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To cite this article: A S Seminenko *et al* 2018 *IOP Conf. Ser.: Mater. Sci. Eng.* **327** 042097

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# ANALYSIS OF THERMAL DESIGN OF HEATING UNITS WITH METEOROLOGICAL CLIMATE PECULIARITIES

A S Seminenko, Y V Elistratova, M I Pererva, M V Moiseev

Belgorod State Technological University named after V.G. Shoukhov, 46, Kostyukov St., Belgorod, 308012, Russia

E-mail: tgv.bel@gmail.com

**Abstract.** This article is devoted to the analysis of thermal design of heating units, one of the compulsory calculations of heating systems, which ensures their stable and efficient operation. The article analyses the option of a single-pipe heating system with shifted end-capping areas and the overhead supply main; the difference is shown in the calculation results between heat balance equation of the heating unit and calculation of the actual heat flux (heat transfer coefficient) taking into account deviation from the standardized (technical passport) operating conditions. The calculation of the thermal conditions of residential premises is given, the deviation of the internal air temperature is shown taking into account the discrepancy between the calculation results for thermal energy.

## 1. Introduction

The problem of energy saving, in particular, of heat supply systems has recently been reflected in the goals and objectives of a number of government bills and decisions. Thus, Federal Law No. 261-FZ "On Energy Saving, on Improving Energy Efficiency and on Amending Certain Legislative Acts of the Russian Federation" predicts a significant reduction in energy consumption by heating and ventilating systems in residential buildings [1, 3-5, 9] based on the established objectives of this document.

To date, residential buildings in most cases are equipped with vertical single-pipe heating systems. One of the measures to increase energy efficiency of such systems is to install thermostatic valves on the heating units, which automatically maintain constant temperature in the premise at a given level. At the same time, up to 20% of heat savings are achieved primarily due to the reduction of unproductive heat costs (overflow, etc.) [6, 2]. However, individual regulation with the use of thermostatic valves has certain drawbacks which can affect the operation of the entire heating system. The article [9] considers some restrictions and drawbacks of the operation of single-pipe systems equipped with thermostats.

However, selection of proper thermostatic valves requires data about dependence of the heat transfer of the heating units on external and internal factors and the ability of considering them.

## 2. Objective of the study.

Analysis of thermal design of heating units of a single-pipe heating system for residential premises.

The objective of the study defines its tasks:

- to conduct analytical research of the factors influencing heat transfer of heating units;
- to carry out theoretical studies of the temperature head influence on heat transfer of radiators;



- to determine values of the internal air temperatures in the premises under consideration via computer numerical simulation [11-12].

### 3. Materials and methods

The research methods included analytical generalization of known scientific and technical results, collection of actual data on the characteristics of a residential building and its functioning heating system, computer numerical modeling of the change in the temperature parameter of internal air. [11-12]

Observance of the required heat transfer of the heating unit for each premise with known heat loss is provided by selecting the radiator area taking into account the average temperature head according to formula [2], W:

$$Q = K_{unit} \cdot F \cdot \Delta t_{av} \quad (1)$$

where  $K_{unit}$  - heat transfer coefficient of the heating unit,  $W / m^0 C$ ;  $F$  - area of the heating device  $m^2$ ;  $\Delta t_{av}$  - temperature head,  $^0 C$ .

### 4. Body of the study

Let us perform thermal design of the units taking into account the regulation of the heat flow for the riser passing through the living rooms of single-type flats in a 14-storey residential building with the upper heating agent supply.

Thermal design has the following parameters:  $t_{air}$  - temperature of the internal air according to GOST 12.005-88 and the design standards for the corresponding buildings and structures,  $t_{int} = 20^0 C$ ;  $t_{ext}$  is the calculated winter temperature of the external air, in  $^0 C$  equal to the average temperature of the coldest five-day period of 0.92 [2]  $t_{ext} = -23^0 C$ ; the area of the outer wall -  $F_{ew} = 19.8 m^2$ , the coefficient of heat transfer of the external wall -  $K_{ew} = 0.28 W / (m^2 \cdot ^0 C)$ ; the area of the window opening -  $F_{wo} = 4.5 m^2$ , the coefficient of heat transfer of the window opening -  $K_{wo} = 1.62 W / (m^2 \cdot ^0 C)$ ;  $F_{floor} = 23.4 m^2$ .

