This article presents the developed methodology for the numerical simulation of radiation heat transfer from water panel heaters and optimization results of water temperature in the supply pipeline from the mass flow rate of the heat-carrier and the surface area for a given thermal power of the panel system. A numerical mathematical model is developed in the assumption that heat transfer occurs by means of radiation heat exchange by longitudinal ribs and pipes, which are thermally insulated on top. It is assumed that the temperature of the rib base is equal to the temperature of the outer wall of the pipe. The irregularity of the radiation density in different directions depends on the angle and distance to the irradiated area. The aim of the work is to develop a methodology to simulate the heat transfer processes of a radiation panel water heating system and optimization of design and operating parameters. The radiation intensity is determined by a numerical method using the MATLAB software package. Our results of experimental studies of the radiation flux density are presented and compared with the results of numerical ones. The thermodynamic efficiency of a panel heating system is analyzed using the entropy production method (exergy destruction). The multi-criteria optimization of water temperature in the supply pipeline is performed by LPτ-search. It is found that the unevenness of surface temperature of panels reaches 24.4% as well as for the panels of about 50m in length a decrease in water temperature to 20K is observed, which leads to the unevenness of radiation flux density over the heated area. The area of the cooling system as a function of water temperature and the conditions under which the entropy production in the system is minimal is determined.

Keywords: heating and cooling panel, numerical and experimental techniques, optimization of thermal comfort, energy, heat supply, radiation heating system, mathematical model, heat exchange.

1. Introduction

Water (vapor) radiant ceiling panels are widely used in heating systems of industrial, administrative and public, sport, shopping and entertainment buildings. The heating of warehouses, production shops and workshops of factories, railway stations, swimming pools and concert halls, where the height of buildings is more than 3 m, does not allow using classical (traditional) water heating systems and the use of air heating is inefficient. These systems are more efficient (up to 35-40%), as compared with the air heating systems and can be used both for heating and for conditioning
premises. They are characterized by high comfort and hygiene because of absence of forced air circulation and lower air temperature in the working zone [1-8]. The use of radiant heat transfer principle allows the lower air room temperature in the operating area to be maintained in accordance with Standard ISO 7730, DIN EN 14037-1, -2, -3 requirements [9-11].

2. State of the problem

The specific features of the design and calculation method for cosmic tubular radiators [1, 2, 12-15] with two longitudinal edges are the following:
- the shape of the radiating panels with bilateral radiation (Fig. 1);
- the distance between the pipes is equal to double height of the ribs;
- the heat carrier of the power plant moves in the tubular channels;
- the construction with an internal rib (Fig. 1g) in which the inner part is a convective rib and the outer part is a radiating rib is the most effective one;
- thin-wall pipes with external ribbing are considered separately, i.e. thermal pipe insulation and thermal rib conductivity, assuming that the temperature of the rib base is equal to the temperature of the outer pipe wall pipe.

The cooling systems of space power installations differ from the heating/cooling systems of buildings, but the experience of designing and building the ones can be useful.

Calculation of the intensity of the radiation heat flux is fundamental to the design procedure of radiant heating/cooling systems using the analysis of thermal comfort and the program of building information modeling (BIM).

Radiant panels are widely used in the cooling systems of spacecraft power plants [1, 2, 15]. The methods of their calculation are developed and the optimum parameters are determined. The radiation heat transfer is modeled on the basis of two-dimensional equation of the rib heat-conduction in a diathermia medium in [12]. Different profiles of ribs (rectangular, trapezoidal and triangular) and the surface emissivity have been studied. The optimization of the system parameters (heat-carrier velocity, pipe diameter and number of panels) is done. The emitter parameters, such as the diameter of the pipes, the material, thickness and height of the ribs and various heat-carriers are studied in [13-15]. Heat-carriers are hydrogen, neon and Na-K alloy; temperature range is 624-345 K; rib and pipe material is aluminum-steel; aluminum-titanium; carbon-carbon; design parameters are the following: pipe diameter is from 10/12 to 18/20, the rib height is from 0.02 to 0.1m and rib thickness is 0.001m.

Figure 1 shows the constructive shapes of radiant panels. Figure (1a) is a cylindrical pipe with the attached (soldering, welding) ribs; Figure (1b) is a common rib with the cylindrical impressions, into which cylindrical pipes are stacked and soldered; Figure (1c) is a common bilayer rib into cylindrical impressions of which the cylindrical pipes are stacked and soldered; Figure (1d) is a common rib (flat plate) with cylindrical or oval pipes unilaterally attached thereto; Figure (1e) is a pair of parallel planar common ribs, between which the cylindrical (or oval and flattened) pipes are stacked and soldered to them; Figure (1f) is a cylindrical pipe with ribs attached to it by soldering or welding and outer cylindrical screen, designed for better protection from meteor hazard; Figure (1g) is a rib which presents an effective design, i.e. the inner part of the pipe is convective and the outer part is an emitting rib.
Fig. 1. Structural shapes of the radiating planar panel: 1 is a pipe, 2 is a rib, 3 is an inner pipe, 4 is an outer pipe (protective casing), 5 is an upper half-pipe, 6 is an emitting part of the rib, 7 is a convective part of the rib, 8 is a lower half-pipe.

