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ON THE ENERGY CONSUMPTION OF HIGH- AND LOW-LEVEL AIR SUPPLIES

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Abstract

This paper focuses on the energy consumption of air supply systems for a ventilated room involving high- and low-level supplies. The energy performance will be based on the airflow rate, which is related to the fan power consumption by achieving the same performance for each case. Four different ventilation systems are considered: wall displacement ventilation, confluent jets ventilation, impinging jet ventilation and a high level mixing ventilation system. The ventilation performance of these systems will be examined by means of achieving the same Air Distribution Index (ADI) for different cases.
The widely used high-level supplies require much more fan power than those for low-level supplies for achieving the same value of ADI. In addition, the supply velocity, hence the supply dynamic pressure, for a high-level supply is much larger than for low-level supplies. This gives an additional difference in the power consumption between the two systems.
The paper considers these factors and attempts to provide some guidelines on the difference in the energy consumption associated with high and low level air supply systems. This is useful information for designers and to the authors' knowledge there is not enough information available in the literature on this area of room air distribution.

1. Introduction

The major part of energy consumption for both industrial and residential buildings is due to HVAC systems. This paper focuses on the energy consumption of air supply processes for a ventilated room involving high and low-level supplies. The energy performance will be distinguished by the fan power consumption, which is related to the airflow rate, by achieving the same environmental performance for each case. Four different ventilation systems are considered: wall displacement ventilation, confluent jets ventilation, impinging jet ventilation and a high level mixing ventilation system. The ventilation performances of these systems are examined elsewhere [1-8].
The influence of temperature gradient on energy consumption for mixing, displacement and confluent jets systems was numerically investigated by Vulle [9]. He used the IDA-Indoor Climate and Energy program for simulation of thermal comfort, indoor air quality and energy consumption in buildings [10] in his calculations. He presented a potential energy saving of about 10-15% for confluent jes and displacement systems compared with a traditional mixing air supply system. In contrast to this paper, Vulle [9] does not discuss the relation between temperature gradient and thermal comfort.

2. Theoretical Background

The numerical calculations have been carried out for a test room (2.78m x 2.78m x2.3m) at the University of Reading using VORTEX CFD code with RNG turbulence model [11]. It is worth mentioning that some extensive measurements were also done for all four systems which can be found in [3]. A basis for dimensioning physical parameters for the use in the design of confluent jets ventilation systems was established.
For CFD simulations the supply air flow rate varied from 25 l/s (reference flow rate) up to 50 l/s to achieve the same ADI (defined below) for all systems used. A supply temperature of 18 ^oC and a maximum cooling load of 60W/m^2 were considered.

The local age of air at any point in the room can be calculated using the following expressions:

$$\bar{\tau}_p = \frac{\int_0^\infty C_p(t)dt}{C(0)} \qquad (1)$$

To assess the effectiveness of a ventilation system, 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 [12]:

$$\varepsilon_t = \frac{T_o - T_i}{T_m - T_i} \quad \text{and} \quad \varepsilon_c = \frac{C_o - C_i}{C_m - C_i} \qquad (2)$$

In equation (2), T is temperature (^oC), 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 [13] 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. (3) and (4).

$$PD = 395 \cdot \exp(-1.83\ \dot{v}^{0.25}) \qquad (3)$$

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

Where $\dot{v}$ is the ventilation rate (ls^{-1}) and PMV is the Predicted Mean Vote as defined in ISO 7730 [14] 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.

To examine the quality of a ventilation system a thermal comfort number, N_t , and an air quality number, N_c , may be found by combining relations (2) with PPD and PD respectively [12, 15]:

$$N_t = \frac{\varepsilon_t}{PPD} \quad , \quad N_c = \frac{\varepsilon_c}{PD} \tag{5}$$

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 an Air Distribution Index (ADI), defined as [12]:

$$ADI = \sqrt{N_t \times N_c} \tag{6}$$

In this investigation the above relation is used for comparing the ventilation performance of the ventilation systems.

