Model of heat and mass transfer in the heating system pipe

V O Kaledin, E S Viachkin, A E, Gileva, E A Viachkina, D A Galgin and A D Ulyanov
Department of Mathematics and Mathematical Modeling of Novokuznetsk Institute (Branch) of FGBOU VO “Kemerovo State University”, 23 Tsiolkovsky street, 654000 Novokuznetsk, Russia
E-mail: sedovaea@yandex.ru

Abstract. The paper considers a mathematical model of the temperature distribution over the heating pipe to estimate the actual parameters of the heating system based on the results of temperature measurements. An example of solving the problem of actual flow rate identifying of the coolant in the risers, the temperature at the entrance to each riser and the shunt coefficients of the radiators of the heating system of the building is given. Identification results are used to diagnose the condition of the heating system.

1. Introduction
The operation of heated buildings requires maintaining the operational status of heating systems. This is due to the cost of preventive repairs. Specific features of the heating system are seasonality and inadmissibility of prolonged shutdowns during the winter period due to the danger of pipeline failure. A significant number of publications are devoted to detailed models of heating systems. In works [1, 2], models of independent central heating systems are described in detail. The new hydraulic scheme of the substation is given in [3]. This scheme includes a discharge pipe. A heating system that accumulates heat in underground tanks is described in [4]. In [5], the supply of thermal energy into the room is analyzed in detail.

Therefore, diagnosing the current state and predicting changes in the temperature regime of a building is of great importance. Forecasting is aimed at a rational choice of corrective measures (replacement and flushing of pipes and heating devices) at the end of the heating season. This requires the construction of a model that allows determination of the temperature at each site of the heating tower.

2. Heat balance model of radiator heating
Consider the convective heat transfer equation:

\[ q = h(T_0 - T_1), \]  

(1)

\( q \) is heat flux density, W/m², \( T_0, T_1 \) are surface and ambient temperatures (air temperature in the room and radiator or air temperature in the room and the inner surface of the fence or temperature of the outside air and the outer surface of the building), and \( q \) is considered positive if \( T_0 \) is greater than \( T_1 \); \( h \) is heat transfer coefficient.

Equation (1) should be written for a point on the surface, since the temperature on the surface is usually variable. So, the temperature of the radiator is maximum in the area of coolant supply; during
normal operation, this temperature is minimal in the coolant outlet zone (at the outlet flange), and in
the event of radiator clogging, the temperature may be minimal in clogged sections. Therefore, to
determine the power delivered by the radiator, it is necessary to integrate (1) over the area of the
radiator. Considering the air temperature $T_i$ in the room constant throughout its volume (or
everywhere near the radiator), we obtain the power $N$, W:

$$N = h \int_{S_k} (T_0 - T) dS = h \cdot S_k (\overline{T}_0 - T),$$  \hspace{1cm} (2)

$S_k$ is surface area of the radiator (or convector),
$
\overline{T}_0$ is average integral temperature of the radiator:

$$\overline{T}_0 = \frac{1}{S_k} \int_{S_k} T_0 dS .$$  \hspace{1cm} (3)

Thus, the characteristic of the radiator is the average integral temperature. This characteristic is
determined by the heat balance equations the output thermal power.

The processes of heat and mass transfer in the heating system are determined by the following
factors:
1) coolant flow rate at the entrance to the heating system,
2) coolant temperature at the entrance to the heating system,
3) coolant flow through the Elevator (return of the coolant from the outlet pipe to the inlet),
4) distribution of coolant flow rates over the heating towers,
5) heat losses during coolant flow from the heat source to the risers,
6) shunting radiators.

In addition, the following factors affect heat transfer coefficients from radiators to the room, area of
the radiators and distribution of their temperature over the area.

Secondary (dependent) factors are the temperature of the coolant at the entrance to each radiator
and the temperature of the coolant at the outlet of the heating system.

