NUMERICAL MODELING OF THE INFLUENCE OF ANGLE ADJUSTMENT OF A/C DIFFUSER VANES ON THERMAL COMFORT IN A COMPUTER ROOM

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ABSTRACT

In the paper, based on MS Thesis by Richtr, 2000, the authors describe CFD modeling of a computer room with regard to thermal comfort of workers. The room was equipped with an additional A/C and people felt uncomfortable in certain places after the installation. As input data for CFD simulation, individual heat loads measured on computers and monitors were used. Different scenarios of adjustable vanes of the A/C diffuser and further modifications were studied. Based on the results of velocity and temperature fields, a solution was suggested to improve the both fields in respect to obtain a better thermal comfort in the room.

INTRODUCTION

Using a standard methodology to design or improve thermal comfort in a room is in principle an easy task. But it is a well-known fact that when we design a ventilation system for a room in order to guarantee a thermal comfort, the reality is often very different. After the room is filled with furniture, computers and people, we find that the airflow and temperature are far from being optimal. The solid objects and people obstruct the airflow and affect the velocity and temperature fields. As a result people often feel uncomfortable either by a higher air velocity or a low temperature in working places.

One possible way how to assess and then optimize the designed system of ventilation or A/C is to use CFD modeling. With this tool we can predict the resulting velocity and temperature fields in a room that is filled with different objects and where we can expect that the velocity and temperature fields differ from that under nominal conditions.

Often we meet cases when in a room or a laboratory we need to install an additional A/C system with minimal construction costs. In such a case, for instance the A/C air diffusers are installed in a place where it is possible and not where it is optimal from the point of thermal comfort. Very often the contractor is not willing even qualified to design an untypical installation which differs from a standard methodology. As a result, as soon as the A/C is put on service we find that it creates uncomfortable environment. The room has an acceptable mean temperature corresponding with a recommended value for a given work category and season but locally we meet places with a considerable discomfort - produced either by a high air velocity or a low temperature or combination of both which is the worst variant.

DESCRIPTION OF THE COMPUTER ROOM

The modeled computer room can be seen in the Fig. 1. The room has the dimensions LxWxH = 7 x 3.28 x 3.15m. The whole left wall is formed with a window that is oriented approximately in the W-W-N direction so the afternoon sun impacts on the window. Two long walls are solid walls. The entrance door is located in the second shorter wall, opposite to the window. On both sides of the door are cabinets. The wall mounted A/C diffuser is placed on this wall (the arrow on the right side in the figure 1) 2.3 m above the floor. There is a fan in the unit, which sucks the air from the upper side of the room and discharges it through adjustable vanes into the room. There are 6 workstations SGI and Intergraph and 1 PC. Among high heat load devices belong four monitors 21", one 19" and two 17". Beside the computers there are printers and scanners in the room but heat loads from these devices were neglected.

