LATVIAN JOURNAL OF PHYSICS AND TECHNICAL SCIENCES
2003, N 3

NUMERICAL MODELLING OF AIRFLOW AND TEMPERATURE DISTRIBUTION IN HEATED ROOMS

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The authors investigate the distributions of temperature and averaged turbulent airflows in living rooms in a 2D approximation using ANSYS/FLOTRAN software. The distributions are calculated depending on the placement and temperature of heaters, heat transfer coefficients of building structures and ventilation conditions. The authors analyse the influence of these factors on the air circulation and the related heat flows through building structures. The thermal balance of a room and its dependence on various external factors is considered. As parameters determining conditions of comfort there are analysed the airflow velocities and indoor temperatures. It is shown that it is possible to save heat, maintaining at the same time the conditions of thermal comfort in a room.

1. INTRODUCTION

The placement of heaters and their operating temperature essentially influence the distribution of temperature in living rooms. Such a distribution strongly depends on the behaviour and intensity of airflows, which determine the thermal convection, controllable and uncontrollable heat influxes through the openings of ventilation and holes in the walls of a room, as well as heat transfer through its windows, floor and ceiling. Under such conditions the heat consumption for maintaining thermal comfort increases essentially, which, in turn, leads to an increase in the coefficient U (W/m^{2}K) of heat transfer through boundary constructions, especially through the external wall [1]. The convective heat transfer through boundary layers increases also at intense heat flows through the building structures. As a result, the surface coefficient of heat transfer α_I can differ considerably from the standard value for indoor space ($\alpha_I = 8.1$ W/m^{2}K), with variations depending on the properties of a building structure (especially for external walls and windows). It is possible to regulate the intensity of airflows at the vicinity of boundary surfaces by choosing an appropriate placement for heating facilities and ventilation openings, as well as by adding windowsills to the windows. In such a way it is possible to reduce the coefficient of heat transfer and the total heat leakage of the building at a fixed heat transfer coefficient (U value) of the building structure. The optimal arrangement of the heaters and ventilation system allows maintenance of thermal comfort in the room with reduced heat consumption.

The modelling of thermal situations in individual rooms is also important for specifying an integral model of heat balance of buildings [2]. This model serves for determination of the total heat leakage, i.e. of the heat consumption in a building. The differential modelling can be used for specifying:

- the surface coefficient of heat transfer α_I and heat leakage through external building structures at different characteristic situations;

- the heat exchange among building blocks with different temperatures, e.g. between a living room and a hallway (not only the heat exchange by conductivity, but also through air convection – in the case with openings in the walls and the pressure difference).

The velocity of airflows is one of the key parameters of comfort in a room; therefore, it is important to compare this parameter for various physical characteristics of the room. If the airflow velocity reaches 0.5 m/s, this indicates a significant convection and thermal losses. The average temperature in a room is also a significant characteristic and should therefore be considered. When doing the corresponding calculations, one should estimate the amounts of outgoing and incoming heat, taking into account, in the case of convective heat exchange, the amount of convective heat.

2. MODEL

2.1 Description of a modelled room

The calculations have been performed for the room shown in Fig. 1, with parameters given in Tables 2, 3. The room is 3 m high, with a depth of 6 m. Heat transfer coefficient $U_d = \lambda_{eff}/d$ differs for various boundary structures of the building - here λ_{eff} is the effective heat conductivity, and d is the thickness of the structure. Only one of the walls (M3) has a window (M4) and a boundary with the exterior. A heater is placed near this wall. Heat transfer coefficients of building structures are shown in Table 1. It is important that the heat transfer coefficient is very high in variants G1-1 and G1-2 for the outer wall, while in variant G1-3 it corresponds to building requirements established in Latvia [3]. The heat transfer coefficient for the window is higher in variants G1-4 – G4-2 (shows the difference between the wall and the window). The problem is simplified assuming that the considered room is one of many, i.e., the heat flow through the side walls is practically absent. The temperature in the room above and below the floor is chosen from the condition of thermal comfort, T=20 °C, while the temperature behind the frontal wall W3 is chosen essentially lower: T=12 °C (e.g. in the hallway). The exterior temperature, i.e., behind the wall W4 corresponds to winter conditions, T=-10 °C.

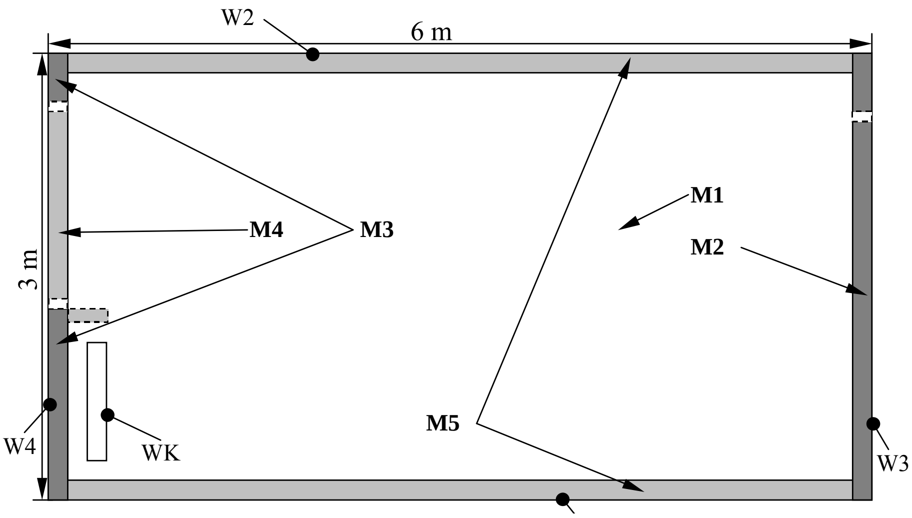


Fig. 1. Layout of building structures in a modelled room

The standard value of the surface heat transfer coefficient for building elements bordering with other rooms is chosen α_I = 8.1 W/m^{2}K, while for the elements bordering with the exterior it is α_I = 23.2 W/m^{2}K. The heat output from the inner surfaces of building structures is determined by modelling. The surface temperature of the convector (WK) is set constant (see WK row of Table 1). The physical properties of the air in the room are shown in Table 3.