In heating systems with an open circuit, part of the coolant is consumed for hot water supply. From
this it follows that the flow rate of the diverted coolant in the outlet pipe is less than the flow rate of
the coolant supplied in the inlet pipe. In this case, the calculation formula for determining the
consumed thermal energy is:

$$Q_0 = M_1 (h_1 - h_2) + M_2 (h_2 - h_x), \hspace{1cm} Q_h = M_h (h_2 - h_x),$$  \hspace{1cm} (4)

$Q_0$ is thermal energy consumption in the heating system,
$Q_h$ is heat energy consumption for hot water,
$M_1$ is mass flow rate at the inlet,
$M_2$ is mass flow rate at the outlet,
$h_1, h_2$ are the specific enthalpy of the coolant at the inlet and at the outlet, respectively (the product of
the specific heat capacity and the temperature),
$h_x$ is specific enthalpy of the coolant at the sewage outlet,
$M_h$ is mass flow rate of the coolant through the hot water system:

$$M_h = M_1 - M_2 .$$  \hspace{1cm} (5)

The value of $Q_0$ includes the flow rate $M_h$ twice: once the total flow at the entrance to the system,
and the second time, as a separate term.

The first term (4) expresses the balance of consumed thermal energy and changes in the enthalpy of
the coolant. A similar balance equation can be written for each radiator of the heating system
separately:

$$N = c \rho Q (T_{in} - T_{out}),$$  \hspace{1cm} (6)

c is specific heat of the coolant,
$\rho$ is heat carrier density,
$Q$ is volumetric coolant flow through the radiator,
$T_{in}$ and $T_{out}$ are coolant temperature at the inlet to the radiator and at the outlet of the radiator.
Equation (6) is more convenient to rewrite. Let us replace the product of the heat carrier density $\rho$ and the volume flow rate $Q$ by the mass flow rate $\mu$ equal to this product:
\[ N = c\mu(T_{in} - T_{out}). \]  

Mass flow rate is constant; it does not depend on the temperature change along the length of the riser. Adding equation (2), we obtain the equations of heat balance. This equation relates to the average integral temperature of the radiator, the temperature in the room, as well as the temperature of the coolant at the inlet to the radiator and at the radiator outlet. These equations can be written for all radiators of one riser. In the absence of radiator shunting, the equations are closed by the initial condition. The initial conditions are the temperature of the coolant at the entrance to the riser.

Some radiators of the simulation object are shunted, as shown in Figure 1.

![Figure 1. Scheme of radiator shunt.](image)

The coolant is fed from above, its flow rate is equal to $V$. The coolant flow forks at point 1: part of the coolant enters the jumper (flow rate through jumper $V_1$), part - into the radiator (flow rate through the inlet $V_2$). The total coolant flow rate at the entrance to the "radiator-shunt" system is equal to the flow rate in the riser $\mu$; it is equal to the sum of expenses through jumper $\mu_1$ and through radiator $\mu_2$. At point 2, the coolant flows merge, and at point 3, the flow rate is, and the speed (without taking into account changes in density) is equal to $V$. The flow of coolant in the shunt is determined by the Bernoulli equation:
\[ z_1 + \frac{p_1}{\rho g} + a_1 \frac{V_1^2}{2g} = z_2 + \frac{p_2}{\rho g} + a_1 \frac{V_2^2}{2g} + h_1, \]

$z_1, z_2$ is height between 1 and 2,
$p_1, p_2$ are pressures at points 1 and 2,
$a_1$ is coefficient for section shape (constant along the length of the shunt),
g is acceleration of gravity,
h_1 is hydraulic losses in the shunt.
For the flow in the radiator:
\[ z_1 + \frac{p_1}{\rho g} + a_2 \frac{V_2^2}{2g} = z_2 + \frac{p_2}{\rho g} + a_2 \frac{V_2^2}{2g} + h_2, \quad (9) \]

\( h_2 \) indicates hydraulic losses in the radiator, 
\( a_2 \) is coefficient for radiator section shape.

Temporarily neglecting the change in the density of the coolant, we find the ratio between the volume flow rate of the coolant in the radiator and shunt. Considering that volumetric flow rates are related to the rate of proportional dependence, the ratio of the flow through the radiator to the flow through the shunt is found as the ratio of the speeds:

\[ \frac{Q_2}{Q_1} = \frac{S_2}{S_1} \sqrt{\frac{h_2}{h_1}}. \quad (10) \]

Here \( S_1 \) is shunt cross-sectional area, \( S_2 \) is cross-sectional area at the entrance to the radiator.

