WIND TUNNEL INVESTIGATION OF JET FAN AERODYNAMICS

by

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ABSTRACT

This study has investigated the performance aerodynamics of jet fans in order to identify and understand the fundamental parameters in their use in mine and tunnel ventilation. Despite their advantages over other ventilation methods, jet fans have not been often used in mining due to an inability to predict performance accurately. They have been used in longitudinal ventilation of vehicle tunnels and other installations with encouraging results despite the fact that such systems have been designed with limited data.

The current studies used a wind tunnel to study jet fan ventilation. The fan was simulated by aluminum pipes of different diameters connected to a centrifugal blower. The aluminum pipes were inserted at the entrance section of the wind tunnel and jet outlet velocities ranging between 20 and 40 m/s were used to produce the flow field. In order to study passage wall effects on the flow the jet fan was traversed from a near-wall position to the wind tunnel axis in equal successive steps. The axial pressure development for all positions were determined together with a detailed velocity distribution and the overall entrainment characteristics. Both the magnitudes of the pressure drop and rise depended on the jet fan position from the walls. Near-wall jet fan positions tended to have initially larger pressure drops and lower pressure rises than positions farther from the wall which had lower pressure drops and higher pressure rises. The consequence of this pressure variation was that generally at near wall positions the jet fan entrained more air into the tunnel than at positions farther from the wall. The smaller diameter jet fan produced higher friction losses (as much as 15 % at the wall position) than the larger diameter fan with lower outlet velocity which had about 8 %. The flow field was found to develop rapidly with axial distance. The jet axis velocity developed faster than that of a free jet of the same initial velocity and revealed that jet fans can move air over distances greater than 70 jet fan discharge diameters and still maintain a minimum air velocity of at least 0.5 m/s. For fan
positioning at $F_p < 0.4$, a region of backflow was identified. The backflow fraction was 0.72 and 0.55 for the smaller and larger diameter fan respectively. The performance parameter $\xi_{jf}$ of the jet fan determined from pressure and flow (entrainment) ratio considerations $Q_T / Q_j (P_m - P_r) / (P_j - P_m)$ was found to decrease as the jet fan was moved away from the tunnel wall despite higher friction losses at near-wall positions. The jet fan performance parameter is generally below 12 % as verified by mathematical derivations. The larger diameter (lower velocity, $U_j = 21.4$ m/s) jet fan had $\xi_{jf}$ performance values almost twice that of the smaller diameter jet fan ($U_j = 40$ m/s). The $\xi_{jf}$ value ranged between 4.5 to 6 % for the larger diameter fan. High entrainment ratios achieved at near wall positions generally improve jet fan performance.

Theoretical equations based on momentum and energy considerations were formulated. These derivations identified a range of flow ratios ($n = 0.1$ to 0.9) which can be used to design an effective jet fan ventilation system. For each flow ratio ($n$) there is an optimum area ratio ($\alpha$) for maximum induction of secondary flow.

