Selection Criteria for Heat Exchangers for Hot Water Systems

E Kh Kitaytseva

Moscow State University of Civil Engineering, Yaroslavskoe shosse, 26, Moscow, 129337, Russia

E-mail: keh2@bk.ru

Abstract. The article provides an analysis of existing solutions for choosing shell-and-tube heat exchangers for the needs of hot water supply. When constructing a mathematical model of heat transfer, an approximation of the criterial dependencies describing this process was used. The error estimation of approximating dependencies is presented. The mathematical model of heat balance was used in a numerical experiment. The significance of the influence on the heat transfer efficiency of the parameters of the heat exchanger was established by the results of a numerical experiment. The results of a numerical experiment made it possible to develop criteria for choosing heat exchangers. Keywords: hot water supply, heat exchanger, mathematical modeling, selection criteria

1. Introduction

Improving the energy efficiency of heat supply is the most important task of today. This problem can be solved by different methods: switching from central heat points to individual heat points [1-2], creating a pulsed flow regime in the heating circuit [3-4], lowering hot water temperature [5], changing methods for regulating the temperature of hot water [6], etc.

2. Literature review

The efficiency of the entire heat network is affected by the efficiency of the heat exchangers used in independent heating system connection circuits and closed hot water supply connection schemes. The debate about which type is better continues to the present. There are both supporters of plate heat exchangers, for example [7, 8], and supporters of shell-and-tube heat exchangers [9, 10]. An objective, scientifically based approach [11] leaves this debate unfinished: “The task of choosing a design for a specific heat supply system is difficult, and in its entirety it is impossible to algorithmize and solve it using only a qualitative description (to a certain extent subjective) due to the large number of alternative situations”. Work continues on improving shell-and-tube heat exchangers [9, 10].

The mathematical model of the heat exchange process has a great influence on the correct choice of a heat exchanger. The technologies of computer hydro-gasdynamics [12, 13] are not used in the practice of designing heat points. In most cases, the heat transfer coefficient K is found using dimensionless parameters [14, 15, 16, 17].
3. The relevance of research

Analysis of the heat load for the needs of hot water and installed heat exchangers (Ramenskoye, Moscow region) showed the following: more than half of the subscribers have a load of not more than 0.1 Gcal / h; only 10% of subscriber inputs are equipped with plate heat exchangers.

It was found that, in order to relieve exactly the same load, heat exchangers of different models were installed, the heating surface and tube areas of which differ by an order of magnitude. On the other hand, a heat exchanger of the same model with the same number of sections was used to relieve loads, which different several times.

As can be seen from a comparison of the data shown in Figure 1, the surface area of the heat exchanger heating is overestimated compared to the recommended value [17]. With a load of less than 0.1 Gcal/h there are more such cases, and they are more pronounced. At the same time, there are cases of underestimation of the surface area of the heating, and although they are much smaller, they are also more common at loads less than 0.1 Gcal/h. It seems that designers and organizations operating heating networks do not pay enough attention to subscriber inputs with a small load, although their number is quite large.

![Figure 1](image-url)

**Figure 1.** Frequency distribution, the ratio of the actual and estimated heating area of the heat exchanger.

The above examples indicate that the selection of heat exchangers is not given due attention.

For shell-and-tube heat exchangers, the heated water velocity is assumed to be 1 m / s [17]. Then, the cross-sectional area of the tubes (for hot water supply systems) or the annular space (for heating systems) is calculated and a heat exchanger with the closest parameter values is selected. It is not clear from the methodology whether the speed value is the ultimate or optimal one and at what parameters it is necessary to switch to a 2-threaded scheme. The lack of precise recommendations leads to the fact that in practice any heat exchanger can be installed, both with overestimated and understated dimensions.

