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Optimization of parameters of heating system with low-temperature water panels by changes of entropy

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Abstract
The results of a numerical study of the radiant heat-exchange processes of water ceiling panels of industrial premises heating systems are presented. Numerical simulation and parameters optimization of panel system using the method of LPτ-search on the condition of minimum entropy generation has been completed. The influence of the design parameters of the panels, the conditions of their accommodation and regime operation parameters is studied. The minimum entropy production in the system has been taken as one of the optimization criteria. Estimation of non-uniformity of the temperature field of the surface panels and non-uniformity of radiation intensity has been carried out. The energy performance indicators of the system efficiency are estimated.

Keywords: Radiant ceiling panels; Coefficient of determination; Mathematical model; Bio criterion; Heat transfer rate.

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 to use 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 to maintain the lower air room temperature in the operating area in accordance with Standard ISO 7730, DIN EN 14037-1, -2, -3 requirements.

However, the methods of engineering design, recommendations for their location to ensure uniform heating and operation need to be clarified. One of the methods to justify the selection of a rational solution is a thermodynamic method with the determination of the minimum entropy production in the system of heating/cooling the building premises.

2. The state of the problem

Radiant panels are widely used in the cooling systems of spacecraft power plants [9-13]. The methods of their calculation are developed and the optimum parameters are determined. In the work [9] the radiation heat transfer is modeled on the basis of two-dimensional equation of the rib heat-conduction in a diathermia medium. 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. Ambient temperature (outer space one) is assumed to be zero (T=0K). In articles [10-13] the following emitter parameters: the diameter of the pipes, the material, thickness and height of the ribs and various heat-carriers are studied. Heat-carriers are hydrogen, neon and Na-K alloy; temperature range is 624-345K; 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.

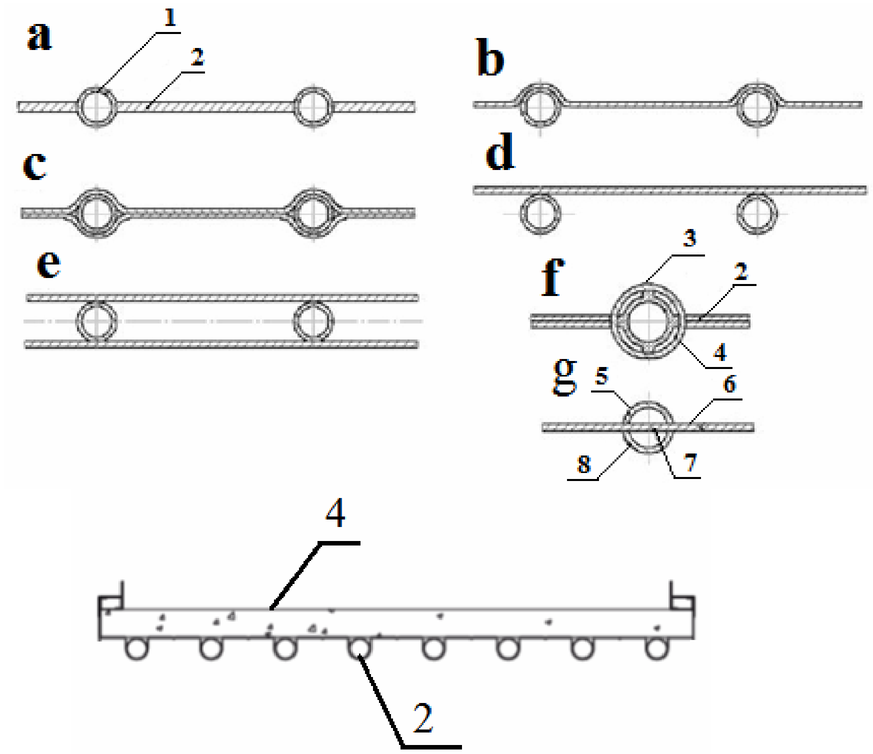

Figure 1. The structural shapes of the radiating planar panel: 1 is a pipe, 2 is a rib, 3 is an inner pipe, 4 is an outer pipe (protective casing), 5 is the upper half-pipe, 6 is an emitting part of the rib, 7 is a convective part of the rib, 8 is a lower half-pipe.