The present studies have established a procedure for jet fan performance analysis using wind tunnel investigations and have provided useful information for jet fan ventilation design.
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<tr>
<td>( A_j ), ( A_t )</td>
<td>Cross sectional area of jet fan and tunnel respectively</td>
</tr>
<tr>
<td>( A_p ), ( A_e )</td>
<td>Area through which secondary flow enters the tunnel (= ( A_t - A_j ))</td>
</tr>
<tr>
<td>( A_{ob} )</td>
<td>Area occupied by obstructions in the tunnel</td>
</tr>
<tr>
<td>( D_j ), ( D_t )</td>
<td>Diameter of jet fan and tunnel respectively</td>
</tr>
<tr>
<td>( D_{tr} )</td>
<td>Diameter ratio (= ( D_j/D_t ))</td>
</tr>
<tr>
<td>( E_{in} )</td>
<td>Energy input from jet fan</td>
</tr>
<tr>
<td>( E_{out} )</td>
<td>Energy in the tunnel outlet flow</td>
</tr>
<tr>
<td>( E_{ml} )</td>
<td>Tunnel mixing loss energy</td>
</tr>
<tr>
<td>( E_{fl} )</td>
<td>Energy due to friction loss</td>
</tr>
<tr>
<td>( E_{jl} )</td>
<td>Jet loss energy at discharge or in the nozzle</td>
</tr>
<tr>
<td>( E_r )</td>
<td>Recirculation or reverse flow energy loss</td>
</tr>
<tr>
<td>( F_p )</td>
<td>Dimensionless jet fan position (= ( Y/D_t ))</td>
</tr>
<tr>
<td>( Y )</td>
<td>Distance across the tunnel</td>
</tr>
<tr>
<td>( F_p )</td>
<td>Dimensionless fan position (= ( Y/D_t ))</td>
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<td>( L_{tr} ), ( L_R )</td>
<td>Backflow length in the tunnel</td>
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<td>( m_e )</td>
<td>Entrained mass flow</td>
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<td>( m_j )</td>
<td>Jet fan discharge mass flow</td>
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<td>( m_t )</td>
<td>Total tunnel mass flow (= ( m_j + m_e ))</td>
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<td>Flow ratio ( (Q_e/Q_j = m_e/m_j) )</td>
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<td>Entrance pressure for entrained flow</td>
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<td>Jet fan total discharge flow</td>
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<td>( P_{tr}, P_m )</td>
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<td>( r_0 )</td>
<td>Radius of jet nozzle</td>
</tr>
<tr>
<td>( S )</td>
<td>Swirl number of jet</td>
</tr>
<tr>
<td>( T )</td>
<td>Angular momentum of jet</td>
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<td>( W )</td>
<td>Axial momentum of jet</td>
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<tr>
<td>( Q_e, Q_s )</td>
<td>Entrained or secondary volume flow</td>
</tr>
<tr>
<td>( Q_j )</td>
<td>Jet fan discharge volume flow</td>
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<tr>
<td>( Q_T )</td>
<td>Total tunnel volume flow</td>
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<tr>
<td>( Q_R )</td>
<td>Backflow volume flow</td>
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<tr>
<td>( U )</td>
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<td>( U_e )</td>
<td>Entrainment velocity</td>
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<td>( U_j )</td>
<td>Jet fan discharge average velocity</td>
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<td>( U_t )</td>
<td>Average tunnel velocity</td>
</tr>
<tr>
<td>( u' )</td>
<td>Velocity fluctuation</td>
</tr>
<tr>
<td>( \sqrt{u'^2}/U_c )</td>
<td>Rms or turbulence level of the fluctuating velocity at the tunnel axis</td>
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<tr>
<td>( W_R )</td>
<td>Width of backflow at any axial distance (metres)</td>
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<td>( W_{tr}/D_t )</td>
<td>Dimensionless backflow width</td>
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<td>( X )</td>
<td>Axial distance from jet fan nozzle (metres)</td>
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<td>( x_o )</td>
<td>Distance from jet where secondary and jet flow start to mix</td>
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\( x_1 \) Distance at which jet flow and secondary flow are fully mixed

\( X / D_j \) Dimensionless distance from jet fan nozzle

\( \eta_i \) Induction efficiency defined by equation 3

\( \Phi \) Tunnel to jet velocity ratio \( U_t / U_j \) ratio

\( \theta \) Thring and Newby similarity parameter \( \left( Q_j + Q_s \right) \left( D_j / 2 \right) / \left( Q_j D_t \right) \)

\( C_t = \theta \) Craya -Curtet parameter for ducted confined jets

\( \alpha \) Area ratio of jet fan to secondary stream inlet area \( = A_j / A_s \)

\( \Omega \) Area ratio of jet fan to tunnel \( = A_j / A_t \)

\( \rho \) Air density (kg/m³)

\( \zeta_n, \zeta_s, \zeta_t \) Loss coefficients of nozzle, secondary flow and tunnel friction coefficient.

\( \zeta_r \) Jet recirculation or backflow fraction of total tunnel flow

\( \xi_j \) Jet fan performance parameter described in Chapter 7
ACKNOWLEDGMENTS

I am most thankful to my supervisor Dr. Allan E. Hall for the conception of the idea of jet fans in mine ventilation at a time when very little was known about the subject. Through his knowledge and wisdom he was able to direct me; leading to the successful completion of this thesis. I also benefited greatly from the many informal discussions during the course of this thesis.

