The use of porous metals in heat exchangers

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Abstract. Heat exchangers are widely used in heat supply systems. To increase the efficiency of heat supply systems, heat exchangers with porous metals are proposed to design. There was a test facility set up to study new types of heat exchangers. The countercurrent flow of heat carriers was activated in those heat exchangers. Freon moved through the heat exchanger pores, and water moved through the inner tubes. It should be noted that the porous materials in the heat exchangers differed in the coefficient of porosity. To be compared, one of the heat exchangers did not contain any porous material. The first test cycle proved the feasibility of using porous metals in heat exchange equipment. Afterwards, a simplified mathematical model of the heat exchanger was compiled. Such an analytical form makes a solution convenient for engineering calculations. Numerical calculations based on this model were compared with the experimental data. Heat transfer intensity of materials with different porosity was compared. This method can be applied in case when the heat exchange surface area is unknown. Moreover, it takes into account the heat capacity and heat of the phase transition of freon, if any.

Keywords: a heat supply system, a heat exchanger, porous metals, a test facility, a mathematical model.

1 Introduction

Centralized heating systems are currently widespread. Heat exchangers of various designs are used to transfer heat from heating system water to a heat carrier. Heat exchangers make one of the main elements in heat supply systems. The efficiency of heat transfer in heat exchangers is characterized by a heat transfer coefficient. Therefore, increase of the heat transfer coefficient stimulates the increase of heat supply systems efficiency.

Basically, heat supply systems use sectional counterflow water heaters or plate heat exchangers. Their heat transfer process is well-known, and it is difficult to obtain significant increase in the heat transfer coefficient. Recently, the problem of designing new heat exchangers (for example with porous metals) has become urgent.

Porous heat-conducting materials made of powdered aluminum, copper and other materials make it possible to create new efficient and compact heat exchangers. Porous metals can significantly increase heat transfer. At the same time, installations with porous metals can be made for various purposes and used not only in heat exchangers of heat supply systems, but become the main elements of refrigeration units, heat pumps, and steam turbine condensers.

There are various design solutions for heat exchangers with porous metals. There are plate heat exchangers where the inter-plate channels for the movement of coolants are filled with porous metal inserts with a high specific area of the inner surface of the frame and small values of the equivalent diameters of the internal channels, providing high heat transfer rates of the operating media. The proposed design increases heat transfer significantly [1-3].

There are porous compact heat exchangers. They intensify heat transfer processes by implementing a porous filler made of high thermal conductivity materials into the ducts. A highly porous material with a varying relative thickness is used for filling in one of the most efficient designs of such heat exchangers. The shell is cylindrical, round or oval in cross section. [4-8]
There are original designs of evaporative elements for cooling fluids in various fields of technology. The evaporation element is a three-layer wall adjacent to the heat transfer surface with different porosity of the extreme and middle layers. The extreme layers are made of porous metal with an average porosity of 0.5, and the middle layer is with low porosity of 0.2-0.25. [9-18]

There are special heat exchangers of the “liquid-gas” type with a phase transition of a heat carrier and without a phase transition. There are also various designs of such devices. For instance, they use a triple layer of porous metal adjacent to a heat transfer surface with a different degree of porosity. They also apply the method of transmitting a hot coolant with a phase transition through a porous metal in order to increase the heat transfer surface and its efficiency. Heat exchangers in the form of porous fins with different angles of inclination are also used. However, it should be noted that sometimes the units are expensive or it is hard to clear them out. [12-18]

Thus, nowadays heat exchangers with porous metals are being designed.

The Department of Industrial Heat Power Engineering in Tyumen Industrial University became a platform for setting up a test facility to study the effectiveness of using such materials.

The test facility is a stand where three heat exchangers with porous inserts and one heat exchanger without them are fixed. Heat exchangers are countercurrent, i.e. water flows through the central copper tubes, and freon moves through the pores of the inserts in the opposite direction. Water is pumped. The water temperature can be changed since the pump is followed by a boiler. Freon moves through a refrigeration circuit consisting of a compressor, an evaporator and a condenser. Thus, two circuits were set up in the test facility: one for water circulating, and the other for freon circulating. The schematic diagram of the test facility is shown in Figure 1.