Figure (1a) is a cylindrical pipe with the attached (soldering, welding) ribs; Figure (1d) is a common rib (flat plate) with cylindrical or oval pipes unilaterally attached thereto; 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 (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. 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:

$$dQ = \varepsilon \cdot \sigma \cdot \eta_{rib} \cdot T_r \cdot dF = M \cdot c_p \cdot dT_{h.t.} \quad (1)$$

T_r is rib degree temperature; M is mass flow rate of the heat-carrier.

$$\Delta p = \varepsilon \cdot \frac{\rho \cdot w^2}{d_{out}} \cdot dx \tag{2}$$

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

$$\eta_{rib} = 1 - \frac{F_r}{F_{rts}} \cdot (1 - \eta_r) \tag{3}$$

η_r is rib efficiency coefficient, which is determined by the relation $\eta_r = f(m)$, where m is the efficiency parameter:

$$m = \frac{2\varepsilon \cdot \sigma \cdot h_r^2 \cdot T_w^3}{\lambda_r \delta_r} \tag{4}$$

The temperature of the radiating pipe wall is defined by the heat-transfer equation:

$$T_w = T_{h.t.} - \frac{dQ}{\pi d \cdot dx} \cdot \left(\frac{\alpha_f}{1 + \varepsilon_d \cdot Bi}\right) \tag{5}$$

where $Bi = \dfrac{\alpha \cdot (d_{output} - d_{input})}{2\lambda}$ is the Bio criterion.

The temperature at the base of the rib is determined by the heat balance equation (in the case of Figure 1g):

$$\lambda_r \cdot f \cdot m_c \cdot (T_{h.t.} - T_w) \cdot t \cdot h \cdot (m_c \cdot l_c) = \varphi \cdot \varepsilon \cdot \sigma \cdot h_r \cdot \eta_r \cdot T_r) \tag{6}$$

where $m_c = \dfrac{\alpha_f \cdot U}{\lambda f}$.

The calculation of the radiator is reduced to the solution of the conjugate thermal-hydraulic problem, described by the equations (1-6).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.

3. Mathematical model of infrared water panel

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. The general view of the panel is shown in Figure 2.

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 numerical integration of all panel sections with different temperatures (Figure 3). 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.

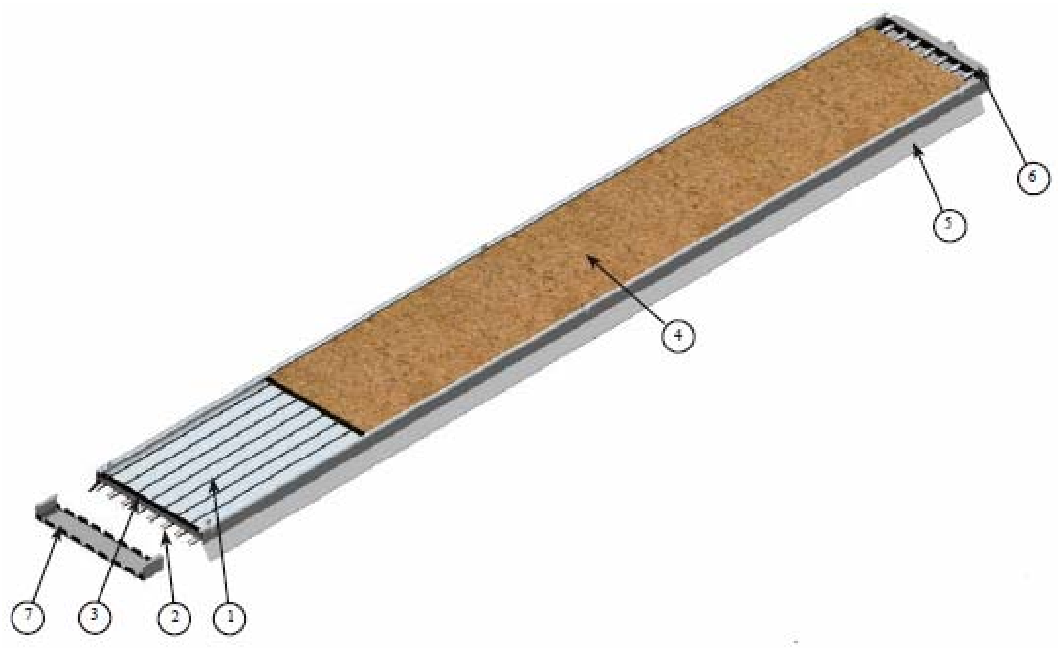

Figure 2. (1) Profiled panel made of coated steel; (2) Pipes of 22mm diameter; (3) Gain traverse; (4) Upper isolating layer; (5) Counter convectional lateral overhang; (6) Square section collector; (7) Waterstrip panel connection element.

