NUMERICAL ANALYSIS
OF THERMAL COMFORT PARAMETERS IN LIVING QUARTERS

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Summary: This paper includes an evaluation of ventilation conditions in a given living quarters – a room in a single-family house, based on local parameters of thermal comfort determined by numerical calculations. Global parameters (Predicted Mean Vote and Predicted Percentage of Dissatisfied) and local parameters (including: Resultant Temperature, Relative Humidity) were determined from numerical solution of transient case of living quarters ventilation in ANSYS-CFX software.

1. INTRODUCTION

Numerical modelling of ventilation in living quarters, office space and utility rooms has been a subject of many scientific papers over the past few years (Abanto et al., 2004; Evola and Popov, 2006; Lin et al., 2007; Stamou and Katsiris, 2006). Doing this kind of calculations allows an in-depth analysis of the ventilation issues already on the project stage, which significantly lowers the cost of the investment.

The goal of this paper is to determine the influence of a chosen ventilation type on the thermal comfort in living quarters by analysing the thermal comfort parameters which are the results of numerical calculations done in ANSYS-CFX software.

Defining the notion of thermal comfort in rooms is difficult, because it is highly individual and subjective. In order to define thermal comfort of large groups of people, special parameters of thermal comfort have been established. These factors were divided by the PN-EN ISO 7730:2006(U) norm into: global, comprehensively determining personal sensations, and local, describing the effect the particular elements of a micro-climate have on the satisfaction or dissatisfaction with the conditions in a given room and describing the negative effects they have on particular body parts.

2. THERMAL COMFORT FACTORS

Thermal comfort and discomfort factors were described in PN-EN ISO 7730: 2006(U) norm. The most important parameters of global comfort were considered to be the PMV – Predicted Mean Vote – expressed in a scale from -3 to +3 and PPD – Predicted Percentage of Dissatisfied. Local comfort and discomfort factors include: Draught Rating, Local Thermal Discomfort Caused by Warm or Cold Floor and Vertical Air Temperature Difference Between Head and Ankles.

PMV is calculated from the formula: (Fanger, 1974):

$$PMV = \left(0{,}352e^{-0{,}042(Q_M/A_{Du})} + 0{,}032\right)\{\frac{Q_M}{A_{Du}}(1-\eta) -$$

$$0{,}35\left[43 - 0{,}061\frac{Q_M}{A_{Du}}(1-\eta) - p_w\right] - 0{,}42\left[\frac{Q_M}{A_{Du}}(1-\eta) - \right.$$

$$\left. 50\right] - 0{,}0023\frac{Q_M}{A_{Du}}(44 - p_w) - 0{,}0014\frac{Q_M}{A_{Du}}(34 - t_w) -$$

$$3{,}4\cdot10^{-8}f_{cl}\left[(t_{cl}+273)^4 - (T_{mrt}+273)^4\right] -$$

$$f_{cl}\alpha_k(t_{cl}-t_w)\} \tag{1}$$

where: A_{Du} is the DuBois area (surface area of a human body) $[m^2]$, Q_M is the metabolic heat $[W]$, η is the physical fitness p_w- partial pressure of water vapour in the surrounding air [mmHg], t_w is the air temperature [°C], f_{cl} is the ratio of the surface area of clothed body to the surface area of exposed body, t_{cl} – mean temperature of the clothed surface of human body [°C], $T_{mrt}-$ mean radiant temperature [°C], α_k is the heat transfer through convection $[W/(m^2K)]$. The temperature t_{cl} is calculated from the formula:

$$t_{cl} = 35{,}7 - 0{,}032\frac{Q_M}{A_{Du}}(1-\eta) - 0{,}18\Lambda_{cl} \tag{2}$$

$$\{3{,}4\cdot10^{-8}f_{cl}\left[(t_{cl}+273)^4 - (T_{mrt}+273)^4\right] +$$

$$f_{cl}\alpha_k(t_{cl}-t_w)\}$$

where: Λ_{cl} is the thermal resistance coefficient of clothing [clo].

