AIR JETS IN VENTILATION
APPLICATIONS

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ABSTRACT

The purpose of air distribution systems for HVAC is to create proper air quality and thermal conditions in an occupied zone. In mixing type air distribution systems air is supplied into a room through various types of outlets and distributed by turbulent air jets. These air jets are the primary factors affecting room air motion. The ASHRAE handbook recognises four major zones of maximum velocity decay along a jet.

Although numerous theoretical and experimental studies have been conducted to develop turbulent air jet theory from the 1930's, air jet performance in the further field from the outlet is still not well understood.

Many studies were therefore carried out, and the following conclusions can be drawn from them:

?? The end centerline velocities of zone 3 for both "free" jet and wall jet could strongly depend on the outlet velocities and room size.

?? The $K$-value of wall jets could be a function of both outlet velocities and outlet size.

?? It is very important to choose suitable sampling time to evaluate jet performance.

?? CFD can not always be used to predict jet behaviour, especially for the jet with low outlet velocity and in the area far away from the outlet. However, for a two-dimension wall jet, CFD could be a powerful tool for designers.

KEYWORDS

air jet, centerline velocity, $K$-velocity, air diffuser, ventilation, measurement, CFD
给所有爱我和我爱的人

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CONTENTS

CONTENTS 9

Nomenclature 11

List of papers 13

1 INTRODUCTION 15

1.1 Overview 15

1.2 Objectives 16

2 AIR JET THEORY 19

2.1 Overview 19

2.2 Turbulent free jet 20

2.2.1 Velocity distribution within zone 3: a literature review 21

2.2.1.1 Velocity distribution in the cross-section of a jet 21

2.2.1.2 Centerline velocity decay 21

2.2.1.3 Effect of exit velocity on $K$-value and virtual origin 24

2.2.1.4 Influence of outlet characteristics on free axial jet (paper 5) 25

2.3 Wall jet 26

2.3.1 Overview 26

3 EXPERIMENTAL FACILITY AND MEASUREMENT TECHNIQUES 29

3.1 Free jet facility 29

3.2 Wall jet facility 31
3.3 Velocity measurement

3.3.1 Overview 33

3.3.2 CTA in our test 34

3.3.2.1 Calibration method for low-speed hot-wire anemometer (paper 1) 34

3.3.2.2 Effect of sampling time for air jet velocity measurement (paper 2) 35

3.3.2.3 Effect of temperature and turbulence intensity on measurement accuracy 36

4 MEASUREMENT RESULTS 39

4.1 Free jet 39

4.1.1 Effect of outlet velocity on the transition point between zone 3 and zone 4 in free jets (paper 3) 39

4.1.2 Influence of turbulence intensity on integral momentum balance 42

4.2 Three-dimensional wall jet (paper 4) 44

5 COMPUTATIONAL FLUID DYNAMICS (CFD) 47

5.1 Introduction 47

5.2 Basic conservation equation 48

5.3 Application of CFD on jet simulation 49

5.3.1 Historical background 49

5.3.2 A CFD study for airflow distributions at floor level in a SISIV room (Paper 6) 50

5.3.3 One jet simulation with standard k-ε model 50

6 SUMMARY 55

REFERENCES 57
Nomenclature

\( A_o \) = outlet area
\( A_R \) = cross-section area of test room
\( c_p \) = specific heat capacity at constant pressure
\( D \) = diameter of nozzle
\( K \) = zone 3 centerline velocity decay coefficient
\( M_o \) = outlet initial jet momentum
\( M_x \) = jet momentum at position \( x \)
\( Q \) = heat source
\( s \) = total spread angle of the jet
\( T \) = temperature
\( u_i \) = velocity vector components
\( U \) = mean velocity in \( x \) direction
\( U_o \) = outlet velocity
\( U_x \) = centerline velocity
\( V \) = mean velocity in \( y \) direction
\( W \) = mean velocity in \( z \) direction
\( x \) = coordinate in axial direction
\( x_i \) = spatial co-ordinate components (\( i=1, 2, 3: x, y, z \) direction)
\( x_p \) = distance from the opening to the virtual origin of the jet
\( y \) = coordinate in radial direction
$y_{0.5} = y$-coordinate where $U/U_x = 0.5$

$\beta = \text{angle between the centerline and the line corresponding to } y_{0.5}$

$\eta = y/y_{0.5} = y/((x-x_p) \cdot \tan \beta)$

$\lambda = \text{thermal conductivity}$

$\rho = \text{air density}$

$\mu = \text{dynamic viscosity}$

$\varsigma = \text{coefficient of volumetric expansion}$
Parts of this thesis are the following papers:


1 INTRODUCTION

1.1 Overview

The main task of a ventilation system is to control the supply and exhaust of air so that a good indoor air quality and thermal environment is obtained in the occupied area. There are three different ventilation principles used to control air distribution within ventilation rooms, see Etheridge and Sandberg (1996):

- **Unidirectional flow ventilation:** the whole ceiling or side walls are used as the air inlet so that a vertical or horizontal unidirectional flow is created from the ceiling to the floor or from one side wall to the other. Contaminated air is then expelled by piston effect from upwind to downwind, so that the occupied region is always kept clean. This method is used mainly for industrial or laboratory clean rooms. Many experimental studies, e.g. Ljungqvist and Reinmüller (1998) and numerical studies, e.g. Chen et al (1991) have been carried out to investigate indoor air quality and thermal comfort in clean-rooms.

- **Mixing ventilation:** conditioned air is normally supplied by an air terminal device and a jet is generated in front of the opening. The jet may either enter the room as a free jet, or it may attach itself to a surface such as the ceiling to form a wall jet. The region between the ceiling and the occupied zone serves as an entrainment region. Due to the entrainment of room air, the supply velocity and temperature difference between supply air and room air become smaller and smaller. The total air (the mixture of discharge air and entrained air) then dilutes the contaminants in the occupied zone in order to achieve an acceptable air quality. This method is widely used in offices and commercial buildings.

- **Displacement ventilation:** low-temperature air is supplied with a low-velocity diffuser located at or close to floor level. The flow pattern in a room is governed by the buoyancy forces and can be shown as a well-defined flow from the lower part of the room to the upper level. When designing a displacement ventilation system the resulting vertical temperature gradient in
the room is of great importance, see Mundt, E. (1996). Displacement ventilation is traditionally used in industrial buildings where a special indoor contaminant control is required. Its application has also been extended to office buildings, restaurants and lecture halls.