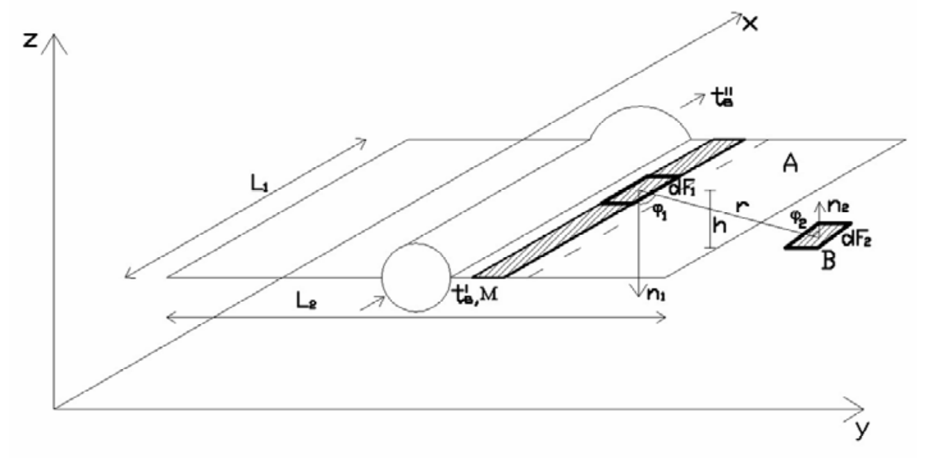

Figure 3.The module design and calculation-model element of the radiating surface.

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

$$\lambda\delta\frac{d^2t}{dy^2} = c_0\varepsilon\left[\left(\frac{T}{100}\right)^4 - \left(\frac{T_0}{100}\right)^4\right] + \alpha(t - t_0) \tag{7}$$

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

$$t(0) = t_w; t'(1) = 0 \tag{8}$$

where t_w is water temperature; $t'=dt/dy$.
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 the pipes, is determined by the formula:

$$q_{id} = l\left[c_0\varepsilon\left[\left(\frac{T_W}{100}\right)^4 - \left(\frac{T_0}{100}\right)^4\right] + \alpha(t_w - t_0)\right] \tag{9}$$

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

$$q_r = -\lambda\delta\left(\frac{dt}{dy}\right)_{y=0} \tag{10}$$

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

$$\eta = \frac{q_r}{q_{id}} \tag{11}$$

Total heat flow from the panel ribs, *W*, is:

$$Q_r = n_r L_1 q_r \tag{12}$$

Total heat flow from the pipes, *W*, is:

$$Q_{pipe} = N \cdot \frac{\pi}{2} D L_1 \cdot \left\{c_0\varepsilon \cdot \left[\left(\frac{T_W}{100}\right)^4 - \left(\frac{T_0}{100}\right)^4\right] + \alpha(t_W - t_0)\right\} \tag{13}$$

Total heat flow from the panel as a whole, *W*, is:

$$Q = Q_r + Q_p \tag{14}$$

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 formula:

$$q_{rad.p.} = n_r L_1 c_0 \varepsilon\left[\left(\frac{T}{100}\right)^4 - \left(\frac{T_0}{100}\right)^4\right] \tag{15}$$

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 formula:

$$Q_{rad.p.} = NL_1 D\frac{\pi}{2}c_0\varepsilon \cdot \left[\left(\frac{T_W}{100}\right)^4 - \left(\frac{T_0}{100}\right)^4\right] \tag{16}$$

Radiation flow from the panel, *W*, is:

$$Q_{rad} = Q_{rad.r.} + Q_{rad.p.} \tag{17}$$

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 band radiation and dF_2elementary area is determined by integrating dH_{12} along the coordinate x_A, i.e., along the strip:

$$dH_{dF-dF_2} = \frac{h^2 dy_A dF_2}{\pi} \int_{-L_{1/2}}^{L_{1/2}} \frac{dx_A}{\left[(x_B - x_A)^2 + (y_B - y_A)^2 + h^2\right]^2} \tag{18}$$

