Simulation of thermal characteristics of heat supply systems in variable operating modes

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Abstract. The operation of heat supply systems is often associated with the inefficient use of heat energy. New formulas have been obtained for calculation of variable thermal and hydraulic modes of heat exchangers of heat supply systems. The rather high correlation coefficient of the operational and design parameters of the heating system operation makes it possible to use the obtained formulas to optimize the variable operating modes of heat exchangers when using programmable regulators.

1. Introduction

One of the main problems of heating systems is the problem of efficient use of thermal power. Heat output is determined by the flow rate and temperature difference of the heat carrier

\[ Q = Gc(t_1 - t_2) = W(t_1 - t_2), \]

where \( Q \) is the thermal power, W; \( G \) is heat carrier flow rate, m\(^3\)/s; \( c \) and \( \rho \) are specific heat capacity, J/(kg\( \cdot \)K) and density, kg/m\(^3\) of heat carrier; \( t_1 \) and \( t_2 \) are initial and final temperatures of the heat carrier, °C; \( W \) is the equivalent of the heat carrier flow rate, W/K.

In order to increase energy efficiency at the heat source, temperature control graphs are applied. The main ways of central regulation are:

- qualitative regulation when the water temperatures in the supply \( t_{p1} \) and return \( t_{p2} \) mains of the heating network change depending on the outdoor temperature tout with a constant flow of the supply water \( G_p \);
- quantitative regulation when the temperature of the water in the heat supply network \( t_{p1} \) is constant during the entire heating period and the flow rate of the supply water \( G_p \) varies depending on the outdoor temperature;
- qualitative and quantitative regulation when both the temperature and the flow rate of the supply water vary depending on the outdoor temperature.

In Russian heat supply systems basically a qualitative method of central regulation is applied [1, 2] with an estimated temperature of heat carrier in the supply line from 150 to 95°C, and 70°C in the return line. In Denmark, particularly in Copenhagen, quantitative regulation is normally applied. The temperature graph of the heating networks is 120/50°C with a working pressure of the coolant of 2.5 MPa. Low-temperature heat supply from district heat sources using heat pumps is also used; during the heating season, the water temperature in the supply line is constant at 80°C, and during the non-heating season it is at 65°C [3]. In Germany in some cities the district heating sources, which operate
on temperature graphs with a temperature in the supply pipe of heating networks of 130-100°C depending on the outdoor temperature, are used. Qualitative and quantitative regulation carried out, as a rule, steps. For example, the temperature graph in Dresden consists of seven steps [4]. In China district heating systems and methods for central regulation of heat loads are very diverse. For example, the low-temperature graphs of qualitative regulation of 65/55°C, where the temperatures of water in the supply and return lines of the heating network vary linearly, are applied [5]. There are also the systems with a sufficiently high estimated heat carrier temperature of 120-110°C [6] with qualitative and qualitative-quantitative regulation and stepwise regulation. Temperature graphs are calculated according to the same equations that are adopted in Russian heating systems.

At the same time it is impossible to ensure rational modes of operation of heating systems only by optimizing the graphs of central regulation since heat networks carry a different load requiring different flow rates and temperatures of network water. Therefore, often the heat capacity of the heating system is not fully used by local heating and hot water systems which causes an excess of the return network water temperature, reduces generation of electrical energy, increases the cost of fuel for generation of thermal energy and reduces the overall efficiency of the heat source.

For example, Figures 1-4 show the operating modes of the Novosibirsk combined heat and power plant (CHP) on December 23-29, 2018. Depending on the outside temperature (Fig. 1), the temperature of the water in the supply and return pipelines of the heating network (Fig. 2) and the flow rate of the network water (Fig. 3) changed.

![Figure 1. Outdoor temperature on December 23-29, 2018.](image1)

![Figure 2. Network water temperature: $t_{p1}^{gr}$ – in the supply line on the graph of the heating network; $t_{p1}^{op}$ – the same operational; $t_{p2}^{gr}$ – in the return line on the graph of the heating network; $t_{p2}^{op}$ – the same operational; $t_{p2}^{cal}$ – the same calculation using the formula (9).](image2)
Figure 3. Network water flow rates: $G_{p1}^{op}$ – operational in the heating system supply line; $G_{p1}^{cal}$ – calculated in the supply line of the heating system by the formula (10); $G_{p2}^{op}$ – operating in the return line of the heating network.

The operating temperature of the return mains water $t_{p2}^{op}$ was higher than the required temperature according to the central regulation graph $t_{p2}^{gr}$ which indicates the irrational use of the thermal power of the heat supply system (see Fig. 2). Figure 4 shows that the thermal capacity produced by the CHP $Q_{prod}$ is more than that used by the heating and hot water supply systems $Q_{used}$ even taking into account the heat losses in the heat network $Q_{hl}^{used}$.