Let us consider a change in heat content (enthalpy) of the coolant. As it moves through the radiator, the coolant cools down; in the shunt, the temperature decreases also, but to a lesser extent. Let at point 1 the temperature of the coolant is \( T_1 \), at the exit of the jumper (at point 2 of the jumper) the temperature is \( T_{21} \), and at the exit of the radiator (at point 2 of the outlet pipe) \( T_{22} \). At point 2 immediately after the convergence of the pipes, the coolant is mixed. According to the rule of mixtures, we define the specific enthalpy \( E \):

\[ E = c\rho T = \frac{Q_1 E_1 + Q_2 E_2}{Q}, \quad (11) \]

\( c \) – specific heat. 

From here we get the temperature of the coolant after the passage of a shunting radiator \( T_2 \):

\[ T_2 = \frac{Q_1 T_{21} + Q_2 T_{22}}{Q}. \quad (12) \]

Returning to the mass flow, we have:

\[ T_2 = \frac{\mu_1 T_{21} + \mu_2 T_{22}}{\mu}. \quad (13) \]

Here, the coolant flow rates through the shunt and the convecter, or rather, the share of the total flow rate \( \mu \), which is also unknown, are unknown. Taking into account that \( \mu = \mu_1 + \mu_2 \) going to the relative values of the flow (in fractions of \( \mu \)), in the last equation we can leave only one independent unknown:

\[ T_2 = \frac{\mu_1}{\mu} T_{21} + \left(1 - \frac{\mu_1}{\mu}\right) T_{22}. \quad (14) \]

Deciding on \( \frac{\mu_1}{\mu} \), will get:

\[ \frac{T_2 - T_{22}}{T_{21} - T_{22}} = \frac{\mu_1}{\mu}. \quad (15) \]

The result obtained allows us to estimate the proportion of volumetric flow through the shunt according to the measurement of temperatures.

Let us analyze the change in the proportion of flow through the jumper, using hydraulic calculation. According to formula (10), the flow rate ratio through the convecter and through the shunt is proportional to the square root of the hydraulic loss ratio:

\[ \frac{\mu_2}{\mu_1} = \frac{S_1}{S_2} \sqrt{\frac{h_1}{h_2}}. \quad (16) \]
Hydraulic losses in the shunt will be presented in the form:

\[ h_1 = h_i + h_u = \left[ \lambda_t \frac{L}{d} + \zeta \right] \frac{V_1^2}{2g}, \quad (17) \]

$L, d$ are the length and internal diameter of the shunt, $\lambda_t$ is coefficient of hydraulic friction against the walls, $\zeta$ is coefficient of local hydraulic losses at the entrance to the shunt and at the exit from the shunt.

The coefficient $\lambda_t$ depends on the flow velocity $V_1$. For a laminar flow, it is inversely proportional to the Reynolds number (and hence the velocity), for a turbulent flow it is equal to the fourth-degree root of the Reynolds number. Local resistances (input, output and aperture) are proportional to the velocity head with a constant coefficient.

The hydraulic losses in the convector are calculated similarly to local resistances (the coefficient of hydraulic losses at a flow rate of 0.1 kg/s is 1.5-1.8).

Thus, the less the coefficient of hydraulic friction changes and the shorter the length, the lesser the change in ratio $h_1/h_2$ with a change in the velocity of the coolant flow. In the first approximation, we can take this relationship as constant. Then the ratio of flow rates through the radiator and jumper can be also taken constant.

The temperatures at the inlet and at the outlet of the shunted radiator (radiator-shunt system) and the power delivered by the radiator to the room are linked by a balance ratio similar to (2). To avoid disagreement in the notation, the expression (2) is written in the form:

\[ N = h \cdot S_0 \left( \bar{T} - T_w \right), \quad (18) \]

where $T_w$ is indoor air temperature, $\bar{T}$ is average integral temperature of the radiator.

The average integral temperature is determined, firstly, by the temperatures of the coolant at the inlet and outlet of the radiator $T_1, T_22$, respectively, and secondly, by the velocity distribution of the coolant over the radiator. In the ideal case, assuming that each particle of the coolant in motion along the convector is cooled to the same temperature, and the rate of change in its temperature is constant, the average integral temperature will be equal to the arithmetic average of the inlet and outlet temperatures. In fact, the average integral temperature is lower because the temperature is not evenly distributed over the sections.

We approximate the dependence of the average integral temperature on the inlet and outlet temperatures by a quadratic function:

\[ \bar{T} = \frac{1}{2} (T_1 + T_{22}) - \frac{\chi}{2} (T_1 - T_{22}) = \frac{1 - \chi}{2} T_1 + \frac{1 + \chi}{2} T_{22}, \quad (19) \]

where the coefficient $\chi$ is a radiator parameter that takes into account the uneven distribution of temperature over its area (coefficient of temperature unevenness).

