NUMERICAL MODELLING OF TEMPERATURE AND AIR FLOW DISTRIBUTION IN ENCLOSED ROOM

Igor Bonefacic[1], Bernard Frankovic[2], Ivan Vilicic[3], Vladimir Glazar[4]

Faculty of Engineering, Vukovarska 58, Rijeka, Croatia,
e-mail: [1]igor.bonefacic@riteh.hr, [2]bernard.frankovic@riteh.hr, [3]ivan.vilicic@riteh.hr, [4]vladimir.glazar@riteh.hr

In the paper three-dimensional case of heat transfer and air flow has been presented for enclosed space heating and cooling with unit mounted on the wall. Numerical modelling has been used to analyze effect of space height on temperature and air velocity distribution under standard conditions. Physical processes in two rooms with different heights are modelled using specialized computational fluid dynamics (CFD) software FLUENT. Calculations are carried out in an empty room without any human or mechanical activity and compared with previous experimental measurements.
Numerical analysis has shown distinct relationship between room dimensions (height in this case) and state of comfort in the room. Although average values of temperature and air velocity in the room are satisfactory there are local distortions from the standard optimal comfort values. Considerable differences in temperature distribution are observed especially in heating conditions where temperature drop in lower layers of higher room occur due to buoyancy effects. To cancel buoyancy effects it is necessary to increase air inlet velocity from the unit which could lead to local air velocity distortions resulting with disturbed optimal comfort values.

1. Introduction

Primary goal during air-condition designing of enclosed working space is to find adequate microclimate for people residing in that room. Basically, it means that state of thermal comfort has to be achieved. That implies thermodynamical balance between human body and the environment during different physical activities. Physical quantities of greatest relevance for microclimate are air temperature in the room, air humidity, local air velocity and surface temperature of surrounding walls, windows and heating surfaces.
In recent years, extensive research effort has been made in the development of general fluid flow and heat transfer programs for solving the flow of air in a room. Similar studies had been carried out, Lee et al.(1997) applied the Finite element method to study the characteristics of forced and mixed convection in an air-cooled room for both the laminar and turbulent regimes, Sinha (2001) deals with case of two-dimensional numerical prediction of an airfow configuration in a room (enclosure) with inlet and outlet as applicable to cooling and ventilation. Three-dimensional model of a computer room with regard to thermal comfort was presented by Richtr et al. (2001) using standard k-ε turbulence model for modelling air flow. Rutman et al. (2002) give extensive view on using numerical modelling tools for improving indoor environment quality, component optimisation and complete system optimisation.

Air temperature

A general recommendation for design of heating and cooling processes for working offices and enclosed areas, where no major physical activities are carried out, is that the temperature has to be held constant in between 21-23°C. During summertime, when outdoor temperatures are higher, it is advisable to keep air-conditioned offices slightly warmer then in winter periods to minimize the high temperature difference between indoors and outdoors. Recommendation is that difference should not be higher than 7°C. Thermal comfort depends also on vertical temperature distribution in the room. Allowed vertical air temperature difference between 1.1m and 0.1m above the floor (head and ankle respectively of the person in sitting position) or between 1.7m and 0.1m above floor (head and ankle respectively of the person in upright position) is less than 3°C (ISO 7730 -1994).). According to Zukowski (2003), the most comfortable state is that where floor temperature is the same or slightly higher than air temperature in head level.

Mean Radiant Temperature

State of thermal comfort cannot be determined simply by knowing air temperature in the room. It is known fact that at least 50% of all thermal energy between the environment and a human body exchanges through radiation. A human body is constantly exchanging radiant heat with all objects that surrounds it due to temperature difference between them. Mean Radiant Temperature (MRT) of a space is really the measure of the combined effects of temperatures of surfaces within that space on human body. A scale of seven degrees from −3 to +3 can be used to measure thermal sensation based on the scale of ASHRAE (1992) and Nagano, Morchida (2004). The larger the surface area and the closer one is to it, the more effect the surface temperature has on the individual. MRT thus changes with position in the room and with it corresponding air temperature which provides thermal comfort. In general if MRT drops by 1°C, air temperature has to be raised by 1 - 1.5°C.

