1. Introduction

The major part of energy consumption in both industrial and residential buildings is for 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 assessment is based on the airflow rate, which is related to the fan power consumption by achieving the same environmental quality 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 further increases the power consumption for high-level supply 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 will be useful information for designers and to the authors’ knowledge there is a lack of information available in the literature on this area of room air distribution.

The energy performance of the above-mentioned ventilation systems has been evaluated on the basis of the fan power consumed which is related to the airflow rate required to provide equivalent indoor environment. The Air Distribution Index (ADI) is used to evaluate the indoor environment produced in the room by the ventilation strategy being used. The results reveal that mixing ventilation requires the highest fan power and the confluent jets ventilation needs the lowest fan power in order to achieve nearly the same value of ADI.

**Key words:** ventilation, air distribution index, energy, CFD, modelling, indoor environment
dropower generation.

The EU Commission states in the directive for energy efficiency in the built environment (EPBD) that the building sector has to decrease its use of energy to reduce CO₂ emissions. In Sweden, the Environmental Advisory Council have stated, within the framework of “Bygga Bo Dialogen”, (Build Housing Dialogue) that the demand for purchased energy in the building sector should decrease at least by 30% by 2025 compared to 2000 and energy use in 2010 should be lower than in 1995.

The link between the increased CO₂ emissions and the use of energy is also a motive to render a more efficient energy usage, and lower the total energy demand. As a result, the need for energy conservation and reduction of electrical energy usage in the built environment is very strong. Ventilation systems, thermal comfort and air quality within the built environment are important issues as they relate both to energy consumption and the health of the occupants.

The aim of this study is to fulfill the above-mentioned goals and is thus a step towards a sustainable building sector. This paper focuses on the energy usage of air supply processes for a ventilated office space involving high- and low-level supplies. The energy performance will be based on the fan power consumption, which is related to the airflow rate, by achieving the same indoor environmental performance in 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.

The influence of temperature gradient on energy usage for mixing, displacement and confluent jets systems was numerically investigated by Vulle. He used the IDA-Indoor Climate and Energy program for simulation of thermal comfort, indoor air quality and energy usage in buildings in his calculations. He presented a potential energy saving of about 10–15% for confluent jets and displacement systems compared with a traditional mixing air supply system. In contrast to this paper, Vulle does not discuss the relation between thermal comfort and air quality with energy usage.

2. Computational set-up and numerical procedure

The room under consideration, which is furnished like an office for a single occupant, is shown in Figure 1. The room has the dimensions 2.78 × 2.78 × 2.3 m. The boundary conditions for inlet velocity profile and turbulent intensity as well as the interior wall surface temperatures have been derived from the measurements by Cho et al. The supply air temperature is 18°C. The heat load consists of the heat gain at the exterior wall (55 W), the window (120 W) and the internal heat generation (100 W), corresponding to a cooling load of 35 W/m².

The VORTEX CFD code is used for both grid generation and the numerical solution of the proposed problem. The generated mesh is a 3-D structured grid with 61 × 58 × 63 points (in x, y, z) for mixing, 58 × 44 × 51 for confluent jets, 60 × 60 × 65 for displacement and 67 × 60 × 69 for impinging jet, giving a total of about 222 894, 130 152, 234 000 and 277 380 cells respectively. Clustering the mesh towards the walls has been controlled in such a way that the employed wall-function treatment is properly

![A sketch of the office room.](image-url)
applied. The Renormalization Group (RNG) $k$-$\varepsilon$ model has been used to predict the turbulent behaviour of the flow within the room. The governing equations are solved using a segregated scheme. The equations for the momentum, energy, concentration, kinetic energy and turbulent dissipation rate are discretized spatially with a QUICK scheme. The pressure-velocity coupling algorithm SIMPLE is used to solve the continuity equation. The local criterion for numerical convergence, i.e. the maximum relative difference between two consecutive iterations for any local variable, is less than $10^{-3}$. To extend the degree of accuracy in the solution a calculation of the energy balance is also made.