Let's consider the integral:

$$I = \int_{-L_{1/2}}^{L_{1/2}} \frac{dx_A}{\left[(x_B - x_A)^2 + (y_B - y_A)^2 + h^2\right]^2} \tag{19}$$

Let:

$$D^2 = (y_B - y_A)^2 + h^2; x = x_A - x_B, \tag{20}$$

Then

$$I = \int_{-L_{1/2}-x_B}^{L_{1/2}-x_B} \frac{dx}{\left[x^2 + D^2\right]^2} = f\left(L_{1/2} - x_B\right) - f\left(-L_{1/2} - x_B\right), \tag{21}$$

where

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

Angular emissivity is:

$$\varphi_{dF-dF_2} = \frac{dH_{dF-dF_2}}{dF}, \tag{23}$$

or, given that $dF = L_1 dy_A$

$$\varphi_{dF-dF_2} = \frac{h^2 dF_2}{nL_1} \cdot I \tag{24}$$

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_2 area. If in addition to take into consideration that $c_1 = \varepsilon_1 c_0; c_2 = \varepsilon_2 c_0$ where ε_1 and ε_2 are the emissivities of the corresponding surfaces, we finally obtain:

$$E = \frac{h^2}{\pi} \varepsilon_1 \varepsilon_2 c_0 \int_{-L_2}^{L_2} \left[\left(\frac{T(y_A)}{100}\right)^4 - \left(\frac{T_0}{100}\right)^4\right] \cdot I(y_A) dy_A \tag{25}$$

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:

$\int_{-L_2}^{L_2} I(y_A) dy_A$. Since $I(y_A)$ is integral, in fact the double integral is calculated:

$$I = \int_{-L_2/2}^{L_2/2} \int_{-L_1/2}^{L_1/2} \frac{dx_A dy_A}{\left[\left(x_B - x_A \right)^2 + \left(y_B - y_A \right)^2 + h^2 \right]^2} \tag{26}$$

Designating $x = x_A - x_B ; y = y_A - y_B$ we obtain:

$$I = \int_{-L_2/2 - y_B}^{L_2/2 - y_B} \int_{-L_1/2 - y_B}^{L_1/2 - y_B} \frac{dxdy}{\left[x^2 + y^2 + h^2 \right]^2} \tag{27}$$

Then we get the following at panel constant temperature T:

$$E = \frac{h^2}{\pi} \varepsilon_1 \varepsilon_2 c_0 \left[\left(\frac{T}{100} \right)^4 - \left(\frac{T_0}{100} \right)^4 \right] \cdot I \tag{28}$$

The I integral can be determined more easily by the numerical method. For this purpose the *dblquad* function of MATLAB program complex is used.
The formulas (25) and (28) 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 formula (29) similarly as in [14].

$$\Delta S = W \ln \left(\frac{T''}{T'} \right) + \frac{Q}{T_0} + \Delta S_{\Delta p} \tag{29}$$

where T' and T'' are heat-carrier temperatures at the inlet and outlet of the panel system, respectively, K; Q are thermal power of the panel system, W; T_0 is the temperature of fencing construction, K.
The first term of this formula is a decline in the water entropy, the second one is an increase in the entropy of the consumer. Given the thermal power Q, the second term of formula (28) is constant. The first term can be written as $W \ln \left(1 - \frac{Q}{WT'} \right)$ Since the T''/T' ratio is close to the unity, then:

$$W \ln \left(1 - \frac{Q}{WT} \right) \sim -\frac{Q}{WT'}. \text{ then we get } W \ln \left(\frac{T''}{T'} \right) \sim -\frac{Q}{T'} \tag{30}$$