Air velocity (draft)

Air flow velocity is important factor in determining thermal comfort because increased air velocity can cause discomfort due to increased convective heat transfer from surface of a human body. ISO 7730-1994 limits local air flow velocity in offices and spaces where no major physical activites occur at 0.15 m/s during winter periods (heating) and 0.25 m/s during summer (cooling). Sense of draft is more pronounced with decreased phyisical activity. Also, discomfort caused by draft increases with air temperature drop. Contrary, if air velocity in enclosed space is to low it could couse feeling of stuffiness and bad odour especially in winter (heating) periods. It is, therefore, necessary to assure optimal level of air circulation in the room.

2. Mathematical model

The problem is solved using mathematical model of threedimensional turbulent flow of uncompressible fluid which is described using general differential equation:

$$\frac{\partial}{\partial t}(\rho\phi)+\frac{\partial}{\partial x_i}(\rho u_i\phi)=\frac{\partial}{\partial x_i}\left(\Gamma\frac{\partial}{\partial x_i}\right)+S_\phi\,, \tag{1}$$

where variable ϕ represents velocity components *u*, *v*, *w* and temperature T.

The importance of buoyancy forces in a mixed convection flow can be measured by the ratio of the Grashof and Reynolds numbers:

$$\frac{\mathrm{Gr}}{\mathrm{Re}^2}=\frac{\Delta\rho g h}{\rho v^2}\,, \tag{2}$$

which exceeds unity, showing that strong buoyancy effects in the flow could be expected.

Performances of different turbulence models in similar studies were examined in Rouaud, Havet (2002). Turbulence was modeled using realizable k-ε model which consist of transport equation for turbulent kinetic energy k, and its dissipation rate ε.

$$\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_i}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k+G_b-\rho\varepsilon\,, \tag{3}$$

$$\frac{\partial}{\partial x_i}(\rho\varepsilon u_i)=\frac{\partial}{\partial x_i}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]-\rho C_2\frac{\varepsilon^2}{k+\sqrt{\nu\varepsilon}}+C_{1\varepsilon}\frac{\varepsilon}{k}C_{3\varepsilon}G_b\,. \tag{4}$$

In these equations G_b represents generation of turbulence kinetic energy due to buoyancy effects. C_2, $C_{1\varepsilon}$ and $C_{3\varepsilon}$ are constants. σ_k and σ_ε are the turbulent Prandtl numbers for k and ε, respectively (FLUENT Inc., 1998).

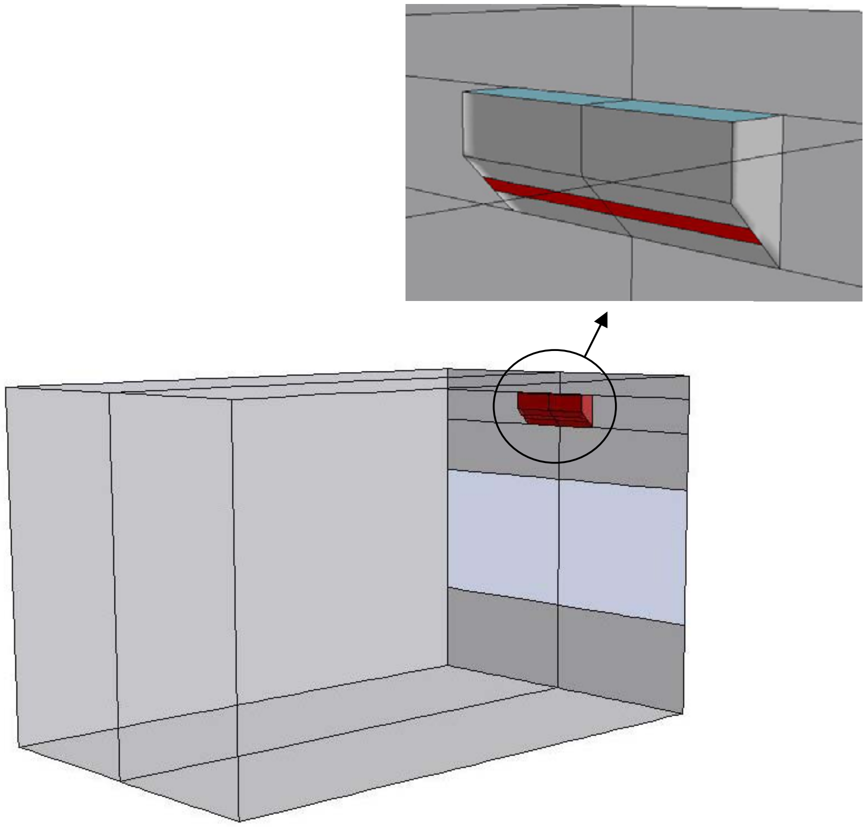
System of transport equations was solved using CFD software FLUENT based on finite volume method (Patankar, 1980).