The main emitting panel surface is the surface of longitudinal flat ribs, through which the major share of the radiation heat flow is emitted. The energy equation for the ribbed-surface element is as follows [6, 7]:

$$dQ = \varepsilon \sigma_0 \eta_{rib} T_r^4 dF = M c_p dT_{\text{h.t.}}.$$  \hspace{1cm} (1),

where $T_r$ is rib degree temperature; $M$ is mass flow rate of the heat transfer; $T_{\text{h.t.}}$ is coolant temperature.

The loss of water pressure in the pipe is defined by the equation:

$$dp = \xi \frac{\rho \omega^2}{d_{\text{input}}} dx.$$  \hspace{1cm} (2)

The efficiency coefficient of the ribbed radiating surface is defined by the equation:

$$\eta_{rib} = 1 - \frac{F_r}{F_{\text{rt.s}}} (1 - \eta_r).$$  \hspace{1cm} (3),

Where $\eta$ is a rib efficiency coefficient, which is determined by the relation, where $m$ is an efficiency parameter:

$$m = \frac{2 \varepsilon \sigma_0 \rho \omega^2 T_w^3}{\lambda r \delta r}$$  \hspace{1cm} (4)

The calculation of the radiator is reduced to the solution of the conjugate thermal-hydraulic problem, described by the equations (1-4). Radiant water panels which are used in heating systems are different from the space ones by materials and structurally, by the temperature level of heat-carriers and heat-exchange conditions, by hydraulic and operational modes. However, the available experience of creating the space cooling systems can be used at the same time. Therefore, simulation and optimization techniques of the radiant water panel parameters in the heating systems need to be
clarified and improved. The problems of temperature-mode irregularity of the panel and working area heating, selecting the optimal panel location height and their surface area need examining.

Radiant panel water systems of buildings were studied in [3-8, 17-28]. The [9-11,16-19] provides design guidelines panel systems.

The work [17] studied the performance of radiant ceiling panel in the classroom. The overall heat transfer coefficient for heating and cooling mode were 3.7 Wm\(^{-2}\)K\(^{-1}\) and 4.8 Wm\(^{-2}\)K\(^{-1}\). The downward heat flux ratio was 61-65% for heating and 65-72% for cooling mode.

This article aims at determining an optimal simulation method for heat transfer calculation of slow thermal mass hot-water radiant heating/cooling panels. The results applied to a large range of panel geometries and operation parameters are given in [18].

The work [19] presents the findings of a field study of occupant thermal comfort with radiant cooling system. The results indicated that there may be lower limits on air speeds to occupants. Statistical analysis indicated that thermal occupant loadings were free of significant correlation with personal, contextual and psychological factors.

One of the primary goals of the radiant heating/cooling community for the past two decades has been the fair and accurate comparison of radiant space conditioning systems with forced convention air systems [20].

The work [21] reviews the practical applications in four major projects involving different building types located under different climatic conditions and presents the associated considerations in the system design.

The work [22] investigates various characteristics of radiant ceiling panel system and their practical application to office building.

The results of the investigation showed that the open-type cooling radiant ceiling panel could (CRCP) provide 54-80% higher nominal cooling than a conventional closed-type CRCP [23].

The results of a hybrid numerical optimization study of a heating radiant panel system are given in [24]. The temperature control of the fluid is shown to be the most important parameter for maximizing comfort and minimizing energy consumption of hot-water heating radiant panels. The work presents the numerical study on radiant heating panels placed at different locations in the room as well as their effects on thermal comfort and the results of the numerical CFD analysis.

This experimental study is conducted under the location configurations of the three different wall panel arrangements for the seven water slow temperature values ranging from 30°C to 42°C. The total heat transfer coefficient of 8.4 Wm\(^{-2}\)K\(^{-1}\) is defined [25]. The heat transfer capability by radiation amounts to approximately 68% of the total number of different studies in question. We studied the effect of such parameters as mass flow rate, supply water temperature, fenestration and ventilation system effects and thermal load distribution [26, 27]. The application of water ceiling panel systems in the north of Europe is shown in [28]. The results of the experimental analysis are given.