Thus, ΔS is practically independent from the W flow-rate heat capacity and, hence, from water flow-rate, and the formula (29) is converted to (31):

$$\Delta S = Q \left(\frac{1}{T_0} - \frac{1}{T'} \right) \tag{31}$$

or $\Delta S = -(\Delta S_{rad} + \Delta S_{conv}) + \frac{Q}{T_0} + \Delta S_{\Delta p}$ (32)

$$\Delta S_{rad} = \Delta S_{rad.r.} + \Delta S_{rad.p} \tag{33}$$

$$\Delta S_{conv} = \Delta S_{conv.rib} + \Delta S_{conv.pipe} \tag{34}$$

$$\Delta S_{rad} = \int \frac{q(x)_{rad}\, dx}{T(x)} + \frac{Q_{rad.p.}}{T_w} \tag{35}$$

$$\Delta S_{conv} = \int \frac{q(x)_{conv}\, dx}{T(x)} + \frac{Q_{conv.p.}}{T_w} \tag{36}$$

$$\Delta S_{\Delta p} = \frac{M_1 \Delta p}{\rho_1 T_1} \tag{37}$$

Simulation and optimization were completed with using the method of $LP\tau$-search [15-18].

4. Results and discussion

The graph of a typical temperature distribution along the rib height across the panel is given below (Figure 4).

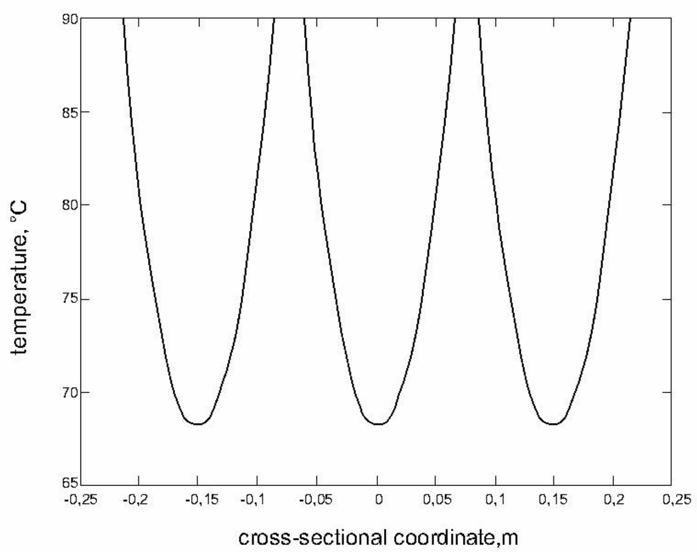

Figure 4. Temperature distribution across the panel at water temperature 90°C, panel width being 0.472m.

As it can be seen from Figure 4 the distance between the pipes (rib width) substantially affects the panel surface temperature-field irregularity, since at the water temperature of 90°C, the temperature of rib top is 68°C, i.e. decreases by 24.4%. For panels of about 50 m long or more water temperature drop along the length is observed. The water temperature in the delivery pipe may be different from the temperature in the return pipe by $\Delta t = t'_p - t'_0 = 10 - 20K$, which leads to density irregularity of the radiant flow in the room space.

Figure 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 [19]. As it can be seen, there is a satisfactory agreement between the calculated and experimental data.

The distance between the individual panels also affects the regularity of the radiant flow density in the room space. Figure 6 shows the distribution of the radiant flow density under the panel, set at height of 3m, and water temperature of 70°C.

As it can be seen, at a distance of 3m from the normal the heat flow density is reduced from 12 W/m^2 to 4 W/m^2. Figure 7 shows the change in the density of the radiant flow under two panels located at the distance of 4m from each other, the height of panel installation being 3m and water temperature being 70°C.

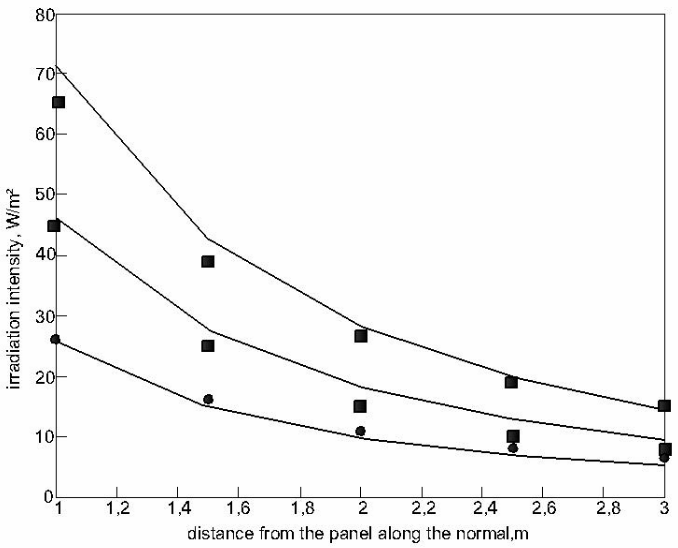

Figure 5. Comparison of the experiment (marker) and calculation at water t = 50,70,90°C [15].

