



## Influence of heat resistance of the enclosing structures of a heated room on the choice of the type and location of the heating device

Sargsyan S.V.<sup>1</sup> , Agafonova V.V.\*<sup>1</sup> 

<sup>1</sup> Moscow State University of Civil Engineering (National Research University), Russia

**Abstract.** The article examines the influence of heat resistance of enclosing structures on the choice of type and location of heating devices in the room. The main attention is paid to the relationship between the thermal characteristics of fences and the method of heat transfer of heating devices. The paper considers two main types of heat transfer: convective and radiative. It is shown that at low thermal stability of structures, it is advisable to use devices with a predominant convective heat transfer (convectors, panel radiators), and at high – with radiation (radiators, underfloor heating, infrared heaters). A mathematical model of spatial non-stationary heat transfer in a heated room is developed, taking into account various thermophysical characteristics of enclosing structures and the type of heat release sources. The study of the amplitude of fluctuations in the temperature of air and internal surfaces, depending on the type of heating device, is carried out. The practical significance of the work is the development of scientifically based recommendations for the selection and placement of heating devices, taking into account the heat resistance of enclosing structures. The results of laboratory tests confirm the theoretical conclusions and demonstrate the nature of the temperature distribution in the room under various heat exchange conditions. The results of the study show that the correct choice of the type and location of the heating device allows you to ensure uniform heating of the room, minimize energy consumption for heating and create comfortable microclimate conditions.

**Keywords:** heat stability, enclosing structures, heat transfer, boundary conditions, convective heat transfer, radiant heat transfer, heating devices

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### 1. INTRODUCTION

In modern conditions of energy efficiency, the rational choice of the type and location of heating devices is of particular importance, which directly affects the comfort of the internal microclimate and the efficiency of energy consumption.

\*Corresponding author E-mail: [agafonovavv@mgsu.ru](mailto:agafonovavv@mgsu.ru)

The thermal stability of enclosing structures has a significant impact on the choice of the type and location of heating devices, since it determines the rate of heat transfer inside the room and heat loss through external enclosing structures.

Heat resistance characterizes the ability of the enclosing structure to maintain a constant temperature on the inner surface of the structure and prevent sudden fluctuations in the indoor air temperature. It depends on the thickness and material of the walls, the thermal characteristics of windows, doors, floors and floor coverings of the heated room. High thermal stability ensures a comfortable microclimate, minimizes heat energy costs and reduces the possibility of condensation on the inner surface of enclosing structures. A number of works by domestic and foreign authors are devoted to the study of heat stability of enclosing structures [1-17].

The thermal stability of enclosing structures affects the distribution of indoor temperatures and, accordingly, determines the features of the choice of heating devices and their placement.

Heating devices are classified according to the dominant method of heat transfer: convective and radiative [1-4]. Heating devices, where convective heat transfer dominates, heat the air of the heated volume of the room. This leads to the formation of ascending convective flows of warm air, which are evenly distributed over the entire volume of the room [5-10]. Examples of such devices include heating convectors. For optimal distribution of warm air, it is recommended to place such devices under windows or along external enclosing structures with low resistance to heat transfer, in order to prevent the influence of falling convective flows formed along the cooled surfaces.

Heating devices, where radiation heat transfer dominates, transfer heat directly to surfaces, by means of infrared radiation. Providing the desired resulting temperature at a relatively low ambient temperature inside the room.

If the heat resistance of exterior walls, windows, doors, floor and ceiling coverings is low, it is necessary to choose heating devices with a predominant convection form of heat transfer. Convection quickly distributes heated air throughout the room, compensating for rapid heat losses through the walls. Examples of such appliances include panel radiators, floor-mounted, wall-mounted, овнутрипольные indoor convectors with natural or forced convection.

With high thermal stability of exterior walls, windows, doors, floor and ceiling coverings, heating devices with a radiation-based heat transfer method will be most effective. Radiation heat transfer prevails when used as heating devices for underfloor heating, infrared heaters and water panels. Radiation allows you to maintain a uniform temperature of the internal surfaces, providing the resulting temperature in the comfort zone, when the internal air temperature is low within acceptable limits.

When choosing the type of heating device, it is necessary to take into account the heat transfer resistance of external enclosing structures. In conditions of a decidedly low value of the heat transfer resistance of external enclosing structures of windows, doors, coverings and floor or ceiling coverings, it is recommended to use heating devices with dominant convective heat transfer. This is due to the need for rapid distribution of heat over the area and height of the heated room, providing the required air parameters in the serviced area of the heated room in conditions of increased heat losses.

In the case of high thermal stability of enclosing structures, it is advisable to use heating devices with a predominance of radiation heat transfer. Such devices ensure an even distribution of heat and maintain a comfortable room temperature even at low air temperatures.

When determining the location of heating devices, it is necessary to take into account the thermal inertia of enclosing structures. For rooms with low heat resistance, it is recommended to install devices near cold areas (windows, doors), which helps to compensate for local heat losses. In rooms with high heat resistance, it is possible to evenly distribute heating devices around the perimeter of the room or to place them hidden, including underfloor heating systems.

The location of heating devices also plays an important role in ensuring the comfort and efficient operation of the heating system. Heating devices with a predominance of convective heat transfer are recommended to be placed near external walls and windows to compensate for heat losses through these elements. Heating devices with the predominance of the radiation form of heat transfer, on the contrary, are most effective when placed at a height, directing infrared radiation to the area of the heated room, where it is necessary to provide the required air parameters.

