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Simulation of Heat Transfer by Water Heating Convectors

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ABSTRACT

Nowadays heat protection of buildings is being improved together with calculation procedure of heat demand. Herewith, classical approach to development of heating system remains unchanged for quite long time, which leads to significant errors. Experimental data available to designers for selection of heating appliances do not provide complete picture for their reasonable selection. This is stipulated by existing empirical coefficients, which are assumed to be conventionally constant at any temperature regimes of operation of heating appliances. This work analyzes the sequence of development of mathematical model of convector, which facilitates more accurate estimation of heating capacity upon quantitative adjustment of heating medium.

Key words: Water heating system, heating appliances, adjustment of heating capacity, finite differences, heat exchange coefficient.

1. INTRODUCTION

Heat protection designs of buildings are constantly improving [1, 2] and prescribed predictions of heat losses become more accurate [3, 4]. However, classical procedures of development of engineering systems, water heating in particular, have not been revised for sufficiently long time, which leads to significant errors in the calculations. This can be exemplified as follows: normative (basic) specific consumption of thermal energy for heating and ventilation in 2018 according to Specifications SP 50.13330 for nine-story residential building is 0.319 W/(m^3·°C). That is, even estimated heat demand for a 60 m^3 room (peculiar for moderate size residential room in new districts of Moscow) equals to 861 W, and if the temperature schedule is 90/65 (peculiar for heating systems with nonmetal pipes), the consumption of heating medium will be 29.6 kg/h. First of all, for such estimated consumption, upon connection of heating appliance using minimum practical size of cross-linked polyethylene pipe Ø16×2 mm, the speed of heating medium will be 0.074 m/s, therefore, the flow regime is in the transient laminar area (the Reynolds number is Re = 2440), hence, the classical prediction of hydraulics of heating system [5] is characterized by significant error. Then, significant decrease in the heating medium consumption decreases the accuracy of heat transfer of heating appliances.

Nominal heat transfer and empiric coefficients used for calculation of actual heat transfer of heating appliances are determined experimentally [6, 7]. As a rule, according to experimental results, the empirical coefficients are averaged in wide consumption range and are provided to designers in fixed form. This significantly simplifies predictions for engineers, however, leads to errors, especially upon analysis of heat regime of buildings.

For instance, it was attempted in [8] to determine actual heat transfer of heating appliance during quantitative adjustment. The actual heat transfer Q_h, W, was determined as follows.

$$Q_h = \frac{Gc_w\left(t_{in} - t_a - \left(\frac{3.6Q_h^{nom}n}{360^p 70^{1+n} c_w (G)^{1-p}} + \frac{1}{(t_{in}-t_a)^n}\right)^{-\frac{1}{n}}\right)}{3.6},$$

where G was the heating medium consumption, kg/h; c_w was the water heat capacity, kJ/(kg·°C); Q_h^{nom} was the nominal thermal capacity of heating appliance, W; t_{in} was the temperature of water supplied to the heating appliance, °C; t_a was the temperature of heating air, °C.

This equation was derived with the coefficients n and p as constants.

The following curve was obtained on the basis of calculations for several types of heating appliances (Fig. 1).

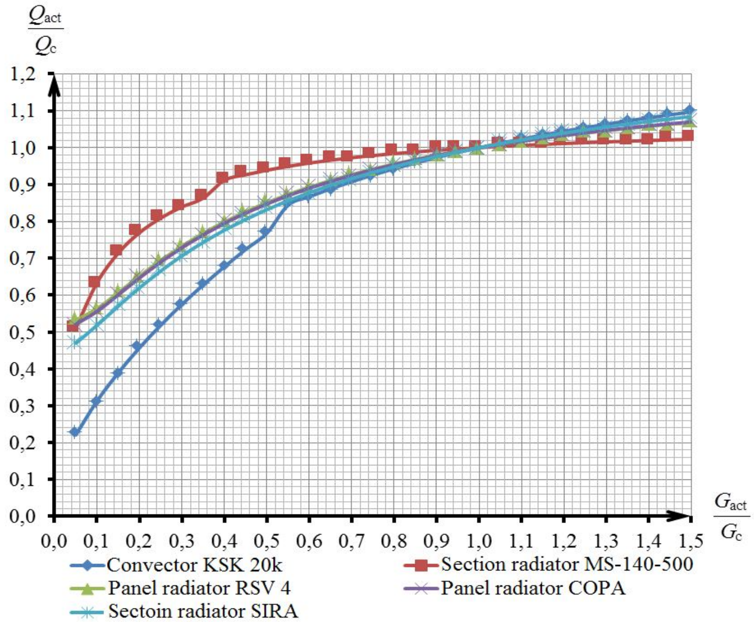


Figure 1: Calculated actual heat transfer of various heating appliances as a function of heating medium consumption