Table 1

Heat transfer coefficients of building structures for different variants (see Fig. 1)

Geometry			G1			G2	G3	
Variant			G1-1	G1-2	G1-3	G2-1	G3-1	G3-2
	Symbol	Description						
U, W/m²K	M1	*Air*	*					
	M2	*Wall with another room*	1.2	1.2	1.2	1.2	1.2	1.2
	M3	*Exterior wall*	1.5	1.5	0.33	0.33	0.33	0.33
	M4	*Exterior window*	2.5	2.5	2.5	6.0	6.0	6.0
	M5	*Floor and ceiling*	0.8	0.8	0.8	0.8	0.8	0.8
T, °C	WK	*Convector*	60	40		50		

* – see Table 3.

Table 2

Outer surface heat transfer coefficients of building structures (see Fig. 1)

Parameter	W1, W2	W3	W4
	Floor and ceiling	Walls to another room	Exterior wall
Heat exchange coefficient at surfaces α, W/m²K	8.1	8.1	8.1
Bulk temperature T_∞, °C	20	12	–10

Table 3

Properties of material M1 (air)

Parameter	Symbol	Value	Units
Density	ρ	1.11	kg/m³
Dynamic viscosity	μ	$2\cdot10^{-5}$	kg/(m·s)
Thermal expansion	α	0.0034	K^{-1}
Heat conduction	λ	0.02454	W/(m·K)
Heat capacity	c_p	1007	Ws/(kg·K)

2.2 Physical model

Airflow in a room depends both on the convection created by the temperature difference and on the air exchange between openings in building structures (slots, openings, natural and artificial ventilation, etc.).

To determine airflow characteristics, the following dimensionless numbers are employed [4]:

a) the Reynolds number, $\mathrm{Re} = v_0 L/\nu$ (v_0 is the characteristic velocity, L is the characteristic size of a room and ν is the cinematic viscosity), which shows the type of flow (laminar or turbulent). Simple calculations (with a velocity of airflow that can exceed 0.1 m/s under typical conditions) show that in our case the Reynolds number is approx. 10^4, which corresponds to the turbulent character of airflow;

b) the Peclet number, $\mathrm{Pe} = vL/a$ (a is the temperature conductivity), characterising the proportion of convection and molecular conduction by heat transfer in gases or liquids. In our model it is approx. 10^5, which means that in the heat exchange there dominates convection;

c) the Prandtl number, $\mathrm{Pr} = \nu/a$, which characterises the proportion of molecular friction and heat exchange. In the case of turbulent airflow the turbulent number (Pr_T) should be used. In our calculations $\mathrm{Pr}_T = 0.85$ was used.

To describe the quasi-stationary behaviour of temperature and averaged turbulent flows, traditional differential equations are employed [5]:

- Navier-Stokes' equation;
- average continuity equation;
- equations for specific turbulence energy k and dissipation rate of this energy ε;
- energy conservation equation.

The turbulent viscosity ν_T is calculated by using the k-ε turbulence model:

$$\nu_T = c_\nu k^2/\varepsilon,$$

where $c_\nu = 0.09$ is an empirical constant, under traditional boundary conditions [6].

The temperature distribution should be determined both inside the room and in the building structures, because the convection type boundary conditions $\lambda \cdot \partial T/\partial n = \alpha(T - T_\infty)$ are set for the temperature at the outer boundary of these structures (see Table 2). The solar heat radiation through the window is ignored in order to simplify the model. In the case of a model with openings there is airflow from/to the outside thanks to pressure boundary conditions at openings. For all surfaces, except openings, non-slip boundary conditions ($v = 0$) are used.

2.3 Numerical realisation

In the modelling, the software package ANSYS/FLOTRAN was applied to obtain both the stationary temperature distribution and averaged airflows in the approximation of the k–ε turbulence model. The discretisation was performed with triangular elements of varying size. The size of elements in the centre of a room did not exceed 5 cm. This size was 1–2 mm in the vicinity of solid objects (especially near the convector and the openings in the walls). Therefore, the total number of elements in the 2D area depending on its geometry and the modelling variant was $\approx 3 \cdot 10^5$. An example of triangulation near the convector is shown in Fig. 2.

The boundary conditions of the third type (convection from walls to the outside and to other rooms) for temperature on all surfaces of building structures and the low viscosity of air essentially worsen the convergence of iteration process. The time required for calculations with a 600MHz computer for each variant is 40–60 hours, and 20–40 hours if the problems were solved with a 1.5 GHz computer. The difference between the heat provided by the convector and the heat leakage from the outer surfaces of the building decreases below 10% during each simulation. Comparison of the total heat demands and differences will be given in chapter 3.

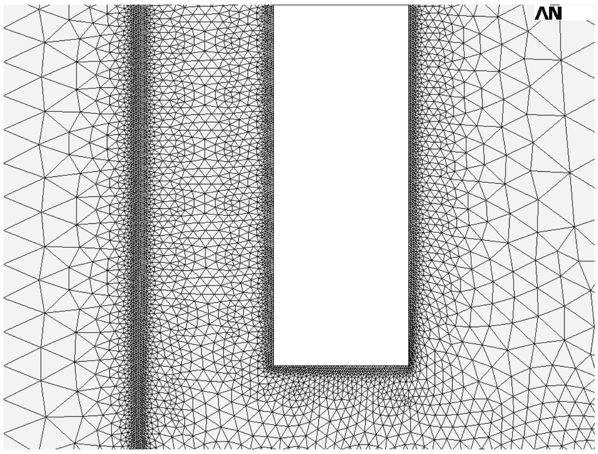


Fig. 2. Discretisation in the finite elements' method in the modelled corner of a room

3. RESULTS OF MODELLING: PHYSICAL FIELDS

Further, we consider some variants of modelling, which evidently show the impact of particular changes in the geometry and heat exchange conditions on the physical fields and characteristic values of temperature and airflows

3.1 Temperature variations of the convector (variants G1-1 and G1-2)

Having assumed a simple geometry for the room (the model without a windowsill – G1-x) we can estimate if there is a noticeable influence of the convector's temperature on the general temperature distribution and airflow circulation (see boundary conditions and properties of materials in Tables 1 and 2). We will compare the air motion in the room for variants with convector temperatures 60°C and 40°C (variants G1-1 and G1-2, respectively), displaying the velocity modulus (Fig. 3*a* and 3*b*). In the case of higher temperature on the surface of the convector, the flow of warm air deviates to the window, i.e. to the greater temperature gradient (since the temperature of air in variant G1-1 is higher than in G1-2). In the case of lower temperature (and a smaller gradient of temperature), the flow of air deviates more in the centre of the room, therefore, on the window surface the cooled air will move more downwards. Motion of this type is caused by thermal expansion of the air – this cause changes of density of air depending on temperature. Because of it there is a movement of air.

