HEAT TRANSFER CHARACTERISTICS OF SCO\textsubscript{2} AND DYNAMIC SIMULATION
MODEL OF SCO\textsubscript{2} LOOP

Gang Xiao\textsuperscript{*}
Zhejiang University
Hangzhou, China
Email: xiaogangtianmen@zju.edu.cn

Kaixiang Xing
Zhejiang University
Hangzhou, China

Jinyi Zhang
EDF R&D China
Beijing, China

Yann Le Moullec
EDF R&D China
Beijing, China

Pan Zhou
EDF R&D China
Beijing, China

Tianfeng Yang
Zhejiang University
Hangzhou, China

Mingjiang Ni
Zhejiang University
Hangzhou, China

Kefa Cen
Zhejiang University
Hangzhou, China

ABSTRACT
Supercritical CO\textsubscript{2} (sCO\textsubscript{2}) Brayton cycle is a promising power generation technology with high efficiency, compact turbo-machineries and simple layout. Heat exchangers play an important role in sCO\textsubscript{2} Brayton cycle systems. Therefore it is important to have a comprehensive understanding on heat transfer characteristics of sCO\textsubscript{2} at various operation conditions, in order to better design the heat exchangers. A sCO\textsubscript{2} test loop is built to investigate the heat transfer characteristics of sCO\textsubscript{2}, which consists of a test section for heat transfer (horizontal straight tube with an inner diameter of 8 mm), an electrical heater of 7.2kW, a heat recuperator of type coil tubes, a back-pressure valve, a CO\textsubscript{2} pump, a CO\textsubscript{2} cooler and a CO\textsubscript{2} storage tank. The test loop is designed to operate with pressure ranging from critical pressure to 15MPa, temperature ranging from critical temperature up to 600 °C, mass flow rate ranging from 20 to 70kg.h\textsuperscript{-1}. With this test loop, the heat transfer characteristics of sCO\textsubscript{2} under different conditions are measured and studied. These measurements are then used in the development of a new heat transfer correlations for sCO\textsubscript{2}. Based on the developed correlation, a dynamic model has been built in Dymola and validated by experimental results. The experimental and simulation results show that: 1) The heat transfer coefficient (HTC) changes greatly for CO\textsubscript{2} at or near the critical point; 2) For sCO\textsubscript{2} at operation conditions far from the critical point, the wall-bulk temperature difference increases between the tube wall and sCO\textsubscript{2} flow with an increasing heating power, and the heat transfer coefficient is relatively stable. The experience gained by the experiment and simulation is then used to design a 150kWe sCO\textsubscript{2} Brayton cycle coupled with a solar solid particle receiver.

INTRODUCTION
Coupled the solar power with the sCO\textsubscript{2} Brayton cycle is one of the promising choices for future power generation. In recent years, many countries have completed the early work for sCO\textsubscript{2} Brayton cycle, such as USA, Korea, China and Japan at el.

The USA, who first suggested to use sCO\textsubscript{2} Brayton cycle in nuclear power system, takes the lead in developing sCO\textsubscript{2} power generation systems. Since 2011, DOE (Department of Energy) of USA started the SunShot Project (65 million dollars invested as research funding) and the Gen 3 Project (72 million dollars invested as research funding) for Supercritical CO\textsubscript{2} power cycle applied in solar thermal system. Sandia National Laboratory (USA) has been implementing and improving a supercritical CO\textsubscript{2} power generation test loop since 2005. Then, a 250kWe recompression closed Brayton cycle is implemented using two turbine-alternator-compressor units consisting of gas foil bearings.[1] In addition, Bechtel Marine Propulsion Co. developed a 100 kW supercritical CO\textsubscript{2} power generation cycle test loop using one turbo-generator and one turbo-compressor.[2] Korea Institute of Energy Research developed a 10 kW-class simple Brayton cycle and an 80 kW-class test loop for waste heat recovery.[3] Tokyo Institute of Technology (Japan) developed a 100 kWe sCO\textsubscript{2} experimental test loop.[4] Recently, researchers from Southwest Research Institute and General Electric Company (USA) successfully tested the axial sCO\textsubscript{2} turbine at temperature over 700 °C.