Further, considering the temperature drop on the jumper is quite small compared to the temperature drop on the radiator, we will take $T_{21} = T_1$, and express the temperature at the radiator outlet through the temperature at the exit from the $T_2$ "radiator-shunt" system. For simplicity, we denote: $\bar{\mu}_2 = \frac{\bar{H}_2}{\mu}$ is the share of mass flow of the coolant through the radiator. Let us express the temperature at the exit of the radiator $T_{22}$ through the temperature at the exit of the jumper $T_2$:

\[ T_{22} = \frac{T_2 - T_1 (1 - \bar{\mu}_2)}{\mu_2}, \quad (20) \]

Substituting this expression in (19), we obtain:

\[ \bar{T} = \frac{1}{2 \mu_2} \left[ 2 \bar{\mu}_2 - (1 - \chi) \right] T_1 + \left( 1 + \chi \right) T_2, \quad (21) \]
We introduce the notation:

\[ \alpha = \frac{2\mu_2 - 1 - \chi}{2\mu_2}, \quad \beta = \frac{1 + \chi}{2\mu_2}. \]  

(22)

Then the expression for the average integral temperature takes the form:

\[ \bar{T} = \alpha T_1 + \beta T_2. \]  

(23)

Note that the amount \( \alpha + \beta = 1 \).

The power delivered by the shunted radiator is expressed in terms of the inlet and outlet temperatures as follows:

\[ N = hS(\alpha T_1 + \beta T_2 - T_w). \]  

(24)

On the other hand, this power is expressed through the increment of the enthalpy of the coolant at the inlet and outlet:

\[ N = \mu(T_1 - T_2). \]  

(25)

Thus, for the system “radiator shunt” we get the balance ratio:

\[ c\mu(T_1 - T_2) = hS(\alpha T_1 + \beta T_2 - T_w). \]  

(26)

This ratio is given to the equation relating to the temperature of the coolant at the inlet and outlet to the “radiator-shunt” system with the room temperature and the mass flow of the coolant in the heating riser:

\[ (c\mu + ahS)T_1 + (c\mu + b\mu hS)T_2 = hS T_w. \]  

(27)

Thus, the equations obtained make it possible to describe stationary thermal fields in the heating system and in a heated building. For their practical use, it is necessary, first, to specify the form, dimensions and actual parameters of the object, and second, to bring the written equations into a form that can be solved on a computer.

3. Heat balance model of the heating system

The heat balance of the heating system is determined by heat and mass transfer processes in which the temperatures in the rooms do not vary, but are assumed to be known (found from the model of the heat balance of the building envelope). The processes of heat and mass transfer occur in the following material objects of the heating system:

- inlet main pipe through which the coolant of a given temperature flows with a known mass flow rate and with a known specific heat capacity;
- elevator, in which the hot coolant is mixed with the colder one, and the flow rate of the cold coolant is unknown or known with great uncertainty;
- pipes of the supply circuit, through which the coolant from the elevator enters the entrance of each heating pipe;
- heating towers, with a “return” pipes, which run vertically along the floors of the building, and the heating radiators included there, either bypassed or bypassless.

The cooled coolant that has passed through all the radiators of the riser enters the outlet pipe of the heat source.

The controlled parameters of the coolant are the inlet temperature (before mixing in the elevator), the outlet temperature, inlet and outlet pressure and flow rate in the inlet and outlet pipes. These parameters are measured by heat meters and are available in tables of hourly and daily heat consumption.

The temperature of the coolant supplied from the main varies over a wide range in accordance with the temperature schedule when the outside air temperature changes. Unlike temperature, inlet and outlet pressures, as well as hourly flow, are fairly stable values. Therefore, the flow rate of the cooled coolant through the elevator can also be considered constant. This allows us to give a reasonable estimate of the temperature decrease at the elevator according to measurements. A similar estimate can be made for the magnitude of the decrease in the temperature of the coolant during its flow from the elevator to the entrance of each riser. Thus, the temperature of the coolant at the entrance to the riser...
can be assumed to be known with a certain degree of uncertainty; this forces us to accept it as a model parameter.