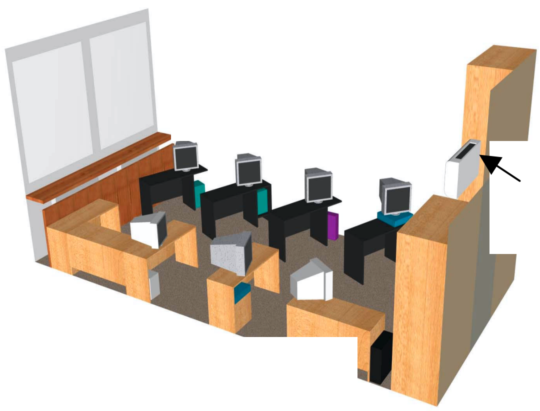

Fig.1 Schematic view of the computer room

The A/C device is a split type MUST10Q Emerson. Schematic view of the evaporator and fan unit is in Fig. 2. In the upper part there is a suction vent grill through which the air is brought into the evaporator. In the lower part there are adjustable vanes through which the cold air is discharged into the room. The vanes have a limited adjustment range. In the horizontal plane the range is -40° to +40° from the middle vertical plane. This range is in fact further limited as the diffuser is from one side bordered by the room wall and from the other side by a non-removable partition, both walls being very close to the diffuser. In the vertical plane the vanes are adjustable in the range of 45° from downward vertical. The limited adjustment range impacts negatively on the thermal comfort in the room namely in the working places located in the vicinity of the diffuser. The total refrigerating capacity of the A/C is 2790 W, maximum volumetric flow rate is 360 $m^3 hour^{-1}$, the cross sectional area of discharge vents is $0.03 m^2$, the mean discharge velocity is 3.33 ms^{-1}.

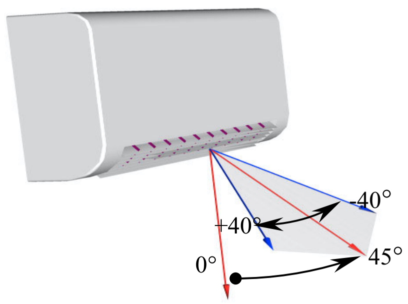

Fig.2 Schematic view of A/C diffuser

INPUT DATA - CALCULATION OF HEAT LOADS

We took into consideration only heat loads from computers, monitors and from solar radiation through the windows. Heat production from people were neglected (an average occupation in the room is 2 to 3 persons).

Monitors and computers:

All monitors have in the upper part of the casing a series of vent grills through which the hot air rises. Heat flux discharged from the monitors was calculated from the measured velocity and temperature of the air leaving the casing. Measurement was done in 9 locations close to the discharge vents with a hot wire anemometer LT Lutron AM-4204 that enables simultaneous measurement of velocity and temperature with resolution of 0.1K and 0.05 m/s^{-1}, respectively. The cross sectional area of flow was taken as 60% of the total grilled upper surface of the monitor. The average ambient temperature during measurement was between 28°C and 30°C. The similar approach was used to measure the heat load from computers. The velocity and temperature of the air leaving the computer casing were measured in different radial position close to the circular vent grills. Then the mean bulk temperature $\overline{T}_{out}$ and mean velocity $\overline{w}$ were calculated. The appropriate heat load was calculated from the formula:

$$\dot{Q} = \overline{w} A \rho c_p \left(\overline{T}_{out} - T_\infty\right)$$

The total heat loads from computers and monitors are in the Tab.1.

Tab.1 Total heat loads from computers and monitors in [W] - (for code numbers see Fig.6)

Device	1	2	3	4	5,6	7	Σ
monitor	80	100	100	80	160	100	620
computer	75	75	100	100	200	75	625

Solar irradiation through window:

Heat flux through the windows has two components - convective and radiative. The overall heat transfer by convection through the window is treated in the CFD model by defining the thermal resistance of the window and the outdoor temperature. For the solar radiation through the window, intensity of radiation must be determined. The calculation was done for the hottest day in the year, which is 21 July following the procedure described in Cihelka, 1994.

- Intensity of direct solar radiation

$$\dot{I}_{Dn} = 1350 \exp\left[-0.1z\left(\frac{\frac{16000-H}{16000+H}}{\sin h}\right)^{0.8}\right]$$

  Coefficient of atmospheric pollution z was assumed *4.4* for urban agglomeration, the altitude of city of Brno is H =250m and h is the elevation of sun for 50° of northern latitude, varying during day.

- Intensity of direct solar radiation on the window surface is $\dot{I}_D = \dot{I}_{Dn} \cos\theta$

  where θ is the angle between the solar radiation beam and the normal to the given surface. The cosine is calculated from the formula:

$$\cos\theta = \cos h - \cos(a - a_s)$$

  where a is the solar azimuth measured clockwise from south and a_s is azimuth angle measured from south to the normal to the surface (in our case 148°).