At the upper surface of the convector, depending on its temperature, the airflow changes direction (Fig. 4*a* and 4*b*), which explains the difference in temperatures between the airflow and the boundary structure. In the case of high temperature of the convector, the air between this convector and the wall is ascending under the influence of hot air (Fig. 4*a*). In the case of low temperature of the convector, cold air is moving as shown in Fig. 4*b*, cooling down near the window in the downward direction, and thus there is no influence of the flow of warm air any more. The flow of air deviates the external wall in the upper part of the convector, and the distribution of temperature varies in other parts of the room.

The characteristic temperature of the room on its average line (at 1,5 m height from floor) in the first and second case reaches 26 °C and 22 °C, respectively. It is obvious that the temperature of 26 °C is too high for living rooms, therefore, in further calculations we assume the convector temperature of 50 °C (the minimum value is not used, because of possible increase in heat losses in some aspects in further modelling variants). The comparison of thermal losses through various building constructions and calculations of the total thermal balance are given in the heat balance results (chapter 3).

a)

b)

c)

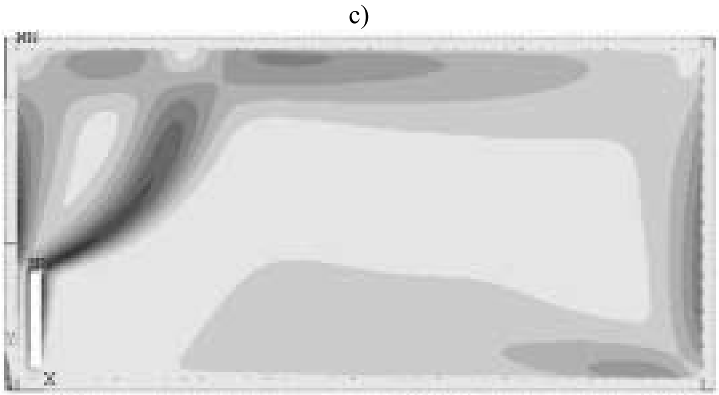


Fig. 3. Stationary distribution of the velocity modulus in the modelled room.
Images *a, b, c* correspond to variants G1-1, G1-2, G1-3.

Another important factor when analysing the climatic conditions ensuring the comfort in a room is the velocity of airflows in this room. The velocity characterises the convective heat exchange by air masses and shows the intensity of heat losses by convection and the rate of air mixing. At a high temperature of the convector, the characteristic air velocity in the room reaches 0.5 m/s, but at lower temperature it is reduced by twice.

To estimate the influence of physical characteristics of building structures on the heat exchange processes in a room, we will consider a variant with changed heat conductivity of a wall (variant G1-3) – in this case it is reduced to meet the requirements of LBN 002-01 [3] complying with real situations occurring in new buildings.

3.2. Heat conductivity variations in the building structures (variant G1-3)

In the above considered variants the accepted thermal resistance of structural elements is low (Table 1), which means that there are considerable heat losses through them. We will analyse the impact of a wall with higher thermal resistance R ($R = 1/U$) (according to LBN 002-01 [3]) on physical processes in the room. In the modelling software the heat conductivity λ ($\lambda = U \cdot d$) is taken, which can be set using U and d values. In this variant, $U = 0.35$ W/(m^2·K) is assumed, which corresponds to λ=0,033 W/(m·K) if the wall thickness is 10 cm and on the surfaces of building structures standard values of heat transfer coefficient are chosen (on the inner surface 8.1 W/(m^2·K) and on the outer – 23.2 W/(m^2·K)).

Distribution of airflows (absolute values) are shown in Fig. 3*c*. As apparent from the figure, in the case with improved insulating properties of a wall the intensity of airflow between the convector and the wall decreases, and thermal losses in this zone also decrease. When thermal resistance R of the wall increases, the velocity of the airflow near the window, which has a low thermal resistance, is also increasing. In this case, the flow of warm air moves more aside the centre of the room (deeper into it). The velocity vectors of airflow near the upper part of a convector are shown in Fig. 4*c*. In this figure there is seen the tendency of reduction in the absolute velocity values in the gap between a convector and a wall; also, the change of airflow direction towards the centre of the room is clearly visible.