The Predicted Percentage of Dissatisfied expresses the heat sensations of a group of people dissatisfied with the thermal conditions in a given room. It is calculated from the formula (Fanger, 1974):

$$PPD = 100 - 95e^{(0{,}03353PMV^4 + 0{,}2179PMV^2)} \tag{3}$$

Draught Rating (DR) is calculated as follows (PN-EN ISO 7730:2006):

$$DR = (34 - t)(v - 0{,}05)^{0{,}62}(0{,}37vT_u + 3{,}14) \tag{4}$$

where: t and v is, respectively, temperature [°C] and mean air velocity $[m/s]$, T_u is the turbulence intensity; formula (4) is applicable within these parameters: $t = 20°C \div 26°C$, $v < 0{,}5\ [m/s]$ i $T_u = 10\% \div 6\%$; for $v < 0{,}05\ [m/s]$ it is presumed that $v = 0{,}05\ [m/s]$, and in case of $DR >$

100%, the value of $DR = 100\%$ is accepted;

An important local indicator of thermal comfort is also the relative humidity RH calculated from the formula:

$$RH = \frac{\rho}{\rho_m} 100\% \ ; \tag{5}$$

ρ – water vapour density, ρ_m – density of saturated water vapour.

The felt air temperature is described numerically by determining the DRT – Dry Resultant Temperature, calculated from the formula (Awbi, 2003):

$$DRT = \frac{t_{mrt}+3{,}17\sqrt{v}}{1+3{,}17\sqrt{v}} \ [°C]; \tag{6}$$

where: t_{mrt} mean radiant temperature [°C], calculated from Stefan – Boltzman law, t – temperature [°C] I v – air velocity $[m/s]$; for $v < 0{,}1 \ [m/s]$ it is presumed that $DRT = (tmrt+t)/2$.

PDV – percentage of dissatisfied from vertical air temperature difference between head and feet – is calculated from the following ratio (PN-EN ISO 7730):

$$PDV = \frac{100}{1+e^{(5{,}76-0{,}856\Delta t_v)}}, \tag{7}$$

where $\Delta t_v < 8°C$ is the difference in temperatures between head and feet measured vertically [°C];

PDF – percentage of dissatisfied caused by warm or cold floor – is calculated from the formula (PN-EN ISO 7730):

$$PDF = 100 - 94e^{(-1{,}387+0{,}118t_f-0{,}0025t_f^2)}, \tag{8}$$

where t_f – floor temperature [°C].

After performing a numerical simulation of the ventilation in ANSYS-CFX, an evaluation of thermal comfort has been conducted. In order to do so, the global, PMV (1) and PPD (3), and local, DR (4), RH (5), DRT (6), PDV (7), PDF (8) parameters were applied.

3. DESCRIPTION OF LIVING QUARTERS

The calculations were done for an actual living quarter located in a single-family house from the 1970's. The geometry of the room with most important features is presented in Fig. 1. It represents the flow domain filled with work fluid. Because of the need to reduce the number of nodes in computational grid, shapes of most of the elements have been simplified. Such simplifications are commonly used in numerical calculations regarding the ventilation problems (Evola and Popov, 2006; Lin et al., 2007; Stamou and Katsiris, 2006), rarely are there publications where the geometry of the elements is more complex (Abanto et al, 2004; Sorensen and Voigt, 2003).

Room dimensions: 4m x 4m x 2,8m. The calculations include the following heat sources: human, radiator, computer and chandelier. Heat penetration through external walls and window was also taken into consideration. Heat gain from sunshine was neglected.