However, the density of panel radiation irregularity in the function of the rib height is not well studied. The density of radiation irregularity from two (or three) panels is also not well understood. Thermodynamic analysis of heat exchange processes in heating/cooling systems is not performed in sufficient detail for various practical applications.
3. Mathematical model of infrared water panels

The purpose of creating a mathematical panel model is to determine its thermal power as well as the object irradiation intensity distribution at any given point. This panel, shown in fig. 2, consists of the following elements, such as a pipes of 22 mm diameter (1), a collector (2), a lateral anti-convection overhang (3), insulation layer (4).

![Fig. 2. General view of the panel](image)

When creating a mathematical model one should take into account changes in temperature across the panel. The intensity of the radiation by means of the panel at a given point is determined by the numerical integration of all panel sections with different temperatures (Fig. 3).

![Fig. 3. Module design and calculation model element of the radiating surface](image)

Fig. 3 shows different geometrical parameters of a studied model, where: \( L_1 \) is length of the panel, m; \( L_2 \) is width of the panel, m; \( T'_w \) is water temperature at the input of the panel system, K; \( T''_w \) is water temperature at the output of the panel system, K; \( r \) is distance of the areas \( dF_1 \) - \( dF_2 \), m; \( n_1 \) is a vector normal to the surface A; \( n_2 \) is a vector normal to the surface B; \( \varphi_1 \) is an angular coefficient of a radiation area of the surface A; \( \varphi_2 \) is an angular emission coefficient of the B surface area; \( dF_1 = dx_A dy_A \) is a differential radiation area of the surface A; \( dF_2 = dx_B dy_B \) is a differential radiation area of the surface B; \( h \) is the height of the panel, m, i.e. the distance between the centers of the elements of the areas \( dF_1 \) and \( dF_2 \).
The temperature distribution across the panel depends on the temperature distribution in the panel rib. Since the temperature distribution in the rib is symmetrical in relation to its center, it is sufficient to determine the temperature distribution in the half-rib.

This distribution is obtained by solving a differential equation of the thermal second-order conductivity:

\[ \lambda \delta \frac{d^2 T}{dy^2} = c_0 \epsilon \left[ \left( \frac{T_w}{100} \right)^4 - \left( \frac{T_0}{100} \right)^4 \right] + \alpha (T_w - T_0) \quad (5) \]

This problem is a boundary one, i.e. the boundary conditions are given at the interval ends (rib boundaries):

\[ T(0) = T_w; T\left( \frac{L_2}{2} \right) = 0 \quad (6) \]

Where \( T_w \) is water temperature. The condition is due to the neglect of heat removal from the end of a thin edge (in other words, the condition is due to the small thickness of the plate). It is assumed that the panel surface temperature is equal to the temperature of water, \( T' = dT/dy \), \( T_0 \) is air temperature, \( \alpha \) - heat transfer coefficient. The value of \( \alpha \) is 3.7 Wm\(^{-2}\)K\(^{-1}\). When \( \alpha \) is equal to 4.5 Wm\(^{-2}\)K\(^{-1}\) relevant data [2] and the calculations match better.

The linear density of the heat flow withdrawn from a perfectly conducting half-rib, i.e. when its temperature equals the \( T_w \) water temperature in pipes, is determined by the equation:

\[ q_{id} = L_1 \left[ c_0 \epsilon \left[ \left( \frac{T_w}{100} \right)^4 - \left( \frac{T_0}{100} \right)^4 \right] + \alpha (T_w - T_0) \right] \quad (7) \]

The linear density of the heat flow supplied to the half-edge and withdrawn from it is equal to:

\[ q_r = -\lambda_r \delta_r \frac{dT}{dy} \] at \( y = 0 \) \quad (8)

Rib efficiency is the ratio of the heat flow being actually eliminated toward the flow being eliminated by the perfectly conducting rib, i.e.

\[ \eta_{rib} = \frac{q_r}{q_{id}} \quad (9) \]

Total heat flow from the panel ribs, \( Q_r \), is:

\[ Q_r = n_r L_1 q_r \quad (10) \]

Total heat flow from the pipes, \( Q_{pipe} \), is:

\[ Q_{pipe} = N \frac{\pi}{2} \cdot d_{out,ptu} L_1 \left[ c_0 \epsilon \left[ \left( \frac{T_w}{100} \right)^4 - \left( \frac{T_0}{100} \right)^4 \right] + \alpha (T_w - T_0) \right] \quad (11) \]

Total heat flow from the panel as a whole, \( Q \), is:

\[ Q = Q_r + Q_{pipe} \quad (12) \]

Radiation heat panel flow is calculated like the total one for the ribs and the pipes separately. The density of radiation heat flow per rib width unit is determined by the equation:
To determine the $q_{rad.r.}$ radiation heat flow from ribs the given density must be integrated along the panel rib width or with the current rib temperature. Proper integration was implemented according to the trapz program of the MATLAB software. The radiation heat flow from pipes is determined according to the equation:

$$Q_{rad.p.} = NL_1 d_{output} \frac{\pi}{2} c_0 e \left[ \left( \frac{T_w}{100} \right)^4 - \left( \frac{T_0}{100} \right)^4 \right]$$  \hspace{1cm} (14)

