Model of a real cycle of a power installation with a real-gas working fluid

A Yu Chirkov¹, K S Egorov¹, K B Ganeev¹ and T R Zuev¹

¹Bauman Moscow State Technical University, 2nd Baumanskaya str. 5/1, Moscow, Russia, 105005

e-mail: chirkov@bmstu.ru, egorovks@bmstu.ru, konstantin.ganeev@gmail.com

Abstract. The modelling of the working cycle of a closed cycle power installation with real gas is considered. The properties of a real gas are investigated using carbon dioxide as an example. The indicators of the real thermodynamic cycle of the installation when working on carbon dioxide are investigated. Particular attention is paid to the simulation of heat transfer in the recuperator.

1. Introduction
In the high power range, steam turbine and gas turbine power plants are usually used. The main obstacle to the use of steam and gas turbines of low power is the low efficiency of blade machines in the low power range. A possible solution in this case is the use of supercritical carbon dioxide cycles [1]. Currently, there is a renewed interest in this area, in particular, in connection with the possibility of using carbon dioxide in power plants with prospective nuclear reactors [2–4], fusion reactors [5–7], neutron sources [8, 9], and other energy applications [10–12]. Note, the promises of power installations with supercritical carbon dioxide for utilization of low-grade heat are of particular interest. The relevance of modelling thermodynamic cycles and workflows is related to the justification of the efficiency of such facilities and the initial data for their design [13].

Dealing with thermophysical properties of carbon dioxide it turned out that the special feature of this gas is the values of critical parameters: critical temperature \( T_c = 31.1 \, ^\circ\text{C} \), critical pressure \( p_c = 7.38 \, \text{MPa} \). This makes it possible to ensure the transfer of heat to the environment at the maximum achievable low temperatures, which increases the efficiency of the cycle. There are two main types of supercritical cycles of carbon dioxide: (i) with condensation (similar to the Rankine supercritical cycle) and (ii) without condensation (the state of the working body is completely above the saturation curve (in the single-phase region), similar to the Brayton cycle, but taking into account significant changes in the thermophysical properties of carbon dioxide in the region of the critical point [14, 15]. In both cases, the compressor operation is significantly lower than the operation of the expansion machine (similar to the Rankine cycle), which distinguishes supercritical cycles on carbon dioxide from Brayton cycles with a working body that is close in properties to an ideal gas.

The essential feature of supercritical cycles on carbon dioxide is the presence of regenerative heat exchange. Thus thermodynamic efficiency of 50% can be obtained [16]. At the same time, the optimal
value of pressure increase in the compressor \( \pi \) is small (\( \pi < 3 \)), which significantly simplifies the design of the compressor (both for cycles with condensation and without condensation). The permissible level of efficiency of the compressor is relatively low, since the amount of operation of the compressor is significantly lower than the operation of the expansion machine.

In the cycles under consideration, the value of the lower pressure is about 7.5 MPa (higher than the critical one), the upper one is about 22 MPa. The low pressure ratio of supercritical cycles leads to a significant simplification of the expansion machine as compared to the steam Rankine cycle. Both compression and expansion can be done in a single-stage machine.

To test the operation of thermal power plants operating on supercritical carbon dioxide cycles, experimental installations were created, in particular, IST (Naval Nuclear Laboratory, USA) with a power of 100 kW [17]. Its tests have successfully demonstrated the possibility of using the supercritical cycle and the effectiveness of the control system in stationary and transient modes [18]. The tests did not reveal the disadvantages associated with the use of a supercritical cycle on carbon dioxide. The relatively low efficiency of the installation was associated with the low efficiency of small-sized turbomachines.

In connection with a dramatic change in the thermodynamic properties of carbon dioxide (heat capacity, compressibility, etc.) in the vicinity of the critical point, the equation of state of a supercritical fluid has a special role in the simulation of cycles. There are two types of state equation models: specialized and general. The most accurate one is the method of calculation based on specialized models. An example of such a model used in the calculation of cycles on carbon dioxide is the Span–Wagner model [19], on the basis of which the data on thermodynamic properties are reproduced.

2. The description of the thermodynamic cycle model and features of heat recuperation

A dramatic change in the heat capacity of carbon dioxide in the vicinity of the critical point causes a decrease in the efficiency of recuperative heat exchange, since the maximum amount of heat transferred in the case of different heat capacities of the hot and cold working fluids at equal mass flow rates is limited to the lowest heat capacity, which makes it impossible to regenerate at the minimum temperature head. A partial solution in this case may be the use of a so-called recompressor and separation of the regenerative heat exchanger into two sections: high-temperature and low-temperature. Part of the flow of the working fluid after exiting the expander and passing through the high-temperature and low-temperature sections is sent to the recompressor (bypassing the cooling heat exchanger), compressed and sent to the input of the high-temperature regenerator. After leaving the low-temperature heat exchanger, another part of the flow is directed to the cooling heat exchanger and to the main compressor inlet, then to the low-temperature regenerator. The connection of flows occurs at the entrance to the high-temperature regenerator (the flow of the recompressor does not pass through the low-temperature heat exchanger).

In the case of a power plant functioning at an upper temperature of 700 °C and a lower temperature of 37 °C, the thermodynamic efficiency of the cycle without recompression (only recuperation) is about 42%. A cycle with a recompressor and flow separation can have efficiency of up to 50% [1]. Although 8% increase in efficiency is quite significant from the point of view of the operation of large-scale power plants, but for low-power installations (100 kW), along with fuel economy, low initial cost and operating costs play an important role. The presence of recompressor significantly increases the cost of installation. It also complicates the installation management system due to the need to control the mutual operation of the main compressor and recompressor and the mutual operation of the high-temperature and low-temperature sections of the heat exchanger. All this leads to an increase in operating costs. It is assumed that in the case of the use of the regenerative cycle without recompression, increased fuel costs will be largely compensated by reducing the initial installation cost and operating costs.