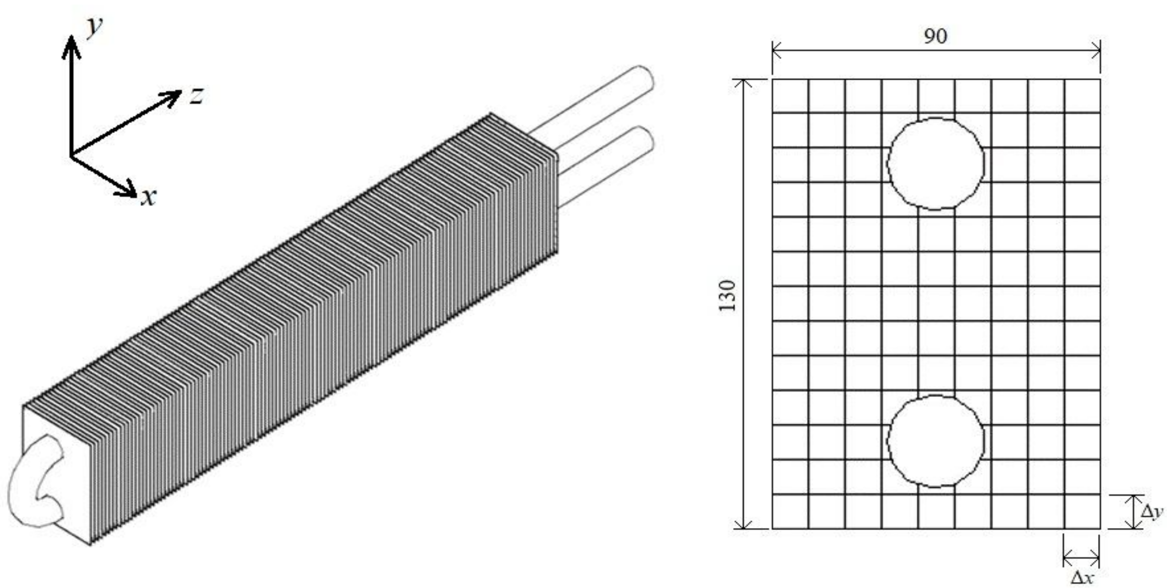
It can be seen that at the heating medium consumption close to 0 kg/h, the curve of heat transfer of the appliances does not reach zero. This evidences that in the range of low consumptions, the proposed procedure is not accurate and should be adjusted.

2. METHODS

This work, aiming at more accurate description of variation of heat transfer of heating appliance upon quantitative adjustment as well as at low estimated consumption of heating medium, proposes mathematical model of convector without casing.

The considered convector is a pipe and plate element based on welded pipes with the sizes of Ø26×2.5 mm with the interaxial distance of 80 mm and with plates with the sizes of 90×130×0.5 mm.

Computational grid was applied onto each plate. Herewith, in the plane of plate surface, the grid increment Δx and Δy was 10 mm, and in the plate thickness, the increment Δz was 0.5 mm, which corresponded to the plate thickness. The flowchart and external view of the appliance are illustrated in Fig. 2.



a) b)

Figure 2: Object of research: a) convector model with coordinate axes; b) application of computational grid on convector plate

The convector heating capacity was predicted according to the flowchart in Fig. 3.