3. Data reduction

To assess the effectiveness of a ventilation system, the effectiveness for heat removal ($\varepsilon_i$) and contaminant removal ($\varepsilon_c$) are used together with the predicted percentage of dissatisfied (PPD) for thermal comfort and percentage of dissatisfied ($PD_{AQ}$) for air quality. $\varepsilon_i$ and $\varepsilon_c$ are defined by [12]:

$$\varepsilon_i = \frac{T_o - T_i}{T_m - T_i}$$

(1)

$$\varepsilon_c = \frac{C_o - C_i}{C_m - C_i}$$

(2)

In Eqs. (1) and (2), $T$ is temperature (°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 1.8 m). $\varepsilon_i$ is similar to a heat exchanger effectiveness and is a measure of the heat-removing ability of the system. $\varepsilon_c$ is a measure of how effectively the contaminant is removed. The values for $\varepsilon_i$ and $\varepsilon_c$ 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.

The expressions for the Percentage of Dissatisfied ($PD_{AQ}$) with the indoor air quality and the Predicted Percentage of Dissatisfied (PPD) with the thermal environment are given by Eqs. (3) and (4), see Fanger [13]:

$$PD_{AQ} = 395 \cdot \exp(-1.83 \cdot \dot{V}^{0.25})$$

(3)

$$PPD = 100 - 95 \cdot \exp[-0.033353 \cdot (PMV)^{4} + 0.2179 \cdot (PMV)^{2}]$$

(4)

where $\dot{V}$ is the ventilation rate (l/s) 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 $-0.5 \leq PMV \leq 0.5$. 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 (1) and (2) with $PPD$ and $PD_{AQ}$ respectively [12 and 15]:

$$N_t = \frac{\varepsilon_i}{PPD}$$

(5)

$$N_c = \frac{\varepsilon_c}{PD_{AQ}}$$

(6)

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]:

$$ADI = (N_t \times N_c)^{0.5}$$

(7)

In addition, the effect of draught in the room is calculated using the equation below, see [16], which gives the percentage dissatisfied due to draught ($PD_{D}$):

$$PD_{D} = (34 - t_a) v - 0.05 \cdot 0.6223 \cdot (3.143 + 0.3696 v \cdot TI)$$

(8)

where $t_a$ is the air temperature (°C), $v$ is the air speed (m/s), and $TI$ is the turbulence intensity (%).

In this investigation the above relations are used for comparing the ventilation performance of the ventilation systems.

The following relations between the flow rate, $q$, pressure difference, $\Delta p$, and the fan power, $E$, are used to evaluate the energy performance of the ventilation systems:

$$\Delta p \propto q^2, \quad E \propto q^3$$

(9)

4. Results

4.1 Comparison between the indoor quality performance of the ventilation systems for achieving the same value of ADI

Figure 2 (a and b) shows a perspective view of the velocity in the occupied zone for the summer case with the iso-velocity $= 0.25$ m/s represented by the floods for the mixing and confluent jets ventilation systems. The occupied zone here is defined by a height of 1.8 m and distances of 0.30 m from each wall and 0.1 m above the floor. Figure 2(a) shows that the mixing system has a small area at the floor with a velocity higher than 0.25 m/s and the confluent jets supply shows a small area under the supply device also at floor level where the velocity is $= 0.25$ m/s. The same effect has also been observed for the dis-
placement system which creates a so-called near-zone. In the case of the impinging jet system, the air velocity is lower than 0.25 m/s in the whole occupied zone. This means that the impinging jet fulfils the accepted criterion for velocities below 0.25 m/s in the occupied zone very well, see Fig. 2(b).

To relate the physical parameters of the indoor environment to the occupant’s comfort, the $PDD$ (Percentage Dissatisfied due to draught) index has been used. The $PDD$ index is illustrated in Fig. 3(a) and (b) for the mixing and the confluent jets. The $PDD$ index is plotted at a plane which is 1.1 m over the floor level, i.e. at about neck height.