3. Boundary conditions

3.1. SUMMER

Constant air inlet velocity boundary condition:

v_{in} =2,0 m/s,
T_{in} =293 K,
φ = 0° (Air flow inlet angle).



Picture 1. Simulated enclosed room

Heat transfer due to convection and radiation is defined by:

$$q = \alpha_V \cdot (T_V - T_Z) + \varepsilon_V \cdot \sigma \cdot \left[\left(\frac{T_\infty}{100} \right)^4 - \left(\frac{T_Z}{100} \right)^4 \right], \qquad (5)$$

where is:

α_V - Convection heat transfer coefficient at the external wall surface;
T_V - External air temperature;
T_Z - External wall surface temperature;
T_∞ - External radiation temperature;
ε_V - External emissivity;
σ - Stefan-Boltzmann's constant.

3.2. WINTER

Constant air inlet velocity boundary condition:

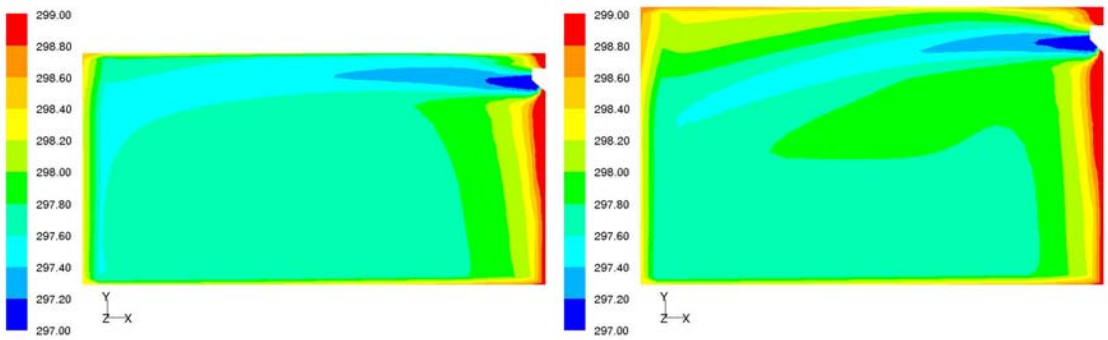
v_{in} =1,7 m/s,
T_{in} =300 K,
φ = 45° (Air flow inlet angle).

Convective heat transfer is defined by:

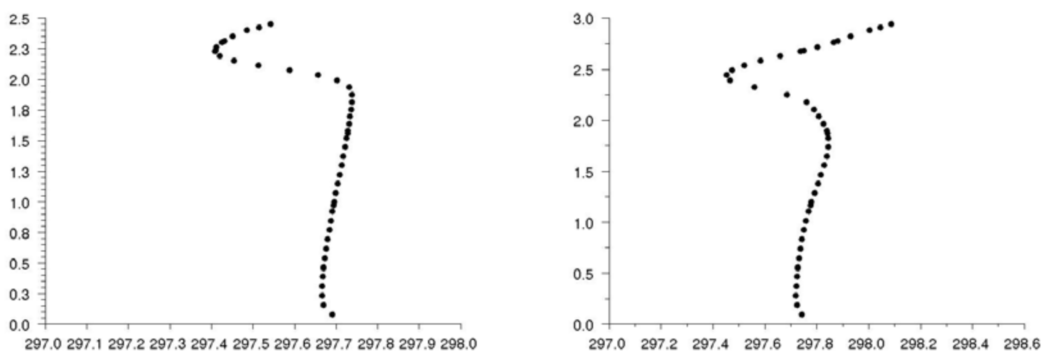
$$q = \alpha_V \cdot (T_V - T_Z)\,. \qquad (6)$$