Thus, the analysis of domestic and foreign sources shows that the choice of the type and location of the heating device should be based not only on the amount of heat loss, but also on the thermal stability of the enclosing structures. At low thermal inertia of the external enclosing structures of the heated room, heating devices with a predominance of the convective form of heat transfer (convectors) placed near "cold" zones are preferred; at high – radiation systems with a uniform distribution of heat. However, there are still no unified methods that combine the thermal engineering characteristics of fences, the type of heat transfer and the spatial location of heat sources into a single calculation model, which determines the relevance of this study.

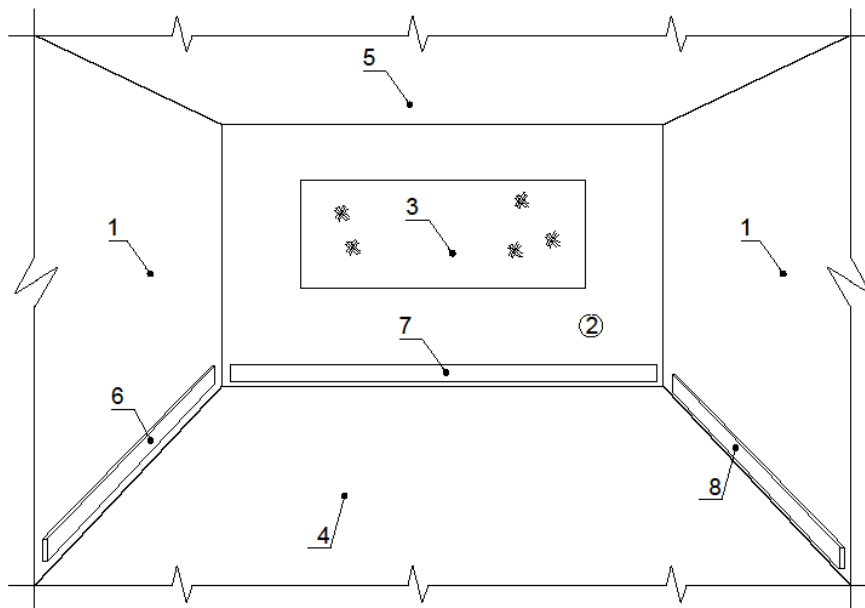
Objective: to develop scientifically based recommendations on the choice of type and rational placement of heating devices in heated rooms, taking into account the heat resistance of enclosing structures.

## 2. METHODS AND MATERIALS

Mathematical modeling of heat exchange and mass transfer processes in heated rooms of buildings for various purposes is an effective tool for optimizing the operation of heating systems. This approach makes it possible to analyze and predict in detail the distribution of air temperature and heat fluxes in heated rooms for various purposes. This approach contributes to creating comfortable conditions in the heated room and improving energy efficiency and energy saving of the heated room as a whole.

A mathematical model of a non-stationary process of spatial heat exchange in a closed volume of a heated room with linear or local heat sources that have inhomogeneous boundary conditions is considered.

Fig. 1 shows the region where heat exchange takes place using the boundary value heat transfer problem.



**Fig. 1.** Heated room with heat sources. 1-wall fences, 2-air of the heated room; 3-window; 4-coating; 5-overlap; 6, 7, 8 - heat sources.

Structural elements of a heated room have certain dimensions with characteristic thermophysical properties. Boundary conditions are established between the internal environment of the heated room and the internal surfaces of external enclosing structures.

Fig. 1 shows a mathematical model of the heat consumption of a heated room. The model takes into account:

- availability of heat sources.

- processes of heat exchange between the internal environment of the room and the internal surfaces of external enclosing structures.

At the same time, the model does not consider heat transfer due to natural or forced convection. The heat transfer process is mathematically described by a system of heat conduction equations that includes nonlinear boundary conditions.

## 2.1 MATHEMATICAL MODEL OF THE PHYSICAL PROCESS OF AIR TEMPERATURE CHANGE IN A HEATED ROOM WITH LOCALLY AND LINEARLY LOCATED HEAT SOURCES

The mathematical model of heat distribution over the height and area of a heated room is described by the heat conduction equation [1].

$$\rho_i \cdot c_i \frac{\partial T_i}{\partial t} = \lambda_i \cdot \left( \frac{\partial^2 T_i}{\partial x^2} + \frac{\partial^2 T_i}{\partial y^2} + \frac{\partial^2 T_i}{\partial z^2} \right), \quad (1)$$

$$i = \overline{1-5}, \quad (2)$$

where  $T=T(t, x, y, z)$ ;  $T_i$  is the temperature at the characteristic point;  $x, y, z$  are the coordinates of the characteristic point;  $c_i$  is the specific heat capacity of the medium at the characteristic point;  $\rho_i$  is the density of the material at the characteristic point;  $\lambda_i$  is the thermal conductivity of the  $i$ -th material of the characteristic point:  $i = 1$ , wall;  $i = 2$ , air environment inside the heated volume;  $i = 3$ , window structures;  $i = 4$ , floor structure;  $i = 5$ , cover structure;  $t$  – time.

Initial condition:

$$T(t, x, y, z)|_{\tau=0} = T_0 = \text{const} \quad (3)$$

Boundary conditions on the boundary between the internal surfaces of external enclosing structures and the air environment of the heated room:

$$-\lambda_i \frac{\partial T_i}{\partial y} = \alpha \cdot (T_{in} - T_i) + \varepsilon \cdot \sigma \cdot (T_{in}^4 - T_i^4), \quad (4)$$

where  $i = 1, 3, 5$  ( $i = 1$  - wall;  $i = 3$ -window;  $i = 5$ -cover).