The object of the research is the heat transfer process in heat exchangers of the "liquid-gas" type with a phase transition of the coolant and without a phase transition. The purpose of the work is an experimental and theoretical study of the heat transfer intensity in heat exchangers of the "water-gas" type with a phase transition of a coolant containing porous inserts of different porosity.

The theoretical basis for the study of a stand heat exchanger (Figure 1) of the “water-gas” type with a phase transition of the coolant is the research of I. Popov et al. [19-20].

2 Materials and methods
The scheme of the test facility for estimating the heat transfer intensity in heat exchangers with porous structures is shown in Figure 1.
Heat exchangers are comprised of 19 copper tubes for water flow. Four cylindrical inserts of aluminum with different porosity are on them. The porosity of the inserts in the first heat exchanger is 0.4901, in the second heat exchanger the porosity of the inserts is 0.6169, in the third one the porosity of the inserts is 0.4739, in the fourth there is no porous insert. The inside of the heat exchanger with porous inserts is shown in Figure 2.

Porous inserts are cylinders of porous aluminum with a height of 50 mm and a diameter of 49 mm. There are 19 holes with a diameter of 6 mm in each of the inserts. Water moves through the tubes installed into the holes. Freon moves through the pores of the insert.

The facility is equipped with valves and cocks creating different experimental conditions for each heat exchanger operating separately, or all heat exchangers at once, or a group of heat exchangers. A ball valve blocks water supply to the heat exchangers out of the experiment. The freon circuit is also equipped with ball valves blocking the flow of freon into heat exchangers that are not tested.

A series of experiments was carried out at the set up. The first part of the experiment was as follows: water consumption was regulated by a ball valve. The temperature changes were measured at the inlet to and from the heat exchanger. The flow rate was also recorded. In order to exclude random errors at each flow rate, temperature readings were taken 10 times. The average value of a series of experiments was found. The obtained values determined the freon mass flow rate and the amount of heat transferred from water to freon [21].

Every part of the experiment was carried out on each heat exchanger with four values of water flow. As a result, the obtained data made it possible to estimate the heat transfer intensity.

![Figure 2. Porous aluminum inserts.](image)

![Figure 3. Results of measurements of heat transfer intensity in heat exchangers with and without porous inserts.](image)
The analyzed data indicated that at the same mass flow rates, the heat transfer intensity is higher for heat exchangers with porous metal inserts. Moreover, heat exchanger 4 without porous inserts has lower heat transfer intensity than other heat exchangers. The results are shown in Figure 3. It can be concluded that heat exchangers with porous metals are more efficient than a conventional counterflow heat exchanger.

The next part of the experiment was carried out with an unvarying flow rate of the cooling heat carrier - water. Water of room temperature of about 20 - 22 °C was cooled to a temperature of about 3 - 50 °C in the circuit with freon running. The water temperature and other measurements were recorded every 2 minutes. Each heat exchanger testing was repeated several times, measurements were made up to 10 times. As a result, the following data are presented in Figure 4.

![Figure 4](image-url)

**Figure 4.** The graph of temperature changes in water cooled by freon.

Summarizing the results of measurements throughout the experiment, we can register more efficient heat transfer in heat exchangers with porous aluminum. Besides, the most effective is the heat exchanger with the largest porosity among the tested heat exchangers.

Then we considered the heat transfer process inside a porous cylinder (porous insert) made of aluminum with a constant coefficient of thermal conductivity $\lambda_c$. The cylinder was well-insulated; therefore, we assumed that there was no heat exchange with the external environment. There are 19 copper tubes inside the porous cylinder. Chilled water flows through them. An inlet temperature of water is $t_{in}$ (Figure 5).
Freon is continuously supplied from the bottom up with a constant specific mass flow rate \( G_c \).