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This thesis is dedicated to my wife Hope Nwaobilo, and my three children Kudakwashe, Imogen and Munashe. They showed great strength and understanding during the years they stayed in England without me. I also wish to mention my mother Stella, father Nevison and my late stepmother Eleanor and grandmother Mbuya Adeke "Deke iro!"
CHAPTER ONE

1. INTRODUCTION

Jet fans are free-standing, unducted axial flow fans used in mine and vehicular tunnel longitudinal ventilation, and are often fitted with a nozzle as shown in Figure 1.1. Their application in mining operations includes pressure boosters and ventilation of development ends, underground workshops, battery charging bays, pump and machine chambers. In industrial applications, they are used for road tunnel ventilation, cooling of furnaces and kilns, degassing of tanks and ship hulls; and ventilation of service tunnels during repairs. Although jet fans have been used successfully in longitudinal ventilation of road tunnels their use in mining is not common because their operation is not well understood. In order to appreciate the reasons for this, it is necessary to review the basics of typical mine ventilation systems.

1.1 Importance of Ventilation

Fresh air is required in subsurface environments such as mining and tunneling because (i) sufficient air is needed for personnel to breathe. (ii) Noxious fumes or gases produced during mining must be diluted to a safe level of concentration, to prevent adverse effects on people exposed to the conditions (iii) The greatest amount of comfort possible should be provided at a reasonable and economic price and (iv) the ventilation provided should be cost effective.

Failure to provide adequate ventilation which meets the legislated standards can result in the closure or suspension of the production operation. The resulting adverse health effects which may appear long term incur significant compensation claims, and thus
environmental control measures are an essential and integral part of the mine production cycle.

Ventilation in underground mining and tunneling is achieved using fans which induce airflow in mine openings. Fans offer the most convenient means of supplying ventilation although compressed air injectors are used in limited circumstances by some mines. Natural ventilation effects may assist or work against the fan system. These effects can be significant in deep mines.

1.2 Mine Ventilation Fans

1.2.1 Main fans

Main fans provide the whole system with air and can either be centrifugal or the axial flow type. Large fan ratings are required particularly in large and deep workings. Each person working underground requires approximately 0.1 m³/s but this is seldom a limiting value in N. America. The extensive use of diesel equipment underground requires the provision of 0.06-0.09 m³/s of ventilation air per kilowatt of engine power. Ventilation circuit head losses are often considerable in mining. Main mine fan installations can be required to overcome pressures beyond 6000 Pascals delivering up to 1000 m³/s into the very large and deep mines. Power consumption for this duty would be 8.57 MW for a fan mechanical efficiency of 70 % and cost over $300 000 per month at a cost of 5 cents/kWh.

Centrifugal fans can have radial, forward or backward curved impeller blades. Figure 1.2 shows a centrifugal fan. In centrifugal fans, air is drawn into a rotating impeller and discharged radially into an expanding scroll casing. The tangential velocity of the air
entering and leaving the impeller increases the centrifugal (static) energy and is proportional to the work done.

In axial flow fans, pressure is produced by imparting a tangential acceleration as it passes through the impeller of the fan. As air leaves the impeller, the energy of rotation is converted into a linear flow of energy. It is usual to locate guide vanes in the diffuser casing following the impeller and these are most effective in converting the rotative energy. The blades of the impeller can either be fixed or have a variable pitch. Most large axial-flow fans are of the vane axial type. Figure 1.3 shows the construction of an axial-flow fan.

The application of main fans to mine circuits is subject to significant uncertainty. According to the advance of the workings, new shafts, roadways, and stopes are added to the circuits so that the air volume as well as the necessary fan pressure is altering constantly. Final dimensions of airways often differ from designed figures because of scaling and overbreak in blasting. Airways are irregularly obstructed by mobile equipment and rock falls. The design of the fan system should allow for these developments and accommodate them adequately. Another complication is that in the mine between the shafts and working faces, leakage may occur through old workings or through strata. The leakage is significant and on average about 45% of short-circuiting can occur. These factors complicate the design specification for the fan.