To choose a suitable air supply method for a specific application, the following criteria need to be considered:

- room floor area and height;
- number of occupants in the room and their location;
- heating/cooling loads, air exchange rate and type and amount of contaminants released;
- technological processes, size of process equipment and space obstruction of this equipment; and
- type of HVAC system used (variable air volume or constant air volume);

The mixing ventilation has been used intensively in the Scandinavian countries because it is an effective method of providing accurate temperature control and good air quality. When designing a mixing ventilation system, knowledge of jet performance from an air terminal device is one of the most important elements, because the recirculating air movement in the room is generated and maintained by the momentum flow from the supply opening, and only to a small extent maintained by the buoyancy force, see Nielsen (1995). Although numerous theoretical and experimental studies have been conducted to develop turbulent air jet theory from the 1930’s, for instance, Rydberg et al (1946), Becher (1949), Koestel et al (1950), Nottage (1951), Tuve (1953), Helander et al (1957), Shepelev (1961), Abramovich (1963) and later Regenscheit (1970), Jackman (1971), Holmes (1973), Nielsen (1973), Skåret (1987), Miller (1990), Malmström et al (1992), Grimitlin (1994) and Malmström et al (1997), air jet performance in the far field from the outlet is still not well understood. When and where does jet disintegration occur, for instance? This problem is of central importance to the ventilation process because at this point the jet flow stops entraining room air and supply air starts to diffuse into the occupied zone.

1.2 Objectives

The main objective of this study is to get improved knowledge about the velocity decay of jets in the far field from the outlet. Therefore, the first part of the project
was to retest the existing knowledge of air jets, especially the study carried out by Malmström et al (1992), which examined the effect of nozzle diameter and supply velocity on the maximum mean velocity decay of the jet and concluded that centerline velocity decay coefficient $K$ is a function of outlet velocity $U_o$ rather than of outlet Reynolds numbers. Based on these studies, further experimental tests of jet behaviour with different outlet sizes were carried out. The goals, more specifically, are:

- to retest the conclusions reached by Malmström et al (1992);
- to investigate the velocity decay mechanism of isothermal air jets, based on parameters such as outlet condition and room size;
- to evaluate CFD-modelling of jets; and
- to propose new methods for measurement and data analysis.

In practice, most air jets used for ventilation and air conditioning purposes have a temperature difference to the room air. This difference will of course influence jet behaviour. However, the current work is limited to isothermal jets in order to get a basis for future work with jets with temperature differences.
2 AIR JET THEORY

2.1 Overview

In HVAC applications, air jet theory is a well-practised basis for analysing and designing mixing ventilation systems. Air jet theory divides the room into a jet region and an occupied zone. The aim is to predict the extreme values of air velocities and temperatures in the occupied zone.

Compared to the "typical" jet studied in fluid mechanics, the jets in HVAC applications generally have comparatively large outlet dimensions and low outlet velocity. The jet flow in room ventilation can be of different types or a combination of types. If the wall, ceiling or obstructions do not influence the air jet it is considered a free jet. If the air jet performance is influenced by reverse flows, created by the same jet entraining ambient air, this is called a confined jet. Particularly, if the air jet is discharged parallel to a wall with one edge of the outlet close to the wall, it is a wall jet. When considering the temperature difference between supply air and room ambient air, air jets can be divided into isothermal jets and non-isothermal jets. This temperature difference generates buoyancy forces in the jet, affecting the trajectory of the jet, the location where the jet attaches and separates from the ceiling or floor and the throw of the jet. The significance of such effects depends on the ratio of thermal buoyancy to inertial force (characterised by Archimedes number or thermal length).

This investigation was concentrated on the axisymmetric isothermal air jet, which is a base for calculation of other types of jets, such as nonisothermal jets (note that the differences between the two jet types are significant).


2.2 Turbulent free jet

Principally, a turbulent free jet is a jet that flows into an infinitely large space, and the phenomenon has been widely studied by many researchers. In the 1850’s, Helmholtz and Kirchhoff were the first to formulate and solve the jet problem. The first experiments on plane turbulent jets were conducted by Förthmann (1936). An early measurement of the turbulence characteristics of circular jets was done by Corrsin (1946). Studies in fluid dynamics were continued by Wygnanski et al. (1969) and Rajaratnam (1976). Recently, an extensive experimental study of momentum balance that received a great deal of attention was conducted by Hussein et al. (1994).

In HVAC application, early studies include those by Rydberg and Norbäck (1946), Tuve (1953), Nottage (1951) and Koestel (1954). Becher (1966) and Grimitlin (1970) studied the influence of grills with diverging vanes on free air jets. Malmström et al. (1992) examined the effect of nozzle diameter and supply velocity on the maximum mean velocity decay of the jet; this study indicated that $K$-value is a function of $U_o$ rather than of Reynolds numbers.

Based on these studies, the development of a jet is divided into four zones, related to centerline velocity decay, see figure 2.1:

- zone 1, a conical zone where centerline velocity is equal to outlet velocity;
- zone 2, a transition zone where the velocity starts to decrease, often approximated as proportional to $x^{-0.5}$, where $x$ is the axial distance;
- zone 3, where transverse velocity profiles are similar at different values of $x$ and velocity decay is assumed proportional to $x^{-1}$; and
- zone 4, the jet terminal zone where centerline velocity rapidly decreases.

Although this zone has been studied by several researchers (such as Madison et al. (1946), and Weinhold (1969)), its different mechanisms are still not well understood.

In an axial jet, the two first zones are strongly influenced by the diffuser, the third zone is the developed jet, and the fourth zone is the zone of jet termination. In the three first zones, room air is entrained into the jet and mixed with supply air. In the fourth zone, the jet collapses inwards from the boundaries and the supply air is distributed to the room air as the jet disintegrates.
2.2.1 Velocity distribution within zone 3: a literature review

2.2.1.1 Velocity distribution in the cross-section of a jet

In the region of fully developed jet flow, axial mean velocity profiles were found to be similar, see Förthmann (1934), Albertson et al (1948), and Reichardt (1942). Based on his experimental studies, Reichardt (1942) proposed that the following Gauss error-function could be used to describe the jet profile within zone 3:

\[
\frac{U}{U_x} = \exp(-\ln 2 \cdot \eta^2)
\]  (2.1)

It has been demonstrated by Ruden (1933), Albertson et al (1948), Taylor et al (1951), Becher (1949), Nottage (1951), Shepelev (1961), Capp et al (1990), Grimmitlin (1994) and other researchers that this Gauss error-function profile is comparable with data obtained in studies of both nozzle jets and air diffuser supply jets.