As shown in Fig. 3(a), draught may cause problems for the mixing system used in the present study as $PDD > 10\%$ for large parts of the plane at neck level. However, the confluent jets system shows acceptable levels of $PDD$ near the occupant although some high values of $PDD$ are observed far from the occupant and close to the jet zone which is not of much interest in this case. The $PDD$ levels for the confluent jets and the other two systems are lower than those of the mixing system and are within the range of recommended values. This could be due to the low ventilation effectiveness of the mixing system which is usually much less than 100% compared with that for the displacement, confluent jets and impinging jet systems which is over 100% in many cases.

Figure 4 shows predicted $PDD$ values at 1.1 m above the floor level for the mixing and confluent jets systems. One can see that the highest levels of $PPD$ are observed close to the occupant for the mixing system. It is worth mentioning that $PPD$ values over 10% are not considered acceptable. In the case of the confluent jets, the highest levels of $PPD$ are observed very close to the occupant and under the supply de-
Fig. 3(a). Percentage Dissatisfied due to draught at 1.1 m over the floor level (mixing ventilation left and confluent jets ventilation right).

Fig. 3(b). Percentage Dissatisfied due to draught at 1.1 m over the floor level (impinging jet ventilation left and displacement ventilation right).

Fig. 4. Predicted Percentage of Dissatisfied 1.1 m over the floor level (mixing ventilation left and confluent jets ventilation right).
vice. It is worth mentioning that in the most parts of
the room the PPD level is below or equal to 10%.
Here one can see the benefits of CFD calculations in
the pre-design stage. With the above observation it
will be recommended to re-position the occupant into
other more favorable zones in the room.

Finally, by analysing the PPD data, it was found
that the three systems—confluent jets, impinging jet
and displacement systems—behave similarly in the
ventilation of a small office with acceptable ranges of
PPD values. But for the market-dominated mixing
system this is not the case and this system should be
avoided when possible due to its high energy con-
sumption and less favorable aspects of the indoor en-
vironmental quality for the occupants.

The temperature and velocity floods of the conflu-
ent jets ventilation system in the mid-plane of the
room are shown in Fig. 5. The highest temperatures
exist at ceiling level and the velocities are within the
recommended values. Therefore the confluent jets
system can combine the positive effects of the dis-
placement system (stratification) and the mixing sys-
tem (rapid entrainment of the surrounding air into the
jet). Another benefit of using a confluent jets system
is that it can be used for both heating and cooling
purposes.

Table 1. PPD and PD_{AQ} values in the occupied zone for all ventilation systems with the same ADI index.

<table>
<thead>
<tr>
<th></th>
<th>Mixing</th>
<th>Displacement</th>
<th>Impinging jet</th>
<th>Confluent jets</th>
</tr>
</thead>
<tbody>
<tr>
<td>PPD (%)</td>
<td>8.4</td>
<td>13.9</td>
<td>11.2</td>
<td>5.0</td>
</tr>
<tr>
<td>PD_{AQ} (%)</td>
<td>2.2</td>
<td>12.9</td>
<td>11.7</td>
<td>1.0</td>
</tr>
</tbody>
</table>

Table 2. The average values of \( \varepsilon \), PPD, \( \varepsilon^\prime \), PD_{AQ} and
ADI in the whole room for all ventilation systems.

<table>
<thead>
<tr>
<th></th>
<th>Mixing</th>
<th>Displacement</th>
<th>Impinging jets</th>
<th>Confluent jets</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \varepsilon ) (%)</td>
<td>99</td>
<td>121</td>
<td>125</td>
<td>123</td>
</tr>
<tr>
<td>PPD (%)</td>
<td>9.0</td>
<td>9.2</td>
<td>8.9</td>
<td>8.5</td>
</tr>
<tr>
<td>( N_t ) (%)</td>
<td>11.0</td>
<td>13.2</td>
<td>14.0</td>
<td>14.5</td>
</tr>
<tr>
<td>( \varepsilon^\prime ) (%)</td>
<td>118</td>
<td>118</td>
<td>120</td>
<td>121</td>
</tr>
<tr>
<td>PD_{AQ} (%)</td>
<td>5.4</td>
<td>6.1</td>
<td>7.0</td>
<td>7.2</td>
</tr>
<tr>
<td>( N_t' ) (%)</td>
<td>21.8</td>
<td>19.3</td>
<td>17.1</td>
<td>16.8</td>
</tr>
<tr>
<td>ADI [-]</td>
<td>15.5</td>
<td>15.9</td>
<td>15.5</td>
<td>15.6</td>
</tr>
</tbody>
</table>