Radiation flow from the panel, $Q_{rad}$, is:

$$Q_{rad} = Q_{rad.r.} + Q_{rad.p.}$$  \hspace{1cm} (15)

Density irregularity of radiation in different directions depends on the angle and distance to the area being irradiated. The mutual area of the $dF_1$ bend radiation and $dF_2$ elementary area is determined by integrating $dH_{12}$ along the coordinate $x_A$, i.e., along the strip:

$$dH_{dF_1-dF_2} = \frac{\pi^2 dY_A dF_2}{-L_1} \frac{dx_A}{\left[ (x_B-x_A)^2 + (y_B-y_A)^2 + D^2 \right]^2}$$  \hspace{1cm} (16)

Let us consider the integral:

$$I = \int_{-L_1}^{L_1} \frac{dx_A}{\left[ (x_B-x_A)^2 + (y_B-y_A)^2 + D^2 \right]^2}$$  \hspace{1cm} (17)

Let:

$$D^2 = (y_B-y_A)^2 + \frac{D^2}{2}; \; x = x_A - x_B$$  \hspace{1cm} (18)

Then:

$$I = \int_{-L_1}^{L_1} \frac{dx}{\left[ x^2 + D^2 \right]^2} = f \left( \frac{L_1 - x_B}{2} \right) - f \left( \frac{-L_1 - x_B}{2} \right)$$  \hspace{1cm} (19)

Where:

$$f(x) = \frac{x}{2D^2(x^2 + D^2)} + \frac{1}{2D^3 \arctg \frac{x}{D}}$$  \hspace{1cm} (20)

Angular emissivity is: $\varphi_{dF_1-dF_2} = \frac{dH_{dF_1-dF_2}}{dF_2}$

or, given that $dF_1 = L_1 \; dy_A$:

$$\varphi_{dF_3-dF_2} = \frac{\pi^2 dF_2}{nL_1} I$$  \hspace{1cm} (21)
To obtain area irradiation intensity it is necessary to integrate \(dQ\) across the panel, i.e., along the \(y_A\) coordinate and divide the result by \(dF\). In addition, if one takes into consideration that \(c_1 = \varepsilon_1 c_0; c_2 = \varepsilon_2 c_0\) where \(\varepsilon_1\) and \(\varepsilon_2\) are the emissivities of the corresponding surfaces, one can finally obtain:

\[
f(x) = \frac{x}{2D^2(x^2 + D^2)} + \frac{1}{2D^3} \arctg \frac{x}{D} \tag{22}
\]

If we neglect the change in temperature across the panel, i.e., its temperature is considered to be constant and equal to the water temperature, then the expression in the square brackets can be taken outside the integral sign and calculated in the following way:

\[
\int_{y_A^1}^{y_A^2} I(y_A)dy_A \quad \text{Since } I(y_A) \text{ is integral, in fact the double integral is calculated:}
\]

\[
f(x) = \frac{x}{2D^2(x^2 + D^2)} + \frac{1}{2D^3} \arctg \frac{x}{D} \tag{23}
\]

Designating:

\[
x = x_A - x_B; y = y_A - y_B,
\]

we obtain:

\[
I = \int_{-L_1-y_B}^{L_1-y_B} \int_{-L_2-y_B}^{L_2-y_B} dx dy
\]

\[
\left[ x^2 + y^2 + \theta^2 \right]^2
\]

Then we get the following at panel constant temperature \(T\):

\[
E = \frac{2}{\pi} \varepsilon_1 \varepsilon_2 c_0 \left[ \left( \frac{T_w}{100} \right)^4 - \left( \frac{T_0}{100} \right)^4 \right] I \tag{25}
\]

The integral can be determined more easily by the numerical method. For this purpose the \textit{dblquad function} of MATLAB program complex is used. The equations (22) and (25) make it possible to determine the radiation intensity of any elementary area with \(x_B, y_B\) coordinates. Entropy production in heat exchange with the (fencing, air) consumer panel system was determined by the equation (26) similarly as in [29, 30]:

\[
\Delta S = M_{cp} \ln \left( \frac{T_w'}{T_w} \right) + \frac{Q_p}{T_0} + \Delta S_{dp} \tag{26}
\]

where \(T_w'\) (K) and \(T_w''\) (K) are heat transfer (water) temperatures at the input and output of the panel system, respectively; \(Q_p\) (W) is thermal power of the panel system, \(T_0\) (K) is the temperature of fencing construction and air. This technique was used for the optimization of heat-exchanger parameters of heat pipes [31].

