



MODERNIZATION OF HYDRAULIC MODES OF HEATING NETWORKS IN TYUMEN

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ABSTRACT

The article presents a proposal for solving the problem of excessive temperature of the return line based on the results of the study of the hydraulic modes of heat networks from the Tyumen CHP-2. The authors propose a modernized scheme of a group heating point (GHP), which is installed at the border of the balance of the main and district heating network. For this scheme, a physico-mathematical model is created, the purpose of which is to derive a modernized formula for the mixing ratio for a GHP, with which it is possible to determine the necessary actions to increase the efficiency of heat supply system control and the amount of time required for this. The calculation is made for the existing heat chambers of the city of Tyumen using the GHP scheme, the results of which revealed certain patterns.

Key words: energy efficiency of the heat supply system, group heating point, main and distribution heat network, mixing ratio.

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1. INTRODUCTION

Currently, the main problem of centralized heat supply in the Russian Federation is the imbalance of the heating network, which over the years impairs the hydraulic stability of heat supply systems. In particular, it is observed in populous cities with very extensive and “hard-to-control” heat supply systems.

Conducting a study to identify the factors that reduce the energy efficiency of centralized heat supply using the example of main heating networks in the circuit of the Tyumen CHP-2 [1] has become a task the solution of which will ensure uninterrupted operation of heat supply systems.

The main factor that reduces the energy efficiency of centralised heat supply in the city of Tyumen is a significant deviation from the norm in terms of flow rate and return temperature. Moreover, excessive flow rate is directly related to excessive temperature of the return pipeline, since this deviation from the norm leads to an increase in the heat carrier flow rate, otherwise the heat carrier parameters are reduced, and the consumer does not get the amount of heat necessary for a comfortable stay in the room.

2. MATERIALS AND METHODS

To solve the problem of the increased return temperature, a control scheme was proposed with the participation of a group heating point (hereinafter referred to as the GHP) located on the border of the balance sheet attribution between the main and distribution heat network. So, having considered the existing GHP scheme, a modernized scheme was proposed (Figure 1) with the possibility of more efficient control of the main parameters of heat supply. In the existing GHP schemes [2] in the form of a temperature controller, a two-way regulating device is used, but for more efficient heat distribution it is proposed to use a three-way regulating device, with installation of pumping equipment on the return heat pipeline.

Thus, it is proposed to create a method for calculating the amount of return heat carrier from the return heat pipeline to the supply one through the use of a three-way regulating device at the GHP.

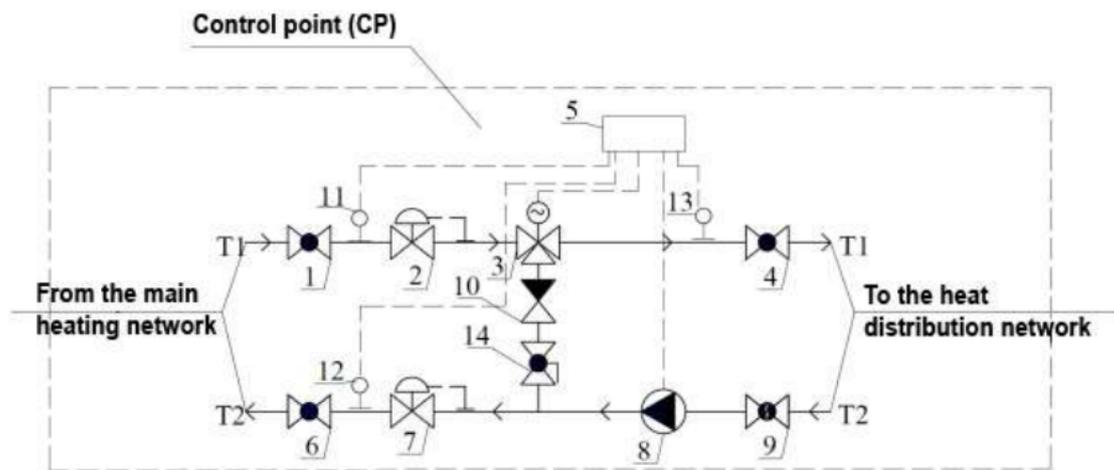


Figure 1 GHP scheme:

1,6,4,9 – locking device; 2 – downstream pressure regulator;
3- three-way control valve; 5 – controller; 7 – upstream pressure regulator;

8 – pump; 10 – check valve; 11 – temperature sensor on the supply pipe of the main heating network; 12 – temperature sensor on the return pipe of the main heating network; 13 – temperature sensor on the supply pipe of the main heating network; 14 - ball valve with manual reconfiguration function.

The essence of the modelling process is to derive a pattern of changes in the parameters of heat relative to the control modes in the considered period of time. The model will show exactly what actions need to be taken to increase the efficiency of control and how long it will take to stabilize the temperature and heat process.

The possibility of deriving regularities of the control process with respect to time is proposed to be realized thanks to the derivation of a coefficient that will fully reflect the mutual influence of changes in parameters during qualitative and quantitative control.

The main parameters in the regulation are flow, pressure and temperature. The flow rate and pressure characterize the capacity of the heat network, and the flow rate and temperature mode characterize the heat load, so it was decided to use the heat balance formula and the capacity of control devices [3].

The factual and calculated structure of the division of heating networks between the main and district heating networks is presented in Figures 2 and 3.

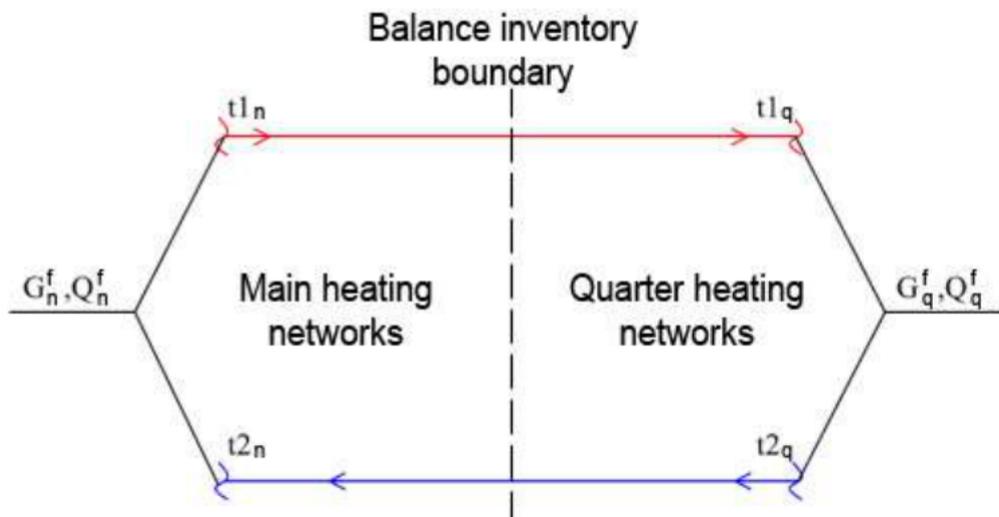


Figure 2. The existing distribution structure of the main and district heating networks

