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Investigation of air quality, comfort parameters and effectiveness for two floor level air supply systems in classrooms

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ABSTRACT

The method of distributing the outdoor air in classrooms has a major impact on indoor air quality and thermal comfort of pupils. In a previous study, Karimipanah *et al.* (2000, 2005) presented results for four and two types of air distribution systems tested in a purpose built classroom with simulated occupancy as well as CFD modelling.
In this paper, the same experimental setup has been used to investigate the indoor environment in the classroom using confluent jet ventilation, see also Cho, *et al.* 2004. Measurements of air speed, air temperature and tracer gas concentrations have been carried out for different thermal conditions. In addition, 56 cases of CFD simulations have been carried to provide additional information on the indoor air quality and comfort conditions throughout the classroom, such as ventilation effectiveness, air exchange effectiveness, effect of flow rate, effect of radiation, effect of supply temperature, etc., and these are compared with measured data.

KEYWORDS

Air distribution index, School Ventilation, Full-Scale Measurements, CFD, Displacement System, Wall Confluent Jets System, Ventilation Effectiveness.

INTRODUCTION

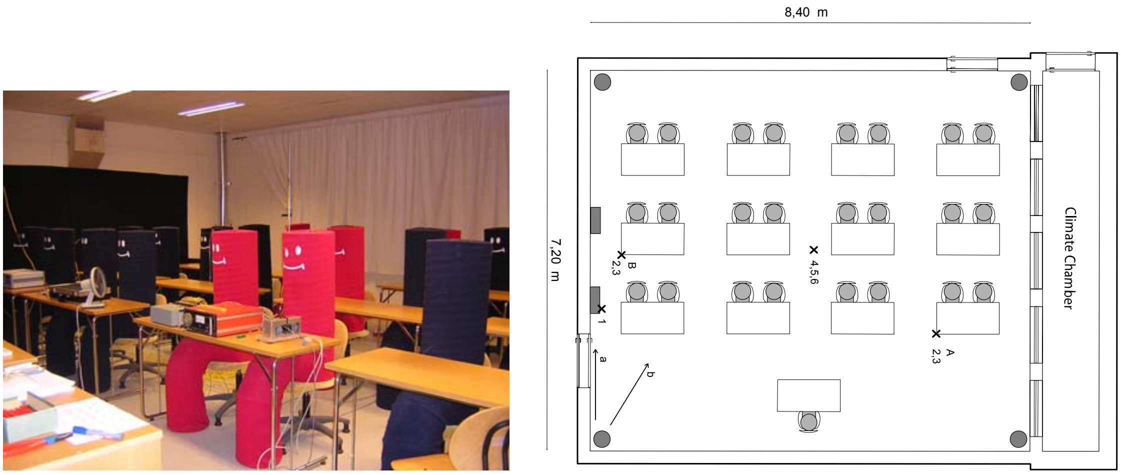
There have been many studies on the indoor environment in office buildings but hardly any has been done in school buildings. Some studies estimate that more than 50% of school children have some kind of allergy or asthma. Hence, there is a need to consider ventilation in schools as these buildings often have much higher occupancy density than office buildings and results from the latter studies may not be applicable to classrooms. Good indoor air quality can actually have a positive impact not just on pupils' health but can also improve learning, Rosenfeld (1989). It has also been shown by Koo *et al.* (1997) in Hong Kong that the frequencies of symptoms among pupils in air-conditioned classrooms were higher than those in naturally ventilated classrooms.
Because of the large number of obstacles and heat sources (occupants and computers, etc.) found in classrooms, air distribution using conventional methods may not be adequate. In order to keep the temperature at reasonable level large quantities of air must be supplied. As a consequence draught may occur using conventional air distribution systems. The study of air distribution in such rooms is also more complicated and challenging. A new ventilation system "Confluent Jets Ventilation" developed in Sweden has been evaluated both by measurement in a mock-up classroom and using computational fluid dynamics (CFD). Some of the results that have been obtained so far are presented in this paper.
Ventilation efficiency and occupied-zone velocities for the new Confluent Jets System were measured and the results compared with CFD calculations and with a displacement ventilation system that was used in the classroom.

EXPERIMENTAL SETUP AND MESAUREMENTS

Test Room

The new air distribution system has the following properties:
Confluent Jets System blowing to the wall at the corners - The supply device is a duct with specially designed circular nozzles in 1 to 6 rows which ends at a certain distance above the floor, see Figs. 1 and 2. The jets are supplied with high velocity (momentum), compared with jets from mixing and

displacement ventilation devices, of up to 12 m/s. Therefore these can be considered to be high momentum supply devices with stratification capability. This method of air distribution combines some positive aspects of both mixing and displacement systems. It produces a clean air zone in the lower part of the occupied zone and exhibits a higher air-exchange effectiveness than a mixing system and slightly better than a displacement system. At the same time it has some of the properties of mixing ventilation by causing entrainment of the ambient air into the jet. Another advantage is that heated air can be supplied with this system in contrast to a displacement system which is only useful for cooling. However, the somewhat higher velocities generated by this system very close to the wall at floor level, although this is outside the occupied zone, can be challenging from thermal comfort considerations.



(a) The classroom with person simulators (b) Plane view of the classroom with supply positions.

Fig. 1 Experimental setup used in the study.