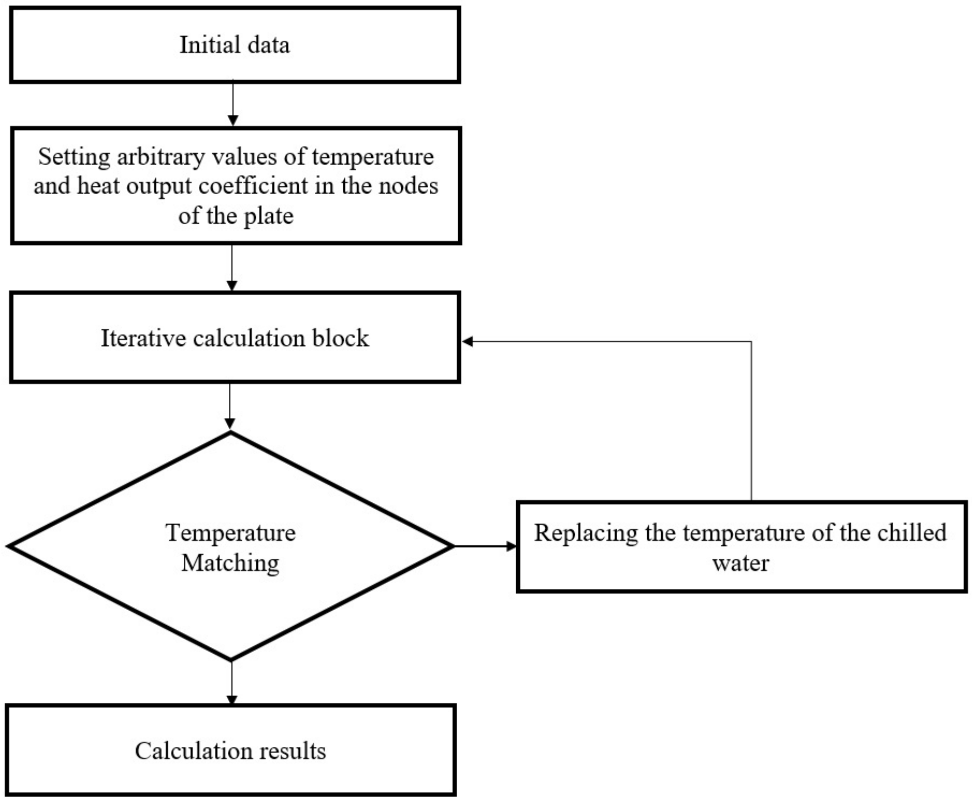


Figure 3: Prediction of convector heating capacity

Let us consider each step separately.

2.1 Initial data

This block presets the consumption of heating medium flowing across the appliance *G*, kg/h, the ambient air temperature t_a, °C, the appliance design and the sizes of the applied grid, the temperature of heating medium supplied to the appliance t_h,°C, and the approximate temperature at output t_c,°C. In addition, the air parameters are used for presetting kinematic viscosity ν_a, m^2/s, and air thermal conductivity, λ_a, W/(m·°C). The convector design also determines the thermal conductivity of plates, λ_m, W/(m·°C), the interplate distance *S*, m, and other geometrical sizes.

2.2 Setting arbitrary values of temperature and heat exchange coefficient in plate nodes

Arbitrary temperature is assigned to each grid node of the appliance. However, the temperature in the nodes, which are the closest to the pipe and inside the pipe, are assumed to be constant equaling to that of heating medium in the considered pipe. One of the simplifications of the model should be mentioned. Since further on we will apply the known equations to determine the coefficient of heat transfer from heated plate, then, conventionally, the convector model is considered as a set of plates with two heat sources in the point where pipes pass through the plate. Unfortunately, equations of the coefficient of heat transfer from such complex design as a set of plates on pipe have not been derived yet and the physics of the process have not been studied in details.

In this regard the heat transfer coefficient is assumed to be the same for all nodes assigned arbitrary at this step, which should be reviewed later.

2.3 Iterative calculations

This block performs the main calculations for determination of temperature distribution over the surface of all convector plates. The temperature in each node of each plate was determined on the basis of stationary temperature field.

$$\frac{\partial}{\partial x}\left[\lambda(x,y,z)\frac{\partial t}{\partial x}\right]+\frac{\partial}{\partial y}\left[\lambda(x,y,z)\frac{\partial t}{\partial y}\right]+\frac{\partial}{\partial z}\left[\lambda(x,y,z)\frac{\partial t}{\partial z}\right]=0, \qquad (1)$$

where $\lambda(x,y,z)$ was the preset thermal conductivity in each node, W/(m·°C); *t* was the temperature in node, °C.

At first let us consider several typical nodes existing on the convector plates.

The first type is comprised of the nodes in the plate corners. Figure 4 illustrates the position of one such node and the arrows show mutual exchange between adjacent nodes and ambient air

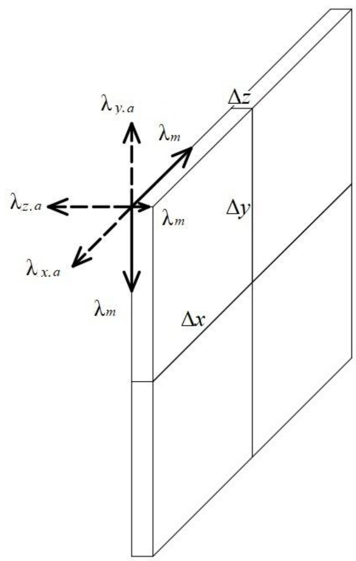


Figure 4: Heat balance of node of the first type

In this case, thermal conductivity of plate material is known exactly, λ_m, W/(m·°C), which was preset in initial data. At this step, the heat exchange between air and plate surface is denoted as $\lambda_{x.a}$, $\lambda_{y.a}$, $\lambda_{z.a}$, W/(m·°C),

While expanding derivatives of the product in Eq. (2), we obtain:

$$\frac{\partial}{\partial x}\left[\lambda(x,y,z)\right]\frac{\partial t}{\partial x}+\lambda(x,y,z)\frac{\partial^2 t}{\partial x^2}+\frac{\partial}{\partial y}\left[\lambda(x,y,z)\right]\frac{\partial t}{\partial y}+\lambda(x,y,z)\frac{\partial^2 t}{\partial y^2}+ +\frac{\partial}{\partial z}\left[\lambda(x,y,z)\right]\frac{\partial t}{\partial z}+\lambda(x,y,z)\frac{\partial^2 t}{\partial z^2}=0. \quad (2)$$