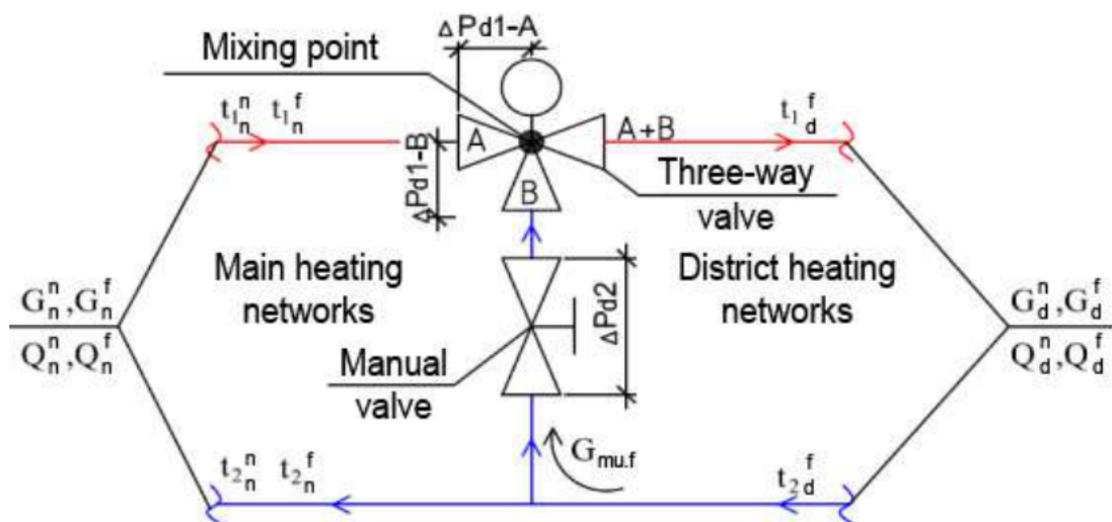


Figure 3. Design scheme for the use of the GHP in a physico-mathematical model:

Where G_n^n - normalized flow rate of the main heating network, G_n^f - factual flow rate of the main heating network, Q_n^n - normalized heat load for the main heating network, Q_n^f - factual heat load for the main heating network, G_d^n - normalized flow rate of the district heating network, G_d^f - factual flow rate of the district heating network, Q_d^n - normalized heat load for the district heating network, Q_d^f - factual heat load for the district heating network, t_{1n}^n, t_{2n}^n - normalized supply and return heat carrier temperature for the main heating network, t_{1n}^f, t_{2n}^f - factual supply and return heat carrier temperature for the main heating network, t_{1d}^f, t_{2d}^f - factual supply and return heat carrier temperature for the district heating network, $\Delta Pd1-A$ - pressure loss at the three-way valve of port A, $\Delta Pd1-B$ - pressure loss at the three-way valve of port B, $\Delta Pd2$ - pressure loss at the manual valve, - - factual heat flow rate for make-up.

It should be noted that when introducing a modernized control scheme, the main parameters of the heat carrier must be divided into factual and normalized (nominal). In case of deviations, the control process begins, in which the make-up of the supply heat line is performed from the return one.

3. RESULTS AND DISCUSSION

The purpose of the algorithm for calculating the physical and mathematical model is to calculate the mixing ratio considering the existing flow rates of the heat carrier and pressure losses at the three-way valve. To do this, we determine the maximum heat flow rate $G_{f,max}$ which is necessary for district heating networks to obtain the normalized heat load Q_d^n without considering mixing.

From the formula of heat balance, we get:

$$G_{f,max} = \left(\frac{Q_d^n}{(t_{1n}^f - t_{2n}^f) \cdot C} \right), \text{ kg/s} \quad (1)$$

where Q_d^n - normalized heat load for the district heating network, kW; t_{1n}^f, t_{2n}^f - factual supply and return heat carrier temperature for the main heating network °C; C - specific heat of the heat carrier in the temperature range t_{1n}^f, t_{2n}^f , kJ/(kg·°C).