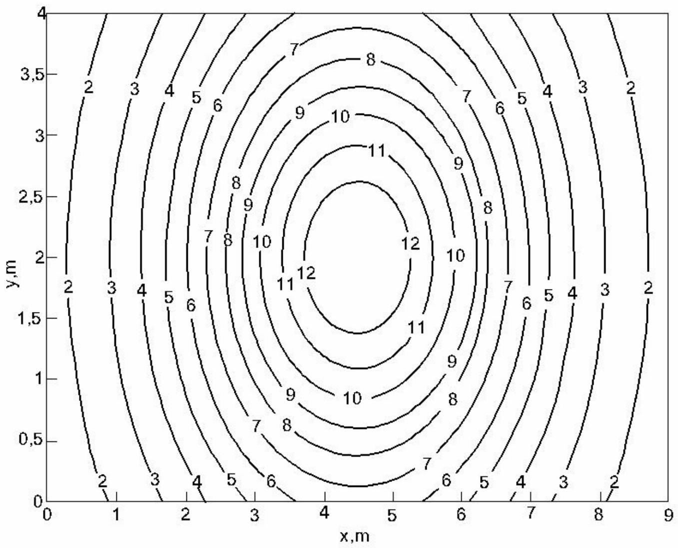

Figure 6. The intensity distribution of the radiation at a distance from the normal under the panel, W/m^2.

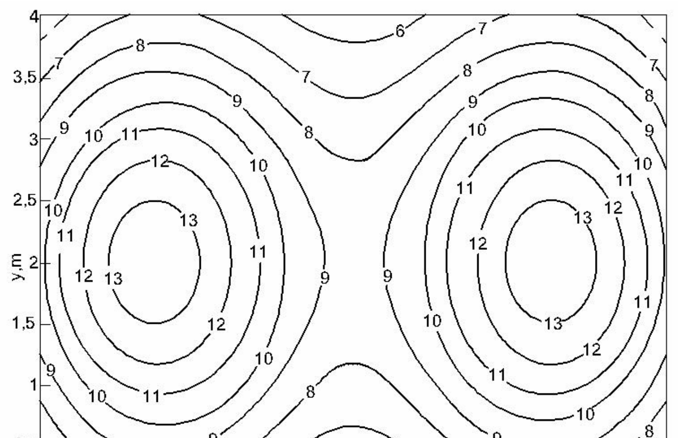

Figure 7. Distribution of irradiance from two panels, W/m^2.

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 Figure 7 radiant flux density decreases from 13 W/m^2 to 9.8 W/m^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. 3×3×5×5=225 experiments. Irradiation intensity E (W/m^2) of the object being under the center of the panel was used as a response. The equation for determining 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, $x_2 = N - 4 x_2 = \frac{t_w - 70}{20}$, where t_w is the water temperature, °C , $x_3 = \frac{\delta - 1{,}5}{0{,}5}$, where δ is thickness of the panel rib, mm, $x_4 = H - 2$, where H is panel height above the object, m.

Total Q_{total} (W) thermal power of the panel was determined by the computing experiment.

The regression equation is:

$$Q_{full} = 508{,}2 + 47{,}5x_1 + 224x_2 + 14{,}2x_3 - 13{,}9x_1^2 + 23{,}8x_1x_2 - 8x_1x_3 + \\ +12{,}1x_2^2 + 7{,}3x_2x_3 - 3{,}9x_3^2 \tag{38}$$

Coefficient of determination $R^2 = 99{,}96\%$

Remainder variance $S^2_{residual} = 25{,}8$ with 15 degrees of freedom.