- Intensity of diffuse solar radiation is calculated from the formula

$$\dot{I}_d = \left(1350 - 0.5\dot{I}_{Dn}\right)\frac{\sin h}{5}$$

The total solar irradiation is then

$$\dot{I} = \dot{I}_D + \dot{I}_d$$

The maximum solar intensity impacting on the window is at 5:00PM, namely $\dot{I}_D = 293$ [Wm^{-2}] at the angle $\theta = 29°$ and $\dot{I}_d = 90$ [Wm^{-2}].

MATHEMATICAL MODEL AND BOUNDARY CONDITIONS

The problem was solved using a set of equations for incompressible, steady turbulent 3D flow with standard k-ε model of turbulence. The equation for a general variable ϕ has the well-known form:

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

where the variable ϕ substitutes velocity components *u, v, w*, temperature *T* and kinetic energy of turbulence *k* and its rate of dissipation *ε*.

The equations for k-ε model of turbulence are as follows:

$$\frac{\partial}{\partial x_i}(\rho u_i k) = \frac{\partial}{\partial x_i}\left(\frac{\mu_{ef}}{\sigma_k}\frac{\partial k}{\partial x_i}\right) + P + P_B - \rho\varepsilon \qquad (2)$$

$$\frac{\partial}{\partial x_i}(\rho u_i \varepsilon) = \frac{\partial}{\partial x_i}\left(\frac{\mu_{ef}}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial x_i}\right) + C_1\frac{\varepsilon}{k}(P + C_3 P_B) - C_2\frac{\rho\varepsilon^2}{k} \qquad (3)$$

where *P* is the production of kinetic energy of turbulence due to shear and P_B is the production due to thermal gradients. Constants are: C_1=1.44, C_2=1.92, C_3=1.44 (if $P_B > 0$), C_3=0 (if $P_B \le 0$), C_μ=0.09, σ_k=1.0, σ_ε=1.3, Pr_t=0.85.

The set of equations was solved using the control volume method and CFD code StarCD. The computational model of the computer room can be seen in Fig. 3. The geometry is simplified in the sense that all desks are modelled as solid boxes with slots on one side representing suction and discharge vents of computers. Monitors are modelled again as solid boxes with upper surface as discharge vents. All vents are prescribed with the "inlet" boundary condition, i.e. the discharge vents are prescribed with measured velocities and temperatures of air. The velocities of air through the suction vents into the computer are calculated from the discharged mass flow and appropriate cross sectional area of suction vents. The discharge vent of the A/C unit is prescribed with velocity and temperature that is approximately by 15K above the ambient temperature. The suction vent into the A/C unit is prescribed with pressure boundary. The partition that is on the right side of the A/C unit was modelled as a baffle with prescribed thermal resistance. The similar condition was used for the additional guiding plate (see the Variant IV) designed and put below the A/C unit to improve the flow and temperature fields in the room. All sidewalls, the ceiling and the floor were prescribed with appropriate thermal resistance and with outside temperature 24°C.