Table 1 shows the parameters of shell-and-tube heat exchangers and calculated limit values of heat loads, flow rates and speeds. With a load of less than 0.122 Gcal / h, it is necessary to install a heat exchanger z-01 (or z-02), and with a load of more than 4.602 Gcal / h, switch to a 2-flow scheme. The values given in the table were obtained for the values of the heated water temperature \( t_h = 60 ^\circ C \), \( t_c = 5 ^\circ C \) and for the values of the heating water temperature \( t_1 = 70 ^\circ C \), \( t_2 = 30 ^\circ C \) (DHW) and for the...
values of the heated water temperature \( t_3 = 95 \, ^\circ C \), \( t_4 = 70 \, ^\circ C \) and for the values of the heating water temperature \( t_1 = 150 \, ^\circ C \), \( t_2 = 76 \, ^\circ C \) (heating system).

In order to determine the effect the velocity of the heated water, of which defines a model of the heat exchanger, and the heating surface area (number of sections selected model) for flow of the heating water, a numerical experiment was carried out. The following conditions have been accepted: \( Q = 0.223 \, \text{Gcal/h} \), the connection diagram is parallel.

Table 1. The main parameters of shell-and-tube heat exchangers.

| Model | Inner diameter of the casing mm | Number of tubes | \( F_h \) m² | Limit values at \( w_2 = 1 \, \text{m/s} \) |
|-------|---------------------------------|-----------------|--------------|-----------------|
|       |                                 |                 | \( Q \) Gcal/h | \( \tau \) t/h | \( G \) Gcal/h/h | \( \tau \) t/h | \( \varphi \) m/s | \( Q \) Gcal/h/h | \( \tau \) t/h | \( G \) Gcal/h/h | \( \tau \) t/h | \( \varphi \) m/s |
| z-01  | 50                              | 4               | 0.38         | 0.122          | 2.2              | 3.05          | 0.73          | 0.104          | 4.17           | 1.41          | 0.64          |
| z-02  | 50                              | 4               | 0.75         | 0.122          | 2.2              | 3.05          | 0.73          | 0.104          | 4.17           | 1.41          | 0.64          |
| z-03  | 69                              | 7               | 0.66         | 0.213          | 3.9              | 5.33          | 0.64          | 0.210          | 8.39           | 2.84          | 0.73          |
| z-04  | 69                              | 7               | 1.32         | 0.213          | 3.9              | 5.33          | 0.64          | 0.210          | 8.39           | 2.84          | 0.73          |
| z-05  | 82                              | 10              | 0.94         | 0.305          | 5.5              | 7.62          | 0.65          | 0.294          | 11.77          | 3.98          | 0.72          |
| z-06  | 82                              | 10              | 1.88         | 0.305          | 5.5              | 7.62          | 0.65          | 0.294          | 11.77          | 3.98          | 0.72          |
| z-07  | 106                             | 19              | 1.79         | 0.579          | 10.5             | 14.48         | 0.80          | 0.450          | 18.02          | 6.09          | 0.58          |
| z-08  | 106                             | 19              | 3.58         | 0.579          | 10.5             | 14.48         | 0.80          | 0.450          | 18.02          | 6.09          | 0.58          |
| z-09  | 158                             | 37              | 3.49         | 1.128          | 20.5             | 28.19         | 0.64          | 1.095          | 43.80          | 14.80         | 0.72          |
| z-11  | 158                             | 37              | 6.97         | 1.128          | 20.5             | 28.19         | 0.64          | 1.095          | 43.80          | 14.80         | 0.72          |
| z-12  | 207                             | 61              | 5.75         | 1.859          | 33.8             | 46.48         | 0.60          | 1.925          | 77.00          | 26.01         | 0.77          |
| z-13  | 207                             | 61              | 11.50        | 1.859          | 33.8             | 46.48         | 0.60          | 1.925          | 77.00          | 26.01         | 0.77          |
| z-14  | 259                             | 109             | 10.27        | 3.322          | 60.4             | 83.06         | 0.75          | 2.769          | 110.77         | 37.42         | 0.62          |
| z-15  | 259                             | 109             | 20.55        | 3.322          | 60.4             | 83.06         | 0.75          | 2.769          | 110.77         | 37.42         | 0.62          |
| z-16  | 309                             | 151             | 14.23        | 4.602          | 83.7             | 115.06        | 0.72          | 4.017          | 160.67         | 54.28         | 0.65          |

4. Methods

In the numerical experiment, a mathematical model of heat transfer was used, which is described below.