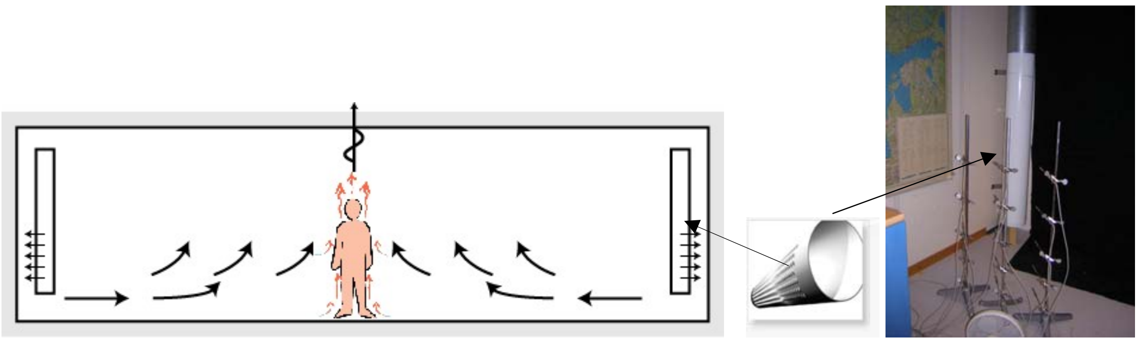


Fig. 2. Confluent Jets directed towards the wall in the room corners

Test Data

A classroom mock-up having dimensions 8.4 x 7.2 x 3.0 m ceiling height with a volume of 170 m was used for the tests, see Figs. 1 and 2. To produce a heat-load corresponding to a fully occupied classroom 25 person-simulators were placed in the room. The classroom has an “outdoor facing” wall, provided with four triple glazed windows, located next to a temperature controlled climatic chamber whose temperature can be varied to simulate winter and summer climates. No heaters below the windows have been used for winter conditions. The heat load consisted of 25x 95 = 2375 W generated by people and 525 W generated by lighting giving a total load of 2 900 W (48 W/m^2).

The air distribution systems tested are as follows:

1. The first case is an original ventilation system consisting of 4 low momentum supply devices with flow rate of 50 l/s per supply device.
2. A high momentum supply device called Softflo S11 (Wall Confluent Jets) blowing to the corners tested with the following configurations: a) 4 devices placed at the corners with 50 l/s

per supply device at 100 Pa. b) The same configuration as case (a) but reducing the flow rate to 30 l/s for each device. c) Two devices placed at two corners of the room in the wall close to the teacher with 100 l/s per device at 100 Pa. d) the same as case (c) but with the flow rate reduced to 60 l/s per device. All test configurations and number of devices are described in table 1.

The temperature in the climate chamber (outdoor temperature) is -16°C

The supply air temperatures to the classroom are 16°C at flow rate 200 l/s and 13°C at 120 l/s.

All measurements were carried out under steady state conditions.

Measurements

The following measurements have been carried out for both the low momentum supply terminals (Displacement System) and the high momentum supply terminals (Confluent Jets System) in operation:

- Air Exchange Efficiency measurement with tracer gas (decay method)
- Local mean age of air measurement with tracer gas (decay method) at 6 points
  Measuring point 1 at exhaust (= nominal time-constant, τ_n)
  Measuring point 2 at position A (Fig. 1b) 0.1 m above the floor level
  Measuring point 3 at position A (Fig. 1b) 1.2 m above the floor level
  Measuring points 4, 5 and 6 in the middle of the room with heights respectively of 0.1m, 1.2m and 1.8m above the floor level
- Air velocity measurements for the near-zone at (a) and (b), Fig. 1b
- 150 measurements for 5 minutes with 2 s intervals.

VENTILATION INDICES

Air Exchange Efficiency

The room mean air exchange efficiency is defined by:

$$\varepsilon_a = \frac{\tau_n}{2\cdot \langle \bar{\tau} \rangle}\cdot 100\,[\%] \qquad (1)$$

where τ_n the nominal time-constant and $\langle \bar{\tau} \rangle$ is the mean age of room air.

The Local Air Change Index (ε_p) $= \frac{\tau_n}{\bar{\bar{\tau}}_p}$ (2)

i.e., ε_p is the ratio of nominal time constant (τ_n) to the local mean age of air ($\bar{\tau}_p$).

Air Distribution Index (ADI)

To assess the effectiveness of a ventilation system in both measurement and CFD simulation, the effectiveness for heat removal (ε_t) and contaminant removal (ε_c) are used together with the predicted percentage of dissatisfied (*PPD*) for thermal comfort and percentage of dissatisfied (*PD*) for air quality. ε_t and ε_c are defined by:

$$\varepsilon_t = \frac{T_o - T_i}{T_m - T_i} \qquad (3)$$

and

$$\varepsilon_c = \frac{C_o - C_i}{C_m - C_i} \qquad (4)$$

In equations (3 & 4), *T* is temperature (^{o}C), *C* is the contaminant concentration (ppm), subscripts *o,i* and *m* denote outlet, inlet and mean value for the occupied zone (to a height of 1.8m). ε_t is similar to a heat exchanger effectiveness and is a measure of the heat removing ability of the system. ε_c is a measure of how effectively the contaminant is removed. The values for ε_t and ε_c are determined by heat and contaminant sources, the method of room air distribution, room characteristics, etc. However, high values do not always give a good indication of the thermal comfort and air quality in the occupied zone.

Fanger (1972) has developed expressions for the percentage of dissatisfied (*PD*) with the indoor air quality and the predicted percentage of dissatisfied (*PPD*) with the thermal environment given by Eqs. (5) and (6).

$$PD = 395 \cdot \exp(-1.83\ \dot{v}^{0.25}) \tag{5}$$

$$PPD = 100 - 95 \exp -\{0.03353\ (PMV)^4 + 0.2179\ (PMV)^2\} \tag{6}$$

where $\dot{v}$ is the ventilation rate (ls^{-1}) and *PMV* is the Predicted Mean Vote as defined in ISO 7730 (1994) and the recommended *PPD* limit for ideal thermal environment is 10%, corresponding to $-0.5 \le PMV \le 0.5$. Thus, low values for both indices guarantee a good indoor air quality and thermal comfort.