2.2.1.2 Centerline velocity decay

Centerline velocity in zone 3 of the jet can be calculated from equations based on the principle of momentum conservation along the jet. (Note that the following text is based on Malmström et al (1998).)
The momentum at any distance, \( x \), is:

\[
M_x = \rho * 2\pi * \int_{jet} U^2 \, dy
\]

(2.2)

The initial momentum with uniform outlet velocity can be expressed as:

\[
M_o = \rho * A_o * U_o^2
\]

(2.3)

For a nozzle, the area can be calculated using:

\[
A_o = \frac{\pi}{4} D^2
\]

(2.4)

According to the principle of momentum conservation:

\[
M_o = M_s
\]

(2.5)

Applying the Gauss function in equation 2.1 with equation 2.2, 2.3, 2.4 and 2.5, the mean centerline velocity \( U_x \) can be solved as:

\[
U_x = \sqrt{\ln 2} \cdot \frac{1}{\tan \beta} \cdot \frac{U_o \cdot D}{(x - x_p)}
\]

(2.6)

If we write:

\[
K = \sqrt{\ln 2} \cdot \frac{1}{\tan \beta}
\]

(2.7)

Then equation 2.6 can be simplified as:

\[
\frac{U_x}{U_o} = K \cdot \frac{D}{x - x_p}
\]

(2.8)

or

\[
U_x = \frac{2K}{\sqrt{x}} \cdot \frac{\sqrt{M_o / \rho}}{x - x_p}
\]

(2.9)

Equation 2.8 is often used to calculate the centerline velocities of fully developed air jets or the throw for a certain distance \( x \), see ASHRAE Handbook 1997.
Obviously, the centerline velocity decay coefficient $K$-value is an important factor for describing jet performance. $K$-values are usually calculated by measuring mean velocities in different centerline positions of the air jet.

Figure 2.2 shows a case with typical values for HVAC ($U_o=9$ m/s) jets and the quotient $U_o/U_x$ plotted against the normalised distance from the outlet, $x/D$. For a jet from a nozzle, equation 2.8 corresponds to the straight line in the graph, fitted to the measured values. The coefficient $K$ (or rather $1/K$) is a measure of the slope of the line, and the distance $x_p$ between the outlet and the “virtual origin” of the jet is defined by the intersection of the line and the x-axis. The part of the jet represented by the linear relationship in figure 2.2 corresponds to zone 3 and at long distances the velocity decay is faster (zone 4). Since zone 3 has no well-defined boundaries, it could be concluded that decisions about which measured points should be included in the fit of the straight line have an influence on the estimated value of $K$ and a big influence on the estimated value of $x_p$.

![Figure 2.2: Centerline velocity variation with distance from the nozzle exit ($D=41mm$, $U_o=9$ m/s), $K=6$ and $x_p/D=0.5$.](image)

However, as shown in equation 2.7, values of $K$ also can be calculated from the velocity profile. Malmström et al (1992) showed that $K$-values re-evaluated from velocity profiles measured by Nottage (1951) are comparable with results from mean velocities at different centerline positions of the air jet; this may be evidence for the validation of this jet model.
2.2.1.3 Effect of exit velocity on $K$-value and virtual origin

One consequence of equation 2.9 is that if the values of $K$ and $M_o$ (and hence also of the outlet Reynolds number) are the same for two jets from different nozzles, the same jet will appear at the same distance from virtual origin $(x - x_p)$. However, as shown in figure 2.3, studies in Malmström et al (1992) clearly indicate that for air jets from round nozzles, the centerline velocity decay coefficient $K$ could be described as a simple function of outlet velocities. No evidence of simple dependence on the outlet Reynolds number is found.

![Graph showing K-values based on outlet velocities for free jets.](image)

Figure 2.3: K-values based on outlet velocities for free jets. (from Malmström et al 1992)

In contrast to $K$-values, the location of virtual origin $x_p/D$ is a parameter that is strongly influenced by the decision about which measurement points to include in the fit of the straight line. However, the initial conditions of the air jets have been shown as one factor affecting the location of this parameter. A disturbance at the outlet, causing increased entrainment in a limited part of the jet, can often be described as an upstream shift of the virtual origin position, see paper 5. In Figure 2.4 the variation of $x_p/D$ as the function of outlet velocities is presented, based on Nottage (1951) and Malmström et al (1992). It indicates that when outlet velocities are below 6 m/s, the locations of virtual origins have a tendency to get close to zero with increasing outlet velocities. In Nottage (1951), a “turbulence promoter” was used to test the effect of outlet turbulence intensity levels upon axial velocities. Two outlet velocities were chosen, $U_o = 1.8$ m/s and $U_o = 12.5$
m/s. Experimental results showed that there was no change in the 12.5 m/s jet for different outlet turbulence levels, but the virtual origin was shifted upstream by 4 to 5 nozzle diameters for the 1.8 m/s jet. The results shown in figure 2.4 are also an indication that jets with lower outlet velocities are more sensitive to initial conditions.

![Figure 2.4: Values of $x_p/D$ as a function of outlet velocity.](image)

2.2.1.4 Influence of outlet characteristics on free axial jet (paper 5)

The subject of this paper is grille influence on free, axisymmetric, isothermal air jets. The most well defined outlet is a conical nozzle. Grilles compared to nozzles seem to have two effects on the momentum flow rate. The first is that, for a diffuser, there is no single outlet velocity $U_o$ but rather velocities within a big interval. The second effect depends on a region of slightly decreased pressure just downstream of the grille causing an adverse force on the flow field and resulting in loss of momentum flow rate.

For a very special type of "diffuser", perforated plates, "momentum loss coefficient" has the value between 0.3 and 0.7. Some studies indicate that "momentum loss coefficient" is a function of Reynolds number for the flow through the perforated holes, of the number of holes, and of the percentage of perforation. More normal diffusers have smaller momentum losses, that is values
of "momentum loss coefficient" closer to one. There are indications that the velocity decay coefficient also is a function of outlet velocities.

If the flow in or close to a nozzle is disturbed, induction of room air may increase. This, typically, is equivalent to the natural induction in the jet over a short distance. The influence can be described as a shift upstream in the location of the apparent source of the jet. The influence of grilles, vanes and other devices seems to be similar.

### 2.3 Wall jet

#### 2.3.1 Overview

Wall jets are created when high velocity air is injected along a surface into a large open space. An example of a wall jet from a nozzle is shown in figure 2.5. However, there is a certain distance between the outlet and the ceiling. The turbulent mixing layer will entrain air from both sides of the jet, and a lower pressure on the upper side will deflect the jet and a wall jet will be established at some distance from the outlet. The effect that generates the deflection of the flow in figure 2.5 is called the Coanda-effect.

![Diagram of a wall jet](image)

Figure 2.5: A nonflush-mounted three-dimensional wall jet.

The earliest three-dimensional wall jet work in HVAC application could be Farquharson (1952) in which the effect of the proximity of a wall on the behavior
of the jet from square orifices was studied. Measurement results showed that the maximum velocity decay coefficient $K = 6.5$ in a free jet should be replaced by 9.0 when the edge of the discharge opening is adjacent to a wall.

This work was followed by that of Tuve (1953), Koestel (1957) and Sforza et al (1970). Sforza et al (1970) presented an extensive experimental investigation of the mean properties of turbulent wall jets from various rectangular orifices. From the results obtained, the wall jet flow field was found to be also characterized by axial velocity decay like a free jet. An analytical approach to estimate the shear stress distribution at the plate was also presented.

Nielsen et al (1988) measured a nozzle–like diffuser ($D=140\text{mm}$) located close to the ceiling under both isothermal and thermal conditions and found that the flow was independent of the Reynolds number for all the measurements ($U_x > 2.9 \text{ m/s}$). The temperature distribution was independent of the Archimedes number for all supply velocities and could also be described by an equation similar to the velocity distribution equations.