The amount of heat, \( Q_1 \), Gcal/h, given off by heating water, in heat exchanger

\[
Q_1 = c G_1 (t_1 - t_2),
\]

where \( c \), Gcal/(kg K), is the heat capacity of water, taken equal to 0.001; \( G_1 \), t/h, - flow rate of heating water passing through the heat exchanger; \( t_1 \), \( ^\circ C \) – the temperature of the heating water at the inlet to the heat exchanger, taken according to the temperature schedule equal to the temperature of the network water in the supply pipe at the current outdoor temperature; \( t_2 \), \( ^\circ C \), - the temperature of the heating water at the outlet of the heat exchanger.

The amount of heat \( Q_2 \), Gcal/h, obtained by heated water, in the heat exchanger

\[
Q_2 = c G_2 (t_3 - t_4),
\]

where \( G_2 \), t/h, is the flow rate of heated water passing through the heat exchanger, is equal to the estimated flow rate;

\[
G_2 = G_{DHW},
\]

\( t_3 \), \( ^\circ C \), is the temperature of the heated water at the outlet of the heat exchanger; when the temperature controller is installed, the temperature is taken equal to the required temperature in the hot water supply system \( t_h \); \( t_4 \), \( ^\circ C \), - the temperature of the heated water entering the heat exchanger is the tem-
perature of the cold water mixture and returned from the DHW system through the circulation pipe, which is determined from the heat balance equation for the mixture point:

$$c_t G_2 = c_t (G_2 - G_{circ}) + c_t G_{circ}. \quad (5)$$

If we assume that the return water flow $G_{circ}$, t/h, is a certain fraction $K_{circ}$ of the calculated flow, and the cooling of the return water $\Delta t_c$ is some constant, then the temperature of the heated water $t_4$ is determined by the formula:

$$t_4 = t_{cw} (1 - K_{circ}) + (t_3 - \Delta t_c) K_{circ}. \quad (6)$$

Heat exchanger characteristic equation:

$$Q = K F_h \left[ (t_1 - t_3) - (t_2 - t_4) \right] \left( \ln \frac{t_1 - t_3}{t_2 - t_4} \right)^{-1}, \quad (7)$$

where $K$, Gcal/(h m $\degree$C), heat transfer coefficient of the heat exchanger; $F_h$, m$^2$, is the area of the heating surface of the heat exchanger.

The heat transfer coefficient $K$ is determined by the formula:

$$K = \phi \beta \left( \frac{1}{\alpha_1} + \frac{1}{\alpha_2} + \frac{\sigma_w}{\lambda_w} \right)^{-1}, \quad (8)$$

where $\phi$ is the heat transfer efficiency coefficient adopted in accordance with [17]; $\beta$ — a coefficient taking into account the pollution of the surface of the pipes depending on the chemical properties of water is taken according to [17]; $\alpha_1$, Gcal/(h m$^2$ C), - heat transfer coefficient from the heating water to the wall of the dividing surface; $\alpha_2$, Gcal/(h m$^2$ C), - heat transfer coefficient from the wall of the dividing surface to the heated water; $\sigma_w$, m, - wall thickness of the dividing surface; $\lambda_w$, Gcal/(h m °C), is the thermal conductivity of the wall material.

The heat transfer coefficient is found from the dimensionless dependencies [15] according to the formula:

$$\alpha = A \lambda a^{-n} d^{m-1} v^{n-m}, \quad (9)$$

where $A$, m, n — experimental coefficients, depending on the totality of parameters [16]; d, m, is the linear size taken for shell and tube heat exchangers equal to the equivalent diameter of the tubes or the equivalent diameter of the annulus; $\lambda$, W/(m °C) is thermal conductivity of water; $v$, m$^3$/s, is the flow rate of water; $\nu$, m$^2$/s, is the kinematic viscosity of water; $a$, m$^2$/s, thermal diffusivity of water.

