Theoretical and experimental investigation of impinging jet ventilation and comparison with wall displacement ventilation

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Abstract

This paper focuses on evaluating the performance of a new impinging jet ventilation system and compares its performance with a wall displacement ventilation system. Experimental data for an impinging jet in a room are presented and non-dimensional expressions for the decay of maximum velocity over the room are derived. In addition, the ventilation efficiency, local mean age of air and other characteristic parameters were experimentally and numerically obtained for a mock-up classroom ventilated with the two systems. The internal heat loads from 25 person-simulators and lighting were used in the measurements and simulations to provide a severe test for the two types of ventilation systems. In addition to a large number of experimental data CFD simulations were used to study certain parameters in more detail. The results presented here are part of a larger research programme to develop alternative and efficient systems for room ventilation. © 2002 Elsevier Science Ltd. All rights reserved.

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1. Introduction

Although the traditional mixing systems have poor ventilation efficiency and are less energy efficient, they still occupy a large portion of the market. When displacement ventilation (DV) was first introduced almost three decades ago, it seemed at the time to be a promising ventilation concept due to its high ventilation efficiency and stratification principle. In DV systems, cool air supplied at low level is entrained by plumes rising from heat sources in the room. To create an effective ventilation system in the occupied zone, there should be a balance between the momentum in the supply air and thermal (buoyancy) forces due to heat sources. In this low momentum displacement flow, the buoyancy forces created by heat sources have a tendency to take over and thus often causing poor ventilation efficiency in some zones of the room [1]. Another disadvantage of a displacement system is that it can only be used for cooling and is not suitable for winter heating. To overcome this problem new systems like ceiling mounted textile bag supply and down-to-floor impinging supply have been developed, see Ref. [2].

A new method of air distribution known as the Air Queen (AQ) has been developed in Sweden, which is based on impinging jet ventilation (IJV) [3]. This method is based on the principle of supplying a jet of air with high momentum downwards onto the floor. As the jet impinges onto the floor it spreads over a large area causing the jet momentum to recede but still has a sufficient force to reach long distances. Unlike the DV system which “floods” the floor with supply air, the resulting flow from an IJV is a very thin layer of air over the floor. This method has the advantages of both the mixing and displacement ventilation systems without known disadvantages. Impinging jet ventilation has lower momentum than mixing and higher momentum than wall displacement ventilation (WDV), i.e. when the air terminal unit is wall mounted at low level. Although higher momentum than WDV, IJV produces a similar flow field and has, therefore, promising applications [2,4].

2. Properties of impinging jet

The impingement of a turbulent jet on a flat plate (wall) has been widely studied with different configurations. In all
early investigations, the prediction of the momentum flux follows the same procedure as in a classical boundary layer. The impinging jet is an interesting test case due to its different flow regions (i.e. the potential core region, the free jet region, the impinging or flow deflection region and the wall jet region).

Some early and recent investigators have also studied the free and impinging jet phenomena. Beltaos and Rajaratnam [5] provided detailed results for the impingement region and enlarged the scope of the results which were available in the free jet and wall jet regions. Gutmark et al. [6] presented an experimental study of the turbulent structure on the centre-line of a two-dimensional impinging jet. They pointed out that the impingement of the jet is not affected by the presence of the plate over 75% of the distance between the nozzle and the plate and also the turbulent properties of the jet change from their equilibrium level close to the impingement region. The impinging jet is somewhat similar to a stagnation flow in which an infinite stream impinges on a finite body. In our investigation we used the impinging jet configuration shown in Fig. 1. The flow field is divided into three regions, see also Ref. [5]: (i) a free jet region, (ii) the impingement region and (iii) the wall jet region. There are also transitional zones between these regions.

The numerical analysis and development of impinging flows have received large attention, not because of their simplicity, but due to the presence of different flow regions. However, the result from the analysis may depend on the turbulence model used.