The room is ventilated using a trickle vent, and the air-exhaust is done through a vent connected to ventilation chimney. Air flow (inflow and outflow) is also possible through a crack under the door leading to the rest of the house.
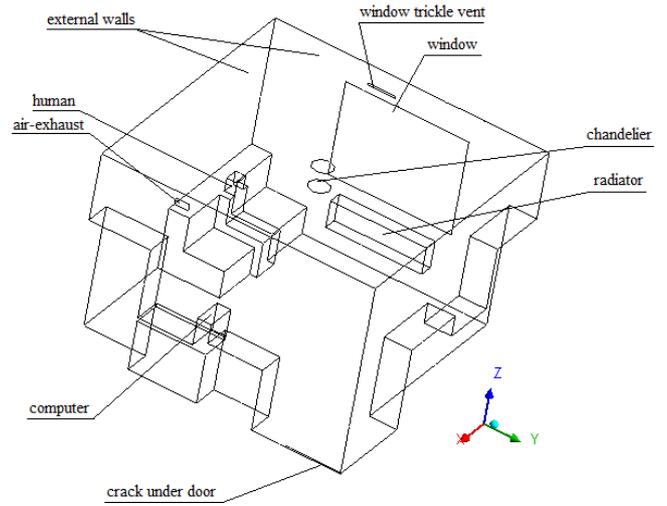
Crack dimensions: 0,8m x 0,02m.

Fig. 1. Geometry of the subject room

4. MATHEMATICAL MODEL

The flow of air (working fluid) in a given room is governed by the basic laws of mechanics: conservation of mass, conservation of momentum and conservation of energy. These are their differential forms:

− continuity equation:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot \left(\rho \vec{V}\right) = 0; \tag{9}$$

where ρ – density of the working fluid, $\vec{V}$ – is the velocity vector;

− Navier-Stokes equation:

$$\frac{\partial (\rho \vec{V})}{\partial t} + \nabla \cdot \left(\rho \vec{V} \otimes \vec{V}\right) = -\nabla p + \nabla \cdot \tau + \vec{f}; \tag{10}$$

where p is the pressure and

$$\tau = \mu(\nabla \vec{V} + \nabla \vec{V})^T - \frac{2}{3}\delta \nabla \cdot \vec{V}); \tag{11}$$

is the stress tensor, $\vec{f}$ – body forces

− energy equation:

$$\begin{aligned}\frac{\partial (\rho h_c)}{\partial t} - \frac{\partial p}{\partial t} &+ \nabla \cdot \left(\rho \vec{V} h_c\right) \\ &= \nabla \cdot (\lambda \nabla T) + \nabla(\vec{V} \cdot \tau) + \vec{V} \cdot \vec{f} + S_E\end{aligned} \tag{12}$$

where $h_c = e + \frac{p}{\rho} + \frac{1}{2}V^2$ – total enthalpy, e – internal energy, λ – thermal conductivity, S_E – heat source.

The equations presented above need to be supplemented with proper initial and boundary conditions.

In case of turbulent flow, solving the Navier-Stokes equations is ineffective, because of the limited capabilities of modern computers, which does not allow for the proper density of computational grid. Because of this, the equations are substituted with Reynolds equations. In order to close the set of equations, additional equations are needed – the 'turbulence model'. In this paper, the $k-\varepsilon$ turbulence model was utilized in an RNG modified form, widely used in this type of calculations (Abanto J et al 2004).

The ANSYS-CFX software solves the fluid flow and heat transfer problems described by RANS equations using the control volume method, utilizing the shape function for estimating the variables within the control volume (ANSYS-CFX, 2006).

5. NUMERICAL MODELLING

Numerical modelling utilizing the ANSYS-CFX requires creating a geometrical model, computational grid, choosing the work fluid, determining the model of heat transfer and the type of flow (the turbulence model), defining the initial and boundary conditions, selecting the solution parameters and conducting the calculations.

In this case, the working fluid is a humid air, consisting of dry air (ideal gas parameters) and water vapour. Water vapour parameters were taken from the IAPWS IF97 library implemented in the ANSYS-CFX software. Additionally, a distinction was made between "fresh" air, flowing from outside, and "old" air, present in the room. As a result, the working fluid was a mixture (ideal) of "old" air, water vapour, and "fresh" air. The distinction of different kinds of air in the mixture allowed an easy analysis of air flowing from outside. Because of the importance of natural convection in the analysed case, the physical flow model included the gravity. Defining the thermal comfort parameters required determining the mean radiant temperature; in the physical model of the process the Discrete Transfer radiation model, recommended for this type of calculations by the ANSYS-CFX documentation, was applied.