Since the calculation is carried out without considering mixing, we assume that:

$$t_{1n}^f = t_{1d}^f = t_{1n}^n \text{ and } t_{2n}^f = t_{2d}^f, \text{ whereas } t_{2n}^n < t_{2n}^f \text{ and } Q_d^n = Q_n^n$$

$$\text{Therefore: } G_{f,max} \geq G_n^f \quad (2)$$

The amount of make-up heat flow rate for admixing:

$$G_{mu} = G_{f,max} - G_n^f, \text{ kg/s} \quad (3)$$

Pressure loss at the three-way and manual control valves will be expressed from the formula of the capacity of control devices [3]

$$\Delta P_v = \left(\frac{G_v}{K_{V,v}} \right)^2, \text{ bar} \quad (4)$$

where G_v - water flow rate through the valve, m³/h; $K_{V,v}$ - valve capacity, (m³/h)/bar^{0.5}

To calculate the mixing ratio in the control mode, it is necessary to calculate the flow rate depending on the capacity and pressure loss at the control device. Figure 3 demonstrates that port A shows the factual flow rate of the main heating network G_n^f . Port B shows the make-up rate G_{mu}^f which, when mixed with G_n^f , compensates for the normalized heat load Q_d^n necessary for the district heating network. The resulting flow rate during mixing will be factual (G_d^f). To obtain a modernized formula, the formula for the mixing ratio (for the heat carrier flow rate) [4] and the transformed formula 4 were taken:

$$\mu = \frac{G_{mu}}{G_n} = \frac{K_{VS-B} \sqrt{\Delta P_{K1-A}}}{K_{VS-A} \sqrt{\Delta P_{K2-B}}} \quad (5)$$

where ΔP_{d1-B} - pressure loss at the port B three-way valve, bar; ΔP_{d1-A} - pressure loss at the port A three-way valve, bar; K_{VS-A} - normalized capacity of the port A three-way valve, m³/h; K_{VS-B} - normalized capacity of the port B three-way valve, m³/h.

To simplify the calculation of the mixing ratio in the nominal valve operation mode, it is necessary to clarify that the nominal capacity is obtained as a result of factory tests, which take place when creating resistance of 1 bar on a control device [3], therefore ΔP_{d1-A} and ΔP_{d1-B} values are subject to reduction and the final formula for the nominal mixing ratio will be:

$$\mu_n = \frac{K_{VS-B}}{K_{VS-A}} \quad (6)$$

Comparing formulas 4, 6 and the flow design scheme of the heat carrier in the mixing modes (Figure 3), we obtain the formula for the factual mixing ratio μ_f which determines the ratio of water admixture at the mixing point of the supply and return heat carrier.

$$\mu_f = \left(\frac{G_{mu}}{\sqrt{\Delta P_{d1-B} + \Delta P_{d2}}} \right) / \left(\frac{G_n}{\sqrt{\Delta P_{d1-A}}} \right) \quad (7)$$

where ΔP_{d1-B} - pressure loss at the port B three-way valve, bar; ΔP_{d1-A} - pressure loss at the port A three-way valve, bar; ΔP_{d2} - pressure loss at the manual valve, bar.

Having obtained the exact value of the mixing ratio, we can determine the temperature of the supply heat carrier regarding μ . For this, it is necessary to derive the t_{1C} value from the formula of the mixing ratio (according to the heat carrier temperature) [4], which in this case is equal to t_{1f} and the value of $t_1 = t_{1n}$ and $t_2 = t_{2n}$, then the desired formula will be:

$$t_{1f} = \frac{t_{1n} + t_{2n} \cdot \mu_f}{(\mu_f + 1)}, \text{ } ^\circ\text{C} \quad (8)$$

In this case, the temperature of the return main heating network t_{2n} will be equal to the factual temperature of the return heating network t_{2f} .

The temperature value t_{1f} of the supply heat line makes it possible to determine the heat load that can be obtained by admixing. This value will be below the norm, as there is a decrease in the temperature difference between the supply and return heat lines, therefore, to determine the time to reduce the heat load, you need to know the factual flow rate that can be supplied in the GHP control mode. For this, you need to calculate the amount of heat left not consumed by the end user because of which there is an excess of the return heat carrier temperature.