Zhejiang University started the research on sCO\textsubscript{2} power cycle since 2013. In order to have a comprehensive understanding about the heat transfer characteristics of sCO\textsubscript{2} from near ambient temperature to 600 °C, a test loop was built in 2017. Heat transfer experiments have been carried out in
2018. Based on the test loop, a dynamic model has been developed to explore the dynamic characteristic of a sCO₂ Brayton cycle. The steady state validation of the model has been done, while the dynamic validation with experiments will be finished in next 6 months.

The experience gained by the experiment and simulation is then used to design a 150kWe sCO₂ Brayton cycle coupled with a solar solid particle receiver. The selected cycle design to be implemented is a simple regenerative cycle, with a small-scale turbine and compressor. The inlet temperature and pressure of turbine are set to be 550°C and 20MPa respectively, and the efficiency of turbine is expected to be over 75%. The inlet temperature and pressure of compressor are set to be 35°C and 7.4MPa respectively, and the efficiency of compressor is expected to be about 75%. A 2MW regenerator is adopted to increase the compressor outlet sCO₂ flow temperature to 389°C at 20.1MPa, then with a particle/CO₂ heat exchanger as the main heater, the flow is heated up to 550°C before entering the turbine. This innovative system is going to be setup in 2019 and demonstrated in 2020 by Zhejiang University, located in a 1MW solar tower.

TEST LOOP IN ZHEJIANG UNIVERSITY

A sCO₂ test loop is developed in Zhejiang University as shown in Figure 1. The CO₂ flow from the gas source passes the filter, and is cooled to liquid phase in the cooling system. The liquid CO₂ flow rate is measured with a flow meter and then pressurized further by the pump. Here, a buffer tank is used to reduce the fluctuation, and a recuperator is used to preheat the liquid CO₂ stream with the exist stream of the sCO₂ from the straight tube test section. Then sCO₂ flow is heated to the designed inlet temperature in the four-stage heater, and flows through the test section. In this section, temperature and pressure of the sCO₂ flow are measured for the heat transfer characteristics investigation. The heat of the sCO₂ flow is reused in the recuperator. Then a heat exchanger is used to further cool down the sCO₂ flow for the safe operation of the back pressure valve. After the pressure is released, sCO₂ flows back to the source. The test loop can offer sCO₂ working fluid temperature and pressure at 50 ~ 550 °C, and 8~ 15 MPa respectively at the outlet of the test section.

CO₂ gas (99.99%wt) is used as the working fluid in experiments and the mass flow rate is controlled by a pump whose frequency ranges from 0 to 50 Hz. A mass flowmeter and thermocouples are used. The experiments have been carried out at 20~600 °C with the mass flow rate of sCO₂ ranged from 20 to 70 kg·h⁻¹ and pressure from 7 to 15 MPa.

HEAT TRANSFER EXPERIMENTS

Heat transfer experiments were carried out for sCO₂ from ambient temperature to 600°C, from 7MPa to 15 MPa. Fig. 2 shows the test section and the locations of the measuring points. The test section was basically a horizontal straight tube with an inner diameter of 8 mm (wall thickness of 3 mm), and it is heated uniformly by a copper column and six heat resistors. The maximum heating power is 7.2 kW. The test section was made of Hac 276 alloy. The horizontal test section was 2.35 m long, and its heated section was 1.75 m. Fifteen K-type thermocouples were attached with 100 mm interval onto the outer surface of the tube to measure the wall temperatures. The test section was heat insulated by insulation cotton. Two pressure sensors and one pressure difference transducer are used to measure the pressure and pressure drop of the inlet and outlet. A straight tube is heated with a maximum heating power of 7.2 kW. Fifteen thermocouples are used to detect outwall temperature along the tube with an interval of 100 mm. The experiments were implemented at steady state. The accuracies and ranges of measuring devices are shown in Table 1. The thermocouples were calibrated by using thermometer, and the maximum error was less than 1.0%. The differences between the corrected value and the thermometer were less than 0.3 °C when thermometer readings were between 20 and 100 °C, respectively. The total uncertainty of the measuring temperature difference is 1.25%. The error in the electric heating power was 4.25%, considering the heat loss. The physical properties of CO₂ were calculated from the NIST package. Based on the calculated formulation, the maximum uncertainty of the heat transfer coefficient was estimated to be ±7.5%.

