Engineering method for calculating the operation modes of the heat supply station with the associated heat supply

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Abstract. The effective use of heat power in heat supply systems requires the development of new methods for central and local regulation of heat supply systems, the use of heat supply schemes with associated heat supply. The development of such technologies requires modeling and systematic calculations of variable modes of joint operation of local heat-consuming systems. Currently used methods for calculating such operating modes of heat exchangers usually use constant parameters of heat exchangers. Such methods are not applicable in cases for large changes in temperatures and flow rates. Therefore, the calculation of variable modes of tied heat exchangers is usually performed by the method of successive approximations and is very time-consuming, which complicates the construction of a control system. In the present work, a fairly simple engineering technique is proposed for calculating the operating modes of heating stations of heat supply systems with associated heat supply. The basis of this technique is the use of previously derived analytical formulas that describe the change in the parameters of heat exchangers for various schemes of heating stations and temperature graphs of central regulation. The developed method allows the calculation of variable operating modes of the heat supply system without resorting to the method of successive approximations. The efficiency of the proposed methodology was tested on the operational data of the central heating station in the Kalinin district of Novosibirsk.

1. Introduction
Nowadays functioning of district heating systems is characterized by a number of problems, one of which is the incomplete use of thermal power. The solution to this problem requires, in particular, the development of new methods for central and local regulation of heat supply systems [1–4]. This requires a systematic calculation of the variable modes of joint operation of heat-consuming systems, since in certain operating modes the heating system may not receive a significant amount of heat. Methods for calculating the variable operating modes of heating stations were developed for unassociated heat exchangers, when the operation of a separate heat exchanger does not affect the operation of other heat exchangers. For such schemes, the consumption of network water will be greater than in schemes with associated heat supply, when heat flows are redistributed between the heat exchangers depending on their operating conditions. Existing theories of calculating variable operating modes of heat exchangers [5, 6] are based on the use of constant dimensionless complexes, one of which is the heat exchanger parameter that determines the heat transfer coefficient through the estimated flow rates of primary and secondary heat carriers:
\[ \Phi_0 = kF / (W_{pc} W_{hc})^{0.5} , \]  

where \( kF \) – is the product of the heat transfer coefficient by the area of the heat exchanger; \( W_{pc}, W_{hc} \) – calculated water equivalents of flows of primary and secondary heat carriers, W/K.

Parameter \( \Phi_0 \) is considered constant regardless of the operating mode of the heat exchanger. Theories based on the use of constant dimensionless parameters make it possible to calculate the operating modes of a group of heat exchangers in conditions of insufficient initial data; however, they are of limited use, since they do not take into account the influence of heat carrier temperatures on the heat transfer coefficient in the heat exchanger. They are not applicable also with a large change in the flow of heat carriers, which is typical for heating stations with associated heat supply.

Earlier in [7], the range of applicability of existing calculation theories based on the use of constant parameters of heat exchangers was determined, and a new calculation procedure was developed that allows one to calculate variable modes of operation of heat supply systems with associated heat supply [8]. However, this technique is constructed by the method of successive approximations and is rather time-consuming; it is not easy to use in real engineering calculations, which complicates the design of schemes of heating stations and the management of their work. To develop a simpler approach, in [9, 10] a systematic simulation of the operation of heat exchangers in various modes was performed and analytical formulas were derived that describe changes in the parameters of heat exchangers for various schemes of heating stations and temperature graphs of central regulation. The aim of this work is to construct an engineering methodology for calculating variable operating modes of a heating station based on formulas obtained in [9, 10]. The proposed method allows us to calculate the variable operating modes of a heating station without using the method of successive approximations.

2. The method of calculating the variable modes of operation of the heating station

In order to develop and verify the calculation methodology, the operational data of the central heating station (CHS) in the Kalinin district of Novosibirsk were used. Heat supply to the central heating station is carried out from the CHP-4 in Novosibirsk. In central heating station was implemented associated supply of heat, the hot water heat exchangers were connected in a two-stage scheme with the limitation of the flow rate of network water (Fig. 1). Estimated heat loads: for heating and ventilation \( Q_{ovmax}=5.22 \text{ MW} \); for hot water supply \( Q_0=3.30 \text{ MW} \).