A jet produced from a rectangular outlet was studied by Kirkpatrick et al (1998). Using a simple jet model, the general nature of jet flow characteristics, such as the occurrence of velocity decay coefficients, virtual origins and spread angles was deduced and compared with previous studies by Sforza and Herbst.

In Sandberg (1998), experimental studies of both three- and two-dimensional jets in rooms were presented and it was found that the traditional wall jet theory, based on the expansion of a jet in an infinite ambient, was also useful in room air flow. These studies also showed that there might be confinement phenomena that have strong influence on jet development.

Finally, a good number of publications on the characteristics of turbulent wall jets can also be found in the field of fluid mechanics, such as Newman et al (1972) and Padmanabham et al (1991). An excellent review of wall jet flows is given in Launder and Rodi (1981) and (1983) and in Abrahamsson et al (1997).
3 EXPERIMENTAL FACILITY AND MEASUREMENT TECHNIQUES

3.1 Free jet facility

One reason for the lack of knowledge about the far field of jet expansion is the difficulty of making reliable measurements. Particular care must be applied in the experimental procedure.

In this study, all the free jet measurements were carried out in the laboratory hall of the Division of Building Services Engineering. The experimental setup consisted of a fan, a settling chamber, and ASME standard long radius nozzles ($D=41$ mm and $D=76$ mm). A fan with a frequency controller and an air-cooling coil was used to deliver the required airflow rate with the same air temperature as the settling chamber, which was 1.2 m long and 0.8 m in diameter. Two internal fine mesh screens were used to produce a uniform velocity profile and reduce the turbulence level, see also figure 3.3. Figure 3.1 shows that the velocity distribution in the exit of a nozzle with supply velocity $U_o = 6.1$ m/s is a true "top hat" profile. Figure 3.2 shows the turbulence intensity at the exit for three nozzle sizes ($D=41$ mm, 76 mm and 152 mm). It is interesting to note that turbulence intensities increase with outlet velocities when outlet velocities are below a certain value.

The nozzle with the settling chamber was located freely in a large laboratory area. The whole laboratory's size is approximately 15m×6m×6m. With the other equipment in this laboratory taken into consideration, the total free space for our measurement was approximately 10m×4.4m×6m. According to Capp et al (1990), the distance from the outlet for which the jet can be considered "free" (that is, independent of the room) depends on the square area of the room and error in jet momentum that can be accepted. A frequent recommendation is that a jet can be considered as free when a distance less than $(A_r)^{0.5}$, where $A_r$ is the area of the room, normal to the jet flow, see Hussein et al (1994) and Malmström et al
(1992). At this distance 98% of the outlet momentum is still contained in the jet flow. In our case, this distance is estimated to be at least 4.4m.

Before measurement, the room’s air movement without an air jet was checked with both smoke and hot-wire probe almost for every test. The smoke test showed that the room air movement is small and the hot-wire probe showed that the room air velocity in the measurement area was always less than 7 cm/s with turbulence intensity always around 50%. A smoke generator was also used to study the recirculation airflow created by the jet itself and the entrainment of secondary air, to make sure that the room’s air motion was not strong enough to influence the free jet.

The test room temperature had been studied at 6 points with a CHINO dot type recorder for one week and the room temperature was always found to be between 21°C and 23°C. During the measurement period, a calibrated Swematemp 360 type thermometer (accuracy=±0.3°C) was used to check the temperature in the nozzle outlet and measurement areas.

![Figure 3.1: Velocity profile in the exit of nozzle (D= 41mm, \(U_o=6.1 \text{ m/s}\).](image)

Figure 3.1: Velocity profile in the exit of nozzle (\(D=41\text{mm}, U_o=6.1 \text{ m/s}\)).
3.2 Wall jet facility

All the measurements were carried out under isothermal condition in a full scale test room located within the laboratory hall of the Division of Building Services Engineering, at the Royal Institute of Technology. The dimensions of the test room were 4.2m×3.6m×2.7m. All four walls of the test room were insulated and the ceiling was smooth. The supply and exhaust were located in the same wall. To study the influence of outlet size, three ASME standard long radius nozzles ($D=43\text{mm}$, $D=76\text{mm}$ and $D=152\text{mm}$) were tested. The nozzles were located as close as possible to the ceiling (about 3 cm from the ceiling to the nozzle edge) so that they could generate three-dimensional wall jets along the ceiling. Air was supplied by a frequency controlled centrifugal fan and the flow was led through a cooling coil before it entered the settling chamber, which was 1.2m long and 0.8m in diameter, see figure 3.3. Two internal fine mesh screens in the settling chamber were used to produce a uniform velocity profile and reduce the turbulence level, see figure 3.4.
Figure 3.3: Experimental set-up for wall jet

Figure 3.4: Velocity profile in the exit of nozzle ($D = 43\text{mm}, U_o = 8.3 \text{ m/s}$).
3.3 Velocity measurement

3.3.1 Overview

Many different measuring techniques have been applied in the velocity field of the turbulent jet, e.g. Pitot tubes, Hot-Wire Anemometry (HWA), Laser-Doppler Velocimetry (LDV) and flying hot-wire anemometry. Pitot tubes were used in the early jet investigation, see Nottage (1951) and Förthmann (1936). HWA technique has been employed in a majority of all measurements since the beginning of 1960's. However, recently a few tests using LDV have been presented, see Hussein et al (1994) and Zou et al (2000).

The velocity measuring techniques may be divided into the following categories, see Taghi (1996):

1. intrusive techniques, which require a probe or a tube at the measuring point, for example a HWA or a Pitot tube.
2. non-intrusive techniques which do not require a probe at the measuring point, for example LDV.

According to the measurement size, velocity instruments can also be divided into:

- point measuring techniques which provide information only at several points. These are very time consuming and almost impossible if one wants detailed knowledge about the whole field. Most of the measurement methods belong to this area; and

- whole field measuring techniques which can provide information about a whole field, for example Particle Image Velocimetry (PIV)

However, choosing a suitable device to measure jet flow in the far field from an outlet is not easy. The difficulties stem from the following facts:

- jet flow can occupy a very large volume;
- jet flow far away from the outlet is very sensitive to any disturbance. Even when no disturbance is apparent the jet is not quite steady;
- the velocity field is three-dimensional;
- the relative turbulence intensity is relatively high (>20%); and
- the velocities are low (in the most interesting cases the mean velocity is <<1 m/s).
3.3.2 Anemometry in our test

In this study, the velocity measurements were made with a constant temperature hot-wire probe having a single, unplated tungsten sensor, 1.5mm long and 5µm in diameter. This probe was operated at an overheat ratio of 1.8 and placed on a stand with wheels. Data was acquired and converted by an An-2000 computerized anemometer system. Typical sampling time was 180 seconds with sampling frequency at 150 Hz.