According to SNiP 41-01-2003 "Heating, ventilation and air conditioning" [16], heating is designed to ensure even heating and normalized indoor air temperature taking into account:

1. losses of heat through enclosing structures;
2. heat consumption for infiltrating air heating;
3. consumer heat input  $Q_c$  supplied regularly from electrical appliances, lighting, pipelines, people and other sources (with at least 10 W per  $1 m^2$  of floor).

Calculation of heat consumption for heating air being infiltrated is applied in two cases: with natural exhaust ventilation not compensated by the influx of heated air  $Q_{ia}$ , W and with the influence of thermal and wind pressures  $Q_{itw}$ , W [2]. At the same time, for determination of  $Q_{ia}$  flow rate of the exhaust air,  $L_h$ ,  $m^3 / h$ , for residential buildings is assumed to be equal to the normative consumption of  $3 m^3 / h$  per  $1 m^2$  of residential premises, and for  $Q_{itw}$  heat air consumption  $G_h$ ,  $kg / h$  is equal to the amount of infiltrating air coming through the leakage of external enclosing structures  $\Sigma G_h$ ,  $kg / h$  [2, p.97].

On the basis of the obtained values of  $Q_{ia}$  and  $Q_{itw}$ , the larger of these values is taken as the calculated indicator of the heat consumption for the infiltrated air heating. In this regard, additional heat losses for infiltration are calculated and in some cases are not equal to the actual needs for air heating.

The flow rate of the heating agent in the riser -  $G_{riser}$ ,  $kg / h$  is determined by the formula:

$$G_{riser} = \frac{3,6 \cdot \Sigma Q_{calc}}{c \cdot (t_{in} - t_{out})}, \quad (2)$$

where  $\Sigma Q_{calc}$ , W - heat loss of premises (serviced by the riser in question), taking into account the fact that  $Q_i = Q_{ia}$ , W;  $c$  is the specific heat of water,  $kJ / (kg \cdot K)$ ;  $(t_{in} - t_{out})$  - temperature of heating agent at the inlet and outlet of the device,  $95, 70^0 C$ , respectively.

Heat losses through external enclosures, W:

$$Q_{ee} = \Sigma(K_{ee} \cdot F_{ee}) \cdot (t_{int} - t_{ext}), \quad (3)$$

$F_{ee}$  - the area of external enclosures (area lights, external walls, floor or ceiling of the basement or attic floor), respectively,  $m^2$ ;  $K_{ee}$  - coefficient of heat transfer of fences (light apertures, external walls, floor or ceiling of the basement or attic floor),  $W / (m^2 \cdot ^\circ C)$ .

According to [2, p.96, f (9.12)] heat consumption for heating, the infiltrated air for each premise from the 1st to 14th floors is  $Q_i = 1018.8$  W. In this case, calculated premise heat loss  $Q_{calc}^{2-13}$  from the 2nd to 13th floor and  $Q_{calc}^{1, 14}$  for floors 1 and 14 with the main heat loss through external enclosures  $Q_{ee}$  and  $Q_c$  are equal to, W:

$$Q_{calc}^{2-13} = (Q_{ee} + Q_i - Q_c) \cdot 12 = (785,8 + 1018,8 - 234) \cdot 12 = 1571 \cdot 12 = 18852, \quad (4)$$

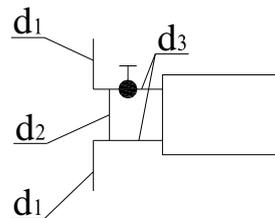
$$Q_{calc}^{1,14} = (Q_{ee} + Q_i - Q_c) \cdot 2 = (995,8 + 1018,8 - 234) \cdot 2 = 2 \cdot 1781 = 3561,2, \quad (5)$$

Then the overall heat loss of the single-type premises from 1<sup>st</sup> to 14<sup>th</sup> floor is  $\Sigma Q_{calc} = 22413$  W. According to formula (2), the heating agent flow in the riser is  $G_{riser} = 770$  kg / h.