Water is supplied to the tubes from top to bottom (counterflow circuit) with a constant specific mass flow rate \( G_b \). The input water temperature was specified \( t_{in} \). While passing through the heat exchanger, the water is cooled and at the outlet has a temperature \( t_{out} \) (found by calculation). The cooled water passes through the water circuit and gets into the heat exchanger with a temperature of \( t_{in1} = t_{out} \). At the outlet of the heat exchanger the temperature will be \( t_{out1} \), so that the value of \( t_{in} \) will be different in the next cycle. It is also found by calculation. Further on, the cycle is repeated manifold. So the temperature of the water circulating through the heat exchanger decreases continuously when the refrigeration circuit is operating. In order to justify such a change in temperature, it is necessary to find a dependence making it possible to determine the temperature of water at the outlet of the heat exchanger, and to compare the calculated data with the experimental ones.

The inner surface area of the tubes \( S \) is known. The volume of the porous insert \( V \) is known. The temperature of the freon at the inlet and outlet is \( t_0 = t_{c1} \) and \( t(h) = t_{c2} \). The specific mass flow rates of freon and water \( (G_c \text{ and } G_b) \) are also known. They are measured with the flow rate meter during the experiment.

Also, the specific heat of freon and water is set - \( c_{pc} \) and \( c_{pw} \).

The porosity of the inserts \( p \) is considered as the ratio of the pore volume to the entire volume of the material. The porosity is considered uniform. Therefore, the cross section for the gas passage on a surface unit normal to the direction of the gas flow \( f_w = p \), and the cross section of the solid skeleton involved in heat transfer is \( f_s = 1 - f_w = 1 - p \).

Taking into account the heat transfer of the porous insert, the heat flow rate in the section \( x \) and \( x + dx \) is written as follows:

\[
q_s = -\lambda_c \frac{dt}{dx} (1 - p)
\] (1)

and

\[
q_{s+dx} = -\lambda_c \frac{d}{dx} \left(t + \frac{dt}{dx} dx\right)(1 - p)
\] (2)

Under static conditions, there will be a change in the heat flow in the section \( dx \) due to heat transfer between the solid and the fluid flowing through the pores, i.e.

\[
dq = q_s - q_{s+dx} = G_c \cdot c_{pc} dt
\] (3)

or

\[
-\lambda_c \frac{dt}{dx} (1 - p) + \lambda_c \frac{dt}{dx} (1 - p) + \lambda_c \frac{d^2 t}{dx^2} (1 - p) dx = G_c \cdot c_{pc} dt
\] (4)

Therefore, the differential equation for \( 0 \leq x \leq h \) is as follows:
So, here is the equation for the one-dimensional problem of cooling a porous body [22]:

\[
\frac{d^2t}{dx^2} - \frac{G_c \cdot c_{pc}}{\lambda_c (1 - p)} \frac{dt}{dx} = 0
\]

(5)

where \( \frac{G_c \cdot c_{pc}}{\lambda_c (1 - p)} = \xi_c \).

The derived equation led to an assumption that all heat inside the porous body is transferred due to thermal conductivity through the solid phase and that the temperatures of the solid and the heat carrier hardly differ from each other at any point of the porous structure. If this assumption is accepted, then Eq. (6) can describe the heat transfer process in a homogeneous porous insert, that is, without tubes with water (Figure 3). In this case, the temperature field can be considered one-dimensional, considering thermal insulation.

The temperature field becomes two-dimensional if there are tubes with water, and Eq. (1) does not work.

This equation is proposed to supplement with a function of distributed heat sources (sinks). The function describes the process of heat transfer from a porous material through the walls of copper pipes to water, with some error though.

\[
q = \frac{\alpha \cdot (t_u - t) \cdot S}{\lambda_a \cdot V_a}
\]

(7)

Consequently, the following equation succeeds:

\[
\frac{d^2t}{dx^2} - \xi_c \frac{dt}{dx} + \frac{\alpha \cdot (t_u - t) \cdot S}{\lambda_a \cdot V_a} = 0
\]

(8)

where \( V_a \) is the volume of porous inserts, \( \alpha \) is the coefficient of heat transfer from the copper wall to water, \( t_u \) is the water temperature at a given value of \( x \).