3.3.2.1 Calibration method for low-speed hot-wire anemometer (paper 1)

The hot-wire anemometer is a popular instrument in fluid mechanics research since it can provide continuous signals of local velocity information as well as information on turbulence properties. Calibration of hot-wire probes at relatively high velocity (e.g. \( U > 2 \text{m/s} \) for airflow) can be easily carried out by measuring the outlet velocity of a calibration nozzle with a “top hat” velocity profile. In the low-speed range, however, the velocity profile is hard to keep uniform. Lee and Budwig (1991) indicated that the velocity data between 0.15 m/s and 0.85 m/s obtained by a calibration jet was about 4-50% lower than the real value. Application of hot-wire anemometry at low velocities calls for special calibration techniques.

In this paper, an improved laminar pipe flow method is presented. The whole calibration consists of two airtight containers, an adjustable pipe in one of them, some flexible plastic connection pipes and a calibration copper pipe, see figure 1 in paper 1.

The purpose of the first container is to provide constant water flow rate to the second container. This is achieved as long as the water level in container 1 is above the opening slit of the air pipe, as the pressure at the level of the horizontal part of the air pipe will be constant. Constant flow is produced through a specially designed air pipe: a copper pipe bent at 90° angles, resembling a half square. The top end of the air pipe is open to outside atmosphere. The lower part of this pipe is parallel to the bottom surface of the container.

When the valve between the two containers is open, water will flow from the first container to the second one. Since the containers are air tight, the pressure at the slit surface will decrease. However, this pipe is connected to the outside atmosphere. When the pressure decreases to the outside air pressure, the air pipe begins to carry outside air and release air to the container to keep the pressure of slit surface always equal to outside air pressure minus pressure loss in the pipe.
The special design of slit helps to release air smoothly and continuously. After that, the pressure difference between the slit of the horizontal pipe and the outlet of the container will be constant $\Delta H \cdot \rho \cdot g$, where $\Delta H$ is the distance from the outlet to the slit of the horizontal pipe, $\rho$ is water density and $g$ is the acceleration of gravity. As long as the slit is below the water surface, the first container will provide a constant water flow rate to the second container. The water flows into the second container at a constant rate and pushes the same volume of air into the calibration tube. This water flow rate can be easily measured with a balance weighing container and a watch.

This method to create stable water flow rates has been used for instance at the laboratory of Heating and Ventilation at KTH. The calibration results of this equipment has been compared with another certified low-velocity calibration test device in SP (Swedish National Testing and Researching Institute) which showed that the greatest difference between these two devices was 3.5 % and the largest deviation was 4 cm/s, see figure 3.5.

![Figure 3.5: Comparison between the reported device and the low-velocity calibration test device in SP.](image)

### 3.3.2.2 Effect of sampling time for air jet velocity measurement (paper 2)

The $K$-value is an important factor for describing jet performance. $K$-values are usually evaluated by measuring mean velocities in different centerline positions of the air jet. In most experiments, sampling time scales of a couple of minutes are...
used in the whole measurement field, which may be too short for measurement points far away from the outlet, perhaps resulting in large measurement errors. In order to study the influence of sampling time, an air jet \((D=41\text{mm}, U_o=11\text{m/s})\) was investigated. Five different centerline velocities at \(x/D=16, 24, 40, 50\) and 60 were measured with the total measuring period for each point being 45 minutes, which was assumed to give a real mean velocity. The measurement results imply that, for an air jet with known outlet conditions, the distance from the outlet to the measurement point is an important factor for the accuracy of mean centerline velocity measurements. Therefore, a couple of minutes could be not long enough as sampling time for the measurements taken in the field far away from outlet.

The second part of the study focuses on the variance of \(K\)-values evaluated from different sampling times. Eight different sampling time scales (1s, 2s, 4s, 8s, 16s, 32s, 64s and 180s) were simulated from the available data of a full 3-minute measurement period and used to evaluate \(K\)-values. When a short sampling time was used to evaluate jet performance, the centerline velocity decay rate maintained the behaviour \(U_x \propto x^{-1}\) for a longer distance but with higher values. This implies that when disturbances play a key role for jet disintegration, it is perhaps possible to use different sampling times to separate the influence of the disturbances from the pure jet behaviour.

### 3.3.2.3 Effect of temperature and turbulence intensity on measurement accuracy

It is well known that air temperature influences the output from a constant temperature anemometer. One of our studies, see paper 1, shows that when the room air temperature changes by one degree, the output of the anemometer could deviate 2% for some air speeds. To avoid measurement error caused by temperature two sets of calibration curves were applied for different temperature ranges, one evaluated from 21.5ºC and another from 22.5ºC.

However, an important limitation on the hot-wire anemometry is the inability to measure accurately in airflow with high turbulence level. The CTA is only capable of measuring at rather low turbulence intensity where the velocity vector stays within an acceptable cone of the CTA probe (often ±30º). When the local turbulence level increases, severe error such as "rectification" and "drop-out" arises. This often results in an overestimated mean velocity and underestimated turbulence intensity.

The effect of high turbulence intensity on hot wires have been discussed by many authors including Hinze (1959), Shabbir et al (1996) and Hussein et al (1994).
With the assumption \( V, W = 0 \), the measurement values of mean velocities \( (U_m) \) and their fluctuating components \( (u_m) \) with a single wire can be expressed by:

\[
U_m = U \left( 1 + 0.5 \left[ \frac{w^2}{U^2} - \frac{u w^2}{U^3} + \frac{u^2 w^2}{U^4} - \frac{1}{4} \left( \frac{w^4}{U^4} \right) + \ldots \right] \right)
\]  
(3.1)

\[
\overline{u_m^2} = u^2 \left[ 1 + \frac{uw^2}{u^2 U} - \frac{u^2 w^2}{u^2 U^2} + \frac{1}{4} \left( \frac{w^4}{u^2 U^2} \right) + \ldots \right]
\]  
(3.2)

In order to estimate the errors in our measurements, considering a case where \( \sqrt{\frac{u^2}{U}} = 0.4 \), \( \frac{w^2}{U^2} = 0.05 \) and \( \frac{uw^2}{U^3} = -0.01 \) (a situation in the far field of a free jet in Hussein et al (1994) measured with LDA), then the mean velocity will be overestimated 2.5% from equation 3.1 and the turbulence intensity will be underestimated by 6.25% from equation 3.2.

In our case, the low centerline velocities in the low velocity jets are thus not due to turbulence influence on the anemometry wire as this effect should increase velocity reading. Also, the velocity fluctuations are so slow that the wire will follow them.

In paper 1 it is also shown that influence of the humidity of the air is small.
4 MEASUREMENT RESULTS

4.1 Free jet

Self-similarity is the most important physical phenomena of the fully developed turbulent jet. Self-similarity or self-preservation means that the velocities and Reynolds stresses become constant with respect to the streamwise direction if they are scaled with local jet length and local centerline velocity. The mean velocity reaches self-similarity at about 10D. However, when further downstream jet velocities are very low, self-similarity does not exist any more and the jet starts to disintegrate. In the ASHRAE handbook this section is called the terminating zone or zone 4. As stated in the ASHRAE handbook 1997, jet behaviour in zone 4 is not well understood. However, jet supply air is distributed to the room air only in the fourth zone as the jet disintegrates. The characteristics of the fourth zone are, therefore, of central importance to understand ventilation and room air processes.