The comfort number, N_t, and the air quality number, N_c, (Awbi 1998a) combined with *PPD* and P*D* respectively are useful to examine the quality of a ventilation system. These are defined as:

$$N_t = \frac{\varepsilon_t}{PPD} \tag{7}$$

and

$$N_c = \frac{\varepsilon_c}{PD} \tag{8}$$

These two numbers can be combined into a single parameter which determines the effectiveness of an air distribution system in providing air quality and thermal comfort in the form of a ventilation parameter defined as:

$$ADI = \sqrt{N_t \times N_c} \tag{9}$$

This parameter is also called the Air Distribution Index (*ADI*), Awbi (2003).

CFD CALCULATIONS

The measured wall temperatures have been used as boundary conditions for the CFD simulations. Predicted and measured quantities are: air velocity, air temperature, ventilation effectiveness and local mean age of air.

The numerical simulations were done using the CFD code VORTEX 4.0 (Awbi, 2005) that has been developed for the simulation of airflow, heat transfer and concentration in enclosures. The code uses two turbulence models: the standard κ-ε and RNG turbulence model. The program has been developed for ventilation research, which may be more suited to ventilation simulations than the more general-purpose codes. In the simulations, the measured mean surface temperatures of all six room surface have been used as boundary conditions. The number of nodes used was 63 x 30x 52 giving a resolution of 0.13 m in the horizontal plane and 0.10 m in the vertical direction. The near-wall nodes were located 10mm from the surfaces as recommended by Awbi (1998b). In the case with Wall Confluent Jets the distance of the duct from the wall was 110 mm and the distance from the floor level for all cases was 300 mm as in the measurements.

In the VORTEX code the mean radiant temperature at each computation cell is calculated using a radiation exchange model that is based on the radiosity of room surfaces (Gan and Awbi, 1994).

Figure 3 shows a CFD model of the test room shown in Fig. 1.

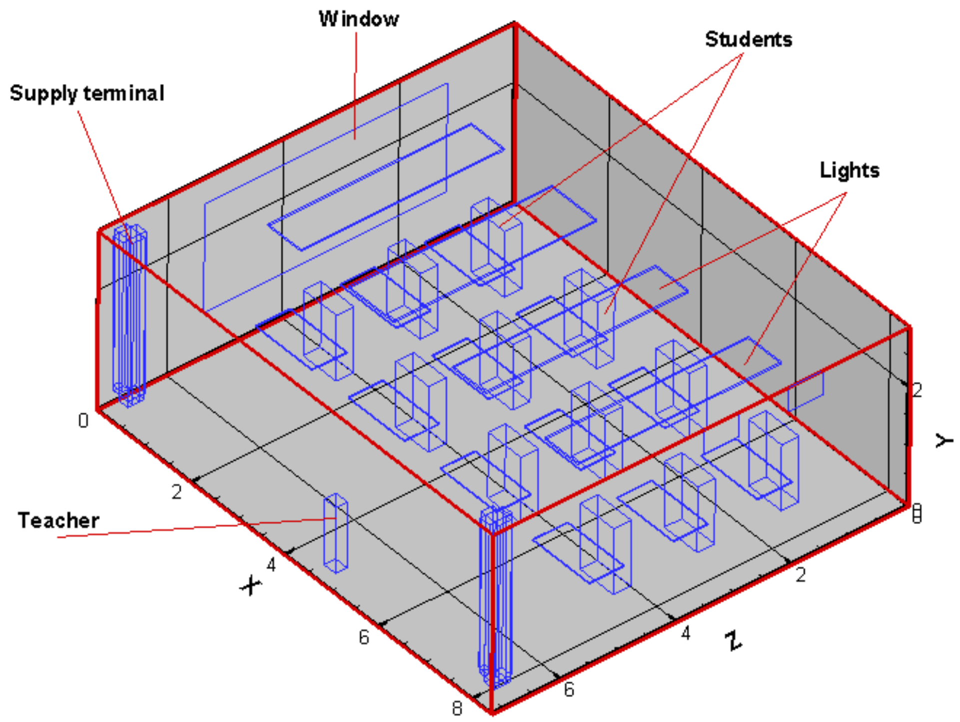


Fig. 3. Model for CFD Calculations. Each box represents 2 students

RESULTS

Experimental Data

Table 1 shows results from the tracer gas measurements for a displacement ventilation system and two types of Confluent Jets systems. Points 2 and 3 are located at position A or B with 0.1 m and 1.2 m above floor level. Points 4, 5 and 6 are located in the middle of the room at 0.1 m, 1.2 m and 1.8 m above floor level. All these positions are shown in Fig. 1b.

The air exchange effectiveness obtained from the CFD calculations is also given in table 1 for comparison. The CFD results are in good agreement with the measured values. The results show that the confluent jets systems give slightly better effectiveness than the displacement system and the values of local air exchange index also indicate good air distribution in all the zones considered.