- for all external borders of the heated room, set the symmetry conditions:

$$-\lambda_i \frac{\partial T_i}{\partial x^k} = 0, \quad (5)$$

$$i = 1, 5, \quad (6)$$

$$k = 1, 3, \quad (7)$$

- on linear heat sources, the following conditions of the 1st kind are set:

$$T = T_{hs} = \text{const}, \quad (8)$$

where:  $\alpha$  is the heat transfer coefficient between the external environment and the area under consideration;  $T_{in}$  is the temperature of the internal environment;  $\varepsilon$  is the reduced degree of blackness of the surface;  $t_i$  is the initial temperature of the object under consideration;  $T_{hs}$  – temperature on the surface of a linear or local heat source;  $\sigma$  is the Stefan-Boltzmann constant.

Linear or local heat sources have a constant temperature. Boundary conditions of the first kind are considered on the surfaces of heat sources. For the numerical solution of this problem, it is advisable to use the finite difference method [2]. When solving the problem, it was taken into account that the latter includes elements with different thermal characteristics.

## 2.2 HEAT ABSORPTION OF EXTERNAL AND INTERNAL ENCLOSING STRUCTURES

The heat flow coming from heating devices equipped with thermal valves with thermal heads is characterized by periodic action.

The air temperature in a heated room changes over time depending on fluctuations in the outside air temperature. These changes can be approximated by harmonic functions, such as sinusoidal ones. As a result of harmonic fluctuations in the outdoor temperature, the indoor temperature also changes harmonically, but with a certain time shift relative to the fluctuations in the outside temperature.

A decrease in the outdoor temperature leads to an increase in the heat flow through the external enclosing structures. This causes a drop in the internal air temperature, which is detected by the sensor of the thermostatic valve installed on the heater. The system automatically increases the flow rate of the heat carrier, increasing the heat output of the heater and ensuring that the set temperature regime in the room is maintained.

The opposite situation is observed when the outside air temperature rises: heat losses through external enclosing structures decrease, which leads to an increase in the internal air temperature. Thermostatic valves respond to this by reducing the flow rate of the heat carrier, thereby reducing the heat output of heating devices.

Consequently, heating devices equipped with thermal valves with thermal heads installed in rooms for various purposes, depending on the type of heating device, are local or linear heat sources of periodic action.

Fluctuations in the internal air temperature are transmitted to the inner surface of the outer fence and then propagate in the thickness of the structure with a gradual attenuation of the amplitude, reaching the outer surface with a certain phase shift.

For a complex solution of the problems of the oscillation amplitude and phase shift between the outdoor and indoor air temperatures, the paper uses a "Symbolic" method based on the use of hyperbolic functions of a complex variable.

The main task of the study is to determine the limits (amplitude) of fluctuations in the temperature of the indoor air in a heated room during periodic heating.

When solving the problem of determining the limits of air temperature fluctuations inside a heated room, it is necessary to simultaneously determine temperature fluctuations on the inner surface of the fence. The distribution of temperature fluctuations inside the enclosing structures is necessary to determine the coefficient of heat perception of the internal surfaces of fences.

A heating device installed in a heated room emits a heat flow, the value of which varies over time according to the law of the correct harmonic function. As a result, the temperature of the indoor air, which is under the influence of an alternating heat flow, also undergoes harmonic fluctuations with an identical period, which will contribute to the appearance of temperature fluctuations on the inner surface of the outer enclosing structure of the heated room.

Figure 2 shows the time dependences of these values: fluctuations in the temperature of the indoor air are shown by the curve  $\theta_{in}$ , temperature changes on the inner surface of the outer enclosing structure are shown by the curve  $\theta_{in,s}$ , and the corresponding heat flow from the air to the surface is shown by the curve  $Q$ .

The indicated curves are shifted relative to each other in phase.

In particular, the heat flow curve  $Q$  determines the position of the temperature curve of the inner surface  $\theta_{in,s}$  with a phase shift  $\varphi_2$ , and the curve  $\theta_{in,s}$ , in turn, lags behind the air temperature curve  $\theta_{in}$  by an angle  $\varphi_3$ . There is the following relationship between the heat flow  $Q$  and the temperature of the inner surface of the outer enclosing structure  $\theta_{in,s}$ :

$$\frac{Q}{\theta_{in,s}} = Y_{ins} \quad (9)$$

where:  $Y_{in,s}$  is the coefficient of heat assimilation of the inner surface of the outer enclosing structure.

$$\frac{Q}{\theta_{in}} = B \tag{10}$$

where:  $B$  is the heat absorption coefficient of the inner surface of the fence.

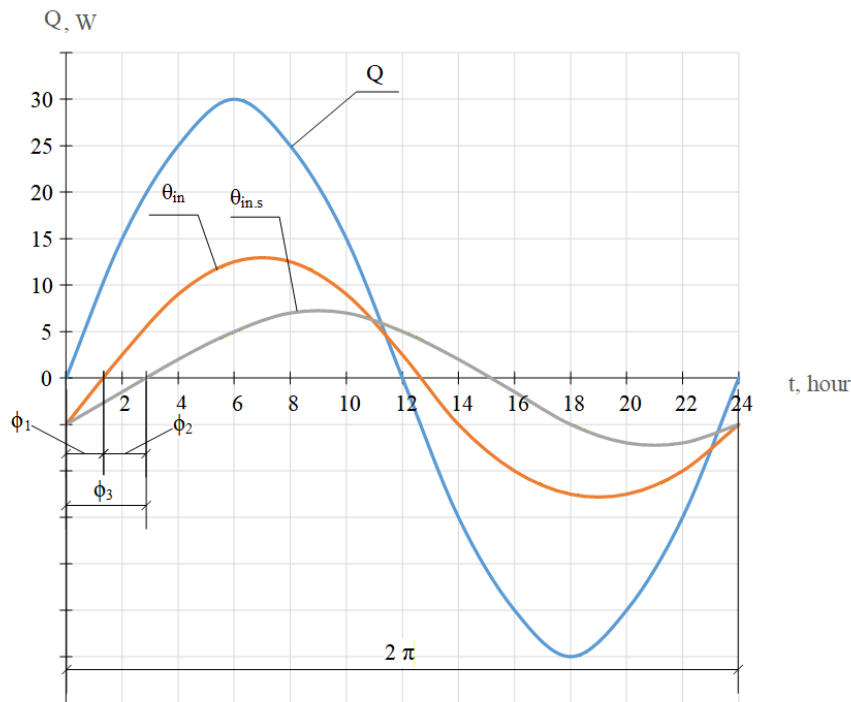
Based on the above, the relationship is valid:

$$Q \sin x = [\theta_{in} \sin(x - \varphi_1) - \theta_{in,s} \sin(x - \varphi_2)] \alpha_{in} \tag{11}$$

where:  $Q$  is the heat flow from the heat release source,  $W$ ;  $\theta_{in}$  is the internal air temperature,  $^{\circ}C$ ;  $\theta_{in,s}$  is the temperature on the inner surface of the outer enclosing structure,  $^{\circ}C$ .

Dependence of changes in the values of  $Q$ ,  $\theta_{in}$  and  $\theta_{in,s}$  by the hours of the day for the period  $z=24$  hours.

$$Q \sin 15t = [\theta_{in} \sin(15t - \varphi_1) - \theta_{in,s} \sin(15t - \varphi_2)] \alpha_{in} \tag{12}$$



**Fig. 2.** Graph of changes in values in a heated room by the hours of the day for the period  $z=24$  hours and  $\alpha_{in} = 8.7 \text{ W}/(\text{m}^2 \text{ } ^{\circ}C)$ : heat flow from the heat source –  $Q$  [W], internal air temperature- $\theta_{in}$  [ $^{\circ}C$ ]; temperature on the inner surface of the outer surface of the enclosing structure -  $\theta_{in,s}$  [ $^{\circ}C$ ].

### 2.3 INFLUENCE OF THE RATIO OF THE TYPE OF HEAT TRANSFER OF A HEATING DEVICE ON THE AMPLITUDE OF AIR TEMPERATURE FLUCTUATIONS IN A HEATED ROOM

The amplitude of air temperature fluctuations in the room is determined by the formula

$$\theta_B = \frac{m \cdot W_{av}}{\sum B \cdot A} \tag{13}$$

where:  $m$  is the coefficient of uneven heat transfer of the heating device;  $W_{av}$  is the time-averaged heat transfer of the heating device;  $\Sigma B \cdot A$  is the heat absorption capacity of the room;  $A$  is the area of a separate fence, internal and external, by internal size, without taking into account the thickness of walls and floors;  $B$  is the heat absorption coefficient of the inner surface of the enclosing structure, defined as

$$B = \frac{A_q}{A_{t\ in}} \quad (14)$$

or

$$B = \frac{1}{\left(\frac{1}{Y_1} + \frac{1}{\alpha_{in}}\right)} \quad (15)$$

where:  $A_q$  is the amplitude of fluctuations in the heat flow passing through the fence surface;  $A_{t\ in}$  is the amplitude of fluctuations in the temperature of the medium (air) surrounding the fence surface caused by fluctuations in the heat flow;  $Y_1$  is the heat absorption coefficient of the inner surface of the fence;  $\alpha_{in}$  is the heat transfer coefficient on the inner surface of the fence.

The heat absorption coefficient of filling light openings is defined as:

$$B = \frac{1}{1,08 \cdot R_t} \quad (16)$$

where:  $R_t$  is the heat transfer resistance of filling light openings ( $m^2 \cdot ^\circ C/W$ )

The coefficient of uneven heat transfer of a heating device is determined by the formula:

$$m = \frac{W_{max} - W_{min}}{2 \cdot W_{av}} \quad (17)$$

where:  $W_{max}$  – heat transfer of the heating device at the moment of maximum heat transfer in  $W$ ;  $W_{min}$  – heat transfer of the heating device at the moment of minimum heat transfer in  $W$ .

The values of  $W_{max}$  and  $W_{min}$  were determined during testing of heating devices in the isothermal chamber of the test laboratory.

The complete heat transfer of the heater to the environment occurs in two ways: convection and radiation. The average time value of the heat transfer of the heater ( $W_{av}$ ), taking into account convective and radiant heat transfer, is defined as:

$$W_{av} = \frac{W_{max} - W_{min}}{m \cdot (W_{max} + W_{min})} \quad (18)$$

The amplitude of fluctuations in the internal air temperature ( $\theta_{in}$ ) is directly affected by the heat transferred to the air environment from the heater by convection. The heat transferred to the air of the internal environment by radiation does not directly affect the value of the amplitude of vibrations of the internal air  $\theta_{in}$ .

The amplitude of heat flow fluctuations is determined by the dominant type of heat transfer of the heater to the environment of the heated room – radiant or convective. Ignoring the type of heating device and the form of the dominant heat transfer mechanism, as well as the assumption that all the heat transferred from the surface of the heating device is used to increase the indoor air temperature, leads to overestimated estimates of the amplitude of heat flow fluctuations compared to actual values.

The heat flux released from the heater to the environment of the heated room by radiation depends on the temperature difference between the internal surfaces of the enclosing structures and the surface of the heater and is determined by the formula:

$$W^R = \alpha_R [t_h - t_{cons}] \quad (19)$$

The heat flow transferred from the surface of the heater to the environment of the heated room by convective heat exchange is defined as:

$$W^C = \alpha_C [t_h - t_{in}] \quad (20)$$

where:  $\alpha_R$ ,  $\alpha_C$  are the radiant and convective heat transfer coefficients, respectively.