4.1.1 Effect of outlet velocity on the transition point between zone 3 and zone 4 in free jets (paper 3)

In this paper, the effect of nozzle exit velocities on the transition point between zone 3 and zone 4 (as defined in ASHRAE Handbook) was studied for a nozzle (D=41mm) with different outlet velocities. Figure 4.1 shows the end centerline velocities of zone 3 for our test jet and Nottage’s results (1951) as a function of outlet velocities. Obviously, these two sets of data follow almost the same curve. For the low outlet velocities ($U_o<10$ m/s), especially, the end centerline velocities of zone 3 are around 10% of the outlet velocities independent of the outlet size. On the other hand, for air jets with $U_o>10$ m/s, the end centerline velocities become relatively constant.
Figure 4.1: End centerline velocities based on the outlet velocities, evaluated from Zou (2000) and Nottage (1951).

Some tests of Malmström et al (1992) carried out in a 12 m long test room with an area of 25 m² perpendicular to the jet centerline (the room was not symmetrical) are also reevaluated, see figure 4.2. These measurement were also done by two ASME standard long radius nozzles, $D=152$ mm and $D=41$ mm. Except for one data point ($D=41$ mm and $U_o=10.9$ m/s), the other end centerline velocities follow a tendency similar to that of the data of figure 4.2: for air jets with $U_o<10$ m/s, the end centerline velocities are always proportional to $U_o$; in this case, however, the end centerline velocity is about 17% of $U_o$. For air jets with $U_o>10$ m/s, the end centerline velocities also became relatively constant. This difference could be caused by the smaller test room used in these tests. However, the tendency is similar, especially for air jets with low outlet velocities.
Figure 4.2: End centerline velocities of zone 3 based on the outlet velocities. Evaluated from Malmström et al (1992).

Figure 4.3 shows $K$-values for the present test jet as a function of outlet velocities. This data has similar tendencies and values to Malmström et al (1992) and also to other researchers, see also figure 2.3.

Figure 4.3: $K$-values based on average outlet velocities
4.1.2 Influence of turbulence intensity on integral momentum balance

In most cases, the size of the test room and the influence of walls and room air recirculation are important factors governing transition from zone 3 to zone 4 and subsequent jet termination due to momentum loss (Nottage (1951), Skåret (1973), Schneider (1985), and Hussein et al (1994)). Hussein et al (1994) made a detailed investigation of turbulent air jet resulting from a round nozzle \((U_o=55 \, m/s, D=25 \, mm, \, Re=10^5)\). Velocity momentum to the third order was obtained using stationary and flying hot-wire and burst-mode Laser-Doppler Anemometry (LDA) techniques. Based on the computation of momentum balance for a free jet, Hussein et al (1994) argued that the emergence of the fourth zone could be attributed to the effects of confinement on the jet. In this case, the jet momentum is partly transferred to the recirculation airflow created by the jet itself and the entrainment of secondary air. This return airflow "steals more momentum from the jet" with increasing distance from the outlet so that the centerline velocities decay more rapidly farther downstream of the air jet. Hussein et al (1994) also purposed the following simple model to estimate momentum loss with relation to room parameters:

\[
\frac{M_x}{M_o} = \left[1 + \frac{4}{K^2} \left(\frac{x}{D}\right)^2 \frac{A_o}{A_R}\right]^{-1}
\]

(4.1)

Zou (2000) (paper 3) also measured centerline velocity decay of a free jet in a large room. Most measurements concentrated on the air jet with outlet velocities less than 10 \(m/s\) because jets for ventilation or air conditioning frequently have low outlet velocity compared to the jets normally used in fluid mechanics experiments. However, calculations with equation 4.1 show that, for all outlet velocities, at the distance of the transfer point between the zones 3 and 4, 99% of the outlet momentum is still contained in the jet flow. Therefore, the effect of jet confinement is not the only main reason for the rapid centerline velocity decay.

Traditionally, the integral momentum equation for the axisymmetric jet only contains the first order terms:

\[
M_x = \rho \int U^2 dy
\]

(4.2)

However, a reasonable model of an axisymmetric jet in an infinite environment should consider that the velocity profiles must satisfy the momentum integral at least to the second order, see also Hussein et al (1994) and Capp et al (1990).
Chapter 4 Measurement results

\[ M_x = \rho \cdot 2\pi \int_{jet} \left[ U^2 + u^2 - \frac{(v^2 + w^2)}{2} \right] ydy \]  \hspace{1cm} (4.3)

where \( u, v \) and \( w \) are the fluctuating velocity components in axial, radial and tangential directions. Note that \( u^2, v^2 \) and \( w^2 \) can directly connect with local mean velocity \( U \) as local turbulence intensity \( \frac{\sqrt{u^2}}{U}, \frac{\sqrt{v^2}}{U} \) and \( \frac{\sqrt{w^2}}{U} \).

Equation 4.3 indicates that the momentum integral of mean velocities can only stay constant at different cross-sections of air jet when the second order contributions due to turbulence stay constant. In other words, if turbulence contribution increases in the downstream section, the whole jet momentum is still constant but the second order terms will "steal momentum from the mean velocity" and the centerline velocities will decay more rapidly.

Figure 4.4 presents the centerline turbulence intensity with distance from the jet \((U_o=5 \text{ m/s, } D=41\text{mm})\). As expected, for an air jet in the fully developed zone \((x/D<47 \text{ in figure 4.4})\), turbulence intensities at the centerline are almost the same, and when turbulence intensities start to increase, centerline velocity cannot decay linearly and the fourth zone appears.

![Turbulence intensity variation with distance from jet](image)

Figure 4.4: Turbulence intensity variation with distance from jet \((U_o=5 \text{ m/s, } D=41\text{mm})\).

One possible reason for the increase of turbulence intensity is, of course, that the centerline velocities are too weak to prevent outside disturbances. In figure 4.5,
nine different centerline mean velocity and corresponding turbulence intensity measurements at the transfer point between zones 3 and 4 \((x/D=47)\) are presented. With only one point with turbulence intensity less than 30%, measurement data shows a tendency that turbulence intensity is inversely proportional to mean velocity. The same behavior was also observed in the air jet with different outlet velocities.

![Graph](image)

Figure 4.5: Centerline velocity and turbulence intensity measurements at \(x/D=47\) \((U_o=5\text{ m/s}, D=41\text{ mm})\).