Table 1. Measurements in Classroom at Centre for Built Environment, BMG, Gävle University

Case	No. of Terminals x flow rate [l/s]	Position (see Fig. 3)	Local Air Exchange Index ($\frac{\tau_n}{\tau_p}$)					Air Exchange Efficiency $\frac{\tau_n}{2\langle\bar{\tau}\rangle}\times 100$	
			Point no. 2	Point no. 3	Point no. 4	Point no. 5	Point no. 6	EXP.	CFD
DV4C_A50	4 x 50	A+middle	1.28	0.68	1.19	0.95	0.79	43	48
CF4C_A50	4 x 50	A+middle	1.195	1.155	1.32	1.35	1.145	56	54
CF4C_B50	4 x 50	B+middle	1.48	1.28	1.10	1.19	1.06	73	54
CF4C_B32	4 x 32	B+middle	1.755	1.61	1.43	1.36	1.26	50	52
CF2_B60	2 x 60	B+middle	1.75	1.52	1.58	1.43	1.22	51	50
CF2C_B100	2 x 100	B+middle	1.20	1.1	1.11	1.12	1.01	51	52

DV = Displacement system ***CF= Confluent Jets System***

CFD Predictions

Tables 2 and 3 show the values of the ventilation indices given by equations (3) to (9) obtained from the CFD calculations for comparison between the two ventilation systems used. A large data base was created and 56 cases have been run for these comparisons. The number of supply devices, supply temperature and the flow rates were varied to study the performance of the two systems. Although the results do not show large differences between the two systems, in all the cases considered the confluent jets ventilation system gives more promising results. An interesting finding here is that when decreasing the flow rate the comfort and air quality parameters are decreased drastically which can be a warning signal to those who contemplating a reduction in outdoor air supply rates as in so doing, pupils' health and their performance could be affected.

To have a better interpretation of the explored CFD results, the cases shown in tables 2 and 3 were compared by means of factors like the effect of radiation, number of supply devices installed at the room corners, effect of inlet temperature and flow rate. A comparison with the experimental results (tables 1 and 4) shows that the velocities and temperature fields are predicted better by using the RNG turbulence model. Thus, analysis of the results is focused on the RNG model data in table 3.

It is worth mentioning that in these tables the case names are related to the following conditions:

CF = Confluent jets system; DV= Displace Ventilation system; 2C and 4C means 2 and 4 supply terminals used at the corners; T13 and T16 means supply temperature 13 °C and 16 °C; 4x50, 2x100, 2x150, 2x60 and 4x30 are number devices times flow-rate per device in l/s.

Figures 4 and 5 show a comparison of the temperature fields between the two air distribution systems using the RNG turbulence model. As shown, the temperature fills spread more evenly with the CJS than with the DVS.

Figures 6 and 7 show a comparison of the velocity distribution in a plane 0.1 m above the floor for the two air distribution systems. Although the velocity for the CJ system is slightly higher than that for the DV system the values are below the recommended ISO 7730 (1994) limit of 0.25 ms^{-1} over most of the floor.

Table 2. Comparison of comfort and air quality between different cases for a displacement system and a confluent jets system (κ-ε turbulence model)

COMFORT AND AIR QUALITY							
VENTILATION EFFECTIVENESS EQUATION (4) ε_c [%]	HEAT REMOVAL EFFICIENCY EQUATION (3) ε_t [%]	COMFORT NUMBER FOR AIR QUALITY N_c [-]	COMFORT NUMBER FOR THERMAL SENSATION N_t [-]	AIR DISTRIBUTION INDEX (ADI) [-]	TURBULENCE MODEL	CASE	NO. OF DEVICES x FLOWRATE (l/s)
102.3	105.1	5.6	12.7	8.5	κ-ε without Radiation	CF4CT13	4 x 50
105.8	111.7	5.8	17.7	10.1	κ-ε with Radiation	CF4CT13	4 x 50
98.5	99.5	5.4	12.4	8.2	κ-ε without Radiation	DV4CT13	4 x 50
99.8	108.7	5.5	16.4	9.5	κ-ε with Radiation	DV4CT13	4 x 50
107.8	125.1	5.9	16.6	9.9	κ-ε without Radiation	CF4CT16	4 x 50
108.4	127.6	6.0	14.1	9.2	κ-ε with Radiation	CF4CT16	4 x 50
98.4	113.5	5.4	15.8	9.2	κ-ε without Radiation	DV4CT16	4 x 50
100.5	122.0	5.5	11.7	8.0	κ-ε with Radiation	DV4CT16	4 x 50
111.8	98.0	6.1	12.2	8.6	κ-ε without Radiation	CF2CT13	2 x 100
112.7	104.1	6.2	13.6	9.2	κ-ε with Radiation	CF2CT13	2 x 100
94.8	92.0	5.2	11.7	7.8	κ-ε without Radiation	DV2CT13	2 x 100
96.6	100.7	5.3	11.9	8.0	κ-ε with Radiation	DV2CT13	2 x 100
110.3	83.8	8.5	10.4	9.4	κ-ε without Radiation	CF2CT16	2 x 150
107.6	83.2	8.2	7.2	7.7	κ-ε with Radiation	CF2CT16	2 x 150
93.7	85.8	7.2	10.2	8.5	κ-ε without Radiation	DV2CT16	2 x 150
97.4	93.9	7.4	12.2	9.5	κ-ε with Radiation	DV2CT16	2 x 150
108.3	114.0	5.9	15.8	9.7	κ-ε without Radiation	CF2CT16	2 x 100
119.7	120.9	6.6	10.5	8.3	κ-ε with Radiation	CF2CT16	2 x 100
92.1	145.2	5.1	14.5	8.6	κ-ε without Radiation	DV2CT16	2 x 100
99.9	122.2	5.5	11.3	7.9	κ-ε with Radiation	DV2CT16	2 x 100
113.5	163.8	4.5	19.7	9.4	κ-ε without Radiation	CF4CT13	4 x 30
102.8	142.0	4.1	11.7	6.9	κ-ε with Radiation	CF4CT13	4 x 30
104.8	151.9	4.0	20.0	8.9	κ-ε without Radiation	DV4CT13	4 x 30
105.3	169.1	4.0	22.7	9.5	κ-ε with Radiation	DV4CT13	4 x 30
117.5	165.4	4.4	20.8	9.6	κ-ε without Radiation	CF2CT13	2 x 60
125.6	161.9	4.8	18.7	9.4	κ-ε with Radiation	CF2CT13	2 x 60
98.8	128.8	3.8	18.7	8.8	κ-ε without Radiation	DV2CT13	2 x 60
97.6	136.5	3.7	7.4	5.2	κ-ε with Radiation	DV2CT13	2 x 60

CF = Confluent Jets; DV= Displacement; T16 and T13 = Supply temp 16and 13°C, 4 and 2 are the number of supplies respectively.