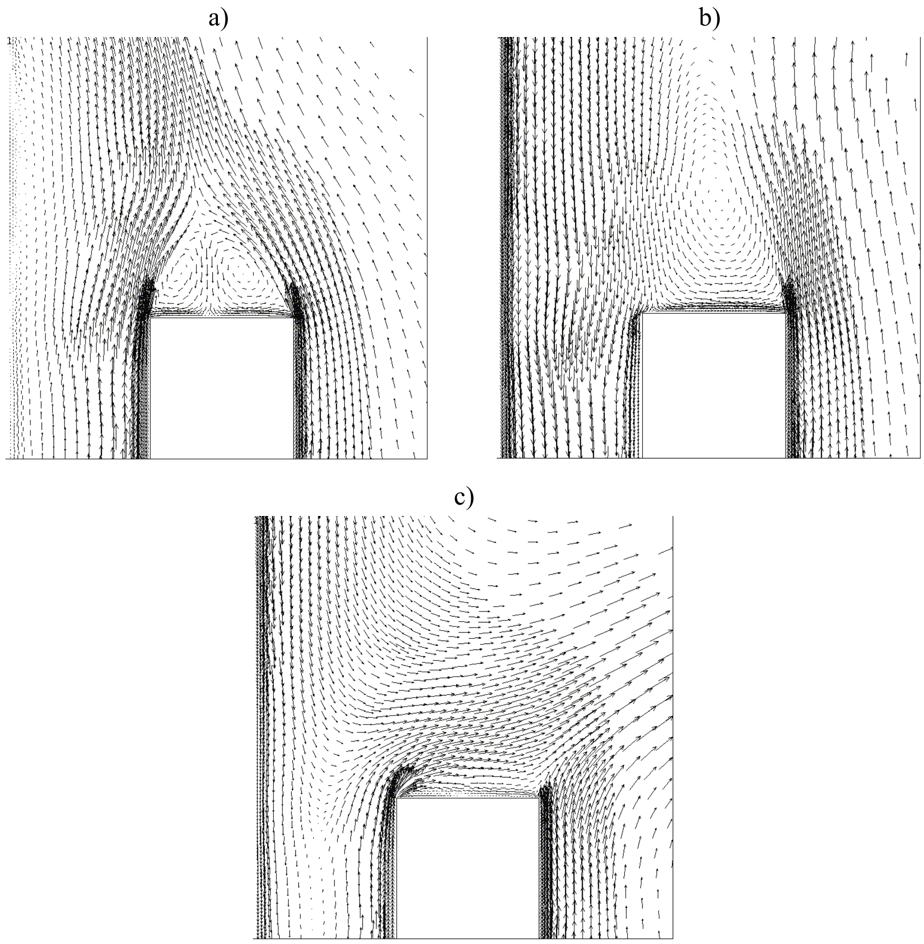


Fig. 4. Airflow vectors above the convector. Images *a, b, c* correspond to variants G1-1, G1-2, G1-3

At the decreased heat conductivity of an outer wall, the thermal losses through this wall also decrease. The lower the air velocity, the lower the intensity of convective heat exchange at surfaces. The characteristic temperature in the centre of a room at the same convector surface temperature has increased up to 23 °C, and the maximum air circulation has moved from the window to the centre of the room. The character of circulation of airflows does not change in the centre of the room, however, it changes in the gap between the convector and the wall, where the velocity begins to reduce and the flows on the opposite surfaces are of reverse directions (Fig. 4*c*). Such effects are attributed to the reduced temperature gradient.

From the viewpoint of geometry, the considered model is too simple and almost never occurs in practice. Since heaters are usually installed in niches (or beneath windowsills) but not in open places, the direct flow of air to the window deflects, which reduces thermal losses. Therefore, in the further models we shall consider a geometry with a windowsill (Fig. 1.) and the increased heat conductivity of the window (up to 6 $W/(m^2 \cdot K)$). Thus, it will be possible to better estimate the contribution of a window to the distribution of airflows. In this connection we will slightly increase the temperature of the convector (up to 50°C), so as not to disturb the total heat balance.

3.3. Changes in the geometry (variant G2-1)

As apparent from the considered modelling variants, the flow of air from a convector is directed upwards near a window, thus promoting intensive heat exchange with cold air near the window with the ensuing considerable thermal losses. One of the ways to avoid this consists in increasing the thermal resistance of the wall and the window, thus reducing the temperature gradient and thermal losses. Another way is to change the geometry in such a manner that the airflow was mechanically reflected from external boundary structures. For example, it can be realised if we employ windowsills (Fig. 1).

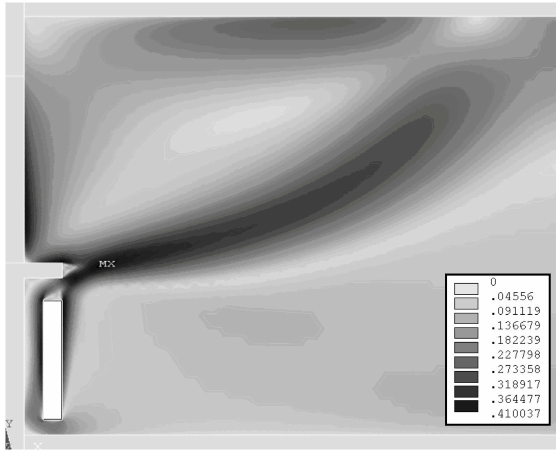
Distribution of airflows and their behaviour in subsequent variants G1-1 – G1-3 essentially differ from those considered earlier. The windowsill now serves as a mechanical barrier for the airflow and provides flow of warm air in the centre of the room (Fig. 5*a*). A high velocity of air is observed near the surface of the window, which is attributed to the high horizontal gradient of temperature $\partial T / \partial x$ in this place, as the window is the most powerful heat conducting element in this model. At the opposite wall of the room (at the wall adjacent to the unheated corridor) the air cools down, therefore the high velocity of air in this region is related to the airflows directed downwards.

Considering the motion of the air at the upper part of the convector, we can see that it is directed to the centre of the room, and on the upper surface there arise two secondary vortices formed by the movement of warm air on the side surfaces of the convector. Similar vortices occur also at the end of a windowsill, however they are formed with flows of warm (the bottom surface of the windowsill) and cold (the top surface of the windowsill) air (Fig. 6*a*). Apparently, small vortices arise also under the windowsill, where inactive masses of warm air are present. Distribution of temperature in the area of the windowsill is shown in Fig. 7*a*.

If the heat transfer coefficient of the window is high and airflow is deflected from it, e.g., by windowsill, then the temperature of the inner surface of the window can considerably decrease (from 19 to 16 °C). As a result, the temperature difference between this surface and the air in the room increases thus leading to a risk of condensate formation.

As the flow of warm air deviates from the window, where the greatest thermal losses are observed, the average temperature in the room also increases. The temperature in this variant is 26 °C (against 23 °C in the case without a windowsill). Thus, the general thermal requirement is also reduced – see the analysis in Part 2. The mentioned temperature cannot be considered comfortable for people, however in real living rooms it is lower owing to convective thermal losses by air exchange.

a)



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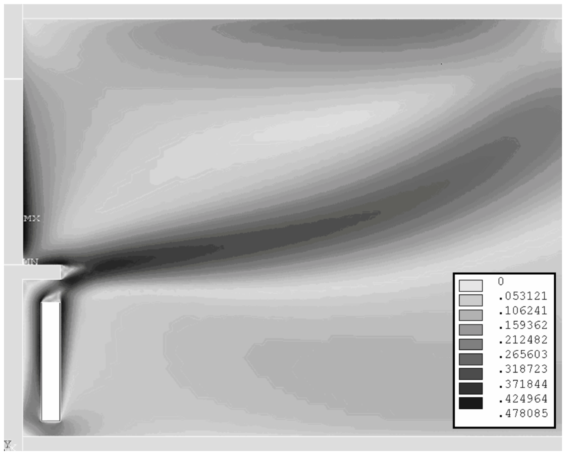