There is a phase shift between the temperature of the internal surfaces of the heated room,  $t_h$ , and the temperature of the internal environment,  $t_{in}$ , as well as between the amplitudes of vibrations of the internal air and internal surfaces. The value that depends on the design features of the fence. In the case when the enclosing structures of a heated room have the same value of the heat absorption coefficient, the following equality occurs:

$$\varepsilon = \varphi_1 \quad (21)$$

The phase shift between the oscillation amplitudes of the internal environment and the internal surfaces of the enclosing structures of the heated room is defined as:

$$\varphi_3 = \varphi_2 - \varphi_1 \quad (22)$$

The total value of the oscillation amplitude of the radiant and convective heat flow from the heater is defined as:

$$A_{os} \approx A_{os}^R + A_{os}^C \quad (23)$$

In the absence of a convective component of the heat flow from the heater, the temperature of the internal environment is formed solely under the influence of the radiation component of the heat flow from the heater. Radiation heats the surfaces of internal enclosing structures to varying degrees, which, in turn, affects the formation of indoor air temperature:

$$\theta_{in}^R = \frac{\sum \theta_{in.s}^R \cdot A_i}{\sum A_i} \quad (24)$$

where:  $\theta_{in}^R$  is the amplitude of air temperature fluctuations in the heated room under the influence of radiation heat transfer from the heater;  $\theta_{in.s}^R$  is the amplitude of temperature fluctuations of the inner surface of enclosing structures in the heated room under the influence of radiation heat transfer from the heater;  $A_i$  is the area of the inner surface of the fence or other surfaces inside the heated room (furniture), where the same temperature is observed over the entire area.

If the radiation heat from the heater is completely transferred to the surrounding surfaces and is not absorbed by the air of the internal environment of the heated room, then

$$\theta_{in.s}^R = \theta_{in}^R \quad (25)$$

Radiation heat from the heater is transferred to the internal surfaces of the heated room, heating them. Further, the heat from the surfaces heats up the air inside the room with secondary convective flows. The value of the heat flow is determined by the formula:

$$Q_{cons}^C = \alpha_{in} (\theta_{in.s}^R - \theta_{in}^R) \quad (26)$$

Another part of the radiant heat is absorbed by the fence structure and causes the temperature of its surfaces to fluctuate:

$$\theta_{in.s}^R = \frac{Q}{Y_{in}} \quad (27)$$

The amount of heat absorbed by the structure will be:

$$Q_{cons}^C = Q_{cons}^R - \alpha_R(\theta_{in.s}^R - \theta_{in}^R) \quad (28)$$

The amplitude of temperature fluctuations of the inner surface of enclosing structures in a heated room under the influence of radiation heat transfer from a heating device is defined as:

$$\theta_{in.s}^R = \frac{Q_{cons}^R - \alpha_R(\theta_{in.s}^R - \theta_{in}^R)}{Y_{in}} \quad (29)$$

or

$$\theta_{in.s}^R = \frac{Q_{cons}^R + \alpha_R \theta_{in}^R}{Y_{in} + \alpha_{in}} \quad (30)$$

As a result of the joint solution of equations (24) and (27), the expression was obtained:

$$\theta_{in}^R = \frac{1}{A} \left[ \sum \frac{Q_{cons}^R A_i}{Y_{in} + \alpha_{in}} + \theta_{in}^R \cdot \sum \frac{\alpha_{in} \cdot A_i}{Y_{in} + \alpha_{in}} \right] \quad (31)$$

$$\theta_{in}^R = \frac{\sum \frac{Q_{cons}^R A_i}{Y_{in} + \alpha_{in}}}{\sum A_i - \sum \frac{\alpha_{in} \cdot A_i}{Y_{in} + \alpha_{in}}} \quad (32)$$

Fluctuation of air temperature under the influence of convection heat:

$$\theta_{in}^C = \frac{A_{cons}^C}{1.08 \cdot \sum B \cdot A_i} \quad (33)$$

Ignoring the phase shift between  $\theta_{in}^R$  and  $\theta_{in}^C$ , we obtain the expression:

$$\theta_{in} = \theta_{in}^C + \theta_{in}^R = \frac{\sum \frac{Q_{cons}^R A_i}{Y_{in} + \alpha_{in}}}{\sum A_i - \sum \frac{\alpha_{in} \cdot A_i}{Y_{in} + \alpha_{in}}} + \frac{A_{cons}^C}{1.08 \cdot \sum B \cdot A_i} \quad (34)$$

To simplify the solution of the problem of radiant heat exchange, it is assumed that radiant heat from the open surfaces of the heater is evenly dispersed throughout the room, falling on all fences with the same intensity. Taking into account the above, it follows that equation (34) can be represented as:

$$\theta_{in} = \frac{Q_{cons}^R \sum \frac{A_i}{Y_{in} + \alpha_{in}}}{\sum A_i - \sum \frac{\alpha_{in} \cdot A_i}{Y_{in} + \alpha_{in}}} + \frac{A_{cons}^C}{1.08 \cdot \sum B \cdot A_i} \quad (35)$$

#### 2.4 PHASE SHIFT OF HEAT ABSORPTION OF ENCLOSING STRUCTURES

Heat flows from the heater to the internal environment of the heated room, neglecting the heat capacity of the air due to its insignificance, are absorbed by the internal surfaces of the enclosing structures.