The first term of this equation is a decline in the water entropy, the second one is an increase in the entropy of a consumer. Given the thermal power \(Q_p\), the second term of the equation (28) is constant. The first term can be written as:

\[
M_{cp} \ln \left( 1 - \frac{Q_p}{T_w} \right) \quad \text{Since the } T_w' / T_w \text{ ratio is close to the unity, then:}
\]

\[
M_{cp} \ln \left( 1 - \frac{Q_p}{M_{cp} T_w} \right) < - \frac{Q_p}{M_{cp} T_w}
\]
Then we get:

\[ Mc_p \ln \left( \frac{T'_w}{T_w} \right) < - \frac{Q_p}{T_w} \]  

(27)

Thus, \( \Delta S \) is practically independent from the \( Mc_p \) flow-rate heat capacity and, hence, from water flow-rate, and the equation (29) is converted to (31):

\[ \Delta S = Q_p \left( \frac{1}{T_0} - \frac{1}{T'_w} \right) \]  

(28)

or

\[ \Delta S = - (\Delta S_{rad} + \Delta S_{conv}) + \frac{Q_p}{T_0} + \Delta S_{ap} \]  

(29)

\[ \Delta S_{rad} = \Delta S_{rad.r} + \Delta S_{rad.p} \]  

(30)

\[ \Delta S_{conv} = \Delta S_{conv.rib} + \Delta S_{conv.pipe} \]  

(31)

\[ \Delta S_{rad} = \int \frac{q(x)_{rad} dx}{T(x)} + \frac{Q_{rad.p}}{T_w} \]  

(32)

\[ \Delta S_{conv} = \int \frac{q(x)_{conv} dx}{T(x)} + \frac{Q_{conv.p}}{T_w} \]  

(33)

\[ \Delta S_{ap} = \frac{M\Delta p}{\rho_1 T_1} \]  

(34)

Simulation and optimization were completed with using the method of LPτ-search [32-35].

4. Results and discussion

The graph of a typical temperature distribution along the rib height across the panel is given below (Fig. 4).
As it can be seen from Fig. 4 the distance between the pipes (rib width) substantially affects the panel surface temperature-profile irregularity, since at the water temperature of 363.15 K, the temperature of rib top is 341.15 K, i.e. it decreases by 24.4%. For panels of about 50 m long or more water temperature drop along the length is observed. Water temperature in the delivery pipe may be different from the temperature in the return pipe by \( \Delta T = T_{\text{in}} - T_{\text{out}} = 10-20 K \), which leads to density irregularity of the radiant flow in the room space. Fig.5 shows the calculated curves of the relation between the object radiation intensity under the panel center and the panel height under the object and the corresponding experimental values [36, 37].

The true value of the being measured radiant flow density with the probability of 0.95 is in the range of (average of two parallel experiments) ± 2 Wm⁻².

As it can be seen, there is a satisfactory agreement between the calculated and experimental data. The radiation intensity with increasing distance from 1 to 3 m decreases from 72 Wm⁻² to 20 Wm⁻².
The distance between the individual panels also affects the regularity of the radiant flow density in the room space. Fig.6 shows the distribution of the radiant flow density under the panel, set at height of 3 m and water temperature of 343.15 K.

As it can be seen, at a distance of 3m from the normal the heat flow density is reduced from 12 Wm$^{-2}$ to 4 Wm$^{-2}$. Fig.7 shows the change in the density of the radiant flow under two panels located at the distance of 4 m from each other, the height of panel installation being 3 m and water temperature being 343.15 K.

According to the developer recommendations [1, 2] regularity of the radiant flow density is provided when installing panels at the distance among them being equal to the height of panel installation. As it can be seen from Fig.7 radiant flux density decreases from 13 Wm$^{-2}$ to 9.8 Wm$^{-2}$, which indicates the validity of the distance between the panels, which may thus be increased by 25-35%. Multi-level full-factorial experiment (FFE) was implemented with the optimization of the parameters of the heating system, involving a complete listing of levels, i.e. $3 \times 3 \times 5 \times 5 = 225$ experiments. Irradiation intensity $E$ (Wm$^{-2}$) of the object being under the center of the panel was used as a response. The equation to determine the intensity of irradiation $E$ was obtained by the least squares method as a result of the reduced computational experiment conducted according to the
second order Hartley plan and containing 25 experiments. The following influencing factors have been taken: \( x_1 = N - 4 \), where \( N \) is the number of pipes in the panel, \( x_2 = (T_w - 70)/20 \), where \( T_w \) is the water temperature, °C, \( x_3 = (\delta - 1.5)/0.5 \), where \( \delta \) is thickness of the panel rib, mm, \( x_4 = H - 2 \), where \( H \) is panel height above the object, m.