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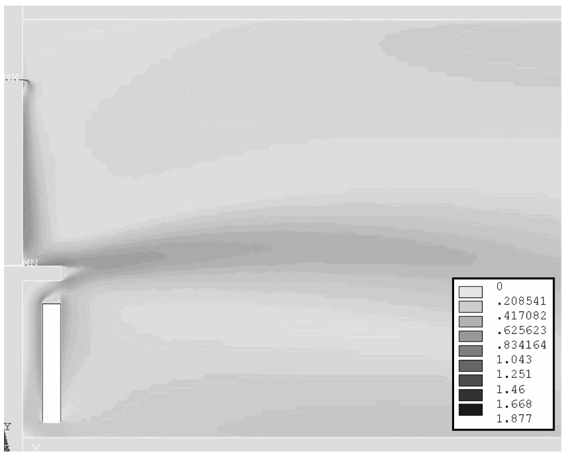
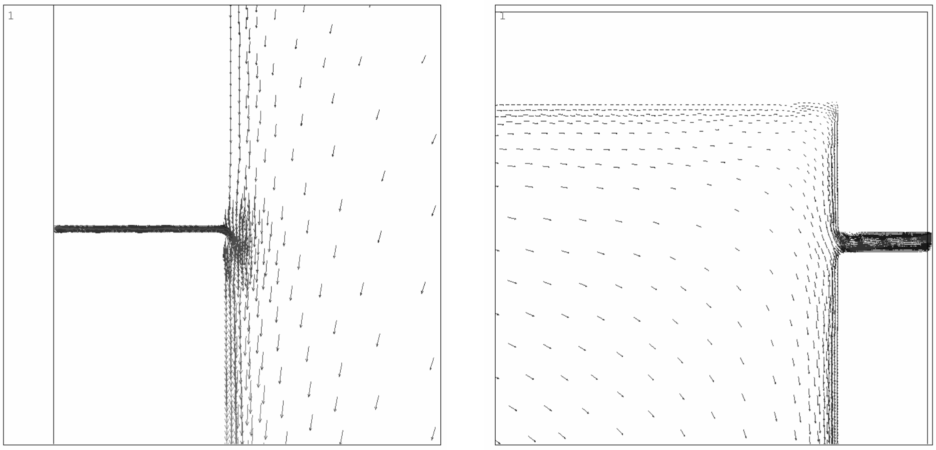


Fig. 5. Distribution of the velocity modulus. Images *a, b, c* correspond to variants G2-1, G3-1, G3-2

The maximum air velocity is not changed in comparison with variants without a windowsill and remains at a level of 0.3 m/s.

a)



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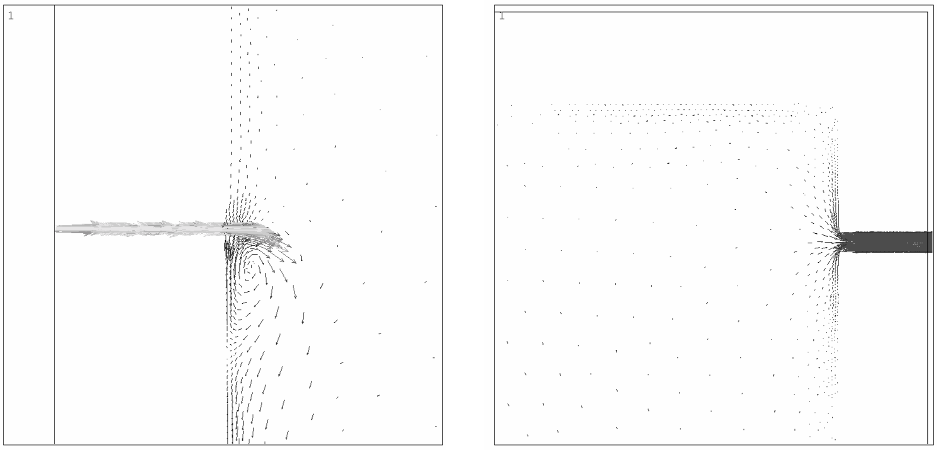


Fig. 6. Vectors of airflow in openings of the room.
Images *a, b* correspond to variants G3-1, G3-2

In all previous variants, a closed room without openings was considered. However, in reality, rooms are not isolated from environment and the airflows compensate the consumed oxygen that is necessary for a person. Therefore, the following designed model of the room accounts for openings in the boundary structures which provide for air circulation. Thus, there are also additional thermal losses that have not been taken into account in the above considered variants.

3.4. Inclusion of convective heat exchange (variants G3-1 and G3-2)

To take into account air exchange with the environment, a room with three openings is considered. They provide air circulation and contact with the outside (Fig. 1). As it usually is, on the perimeter of a window there are cracks, and in the model the location of two openings is between the window and the wall. The sizes of openings should be small to correspond to the real room situation. In the model, 5 mm openings are employed. In the opposite wall one more opening is modelled which corresponds to a 2.5 cm wide ventilation system's opening.

Two air exchange variants with different boundary conditions on the openings are modelled:

- with zero (0 Pa) pressure difference, which is the case when between the opposite walls of the room there is no pressure difference. Also in this case airflows through openings could be observed owing to the nonuniformity of temperature and pressure fields in the rooms;
- with 2 Pa pressure difference. A case is considered when there is under-pressure in comparison with the outdoor air where the pressure is higher than that on the opposite wall.

On the lines in openings that delimit the room from the environment, the pressure values are taken 0 and 0 or 0 and 2 Pa, respectively, as well as the temperature equal to temperature T_∞ assumed for the corresponding surface under boundary conditions of the third type. To allow flowing of air through openings, non-slip conditions are not used on these lines.

Velocity distribution in the room in both variants – near the window and the convector – is shown in Figs. 5*b* and 5*c*, respectively. The inflow of the cold air on both sides of the window as well the outflow of the warm air through the ventilation opening in the opposite part of the room radically changes the airflow distribution in comparison with the closed room model.

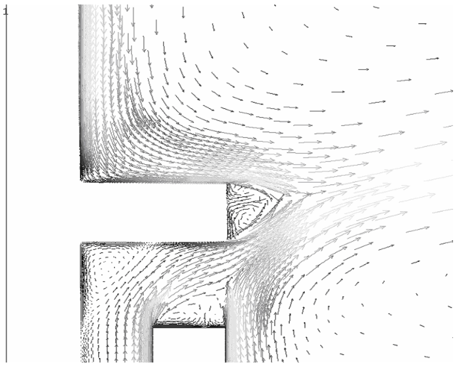
The velocity field changes are especially clearly expressed at the pressure difference ΔP=2 Pa, which actively enhances the inflow of cold air under the over-pressure (in Fig. 5*c* higher airflow velocities are by the window, therefore the velocity field in the room is more uniform on the chosen scale). Now the flows of warm air are changed more to the centre of the room and are determined not only by the windowsill as a mechanical barrier to the flow, but also by the inflow of cold air. The incoming air through the upper opening moves down (because of its higher density) and intensifies this effect.