The simulation was divided into two stages. The first stage was a transient simulation of natural convection caused by an abrupt "switching on" the internal heat sources: radiator (350W), chandelier (50W) and computer (95W). In still air, which was initially thermally homogeneous (20°C) with relative humidity of 40%, the human influence was also included. A person resting in seated position generates a heat flux (20W) and mass flow rate of water vapour (41 g/h) (Fanger, 1974). The calculations included the heat losses through non-transparent boundaries (external walls) with heat transfer coefficient of building build between 1976-1983, equalling 1,163 $W/(m^2K)$ established according to PN-74/B03404 norm; and heat transfer coefficient of windows equalling 1,1 $W/(m^2K)$. Reference pressure was 0.1MPa. For the purpose of these calculations, the outside temperature was assumed to be -5°C. The door to the room was closed, but the air flow was possible through the crack under the door (Fig. 1) with pressure difference equalling 0Pa. The air flowing out from the room had a temperature of 20°C and relative humidity of 40%.

In the second stage, 10 seconds after "switching on" the heat sources, the window vent was opened, which allowed the inflow of outside air of temperature -5°C and relative humidity 100%, as well as the air exhaust (outflow with 0.5Pa pressure difference) connected to ventilation chimney. The inflow of the outside air had a constant velocity of 0.5m/s, normal to the inflow surface. The calculations were continued until t=55s.

The calculations used RNG k-ε turbulence model with scaled wall function; the intensity of turbulence was set to 5%.

6. RESULTS. DISCUSSION

The calculations used a grid with 169338 nodes. The simulation was conducted until t=55s. The results presented in this paper reflect the thermal comfort for that moment.

In the first stage of simulation, that is during the natural convection over internal heat sources: human, radiator, computer and chandelier, a convective current moving up towards ceiling could be observed. The most intensive ascending stream was located over the radiator, parallel to the surface of window.
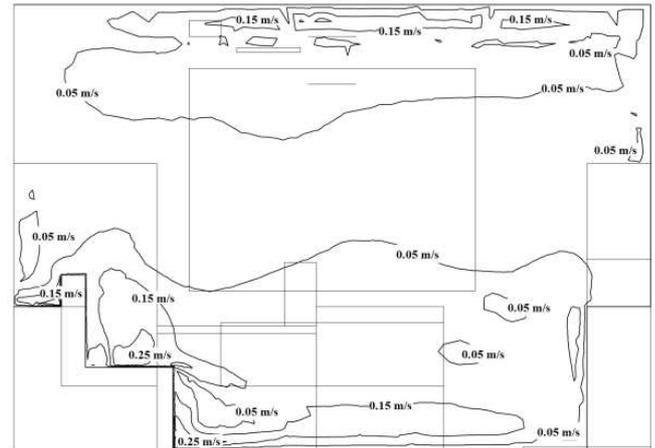

Fig. 2. Velocity isolines in plane x=1m,
going through the resting person, for *t*=55s

After opening the air vent and air-exhaust, cold air of temperature -5°C began to flow inside. Relatively strong convection current in the plane of window, moving upwards, caused a turbulence of air close to the ceiling. Cold air, warmed after contacting the warmer convection current moved from the window vent towards the ceiling, towards the corner of the room on the left side of the window and further down the wall towards the sitting person or moved directly down along the window. The movement of air in the room had significant influence on the thermal comfort in the room.

Fig. 2. presents the velocity isolines in plane x=1 going through the resting person and parallel to the window. Although the plane was distanced from the window (where the air velocity reached 0.68m/s), an eddy flow close to the ceiling could be observed, caused by the convection current which in turn was caused by the radiator. The highest velocity – around 1.36m/s – was reached close to the air-exhaust. The air flow through the door crack reached 0.96m/s. In the area around the sitting person the air velocity equalled 0.05÷0.35m/s and in some places exceeded velocity acceptable in winter conditions, that is 0.2÷0.3m/s.