The tests were conducted with a change of the mass flux and the heat flux at a given pressure. In order to investigate the effects of the pressure on heat transfer, the experiments were carried out at several pressures: 7.6, 8.3, 9 and 11 MPa. Table 2 shows the range of the test conditions. The range of the Reynolds number based on the inlet condition is from 2×10⁴ to 1.1×10⁵. The working temperature of the test section is controlled from 30 to 600 °C, which is currently working temperature for the solar receiver.

![Figure 1: Layout scheme of the Test Loop.](https://example.com/figure1.png)
Figure 2: Test section and the locations of the measuring points.

Table 1 Range and accuracy of instruments.

<table>
<thead>
<tr>
<th>Instruments</th>
<th>Range</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure difference transducer</td>
<td>100 kPa</td>
<td>0.1%</td>
</tr>
<tr>
<td></td>
<td>500 kPa</td>
<td>0.1%</td>
</tr>
<tr>
<td></td>
<td>1000 kPa</td>
<td>0.1%</td>
</tr>
<tr>
<td>Pressure transducer</td>
<td>10 MPa</td>
<td>0.1%</td>
</tr>
<tr>
<td></td>
<td>25 MPa</td>
<td>0.1%</td>
</tr>
<tr>
<td>Thermocouple</td>
<td>220~1440 K</td>
<td>0.75%</td>
</tr>
<tr>
<td>Mass flow meter</td>
<td>0~300 hg.h⁻¹</td>
<td>0.2%</td>
</tr>
</tbody>
</table>

Table 2 Parameters of the test conditions.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>CO₂</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow direction</td>
<td>Horizontal</td>
</tr>
<tr>
<td>Working Pressure (MPa)</td>
<td>7.6, 8.3, 9, 11, 15</td>
</tr>
<tr>
<td>Mass flow rate (kg/h)</td>
<td>20-60 kg/h</td>
</tr>
<tr>
<td>Inlet temperature, °C</td>
<td>30~450 °C</td>
</tr>
<tr>
<td>Heat flux q, kW/m²</td>
<td>0~60</td>
</tr>
<tr>
<td>Reynolds number</td>
<td>2×10⁴~1.1×10⁵</td>
</tr>
</tbody>
</table>

EXPERIMENTAL RESULTS AND DISCUSSION

In Figure 4, several plots are shown for the mass flux from 34 kg/h to 60 kg/h, while the pressure is kept constant at 7.6 MPa, respectively. As the mass flux increases, the peak HTC increases. The HTC reaches the peak value near the pseudo-critical temperature. When fluid temperature (Tb) is higher than 373 K (100 °C), the HTC is quite stable as shown in Figure 5. As bulk fluid temperature increases, the HTC increases gradually.

Pressure effect is investigated by comparing the test results for 8.3 and 9.1 MPa. In Figure 6, the results show that higher pressure results in lower peak HTC. As the temperature reach the pseudo-critical point, the effect of pressure becomes outstanding, the HTC reach the peak. Especially in the vicinity of the critical pressure, the HTC is several times higher than HTC of conditions far away from pseudo-critical point, due to the steep variation of the physical properties there. In Figure 7, when fluid temperature is higher than 100 °C, the effect of pressure is not significant. As the pressure increases, the wall temperature decreases.

Figure 4: Effect of mass flux on the HTC (pseudo-critical state).

Figure 5: Effect of mass flux on the HTC (high temperature).