A jet approaching a plate will at some distance from the plate begin to “feel” the presence of the plate. For example the mean velocity on the centre line is similar to a free jet up to some distance and then decreases (faster) to the zero value at the impingement point, see Fig. 2. The distance from the plate where the centre line velocity starts to deviate from the free jet curve is taken as the location of the end of the free jet region and the beginning of the impingement region. From the figure one can see that the effect of the plate is felt when the distance from the plate is < 0.14h, which is in agreement with the results obtained by Beltaos and Rajaratnam [5].
3. Impinging jet ventilation system

A test method has been developed for impinging jet supply devices, which is based on the radial impinging jet theory, see Ref. [7]. It was based on the assumption that the velocity distributions on the impinging surface have the same profiles as the wall jet region illustrated in Fig. 3. Independent tests show that the radial velocity ($U$) in the $y$-direction with different radial distances from the origin give a similar velocity profile described in dimensionless form, see Ref. [8].

Using these similarity criteria, the following proportionality relation is obtained:

$$\log \left( \frac{U_{\text{max}}}{U_o} \right) \propto \log \left( \frac{x}{\sqrt{A_{\text{out}}}} \right),$$

where $U_{\text{max}}$ is the maximum velocity in radial direction for a specific point located at horizontal distance $x$ from origin (impingement point), $U_o$ is supply velocity and $A_{\text{out}}$ is the supply outlet area.

The resulting empirical relation based on the measured $U_{\text{max}}$ at different distance ($x$) in the radial direction, can give a basis for use of impinging jet supply system.

3.1. Methodology and test conditions

Velocity field measurements were carried out in a test room (“classroom”) at the Centre för Built Environment in Gävle, Sweden. The supply device “Air Queen” model AQ-D54/24 was used in the tests. Two impingement heights of 300 and 950 mm were studied, see Fig. 4(a).

The aim of velocity measurements was to establish a basis of dimensioning physical parameters for design purposes of the impinging jet supply. This was to investigate the variation of the maximum air velocity on the floor with the jet outlet velocity and to establish the vertical height above the floor where this maximum occurs as this will be important for draught considerations. The tests were also aimed to show the influence of impingement height on the flow pattern over the floor.

The supply airflow rate was regulated by a variable speed fan and the supply temperature was set according to the cooling load in the test room. The room temperature was kept constant by cooling the room walls. The airflow was measured by an orifice plate. During the tests an airflow of 1501/s, supply temperature of 18°C and a room temperature of 21.5°C were used, see Ref. [7] for further details.

For supply heights of $h = 950$ and 300 mm, velocities were conducted at 12 measuring points with different distances ($x$) and 3 angles from the impingement point, see Fig. 4(b). At each measuring point the traversing for velocity was carried out from the floor to 300 mm with 10 mm between traversing intervals. At each point in the traversing interval, 480 samples were recorded for a sampling interval of 0.5 s to give the mean velocity at that point.
The following measuring instruments were used in the tests:

- Air velocity measuring instrument of type Swema Air 300 with hotwire anemometer of type CTDA.
- Micromanometer of type Alnor MP3 KDS (pressure measurement for supply duct).
- Temperature measuring instrument of type Mitec AT30 with thermistor.

The measuring errors were estimated to be:

- Supply air-6Row (orifice) ±4%.
- Room temperature ±0.5 K.
- Supply temperature ±0.5 K.
- Air temperature in the jet ±0.2 K.
- Air velocity ±0.05 m/s.

The measured parameters for the two cases considered were:

**Supply height 950 mm**

Supply airflow (orifice) $q_o = 0.15 \text{ m}^3/\text{s}$.
Supply outlet area, $A_{out} = 0.094 \text{ m}^2$ ⇒ characteristic value $\sqrt{A_{out}} \approx 0.30 \text{ m}$.
Supply air velocity at outlet, $U_o = 1.60 \text{ m/s}$.

**Supply height 300 mm**

Supply airflow rate, $q_o = 0.15 \text{ m}^3/\text{s}$.
Supply outlet area, $A_{out} = 0.117 \text{ m}^2$ ⇒ characteristic value $\sqrt{A_{out}} \approx 0.34 \text{ m}$.
Supply air velocity at outlet, $U_o = 1.29 \text{ m/s}$.