So, we take temperatures in the rooms as the input variables of the heat and mass transfer model. The structural parameters of the model are: the coolant temperature at the entrance to each of the risers, the coolant flow rate in each riser, the area and heat transfer coefficient of each radiator, as well as the irregularity coefficients of temperature distribution over the area of each radiator and shunting factors included in equation (27).

Let us consider a separate heating riser, including $n$ shunted radiators connected in series with a common pipe. Coolant flow rate and its inlet temperature are known. Then for each convector with the number $i$, $1 \leq i \leq n$, we write equation (27) and get a system of difference equations:

$$(-c\mu + \alpha_i h_i S_i) T_{i-1} + (c\mu + \beta_i h_i S_i) T_i = h_i S_i T_{\infty}, \ 1 \leq i \leq n.$$  \ (28)

Let us supplement this system with the equation determining the temperature of the coolant at the entrance to the riser:

$$T_0 = T_{in}.$$  \ (29)

So, we get a system of linear algebraic equations. Its coefficient matrix is two-diagonal:

$$A = \begin{bmatrix}
1 & 0 & \cdots & 0 \\
-c\mu + \alpha_1 h_1 S_1 & c\mu + \beta_1 h_1 S_1 \\
& \cdots & \cdots & \cdots \\
& & & -c\mu + \alpha_n h_n S_n & c\mu + \beta_n h_n S_n \\
\end{bmatrix},$$  \ (30)

and the right parts:

$$B = \begin{bmatrix}
T_{in} \\
h_1 S_1 T_{\infty} \\
\cdots \\
h_n S_n T_{\infty}
\end{bmatrix}.$$  \ (31)

Solving this system, at known indoor temperatures, we obtain the distribution of coolant temperature along the riser.

Equation (27) is applicable to a shunted radiator, and for a radiator without a jumper; in the latter case, the coefficient $\mu_2$ in formula (22) is taken equal to one. We consider separately the case of parallel connection of several radiators to one common riser (Fig. 2).

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{parallel_radiators}
\caption{A pair of parallel radiators without lintels.}
\end{figure}
The total mass flow rate of the coolant through the riser is $\mu$, the flow through the first (left in the figure) radiator is 1, the flow through the second radiator is $\mu_2$. The temperature of the coolant at the entrance to each radiator is $T_1$. Let the temperature of the coolant at the outlet of the first radiator be $T_{12}$, at the outlet of the second it is $T_{22}$. Then the temperature at the total output after mixing will be equal to

$$T_2 = \frac{\mu T_{12} + \mu_2 T_{22}}{\mu} = \frac{\mu_1 T_{12} + \mu_2 T_{22}}{\mu},$$  \hspace{1cm} (32)

or after transformation

$$T_2 = (1-\mu_2)\overline{T}_{12} + \mu_2 T_{22} = T_{12} + \mu_2(T_{22} - T_{12}).$$  \hspace{1cm} (33)

This shows that the proportion of flow through each radiator can be found from temperature measurements only if the temperatures $T_{12}$ and $T_{22}$ are different. In this case, we have:

$$\mu_2 = \frac{T_{22} - T_{12}}{T_{22} - T_{12}}, \quad \mu_1 = 1 - \mu_2 = \frac{T_{22} - T_{2}}{T_{22} - T_{12}}.$$  \hspace{1cm} (34)

Conversely, with a known flow rate, we can find the temperature at the outlet of each radiator, expressing it through the inlet and outlet temperatures.

The coefficient of temperature non-uniformity for the first radiator is denoted by $\chi_1$, for the second radiator it is $\chi_2$. Then the average integral temperatures of the first and second radiators $T_1^*$ and $T_2^*$, accordingly, will be determined as follows:

$$T_1^* = \frac{1}{2}(T_1 + T_{12}) - \frac{X_1}{2}(T_1 - T_{12}) = \frac{1-\chi_1}{2} T_1 + \frac{1+\chi_1}{2} T_{12},$$  \hspace{1cm} (35)

$$T_2^* = \frac{1}{2}(T_1 + T_{22}) - \frac{X_2}{2}(T_1 - T_{22}) = \frac{1-\chi_2}{2} T_1 + \frac{1+\chi_2}{2} T_{22}. $$  \hspace{1cm} (36)