This equation can be solved by substitution of finite differences, then we obtain:

$$\frac{\lambda_{x-\Delta x,y,z}-\lambda_{x,y,z}}{\Delta x}\frac{t_{x-\Delta x,y,z}-t_{x,y,z}}{\Delta x}+\lambda_{x,y,z}\frac{t_{x-\Delta x,y,z}+t_{x+\Delta x,y,z}-2t_{x,y,z}}{\Delta x^2}+\frac{\lambda_{x,y-\Delta y,z}-\lambda_{x,y,z}}{\Delta y}\frac{t_{x,y-\Delta y,z}-t_{x,y,z}}{\Delta y}+ +\lambda_{x,y,z}\frac{t_{x,y-\Delta y,z}+t_{x,y+\Delta y,z}-2t_{x,y,z}}{\Delta y^2}+\frac{\lambda_{x,y,z-\Delta z}-\lambda_{x,y,z}}{\Delta z}\frac{t_{x,y,z-\Delta z}-t_{x,y,z}}{\Delta z}+\lambda_{x,y,z}\frac{t_{x,y,z-\Delta z}+t_{x,y,z+\Delta z}-2t_{x,y,z}}{\Delta z^2}=0 \quad (3)$$

Upon application of grid, it was assumed that Δx = Δy, then, after some transformations:

$$\frac{\lambda_{x-\Delta x,y,z}(t_{x-\Delta x,y,z}-t_{x,y,z})+\lambda_{x,y,z}t_{x+\Delta x,y,z}-2\lambda_{x,y,z}t_{x,y,z}+\lambda_{x,y-\Delta y,z}(t_{x,y-\Delta y,z}-t_{x,y,z})+\lambda_{x,y,z}t_{x,y+\Delta y,z}}{\Delta x^2}+ +\frac{\lambda_{x,y,z-\Delta z}(t_{x,y,z-\Delta z}-t_{x,y,z})+\lambda_{x,y,z}(t_{x,y,z+\Delta z}-t_{x,y,z})}{\Delta z^2}=0 \quad (4)$$

Since calculations are carried out for the node of the first type located on the left side of plate, then tx-Δx,y,z = tx,y-Δy,z = tx,y,z-Δz = ta, because heat exchange occurs between ambient air and plate surface. In addition, the thermal conductivity λx-Δx,y,z, λx,y-Δy,z, λx,y,z-Δz can be expressed in terms of boundary condition [9], for instance, along the *x* axis:

$$-\lambda_{x-\Delta x,y,z}\frac{\partial t}{\partial x}=\alpha_x\left(t_{x,y,z}-t_a\right), \quad (5)$$

where $\lambda_{x-\Delta x,y,z}$ is the thermal conductivity of conventional air layer between ambient air and considered node, W/(m·°C); α_x is the coefficient of convective heat transfer from the considered node to ambient air, W/(m^2·°C).

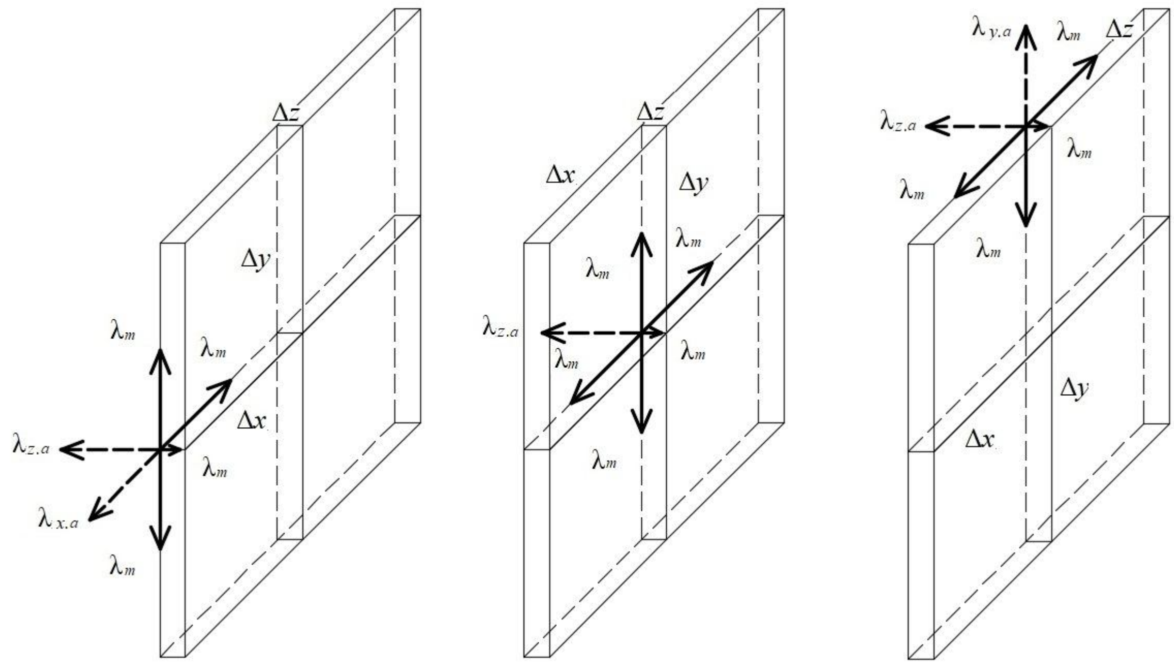
While solving this equation by substitution of finite differences, after certain transformations and under the condition that Δx = Δy, we obtain:

$$\begin{cases}\lambda_{x-\Delta x,y,z}=\alpha_x\Delta x\\ \lambda_{x,y-\Delta y,z}=\alpha_y\Delta x.\\ \lambda_{x,y,z-\Delta z}=\alpha_z\Delta z\end{cases} \quad (6)$$