To calculate the thermodynamic properties, several equations of state were considered. The results of comparing the accuracy of the equations of state with near-critical parameters are illustrated in figure 1. Table 1 shows the deviations of the calculated enthalpy values from the reference values corresponding to the NIST database.
Table 1. Deviation of enthalpy values from reference values.

| Equation of state         | $T = 305$ K, $p = 12$ MPa | $T = 310$ K, $p = 12$ MPa | $T = 316$ K, $p = 7.5$ MPa |
|---------------------------|---------------------------|---------------------------|---------------------------|
| Lee–Kessler               | 2.27 %                    | 2.07 %                    | 0.75 %                    |
| Redlich–Kwong–Soave       | 2.18 %                    | 2.63 %                    | 0.53 %                    |
| Lee–Erbar–Edmister        | 3.86 %                    | 3.02 %                    | 0.53 %                    |
| Redlich–Kwong             | 14.69 %                   | 13.44 %                   | 2.93 %                    |

Figure 1. Molar enthalpy (a) and isobaric heat capacity (b) depending on temperature at a pressure of 7.5 MPa according to different equations of state: 1 – Span–Wagner, 2 – Lee–Kessler, 3 – Redlich–Kwong–Soave, 4 – Lee–Erbar–Edmister, 5 – Redlich–Kwong.

Figure 2. Scheme of power plant [8] (left) and supercritical CO$_2$ cycle with recompression (right).

It should be noted that, the Redlich–Kwong–Soave equation suits better for thermodynamic properties with derivatives. However, the heat capacity values obtained using this equation are two times smaller than the reference ones. When calculating such state functions as enthalpy, the Lee–Kessler equation shows good accuracy. Properties calculated using the Span–Wagner equation differ significantly from properties calculated using other equations of state.

The scheme of the power plant [20] and supercritical CO$_2$ cycle with recompression is shown in figure 2. The areas under curves 2–a and 4–d correspond to the recuperated heat. As one can see, the average temperature in the process 2–a is higher than in the process d–4, even with limit recuperation. This temperature difference can is the excessive temperature gap. Due to the high difference between the average temperatures of the hot and cold flows in the recuperative heat exchanger, the limiting regeneration in such a cycle is accompanied by a non-zero (as in Brayton ideal-gas cycle), and the finite temperature gap. Excessive temperature gap can be several tens of degrees, which exceeds the requirements for the heat exchange in the recuperator. On the other hand, it is possible to reduce the sizes of the recuperative heat exchanger. The problem of excessive temperature gap is removed in a
cycle with recompression, but the presence of a recompressor, as it has already been mentioned, is undesirable due to the installation cost.

3. Simulation results
Figure 3 shows the dependences of thermodynamic efficiency on the degree of pressure increase in the compressor $p$. As it can be seen, when $\pi > 3$, the efficiency of the supercritical cycle increases slightly. Moreover, at high degrees of pressure increase, great influence of the efficiency of the compressor is effective, and the efficiency of the actual installation cycle is reduced. Figure 4 shows the ratio of the compression work to the expansion work. Taking into account the efficiency of the compressor (about 80%) and the turbine (90%), this value should not be large (under these conditions, not more than 0.7). As you can see, at a given magnitude, the supercritical cycle is the most profitable. In addition, the supercritical cycle has the greatest specific work cycle.

![Figure 3. Dependence of the efficiency of Brayton (---), transcritical (- - - -) and supercritical (∙∙∙∙∙∙) cycles with recuperation on pressure increase at turbine inlet temperatures: $1 - T_1 = 500$ K, $2 - T_1 = 1000$ K.](image1)

![Figure 4. Dependence of the ratio of the compressor work to the turbine work on the temperature at the turbine inlet for Brayton (---), transcritical (- - - -) and supercritical (∙∙∙∙∙∙) cycles with $\pi = 2.7$.](image2)

The efficiency of cycles with recompression is higher due to the possibility of using a lower temperature gap in the heat exchanger. With an increase in the upper temperature, the efficiency of the cycle with recompression exceeds the efficiency of the Brayton cycle at the same temperature. Calculations have shown that the use of recompressor increases the efficiency of the cycle with the limiting regeneration from 40 to 47%. The degree of regeneration of the supercritical cycle with recompression is less dependent on the average temperature difference compared with a simple cycle.

4. Conclusions
As the analysis showed, some equations of state can be used to describe properties with near-critical parameters qualitatively, but their accuracy is incomparably less than the accuracy of specialized state equations (for example, Span–Wagner equations in the case of carbon dioxide). The supercritical CO$_2$ cycle has the least compression work and the most work. The simulation demonstrated a special feature of heat recuperation, that is the presence of a significant average temperature gap even at the limiting regeneration (about 30 K at $T_1 = 600$ K). Such an excessive temperature gap means a decrease in the efficiency of the cycle as a result of exergy losses, but the use of recompression is not always justified due to the increase in the size of the regenerator and the increase in the cost of installation.

5. References
[1] Brun K, Friedman P and Dennis R 2017 Fundamentals and applications of supercritical carbon dioxide (sCO$_2$) based power cycles (Woodhead Publishing)
[2] Lee J, Lee J I, Yoon H J and Cha J E 2014 Nucl. Eng. Des. 270 76
[3] Ahn Y, Lee J I 2014 Nucl. Eng. Des. 276 128
[4] Bae S J, Lee J, Ahn Y and Lee J I 2015 Ann. Nucl. Energy 75 11
Acknowledgments
This work was partly supported by the Russian Science Foundation (project no. 17-08-01233) and Russian Ministry of Science and High Education (state order no. 13.5521.2017/BC).