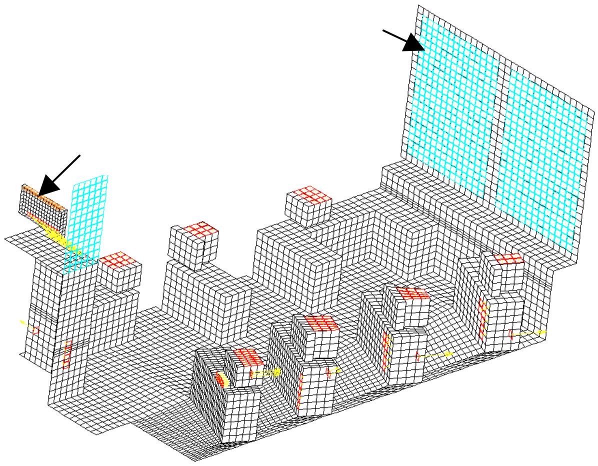

Fig. 3 Computational model of the room

SOLUTION SCENARIOS

The principal problem of the computer room is that in certain working places close to the discharge from the A/C unit, the temperature is too low and people working there are feeling a cold draught. Several scenarios were set to predict both velocity and temperature fields in the room in several vertical and horizontal planes - see Fig. 4 and 5. Namely we were interested in the flow and temperature fields at the level of ankle (horizontal plane 7 at 0.1m from the floor), abdomen (plane 6 at 0.65m) and head (plane 5 at 1.2m) of a sitting person. The vertical planes are indicated as 1, 2, 3, and 4 and are drawn through the working positions.

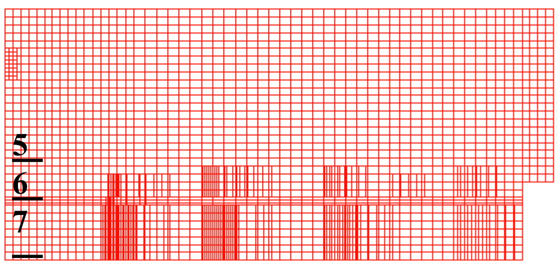

Fig.4 Horizontal planes

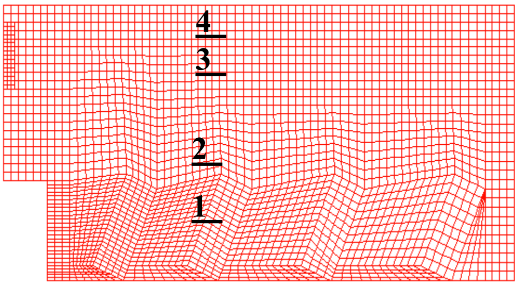

Fig.5 Vertical planes

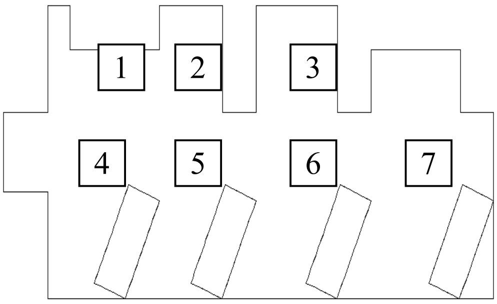

Fig.6 Position of working places

The individual working places are indicated in Fig.6. In total 5 basic variants were solved, several of them with sub-variants - see Tab.2.

Tab.2 Scenarios

Variant		Description
I		A/C off, no solar radiation through window
II		A/C off, solar radiation through windows
III	**A**	A/C on, A/C vanes 45° down, 0° in horizontal plane
	B	A/C on, A/C vanes 45° down, +30° towards the centre of the room in horizontal plane
IV	**A**	Guiding plate under A/C discharge, otherwise as III.A (for plate location see Fig.7)
	B	Guiding plate under A/C discharge, otherwise as III.B
V	**A**	Reverse suction and discharge of A/C, otherwise as III.A, vanes 45° up
	B	Reverse suction and discharge of A/C, otherwise as III.B, vanes 45° up
	C	Reversed suction and discharge of A/C, discharge split into two streams, one in 0°, one in -30° in horizontal plane, vanes 45° up

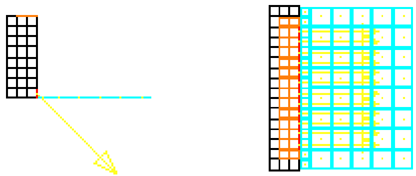

Fig.7 Elevation and plane views of the guiding plate under the A/C

RESULTS AND THEIR DISCUSSION

Variant I:

This is the variant most frequently met before the A/C was installed. In Fig. 8 we can see the temperature field in vertical planes 1, and 4 (for code numbers see Fig.5). We can see hot plumes rising from computers and monitors. Temperatures in the zone occupied by sitting people are high - up to 31°C in the level of head.