Substituting Eq. (6) into Eq. (4) and after transformations, it is possible to obtain equation for determination of temperature in the considered plate node:

$$t_{x,y,z}=\frac{t_a\left(\alpha_x\Delta z+\alpha_y\Delta z+\alpha_z\Delta x\right)+\lambda_m\left(t_{x+\Delta x,y,z}\frac{\Delta z}{\Delta x}+t_{x,y+\Delta y,z}\frac{\Delta z}{\Delta x}+t_{x,y,z+\Delta z}\frac{\Delta x}{\Delta z}\right)}{\alpha_x\Delta z+2\frac{\lambda_m}{\Delta x}\Delta z+\alpha_y\Delta z+\alpha_z\Delta x+\frac{\lambda_m}{\Delta z}\Delta x}. \quad (7)$$

Let us consider similarly other types of nodes (Fig. 5).



a) b) c)

Figure 5: Heat balance of node of the second (a), the third (b), and the fourth (c) type.

By analogy with the first type, the equations of node temperature were derived for these nodes.

For the node of the second type (Fig. 5a):

$$t_{x,y,z}=\frac{t_a\left(\alpha_x\Delta z+\alpha_z\Delta x\right)+\lambda_m\left(t_{x+\Delta x,y,z}\frac{\Delta z}{\Delta x}+t_{x,y+\Delta y,z}\frac{\Delta z}{\Delta x}+t_{x,y,z+\Delta z}\frac{\Delta x}{\Delta z}\right)}{\alpha_x\Delta z+3\frac{\lambda_m}{\Delta x}\Delta z+\alpha_z\Delta x+\frac{\lambda_m}{\Delta z}\Delta x}. \quad (8)$$

For the node of the third type (Fig. 5b):

$$t_{x,y,z}=\frac{\alpha_z t_a\Delta x+\lambda_m\left(t_{x-\Delta x,y,z}\frac{\Delta z}{\Delta x}+t_{x+\Delta x,y,z}\frac{\Delta z}{\Delta x}+t_{x,y+\Delta y,z}\frac{\Delta z}{\Delta x}+t_{x,y-\Delta y,z}\frac{\Delta z}{\Delta x}+t_{x,y,z+\Delta z}\frac{\Delta x}{\Delta z}\right)}{4\frac{\lambda_m}{\Delta x}\Delta z+\alpha_z\Delta x+\frac{\lambda_m}{\Delta z}\Delta x}. \quad (9)$$

For the node of the fourth type (Fig. 5c):

$$t_{x,y,z}=\frac{t_a\left(\alpha_y\Delta z+\alpha_z\Delta x\right)+\lambda_m\left(t_{x-\Delta x,y,z}\frac{\Delta z}{\Delta x}+t_{x+\Delta x,y,z}\frac{\Delta z}{\Delta x}+t_{x,y+\Delta y,z}\frac{\Delta z}{\Delta x}+t_{x,y,z+\Delta z}\frac{\Delta x}{\Delta z}\right)}{3\frac{\lambda_m}{\Delta x}\Delta z+\alpha_y\Delta z+\alpha_z\Delta x+\frac{\lambda_m}{\Delta z}\Delta x}. \quad (10)$$

In Eqs. (7–10), the heat transfer coefficient is unknown. Herewith, four types of this coefficient should be highlighted:
- the coefficient of heat transfer between ambient air near external side of the first and the last plate (α_z for the first and the last convector plate);
- the coefficient of heat transfer between the air passing between the convector plates and plate surface (α_z for all plates except for the first and the last convector plates);
- the coefficient of heat transfer between ambient air and horizontal side of plate (α_y for all plates);
- the coefficient of heat transfer between ambient air and vertical side of plate (α_x for all plates).

As mentioned above, the difficulty of determination of these coefficients is that the convector design is not typical for solution to classical problems of thermodynamics. The convector plates are heated non-uniformly, hence, the equations of heat transfer coefficient of uniformly heated plate [10] cannot be applied directly.

Unfortunately, more accurate equations were not obtained for this case. Therefore, this work proposes another simplification. The heat transfer coefficient was determined by successive approximation as for uniformly heated plate with averaged temperature across vertical nodes of the grid.

That is, the average temperature of plate across vertical nodes of the grid was determined as follows:

$$t_{av.y} = \frac{\sum_{i=0}^{k-1} t_{x,y+i\Delta y,z}}{k}, \tag{10}$$

where k was the number of nodes of the plate in vertical direction (from the lower node upward along the y axis).