*Table 3. Comparison of comfort and air quality between different cases for a displacement system and a confluent jets system (**RNG** turbulence model).*

COMFORT AND AIR QUALITY							
VENTILATION EFFECTIVENESS EQUATION (4) ε_c [%]	HEAT REMOVAL EFFICIENCY EQUATION (3) ε_t [%]	COMFORT NUMBER FOR AIR QUALITY N_c [-]	COMFORT NUMBER FOR THERMAL SENSATION N_t [-]	AIR DISTRIBUTION INDEX (ADI) [-]	TURBULENCE MODEL	CASE	NO. OF DEVICES x FLOWRATE (l/s)
114.8	112.9	6.3	12.5	8.9	RNG without Radiation	CF4CT13	4 x 50
104.6	103.4	5.7	12.6	8.6	RNG with Radiation	CF4CT13	4 x 50
87.9	58.4	4.8	8.9	6.6	RNG without Radiation	DV4CT13	4 x 50
94.4	96.2	5.2	10.6	7.4	RNG with Radiation	DV4CT13	4 x 50
113.6	129.8	6.2	16.7	10.2	RNG without Radiation	CF4CT16	4 x 50
118.9	130.4	6.5	14.8	9.8	RNG with Radiation	CF4CT16	4 x 50
101.6	115.0	5.6	15.7	9.3	RNG without Radiation	DV4CT16	4 x 50
105.3	124.8	5.8	12.3	8.4	RNG with Radiation	DV4CT16	4 x 50
107.9	122.8	5.9	12.6	8.6	RNG without Radiation	CF2CT13	2 x 100
124.9	105.9	6.9	14.0	9.8	RNG with Radiation	CF2CT13	2 x 100
98.2	94.6	5.4	11.5	7.9	RNG without Radiation	DV2CT13	2 x 100
101.1	102.9	5.5	12.5	8.3	RNG with Radiation	DV2CT13	2 x 100
104.2	71.8	7.9	9.5	8.7	RNG without Radiation	CF2CT16	2 x 150
113.3	84.2	8.6	7.2	7.9	RNG with Radiation	CF2CT16	2 x 150
97.6	88.3	7.4	10.3	8.7	RNG without Radiation	DV2CT16	2 x 150
102.7	95.7	7.8	12.4	9.8	RNG with Radiation	DV2CT16	2 x 150
119.5	121.5	6.6	16.1	10.3	RNG without Radiation	CF2CT16	2 x 100
134.9	122.6	7.4	10.6	8.9	RNG with Radiation	CF2CT16	2 x 100
95.7	104.5	5.3	14.7	8.8	RNG without Radiation	DV2CT16	2 x 100
106.3	121.5	5.8	10.3	7.8	RNG with Radiation	DV2CT16	2 x 100
98.2	112.2	3.9	16.3	7.9	RNG without Radiation	CF4CT13	4 x 30
105.6	137.1	4.2	9.0	6.1	RNG with Radiation	CF4CT13	4 x 30
109.2	156.2	4.1	19.9	9.1	RNG without Radiation	DV4CT13	4 x 30
109.8	153.1	4.2	12.0	7.1	RNG with Radiation	DV4CT13	4 x 30
129.9	167.6	4.9	20.6	10.1	RNG without Radiation	CF2CT13	2 x 60
101.2	123.2	3.8	3.8	3.8	RNG with Radiation	CF2CT13	2 x 60
95.3	115.7	3.6	17.4	7.9	RNG without Radiation	DV2CT13	2 x 60
95.4	124.6	3.6	4.3	3.9	RNG with Radiation	DV2CT13	2 x 60

CF = Confluent Jets; DV= Displacement; T16 and T13 = Supply temp 16℃ and 13℃ respectively.

Table 4. Measured and calculated velocities [m/s] and air temperature for Confluent Jets Supply terminals for Case CF4CT16 (Four devices placed at the corners and a flow rate of 50 l/s for each device) at a distance of 30 mm from the wall.