The water flow through unit  $G_{unit}$ , kg / h depends on the coefficient of the heating agent flow into unit  $\alpha$  and  $G_{riser}$ , kg / h:

$$G_{unit} = \alpha \cdot G_{riser}, \quad (6)$$

According with the table 51 [10] for the diameter of the riser pipe,  $d_{riser} = 25$  mm and  $G_{riser} = 770$  kg / h of water speed  $v_w = 0.385$  m / s, then according with the table 38 [10],  $\alpha = 0,312$  in accordance with conditional diameters  $d_1 = 25$ ,  $d_2 = 20$ ,  $d_3 = 20$  mm (Fig. 1).



**Figure 1.** Heating unit bundling scheme

Let us define  $G_{unit}$ , kg / h taking into account the values of  $\alpha$  and  $G_{riser}$ , kg / h:

$$G_{unit} = \alpha \cdot G_{riser} = 0,312 \cdot 770,2 = 240,3 \quad (7)$$

Heat losses of premises through enclosing structures are calculated according to formula (3) if  $t_{ext} = -23$   $^\circ C$ . Let us calculate heat loss through the external enclosures fence with  $t_{ext} = -20$ ;  $t_{ext} = -10$ ;  $t_{ext} = -5$ ;  $t_{ext} = 0$ ;  $t_{ext} = +8$   $^\circ C$ . The data obtained are summarized in Table. 1.

**Table 1.** Changing  $Q_{calc}$  of premises according to  $t_{ext}$ ,  $^\circ C$ .

$t_{ext}$ , $^\circ C$	-23	-20	-10	-5	0	+8
$Q_{calc}^{2-13}$ , W	1571	1445	1025	816	606	270
$Q_{calc}^{1 \text{ and } 14}$ , W	1781	1655	1235	1026	816	480

For single-pipe heating systems, the temperature of the heating agent outlet from the unit on the upper 14th floor is calculated by formula [20],  $^\circ C$ :

$$t_{out} = t_{in} - \left( \frac{Q_{calc} \cdot 3,6}{c \cdot G_{unit}} \right) = 88,63, \quad (8)$$

$$t_{out} = \alpha \cdot t_{out} + (1 - \alpha) \cdot t_{in} = 93,01, \quad (9)$$

The temperature of the heating agent inlet to the device on the 13th and upper floors is equal to the temperature of the water outlet from the unit located on the previous floor, respectively.

On the basis of the formulas [1], the number of sections in the radiator providing the required heat transfer is determined by the following relationship:

$$n = \frac{Q_{calc}^{\beta}}{a_{sec} \cdot q_a \cdot \beta_3}, \quad (10)$$

where  $Q_{\text{calc}}^{\text{fl}}$  - calculated heat loss of the floor under consideration equal to the desired heat transfer of the radiator, W;  $a_{\text{sec}}$  - heating surface of one section, for radiators MC-140  $a_{\text{sec}} = 0.244 \text{ m}^2$ ;  $\beta_4$  - correction factor taking into account the method of installing the radiator [7, p.47], according to (Fig. 1); let us take  $\beta_4 = 1$  [7, p. 47];  $\beta_3$  - coefficient depending on the number of sections in the radiator according to the recommendations [8, p. 47];  $q_a$  is the actual density of the heat flow of the heating device for conditions different from the standard ones [1, p.158],  $\text{W} / \text{m}^2$ :

$$q_a = q_{\text{nom}} \cdot \left( \frac{\Delta t_{\text{av}}}{70} \right)^{1+n} \cdot \left( \frac{G_{\text{unit}}}{360} \right)^p, \quad (11)$$