Figure 4. Thermal power of the heat supply system.

Irrational modes of operation of heat supply systems are related to the fact that local heating and hot water supply systems of consumers are joined according to different schemes (one-step and two-step) and operate according to different temperature graphs and at variable flow rates of heat transfer media. In this case, the regulatory system is calculated to maintain only the operating parameters. Therefore, it is necessary to develop effective methods for regulating and redistributing heat flows in variable operating modes in order to use the thermal energy rationally.
2. Materials and methods
When the outside temperature changes, the heat power of heat exchangers, flow rates and temperatures of heat carriers are changing depending on the mode of operation:

- the amount of heat given by the primary heat carrier
  \[ dQ = -G_p c_p \rho d\tau_p; \quad (2) \]

- the amount of heat received by the secondary heat carrier
  \[ dQ = G_h c_h \rho d\tau_h; \quad (3) \]

- the amount of heat transferred by heat exchange through the separating wall
  \[ dQ = kF \Delta t^\prime, \quad (4) \]

where \( k \), W/(m²K) and \( F \), m² are the heat transfer coefficient and heat exchanger area, \( \Delta t^\prime \) is the average logarithmic temperature difference of heat carriers, °C.

In variable operation mode the temperatures of the primary \( t_2 \) and secondary \( t_2 \) heat carriers at the outlet of the heat exchanger may be unknown; as well as the temperature at the outlet of the heat exchanger and the flow rate of one of the heat carriers or the temperature of one of the heat carriers and the flow rate of another heat carriers, which makes it impossible to use the balance equations (2)-(4) for calculating the variable modes. Therefore, the existing theories of calculating variable modes of heat exchangers are based on the use of constant dimensionless parameters [7] which reduce the number of unknown variables. So, for heat exchangers operating at equal and constant flow rates of heat carriers, the dimensionless heat transfer coefficient (the number of transfer units) \( NTU \) is used which can be determined by the formula [8, 9]

\[ NTU = (kF)_c / W_{min}, \quad (5) \]

where \((kF)_c\) is the product of the heat transfer coefficient and the heat exchanger area in the installation mode of operation, W/K; \( W_{min} \) is the minimum equivalent flow rate, W/K.

In heat exchangers of heat supply systems, the flow rates of heat carriers changes very significantly, therefore \( NTU \), also called the heat exchanger parameter, \( \Phi_0 \), is determined by [7]

\[ NTU = \Phi_0 = (kF)_c / \left( W_{pc} W_{hc} \right)^{0.5}, \quad (6) \]

where \( W_{pc} \) and \( W_{hc} \) are installation (estimated) equivalents of the flow rates of the primary and secondary heat carriers.

Formulas (5) and (6) do not include heat carrier temperatures, which reduces the number of unknown variables. At the same time, \( NTU \) can be affected by all quantities that determine the change in heat transfer coefficient. To determine the applicability of relations with constant parameters, simulation of various variable operating modes of a counter current water-to-water heat exchanger was performed when its thermal power \( Q \) was changed. Based on the numerical dependencies \( NTU(Q) \) obtained in [10], using mathematical simulation we obtained the formulas allowing the description of the variable parameter of the heat exchanger in any mode of operation.

The influence of heat carrier flow rates on the heat exchanger parameter can be described as

\[ NTU(W) = \frac{Q}{\Delta t^\prime (W_p W_h)^{0.5}} \left\{ A_1 \left[ \left( \frac{W_p}{W_h} \right)^{0.5} Q + C_1 \right] \left( \frac{W_p W_h}{W_{pc} W_{hc}} \right)^{0.5E_1} + \frac{Q}{Q} \left( \frac{W_p W_h}{W_{pc} W_{hc}} \right)^{0.5} \right\}, \quad (7) \]

where \( A_1, B_1, C_1, D_1, E_1 \) are coefficients, taken depending on the mode of operation of the heat exchanger (temperature graph) in Table 1.
Table 1. The coefficients in formula (7).

| The coefficients | Heat exchanger operation mode / Temperature graph |
|------------------|--------------------------------------------------|
|                  | I stage of heat exchanger of hot water supply |
|                  | $t_h=\text{const}$ | II stage of hot water heat exchanger and heating heat exchanger |
|                  | $t_p=\text{const}$ | Qualitative regulation $t_p=\text{const}$ | Qualitative and quantitative regulation $t_p\neq\text{const}; t_h=\text{const}$ |
| $A_1$            | -1               | 1               | 1               |
| $B_1$            | 0.008            | 0.008           | 0.04            |
| $C_1$            | 0.18             | 0.012           | 0.015           |
| $D_1$            | -0.2 для $(t_{pc1}-t_{hc1})\geq 50 ^\circ C$ | 0               | 0               |
| $E_1$            | 0                | 0               | 1               |