Figure 6: Effect of pressure on HTC (near pseudo-critical temperature).
Dittus-Boelter correlation and Jackson correlation. As shown in Figure 10, at high temperature range, Gnielinski correlation gives better evaluation of the HTC for straight tube than Jackson correlation. The average error between the experiment results and Gnielinski correlation is near 12% (over 20% for Jackson correlation). Near the pseudo-critical temperature (T_p), the Jackson correlation is closer to the experiment result than Gnielinski correlation, as shown in Figure 11. However, the error between experiment results and Jackson correlation is larger than 30%. A new correlation is needed for this working conditions to evaluate the designed test loop.

In order to evaluate the effect of the heat flux, an experiment was performed with the variation of heat flux while keeping the mass flow rate and pressure constant. Figure 8 shows the experimental results for a mass flux of 50 kg/h at a pressure of 11.2 MPa. The wall temperature is plotted along the tube length. The heat flux varies between 0 and 44 kW.m⁻². As the heat flux increases, the wall temperature increases. At high temperature, the effect of heat flux is ignorable, as shown in Figure 9.

Jackson[5] proposed a friction factor equation contained the Prandtl number, Reynolds number, density ratio and specific heat ratio. The Gnielinski correlation contains Reynolds number, the Prandtl number and the temperature difference facto. To get a better correlation, more factors should be considered. Therefore, the Nusselt number can be represented as a function of the following non-dimensional parameters:

\[ Nu = \frac{hcD}{\lambda_{co}l} = f(Re_b, Pr_b, \frac{\rho_w}{\rho_b}, \frac{c_p}{c_{p,b}}, q^+, c_i) \]  
\[ q^+ = \frac{q \beta_b}{Q c_{p,b}} \]  
\[ c_i = \frac{T_b}{T_w} \]

Where Nu is the Nusselt number, Re is the Reynold number, Pr is the Prandtl number, he is the thermal conductivity, c_p is the specific heat, q^+ is the accelerate number, q is the heat flux ratio, c_i is the temperature factor, Q is the mass flux and \( \beta_b \) is volume expansion ratio.

Log-based non-linear curve fitting was employed to correlate the Nusselt numbers from our experiments, and a new heat-transfer correlation was constructed as follows:
\[ Nu = 32.39 \text{Re}_b^{1.0084} \Pr_b^{1.9876} \left( \frac{\rho_b}{\rho_{w}} \right)^{-0.2123} \frac{C_{p,w}}{C_{p,b}}^{0.7389} q^{-0.9935} \ell_i^{6.1590} \] (1.4)

The new correlation gives better result for this working conditions, especially the near the critical point as shown in Figure 13 and 14. The average error of the new correlation is near 5% at high temperature range.

Besides, the present correlation is compared with the Experiment results from Korean Atomic Energy Research Institute (KAERI)[11]. As shown in Fig. 14, the correlation can reflect the difference of HTC better than Dittus-Boelter correlation and Jackson correlation. However, this correlation cannot give a good prediction of the experiment results at that conditions. The mass flow rate and heat flux of the correlation is limited.
DYNAMIC MODEL SIMULATION TOOL

For the sCO2 Brayton cycle research, a dynamic model is built for this sCO2 test loop in Dymola Software based on the dynamic module library ThermoSysPro developed in EDF R&D. The thermodynamic properties of CO2 is calculated by dynamic calling to NIST REFPROP in real time during simulation. The interface between Dymola and NIST REFPROP is done by MediaTwoPhaseMixture package developed by Henning Francke. The dynamic model consists of a cooler, a four stages heater, a condenser, a recuperator, a back pressure valve, some control valves, a heating section and a three-plunger pump. In the test loop, back pressure valve is used to replace the turbine, while the friction coefficient change due to the open degree of the valve is regard as the load change during the operation. This dynamic model simulation is used to analyze the dynamic characteristics of different components.

In order to be consistent with the result of the experimental data, some modeling approach is considered. The following assumptions are made for this study:

(1) Pressure losses in the mass flow meter, filter and sensors are replaced by singular pressure loss to reduce the complexity of the loop model for this study;
(2) Expansion process is adiabatic.
(3) Heat loss to the ambient air is estimated by constant heat exchanger coefficient.
(4) The heat conduction between solid media is considered, while the heat conduction within CO2 is ignored.