Single-valuedness conditions:

\[
0 \leq x \leq h, t(0) = t_1, t(h) = t_2
\]

(9)

When deriving Eq.(8), the thermal resistance of the copper walls was neglected.

The solution is greatly simplified if we assume that \( t_u - t = \) const. This assumption is quite valid for a small height \( h \) of an insert and with a counterflow circuit. The value of \( \alpha \) is found from the equations of convective heat transfer in pipes.

\[
\frac{\alpha \cdot (t_u - t) \cdot S}{\lambda_a \cdot V_a} = A
\]

We denote

\[
\frac{\alpha \cdot (t_u - t) \cdot S}{\lambda_a \cdot V_a} = A
\]

(10)

Boundary value problem Eq.(8) and (9) was solved by standard methods. The function of temperature changes of the porous metal in the heat exchanger along the \( x \) axis was obtained:

\[
t = t_{e1} + \frac{A}{\xi_c} x + \left(e^{\xi_c x} - 1\right) \frac{t_{c2} - t_{c1} - \frac{A}{\xi_c} h}{e^{\xi_c h} - 1}
\]

(11)

We differentiate the obtained temperature function on variable \( x \):
With function Eq. (12) known, we derive the formula for the heat flow rate:

$$q = -\lambda_e \cdot (1 - p) \cdot \left( \frac{A}{\xi_e} x + \frac{t_c - t_{c1}}{\xi_e} \cdot e^{x \xi - A \cdot h} \cdot e^{x \xi - 1} \right)$$

(13)

The quantity $\Delta Q = \Delta q \cdot S = (q (h) - q (0)) \cdot S$, where $Q$ is the amount of heat. It is equal to the heat transferred from water to freon without heat loss. The volume of water in the tubes is $V_t$. This is the capacity of the tubes. The heat capacity and density of water are $c_{pv}$ and $\rho_v$ respectively. The initial water temperature is known and is equal to $t_{in}$. The outlet water temperature will be found at a known $\Delta Q$ by the formula:

$$t_{in} = t_{out} - \Delta t_t$$

(14)

where

$$\Delta t_t = \frac{\Delta Q}{c_{pv} \cdot V_t \cdot \rho_a}$$

(15)

3 Results and Discussion

The developed model was verified through a series of experiments evaluating its accuracy and correctness. The experiments were carried out with the following values of the main parameters: the diameter of the porous sample - $d = 0.049$ m; the total length of the porous insert is $h = 0.2$ m; number of tubes - $n = 19$; the inner diameter of the tubes is $d = 0.004$ m; heat capacity of water - $c_{f} = 4187$ J/kg/K; thermal conductivity coefficient of aluminum - $\lambda_s = 209.3$ W/m/K. The same values of the main parameters were specified for theoretical calculations.

The inlet water temperature value was determined from the experimental data; inlet and outlet freon temperature proceeded from the results of experimental measurements.

Comparative graphs below present theoretical calculations and the results of experimental measurements for each of the heat exchangers.
Figure 6. Theoretical and empirical lines of temperature changes in water cooled by freon: 6a heat exchanger 1, p = 0.4901; 6b heat exchanger 2, p = 0.6169; 6c heat exchanger 3, p = 0.4739; 6d heat exchanger 4, without porous metal.

The graphs show that the empirical and theoretical curves almost coincide. This verifies the model having small errors within acceptable values.

4 Conclusions

Based on the research and calculations, the following conclusions can be drawn:

1. The heat transfer rate is higher in heat exchangers with porous metal inserts compared to heat exchangers without porous metal inserts.
2. The heat transfer intensity is higher in the heat exchanger with the highest porosity p = 0.6169 compared to other tested heat exchangers with porous inserts.
3. A mathematical model has been developed to describe the heat transfer process in porous inserts of a counterflow heat exchanger.
4. A calculation equation has been derived to determine cooling degree of the hot heat carrier at the outlet of a heat exchanger, as well as determine temperature in the porous structure in any section of a heat exchanger.
5. Laboratory research proves the possibility of designing new heat exchangers with porous metals inside, they can be used in the power industry, namely in heat supply systems.

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