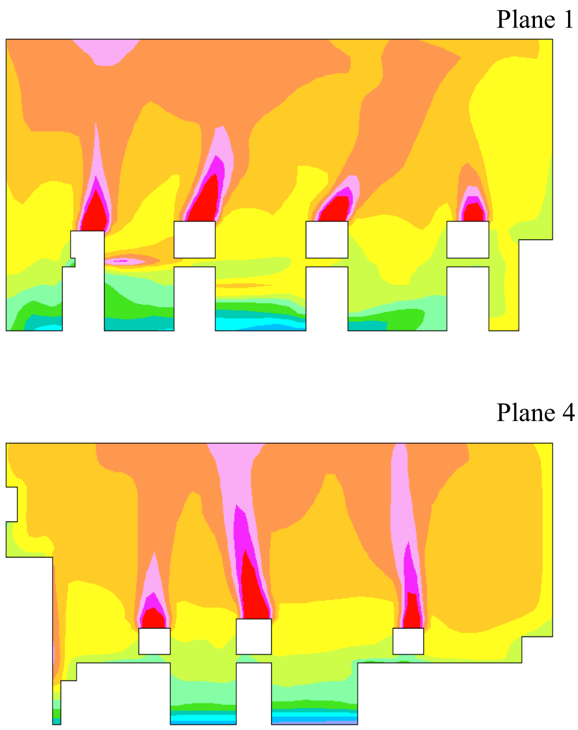

Fig. 8 Temperature field in the vertical planes 1 and 4 for Variant I

Note: The temperature scale for the basic Variant I in the Fig.8 is different from all other Variants II to V (their temperature scale is in Fig.19) due to very different level of temperatures. If the scale is kept the same we are not able to see all details of hot plumes.

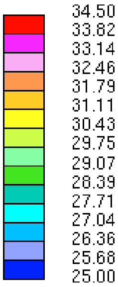

Fig. 8b Temperature scale for Variant I

Variant II:

The temperature field looks similar to that of the Variant I but temperatures are approximately by 1.5K higher. Also the plumes rising from the computers are less distinctive due to the lower temperature difference and lower driving buoyancy forces.

Variant III.A:

In this case the cold air from the A/C discharge is directed into the room under the angle of 45° down but not deflected in the horizontal plane. A typical temperature field is in Fig.9 (vertical plane 3) where we can see a very distinctive cold jet discharged from the A/C and penetrating far into the room. The working positions 1, 2 and 4 (for code numbers see Fig.6) are particularly attacked with the cold air and people sitting there feel uncomfortable. This can be seen from Fig.10. Temperature at the head level in these positions is about 22°C (for temperature scales see Fig. 19). Velocities in these positions approach 0.5m/s^{-1} - see Fig. 11 with horizontal plane 6 (for velocity scales see Fig.19). Particularly, people working in the positions 1 and 2 where they feel a cold draught of 22°C with velocity of 0.4m/s^{-1} can have health problems.

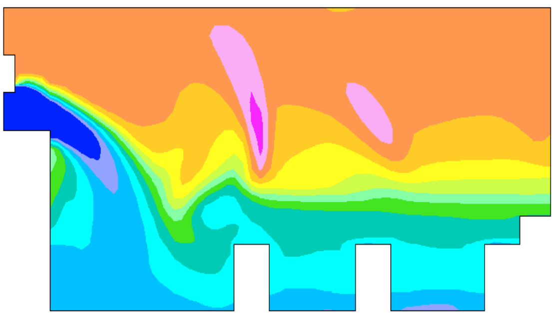

Fig.9 Temperature field in the vertical plane 3 - variant III.A

Variant III.B:

This variant differs from the previous one only in that the discharged cold air from the A/C is deflected in the horizontal plane by 30° into the middle of the room. Both, the temperature and velocity fields are very similar as in the Variant III.A and we do not show them here.