![Figure 1. The scheme of Central Heating Station with associated heat supply.](image-url)
In fig. 1 are shown:
- Temperatures of network water:
  \( t_{p1} \) – in the heat supply line; \( t_{h2}^l \) – in the supply line of the heating network; \( t_{h1}^h \) – in the return pipe of the heating system; \( t_{p2} \) – in the return heating network; \( t_{p1}^h \) and \( t_{p2}^h \) – at the inlet and outlet of II stage of HWS heater; \( t_{p1}^h \) and \( t_{p2}^h \) – at the inlet and outlet of heating heat exchanger.
- Temperatures of heated water:
  \( t_{h1}^l \) – in the cold water supply system at the inlet to the I stage of HWS heater; \( t_{h2}^l \) – after the I stage of HWS heater; \( t_{h1}^l \) and \( t_{h2}^l \) – at the inlet and outlet of II stage of HWS heater; \( \Delta t^{opt} \) – in the circulating line of HWS.
- Equivalents of network water flow rates:
  \( W_p \) – in the heat supply network; \( W_p^h \) – at II stage of the HWS heater; \( W_{p1}^h \) – from the heat supply network to the heating system; \( W_{h}^h \) – in the heating system.
- Equivalents of heated water flow rates:
  \( W_h^1 \) – in I stage of the HWS heater from cold water supply; \( W_{h}^{cir} \) – in the circulation line of HWS; \( W_{h}^h \) – in II stage of the HWS heater.

When calculating the variable operating modes of the heating stations of heat supply systems, various tasks arise, for the solution of which it is necessary to determine the temperatures and flows of the primary and secondary heat carriers after each heat exchanger, as well as the redistribution of heat flows between the heat exchangers depending on their operating modes. Such particular problems arise when it is necessary to regulate non-optimal modes of joint operation of heat-consuming systems. The main tasks that determine the efficiency of the heat supply system are: determining the temperature of the return network water, which shows the degree of use of the available heat capacity and determining the flow rate of water in the heat supply network, the value of which depends on the cost of pumping the heat carrier. This article is devoted to the solution of these problems.

2.1. Determination of temperature of water returned to the heat supply network
So, the first task is to determine the temperature of the return network water. It can be determined if equivalents of the primary \( W_p \) and secondary \( W_h \) flow rates of heat carriers are known. When two-stage schemes of hot water supply heaters are installed at consumers’ heating stations, the temperature of the water returned to the heat supply network is the temperature of the water after the first-stage hot water heat exchanger: \( t_{p2}^l = t_{p2} \), see fig. 1.

The heat capacity of the heat exchanger can be found as follows [11]:

\[
Q_h = kF\Delta t = \varepsilon W_{min}D_I = W_{max}\Delta t_{min},
\]

where \( \Delta t \) – is the average temperature difference of the heat carriers in the heat exchanger; \( \varepsilon \) – heat exchanger efficiency; \( W_{min}, W_{max} \) are the smaller and the larger from the two equivalents of the flow rates of heat carriers; \( D_I \) – the temperature difference between the primary and secondary heat carriers at the inlet to the heat exchanger; \( \Delta t_{min} \) – a smaller decrease in the temperature of the carrier in the heat exchanger. For the 1st stage heat exchanger of: \( D_I = t_{p1}^l - t_{h1}^l \); \( \Delta t_{min} = t_{p1}^l - t_{p2}^l = t_{p1}^l - t_{p2} \).

For the heat exchanger of the 1st stage, the flow rate of heated water will always be lower, since in the heat exchanger of the 1st stage the flow rate of the network water is not less than the flow rate of the water for the heating system (if no water consumption in the hot water supply system). Moreover, the scope of these schemes is limited by the ratio of heat fluxes of hot water supply and heating \( Q_h/Q_a \leq 1 \). The temperature of the return water will depend on the change in the thermal power of the
heat exchanger of the first stage, as illustrated in Fig. 2. In turn, this thermal power will change with a change in the temperature of the outdoor air $t_{ext}$ and the amount of water consumption in the hot water supply system during the day $W_h(z)$.

In Fig. 2 shows the thermal power of the heating system $Q_o$, heat exchangers of hot water supply of I and II stages: $Q_{h1}, Q_{h2}$ depending on the outdoor temperature $t_{ext}$.

In fig. 3 shows the graphs of changes in the thermal power of hot water heat exchangers depending on water consumption. The magnitude of water consumption $W_h(z)$ is proportional to the load of the hot water heat exchanger of the first stage at the current outdoor temperature: $Q_{h1}(t_{ext}, z) = Q_{h1}(t_{ext})Q_h(z)$.

The temperature of the return water can be determined from expression (2) depending on the outdoor temperature $t_{ext}$ and time $z$:

$$t_{p2}(t_{ext}, z) = Q_{h1}(t_{ext}, z) \left( \frac{1}{e^{t_{ext}W_h(z)}} - \frac{1}{W_p(z)} \right) + t_{h1}.$$  
(3)
The efficiency of the heat exchanger can be found by the formula [11]:

\[
\varepsilon(t_{ext}, z) = \frac{1 - \exp \left[-\Phi(t_{ext}, z) \sqrt{\frac{W_p}{W_h} \left(1 - \frac{W_h}{W_p}\right)}\right]}{1 - \frac{W_h}{W_p} \exp \left[-\Phi(t_{ext}, z) \sqrt{\frac{W_p}{W_h} \left(1 - \frac{W_h}{W_p}\right)}\right]} \leq 1, \quad (4a)
\]

or according to the simplified formula proposed by E.Ya. Sokolov [5]:

\[
\varepsilon(t_{ext}, z) = \left( a \frac{W_h(z)}{W_p} + b + \frac{1}{\Phi(t_{ext}, z)} \sqrt{\frac{W_h(z)}{W_p}} \right)^{-1} \leq 1, \quad (4b)
\]

where \(a = 0.35; b = 0.65\) – are constant coefficients taken according to [5] for the countercurrent flow of heat carriers, but the heat exchanger parameter is not taken as a constant value, but is determined by the formula proposed in [9]. Given that in the first stage of the heater, the temperature of the heated water \(t_{h1}\) at the inlet is constant, equal to the temperature of the water in the cold water supply, and the temperature of the heating water \(t_{p1}\) increases with increasing thermal power of the heat exchanger, this formula will have the form:

\[
\Phi(t_{ext}, z) = \frac{Q_h^1}{\Delta t \sqrt{W_p W_h(z)}} \left[0.2 - \left(\frac{W_p}{W_h(z)}\right)^{0.08 + 0.18 \left(\frac{Q_h^1}{Q_h^0}\right)} + \frac{Q_h^0}{Q_h^1} \left(\frac{W_p W_h^1(z)}{W_p W_h^0}\right)^{0.5}\right], \quad (5)
\]

where \(Q_h^0\) and \(\Delta t_e\) are the installation values of thermal power and temperature head in the heat exchanger, determined during the design calculation.

2.2. Determination of network water flow

In the case when the temperatures of the heat carriers are known and their flow rates are unknown, a formula for determining the variable parameter of the heat exchanger was obtained in [10]. In the heat exchanger of hot water supply of the 1st stage, the total consumption of network water always passes. This flow rate must be determined. For the operating conditions of the heat exchanger of the 1st stage, the formula of the variable parameter of the heat exchanger will have the following form:

\[
\Phi(t_{ext}, z) = \frac{Q_h^1}{(t_{p1}^1 - t_{h1}^1) W_p W_h} \frac{1}{1.5} \left[2.6 - \left[1.26 + 0.02(t_{p1} - t_{h1}) \left(\frac{\Delta t_{min}}{\Delta t_{max}}\right)\right] \right] + 1.6. \quad (6)
\]

In the formula (6), \(t_{p1}^1\), \(t_{h1}^1\) are calculated (installation) temperatures of the network and heated water at the inlet to the heat exchanger, \(\Delta t_{min} = t_{p1} - t_{h1}\); \(\Delta t_{max} = t_{p2} - t_{h1}\) are smaller and larger temperature difference of the heat carriers at the ends of the heat exchanger.

Using formulas (6) and (2), the equivalent flow rate of network water, W/K, can be determined by the formula:

\[
W_p = W_h(z) \left[\Phi(t_{ext}, z) \frac{\Delta t}{\Delta t_{min}}\right]^2. \quad (7)
\]

3. Results and discussion

For the selected range of outdoor temperatures on January 21-28, 2018, presented in Fig. 4, the operation modes of the CHS were calculated.
The calculated return water temperature $t_{p2}^{cal}(t_{ext})$ was compared with the operating temperature $t_{p2}^{exp}(t_{ext})$ (Fig. 5). The estimated flow rate of network water, $G_p^{cal} = \frac{3.6}{c} W_p \ t/h$, was compared with the operational flow rate of network water (Fig. 6) ($c$ – is heat capacity of water, J/(kg·K)).

The correlation coefficient was determined by the equation:

$$r = \left( n \sum x_i y_i - \sum x_i \sum y_i \right) \left[ \left( \sum x_i^2 - \left( \sum x_i \right)^2 \right) \left( \sum y_i^2 - \left( \sum y_i \right)^2 \right) \right]^{0.5},$$

where $n$ – is the sample size; $x_i$ – are operational values; $y_i$ – are calculated values.

In Fig. 5 shows the temperatures of the supply water in the supply $t_{p1}^{exp}$ and return $t_{p2}^{gr}$ pipelines according to the temperature graph of the central regulation; the operational temperatures of the water in the supply $t_{p1}^{exp}$ and return $t_{p2}^{exp}$ pipelines, as well as the calculated return water temperature $t_{p2}^{cal}$ according to the formula (3).

The correlation coefficient for return water temperature $r_t = 0.735$.

In fig. 6 shows the operating flow rates of network water in the supply $G_{p1}^{exp}$ and return $G_{p2}^{exp}$ pipelines and the estimated flow rate of network water $G_{p1}^{cal}$, determined by the formula (7).

The correlation coefficient for network water flow $r_G = 0.806$. 
4. Conclusion
The engineering method proposed in this paper for calculating the operating modes of heating stations of heat supply systems with associated heat supply is quite simple. In fact, it can be used online for the operational management of heat supply networks. The efficiency of the method is demonstrated on real operational data. Simulation data are consistent with experimental data with fairly high accuracy. Thus, the proposed technique allows configuring the regulation system and adjusting the irrational modes of operation of heating stations.

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