Height above floor (m)		**Distance from supply (m), along the directions shown by the arrows in Fig. 1 b**											
		0.5		1.0		1.5		2.0		2.5		3.0	
		Temp (°C)	Vel. (ms^{-1})	Temp (°C)	Vel. (ms^{-1})	Temp (°C)	Vel. (ms^{-1})	Temp (°C)	Vel. (ms^{-1})	Temp (°C)	Vel. (ms^{-1})	Temp (°C)	Vel. (ms^{-1})
0.10	Exp.	19.5	0.65	19.9	0.55	20.3	0.48	20.5	0.47	20.6	0.43	21.0	0.33
	κ-ε	19.2	0.6	20.1	0.40	20.2	0.35	21.2	0.32	21.3	0.30	21.7	0.29
	RNG	19.0	0.56	19.7	0.57	20.6	0.51	21.1	0.39	21.3	0.32	21.6	0.34
0.25	Exp.	19.7	0.59	20.2	0.59	20.5	0.46	20.6	0.41	21.0	0.32	21.2	0.24
	κ-ε	19.8	0.31	21.6	0.36	21.9	0.29	22.1	0.21	22.1	0.20	22.1	0.10
	RNG	19.6	0.45	21.4	0.51	21.9	0.42	22.0	0.39	22.0	0.30	22.0	0.20
0.40	Exp.	19.8	0.88	20.3	0.64	20.8	0.43	21.0	0.36	21.2	0.29	21.4	0.21
	κ-ε	22.5	0.35	21.9	0.31	22.0	0.25	22.1	0.18	22.2	0.18	22.3	0.08
	RNG	22.3	0.70	21.6	0.52	21.7	0.39	21.8	0.30	22.0	0.20	22.1	0.18
0.60	Exp.	20.3	0.66	20.6	0.59	21.0	0.40	21.2	0.29	21.4	0.25	21.6	0.19
	κ-ε	22.6	0.30	22.5	0.21	22.4	0.25	22.3	0.10	22.5	0.06	22.5	0.08
	RNG	22.6	0.58	22.4	0.55	22.2	0.36	22.3	0.30	22.3	0.20	22.3	0.15
0.80	Exp.	20.6	0.54	21.0	0.52	21.3	0.32	21.5	0.26	21.7	0.22	21.9	0.15
	κ-ε	22.7	0.11	22.8	0.15	22.8	0.10	22.8	0.08	22.7	0.03	22.7	0.03
	RNG	22.7	0.46	22.6	0.41	22.5	0.28	22.6	0.23	22.5	0.19	22.5	0.09