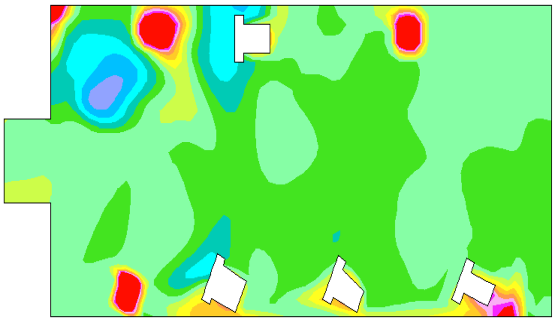

Fig. 10 Temperature field in the horizontal plane 5 - variant III.A

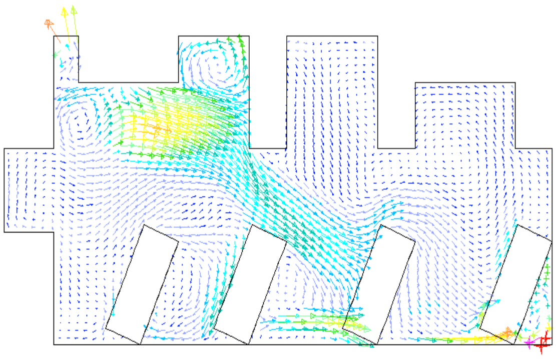

Fig. 11 Velocity field in the horizontal plane 6 - variant III.A

Variant IV.A:

This variant was designed on the basis of the previous predictions and was expected to improve thermal comfort in the room mainly in the working positions 1 and 2. Under the discharge vent of the A/C was placed a guiding plate whose purpose was to guide and throw the cold air into a more distant area in the room before it falls down. On its way the cold air mixes with the hot plumes rising from computers and warms up. In Fig.12 we can see the temperature field in the plane 3 (compare with Fig.9). The cold jet passes a longer way and touches the working positions with temperatures higher by approximately 2K but with a high velocity of 0.45m/s^{-1} (for temperature and velocity scales see Fig. 19). The zone of high velocity as was seen in Fig.11 moves up from the horizontal plane 6 to the plane 5 - see Fig.13.

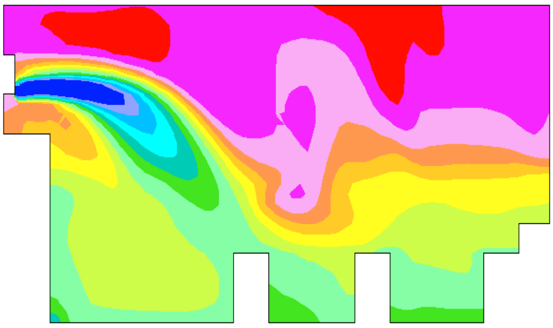

Fig. 12 Temperature field in the vertical plane 3 - variant IV.A

The variant IV.A did not resolved completely the problem of a cold attack on sitting people. The jet from the A/C touches down with temperature of 24°C but still with high velocity. The situation improves in working places 4 to 7 where the temperature is in the range of 24-24.5°C with low velocities.

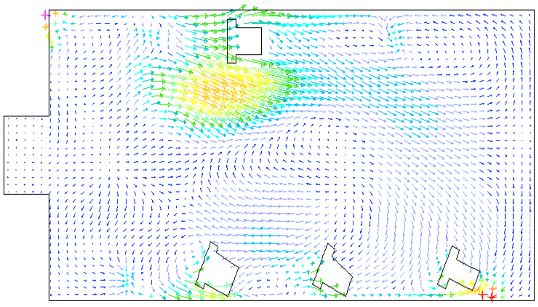

Fig. 13 Velocity field in the horizontal plane 5 - variant IV.A

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Variant IV.B:

This variant is same as the Variant IV.A (with guiding plate under the A/C discharge) and as variant III.B (A/C vanes 45°down, +30° towards the centre of the room in the horizontal plane). This variant shows as the best from all the previous. The temperature field in the vertical plane 3 that is most critical is well stratified and in the limits of 25 to 25.5°C - see Fig. 14. The velocity field shows its maximum off the working places, in the middle of the room -see Fig. 15. In this variant the temperature field is almost optimal in the level of sitting people (24 to 26°C). From the point of velocity, only at working places 5 and 6 the velocity close to the floor is up to 0.25 m/s^{-1} what might create a feeling of a cold draught.