$q_{\text{nom}}$  - surface heat flux density of the heating unit under standard operating conditions,  $\text{W} / \text{m}^2$ , for cast iron radiators MC-140  $q_{\text{nom}} = 758 \text{ W} / \text{m}^2$  [8, app. 9];  $n$  and  $p$  - experimental values of the exponents of the degree are assumed to be equal to  $n = 0.3$  and  $p = 0$  [7, Table 9.2];  $\Delta t_{\text{av}}$  - average temperature head in the heating unit on the 14th floor,  $^{\circ}\text{C}$  equal to:

$$\Delta t_{\text{av}} = \frac{t_{\text{in}} + t_{\text{out}}}{2} - t_{\text{int}} = \frac{95 + 93,01}{2} - 20 = 71,32', \quad (12)$$

where  $\Delta t_{\text{in}}$  and  $\Delta t_{\text{out}}$  - the temperature of the heating agent,  $^{\circ}\text{C}$ , at the inlet and outlet of the heating unit located on the corresponding floor,  $^{\circ}\text{C}$ .

Consequently,  $q_a$  of the radiator on the 14th floor, W, is:

$$q_a = 758 \cdot \left( \frac{71,32}{70} \right)^{1+0,3} \cdot \left( \frac{240,3}{360} \right)^0 = 783,7, \quad (13)$$

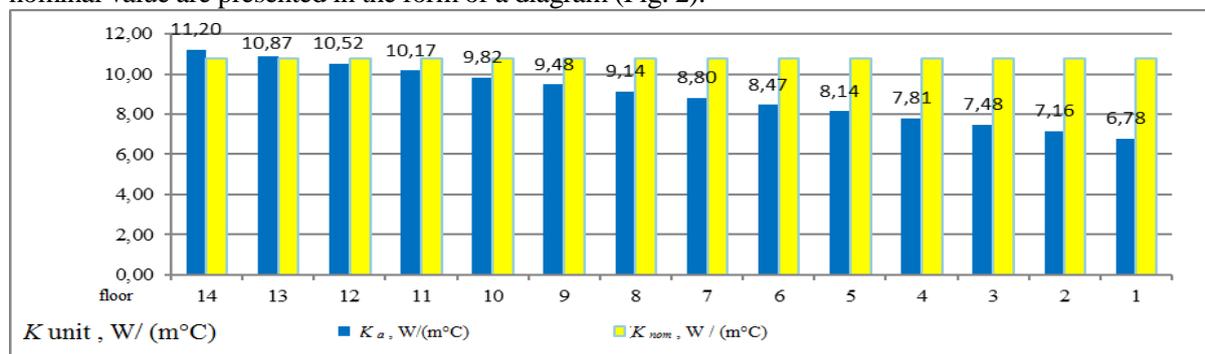
In this case, the number of the heating unit sections on the 14th floor is taken according to formula (10)  $n = 10$ . For the remaining floors, the calculation of the required number of radiator sections is performed in the same way as for the 14th floor.

Taking into account the formula (11), heat transfer of the heating unit depends on heat transfer coefficient  $K_{tr}$ ,  $\text{W} / (\text{m} \cdot ^{\circ}\text{C})$ . The value of this coefficient is determined by the type of radiator chosen and, as a rule, given in the reference literature. So nominal value  $K_{\text{nom}}$  for cast iron radiator MC-140 [7, Table 9.7] is  $(10.83 \text{ W} / \text{m} \cdot ^{\circ}\text{C})$ .

Following recommendations [8], unit heat transfer coefficient  $K_a$ ,  $\text{W} / (\text{m}^2 \cdot ^{\circ}\text{C})$ , under conditions different from normal ones (when  $\Delta t_{\text{av}} = 70^{\circ}\text{C}$  and  $G_{\text{unit}} = 360 \text{ kg} / \text{h}$ ), is calculated by the formula:

$$K_a^{14} = K_{\text{nom}} \left( \frac{\Delta t_{\text{av}}}{70} \right)^{1+n} \cdot \left( \frac{G_{\text{unit}}}{360} \right)^p = 10,83 \left( \frac{71,32}{70} \right)^{1+0,3} \cdot \left( \frac{240,4}{360} \right)^0 = 11,2, \quad (14)$$

The results of the change in the actual heat transfer coefficient over the floors and comparison with the nominal value are presented in the form of a diagram (Fig. 2).