The heat released from the heater is completely absorbed by the internal surfaces of the enclosing structures, which leads to a complete absence of phase shift of the curves describing the processes of heat transfer from the heater and heat absorption by structures.

In a heated room, in which all enclosing structures have the same thermophysical and structural characteristics, the heat transfer coefficient  $\alpha_R$  is constant and the same for all structures, the amplitude of fluctuations in the heat flow from the heater is defined as:

$$A_{os} = m \cdot W_{av} \quad (36)$$

Since the heat flow from the heater is completely absorbed by the enclosing structures with an area of  $A_h$ , with the same value of the heat transfer coefficient, therefore, the expression is valid:

$$A_{os} = \sum Q \cdot A_h \quad (37)$$

Based on the analytical dependences, calculations were made and graphs of changes in air temperature and temperature on the inner surface of external enclosing structures were constructed, depending on the degree of heating of the heater.

It is assumed that heating devices can be placed in a heated room in three characteristic areas:

- free installation.
- installation of the heater along the wall;
- installation of the heating device in the corner of the heated room.

Since the absolute value of the amplitude of the heat transfer fluctuations of the heating device is not affected by the correction coefficients, this value can be assumed to be equal to  $A_k = 100$ .

Three cases of installing a heating device inside a heated room are considered: installation in a free zone; installation along fences; installation in a corner (Table 1).

**Table 1.** Dependence of the amplitude of fluctuations in the air temperature of the heated room on the inertia of the fences and the amplitude of fluctuations in the heat transfer of the heating device.

№	Inertia of the fences of the room	Location of the heater	$A_{cons}^R = 0$		$A_{cons}^R = 0.6 A_{os}$		$A_{cons}^R = A_{os}$	
			$\theta_{in}$	%	$\theta_{in}$	%	$\theta_{in}$	%
1.	High inertia	Free installation of the heater	0.386	100	0.273	71.0	0.209	54
		Installation of the heater along fences	0.386	100	0.287	74.3	0.209	54
		Installation of the heater in the corner of the heated room	0.386	100	0.306	79.3	0.209	54
2.	Low inertia	Free installation of the heater	0.695	100	0.557	80.0	0.49	71.8
		Installation of the heater along fences	0.695	100	0.593	85.3	0.49	71.8
		Installation of the heater in the corner of the heated room	0.695	100	0.617	88.8	0.49	71.8

In the laboratory of the Scientific and Educational Center for Heat and Gas Supply and Ventilation (SEC TGV) The National Research Moscow State University of Civil Engineering (NRU MGSU) conducted experimental studies to determine the nature of the temperature distribution in the volume of the room. The aim of the experiment was to study the temperature fields in the volume of the heated room and on the internal surfaces of external enclosing structures, depending on the type of heat transfer dominance and the location of the heating device at a given value of the heat resistance of external enclosing structures.

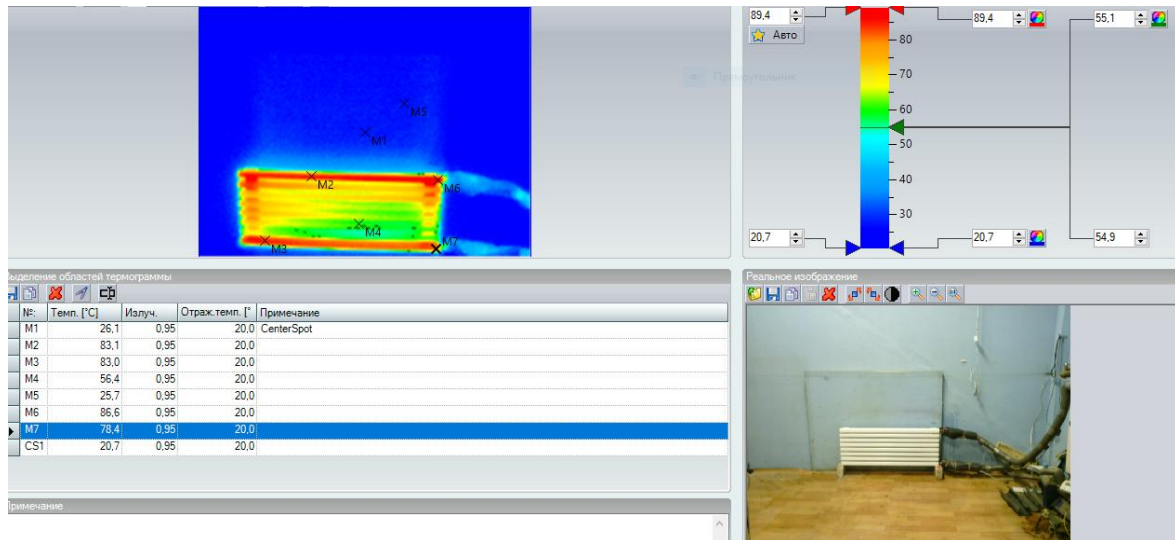
Fig. 3 shows a laboratory stand for determining the nature of the temperature distribution over the height and area of the room under study with linear or local heat sources.



**Fig. 3.** Isothermal chamber with a laboratory stand for determining the nature of the temperature distribution in the study area, in the presence of spatial heat exchange.