The formula for calculating the unused amount of heat is:

$$Q_{\text{mu}} = G_n^f \cdot (t_{2n}^f - t_{2n}^n) \cdot C, \text{ kW} \quad (9)$$

where t_{2n}^n - normalized return heat carrier temperature for the main heating network, $^\circ\text{C}$; t_{2n}^f - return heat carrier temperature for the main heating network, $^\circ\text{C}$; G_n^f - heat flow rate in the main heating network; c - heat capacity equal to 4.187, kJ/kg \cdot $^\circ\text{C}$.

It should be noted that $t_{2n}^f \leq t_{2n}^n$ and $t_{2n}^f = t_{2d}^f$.

In formula 9, the temperature mode was taken based on the physical movement of the heat carrier from the return heat line to the supply one (Figure 2). The flow rate of heat return to the supply line will be calculated based on the heat balance formula and will be:

$$G_{\text{superheating}} = \left(\frac{Q_{\text{mu}}}{(t_{1n}^f - t_{2n}^f) \cdot C} \right), \text{ kg/s} \quad (10)$$

where t_{1n}^f - supply main heating network temperature, $^\circ\text{C}$.

Initially, it is necessary to determine how much heat can be returned from the return heat line to the supply one. This value will be the heat flow rate from the superheated return heat line $G_{\text{superheating}}$. In turn, $G_{\text{superheating}}$ is determined based on the heat load $Q_{\text{superheating}}$, which is in the flow rate of the heating network G_n .

In this regard, the superheated flow rate $G_{\text{superheating}}$ will affect the make-up heat flow rate G_{mu} , which in fact will go to feed into the supply heat line for admixing and will be able to take this effect into account and is the factual flow rate of the supply heat line from the return heat carrier to the supply heat line $G_{\text{mu.f}}$.

$$G_{\text{mu.f}} = G_{\text{superheating}} - G_{\text{mu}}, \text{ kg/s} \quad (11)$$

The factual heat flow rate for the make-up $G_{\text{mu.f}}$ may be negative. This is because the flow rate from the superheated heat carrier may not be enough, and this will affect the heat flow rate that goes into the heating network.

Note that $G_{\text{mu.f}}$ will show the possibility of effective redistribution of the heat flow from the superheated return heat carrier to the supply one. This value can be either positive or negative. In this regard, for negative values of $G_{\text{mu.f}}$, the formula for determining the factual flow rate G_f^* will be:

$$G_f^* = G_{\text{mu}} + G_{\text{mu.f}} + G_n, \text{ kg/s} \quad (12)$$

For positive values of $G_{mu.f}$, the formula will be:

$$G_f^{**} = G_{mu} + G_n, \text{ kg/s} \quad (13)$$

For general physical modelling, the time factor over which the superheating of the return heat carrier will be eliminated is of great importance. The formula for calculating the time spent on eliminating superheating, based on the formula of heat balance, and considering the number of hours of operation, has the form:

$$n = \left(\frac{Q_n^n}{G_f^{**} \cdot (t_{1f} - t_{2f}) \cdot C} \right), \text{ h} \quad (14)$$

During the time of operation of the heat exchanger (n), a decrease in the heat load below the normalized one will be observed; therefore such control is possible during the night time when the heat load on the hot water system will be minimal.

4. CONCLUSIONS

1. The current norms regulating the calculation of the heat losses of buildings, taking into account the single ventilation of heated rooms, have caused a situation in which the design heat capacity of heating systems exceeds the actual requirements of buildings in the heat energy for heating.
2. If the parameters of the heating network are oriented towards a European approach to apartment ventilation, there is no need to raise the maximum temperature of the heat carrier to a virtually unrealistic design value of 150 °C. For a satisfactory heat supply, 124 °C would be sufficient.
3. The technical specifications for heat supply must be designed in such a way that they strictly regulate the real obligations of the buyer and seller of the thermal energy and stimulate its rational use while maintaining the existing hydraulic conditions of the network.
4. Maintaining the first comfort condition in residential buildings while meeting energy efficiency requirements can be achieved by combining sanitary and hygienic requirements and reducing energy parameters.

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