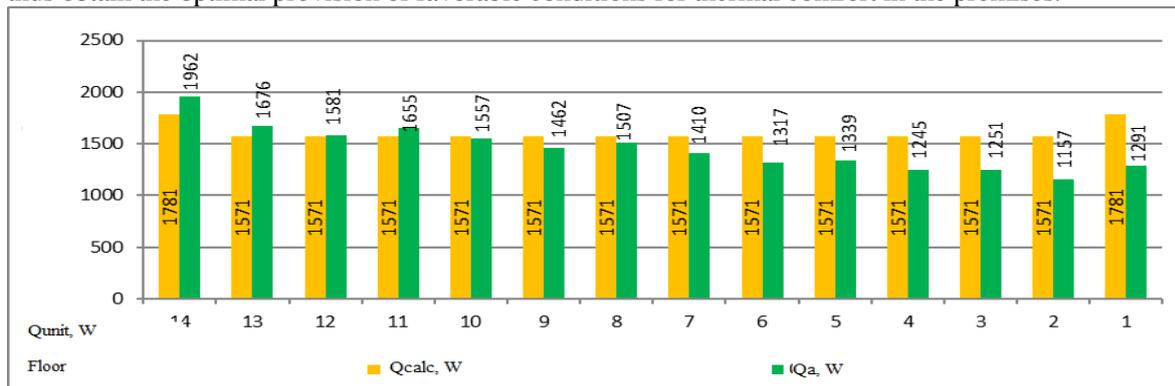


**Figure 2.** A diagram of the change in the actual heat transfer coefficient of the radiators in the riser when the heating agent moves "from top to bottom"

Thus, according to the calculated data (Fig. 2), the coefficient of heat transfer of the unit varies along the path of the heating agent flow and depends on the average temperature head in heating unit  $\Delta t_{\text{av}}$ ,  $^{\circ}\text{C}$  and  $G_{\text{unit}}$ ,  $\text{kg} / \text{h}$ .

Let us calculate the heat transfer from heating units  $Q_a$ , W according to formula (1), but taking into account the change in  $K_a$ ,  $W / (m^2 \cdot ^\circ C)$ , under conditions different from the normal ones, and compare the obtained data with the calculated values of the heat transfer of units  $Q_{calc}$ , W (numerically equal to the amount of heat energy of the heating units due to cooling of water from  $t_{in}$  to  $t_{out}$  with a known water flow). The results are presented in the form of a diagram (Fig. 3).

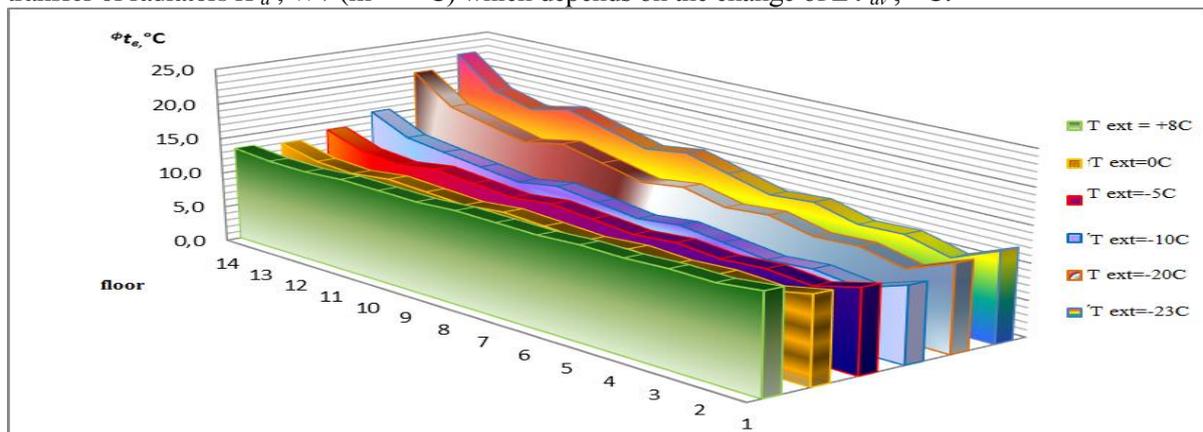
The diagram (Figure 3) shows that the values of  $Q_a$  differ from calculated indicators  $Q_{calc}$  due to changes in the heat transfer coefficient through the riser radiators. Thus, thermal design of the heating system radiators should consider  $K_a$ ,  $W / (m^2 \cdot ^\circ C)$ , depending on changes of parameters  $\Delta t_{av}$ ,  $^\circ C$  and  $G_{unit}$ , kg / h in a unit as it can determine more accurately the required number of sections of the radiator and thus obtain the optimal provision of favorable conditions for thermal comfort in the premises.