4. Simulation results

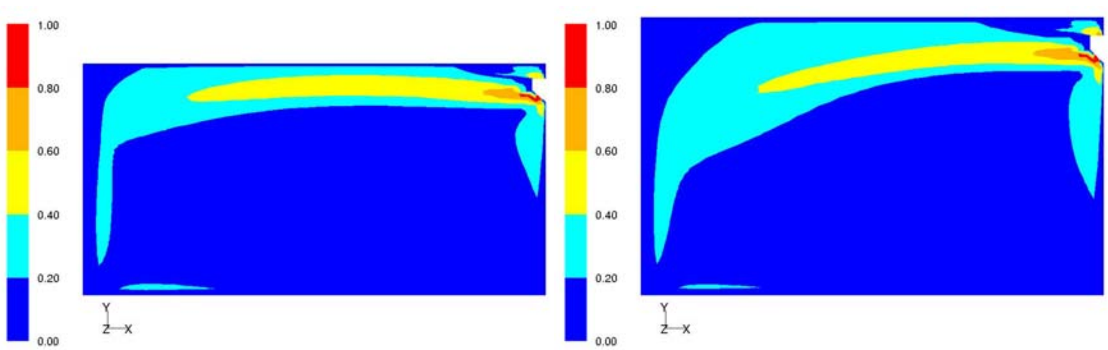
4.1. SUMMER



Picture 2. Temperature field for rooms with different heights, 2.5m (left) and 3m (right)

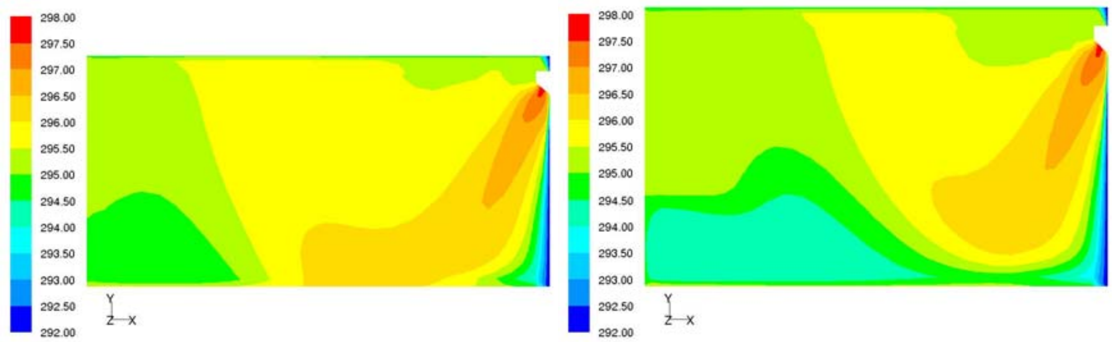


Picture 3. Vertical temperature distribution in the middle of the room

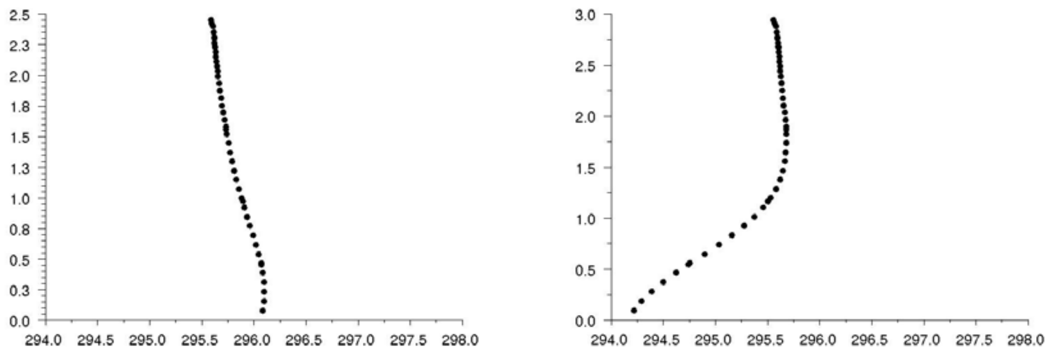


Picture 4. Air velocity field

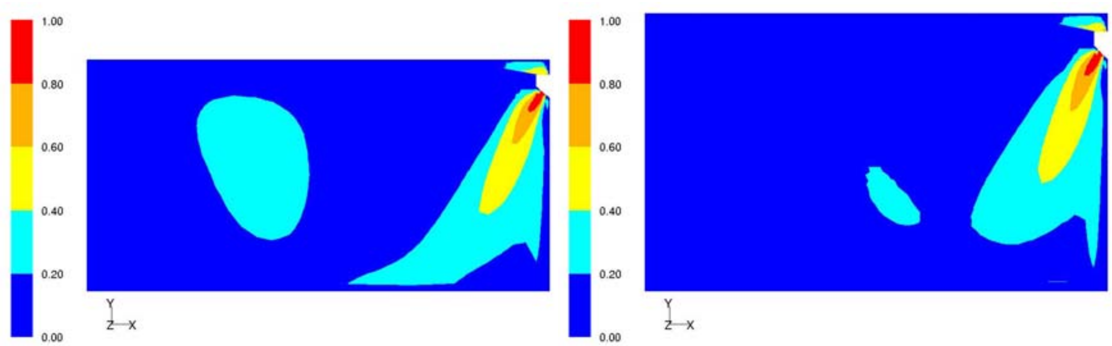
4.2. WINTER



Picture 5. Temperature field for rooms with different heights, 2.5m (left) and 3m (right)