The effect of heat carrier temperature on the heat exchanger parameter in its variable operation modes can be determined by the following expression

$$NTU(t) = \frac{Q}{(t_{pc1}-t_{h1})\left[\frac{W_{pc}W_{hc}}{Q_c}\right]^{0.5}} + A_2 + B_2\left[\frac{\delta t_{\min}}{\delta t_{\max}}\right] + C_2 + D_2 + E_2,$$

where $\delta t_{\min}$ and $\delta t_{\max}$ are minimum and maximum temperature difference between the heat transfer media at the ends of the heat exchanger; $A_2, B_2, C_2, D_2, E_2$ are coefficients, taken in Table 2.

Table 2. The coefficients in formula (8).

| The coefficients | Heat exchanger operation mode / Temperature graph |
|------------------|--------------------------------------------------|
|                  | II stage of hot water heat exchanger and heating heat exchanger $t_{h1}=\text{const}; Q=Q_c$ |
|                  | I stage of heat exchanger of hot water supply $t_p=\text{const}$ |
|                  | Single stage hot water heat exchanger $t_p$ decreases |
|                  | Heating heat exchanger. Qualitative and quantitative regulation $t_p$ increases |
|                  | II stage of heat exchanger of hot water supply $t_p$ decreases |
| $A_2$            | 1               | 2.3            | 1.3            | 1.7           | 0.98          |
| $B_2$            | \begin{align*} & (t_{pc1}-t_{hc1})>100 ^\circ C \quad & (t_{pc1}-t_{hc1})>100 ^\circ C \\ & \frac{[W_{pc}W_{hc}]}{Q_c} & \frac{[W_{pc}W_{hc}]}{Q_c} \end{align*} & \begin{align*} & (t_{pc1}-t_{hc1})\leq 100 ^\circ C \quad & (t_{pc1}-t_{hc1})\leq 100 ^\circ C \\ & -1 & -1 \end{align*} |
| $C_2$            | \begin{align*} & (t_{pc1}-t_{hc1})>50 ^\circ C \quad & (t_{pc1}-t_{hc1})>50 ^\circ C \\ & 0.9 & 0.9 \end{align*} & \begin{align*} & (t_{pc1}-t_{hc1})\leq 50 ^\circ C \quad & (t_{pc1}-t_{hc1})\leq 50 ^\circ C \\ & -0.98 & -0.98 \end{align*} |
| $D_2$            | \begin{align*} & (t_{pc1}-t_{hc1})>50 ^\circ C \quad & (t_{pc1}-t_{hc1})>50 ^\circ C \\ & -0.009 & 0.9 \end{align*} & \begin{align*} & (t_{pc1}-t_{hc1})\leq 50 ^\circ C \quad & (t_{pc1}-t_{hc1})\leq 50 ^\circ C \\ & -0.67 & 1 \end{align*}
3. Results and discussion
At known flow rates of heat carriers, the return water temperature
\[
t_{p2} = \frac{Q_e}{\varepsilon W_h} + t_{b1},
\]
where \(\varepsilon\) is heat exchanger efficiency, defined by formula from [7]
\[
\varepsilon = \frac{1}{\left[ a (W_h / W_p) + 1/NTU(W) \cdot (W_h / W_p)^{0.5} \right]}.
\]
where \(a = 0.35, b = 0.65\) are constant coefficients taken according to [7], NTU(\(W\)) calculated by formula (7).
At known heat carrier temperatures, the equivalent of network water flow is (NTU(\(t\)) calculated by (8)):
\[
W_p = \left( \frac{Q_e}{NTU(t) \Delta t} \right)^2 \cdot 1/W_h,
\]
Using the obtained relations (9) and (11), the temperature and flow rate of the return water were calculated (see Fig. 2, 3) and correlation and regression analysis was performed. The correlation coefficients were determined by the equation
\[
r = \left( \frac{n \sum x_i y_j - \sum x_i \sum y_j}{\left[ n \sum x_i^2 - \sum (x_i)^2 \right] \left[ n \sum y_j^2 - \sum (y_j)^2 \right]} \right)^{-0.5},
\]
where \(n\) is the sample size; \(x_i\) is the values of operating variables; \(y_i\) is values of calculated variables.
The correlation coefficient for the return water temperature is \(r_t = 0.802\); for heat carrier flow rate \(r_G = 0.981\). Analysis of the results showed that the correlation relationship between the variables is medium and strong, which indicates sufficient accuracy of the obtained relations.

Conclusions
New formulas have been obtained that allow one to describe the effect of flow rates and temperatures of heat carriers on the heat exchanger parameter in variable operating modes. The resulting formulas can be used to configure programmable regulators which will allow correcting inefficient modes of operation of the heating system.

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