Variants V.A, B, C:

In these variants the operation of the A/C unit was reversed. The suction vents now work as discharge vents and vice verse. The discharged air is directed

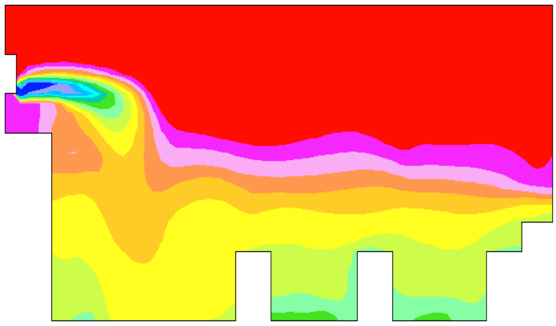

Fig. 14 Temperature field in the vertical plane 3 - variant IV.B

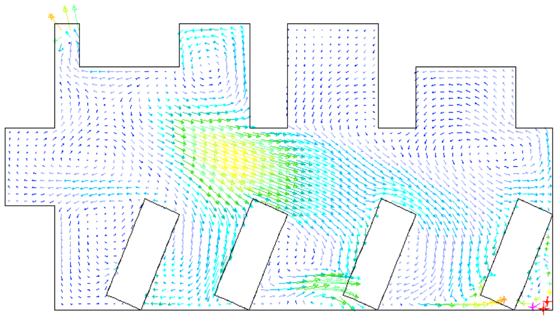

Fig. 15 Velocity field in the horizontal plane 6 - variant IV.B

under the angle of 45° towards the ceiling from which it is expected to rebound and fall down into the room. The variants V.A and V.B do not show any improvements and we will not describe their results in details. The variant V.C in which the discharged jet is split into two streams under various angles (see Tab. 2) shows the best features. The temperature field in the vertical planes 2 and 3 are well stratified in the range of 22 to 24.5 °C - see Fig. 16.

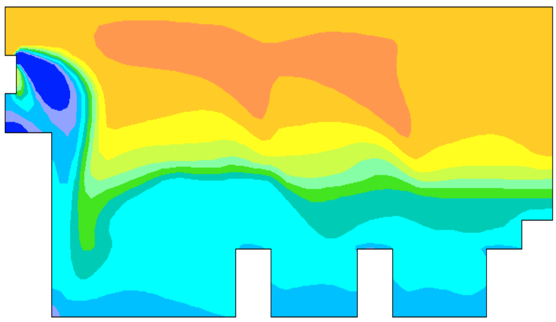

Fig. 16 Temperature field in the vertical plane 3 - variant V.C

The velocity field in all working places at the level of head is favorable and is kept below 0.15 m/s^{-1} - see Fig. 17. Close to the floor - see Fig.18 - the velocity is somewhat higher in working places 5 and 6 (about 0.3 m/s^{-1}).