Main ventilation systems usually consist of independent circuits of differing resistance. High resistance mining circuits may require the provision of a booster fan to overcome the head loss. Provision of a booster fan is preferable to increasing the pressure of the main fan and regulating the other circuits because it requires less energy to be supplied. A typical mining booster fan is shown in Figure 1.4. Because of the large pressure difference
across the fan, partitions are used to prevent uncontrolled recirculation and to ensure maximum development. Wallis (1983), gave some useful features of large fan installations. An example of the magnitude of mine ventilation flow losses is shown in Figure 1.5 (Linsell, 1953). In Figure 1.6, air is introduced through the mine shaft A and is exhausted to the atmosphere through shaft B. Airways ventilated by the main and booster fans are designated as being in a condition called through-flow. Closed end portions and new advancing workings of the mine circuit cannot be ventilated by through ventilation and require the provision of auxiliary ventilation systems.

1.2.2 Auxiliary Fans

An auxiliary fan can either be used attached to suitable ducting commonly known as ventilation tubing or as a "stand alone" fan i.e. ductless. In the latter case when used in this mode it is called a jet fan. Most auxiliary fans including jet fans are designed as either single stage or multistage axial flow fans. The auxiliary fans used in mining have diameters ranging between 300 and 1400 mm. Axial flow type auxiliary fans vary in size and duty. They can range from a rating of 4.5 kW, 1 m³/s units to 11 m³/s or more. These are used in the ventilation of dead-end workings, where mining is taking place and is most needed and by doing so they move the air to where it is needed. Auxiliary fans are used in almost all underground mines for both development work and exploitation.

1.2.2.1 Ducted auxiliary fans

In metal mines; drifts, raises, shafts, winzes and stopes with one entrance require auxiliary ventilation. In coal mines auxiliary ventilation is required in all entries beyond the last (connecting) crosscut and the auxiliary fans fitted with tubing ventilate the dead-end workings which are often partitioned using line brattice. The auxiliary ventilation system
must be able to purge the area of harmful gases and dust and maintain adequate ventilation standards required by legislation. Figures 1.7 and 1.8 illustrate auxiliary ventilation systems commonly used in mines. Hartman et al. (1982) covers the subject of mine ventilation adequately and explains the various ventilation schemes practiced in mining.

Tubing and line curtain used in auxiliary ventilation do not have as high a capital cost as the fan. The problem of these attachments is that solid ducting is difficult to transport and store in mines. Flexible ducting is easier to handle but is readily torn or damaged. This results in excessive leakage and significant sections of tubing are frequently replaced because of damage. This increases the cost of the system and labour required for maintenance. In addition the face conditions are unsatisfactory because of the leakage and the requirement to shut down the system to replace the defective ducting.

Ducted systems and line curtains provide an obstruction in the airway cross-section and the airway is frequently driven over-size to its final required dimension to accommodate a ducted ventilation system during construction. Inadequate installation and clearances from moving equipment can cause damage to the tubing during mining operations. It is not uncommon for operators to tie off parts of the ducting to clear mobile equipment. This may reduce the available flow area by up to a third and throttle the airflow resulting in shock and friction losses. It is therefore attractive to mines to consider the use of ventilation systems without ducting.

1.2.2.3 Ductless Fans (Jet fans)

A jet fan operates by discharging an incompressible turbulent air jet in the area to be ventilated. The jet exchanges its momentum with the surrounding secondary air in exactly the same manner as a jet pump; this causes air pressure to fall below ambient and
consequently surrounding air is entrained by the jet and a continuous airflow is established. In underground mine auxiliary ventilation, jet fans are used to boost air pressure and to direct airflow where mining operations are in progress. The presence of a duct as shown in Figures 1.7 and 1.8 may interfere with mining operations and there are usually restrictions on its size. High velocities in long ducts of small area result in large energy costs. Because of leakages due to damage, ducts have to be replaced frequently at an additional cost.