**Figure 3.** A diagram of comparison of heat transfer calculated value  $Q_{calc}$  and heat transfer of unit  $Q_a$ , taking into account the change in the heat transfer coefficient of the radiators in the riser when the heating agent moves "from top to bottom"

**5. Results**

According to the given estimated dependences, let us calculate internal air temperature  $t_{int}$  with the real value of heating units area by floors, denote it as the "actual" one, but with the "actual" coefficient of heat transfer of radiators  $K_a$ ,  $W / (m^2 \cdot ^\circ C)$  which depends on the change of  $\Delta t_{av}$ ,  $^\circ C$ .



**Figure 4.** The diagram of the calculated actual temperature of the indoor air  $t_{int}$ ,  $^\circ C$  in the premises from 1 to 14 floor inclusive.

The formula determining the actual internal air temperature  $t_{int}$ , will get from the heat balance equation, wherein the amount of heat transfer of room  $Q_1$  is equal to the amount of heat input  $Q_2$ , W:

$$Q_1 = Q_2, \tag{15}$$

$$Q_1 = Q_{calc} = Q_{ee} + Q_i - Q_c, \tag{16}$$

$$Q_2 = F_{unit} \cdot K_a \cdot \Delta t_{av} = n \cdot a_{unit} \cdot K_a \cdot \Delta t_{av}, \tag{17}$$

$Q_{calc}$ , W depend on  $t_{air}$  of the indoor air temperature, therefore equating  $Q_1 = Q_2$ , one obtains the formula for determining the actual temperature of the internal air, °C:

$${}^a t_{int} = \frac{n f_{unit} t_{ev} K_a + t_{ext} (F_{ee} K_{ee} + F_{cm} K_{cm} + 0,28 \cdot G_i \cdot \rho_{air} \cdot c_{air})}{n f_{unit} K_a + (F_{ee} K_{ee} + F_{cm} K_{cm} + 0,28 \cdot G_i \cdot \rho_{air} \cdot c_{air})}, \quad (18)$$

$\rho_{air}$ ,  $c_{air}$  - density, kg / m<sup>3</sup> and heat capacity, kJ / kg · °C, of internal air, respectively.

By putting the appropriate values into formula (18) to obtain the value for the floor 14, let us obtain value  ${}^a t_{air}^{14} = 22,26$  °C, which is 2.26 °C higher than the calculated value of  $t_{int} = 20$  °C. Let us calculate  ${}^a t_{air}$  for 13-1 floors, similarly to  ${}^a t_{int}^{14}$  °C for  $t_{ext} = -23$  °C. Let us also calculate the actual value of the internal air temperature for premises of 1-14 floors at  $t_{ext} = -20; -10; -5; 0$ ; and  $+8$  °C, the results are presented in the form of a diagram (Fig. 4).

## 6. Conclusions

Taking into account the estimates of actual indoor air temperature  ${}^a t_{int}$ , °C under various conditions, one can observe dependence of this parameter not only on value  $t_{ext}$ , °C, but also on the conditions and the accuracy of thermal design of heating units. Thermophysical properties of the heating agent and heating units also determine the indicator for heating surface area; therefore their deviation from the nominal coefficients used in the standard thermal calculations for the systems leads to the wrong decision to establish the required area of the radiators, which may lead to the actual temperature inside the premises mismatching the accepted standards.

## 7. Acknowledgments.

The article was prepared within the development program of the Flagship Regional University on the basis of Belgorod State Technological University named after V.G. Shoukhov, using equipment of High Technology Center at BSTU named after V.G. Shoukhov.

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