The equation obtained by the specified plan is the following:

\[
E = 17.59 + 1.96x_1 + 12.18x_2 + 0.79x_3 - 17.85x_4 - 0.54x_1^2 + \\
+ 1.13x_1x_2 - 1.38x_1x_4 + 0.66x_2^2 - 8.39x_2x_4 - 0.14x_3^2 + 9.26x_4^2
\]  

(35)

The coefficient of determination is \( R^2 = 99.6\% \). Residual variance with 13 degrees of freedom is \( S_{res}^2 = 3.87 \).

According to the results of computational experiment, the total thermal power of the \( Q_{\text{total, panel}} \) (W) and its radiation thermal power \( Q_{\text{rad}} \) (W) were determined according to the equations (12) and (15).

The corresponding equations are:

\[
Q_{\text{total}} = 508.2 + 47.5x_1 + 224x_2 + 14.2x_3 - 13.9x_4^2 + 23.8x_1x_2 - \\
-8x_1x_3 + 12.1x_2^2 + 7.3x_2x_3 - 3.9x_3 \\
R^2 = 99.96\%. S_{res}^2 = 25.8 with 15 degrees of freedom.
\]

\[
Q_{\text{rad}} = 327.4 + 32.7x_1 + 155.1x_2 + 9.9x_3 - 9.2x_4^2 + 17.4x_1x_2 - \\
-5.4x_1x_3 + 12.3x_2^2 + 5.4x_2x_3 - 2.7x_3^2 \\
R^2 = 99.95\%. S_{res}^2 = 13.8 with 15 degrees of freedom.
\]

The factor \( x_4 \) (height) is not included in the equations for \( Q_{\text{total}} \) and \( Q_{\text{rad}} \), since the thermal powers of the panel do not depend on the height.

Fig. 8 shows thermal power of the panel in the function of temperature head [1].

![Fig. 8 Heat emission of the panel in the function of temperature difference (\( \Delta T_m \))](image)

Temperature head is:

\[
\Delta T_m = \frac{(T_w' + T_w'')}{2} - T_0
\]  

(38)

As you can see, the results of comparing the data of mathematical simulation are in satisfactory agreement with the results of the calculated and experimental data of other authors. Tables 1 and 2 show the results of numerical simulation and optimization of the radiant heating/cooling system.
parameters in the function of the influencing parameters with minimum entropy production. The relationship between water temperature in the \( T'_{w} \) delivery pipe and the \( M \) flow rate at a given thermal power of the \( Q_0 \) panel system was estimated in the process of simulation and optimization. The following equation was solved for this purpose:

\[
Q(M, T'_{w}, n, k) = Q_0 \text{ with } \Delta S \rightarrow \min
\]

(39),

where \( n \) is the number of panels in the bend, \( k \) is the number of panel bends. Only those equations for which \( T'_{w} - T''_{w} \leq 20K \) stand out of the total number of solutions of the equation (32). Fig. 9 shows the level curves for a given thermal power \( Q_0 = 80000 \text{ W} \) when the number of panels is 8, 10, 12 and 15.

Fig. 9. Heating panel area and entropy production in the function of water temperature
(thermal power of 80000 W)

Entropy production \( \Delta S \) along the level line with a given number of bends is almost unchanged, but with an increase in the number of bends, \( \Delta S \) is reduced since it is possible to decrease the temperature \( T'_{w} \). Pressure loss in the panel system \( \Delta p \) was calculated for each calculation variant. The entropy production as a function of water temperature and panel area for a given cooling power is shown in Fig. 10.
Fig. 10. Cooling system area and the entropy production in the function of water temperature (cooling power of 15000 W)

Since the pressure losses increase with the increase of the water flow rate, the minimum pressure losses occurring at the lowest water consumption, being equal in this case 1 kgs\(^{-1}\) are shown for each level line. Note that if we make the requirements for temperature difference more stringent, for example., by increasing the minimum flow-rate up to 2 kgs\(^{-1}\), pressure losses will increase dramatically as well. Thus, the data shown in Tables 1 and 2 and in Fig. 11 allow the choice of the area of heating panels, water flow-rate and temperature in the supply pipeline to be defined and justified, which provide minimum entropy production. Fig.11 shows: (1) is 8 panels, \(\Delta p_{\text{min}} = 9263\) Pa, entropy production is 45.9 WK\(^{-1}\); (2) is 10 panels, \(\Delta p_{\text{min}} = 5841\) Pa, entropy production is 39.0 WK\(^{-1}\); (3) – is 12 panels, \(\Delta p_{\text{min}} = 4517\) Pa, entropy production is 33.0 WK\(^{-1}\); (4) is 15 panels, \(\Delta p_{\text{min}} = 3462\) Pa, entropy production is 28.3 WK\(^{-1}\).