Fig. 1.9 shows a jet fan situated at the upstream corner of the last connecting airway (or crosscut) and discharging air in the form of a turbulent jet along the wall into the opening. The jet expands with increasing distance from the fan until, ideally, air is flowing in half the width of the opening and it exhausts in the other half back to the main airway. The velocity of the flow reduces downstream because of the increasing mass of air being entrained and by this process the jet fan delivers a volume much greater than its own inlet volume. Some of the entrained air is recirculated at the end wall of the opening. Recirculation can also occur at the inlet of the opening and can be as much as 20 to 40%. Recirculation is not a major problem in longitudinal ventilation of vehicle tunnels because tunnels are open ended. The presence of sharp curves may cause pockets of local recirculation to be established. In tunnel ventilation it is necessary to locate jet fans in a position that allows enough airflow to go through. Fans are inserted in a short circular duct which is streamlined and sound insulated to reduce noise levels. They are placed at regular distances along the tunnel. In some mining operations jet fans are placed in an airway in order to boost the pressure of the flow as shown in Figure 1.10. In this case a desired pressure rise is achieved.
Jet fan ventilation is very effective because it uses the opening itself as a duct and on average air velocities in the opening, typically are higher between 1 and 2 m/s. This type of ventilation can be operated remotely with the aid of computer control.

Jet fan (or induction) ventilation is simple in principle but the mechanisms by which the jet interacts with the induced flow in confined places is still not well understood. Investigation of the subject is still incomplete and there are no established guidelines for the use of jet fan ventilation systems. Jet fan ventilation design requires a rigorous knowledge of the principles of fluid flow and momentum exchange between the primary air from the fan and that of the secondary stream. Only a well developed theory supported by experimental investigations gives insight to designing induction systems. Any empirical rules may lead either to an oversized system or to a faulty ventilation system.

The studies described in this thesis recognize that the key to the design of jet fan ventilation systems rests in the characterization of the complex nature of the aerodynamics of the discharged turbulent jet from the fan. Objectives of this study have been formulated in order to provide knowledge which have either been unavailable previously or provide full attention to the subject of jet fan performance in mine ventilation particularly overall flow and pressure characteristics of jet fans. The data obtained will therefore contribute considerable information on the subject particularly in mine ventilation where jet fan application has very been limited. In vehicle tunnel ventilation previous work has not provided an adequate data base for effective future jet fan ventilation designs. Future ventilation systems should be inexpensive, effective and easy to commission.
1.3 Research Objectives

The objectives of the present study have been formulated after a comprehensive review of the current literature on the subject in both road tunnel and mine ventilation. Consideration of mine situations shows that there are large numbers of configurations of airways which can be ventilated by jet fans. Ventilation problems may be dilution of strata gases which require a certain air velocity to be provided at the point of emission or dust or fume mitigation which requires a larger quantity of air to be provided at a particular working face area. It is clear that there are an infinite number of individual problems which may occur in mines. Previous research had examined some specific problems and the results had limited application to general principles of jet fan performance and positioning. There are two major ways in which a jet fan can be used in mine ventilation. (i) It can be used to ventilate a dead end working as shown in Figure 1.9 (i.e. closed end case). or (ii) It can be used in an open ended situation (Figure 1.10). When used in the open ended situation it can be used to increase the pressure and also achieve the desired flow volume in the airway. Most previous studies have examined jet fan performance in existing installations where it was difficult to obtain general data which could be used in other situations. In this study, a wind tunnel and jet fan test facility for modelling jet fans in underground mine environment and tunnel conditions is set up. The test conducted is for an open-ended situation but it is considered that the results obtained can also be used in the closed-end mine development situation. The objectives of this research can be stated as follows.

(1) To set up a wind tunnel- jet fan facility
(2) To establish pressure and flow conditions of jet fans applied to geometry appropriate to mine openings with respect to wall interactions.
To formulate a mathematical representation of the jet fan performance based on momentum and energy principles.

To show how the observed results can be applied to a mine ventilation problem.