DYNAMIC MODEL INTRODUCTION

The pump is used to compress the liquid CO2 from gas bottle pressure (around 5 MPa) to high working pressure (7~15 MPa) by plunger movement. It uses mechanical work to push the plunger, thus accelerating the liquid. Here we simplified the calculation of pump with constant isentropic efficiency. The relationship between the mass flow rate Q and the pump frequency, pressure ratio Pe is experimentally measured. A map is generated from a steady state situation for a certain temperature, pressure and frequency. The correlations are concluded as follow (according to the pump frequency from 20 to 50):

\[
Q = \frac{30.934 - 4.2687Pe}{3600} \quad \text{for frequency=20 HZ} \quad (1.5)
\]
\[
Q = \frac{45.503 - 5.8544Pe}{3600} \quad \text{for frequency=30 HZ} \quad (1.6)
\]
\[
Q = \frac{57.802 - 6.21Pe}{3600} \quad \text{for frequency=40 HZ} \quad (1.7)
\]
\[
Q = \frac{71.459 - 7.21Pe}{3600} \quad \text{for frequency=50 Hz} \quad (1.8)
\]

As shown in Figure 15, the correlation concluded for the mass flow rate is closed to the experimental result at different pressure ratios and frequencies. The maximum error of the mass flow correlation is ±1.67%.

We defined the pressure ratio \( Pe \) as:

\[
Pe = \frac{P_{in}}{P_{out}} \quad (1.9)
\]

Where \( P_{in} \) is the pressure at the inlet, \( P_{out} \) is the pressure at the outlet.

The isentropic efficiency \( \eta \) is setting as constant \( \eta = 0.7 \) to reducing the model complexity. Then the power can be calculated by equation:

\[
W = Q(h_{out} - h_{in}) / (1 - \eta_f) \quad (1.10)
\]

Where \( h \) is the enthalpy, \( \eta_f \) is the friction loss percentage. Here, \( \eta_f \) is 0.1.

The outlet enthalpy of the pump can be calculated by following equations and database:

\[
\eta(h_{in} - h_{out}) = h_{out} - h_{in} \quad (1.11)
\]
\[
s_{out} = s = s_{in} \quad (1.12)
\]

Where \( h_{is} \) is the isentropic enthalpy at the outlet calculated by the outlet pressure and outlet entropy. Other properties can be calculated by calling REFPROP based the outlet pressure and enthalpy.

Comparison between simulation result and experimental measure value is conducted, and the results are shown in Figure 16. The maximum outlet temperature error is ±1K or ±0.4%, within the measure error of thermocouples (±2.2K or ±0.75%).

![Figure 15: Mass flow map of the three-plunger pump.](image)

Five heat exchangers are utilized in this test loop, including a recuperator, a cooler, a condenser, a four-stage heater and the heat section (test section). These heat exchangers consists of CO2 pipe and relative wall boundary. The working conditions and pipe geometries are listed in Table 3.
For a normal pipe, pressure loss equation is simplified as:

\[ \Delta P_f = \frac{Q^2}{2\rho} K \]  \hspace{1cm} (1.13)

Where K is the pressure drop coefficient, which can be calculated from the experiment results.

The mass flow model and heat transfer model for the dynamic pipe is shown in Figure 17. The dynamic pipe is divided into n sections. Temperature and pressure of each section is assumed uniform.