### 4.2 Three-dimensional wall jet (paper 4)

As shown in figure 2.5, a wall jet may be considered a combination of two layers, an inner turbulent boundary layer and outer free jet flow layer. The boundary layer at the ceiling is thin compared to the turbulent mixing layer towards the free room. Therefore, the flow in a three-dimensional wall jet with a supply area of \(A_o\) will be similar to a flow in a free jet with a supply area of \(2A_o\) (the outlet and its mirror image). This fact indicates that the flow characteristics in free jets could be used as a reference case for those in wall jets.

The development of a wall jet can be also divided into 4 regions in terms of the maximum velocity decay. Closest to the outlet is the potential core or zone 1, where maximum velocity equals outlet velocity. The next region is the characteristic decay region or zone 2 where maximum velocity starts to decay
from its initial value. The third zone is the fully developed jet and the fourth zone is the zone of jet termination. In the first three zones, room air is entrained into the jets and mixed with supply air. In the fourth zone supply air is distributed to room air as the jet distribution.

Zone 3 is always considered a long zone with major engineering importance and equation 2.8 is also used to determine the maximum velocity decay of zone 3 for wall jets from round nozzles. If we treat the wall jet with the method of image (see paper 4), then the $K$-value (of the wall jet) can always be roughly estimated by multiplying $\sqrt{2}$ with that of a free jet with the same exit area. However, experimental results in paper 4 have shown that, for the case of big nozzles, $K$-values of the wall jet can be estimated by multiplying those of a free jet with the same exit area with $\sqrt{2}$, while for the small-size nozzles, $K$-values become lower than expected, see figure 4.6.

![Figure 4.6: $K$-values based on outlet velocities for wall jets.](image)

In figure 4.7, the location of virtual origin $x_p/D$ is also presented as the function of outlet velocity. Although the location of $x_p/D$ is strongly dependent on the decision regarding which measurement points to fit the straight line, the initial outlet condition still shows strong influence on this parameter. When the outlet velocity is less than 6 m/s, $x_p/D$ data have a tendency to increase for small size nozzles, which is similar to free jet performance, see figure 2.4. It might be worth noting that $x_p/D$ values for $D=43mm$ seem to be different from those for $D=76mm$ and $D=152mm$, which are quite similar to the $K$-value performance, as shown in figure 4.6.
Figure 4.7: \( x_p/D \) as a function of outlet velocity.
5 COMPUTATIONAL FLUID DYNAMICS (CFD)

5.1 Introduction

Due to the complex nature of fluids such as compressibility, turbulence, unsteadiness, momentum effects, buoyancy, etc, the character of fluid flow is usually complex. The analysis of a flow process could be very difficult and some special methods are needed to deal with this. Experimental measurement and order of magnitude analytical calculations are generally used but these can be very expensive and approximate, respectively.

In Computational Fluid Dynamics (CFD), the equations of fluid dynamics and heat transfer are solved numerically by discretizing the equations and then solving them on a fine grid which covers the flow domain. By using CFD fluid flow can be simulated and visually represented on a computer screen. Due to the fact that fundamental equations are solved in every part of the geometry, CFD solutions can pick up the character of fluid flow and heat transfer that might not otherwise have been predicted. The results of a simulation contain all the relevant flow variables such as velocities, pressures, temperatures, densities, etc. The unique advantages of using CFD can be summarised as:

- substantial reduction in the cost of new designs;
- the ability to study systems where controlled experiments are difficult to carry out;
- the ability to study hazardous systems at and beyond their normal performance levels; and
- the unlimited detail of results and analysis options.

The application of CFD to solve fluid flow problems has increased dramatically over the last twenty years. This is due to considerable improvements in computer hardware performance, user-friendly interfaces and mathematical methods. The availability of affordable powerful computers nowadays has brought CFD
technology within the reach of every engineer and analyst. This has led to the application of CFD to almost every field in fluid and heat transfer engineering.

### 5.2 Basic conservation equation

The equations of airflow motion in a Cartesian and time averaged form may be written as:

**Continuity equation:**

\[
\frac{\partial u_i}{\partial x_i} = 0
\]  \hspace{1cm} (5.1)

**Momentum equation:**

\[
\rho \frac{\partial u_i}{\partial t} + \rho u_i \frac{\partial u_i}{\partial x_i} = -\frac{\partial P}{\partial x_i} + \mu \frac{\partial^2 u_i}{\partial x_i \partial x_j} + \frac{\partial}{\partial x_j} (-\rho u_i u_j') - \rho g_i \xi \Delta T
\]  \hspace{1cm} (5.2)

**Energy equation:**

\[
\rho c_p \frac{\partial T}{\partial t} + \rho c_p U_j \frac{\partial T}{\partial x_j} = \lambda \frac{\partial^2 T}{\partial x_i \partial x_j} + \frac{\partial}{\partial x_j} (-\rho c_p u' u_j') + Q
\]  \hspace{1cm} (5.3)

For the isothermal case, the last term on right-hand side of the momentum and energy equations vanishes.

The averaged equations contain the Reynolds stress terms \(u_i'u_j'\) and the turbulence heat flux \(u'T'\). Their determination requires the introduction of a turbulence model.

Engineers are normally interested in only a few quantitative properties of turbulent flow, such as mean maximum centerline in an air jet, and do not need to know the effect of each and every eddy in the flow. This is why the Reynolds-average method is so extensively used in different fluid dynamical problems.

The most common Reynolds Average Navier-Stokes (RANS) turbulence model used in ventilation is the so-called “standard \(k-\varepsilon\) model”. In this method, the Boussinesq approach (eddy-viscosity or isotropic viscosity hypothesis) is used to simplify the Reynolds stress term. The Navier-Stokes equations are then solved with two further equations—one for the kinetic energy of turbulence (\(k\)) and the other for its dissipation rate (\(\varepsilon\)). Although the results are acceptable in some flow fields, this model is handicapped in predicting non-fully turbulent flow, for
instance room air motion in the region very close to the wall. In order to predict the near-wall turbulence in ventilated rooms, the “Low Reynolds Number (LRN)” turbulence model is introduced in which wall function is applied to ensure that viscous stresses are considered at low Reynolds numbers and in the viscous sub-layer adjacent to solid walls, see Patel et al (1985).

5.3 Application of CFD on jet simulation

5.3.1 Historical background

The simulation of a turbulent air jet is not an easy task. The performance of numerical simulations does not depend only on the turbulence model, since a simulation also requires a computer code which allows grid independent solutions. Craft (1991) used a model of the press-scalar gradient correlation to predict some properties of free and impinging jets. Craft (1992) replaced the conventional production term in the standard $k$-$\varepsilon$ model by a mean strain-dependent source term and showed better agreement with experimental data. Based on an assumed asymptotic behavior of the subgrid scale stress, Olsson (1995) used the Large Eddy Model to simulate a spatially developing circular jet; the numerical result was also compared with experimental data. No jet simulation, however, has been carried out in the region far away from the outlet.