1.4 Rationale

The wind tunnel to be used in this study had to be designed, constructed and tested because no other suitable apparatus for this type of study existed at University of British Columbia. No one has ever used a wind tunnel for jet fan studies in mine ventilation. Setting up an experimental test facility for jet fans and using a wind tunnel to create similar conditions as in mine and tunnel ventilation is the most effective way to perform the investigations because several geometric and dynamic factors can be varied with ease. A wind tunnel enables flow and pressure parameters to be measured more accurately and creates a convenient atmosphere for the study to be carried out while maintaining good simulation of a real situation. Computer simulations need experimental data for validation. Reliable data is not available at present.

Parameters such as positioning of the jet fan in cross section are known to affect the aerodynamics of the jet fan but there is no data to quantify the effects. Walls affect the behaviour of many flows of engineering importance in any application and ventilation is not an exception. It is important to study the influence of the tunnel walls on the flow and pressure field.

The ratio of the jet fan diameter to tunnel diameter is important because it specifies geometric ratios for an optimum design. It is thus important to study how this parameter
influences performance of the ventilation system. A knowledge of the effects of the
insertion of the jet fan inside the tunnel is important in evaluating performance.

The parameters defining the flow or velocity ratios of the jet fan to that prevailing in the
tunnel are important for assessment of the system efficiency. Entrainment data can be
determined from the flow field and its effects on the axial static pressure variation or vice
versa can also be assessed. From these measurements, it is then possible for the overall
losses of the jet fan ventilation system to be determined from momentum or energy
considerations.

One question which needs to be addressed is to what extent the axial static pressure
gradient influences the aerodynamics of the flow. Since jet fans discharge turbulent air jets
in the ventilated area; it is interesting to determine how free and confined turbulent jet
theory can be applied to explain observed results of this study. Development of a
theoretical framework can generate tools for use in solving mine ventilation problems.
These tools can be developed in the future by performing additional testwork with the
apparatus constructed in order to verify the applicability of mathematical equations
derived in this work under all conditions which might be met in mine ventilation.

It is important to apply the results obtained to a real mining case of jet fan ventilation in
order to show how the results can be beneficial. The results and analysis of this study can
provide design data for jet fan ventilation in underground mines and tunnels.

In this chapter an introduction to the present research has been given in which the
objectives are clearly stated. The next chapter deals with previous work on the subject and
any information pertinent to the current research.
1.6 Remaining Chapters

In chapter two a general description of turbulent jets is presented together with some of the most outstanding previous work related to jet fan ventilation in vehicular tunnels and mines. Chapter three gives a full description of the experimental program that was followed in this study. In chapter four the data reduction and analysis is presented in order to make the results of this study easier to follow. In chapters five and six the discussion of the jet fan pressure and velocity field developments respectively are presented. Chapter seven discusses the jet fan jet flow entrainment process. Chapter eight gives a theoretical treatment of the jet fan flow based on momentum and energy considerations. Chapter nine gives two examples of jet fan application in an open end and closed passage. In Chapter ten a declaration of the achievements of this work is stated. The conclusion and recommendations of this work are given in Chapter 11 and 12 respectively.
TWO STAGE JET FAN

Figure 1.1 Schematic layout of a jet fan
Fig. 1.2 Construction of a centrifugal fan also showing blade types
Fig. 1.3 Construction of an axial flow fan
Fig. 1.4 A two booster fan system in an airway
Input power = 475 kW = 100%

Air power
= 252 kW
= 53%

Loss in mine fan
223 kW = 47%
(less for modern fan)

Ventilation short circuits
47.5 kW = 10%

Sundry flows
66.5 kW = 14%

Throttling in sundry flows
16.6 kW = 3.5%

Throttling in ventilation section
35.2 kW = 7.4%

Net air power
86.2 kW = 18.1%

Figure 1.5 Typical representation of flow losses (Linsell, 1953)
Fig. 1.6 A schematic diagram of a mine ventilation system.
Figure 1.7 Ducted auxiliary ventilation fan operating in the exhaust mode
Figure 1.8 Ducted auxiliary ventilation fan operating in the forcing mode
Figure 1.9 Jet fan ventilation in a mine heading
Fig. 1.10 Illustration of a tunnel ventilated by a jet fan
CHAPTER TWO

LITERATURE SURVEY

This chapter discusses the behaviour of turbulent jets as they apply to jet fan ventilation. The major part of the chapter presents a review of the previous work known to have been carried out on the subject in both mine and tunnel ventilation.