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

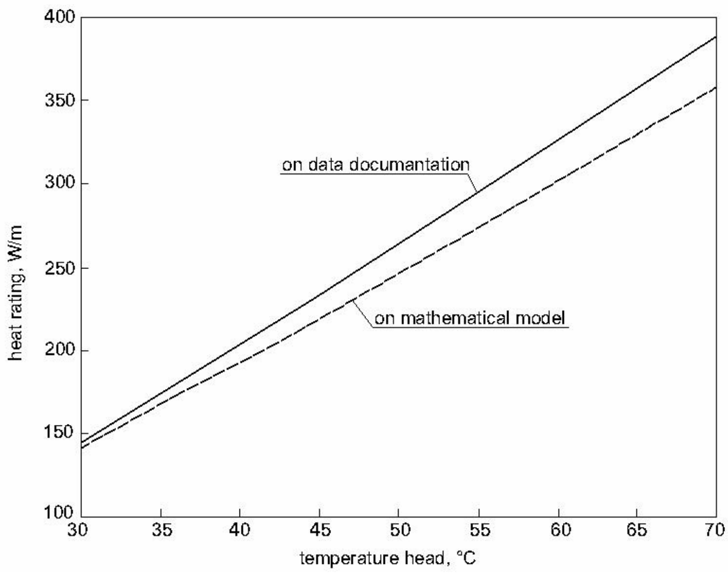

Figure 8. Comparison of thermal Waterstrip series panel (WP2-060) powers.

The calculation model does not take into account the counter convective surface area that is 15% of the total area. This leads to decreased computational results.

Where temperature head is $\Delta t_h = \frac{(t_p + t_0)}{2} - T_W$

Tables 1 and 2 show the results of numerical simulation and optimization of the radiant heating/cooling system parameters, depending on the influencing parameters with minimum entropy production.

Table 1. Parameters of the heating system at a given thermal power of 80000 W.

The number of panel bends/Total width, m/Total panel bend area, m^2	Water temperature in the supply pipeline,°C / Water temperature in the return pipeline, °C	Water flow-rate, kg/s /Pressure loss, kPa	Total thermal power of panel bends, W	Entropy production, W/K
8/2.56/122.88	88.3/70.4	1.07/2.85	79920	45.9
9/2.88/138.24	82.8/64.5	1.04/2.24	79920	42.1
10/3.20/153.6	78.2/59.8	1.03/1.83	79920	39.0
11/3.52/168.96	74.1/56.0	1.05/1.59	79920	36.3
12/3.84/184.32	71.0/52.6	1.03/1.33	79920	33.9
13/4.16/199.68	68.1/49.7	1.03/1.15	79920	31.9
14/4.48/215.04	65.7/47.3	1.03/1.01	79920	30.1
15/4.80/230.4	63.3/45.3	1.05/0.93	79920	28.5

Heating: thermal power 80000 W

Table 2. Parameter calculation results of the cooling systems for a given power of 15000W.

The number of panel bends/Total width, m/Total panel bend area, m^2	Water temperature in the supply pipeline,°C / Water temperature in the return pipeline, °C	Water flow-rate, kg/s /Pressure loss, kPa	Total thermal power of panel bends, W	Entropy production, W/K
9/2.88/138.24	5.3/10.4	0.7/4.1	-14985	2.2
10/3.20/153.6	6.7/11.5	0.75/3.9	-14985	1.96
11/3.52/168.96	7.9/12.4	0.8/3.7	-14985	1.77
12/3.84/184.32	8.9/13.1	0.85/3.6	-14985	1.62
13/4.16/199.68	9.7/13.7	0.9/3.5	-14985	1.49
9/2.88/138.24*	15.2/18.3	1.15/9.76	-14985	1.59
9/2.88/138.24*	16.1/17.5	2.5/43.6	-14985	1.59

* - air temperature in the room 26°C.

The relationship between water temperature in the t' 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:

$$Q(M, t', n, k) = Q_0 \text{ with } \Delta S \rightarrow \min \tag{39}$$

Where n is the number of panels in the bend, k is the number of panel bends. Only those equations for which $t' - t'' \leq 20°C$ stand out of the total number of solutions of the equation (37). Figure 9 shows the values of entropy production heated panels 80 kW. Figure 10 shows the values of entropy production of a cooling panel 150 kW.
Figure 11 shows the level curves for a given thermal power Q_0 = 80000 W when the number of panels is 8,10,12 and 15.