4.2 Comparison between the energy consumption of ventilation systems for achieving the same value of ADI

The minimum, mean and maximum values of PD_{AQ} and PPD in the occupied zone are summarized in Table 1. The results show that the displacement, impinging jet and confluent jets supply systems provide similar performance in terms of indoor air quality and thermal comfort due to the similar range of values of PD_{AQ} and PPD, while the mixing ventilation system has the highest PD_{AQ} and PPD values in the occupied zone, see also [17].

The average values of \( \varepsilon \), PPD, \( N_t \), \( \varepsilon^\prime \), PD_{AQ} and ADI in the whole room are summarized in Table 2. Taking the ventilation space as a whole (including the jet region), the results reveal that the highest values of the effective, \( \varepsilon \), and \( \varepsilon^\prime \) were produced by the
displacement, impinging and confluent jets systems. The same also applies to the values of the thermal comfort number, \(N_c\). However, the mixing ventilation achieved the highest value of the air quality number, \(N_t\), due to the low value of \(PD_{AG}\) although the value of \(\varepsilon_c\) for the mixing system is one of the lowest among the four systems.

Table 3 compares the air flow rates required for achieving an Air Distribution Index (ADI) of about 16. The confluent jets system requires the lowest flow rate (0.025 m\(^3\)/s) of the four systems considered. To obtain nearly the same ADI value for a mixing system we need 1.8 times greater flow rate. Using relation (9), this gives 5.83 times higher fan power consumption. For the impinging jet system the flow rate is 1.4 times higher and demands 2.74 times more power. The power consumption of the impinging jet system is almost half of the mixing system. The displacement system requires 1.1 times greater flow rate and 1.33 times higher power consumption compared to the confluent jets. Although the power consumption is lower than the other two systems, it is still higher than the confluent jets system.

5. Concluding remarks
The choice of air supply system in a room has a major impact on the energy consumption of the ventilation system because of the large differences in the thermal and air quality effectiveness between the systems as well as in the fan power consumption which constitutes a large component of the total HVAC systems energy requirement. The overall performance of the confluent jets air supply system is somewhat better than the displacement and impinging jets systems but superior to the mixing system. The confluent jets behave like the displacement and the impinging jet ventilation systems combined. To obtain the same Air Distribution Index for a mixing system 1.8 times greater flow rate is required and 5.83 times more fan power is used. The displacement system uses 1.33 times and the impinging jet system uses 2.74 times the power used by the confluent jets system, but they still perform better and use less power than the traditional mixing system. The authors believe that for the purpose of energy saving, new developments in low-level air supply systems are needed to reduce the reliance on traditional mixing systems, which have inferior performance compared to low-level air supply systems.

References

<table>
<thead>
<tr>
<th>System</th>
<th>ADI</th>
<th>Total flow rate</th>
<th>Energy usage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mixing ventilation</td>
<td>15.5</td>
<td>0.045 m(^3)/s</td>
<td>580%</td>
</tr>
<tr>
<td>Difference compared to confluent jets</td>
<td>180%</td>
<td>0.035 m(^3)/s</td>
<td>270%</td>
</tr>
<tr>
<td>Impinging jet ventilation</td>
<td>140%</td>
<td>0.0275 m(^3)/s</td>
<td>130%</td>
</tr>
<tr>
<td>Difference compared to confluent jets</td>
<td>110%</td>
<td>0.025 m(^3)/s</td>
<td>100%</td>
</tr>
<tr>
<td>Displacement ventilation</td>
<td>10%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Difference compared to confluent jets</td>
<td>10%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Confluent jets ventilation</td>
<td>10%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3. Comparison between the fan power consumption of the ventilation systems for the same ADI.