The reason for varying the supply outlet area and the velocity in the two chosen heights was because the extended noise damping cover of supply duct in the case with supply height of 300 mm was missing.

### 3.2. Test results

The results from these tests are given in Figs. 5–10. The equation in Fig. 6 shows a best line fit through the measured values. This is used to calculate the jet spread rate, i.e. 0.081 in the equation and also the virtual origin which is $-0.34$. The same procedure is repeated for other angles, see Figs. 8 and 10, which show the difference in the jet spread and the location of virtual origin for the two angles. The differences obtained are due to the vicinity of the wall and the entrained air into the jet boundaries for the measuring cases with $30^\circ$ and $90^\circ$ which leads to a higher spread rate, 0.081 and 0.074, respectively. These values are very close to the wall jet spread rate obtained by Karimipanah [3]. For the $60^\circ$ case the jet spread rate is as low as 0.067, see Fig. 8 and the configuration in Fig. 4. For the later case, the confinement effects are less important and the jet seems to be almost unaffected by the walls.

### 3.3. Empirical equations

The empirical equations for each angle given below show how the maximum velocity varies with distance from the supply device as a function of supply area and outlet velocity. The equations have been derived from the data in Figs. 5–10.

**Angle 30°**

$$U_{max} = 3.38U_o \left( \frac{x}{\sqrt{A_{out}}} \right)^{-1.52}$$

(2)

**Angle 60°**

$$U_{max} = 2.66U_o \left( \frac{x}{\sqrt{A_{out}}} \right)^{-1.19}$$

(3)

**Angle 90°**

$$U_{max} = 2.45U_o \left( \frac{x}{\sqrt{A_{out}}} \right)^{-1.10}$$

(4)
The absolute values for the distance of the virtual origin are also determined from the figures for each angle. The following relations show the spread of jet with distance from the supply jet centreline and the distance of the virtual origin for angles of 30°, 60° and 90°. It is shown in Figs. 6, 8 and 10 that the virtual origin is always downstream of the jet centreline.

\[
\frac{y_{v}}{x_{0}} = \frac{1}{\sqrt{A_{out}}} \approx 0.081 + 0.0278 \sqrt{A_{out}}
\]

Angle 30° \[ \frac{x_{0}}{\sqrt{A_{out}}} \approx 0.34 \],

Angle 60° \[ \frac{x_{0}}{\sqrt{A_{out}}} \approx 1.90 \],

Angle 90° \[ \frac{x_{0}}{\sqrt{A_{out}}} \approx 0.95 \].

3.4. Discussion of results

The distance (x) form the impingement point up to the actual measuring point gives the best parameter for dimensionless representation of the experimental data. The results show that velocities for different angles considered are somewhat different. Although the largest velocities occur for angles of 60° and 90°, the velocities for these two angles at the same measuring points are almost the same. The small variations are due to measuring errors. The lower velocities for angle 30° are due to the damping effect of the adjacent wall.

Since the aim is to produce dimensionless correlations that can be used for selecting impinging jet devices the results
for angle $90^\circ$, which gives somewhat higher values, will be used for this purpose. Thus, if one considers a maximum velocity at a specific distance from the supply device, it is clear that the value at the same distance for other directions may be lower than that for $90^\circ$.

Based on this assumption, the equation for the maximum velocity at a distance $x$ from the jet centreline is

$$U_{\text{max}} = 2.45U_0 \left( \frac{x}{\sqrt{A_{\text{out}}}} \right)^{-1.10}.$$  \hspace{1cm} (8)

The exponent for an ideal case should be $-1.0$. However, due to friction losses and entrainment of the room air into the jet and also because the jet momentum cannot be conserved in a confined enclosure, a value of $-1.10$ is obtained from measurements.

Concerning the supply device height above the floor, the results show that the velocity profiles for both 300 and 950 mm supply heights are almost the same. The marginal difference in the maximum velocities is probably due to the smaller supply area in the case with $h = 950$ mm which gives somewhat higher outlet velocity. A higher supply outlet velocity results in a higher momentum and can affect the jet velocity spread over the floor. Therefore the height of the supply outlet has no significant influence on the velocity profiles over the floor, but small influences on the ventilation efficiency [4].