Fig. 11. Water temperature in the function of flow rate for heating systems with a different panel area (heating power 80000 W) parameters of the heating system at a given thermal power of 80000 W
Table 1

<table>
<thead>
<tr>
<th>N; L_2, m; ΣF, m^2</th>
<th>T'_w, K; T''_w, K</th>
<th>m, kgs^-1; Δp, kPa</th>
<th>ΔS, WK^-1</th>
</tr>
</thead>
<tbody>
<tr>
<td>8; 2.56; 122.88</td>
<td>361.45; 343.55</td>
<td>1.07; 2.85</td>
<td>45.9</td>
</tr>
<tr>
<td>9; 2.88; 138.24</td>
<td>355.95; 337.65</td>
<td>1.04; 2.24</td>
<td>42.1</td>
</tr>
<tr>
<td>10; 3.20; 153.6</td>
<td>351.35; 332.95</td>
<td>1.03; 1.83</td>
<td>39.0</td>
</tr>
<tr>
<td>11; 3.52; 168.96</td>
<td>347.25; 329.15</td>
<td>1.05; 1.59</td>
<td>36.3</td>
</tr>
<tr>
<td>12; 3.84; 184.32</td>
<td>344.15; 325.75</td>
<td>1.03; 1.33</td>
<td>33.9</td>
</tr>
<tr>
<td>13; 4.16; 199.68</td>
<td>341.25; 322.85</td>
<td>1.03; 1.15</td>
<td>31.9</td>
</tr>
<tr>
<td>14; 4.48; 215.04</td>
<td>338.85; 320.45</td>
<td>1.03; 1.01</td>
<td>30.1</td>
</tr>
<tr>
<td>15; 4.80; 230.4</td>
<td>336.45; 318.45</td>
<td>1.05; 0.93</td>
<td>28.5</td>
</tr>
</tbody>
</table>

Table 2: Parameters of the cooling system at a given thermal power of 15000 W

<table>
<thead>
<tr>
<th>N; L_2, m; ΣF, m^2</th>
<th>T'_w, K; T''_w, K</th>
<th>m, kgs^-1; Δp, kPa</th>
<th>ΔS, WK^-1</th>
</tr>
</thead>
<tbody>
<tr>
<td>9; 2.88; 138.24</td>
<td>278.5; 283.6</td>
<td>0.7; 4.1</td>
<td>2.2</td>
</tr>
<tr>
<td>10; 3.20; 153.6</td>
<td>279.9; 284.7</td>
<td>0.75; 3.9</td>
<td>1.96</td>
</tr>
<tr>
<td>11; 3.52; 168.96</td>
<td>281.1; 285.6</td>
<td>0.8; 3.7</td>
<td>1.77</td>
</tr>
<tr>
<td>12; 3.84; 184.32</td>
<td>282.1; 286.3</td>
<td>0.85; 3.6</td>
<td>1.62</td>
</tr>
<tr>
<td>13; 4.16; 199.68</td>
<td>282.9; 286.9</td>
<td>0.9; 3.5</td>
<td>1.49</td>
</tr>
<tr>
<td>9; 2.88; 138.24*</td>
<td>288.4; 291.5</td>
<td>1.15; 9.76</td>
<td>1.59</td>
</tr>
<tr>
<td>9; 2.88; 138.24*</td>
<td>289.3; 290.7</td>
<td>2.5; 43.6</td>
<td>1.59</td>
</tr>
</tbody>
</table>

*air temperature in the room is 299,15 K

5. Conclusions

The development of radiation panel heating systems requires the improvement and creation of sufficiently accurate and easy-to-use calculation methods. A method has been developed for calculating radiation heat transfer, which makes it possible to determine the heating being uniform in height and area in the working area of production, shopping, sports and other premises. The optimization of heating system panel parameters showed the influence of the mass flow rate of water, the distance between the panel pipes and the distance between the individual panels. A significant decrease in the radiation intensity density of the panel at a distance from the normal is shown as well. So when installing the panel at a height of 3 m, the heat flux density decreases from 12 W/m² to 4 W/m² at a distance of 3 m from the normal. The effect of the distance between two panels on the radiation intensity has been determined. The distance value between two panels being equal to the height of the panels provides a change in the radiation intensity density over the heating area from 13 W/m² to 8-9 W/m², which is permissible. A regression equation has been obtained to determine the heat capacity of the panel system depending on the number of pipes in the panel, the thickness of the panel rib and water temperature. The results of numerical optimization of heating / cooling systems with the use of minimization of entropy production as a criterion allow us to justify the parameters of the panel system, namely the number of panels and their area, mass flow rate and water temperature. The results of simulation provide an effective method of numerical study and optimization for the selection of a panel radiation-heating system using an entropy generation model.
The method of simulation and optimization of water ceiling panels of radiation heating systems by a search method LPτ, taking into account the minimum entropy production has been developed. Designing and operating parameters of the system for heating and cooling buildings have been estimated. The effect of non-uniformity of the temperature field of radiant panels, the height of the panel placement and the distance among them as well as the temperature in the supply pipeline has been shown. The conditions under which the entropy production in the system is minimal are determined.