Then, aiming at determination of the coefficient α_z for the first and the last convector plate as well as of the coefficient α_x for vertically located plates according to [10] the Nusselt, Prandtl, and Grashof, similarity numbers were determined:

$$Gr = \frac{g\,\beta\,\Delta t\, l^3}{\nu^2}, \tag{11}$$

where g was the acceleration gravity, m/s^2; l was the plate height, m; β was the temperature coefficient of bulk air, 1/K, determined as follows:

$$\beta = \frac{1}{(t_a + 273{,}15)}; \tag{12}$$

where Δt was the temperature head, °C, determined as follows:

$$\Delta t = t_{av.y} - t_a; \tag{13}$$

where ν was the kinematic viscosity of air, m^2/s.

In all computations, the Grashof number was in the range of $10^3 < GrPr < 10^9$ (laminar air flow), then the Nusselt similarity number was determined as follows:

$$Nu = 0.695 Gr^{0.25}. \tag{14}$$

Based on the obtained data, the heat transfer coefficient for overall plate height for the nodes above each other (along the y axis) α_z, W/(m^2·°C), and the coefficient α_x, W/(m^2·°C), were calculated as follows:

$$\alpha = \frac{Nu\lambda_a}{l}. \tag{15}$$

The coefficient of heat transfer between ambient air passing between the plates and the plate was calculated as follows [11]:

$$Nu = \left[\frac{576}{\left(Ra_S S/l\right)^2} + \frac{2.873}{\left(Ra_S S/l\right)^{0,5}}\right]^{-0.5}, \tag{16}$$

where S was the distance between the plates equaling to 0.005 m for the considered case; Ra_S was the Rayleigh similarity number determined as follows [11]:

$$Ra_S = \frac{g\,\beta\,\Delta t\, S^3}{\nu^2} Pr. \tag{17}$$

Based on the obtained data, the heat transfer coefficient was determined for overall plate height for the nodes above each other (along the y axis) contacting with air in the interplate space, α_z, W/(m^2·°C) [11]:

$$\alpha_z = \frac{Nu\lambda_a}{S}. \tag{18}$$

The coefficient of heat transfer between ambient air and lower horizontal side of plate $\alpha_{y.b}$, W/(m^2·°C), was determined as follows [12]:

$$Nu = 0.27 Ra_L^{0.25}, \tag{19}$$

and the coefficient of heat transfer between ambient air and lower horizontal side of plate $\alpha_{y.t}$, W/(m^2·°C), was determined as follows [12]:

$$Nu = 0.54 Ra_L^{0.25}, \tag{20}$$

where Ra_L was the Rayleigh similarity number determined as follows [12]:

$$Ra = \frac{g\,\beta\,\Delta t\, l^3}{\nu^2} Pr. \tag{21}$$

After determination of the Nusselt numbers by Eqs. (20) and (21), the heat transfer coefficients were determined by Eq. (15).

It should be mentioned that for the upper and the lower sides, the average temperature of nodes on one x axis was considered:

$$t_{av.x} = \frac{\sum_{i=0}^{z-1} t_{x+i\Delta x,y,z}}{z}, \tag{22}$$

where z was the number of nodes on plates in horizontal direction (along the x axis).

Another boundary condition for these calculations is the temperature of heating medium in pipes, hence, the temperature in the stipulated nodes of each plate. However, only the temperature of heating medium at the input to heating appliance is known on the basis of the initial data. In the first approximation, the temperature in the stipulated nodes (the nearest nodes with respect to pipe circumference) was assumed to be 95°C for the upper pipe and 70°C for the lower pipe.

When all variables in Eqs. (8–10) were known, as well as all boundary conditions were preset in the first approximation, the temperature in each grid nodes was calculated by iterations with subsequent recalculation of auxiliary coefficients of heat transfer. The iterations were repeated until the temperature in each grid node differed from previous iterations not higher than by 0.1ºC.

Then the heat transfer of each plate Q_{pl}, W, was determined as follows. At first the average temperature of each element of plate t_{el},°C, with the sizes of $\Delta x\times\Delta y\times\Delta z$, restricted among eight nodes of applied grid was determined as mean arithmetic value of known temperatures in these nodes.

Herewith, the heat transfer from this element was determined from the sides contacting with air, Q_{el}, W, as follows:

$$Q_{el} = \left(t_{el} - t_a\right)\sum_{i=1}^{m} F_{i.f}\alpha_{i.f}, \qquad (23)$$

where m was the number of sides of the element contacting with air; $F_{i,f}$ was the surface area of heat transferring side, m; $\alpha_{i.f}$ was the coefficient of heat transfer of the side determined as the mean arithmetic value of the coefficients of heat transfer in four nodes restricting this side, W/(m^2·°C).