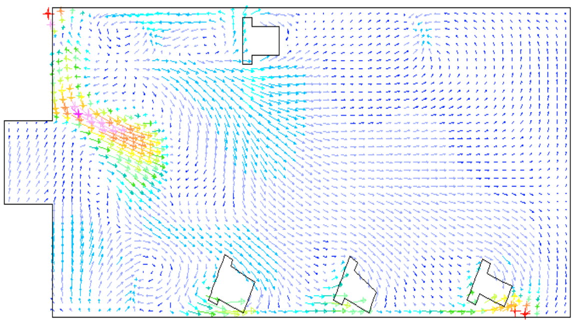

Fig. 17 Velocity field in the horizontal plane 5 - variant V.C

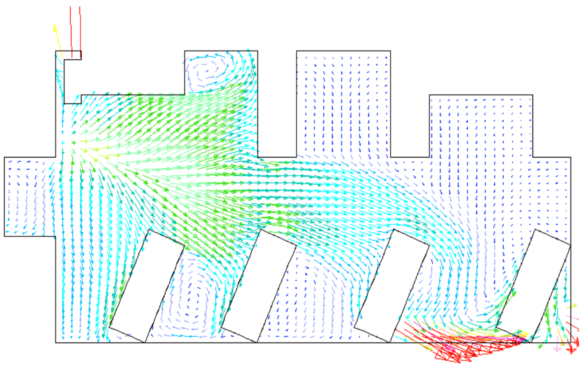

Fig. 18 Velocity field in the horizontal plane 7 - variant V.C

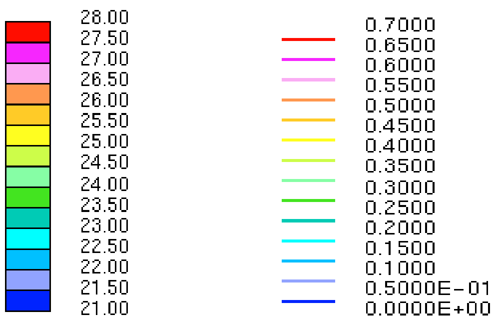

Fig.19 Temperature (left) and velocity (right) scales for Variants II to V.

CONCLUSIONS

The main objectives of the work were to predict and improve the temperature and velocity field in the computer room using CFD modelling. Commercial CFD code StarCD was used. The scenarios included different adjustments of the A/C discharge vanes, an additional guiding vane under the A/C discharge and reversed operation of the A/C.

Assessment of the results was done on the basis of the recommended values of temperatures and velocities for the specific category of work from Jokl, 1997. For the computer room, the optimal operational temperature is in the range 23-26°C (acceptable values are 20-28°C) and for velocities the range is 0.1-0.2 m/s^{-1}.

From the predictions we can conclude:

1. Discharge vanes should at least enable an adjustment angle of 0° to horizontal
2. If not, an additional guiding plate should be placed under the A/C discharge.
3. An alternative solution is to discharge the cold air upwards, e.g. to reverse the operation of the A/C unit
4. Non-standard design is often required when installing the A/C units. The installation should extend the trajectory of the cold air and let him to mix with the hot air rising from local heat sources before it falls down.

ACKOWLEDGEMENT

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REFERENCES

Cihelka, J., "Solar thermal technique" (in Czech), Malina publishing House, Prague, 1994

Jokl, M, "Optimal microclimate conditions in working places" (in Czech), Heating, Ventilating, Installation, vol.2, p.69-72, 1997

Richtr, J., Study of the influence of adjustment of A/C vanes in a computer room, MS Thesis, Brno University of Technology, 2000

StarCD, User Guide, version 3.100a, Computational Dynamics Ltd., London

NOMENCLATURE

A	cross sectional area [m^2]
c_p	isobaric thermal capacity [Jkg^{-1}K^{-1}]
k	turbulent kinetic energy [m^{2}s^{-2}]
$\dot{Q}$	heat flux [W]
S_ϕ	source term in eq.1
T_∞	ambient temperature [°C]
$\overline{T}_{out}$	mean bulk outlet temperature [°C]
u_i	velocity components [ms^{-1}]
$\overline{w}$	mean velocity [ms^{-1}]
x_i	Cartesian coordinate system

Greek letters

ε	dissipation rate of turbulent kinetic energy [m^{2}s^{-3}]
ϕ	general variable
ρ	density of air [kgm^{-3}]
Γ	diffusion coefficient in eq.1