The energy equation for the fluid can be written as:

\[ A\left(-\frac{dp[i+1]}{dt} + \rho[i].\frac{dh[i+1]}{dt}\right) dx = Q[i]h_b[i] - h_b[i+1].Q[i+1] + dW_i[i] \]  \hspace{1cm} (1.14)

Where \( h_b \) is the average specific heat capacity of flowing of the fluid, \( \rho \) is the density of the fluid, \( Q \) is the mass flow rate of the fluid, \( dW \) is the heat transfer to the pipe wall. The thermal conduction during two fluid area is ignored.

\[ dW_i = k ds[i] (T_p[i] - T_w[i]) \]  \hspace{1cm} (1.15)

Where, \( k \) is the heat transfer coefficient of the CO\(_2\) pipe, \( ds \) is the inner wall area per node, and \( dW_1 \) is calculated by the new correlation (1.2).

\[ k = 32.39 Re_b[i]^{1.0084} Pr_b[i]^{0.9876} \left( \frac{\rho_b[i]}{\rho_h[i]} \right)^{2.2123} \]  \hspace{1cm} (1.16)

\[ \left( \frac{c_p,w[i]}{c_p,b[i]} \right)^{0.7389} q^*[i]^{0.9935} c_i[i]^{6.159} \lambda_{CO2}/D \]

\[ Re_b[i] = 4 \frac{Q[i] + Q[i+1]}{2 \pi i D} \mu_b[i] \]  \hspace{1cm} (1.17)

\[ c_i[i] = \frac{T_b[i]}{T_w[i]} \]  \hspace{1cm} (1.18)

\[ q^*[i] = \frac{qD}{c_p,b[i]T_b[i] \mu_b[i] Re_b[i]} \]  \hspace{1cm} (1.19)

Where \( b \) means the bulk temperature, \( w \) means wall temperature, \( \lambda_{CO2} \) is the thermal conductivity of CO\(_2\), \( q \) is the heat flux density, \( Re \) is the Reynolds number, \( Pr \) is the Prandtl number, \( c_i \) is the temperature factor and \( q^* \) is the accelerate factor. The properties can be attained from the REFPROP database.

### Table 3 Working condition for heat exchangers

<table>
<thead>
<tr>
<th>Heat exchanger</th>
<th>Inner/Outer (mm)</th>
<th>Pipe length</th>
<th>Wall condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>recuperator</td>
<td>6/10</td>
<td>12</td>
<td>Normal heat loss to ambient ( T_{wall}=15^\circ C )</td>
</tr>
<tr>
<td>heater</td>
<td>6/10</td>
<td>12×4</td>
<td>Heating power ( (500W*4) ) PID control; Normal heat loss to ambient ( T_{wall}=15^\circ C )</td>
</tr>
<tr>
<td>Cooler</td>
<td>4/6</td>
<td>6</td>
<td>Heat loss to cooling water ( T_{wall}=0-10^\circ C )</td>
</tr>
<tr>
<td>Condenser</td>
<td>4/6</td>
<td>12</td>
<td>Heat loss to cooling water ( T_{wall}=5-4^\circ C )</td>
</tr>
<tr>
<td>Heat section</td>
<td>8/14</td>
<td>2</td>
<td>Heating power ( (0-7.2 kW) ); Normal heat loss to ambient ( T_{wall}=15^\circ C )</td>
</tr>
</tbody>
</table>

**Figure 17:** Heat transfer and mass flow model for dynamic pipe.

The mass balance equation for fluid is written as:

\[ Adx\left( \frac{d\rho}{dp}_b[i] \frac{dp[i+1]}{dt} + \frac{d\rho}{dh}_p[i] \frac{dh[i+1]}{dt} \right) \]

\[ = Q[i] - Q[i+1] \]  \hspace{1cm} (1.20)

The wall is divided into n sections. Temperature and pressure of each section is assumed uniform. The heat transfer model for wall is shown in Figure 18.

The energy equations for solid at node i can be written as:

\[ A\rho_p \frac{dT_w[i]}{dt} dx = dW_w[i] + dW_i[i] + A\lambda(2T_w[i] - T_w[i-1] - T_w[i+1]) / dx \]  \hspace{1cm} (1.21)
Where $dW_2$ is the energy transferred from the ambient, $dW_1$ is the energy transferred from the fluid. Thus, in the pipe, the equation of $dW_2$ and $dW_1$ are calculated as:

$$dW_2[i] = 2\lambda pi(T_w[i] - T_f[i])/(\log(1 + e/E))dx$$  \hspace{0.5cm} (1.22)

$$dW_2[i] = hc(2e + D) pi(T_o[i] - T_f[i])dx$$  \hspace{0.5cm} (1.23)

$$dW_1[i] = 2\lambda pi(T_w[i] - T_f[i])/(\log(1 + e/E))dx$$  \hspace{0.5cm} (1.24)

Where $hc$ is the heat transfer coefficient between wall and pipe fluid, $T_w$ is the temperature of the wall boundary at the ambient side, $T_o$ is the temperature of the wall boundary at the inner pipe wall side, $e$ is the wall thickness, and $D$ is the inner pipe diameter.