Nomenclature

\( c_0 \) is emissivity of an absolutely black body, [Wm\(^{-1}\)K\(^4\)]
\( p \) is pressure, [Pa]
\( w \) is speed, [ms\(^{-1}\)]
\( q_d \) is flux density, [Wm\(^{-1}\)]
\( q_f \) is the linear density of the heat flow, [Wm\(^{-1}\)]
\( Q \) is total heat flow from the panel ribs, [W]
\( Q_{pipe} \) is total heat flow from the pipes, [W]
\( Q \) is total heat flow from the panel as a whole, [W]
\( q_{rad} \) is radiation heat panel density, [W]
\( Q_{rad,p} \) is the radiation heat flow from pipes, [W]
\( Q_{rad} \) is radiation flow from the panel, [W]
\( c_p \) is molar heat capacity at constant pressure, [Jmol\(^{-1}\)K\(^{-1}\)]
\( d_{output} \) is output diameter of the pipe, [m]
\( F \) is area, [m\(^2\)]
\( E \) is intensity distribution of radiation, [Wm\(^2\)]
\( F_r \) is ribbing area, [m\(^2\)]
\( F_r \) is radiating total surface, [m\(^2\)]
\( h_r \) – height of the radiating rib part, [m]
\( L \) is length of heating panel, [m]
\( M \) is mass flow of a heat-carrier, [kg\(s^{-1}\)]
\( N \) is number of pipes in the panel, [-]
\( n_r \) is number of half-rubs, [-]
\( Q_p \) is thermal power, [W]
\( q_p \) is the linear density of the heat flux supplied to and withdrawn from the half-ruble, [Wm\(^{-1}\)]
\( \Delta S \) is entropy production during the heat exchange of the panel system with the consumer, [W\(K^{-1}\)]
\( \Delta S_{sp} \) is entropy production when water moves in a pipe, [W\(K^{-1}\)]
\( \Delta S_{rad} \) is entropy production during radiation heat exchange of a panel system with a consumer, [W\(K^{-1}\)]
\( \Delta S_{conv} \) is entropy production at convective heat exchange of the panel system with the consumer, [W\(K^{-1}\)]
\( \Delta S_{rad.r} \) is entropy production during radiation heat exchange of the ribs of the panel system with the consumer, [W\(K^{-1}\)]
\( \Delta S_{rad,p} \) is entropy production during radiation heat exchange of pipes of a panel system with a consumer, [W\(K^{-1}\)]
\( \Delta S_{\text{conv.rib}} \) is entropy production at convective heat exchange of the ribs of the panel system with the consumer, \([\text{W} \cdot \text{K}^{-1}]\)

\( \Delta S_{\text{conv.pipe}} \) is entropy production at convective heat exchange of pipes of a panel system with the consumer, \([\text{W} \cdot \text{K}^{-1}]\)

\( dT_{\text{ht.}} \) is changing the temperature of the heat carrier along the length of the pipe, \([\text{K}]\)

\( T_0 \) is air and irradiated object temperature, \([\text{K}]\)

\( T_r \) is temperature of the rib, \([\text{K}]\)

\( T_w \) is water temperature, \([\text{K}]\)

\( H_{1,2} \) - angular emissivity \([-\] \)

**Greek symbols**

\( \alpha \) is a convective heat irradiation coefficient from the horizontal surface of the panel (facing downward) toward the air, \([\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}]\)

\( \delta_r \) is rib thickness, \([\text{m}]\)

\( \varepsilon \) is emissivity, \([-\] \)

\( \xi \) is coefficient of local resistance, \([-\] \)

\( \sigma_0 \) is Stefan-Boltzmann constant of an absolutely black body, \([\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-4}]\)

\( \eta_t \) is rib efficiency coefficient, \([-\] \)

\( \eta_{\text{rib}} \) is ribbed-surface efficiency, \([-\] \)

\( \lambda_r \) is coefficient of thermal conductivity of the rib, \([\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}]\)

\( \rho \) is density of water, \([\text{kg} \cdot \text{m}^{-3}]\)

**References**


[22] Imanari T., et al., Thermal comfort and energy consumption of the radiant ceiling panel system. Comparison with the conventional all-air system, Energy and Building, 1999, Vol. 30(2), pp.167-175


[34] Sobol I.M., Statnikov R.B., Choosing the optimal parameters in problems with many criteria, Moscow: Bustard, 2006.