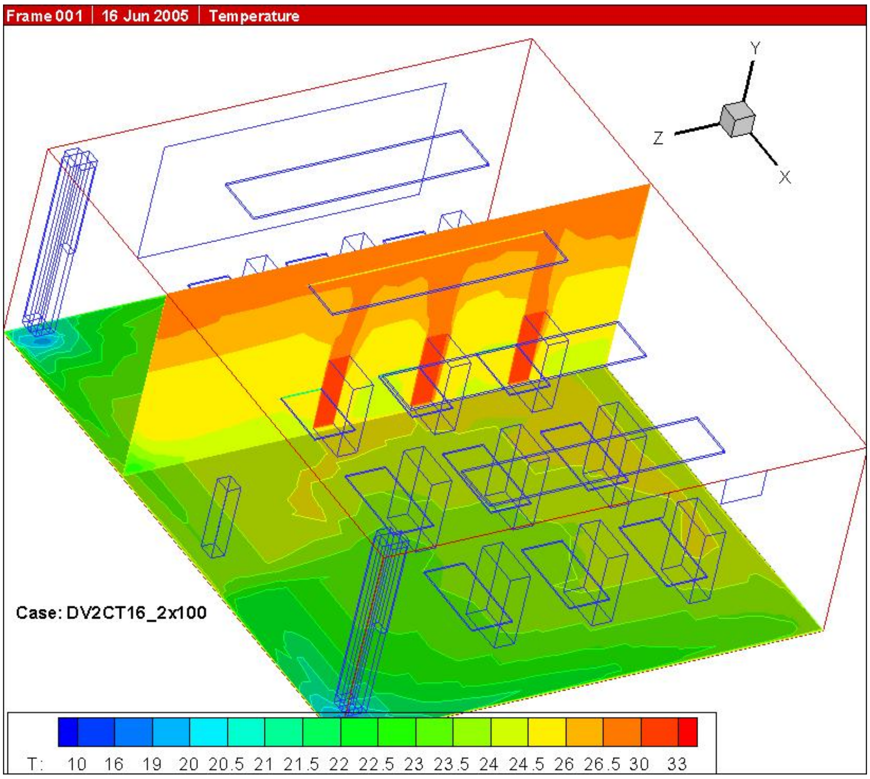


Fig. 4. Temperature field for Displacement system

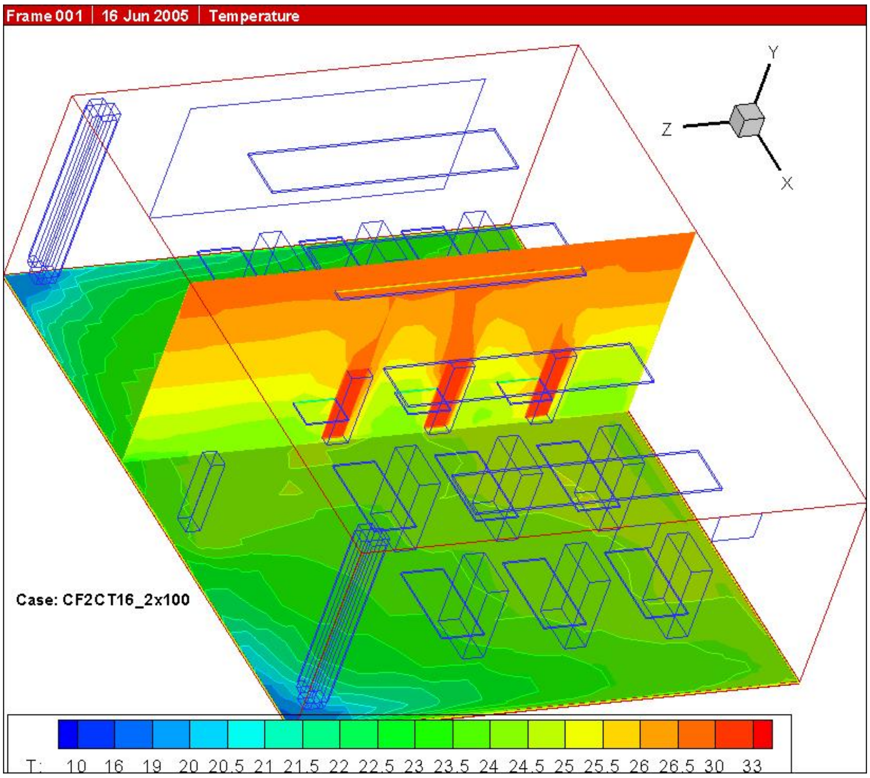


Fig. 5. Temperature field for Confluent Jets system

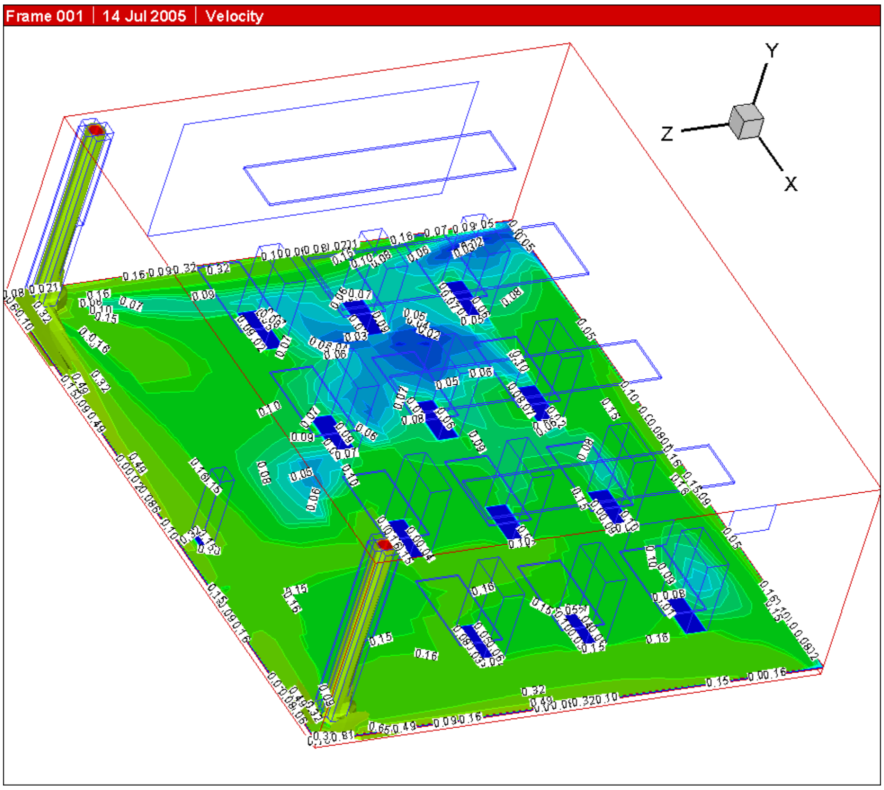


Fig. 6. Velocity field for Displacement system at 0.10 m from floor level

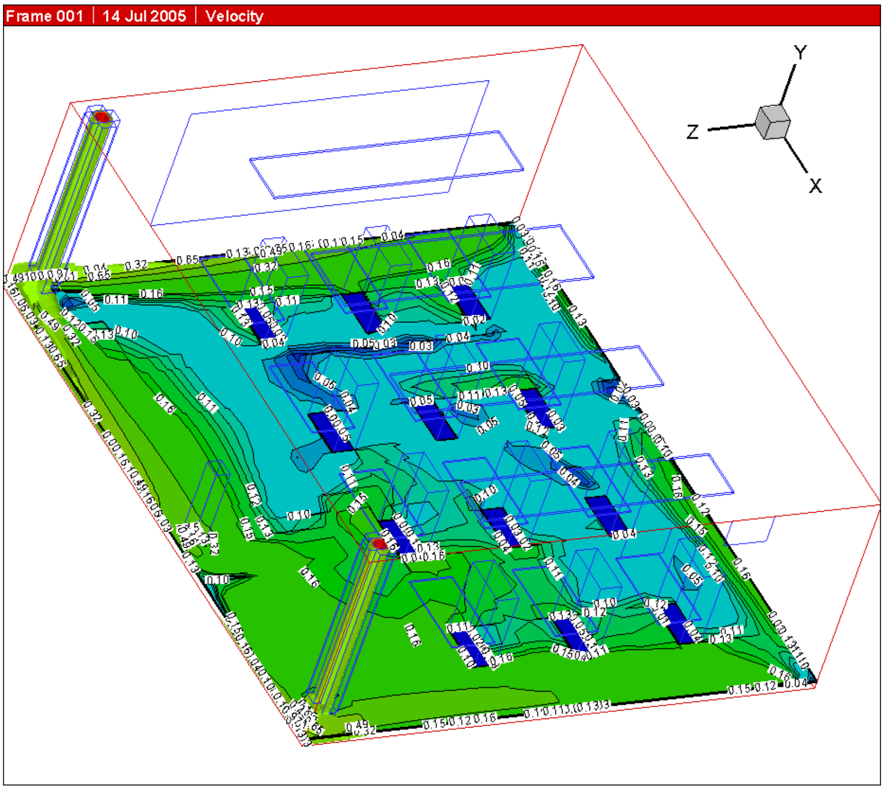


Fig. 7. Velocity field for Confluent Jets system at 0.10 m from floor level

Effect of radiation

Although it is known that the κ-ε model has a major weakness to predict the streamline curvatures and flow pattern of populated classroom with strong plumes, the inclusion of radiation in the calculations may improve the predicted results. It is also known that the more sophisticated turbulence models are costly and time consuming for the simulation of large ventilated enclosures; therefore the RNG turbulence model can be a compromise solution in this case.