Then, the amount of thermal capacity released by one plate could be determined as the sum of heat transfer from all elements of plate Q_{pl}, W:

$$Q_{pl} = \sum_{i=1} Q_{i.el}. \qquad (24)$$

Knowing the heat transfer of each plate, it was possible to determine the temperature of water after the plate, $t_{i.h}$,°C, as follows:

$$t_{i.h} = t_{i-1.h} - \frac{3.6Q_{i.pl}}{2Gc_w}, \qquad (25)$$

where i was the number of plate; $t_{i.h}$, was the temperature of water before the plate, °C.

For the lower pipe with colder heating medium, the temperature of water was determined in reverse order from the preset value of 70°C at output and to the last plate using the equation:

$$t_{i.c} = t_{i-1.c} + \frac{3.6Q_{i.pl}}{2Gc_w}, \qquad (26)$$

where i was the number of plate; $t_{i.c}$, was the temperature of water after plate (along the water flow),°C.

Another simplification is made in Eqs. (25) and (26): heating medium passing through the plate in forward and reverse direction during each pass loses half of thermal energy of total heat transfer of the plate. This simplification is not absolutely valid and should be corrected later.

After verification of the temperature of heating medium, all calculations, starting from determination of temperature in the nodes, are repeated until temperature variations in each grid node differ from previous iteration not higher than by 0.1ºC.

2.4 Verification of temperature matching

This block verifies matching of the temperature of heating medium at output from the last plate via the upper pipe to the temperature of heating medium at input to the last plate via the lower pipe. The temperature loss in the 180° pipe bend is neglected.

If the temperatures are not equal (the accuracy is 0.1ºC), then the calculations are repeated with substitution of t_c, and recalculation of the iteration block until the matching of the aforementioned temperatures is achieved.

When the required matching is achieved, the calculations are terminated.

3. RESULTS

The adequacy of predictions was verified using the proposed model. Heating capacity of the considered convector was predicted at the input temperature of heating medium equaling to 95°C, ambient temperature equaling to 20°C, and successive variation of the consumption from 30 kg/h to 10 kg/h with the increment of 5 kg/h, and from 10 kg/h to 2.5 kg/h with the increment of 2.5 kg/h.

The acquired predictions were compared with similar calculations proposed in [8]. Relative variation of convector heat transfer $\bar{Q}$ as a function of relative variation of heating medium consumption $\bar{G}$ are illustrated in Fig. 6.

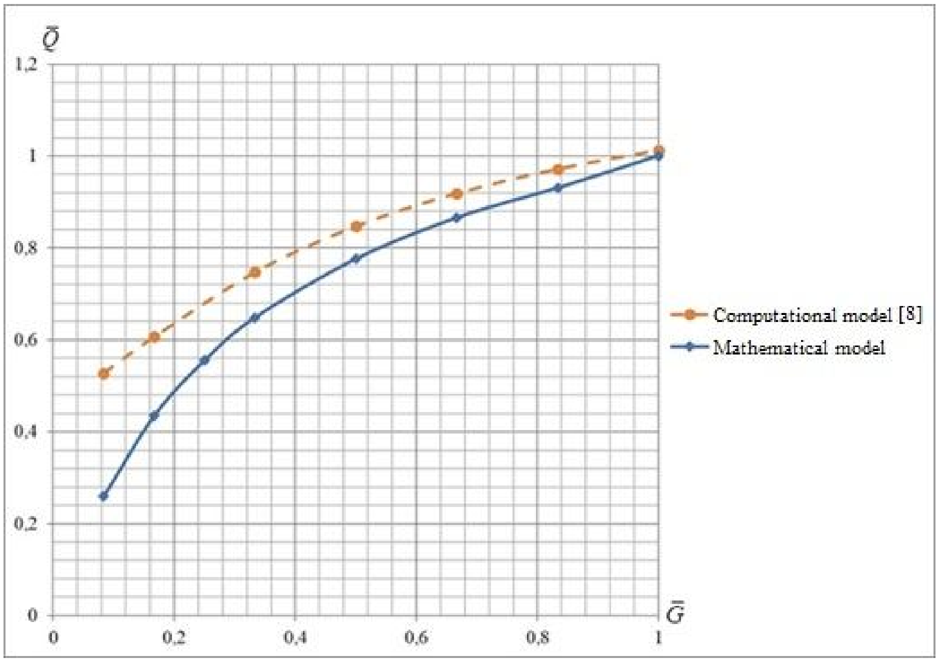


Figure 6: Relative variation of convector heat transfer $\bar{Q}$ as a function of relative variation of heating medium consumption $\bar{G}$

4. DISCUSSION

It can be seen in the plot that, on the one hand, the proposed computational model [8] reflects less accurate the actual variation of convector heat transfer during quantitative adjustment since the curve does not tend to $\bar{Q} = 0$ at $\bar{G} = 0$. This evidences insufficient accuracy of the proposed procedure.

On the other hand, we cannot speak about insufficient accuracy of the mathematical model proposed in this work without appropriate experimental studies on operating appliance.