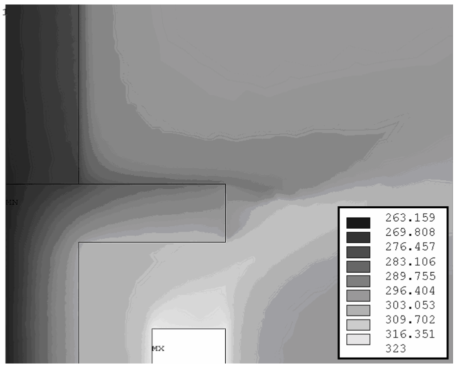
The inflow through openings in the upper part of the window and the outflow through the ventilation opening in both the modelling variants are shown in Figs. 6*a* and 6*b*. In the case when no pressure difference is assumed, the flow through openings is relatively weak (up to 15 cm/s), however a pressure difference of 2 Pa significantly increases the airflow velocity, which reaches even 1.5 m/s and greater in the external wall openings. The air circulation at the top of the convector and near the windowsill is shown in Fig. 7, and the temperature field – in Fig. 8. As is seen, the character of the circulation over the convector and on the windowsill has not changed qualitatively, however the absolute values for the flow through openings have increased, therefore the share of heat losses by convection in the heat balance is radically increasing, which will be shown in the heat balance calculation.

As the convective heat losses increase, the characteristic room temperature decreases. This temperature with zero pressure difference remains at the 25–26 °C level, however it noticeably (explicitly) decreases to 15°C when a 2 Pa pressure difference is taken between the opposite walls. As the convector's surface temperature in these cases is the same as in the previous calculation variants (see Table 1), but the room temperature has significantly decreased, the proportion of the convective heat losses in the heat balance can be clearly seen (see the next chapter).

If the convective air exchange modelling is included in the calculations, the convergence process becomes unmonotonic and unstable; in this case it is difficult to introduce the convergence criteria. In the second part of the publication the thermal comfort conditions are considered, the heat balance is analysed, and the outflow and inflow air quantities are compared.

a) a)





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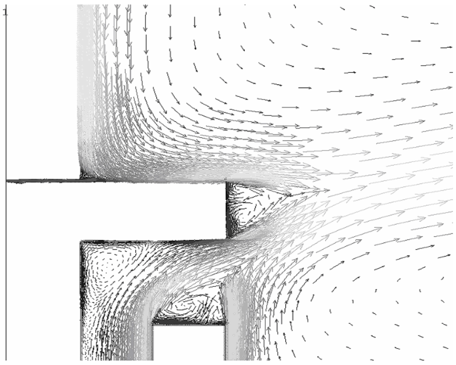

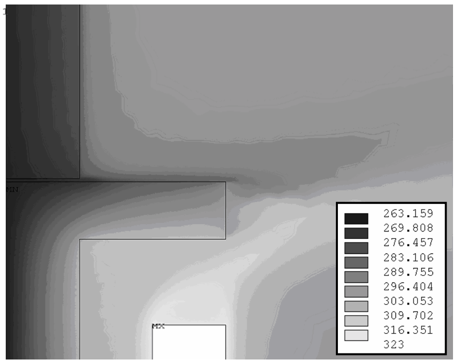

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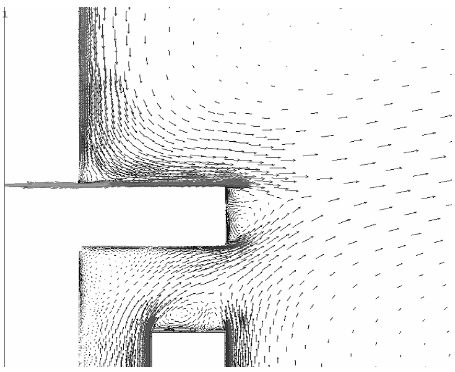

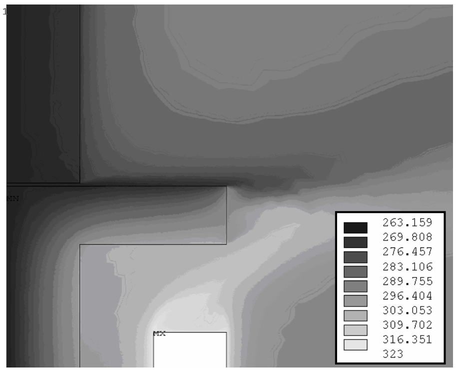

Fig. 7 (left). Vectors of airflow above the convector.
Images *a, b, c* correspond to variants G2-1, G3-1, G3-2
Fig. 8 (right). Temperature distribution.
Images *a, b, c* correspond to variants G2-1, G3-1, G3-2

4. RESULTS OF MODELLING: THE HEAT BALANCE

The control of calculation results can also be performed by the heat balance of a room. Here, the heat flowing to a room from the convector is compared with the heat that flows out of this room by heat conduction. This comparison is valid for the variant without openings, which allows for air inflow or outflow. The corresponding control variants of the heat balance model with convective exchange of air will be considered later. Heat balance calculations enabled us to analyse the spatial distribution and type of heat losses.

The comparison of results will be performed considering a heat flow from the convector surface (its temperature and the temperatures of air in another rooms being given). The heat flow from the outer surfaces of walls, floors and ceiling can be calculated further if we know, from the modelling, the temperatures of these surfaces, as well as temperatures outside the thermal boundary layer and heat transfer coefficients of the surfaces. If there are no other kinds of heat transfer through boundary structures, these quantities in the stationary state should be equal to each other. Comparisons of heat flow data in various variants of modelling are summarised in Table 4. The values of heat flow to the inner surfaces of building structures are calculated in addition to those of heat flow at outer surfaces. The graphical interpretation of the results without convective losses is shown in Fig. 9.