Concerning the influence of radiation on the flow pattern and comfort parameters studied, it is clear from tables 2 and 3 that for both turbulence models used and for most cases, neglecting radiation over-predicts the parameters except for few cases. These over-predictions vary between 5 to 30 % depending on the inlet temperature, number of supply terminals used and total flow rate. Therefore, for simplicity and reliability of predictions the CFD simulations were carried out using the RNG model with a radiation algorithm, see tables 2 and 3.

Effect of number of supply terminals

For the same flow rate but two or four supply terminals installed at the corners, the following cases are compared (see table 3):

1) CF4CT13_4 x 50 with CF2CT13_2 x 100 and DV4CT13_4 x 50 with DV2CT13_2 x 100: According to table 3 the Air Distribution Index (ADI) increased when using two supply terminals instead of four. The supply temperature is 13 °C.
2) CF4CT16_4 x 50 with CF2CT16_2 x 100 and DV4CT16_4 x 50 with DV2CT16_2 x 100: According to table 3 the Air Distribution Index (ADI) decreased when using two supply terminals instead of four. The supply temperature is 16 °C.

These comparisons show that with a supply temperature of 13 °C, the two supply devices create more comfortable environment in a room than with four supply devices. However, for the other cases with 16 °C supply temperature, the four supply devices in the room produce better conditions than the two devices.

Effect of flow rate

By using the total flow rate of 120 l/s (2x60 and 4x30), 200 l/s (2x100 and 4x50) and 300 l/s (2x150), the following cases are compared (see table 3):

3) CF2CT13_2 x 60 with CF2CT13_2 x 100 and DV2CT13_2 x 60 with DV2CT13_2 x 100. According to table 3 the Air Distribution Index (ADI) increased by increasing the total flow rate from 120 l/s up to 200 l/s (5 to 8 l/s/person). This means that using a flow rate below the recommended values may not provide good comfort and air quality. The same phenomena were observed when comparing CF4CT13_4 x 30 with CF4CT13_4 x 50 and DV4CT13_4 x 300 with DV4CT13_4 x 50.
4) By increasing the total flow rate up to 300 l/s, that is 12 l/s /person, and comparing CF2CT16_2 x 100 with CF2CT16_2 x 150 and DV2CT16_2 x 100 with DV2CT16_2 x 150, surprisingly ADI decreases for both Confluent jets and Displacement systems. Considering table 3 and other parameters for the cases being compared, one can see that the comfort number, N_t, and the air quality number, N_c do not change. However, the effectiveness for heat removal (ε_t) and contaminant removal (ε_c) decreased by increasing the total flow rate. The answer may be found in equations (3) and (4). By increasing the flow rate the room-mean and outlet values for temperature and concentration change in such a way that affects the stratification and causes short circuiting of supply air.

These results show that with the present supply modes, both the Confluent Jets and Displacement systems provide acceptable conditions if the recommended air flow rates are used. Confluent jets behave slightly better in terms of comfort and air quality requirements. However, to handle a higher flow rate the inlet area of supply terminals should be increased to improve comfort and air quality in school environment in particular and other ventilated enclosures in general.

Effect of supply temperature

Using the two inlet temperatures of 13 °C and 16 °C, the following cases are compared (see table 3):

5) CF4CT13_4 x 50 with CF4CT16_4 x 50 and DV4CT13_4 x 50 with DV4CT16_4 x 50: According to table 3, the Air Distribution Index (ADI) increased by increasing the supply temperature from 13 °C to 16 °C. This was expected because in our estimation of total heat balance, see Karimipanah et al. (2000), a supply temperature of 16 °C was given and a lower temperature can decrease the comfort parameters.
6) CF2CT13_2 x 100 with CF2CT16_2 x 100 and DV2CT13_2 x 100 with DV2CT16_2 x 100: One can see from table 3 that the Air Distribution Index (ADI) decreased by increasing the supply temperature from 13 °C to 16 °C. This means that using two supply terminals instead of four require a lower supply temperature. Although all these factors are depended on the heat loads, using a VAV (Variable Air Volume) system is necessary to ensure a suitable inlet temperature for creating a comfortable indoor air environment.

A more general look at the results show that ventilation of a school environment is a difficult task and is depended on many parameters that need to be adjusted and controlled. This may mean more responsibility for ventilation engineers and designers to be aware of the fact that a system that works well in a certain environment does not mean it can be adapted for general use.

Comparison between Measurement and CFD Results

Table 4 gives comparisons between measured and predicted temperatures and velocities in the classroom at low level using the Confluent Jets systems. From these tables one can see that the overall agreement between the measured and predicted (CFD) velocities and temperatures is good. As is generally known the κ- ε turbulence model is not able to predict velocities in the occupied zone very well, but the temperature predictions are acceptable. However, the RNG turbulence model shows better capability of predicting the velocities but the temperatures are similar to those predicted by the κ- ε model.

CONCLUSIONS

Both the experimental and the CFD results confirm that the Confluent Jet System in most cases performed better than the displacement system.

The discrepancies between the measured and predicted velocities may be due to a number of reasons:

- Simplifications in the modelling of the real enclosure for the CFD solution
- The presence of strong buoyant flows which cannot accurately be represented by the standard κ-ε model
- The RNG turbulence model gives promising results in the simulation of confluent jets but one can not always be certain of capturing the highly buoyant flow patterns in case of classrooms with many heat sources
- Weakness of the wall functions used in the CFD code
- The influence of local buoyant flows, particularly in low velocity regions, on the measuring accuracies of the instruments used
- Errors in measurements particularly resulting from the short air velocity sampling period.

However, the CFD program gave very promising results. This is probably due to the fact that the measured surface temperatures have been used as boundary conditions in the CFD model.

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