3. Results and discussion

It is known that the widely used high-level air supplies need much more fan power than those for low-level supplies. To achieve an acceptable limit of CO_2 concentration (e.g. 1000 ppm), the traditional ventilation guides recommend 8 l/s/person for low-level and 10 l/s/person for a high-level supplies, i.e. a factor of 1.25.
The following relations between the flow rate (q), pressure difference (Δp) and the fan power (E) are used:

$$\Delta p \propto q^2 \qquad E \propto q^3 \tag{7}$$

Thus, Δp for a high-level supply is higher by a factor of $1.25^2 = 1.56$ (i.e. 56%) and the fan power (E) by a factor of $1.25^3 = 1.95$, which gives 95% difference in the energy consumption. In addition, the supply velocity and hence the supply dynamic pressure for high level supplies are much larger than for low-level supplies. This gives an additional difference in the power consumption between the two types of systems in addition to the larger flow requirements just described.

Using four different air distribution methods, see Table 1 and the Air Distribution Index (ADI) for comparison, one can see that there is no need for higher flow rate if one uses a better air supply method. According to Table 1, the wall confluent jet system gives the best ADI (i.e. 13.5) for the minimum flow rate of 0.025 $m^3 s^{-1}$ that was considered.

To obtain the same index for mixing system we need 1.8 times more flow rate. Using relation (7) this gives $1.8^3 = 5.83$ which means larger energy consumption. For a displacement system this gives $1.10^3 = 1.33$, although less energy compared with the mixing system, it is still higher than the confluent jets system. For the impinging jet system this gives $1.4^3 = 2.74$ which is almost double that for displacement but half that of the mixing system. The impinging jet system performs very well at higher flow rates and is a good competitor for displacement and confluent jet systems.

The temperature and velocity contours of Confluent Jets Ventilation system are shown in Figures 1 and 2. The highest temperatures occur at ceiling level and the velocities are within the recommended standard values. Therefore the confluent jets system can combine the positive effects of displacement system (stratification) and mixing system (entrainment of the surrounding air into the jet). Another benefit of using confluent jet system is that it can be used for both heating and cooling purposes.

Table 1. Data from the CFD simulations

Ventilation system	Total flow rate [m^3/s]	Air Distribution Index (ADI)
Mixing	0.025 (ref. flow rate$^+$)	10.9
Mixing	0.040 (1.6x0.025)	12.2
Mixing	**0.045 (1.8x0.025)**	**13.5**
Mixing	0.0475 (1.9x0.025)	14.2
Mixing	0.050 (2x0.025)	15.4
Displacement	0.025 (ref. flow rate)	12.0
Displacement	0.0277 (1.11x0.05)	13.9
Displacement	**0.0275 (1.10x0.05)**	**13.6**
Displacement	0.050 (2x0.025)	22.9
Confluent jets	**0.025 (ref. flow rate)**	**13.5**
Confluent jets	0.050 (2x0.025)	23.9
Impinging jet	0.025 (ref. Flow rate)	11.3
Impinging jet	0.030 (1.2x0.025)	12.4
Impinging jet	**0.035 (1.4x0.025)**	**13.6**
Impinging jet	0.0375 (1.5x0.025)	14.4
Impinging jet	0.050 (2x0.025)	20.8

$^+$the reference flow = 0.25 $m^3\ s^{-1}$

4. Concluding remarks

- The choice of air supply system has a major impact on energy consumption.
- A confluent jets air supply system performs much better than the displacement, impinging or mixing systems.
- To obtain the same Air Distribution Index for a mixing system 1.8 times more flow rate is required and 5.83 times more energy is consumed.
- A displacement system uses 1.33 times and an impinging jet system uses 2.74 times the energy used by the confluent jets system but they still perform better and use less energy than the traditional mixing system.
- The authors believe that for energy saving new developments of low-level air supply systems are needed as well as reducing reliance on the traditional mixing systems which perform worse than low-level air supply systems.