2.1 Theoretical Considerations of Incompressible Turbulent Jets

The behaviour of the mixing of a jet in a confined flow is of great engineering importance, e.g. in ejector design, combustors and design of any device involving the transfer of momentum and energy.

Since their flow is confined, jet fans can be characterized approximately by confined turbulent jet phenomenon. Due to the complex rough geometry of mine airways and tunnels, where jet fans are used it is necessary to carry out more studies which can provide adequate information for ventilation design purposes. Jet fans fall in the category of induction systems because they move air by entrainment and momentum transfer. Approximate theoretical equations have been developed for inducted systems involving jet fans. The theoretical treatment of the subject and results of previous work are discussed in this section.

The subject of jets is well covered by Abramovich (1963), Rajaratnam (1976) and Blevins (1984), among others. A simplified summary of the most salient features on jets is given for brevity. Figure 2.1 shows a submerged free jet. Mass, momentum and energy enter the
control volume by means of a nozzle which transports fluid of velocity $U_0$ through area $A_0$, and concentration of species $C_0$ above or below the ambient level of the reservoir. The concentration of pollutants or temperature of fluid is represented by $C_0$. The equations of conservation of mass, momentum and species can be applied to the non deformable control surface of Figure 2.1, and are represented by equations 2.1, 2.2 and 2.3 respectively. The entrainment velocity is represented by $U_e$, and the flow passes through the lateral sides of the control surface into the jet with this velocity. Equations 2.1 and 2.2 give conservation of mass and momentum respectively;

$$
\rho(U_0A_0 + U_eA_s) = \rho\int UdA \quad (2.1)
$$

$$
\rho U_e^2 A_s = \rho\int U^2 dA \quad (2.2)
$$

dA is an element of the area of the right hand side of the control surface through which the jet exits, $A_s$ is the area of one lateral side of the control surface and $\rho$ is fluid density. A comparison of equations 2.1 and 2.2 dictates the existence of entrainment to satisfy the balance of momentum and mass conservation. The fact that momentum has to be conserved in free jets is very important in their analysis. Equation 2.2 states that the flux of axial momentum $M_0$ (i.e. momentum passing through a plane per unit time) of a jet is conserved even as the jet disperses. Once past the nozzle jets develop free of externally applied constraints and at some downstream point many jets become "self preserving" i.e. the flows at various axial stations are dynamically similar when non dimensionalised by local length and time scales. To obtain the properties of the fully developed jet, these scales and their evolution with the flow must be determined.
Equation 2.3 describes the behaviour of self preserving jets where \( r \) is the distance from the origin of the jet and \( x \) is the axial distance. \( U_m \) is the centreline axial velocity of the jet where \( r=0 \), and is maximum i.e. \( f(0) = 1.0 \).

\[
\frac{U}{U_m} = f\left(\frac{r}{x}\right) \tag{2.3}
\]

Equation 2.3 can be rewritten in a form that incorporates equation 2.2:

\[
\int_A U^2 dA = U_m^2 \int_A \left(\frac{U}{U_m}\right)^2 dA = U_o^2 A_o = \text{Const.} \tag{2.4}
\]

\( U_o^2 A_o \) is the initial momentum flux of the jet. The element of the control surface is an annular ring for an axisymmetric (round) jet and \( dA = 2\pi r dr \). Equation 2.3 becomes:

\[
U_m^2 x^2 \int_0^x 2\pi \left(\frac{U}{U_m}\right)^2 \frac{r}{x} \frac{d}{x} = U_o^2 A_o = C \tag{2.5}
\]

Therefore \( U_m^2 x^2 = \text{const.} \)