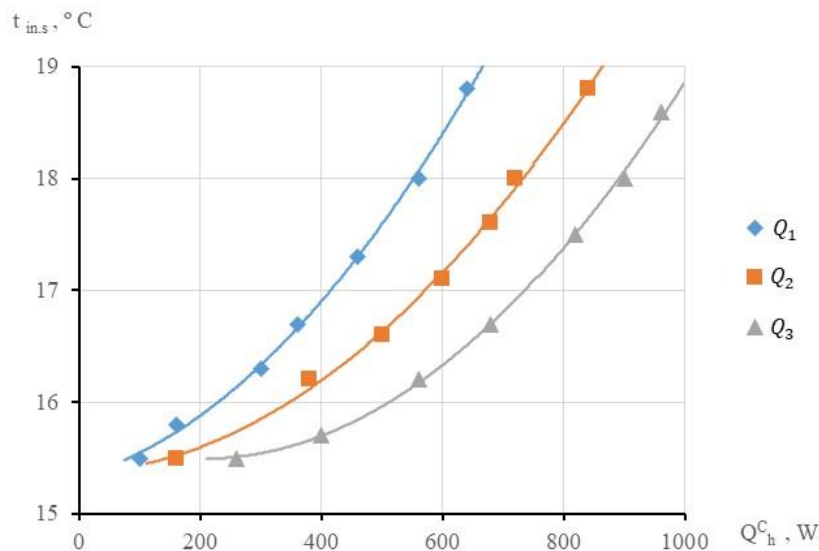
A heating device is installed in the isothermal chamber as a local or linear heat source. The purpose of the experiment is to determine the nature of changes in air temperature, as well as the temperature on the internal surfaces of external enclosing structures, at a certain given value of the nominal heat flux. A heating device is used as an example of a local or linear heat source in experimental studies. The conducted experimental studies revealed the nature of changes in the dependence of changes in air temperature and temperature on the internal surfaces of external enclosing structures, on the form of heat transfer dominance of the heating device.

The thermogram of heating of the heating device depending on the form of heat transfer dominance, at the temperature of the coolant  $t \approx 90^\circ\text{C}$ , is shown in Fig. 4.



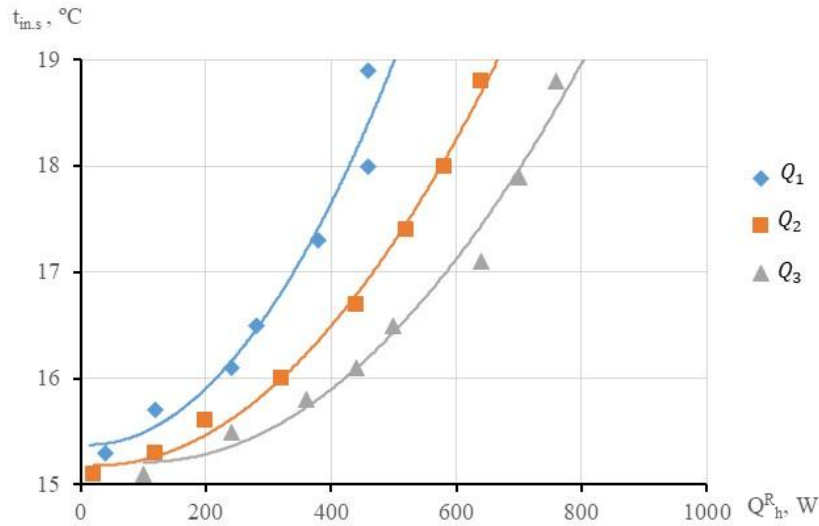
**Fig. 4.** Thermogram of heating of a heating device depending on the form of heat transfer dominance at the coolant temperature  $t \approx 90^\circ\text{C}$ .

Obtained as a result of experimental studies, graphical dependence of the change in the temperature of the inner surface of enclosing structures with the dominance of the convective form of heat exchange of the heating device ( $Q_{1,2,3} > Q_{hc} > 0.75 \cdot Q_{1,2,3}$ ) depending on the value of the nominal heat flow of the heating device:  $Q_1 = 1250 \text{ W}$ ;  $Q_2 = 1500 \text{ W}$ ;  $Q_3 = 1750 \text{ W}$ , shown in Fig. 5.



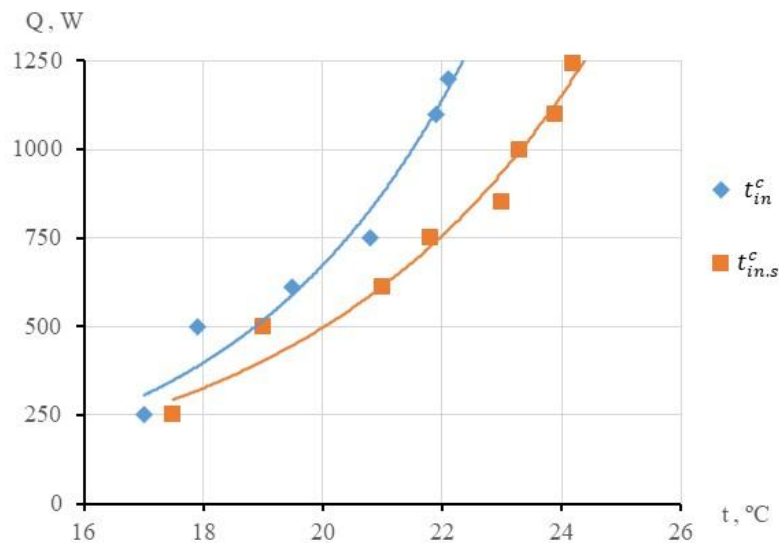
**Fig. 5.** Change in the temperature of the inner surface of enclosing structures with the predominance of convective heat exchange ( $Q_{1,2,3} > Q_{hc} > 0.75 \cdot Q_{1,2,3}$ ) depending on the power of the nominal heat flow of the heater:  $Q_1 = 1250 \text{ W}$ ;  $Q_2 = 1500 \text{ W}$ ;  $Q_3 = 1750 \text{ W}$ .