Picture 6. Vertical temperature distribution in the middle of the room



Picture 7. Air velocity field

5. Conclusion

In this work heating and cooling of the enclosed room is simulated (Picture 1). Numerical analysis based on finite volume method is used to solve threedimensional unsteady flow in enclosed space and resulting temperature distribution and air velocity field for two rooms with different height is given. Analysis shows strong influence of a/c unit position and room geometry (room height in this case) on thermal comfort. Despite the fact that mean values of

temperature and air flow velocity are satisfactory there are some deviations in local values. Results of numerical analysis show considerable differences in temperature distribution beetwen rooms with different height especially in winter period (Picture 5) . In case of 3m high room the warm air cannot reach lower levels due to buoyancy and as result local air temperature below 1m from the floor is not satisfactory (Picture 6). To compensate for buoyancy effects it is necessary to increase air inlet velocity from the unit which, on the other hand, leads to disturbed thermal comfort because of draft.
Conducted numerical analysis can be relatively easy expanded on similar technical problems from the field of heating, ventilation and climatization. It could inlude humidity or calculation of mean radiant temperature (MRT). Also, time dependent boundary conditions could be included which leads to more realistic simulations allowing analysis of system regulation. Major limiting factor for simulating more complex problems is, of course, available CPU power i.e. calculation time.

NOMENCLATURE

C -	constant
g -	acceleration due to gravity, m/s^2
G -	generation of turbulent kinetic energy, kg/ms^3
h -	height, m
k -	turbulence kinetic energy, m^2/s^2
q -	heat flux, W/m^2
S -	source term
t -	time, s
T -	temperature, K
u,v,w -	velocity components, m/s
x,y,z -	Cartesian coordinates, m
	greek letters
α -	heat transfer coefficient, W/m^{2}K
Γ -	diffusivity
ε -	turbulent dissipation rate, m^2/s^3
ε -	emissivity
μ -	dynamic viscosity, Pa·s
ν -	kinematic viscosity, m^2/s
ρ -	density, kg/m^3
σ -	Stefan-Boltzmann constant, $5.67 \cdot 10^8$ W/m^{2}K^4
σ_k -	turbulent Prandtl number for k
σ_ε -	turbulent Prandtl number for ε
φ -	angle, °
Φ -	transported scalar
	subscripts
in -	inlet
Z -	wall
V -	external

REFERENCES

Rutman, E., Fournier, F., Jurkowski, R., Bailly, A., Franković, B., Viličić, I., 2002, Industrial Approach for CFD Modelling Applications for Air-conditioning and Heat Exchanger Systems, *Energy and the Environment 2002*, Vol. 2, Opatija, Croatia, pp.11-18

Lee, S.C., Cheng, C.Y., Chen, C.K., 1997, Finite elements solutions of laminar and turbulent flows with forced and mixed convection in an air-cooled room, *Numerical Heat Transfer, Part A, 31*, pp. 529–50.

Sinha, S. L., 2001, Behaviour of Inclined Jet on Room Cooling, *Building and Environment 36*, pp. 569-578.

Richtr, J., Katolicky, J., Jicha, M., 2001, Numerical modeling of the influence of angle adjustment of A/C diffuser vanes on thermal comfort in a computer room, *Seventh International IBPSA Conference*, Rio de Janeiro, Brazil, pp. 1135-1141.

Zukowski, M., 2003, A Numerical Analisys of Heat and Mass Transfer in a Room with the Air-Underfloor Heating, *8th International IBPSA Conference*, Bialystok Technical University, Poland.

ASHRAE ANSI/ASHRAE Standard 55-1992 Thermal environmental conditions for human occupancy, 1992.

FLUENT 6.1 User's Guide, FLUENT Inc., 1998.

Rouaud, O., Havet, M., 2002, Computation of the Airflow in a Pilot Scale Clean Room Using k-ε Turbulence Models, *International Journal of Refrigeration 25*, pp. 351-361.

Patankar, S.V., 1980, Numerical heat transfer and Fuid Flow, Washington DC, Hemisphere.