Draught Rating (predicted percentage of people dissatisfied with the draught) reached its highest value around the door crack – up to 40%. *DR* was around 10÷13% near the window, that is close to the strongest convective currents and close to window vent and air-exhaust. Fig. 3. presents the area of constant value of the *DR*=3%. In most areas of the room the *DR* value was much lower.
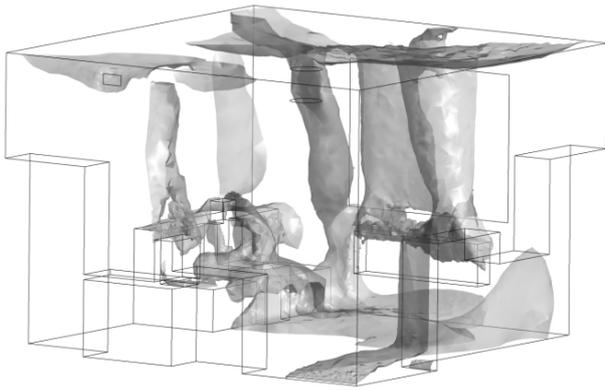
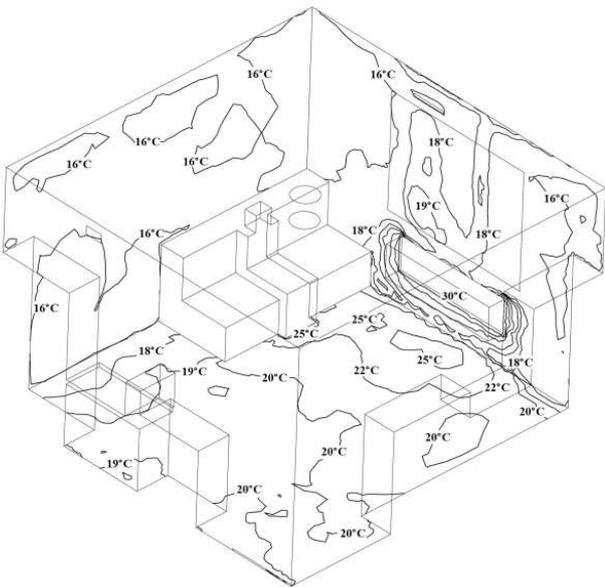**Fig. 3.** Surface with constant indicator value *DR*=3% for *t*=55s

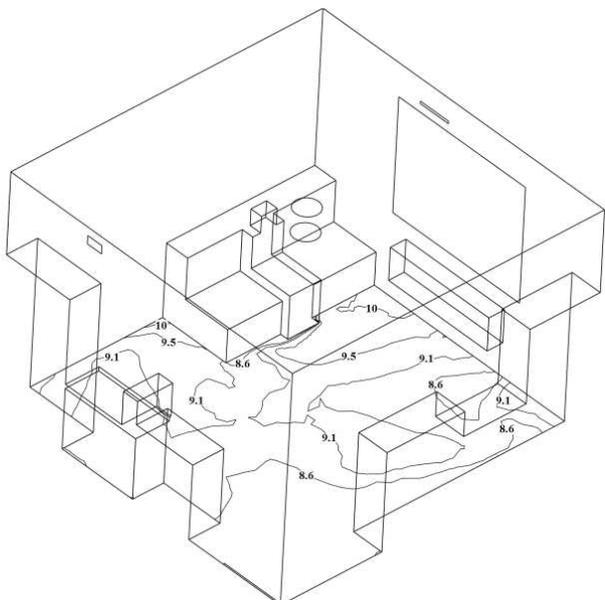**Fig. 4.** Isotherms on the floor and external wall surface for *t*=55s