Water parameters $\lambda$, $\nu$, $a$ are determined by the average temperature $t_m$, and the expression $A \lambda a^{-n} v^{n-m}$ is approximated by a parabola (Figure 2). Parabola coefficients differ among different authors [17–20]. Using the coefficients from [18, 20] leads to a significant error. At the same time, when using the coefficients from [17, 19], the error is less than 2%.

As a result, heat transfer coefficient is found by the formula:

$$\alpha = D_d \left[ A_d + B_d m - C d^2 m \right]^m d^{m-1}. \quad (10)$$

From the heat balance equation for the heat exchanger we have:

$$c G_1 (t_1 - t_2) = c G_2 (t_3 - t_4) \quad (11)$$

and

$$c G_2 (t_3 - t_4) = K F_h \left[ (t_1 - t_3) - (t_2 - t_4) \right] \left( \ln \frac{t_1 - t_3}{t_2 - t_4} \right)^{-1}. \quad (12)$$
Thus, to analyze the operation of a heat exchanger it is necessary to solve the nonlinear equation (12). Equation (12) was solved with respect to the temperature of the heating water at the outlet of the heat exchanger \( t_2 \). The consumption of heating water \( G_1 \) is determined by the formula

\[
G_1 = G_2 \left( \frac{t_3 - t_4}{\eta_1 - \eta_2} \right). \tag{13}
\]

As a method of numerical solution, Newton's method was used.

![Figure 2. Error of approximation of expression by 1-parabola (least squares method); 2-5 – by dependencies presented respectively in [17-21].](image)

5. Results
The experiment used the parameters of heat exchangers of all models with the tubes length of 2 m (table 1) and the number of sections 5, 7, and 10. The consumption of heating water \( G_1 \) was determined by the formula (13) after solving equation (12) with respect to the heating water temperature \( t_2 \). The temperature of the heating water at the inlet to the heat exchanger was taken equal to 70 °C. The pollution coefficient of the heat exchanger was taken equal to \( \beta = 0.8 \). The results of a numerical experiment are presented in Figures 3-4.

With a decrease in the heating surface area \( F_h \), the flow of heating water (Figure 3) tends to infinity. In real conditions, the value is limited by the available pressure at the subscriber input and the resistance of the fully open temperature controller.

As the area \( F_h \) increases, the flow asymptotically tends to the value \( G_1 = \frac{1000Q}{(t_1 - t_h)} \). With equal areas of \( F_h \), a heat exchanger with a smaller cross-section of the annular space provides heating of water to the required value with a lower consumption of heating water. Equal flowses \( G_1 \) can be obtained using heat exchangers of different models. Moreover, the larger the annular space, the more sections are required.

In all cases, the value of \( t_2 \) is bounded above by the value of the temperature of the heating water \( t_1 \) and from below by the temperature of the heated water \( t_4 \) at the entrances to the heat exchanger (Figure 4). For exceeding \( t_2 \) relative to the temperature schedule approved by the heat supply organization, penalties are prescribed in the contracts of subscribers with the heating network. With an incorrectly installed heat exchanger, flow restriction will lead to under-heating of cold water.
The heating surface area, chosen as an argument when plotting the graphs (Figures 3-4), varies discretely with a step depending on the model. The minimum value of $F_h$ for each model is different and is limited to an area of 1 section. The increase in the number of sections leads to an increase in pressure losses in the heat exchanger both on the heating and on the heated side, therefore, it is also not necessary to speak of an infinite increase $F_h$.

Figure 3. Change in the flow of heating water providing the required conditions.

Figure 4. Change in the heating water temperature at the outlet of the heat exchanger.
6. Conclusions
Having analyzed the presented results, we can draw the following conclusions:
- the maximum allowable speeds of the heated and heating water must be determined by the available pressure drops along the heating and heated sides of the heat exchanger;
- when choosing a heat exchanger, a heat exchanger with a smaller cross-section of the tubes and the annular space should be preferred;
- when choosing heat exchangers with an equal number of sections of different models, preference should be given to model, in which the heat exchanger has a smaller cross section.

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