![Figure 18: heat transfer model for the heat exchanger wall.](image)

The recuperator is then modelled as two parallel pipes with a wall in between. The condenser and cooler are both modeled with a dynamic pipe and a cooling wall. The test section and four-stage heater are modeled with a dynamic pipe and a heating wall. The models’ layout is shown in Figure 19.

![Figure 19: Layout of the heat exchangers: recuperator (left), cooler (right upper) and heater (right down).](image)

The steady state validations of cooler and condenser are performed. The maximum errors of outlet pressure and outlet temperature between simulation and experiments for cooler are separately ±0.1% and ±0.3%, as shown in Figure 20, while the measure errors of the temperature and pressure sensors are separately ±0.75% and ±0.1%. The maximum errors of outlet pressure and outlet temperature between simulation and experiments for condenser are separately ±0.03% and ±0.5%, while the measure errors of the temperature and pressure sensors are separately ±0.75% and ±0.1%.

Validations of the heat section with the experimental results have been conducted for the influence of mass flow rate and working pressure. The maximum error of the outlet temperature and pressure are separately ±1.6% and ±0.2%, as shown in Figure 21.

The back pressure valve is modified from the check valve by setting the pressure loss coefficient. The expanding process in back pressure valve is assumed to be isenthalpic. The pressure loss is calculated by equation (1.11).
Figure 20: Comparison of the experiment and simulation result for cooler (outlet temperature and pressure).

Figure 21: Comparison of the experiment and simulation result for heating section (outlet temperature and pressure).

150KWE sCO₂ BRAYTON CYCLE COUPLED WITH A PARTICLE RECEIVER

The experience gained by the experiment and simulation is then used to design a 150kWe sCO₂ Brayton cycle coupled with a solar solid particle receiver, as shown in Figure 22. The selected cycle design to be implemented is a simple regenerative cycle, with a small-scale turbine and compressor. The inlet temperature and pressure of turbine are set to be 550°C and 20MPa respectively, and the efficiency of turbine is expected to be over 75%. The inlet temperature and pressure of compressor are set to be 35°C and 7.4MPa respectively, and the efficiency of compressor is expected to be about 75%. A 2MW regenerator is adopted to increase the compressor outlet sCO₂ flow temperature to 389°C at 20.1MPa, then with a particle/CO₂ heat exchanger as the main heater, the flow is heated up to 550°C before entering the turbine. This innovative system is going to be setup in 2019 and demonstrated in 2020 by Zhejiang University, located in a 1MW solar tower, as shown in Figure 23.

CONCLUSIONS

Zhejiang University has built a sCO₂ test loop since 2017. The heat transfer experiments have been done. The results show that: (1) existing correlations have significant error near the critical point; (2) at high temperature range, heat transfer characteristics of sCO₂ is quite stable. Base the experimental results, a new correlation is got, and used in the dynamic model developed in Dymola software. The steady state validation has been done. The error between simulation and experiments is smaller than 5%.

Then, in next 6 months, dynamic validation of the model will be completed. And the design experience will be used to design a 150 kWe sCO₂ simple Brayton cycle couple with the 2MW solar tower system built in Zhejiang University. The 150kWe test loop will be constructed in next two years.

Figur 22: Complete model of the test loop.

Figure 23: Layout of the 150kWe sCO₂ Brayton cycle.

Figure 24: Picture of the 1 MW Solar tower in Zhejiang University.

NOMENCLATURE

A  Area,㎡
Cv  Flow coefficient
**REFERENCES**