However, the fact that this plot is better correlated with the data in [13] regarding variation of heat transfer upon qualitative adjustment allows to state that the procedure of development of mathematical model is generally valid.

It would be required to estimate the accuracy of the proposed mathematical model, to consider not only for convective heat transfer from convector plates but also for radiant interchange with room as well as for interaction of this appliance with configuration of heated room and place of its location.

5. CONCLUSION

This work has solved the following objectives of theoretical importance:

1. The mathematical model of convector without casing is proposed based on classical laws of thermodynamics eliminating the use of empirical coefficients erroneously considered as constant.
2. Procedure of determination of temperature distribution across convector plates has been developed.
3. Procedure of determination of heat exchange coefficient is proposed for each side of plate heated heterogeneously and having two heat sources.

Using the proposed model, the actual heat exchange of heating appliance has been determined during quantitative adjustment, which is significantly correlated with known laboratory tests upon determination of heating capacity of heating appliances.

Moreover, the mathematical model can be proposed for improvement of convector design by variation of plate material, location of pipes with respect to the plates, and plate sizes. Therefore, it is possible to determine optimum convector design providing maximum thermal intensity of heating appliance.

REFERENCES

1. A.M. Bekov. **Energoeffektivnye zdaniya. teplomodernizatsiya fasadov zdaniya** [Energy efficient buildings: thermal retrofitting of facades]. Proceedings: Intelligent structural composites for green building: International Conference. Belgorod State technical University, pp. 90-94, 2016.
2. A.A. Magai, N.V. **Dubinin. Sovremennoe steklo svetoprozrachnykh fasadov mnogofunktsional'nykh vysotnykh zdanii** [Modern glass of transparent facades of multifunctional high-rise buildings]. Vestnik MGSU, 3, pp. 36-42, 2010.
3. V.G. Gagarin, A.Yu. Neklyudov. **Uchet teplotekhnicheskikh neodnorodnostei ograzhdenii pri opredelenii teplovoi nagruzki na sistemu otopleniya zdaniya** [Accounting of thermal heterogeneities of

guards upon prediction of thermal load on heating system]. Zhilishchnoe stroitel'stvo, 6, pp. 3-7, 2014.

4. V.G. Gagarin, V.V. Kozlov, V.N. Gagarina, Ch. Chzhou. **Opredelenie potrebleniya energii dlya otopleniya s ispol'zovaniem udel'noi kharakteristiki zdaniya** [Prediction of energy consumption for heating with consideration for specific performances of building]. Proceedings of the International Academic Forum AMO – SPITSE – NESEFF. Affiliate of Moscow Power Engineering Institute in Smolensk, pp. 112-113, 2016.
5. L.M. Makhov. **Otoplenie** [Heating]: Guidebook for higher schools. Moscow: ASV, 2014.
6. V.I. Sasin. **Optimizatsiya kolichestva teplovykh ispytanii otopitel'nykh priborov pri ikh sertifikatsii** [Optimization of number of heat tests of heating appliances during their certification]. AVOK: Ventilyatsiya, otoplenie, konditsionirovanie vozdukha, teplosnabzhenie i stroitel'naya teplofizika, 1, pp. 58-61, 2019.
7. V.I. Sasin, G.A. Bershidskii, T.N. Prokopenko. **Deistvuyushchaya metodika ispytaniya otopitel'nykh priborov - trebuetsya li korrektirovka?** [Existing testing procedure of heating appliances - Any adjustment required?]. AVOK, 4, pp. 76-86, 2007.
8. S.M. Usikov. **Diapazon regulirovaniya teplootdachi otopitel'nykh priborov pri kolichestvennom avtomaticheskom regulirovanii** [Range of adjustment of heat transfer of heating appliances during quantitative automatic adjustment]. Naukovedenie, On-line version, 9(3), pp. 34-42, 2017.
9. E.G. Malyavina, O.D. Samarin**. Stroitel'naya teplofizika i mikroklimat zdanii** [Thermal physics for construction and building microclimate]: Guidebook. Moscow: MISI-MGSU, EBS ASV, 2018.
10. M.A. Mikheev, I.M. Mikheeva. **Osnovy teploperedachi** [Foundations of heat exchange]. Moscow: Energiya, 1977.
11. A. Bar-Cohen, W.M. Rohsenow. **Thermally Optimum Spacing of Vertical Natural Convection Cooled Parallel Plates.** Journal of Heat Transfer, 106, pp. 116-123, 1984. https://doi.org/10.1115/1.3246622
12. T. Fujii, H. Imura. **Natural Convection Heat Transfer from a Plate with Arbitrary Inclination.** International Journal of Heat Mass Transfer, 15, pp. 755-764, 1972. https://doi.org/10.1016/0017-9310(72)90118-4
13. V.V. Pyrkov. **Gidravlicheskoe regulirovanie sistem otopleniya i okhlazhdeniya. Teoriya i praktika** [Hydraulic adjustment of heating and cooling systems, Theory and practice]. Kiev: Taki spravi, 2010.