The integral coefficients of the total heat transfer convectors are denoted by $H_1$ and $H_2$, respectively. (These coefficients, as before, are equal to the products of the area by the specific heat transfer coefficient: $H = hS$). Then the power, which is removed from both convectors into the room, is equal to the sum of the powers delivered by each convector:

$$N = h_1S_1T_1^* + h_2S_2T_2^* - (h_1S_1 + h_2S_2)T_0.$$  \hspace{1cm} (37)

Substituting the expressions for average temperatures into the last equality, we get:

$$N = \left( H_1 \frac{1-\chi_1}{2} + H_2 \frac{1-\chi_2}{2} \right) T_1 + H_1 \frac{1+\chi_1}{2} T_{12} + H_2 \frac{1+\chi_2}{2} T_{22} - (H_1 + H_2)T_0.$$  \hspace{1cm} (38)

hence it is seen that the constant term is equal to the product of the total heat transfer coefficient and the room temperature.

When simulating heat and mass transfer, a pair of parallel radiators can be formally combined into one radiator with a total heat transfer coefficient and temperature averaged over the same area. This allows us to save the structure of the difference equation (28) when calculating the temperature distribution along the riser.

Thus, the system of heat balance equations can be represented as association of systems of the form (28) for each riser separately: the general system of equations splits into systems of heat balance equations for individual risers with coefficient matrices (29) and right-hand sides (30). The solution of this system will allow us to determine the temperature of each point of the heating riser.

The constructed model can be used to identify the actual heat transfer coefficients of radiators and the actual coolant flow rate in the riser according to full-scale temperature measurements in the elements of the heating system.

We especially note that the actual costs in risers are determined by measuring the actual temperatures. This allows to determine the inlet temperature and flow rate in each of the risers. It does not require knowledge of the thickness of deposits in the pipes. It does not require knowledge of the sediment thickness in the pipes.
4. Model identification and heating system diagnostics

When identifying indoor temperatures, they were set equal to their measured actual values. The heat transfer coefficients of radiators (per unit surface area) were taken according to the manufacturer.

Figure 3 shows a radiator jumper (shunt) thermogram.

The difference between the pipe outer surface temperature and the coolant temperature was investigated separately. Due to the fact that deposits are distributed unevenly on the inner surface and have the appearance of growths, most of the inner surface of the pipe is covered with a thin layer of deposits. The thickness is on the order of millimeter tenths. Temperature measurements were taken on pipe sections cut from the riser after 20 years of operation. This did not allow us to find a significant difference in the temperature of the outer surface in local zones of sediments and neighboring areas. Evaluation of the temperature difference between the inner and outer surfaces of the pipe gives a value of less than 0.5°C, which is not fixed by thermal imaging equipment. Therefore, the shunt coefficient is determined by the formula (33), in which data from the thermogram are received as estimates of the temperatures in each section.

Similarly, the flow rate of the coolant through the elevator of the heat unit was determined. Figure 4 shows the distribution of the average integral temperature over the radiators of one riser depending on the floor number: the solid line is the calculation at the standard flow rate and the standard temperature of the coolant at the inlet to the riser, the dotted line is the experimental data. The average integral temperature of each radiator was determined by processing the thermogram. Figure 4a shows the calculation results at the standard parameters of the coolant, and Figures 4b and 4c show at the parameters determined by the model identification (after setting). The calculated total flow rate of the coolant in all risers differs from the readings of the heat meter by no more than 2%.
Figure 4. Average temperatures of radiators for one riser: solid line – calculation, dotted line – experiment; a – calculation at the regulatory parameters of the coolant, b, c – calculation after setting up the model.

Figure 5 shows the temperatures at the entrance to a part of the building heating system risers (with numbers 23-42) determined as a result of identification.

The graph shows that the temperature at the inlet to riser 37 is significantly lower than in other risers. This may be a sign of a violation of the insulation of the supply pipe, which is located in an unheated attic.

The data obtained were used in planning the off-season repair of the building heating system.

Conclusions
The heating system balanced model parameters identification by the measured actual temperatures of the radiators allows to determine the actual flow rate of the coolant in each riser and its temperature at the entrance to the riser.
The difference between the total flow rate found and the heat meter readings does not exceed 2%. It does not require a separate deposits thickness assessment.

The adjusted balanced model of the building’s heating system allows predicting the change in the thermal regime in the premises after the scheduled repair work (replacing heating devices, flushing the risers) and determining rational options for the current repair during the off-season period with limited resources of the operating organization.

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