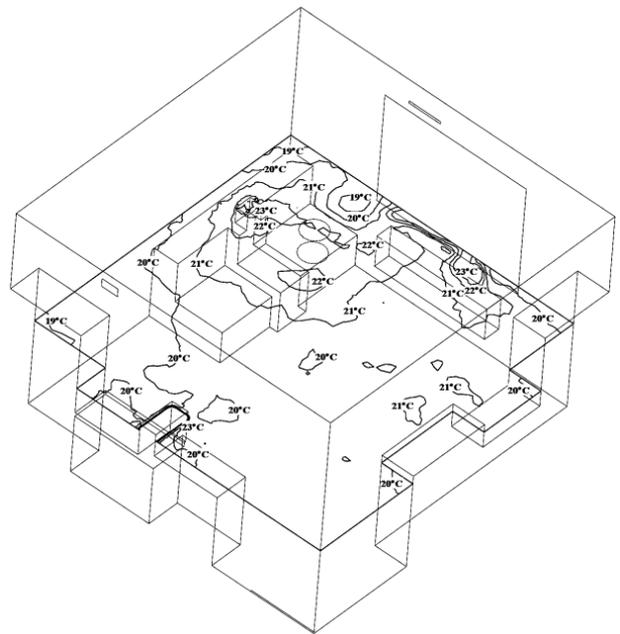**Fig. 5.** Isolines of the *PDF* indicator (predicted percentage of people dissatisfied with floor temperature) for *t*=55s

Fig. 6. *DRT* isotherms for a plane distanced by 1.1m for *t*=55s

The temperature of air in the room varied from -5°C on the surface of air vent up to 50°C on the radiator surface and 45°C on the chandelier's surface. Fig. 4. shows the isotherms on the floor and external walls of the analysed room. Because of the internal heat sources, the temperatures of window, external walls and floor around the radiator rose, which caused a rise in the predicted percentage of dissatisfied with floor temperature – *PDF* – Fig. 5. In the rest of the room this indicator was within the range of 8.5÷9.5%.

Fig.6. shows the isolines of *DRT* resultant temperature in plane parallel to the floor, distanced by 1.1m. This plane is located on the head level of a sitting person. Temperature felt for *t*=55s is slightly higher than the 20÷22°C recommended in winter conditions – this is caused mostly by the heat generated by the internal heat sources.
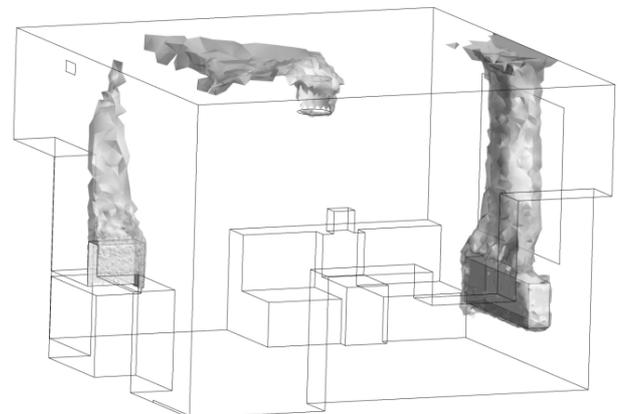

Fig. 7. Relative humidity *RH*≤40% for *t*=55s

Internal heat sources have tremendous effect on the thermal comfort conditions in a room. The effect of such sources is clearly visible in Fig. 7., which presents the part of the room with relative humidity lower than 40% – un-

comfortable for human. Observation of humidity distribution behaviour over time shows a growing area of uncomfortably low relative humidity *RH*, caused by internal heat sources and convection currents caused by them.

The predicted percentage of people dissatisfied with the difference of temperatures between head and ankles (*PDV*) was calculated for a seated person, presuming the ankle level to be 0.1m and head level 1.2m. The *PDV* factor distribution, presented in Fig. 8., shows that the *PDV* is low and for the majority of the room is lower than 1%. With passing time the value of the *PDV* indicator rises slightly in the vicinity of radiator and computer, that is in areas where human presence is physically impossible.
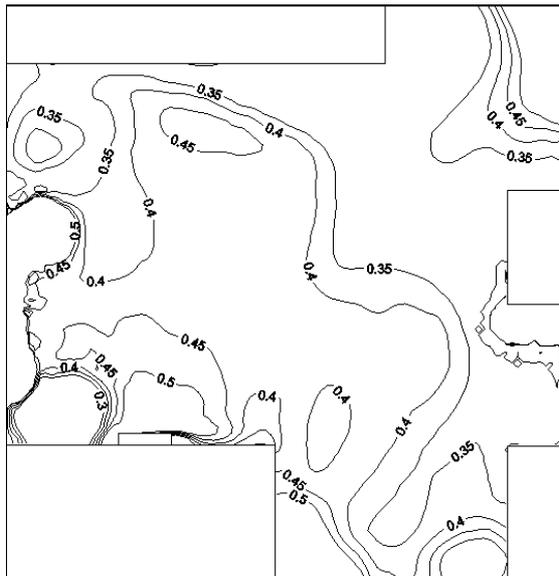