Obtained as a result of experimental studies, the change in the temperature of the inner surface of enclosing structures with the predominance of heat exchange of the radiation form, depending on the power of the nominal heat flow of the heater, is shown in Fig. 6.



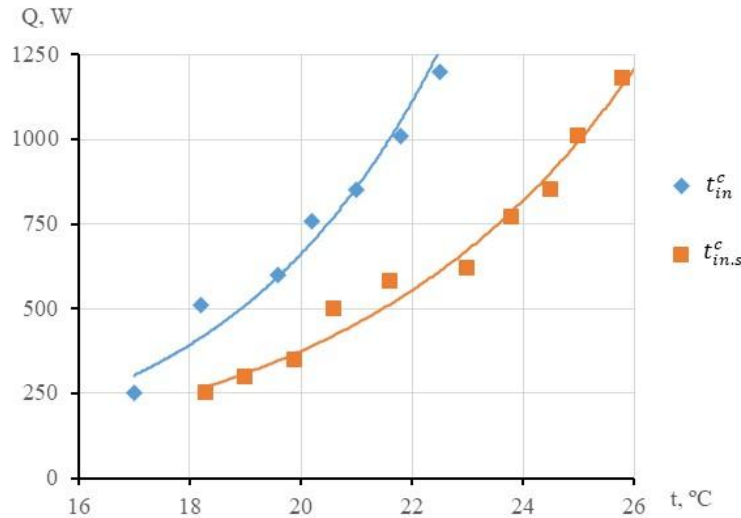
**Fig. 6.** Change in the temperature of the inner surface of enclosing structures with the predominance of heat exchange of the radiation form ( $0 < Q_h^R < 0.75 Q_{1,2,3}$ ) depending on the power of the nominal heat flow of the heater:  $Q_1 = 1250$  W;  $Q_2 = 1500$  W;  $Q_3 = 1750$  W.

The dependence of changes in air temperatures and internal surfaces of the enclosing structures of the heated room on the value of the heat flow of a heating device with a predominance of radiation heat transfer obtained as a result of experimental studies is shown in Fig. 7.



**Fig. 7.** Dependence of changes in air temperatures and internal surfaces of the enclosing structures of the heated room on the value of the heat flow of the heating device with the dominance of radiation heat transfer.  $Q$  is the heat flow from heating devices, W;  $t_{in}^R$  is the air temperature in the heated room, °C;  $t_{in.s}^R$  is the temperature on the inner surface of the enclosing structure, °C.

The dependence of changes in air temperatures and internal surfaces of the enclosing structures of the heated room on the value of the heat flow of a heating device with a predominance of convective heat transfer, obtained as a result of experimental studies, is shown in Fig. 8.



**Fig. 8.** Dependence of changes in air temperatures and internal surfaces of the enclosing structures of the heated room on the value of the heat flow of the heater with the dominance of convective heat transfer.  $Q$  – heat flow from heating devices, W;  $t_{in}^c$  – air temperature in the heated room, °C;  $t_{in.s}^c$  – temperature on the inner surface of the enclosing structure, °C.

#### 4. CONCLUSION

The choice of the location of the heater depends on a combination of two factors: the characteristics of the thermal insulation of the room (thermal resistance of the enclosing structures) and the method of heat transfer of the heater itself (convection or radiation). Following the recommendations, you can ensure efficient use of thermal energy and create comfortable living conditions or stay indoors.

Heating devices equipped with thermal valves with thermal heads installed in heated rooms of buildings for various purposes, depending on the type of heating device, are local or linear heat sources of periodic action.

Heating devices with a predominance of convective heat transfer are recommended to be placed in a heated room with low-inertia enclosing structures.

It is recommended to place heating devices with a dominant radiation form of heat transfer in a heated room with high-inertia enclosing structures.

In rooms with high-inertia enclosing structures, heating devices can be installed so that heat from them can be effectively absorbed by enclosing structures, accumulate in the volume of structures and contribute to its uniform distribution.

To maintain a stable temperature regime during the day, it is necessary to take into account the thermal inertia of the room when choosing the design features and method of heat transfer of heating equipment. With an increase in the thermal inertia of the heated room as a whole, the power of the heating system decreases.

A mathematical model of spatial non-stationary heat transfer in a heated room is developed, taking into account various thermophysical characteristics of enclosing structures and the type of heat release sources.

Laboratory studies of heat exchange in an isothermal chamber with linear or linear heat sources have established a graphical dependence of the nature of changes in air temperature and the temperature of the internal surfaces of external enclosing structures, depending on the inertia of the structures and the dominant form of heat transfer of the heating device.

#### 5. ACKNOWLEDGEMENTS

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### INFORMATION ABOUT THE AUTHORS

**Sargsyan S.V.**, e-mail: [sargsyansv@mgsu.ru](mailto:sargsyansv@mgsu.ru), ORCID: <https://orcid.org/0000-0003-3356-9116>, SCOPUS: <https://www.scopus.com/authid/detail.uri?authorId=57204373040>, Moscow State University of Civil Engineering, Candidate of Engineering Sciences (Ph.D.), Associate Professor, Head of the Department of Heat and Gas Supply and Ventilation, Director of the Scientific and Educational Center «Heat and Gas Supply and Ventilation»

**Agafonova V.V.**, e-mail: [agafonovavv@mgsu.ru](mailto:agafonovavv@mgsu.ru), ORCID: <https://orcid.org/0000-0001-5839-0556>, SCOPUS: <https://www.scopus.com/authid/detail.uri?authorId=57204528191>, Moscow State University of Civil Engineering, Candidate of Engineering Sciences (Ph.D.), Associate Professor of the Department of Heat and Gas Supply and Ventilation