In HVAC applications, most simulations of jets are concentrated on wall jet performance in room ventilation. Nielsen (1973) was the first to introduce CFD to predict airflow in rooms. Nielsen et al (1978) applied the $k$-$\varepsilon$ model to simulate the two-dimensional velocity distribution in a ventilated room without heating sources. The results demonstrated that the precision of calculation was adequate for design purposes. By applying the wall function into the CFD model, Nielsen (1989) also correctly predicted that the maximum velocity of the return flow in the occupied zone was correlated to supply jet momentum. Davidson et al (1998) used Large Eddy Simulation to study the effects of low-Reynolds number in a slot-inlet ventilated room. With the help of a super computer, Murakami et al tested the possibility of using different turbulence models in HVAC applications, especially the airflow pattern in large buildings, see Murakami et al (1983), (1985), (1992) and (1994). Chen also contributed many valuable ideas to this area e.g. how mesh should be generated in supply diffuser simulations. A review of CFD programs used in room airflow simulation can be found in Jones et al (1992).
Unfortunately, all these studies show that no existing turbulence model can accurately predict all the properties of jets. The standard $k-\varepsilon$ model, the most popular in ventilation applications, is proved to be particularly unsuitable for the case of non-fully turbulent flow. With increasing computer capacities sophisticated 3D Large Eddy Simulation (LES) may in the future be extensively used in predicting room air movement.

5.3.2 A CFD study for airflow distributions at floor level in a SOSIV room (Paper 6)

Slot diffusers are widely used in air distribution systems where supply jets form a plane or three-dimensional recirculating flow in a room. Since the maximum velocity of this recirculating flow always happens near floor level, it is essential for the designing of ventilated rooms to fully understand airflow characteristics along floor level.

In this paper, airflow distribution characteristics at floor level in a slot-outlet and slot-inlet ventilated (SOSIV) room was compared between CFD simulation and available measurement results, see Wang et al (1996). With two different outlet/inlet locations, and 6 levels of diffuser height and supply velocities, a total of 48 cases were simulated under isothermal and non-isothermal conditions.

In this study, all simulations were conducted using the $k-\varepsilon$ model developed by Launder et al (1974). Since previous experiments showed that the airflow distributions in an SOSIV room are two-dimensional (height and length), computation was conducted only in two dimensions without considering the effect of width.

For both the vertical velocity profile and horizontal velocity distribution at floor level, simulation results were shown to be in good agreement with available experimental data. However, calculated maximum floor-level velocities were 5% and 10% lower than the measurement data for isothermal cases and non-isothermal cases, respectively. This could be a result of the fact that maximum velocities often happened near the floor and the standard $k-\varepsilon$ model is poor in predicting airflow in this area.

5.3.3 One jet simulation with standard $k-\varepsilon$ model

To test the capabilities and restrictions of the standard $k-\varepsilon$ model in predicting jet behaviour, a jet with diameter 41mm was simulated with the FLO++ code. To
simplify mesh generation this jet was placed freely in an imaginary cylindrical container with diameter 10m and length 30m. Due to the axisymmetry of the whole system, only one quarter was simulated. In this study, 5 different outlet velocities ($U_o = 3, 6, 8, 10$ and $20 \text{ m/s}$) were simulated.

The grid density was $100\times20\times5$ grids along the length ($x$), radial ($r$), and angle ($\theta$), respectively. The grids had smaller spacing at locations where more flow detail was needed: the supply, the exhaust, the jet flow area and the floor level area.

The following boundary conditions were used: (1) solid surface, (2) plane of symmetry, and (3) velocity at an opening. The boundary conditions remained constant throughout each simulation. At the solid surfaces the velocity was assumed to be zero (no slip condition). There is no flow through a symmetry plane, but there can be flow at a symmetry boundary. The finite-difference form of the time-averaged transport equations was obtained by adopting a semi-integral approach to discretize the equations over each control volume of the computational grid using a hybrid-difference scheme. The line-by-line method was used to obtain converged solutions iteratively, whereas relaxation factors were employed to promote stability of the process. The relaxation factors were 0.2, 0.7, and 0.7 for pressure, turbulence kinetic energy and velocity components, respectively. A run was considered converged if the residuals of pressure, velocity components and temperature were less than $10^{-4}$. The velocity field needed about 25,000 iterations to satisfy the convergence criterion, whereas the temperature field required more than 35,000 iterations. The typical real-world computation time for 35,000 iterations on a PC with Pentium-pro 200MZ CPU, 64MB memory, was about 40 hours.

For all cases, in the absence of turbulence data, turbulence kinetic energy $k_{out}$ and its dissipation rate $\varepsilon$ at the diffuser boundary condition were assigned according to Seyedein et al (1994)

$$k_{out} = I U_o^2 \quad \text{(5.4)}$$

$$\varepsilon = \frac{0.09 k_{out}^{3/2}}{0.05 d} \quad \text{(5.5)}$$

Figure 5.1 shows the dimensionless centerline velocity ($U_o / U_c$) simulation results for 5 outlet velocities as a function of dimensionless distance ($x/d$) from the outlet. Centerline velocity decay occurs quite similarly for all simulation cases, $K \approx 5.9$ and $x_p \approx 4$. Obviously, this is not in agreement with existing experiments, see chapters 2 and 4.
Figure 5.1: Centerline velocity simulation results of air jets (d=41mm) with different outlet speeds. For all cases, $K \approx 5.9$ and $x_p \approx 4$.

For air jets with high outlet velocity ($U_o > 9$ m/s), however, the simulation results seem reasonable, at least in zone 3, both in terms of centerline velocity decay and jet velocity profile, see figures 5.2 and 5.3. This could imply that the jet flow is fully turbulent only when $U_o > 9$ m/s. Considering that the jets used in ventilation application always have low outlet velocities, the $k$-$\varepsilon$ model is not recommended for free air jet simulation.

Figure 5.2: Centerline velocity decay comparison between simulation and measurement results (Zou 2000, D=41 mm).
Figure 5.3: Jet velocity profile \( (x/D=20) \) comparison between simulation and LDV-measurement (Capp et al 1990).
6 SUMMARY

The main objective of this study was to explore the velocity decay of jets in the far field from the outlet. Studies of jet behaviour were therefore carried out and the main results obtained were as follows:

- "Free jets" were tested with three different outlet sizes in a large space. The measurement results show that the end centerline velocities of zone 3 strongly depend on the outlet velocities and room size.
- Wall jets were also tested in a standard test room. The results showed that the $K$-value of wall jets is a function of both outlet velocities and outlet size.
- A simple method for HWA calibration was tested and found satisfactory. The influence of natural convection and humidity was also investigated.
- The influence of sampling time for measurement was investigated. The measurement results implied that, for an air jet with known outlet conditions, the distance from the outlet to the measurement point is an important factor for the accuracy of low mean centerline velocity measurements.
- CFD calculations were also carried out in both free jets and wall jets. The simulation results show that it is not always suitable to use CFD to predict jet behaviour, especially for jets with low outlet velocity and in the area far away from outlet. However, for two-dimensional wall jets, CFD could be a powerful tool for designers.
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