Fig. 8. Isolines of the *PDV* indicator (predicted percentage of people dissatisfied with the difference of temperatures between head and ankles) for *t*=55s
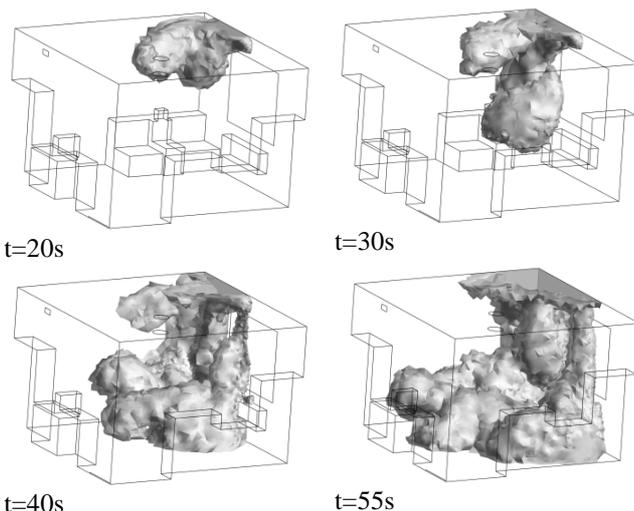

t=20s   t=30s

t=40s   t=55s

Fig. 9. Graphs of volume of air containing at least 0.01 mass fraction of fresh air after t=20, 30, 40 and 55s

Dividing the working fluid into two types: fresh air (outside) and old air (inside) in the physical model allowed visualising the inflow of outside air and observing its distribution in the room. Fig. 9. shows the graphs of volume

of air containing at least 0.01 mass fraction of fresh air after *t*=20, 30, 40 and 55s. These graphs show the process of spreading of fresh air flowing from outside. The air flowing from the window vent moves towards the corners of the room, between the window and the sitting person, flows down, is partially lifted by the convection current from the radiator and moves up, towards the ceiling, along the window plane. Inside the room, after 55 seconds, 22% of the total area in room contains at least 0.01 mass fraction of fresh air.

Determining the global thermal comfort parameters *PMV* (1) and *PPD* (3) requires calculating the body temperature t_{cl}, which is achieved by solving a non-linear equation (2). This requires programming a calculation procedure in Fortran. Because of the lack of Fortran compiler compatible with ANSYS-CFX, this paper is limited to determining the average values of global thermal comfort parameters *PMV* and *PPD* for the whole room. For this purpose the calculator published on www.healthyheating.com/solutions.htm web-page was used.

The calculations assumed the following values: energy flux caused by a person in the room was equal 1 met (for a person resting in a seated position) and the thermal resistance coefficient of clothing was equal 0,6 clo. The following values were taken from the ANSYS-CFX postprocessor : the mean radiant temperature average in the room 20.91°C, the average relative humidity 44.41%, the average air velocity 0.075m/s and average air temperature 20,58°C. The calculations resulted in the following global thermal comfort parameters: *PMV*= - 1.3 , *PPD*=40.3%.

Based on these results it can be said that the conditions in the analysed room would not be comfortable for 40.3% of people staying in this room. The resulting average *PMV* means that a large group of people would describe their heat sensation as cold or relatively cold.

The goal of this paper was to determine the thermal comfort parameters in living quarters using numerical modelling. Numerical modelling allows for a quick analysis of the influence the type of ventilation has on thermal comfort conditions in living quarters. For this kind of calculations, very important is the creating of proper physical model of the fluid flow and heat transfer problem. The quality of the result is heavily influenced by the computational grid. In other words, in order to obtain good quality results, the calculations require dense computational grid and good computer equipment.

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