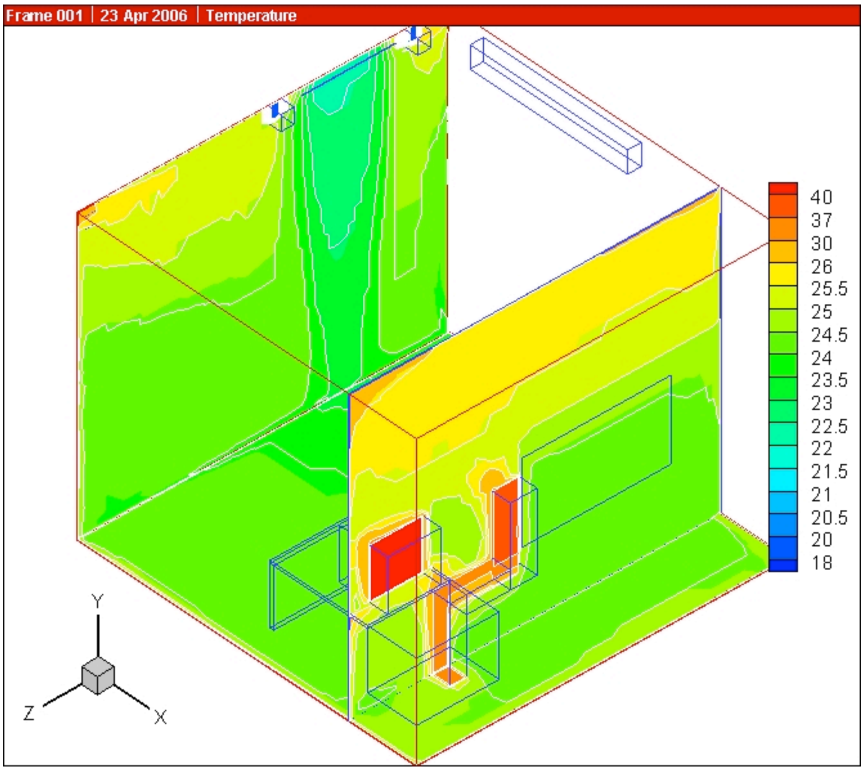

Figure 1 Temperature contours for Confluent jets system

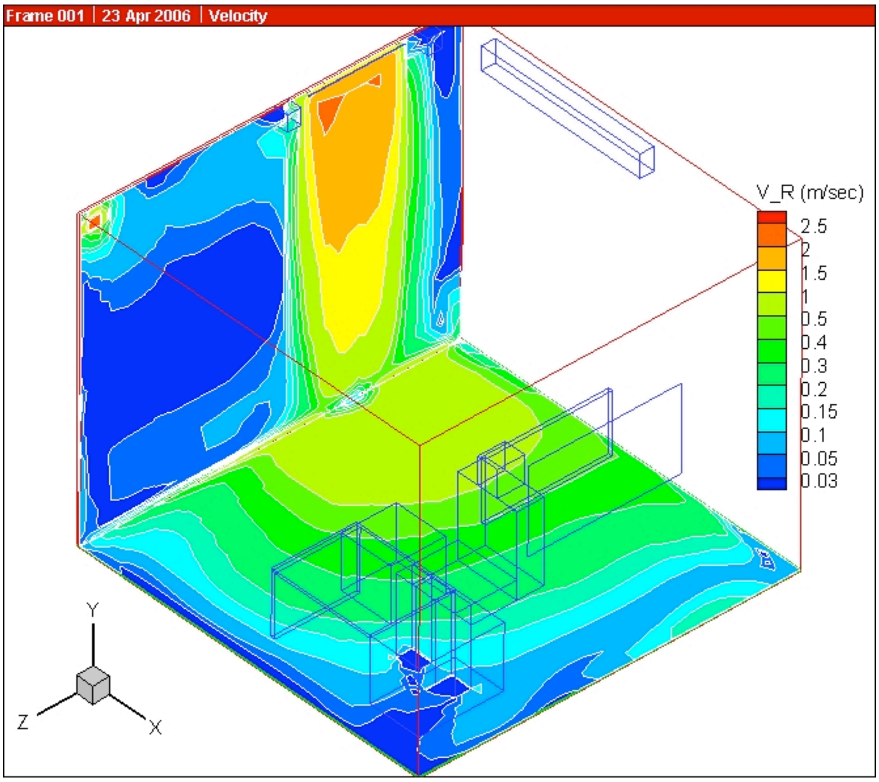

Figure 2 Velocity contours for Confluent jets system

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