Table 4

Total heat conduction losses without air convection

			G1-3	G2-1	G3-1	G3-2
	Modelling variant		G1-3	G2-1	G3-1	G3-2
Heat amount (W)	Convector		196.9	155.0	150.4	253.3
	Outer surfaces	W1	1.6	5.2	7.7	–15.7
		W2	11.9	2.9	10.7	–16.4
		W3	27.0	28.4	29.8	12.0
		W4	135.6	117.9	105.0	59.4
	Outer surfaces - total		176.1	154.5	153.1	39.3
Difference (%)	Convector and outer surfaces		11.4	7.0	0.3	849
Characteristic average velocity in the centre (m/s)			0.5	0.3	0.4	0.5
Characteristic average temperature (°C)			22	26	25	15

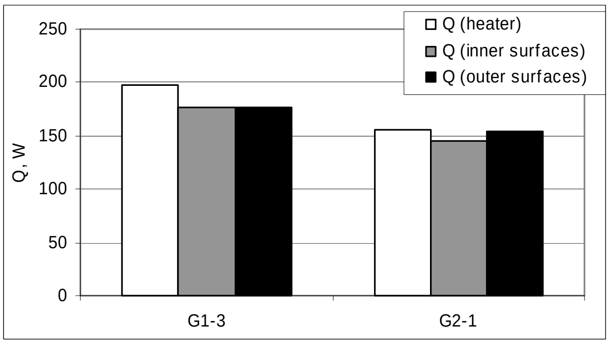

Fig. 9. Comparison of the conduction heat exchange for a confined room without convective losses

In different modelled conditions, indoor airflows change directions and velocities, which is essential for heat exchange in walls and for the total heat amount consumed. The graphical estimation for all boundary structures in different variants is shown in Fig. 10. As one can see, the windowsill in variant G2-1 decreases heat exchange through exterior wall and slightly decreases through wall W1 (Fig. 1). Inclusion of ventilation openings changes the total heat by conduction – now heat losses are mainly by convection. In the modelling variant G3-2 (with 2 Pa pressure difference) the heat flux in walls W1 and W2 change the direction, and now a proportion of heat is going into the room from other rooms. This is associated with temperature difference; the temperature in the modelled room is lower than that set in neighbouring rooms. In the case when convective heat is significant, the temperature in the room is much lower (15 °C) and the total heat losses by conduction are radically decreased (G3-1 in Fig. 10. and Fig.11.).

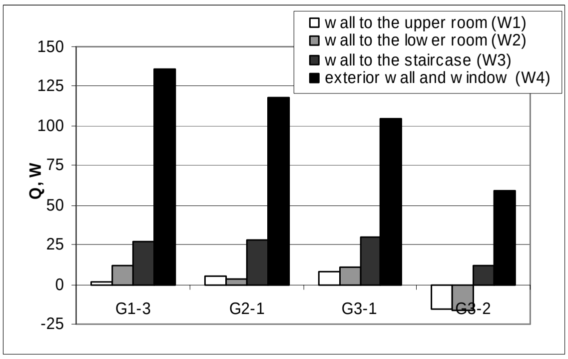

Fig. 10. Comparison of the conduction heat exchange for all boundary structures

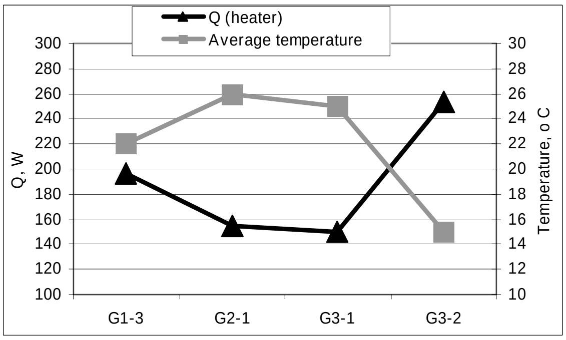

Fig. 11. Characteristic average temperature and velocity in the middle of a room

As can be seen in these figures, the balance of heat fluxes is well satisfied in all the variants without openings. Therefore, we can conclude that the calculations have been done correctly and the obtained results are accurate. However, if the models with openings in a building structure are considered, the heat flow control of the surfaces can be insufficient, which is proved by the calculation results for these variants shown in Table 4. The table and the figures show that the heat that outflows by convection at a zero pressure difference (0 Pa) is less than in the case with a pressure difference of 2 Pa, when the heat amount that outflows by convection is much greater than the heat losses by conduction. Such conclusions can be drawn since the heat delivered by the convector is much greater than that lost through boundary structures by heat conduction. The heat from the convector surface at fixed temperatures in various variants of calculation is related to the temperature regime in its vicinity, when a higher heat amount is required to maintain the temperatures needed.

To include heat losses by convection it is necessary to control the heat fluxes through openings in the building structures. First, the airflows directed inwards and outwards must be equal – the process is stationary and other air inflows and outflows are absent, i.e., conservation of the mass must be satisfied (see also [7]). Second, the heat amount that is transferred by these airflows must be equal to the difference between the heat delivered by the convector and the heat that outflows through building structures by heat conduction. In such a way, the total heat arriving at the convector is equal to the heat losses in the room – both by convection and by conduction.

The airflow (the volume of air per time) through the existing openings in building structures of a room can be determined as $q = \int \vec{v} d\vec{S}$. In this case, the space is discretised. Therefore, we will apply a simplified numerical integration using trapezoidal integration method:

$$q = \sum \frac{v_i + v_{i+1}}{2} \Delta l ,$$

where: v_i is the velocity in chosen point,
Δl is the characteristic size of the element.

The numerical values of flow (m^3/s) for all openings in a structure are given in Table 5. The heat amount for warming up the inflowing cold outdoor air is calculated by the formula $Q = q \cdot c_p \cdot \rho \cdot \Delta T$, where c_p is the specific air heat capacity at constant pressure, ρ is the air density and ΔT is the temperature difference between the average air temperature in the room and the outdoor temperature. The average air temperature in the room can be essentially different (lower) from the characteristic temperature in the middle of the room, e.g. in variant G3-1 the characteristic temperature is approx. 25 °C, but the temperature of a window's inner surface with U = 6 W/(m^2K) is only 3 °C. The corresponding heat amounts transferred through openings are shown in Table 5. The difference for airflows in openings is connected with thermal expansion – the flows are not equal because of density change at different temperatures. The obtained total results for heat balance including heat losses by convection are shown in Table 6.