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Fig. 7. Decay of maximum velocity for angle $60^\circ$ and supply height of 950 mm.

Fig. 8. The rate of spread of the jet for angle $60^\circ$ and supply height of 950 mm.
4. Application to a classroom

A test room of $8.4 \times 7.2 \times 3 \text{ m}^3$ and 25 person-simulators, placed in the room to represent a teacher and 24 students, was used for evaluating the performance of the IJV system. A climate chamber attached to the room was used to simulate extreme winter and summer conditions, see Fig. 11. The heat output from the person-simulators were 2375 and 525 W was considered for lighting giving a total cooling load of 2900 W ($48 \text{ W/m}^2$). In all tests the air flow rate was 10 l/s per person and the supply air temperature was kept constant at $15^\circ \text{C}$. The outdoor temperature was kept at $-21^\circ \text{C}$ to simulate winter conditions and at $+25^\circ \text{C}$ to simulate summer conditions. However, only results for the summer condition are presented here. Results from tests under winter conditions can be found in Ref. [4]. Two types of air supply devices have been evaluated; one is based on the IJV principle and the other on the WDV concept. The devices were tested by measuring the air temperature, air velocity and air quality (local mean age of air) in the occupied zone. The room area was divided into 12 zones and a stand placed in the middle of each zone was used to measure
the required quantities at different heights. The local mean age of air was measured by using the tracer gas decay technique at 1.2 m above floor. Further details can be found in Refs. [2,4].

To assess the effectiveness of the two ventilation systems for measurements and CFD simulations, well-known parameters given below are used [9].

Ventilation effectiveness for heat distribution or removal ($\varepsilon_h$): This is similar to a heat exchanger effectiveness and is defined by

$$\varepsilon_h = \frac{T_o - T_i}{T - T_i}$$

(9)

Ventilation effectiveness for contaminant removal ($\varepsilon_c$): This is a measure of how effective the ventilation system is in removing internally produced contamination. It is defined by

$$\varepsilon_c = \frac{C_o - C_i}{C - C_i}.$$  

(10)

In Eqs. (9) and (10), $T$ is temperature ($\degree C$), $C$ is the contaminant concentration in parts per million (ppm), the subscripts $i$ and $o$ refer to inlet and outlet, respectively, and $(-\overline{)}$ represents the mean value for the occupied zone (to a height of 1.8 m). The values of $\varepsilon_h$ and $\varepsilon_c$ is dependent on the method of room air distribution, room characteristics, heat and contaminant sources, etc.
In addition, expressions for the predicted percentage of dissatisfied (PPD) and predicted mean vote (PMV) are used here and these are defined in Ref. [10].

5. CFD calculations

The CFD program VORTEX [11] has been used to predict the airflow properties in the classroom. This program has been developed for the simulation of airflow, heat transfer, mean age of air distribution, PPD and PMV in enclosures. The code uses the standard $\kappa-\epsilon$ turbulence model and has been developed for ventilation research, which may be more suitable for ventilation simulations than the more general-purpose codes. In the simulations, the measured mean temperatures of the six room surfaces have been used as boundary conditions. The number of nodes used were $49 \times 39 \times 37$ giving a resolution of 0.17 m in the horizontal plane and 0.081 m in the vertical direction. The distance from the floor of the impinging jet outlet was 0.90 m in both the measurements and the CFD simulations.

6. Results from CFD and measurements in classroom

Fig. 12(a) and (b) shows the air flow patterns from the CFD results for the two supply methods at a height of 10 cm from the floor. Although, the air supply velocity of
1.56 ms$^{-1}$ in the case of IJV was much higher than in the displacement ventilation case the velocity close to the floor decays very rapidly and it is only quite large at a short distance from the jet impinging point on the floor. The spread of the impinging jet over the floor produces a velocity near the floor which is similar in magnitude to that of the displacement ventilation system. However, the floor layer in this case is much thinner than that in the case of the WDV.