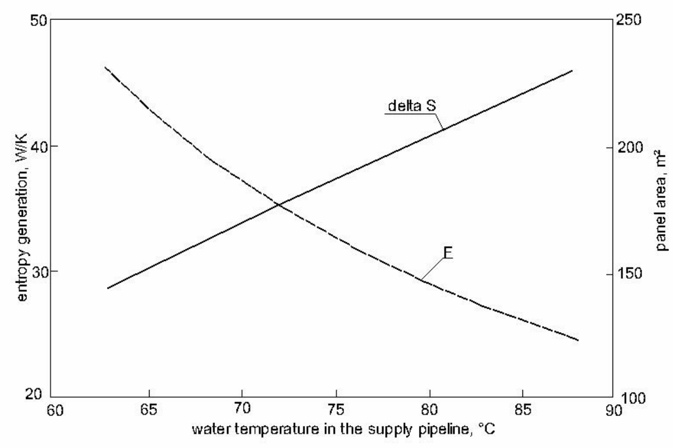

Figure 9. Heating panel area and entropy production v.s. the water temperature (thermal power of 80000 W).

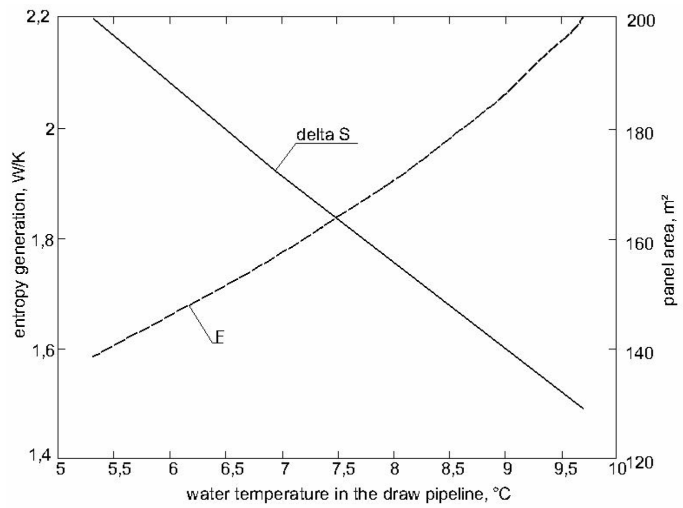

Figure 10.Cooling system area and the entropy production in the function of water temperature (cooling power of 15000 watts).

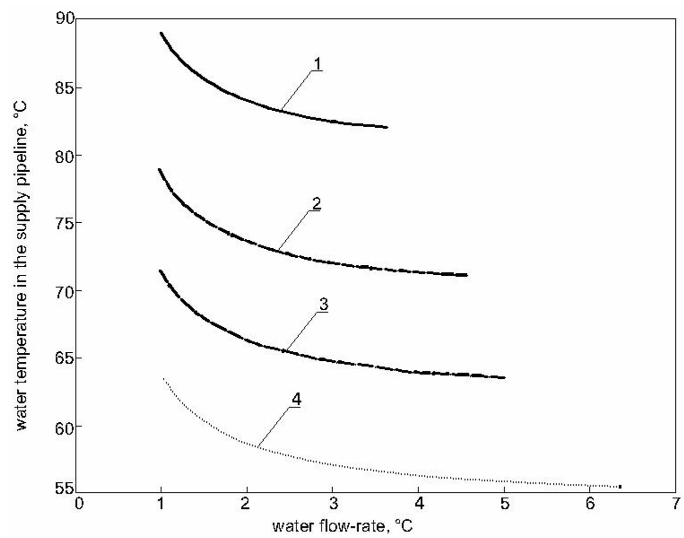

Figure 11. Water temperature in the function of the flow rate for heating systems with a different panel area (heating power 80000 W).

Entropy production *ΔS* along the level line with a given number of bends is almost unchanged, but with an increase in the number of bends,*ΔS* is reduced since it is possible to decrease the temperature *t'*.
Pressure loss in the panel system*ΔP* was calculated for each calculation variant.
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 kg / s are shown for each level line.
Note that if to make the requirements for temperature difference more strict by setting, e.g. $t' - t'' \leq 10°C$, the minimum flow- rate will increase up to 2 kg/s and, hence, the pressure losses will increase dramatically.
Thus, the data shown in Tables 1 and 2 and in Figure 11 allow to define and justify the choice of the area of heating panels, water flow-rate and temperature in the supply pipeline, which provide minimum entropy production.

5. Conclusion

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 influence of non-uniformity of the temperature field of radiant panels, the height of the panel placement and the distance among them, the temperature in the supply pipeline has been shown. The conditions under which the entropy production in the system is minimal are determined.

References

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