Table 5

Airflows and corresponding heat transfer for various pressure differences

Description	Above window	Under window	Opening for ventilation
Size of opening, m	0.005	0.005	0.025
Air flow at 0 Pa, m^3/s	$3.8 \cdot 10^{-4}$	$5.3 \cdot 10^{-4}$	$1.0 \cdot 10^{-3}$
Air flow at 2 Pa, m^3/s	$6.7 \cdot 10^{-3}$	$4.2 \cdot 10^{-3}$	$1.2 \cdot 10^{-2}$
Heat amount at 0 Pa, W	15		
Heat amount at 2 Pa, W	233		

Table 6

Heat from the convector, heat losses by conduction and convection for various pressure differences

	Modelling variant	G3-1	G3-2
Heat amount (W)	Convector	151	253
	Inner surfaces + convection	166	259
Difference (%)	Convector and inner surfaces + convection	9.6	2.3

Since a significant proportion of heat energy is lost by convection at a given difference of pressure, the room temperature depends not only on the heat losses by conduction but also on the dominating heat losses by convection. The average temperature of the room and the corresponding heat amount delivered by the convector are shown in Fig. 11 (see also Table 4). As can be seen, the room temperature is considerably reduced when there are significant heat losses by convection, despite the fact that the heat from the convector at its given surface temperature is much greater.

a) b)

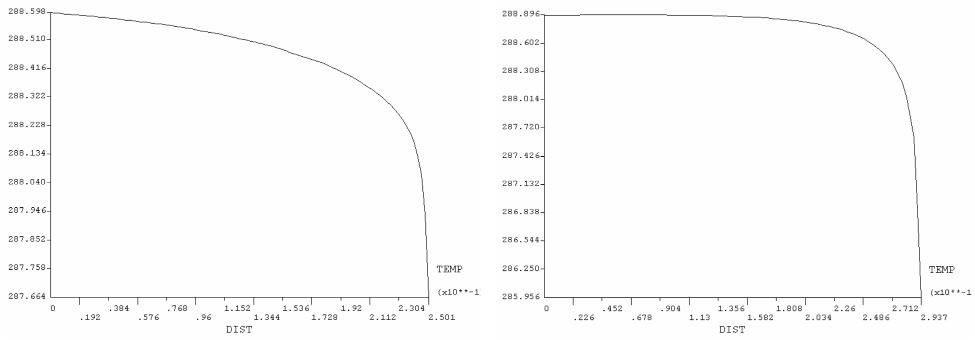

Fig. 12. Temperature profile near the wall for various air circulation regimes

The conditions of heat supply by inner surfaces of building structures also vary depending on the kind and intensity of air circulation in the room. Examples of temperature distribution on the surfaces at low and increased velocities of airflow near these surfaces are shown in Fig. 12. As can be seen, the thermal boundary layer significantly reduces in the case of intense airflow.

The influence of heat transfer both by conduction and by convection on the distribution of various related physical quantities in the room has been investigated in all the discussed variants, with the sources and losses of radiation heat transfer neglected. Heat losses through various elements were compared in order to determine the elements that are critical from the viewpoint of radiation. Heat transfer of the kind is much more intense for transparent surfaces than for non-transparent ones (which are only warmed up, but radiation does not penetrate through them). However, solar radiation can enter the room, while radiation from warm bodies (electric appliances, heating facilities, human beings, etc.) can leave the room. Also the radiation from the convector should be accounted for, in the case of its high surface temperature. However, being not too high this temperature allows the radiation heat transfer from the convector to be neglected.

5. CONCLUSIONS

The 2D calculations of airflows and temperature distribution in a separate room show the influence of rearrangement of structural elements of a room on the character of airflow velocities and directions in this room. As shown, it also influences the temperature field distribution, because of variations in the heat exchange conditions near the building structures. The losses of heat and its distribution are also influenced by variations in the parameters of building structures (namely, in the heat transfer coefficient). This, in turn, changes the intensity of heat conduction processes and therefore the distribution of other physical parameters in the room.

Openings in a room’s walls lead to convective heat losses, which through pressure difference radically change the physical fields’ distribution in the room. In this case additional convective heat losses arise, which are usually greater than those by heat conduction. The distribution of heat losses for rooms is confirmed by measurements of such parameters as airflow velocities and temperatures in different places of a room.

The above-mentioned modelling method enables one to adequately choose (in the design stage) appropriate structural elements - either of the building or its separated rooms – that have the desired values of thermo-technical parameters. The model of a separated room shows the influence of various kinds of factors on the resulting distributions of thermo-physical parameters in the room that are directly related to the conditions of thermal comfort. Since the model allows, at the same time, the heat consumption to be reduced, its use may help to reduce energy production and the related pollution.

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GAISA PLŪSMU UN TEMPERATŪRAS SADALĪJUMA APKURINĀMĀS TELPĀS SKAITLISKĀ MODELĒŠANA

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K o p s a v i l k u m s

Gaisa plūsmu un temperatūru sadalījumu aprēķini telpās uz 2D modeļa bāzes, lietojot ANSYS/Flotran programmatūru, ļauj analizēt dažādu konstruktīvo elementu (sild-elements, palodze, ventilācijas atveres) raksturlielumu un izvietojuma ietekmi gan uz termiskā komforta apstākļiem (gaisa plūsmu ātrumi, temperatūras gradienti) telpā, gan uz siltuma apmaiņas intensitāti caur telpu norobežojošām virsmām , t.i., uz siltuma atdevi caur termiskajiem robežslāņiem.

Parādīts, ka būtiski siltuma zudumu daudzumu un to sadalījumu pa virsmām ietekmē arī būvkonstrukciju siltuma caurlaidības koeficientu izmaiņas. Atveres telpas sienās rada papildus konvektīvos siltuma zudumus pat tad, ja nav ārējās (piem., vēja izraisītas) spiedienu starpības, bet gaisa cirkulācija šajās atverēs būtiski pieaug pat pie nelielas (2 Pa) spiedienu starpības.

Tādējādi variējot konstruktīvu elementu izvietojumu un to siltu mtehniskos parametrus, iespējams samazināt siltuma patēriņu, tajā pat laikā nodrošinot termiskā komforta apstākļus telpā. Izstrādātā pieeja ļauj veikt telpu apsildes un ventilācijas optimizāciju projektu sagatavošanas stadijā.