Modeling heat transfer in a supercritical carbon dioxide flow with greatly variable thermophysical properties

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Abstract. A mathematical model of carbon dioxide isobaric flow under supercritical pressure in the condition of a constant heat flux on the surface of a cooled pipe is discussed. Criterion dependencies for heat transfer are analyzed. The effect of greatly variable heat capacity is shown. A method has been developed for taking this factor into account in numerical calculations. Areas are found with the so-called degraded heat transfer.

1. Introduction

The ways to use carbon dioxide (CO₂) as a low-boiling fluid have been developed for a long time. From a practical point of view, the attractive features of CO₂ are associated with relatively low values of critical pressure and temperature (74 bar, 304 K). Since it is practically difficult to realize the condensation of CO₂ at typical ambient temperatures of about 300 K, cycles in the field of supercritical parameters are of great interest. Greatly variable properties at near-critical parameters generate a wide variety of heat transfer modes for the conditions of using carbon dioxide as a working body of closed thermodynamic cycles of promising power plants [1–7].

A significant amount of research has been devoted to the prospects of using carbon dioxide in nuclear power plants of sufficiently high power. Recently there have been developments of low-power power plants. Nowadays sufficiently large amount of experimental data has been accumulated on the heat exchange of supercritical carbon dioxide during flow in channels [2]. One of the problems that are important for the removal of heat is the so-called degraded modes of heat transfer during cooling [8], which are associated with the variability of thermal physical properties.

In present study, we consider the flow of carbon dioxide in a tube of small diameter at supercritical pressure under isobaric conditions. Changes in thermodynamic parameters and thermophysical properties along the length of the tube are considered. The aim of the research is to assess how great variations in the heat transfer coefficient arise due to the variations of properties, as well as to establish the factors affecting the most significant way.

2. Heat transfer simulation

We considered flow conditions similar to flow conditions in numerical studies [9] and experiments [10] performed for cooling supercritical methane.
To assess changes in thermodynamic parameters, carbon dioxide flow in a tube 2 mm in diameter was considered. The speed at the entrance to the heat exchange site was 40 m/s. The flow regime under the conditions considered is turbulent.

Inlet pressure was assumed to be $p_{in} = 80$, 95, and 110 bar, consequently. The estimation of the change in the thermodynamic parameters of the flow along the length of the tube was made on the basis of the thermodynamic equations for the one-dimensional flow in a channel of constant cross section with constant heat removal [11]. These equations were solved numerically.

The thermodynamic properties were calculated using the Peng–Robinson equation of state, and the transport properties were taken from the NIST database. The pressure drop over the length of the tube is considered to be small, so the flow can be considered isobaric. In our case, the drop over the tube length of 0.6 m was about 0.8 bar, which is much less than the fluid pressure. The heat flux density was set at 1 MW/m$^2$.

To calculate the wall temperature, the heat transfer coefficient was set, which was estimated from two empirical dependencies from [12]. These dependencies are used today in most of the works published on this topic.

In the first case, the heat transfer coefficient corresponded to the dependence for the Nusselt number [13]

$$
Nu_w = Nu_{w0} \left( \frac{\rho_w}{\rho_f} \right)^n \left( \frac{\bar{c}_p}{c_w} \right)^m,
$$

where $Nu_{w0} = (\xi / 8) \text{Re Pr} \left( 12.7 \sqrt{\xi / 8 \text{Pr}^{2/3} - 1} + 1.07 \right)$ is the Nusselt number with no taking into account the variability of properties, $\xi = (1.82 \text{ lg Re} - 1.64)^{-2}$, the indices “w” and “f” indicate the defining temperatures of the wall and fluid (liquid); $\rho_w$ and $\rho_f$ are the densities at the wall and fluid temperature; $\bar{c}_p$ is the average isobaric heat capacity, $c_{pw}$ is isobaric heat capacity at wall temperature, $n$ is the function of the pressure, $m$ is the function of the ratio of $\bar{c}_p / c_{pw}$.

For comparison, another relationship [8, 12] was also used,

$$
Nu_f = 0.028 \text{ Re}^{0.8} \left( \text{Pr} \right)^{0.4+k},
$$

where value of the index $k$ is calculated according to [12].

As the comparison made in [12] showed, both dependencies (equations (1) and (2)) are in good agreement with experimental data. Figure 1 presents the results of a one-dimensional calculation of velocity and density changes along the tube. Since the longitudinal coordinate used for the calculation of the ratio is not included explicitly, the abscissa in figure 1 and the following one, the pressure drop $\Delta p$ in the corresponding section are shown.

The change in the temperature of the fluid, as well as the temperature of the wall in the calculation by equations (1) and (2), is shown in figure 2. The corresponding values of the local heat transfer coefficient are shown in figure 3. As it can be seen from figures 2 and 3, estimates by equations (1) and (2) lead to qualitatively similar results, quantitative differences are relatively small.

The heat transfer coefficient increases dramatically with decreasing temperatures approaching the critical temperature. Next, we consider how this changes the thermophysical properties and how their variations affects the variations of the Nusselt number.

Figure 4 shows the change in the heat capacity of the fluid at different pressures, as well as the heat capacity of the wall temperature and the average integral heat capacity at a pressure close to critical (80 bar). The change in the local Nusselt number is shown in figure 5.

Equation (1) allows us to determine the effect of corrections related to density variability and heat capacity. The effect of these values on the Nusselt number is shown in figure 6. As it can be seen, increasing density leads to an increase in the Nusselt number by about 30% compared with the case without taking into account the variability of properties. The growth of heat capacity, in contrast, gives a significant decrease. Changes in wall and fluid temperatures also affect the change in Reynolds and Prandtl numbers. Since in the formula (1) the wall temperature is crucial for the Reynolds number, it changes significantly. In formula (2) for the Reynolds number, the fluid temperature is crucial. Therefore, it has a constant along the channel length values $Re_f = 2.34 \times 10^5, 2.68 \times 10^5$ and $3.05 \times 10^5$ for pressures of 80, 95 and 110 bar, respectively.
Figure 1. Change in velocity (a) and density (b) of fluid along the length of the tube.

Figure 2. Fluid temperature (solid curves) and wall temperature calculated using equation (1) (dashed curves) and equation (2) (dash-dotted curves).

Figure 3. Local heat transfer coefficient according equation (1) (solid curves) and equation (2) (dashed curves).

Figure 4. Variations of heat fluid capacity (a), the average heat capacity and heat capacity at the wall temperature, calculated using equation (1) (b) and (2) (c).
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3. Conclusion
The estimates obtained show that in promising power plants with supercritical working bodies even relatively small variations of parameters in the cooling channels can lead to noticeable variations in the efficiency of heat transfer, which is especially pronounced at pressures close to critical. This is especially important in the case of carbon dioxide, since the practically achievable values of the lower temperature in the cycle with supercritical CO$_2$ are close to the critical temperature.

4. References
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