The temperature gradients for the two cases are shown in Fig. 13. The agreement between the measurements and CFD simulations are good. The recommended temperature gradient of 3 K or less is achieved.

The mean velocity gradients are shown in Fig. 14. Except for heights up to 15 cm above the floor the velocities are much lower than 0.15 m/s for both cases, indicating a comfortable environment. The velocity distribution over a horizontal plane 10 cm above the floor (see Fig. 11) is plotted in Fig. 15. Only the wall displacement system shows a tendency for a velocity higher than the recommended maximum of 0.15 m/s closed to the walls.

Fig. 16 shows the PPD. One can see from the figure that in a large portion of the room PPD is below 10% for both systems and this is acceptable for such large heat loads [10].
The normalised mean age of air (i.e. the local age divided by that at the exhaust) is shown in Fig. 17 for all the measuring points at a height of 1.2 m above the floor (breathing zone level). There is some agreement between the WDV and the IJV results at some of the points but not at others. This may be due to the difference between the momentum in the two systems and its interaction with the local buoyancy forces. The CFD results show similar trends to measurements.

The temperature and velocity profiles at all measuring points 1.2 m above the floor (breathing zone) are shown in Figs. 18 and 19. Both systems show good temperature distributions but the WDV gives higher values. Considering the velocity field for both cases one can see that the velocities are very low and there is little tendency to draught risk.

A general consideration of all the parameters studied show that both systems can handle the extreme situation they have been exposed to with some small differences between them.

The predicted contaminant removal effectiveness ($\varepsilon_c$), was 140% for WDV and 130% for IJV indicating the ability...
of both systems to remove the contaminants in an effective way. The corresponding value of the predicted ventilation effectiveness for heat distribution for the two cases was 1421 and 142 respectively, which again indicates a high and very close value for the two systems. The PMVs were −0.63 and −0.61, respectively, which are slightly less than −0.5 that is suggested by Fanger [10] to be acceptable.

Fig. 20 shows four sequences of smoke visualisation for an impinging jet supply device. One can see that when the jet reaches the heat source (sitting person) a plume forms and due to sufficient momentum the jet continues along the floor. This is the advantage of impinging jet ventilation compared to a displacement system, in which case the flow sometimes has insufficient momentum to continue pass the heat source and its totally consumed by the plume.

Similar types of impinging jet supply devices (Air Queen) have been installed in different buildings, e.g. schools, offices, industrial buildings, etc. The device is similar in looks to a duct and is mounted onto a wall. It supplies air downward to the floor and contaminated air is exhausted from ceiling level. Four applications of impinging jet ventilation in Sweden are shown in Fig. 21(a)–(d). One of these
represents an installation in a school, one in an office building and two in industrial buildings. Fig. 21(a) shows an installation of impinging jet supply in System 3R International, which was installed to replace displacement devices in a building where precision instruments/tools are manufactured. When displacement ventilation was used there were many complaints from the workers and after their replacement with impinging jet systems no compliment was reported. Fig. 21(b) shows IJV installed in a factory for heavy metal industry. Fig. 21(c) shows the application in the Strömsbro School in Gävle and Fig. 21(d) is in the Swedish Taxes Office in Stockholm. Since installing the impinging jet ventilation system in all these buildings no complaints have yet been reported.
Another advantage of impinging jet ventilation is that the duct itself has a sound damping effect, which reduces aerodynamic noise at the supply terminal.

7. Conclusions

The results presented in this paper have shown that a floor level air distribution can handle a full room heat load in an acceptable manner. Although both the WDV and the IJV systems show similar tendencies, some small differences in their performance were observed. Because of the better balance between buoyancy and momentum forces the IJV system shows slightly better mean age of air and velocity distributions. The experiments show that after a short distance from the impinging region and due to the damping effect of the floor, the jet velocities have decreased to an acceptable level. Furthermore, the IJV system can also be used for both heating and cooling purposes.

In this paper we have evaluated the IJV system under certain conditions which are by no means exhaustive. Therefore, this new system may need further studies by ventilation researchers and designers to fully understand its performance under various conditions experienced in practice.

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