The centreline velocity of an axisymmetric jet must be inversely proportional to the axial distance \( X \):

\[
\frac{U_m}{U_0} = \text{Const.} \frac{A_o^3}{X} \tag{2.6}
\]
Using the above expression for centreline velocity the volume flow of an axisymmetric jet is

\[ Q = \int_A UdA = \int_0^\infty U (2\pi r) dr = U_m X^2 2\pi \int_0^\infty \frac{U}{U_m} \frac{r}{x} d\left(\frac{r}{x}\right) \tag{2.7} \]

From the assumption that \( U / U_m = f(r/x) \) which is based on equation 2.4 the integral on the right hand side of equation 2.7 is a constant independent of \( X \) or \( r \). \( Q \) must be proportional to the quantity \( U_m X^2 \). Substituting equation 2.6 for \( U_m \), the volume flow in an axisymmetric jet increases linearly with axial distance:

\[ \frac{Q}{Q_0} = \frac{\text{Const.} \cdot X}{A_0^{1/2}} \tag{2.8} \]

A similar argument can be used to demonstrate that the rate of decrease of the species concentration \( C \) along the centreline of an axisymmetric jet is:

\[ \frac{\Delta C}{\Delta C_m} = \text{Const.} \cdot \frac{A_0^2}{X} \tag{2.9} \]

The equations generally hold well for both laminar and turbulent flow for \( X \) greater than about 5-8 nozzle diameters.

Submerged turbulent jets are jets of a turbulent fluid flow into a reservoir of similar fluid. A submerged turbulent jet is shown in Figure 2.2 as a sum of three regions: an initial region, a transition region, and the fully developed jet. The initial region has a length \( x \),
and it consists of the core flow and the surrounding shear layer. The velocity in the core flow is equal to the nozzle exit velocity \( U_0 \) for a uniform exit velocity. The core flow is free of shear and the term potential core is often used. The core flow is surrounded by a turbulent shear layer, which forms the boundary between the core flow and the reservoir fluid. The flows in the core, the initial region and the transition region bear the imprint of the nozzle details. At some point in the transition region, the turbulent eddies in the shear layer will efface the details of the nozzle core flow.

The boundary surface between the jet and the reservoir fluid is called the intermittency surface. The reservoir fluid, on the outside of the intermittency surface is non turbulent and is irrotational. The jet fluid within the intermittency surface, is both turbulent and rotational. Approximate solutions for turbulent jets have been developed by a number of workers and are based on the boundary layer time-averaged forms of the Navier-Stokes equations. For axisymmetric flow at constant pressure, these equations are

\[
U \frac{\partial U}{\partial x} + V \frac{\partial U}{\partial r} = \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial U}{\partial r} - r \frac{u'v'}{r} \right) \tag{2.10}
\]

The axial and transverse components of the flow velocities are the sum of the time averaged component and time-dependent deviation. The time averaged values \( u' \) and \( v' \) are zero but their averaged product \( u'v' \) is not. The kinematic viscosity is denoted by \( \nu \). Turbulent shear stress \( u'v' \) models have been developed to solve jet flow problems and the simplest of these models applicable to the axisymmetric jet is the eddy viscosity model:

\[
\overline{u'v'} = -\varepsilon \frac{\partial u}{\partial y}
\]

For this model the quantity \( U_{max} - U_{min} = U_m \), equals the centreline mean velocity.
Using $u'v' = -\varepsilon \frac{\partial u}{\partial y}$, substituting this equation in (2.10) and neglecting kinematic viscosity $v$ in comparison with eddy viscosity of turbulence $\varepsilon_0$ result in the following equation:

$$u \frac{\partial u}{\partial x} + \nu \frac{\partial u}{\partial r} = \varepsilon \frac{\partial}{\partial r} \left( r \frac{\partial u}{\partial r} \right)$$

(2.11)

The species profile in the jet is a simple function of the axial flow velocity profile:

$$\frac{\Delta C}{\Delta C_m} = \left( \frac{U}{U_m} \right)^{\frac{\varepsilon}{\varepsilon_T}}$$

(2.12)

Thus if the axial velocity is known for a plane or axisymmetric jet, then the species profile can be calculated from equation 2.12, and agree with experimental data for an axisymmetric jet $\varepsilon / \varepsilon_T = 1/1.4$ and $\varepsilon / \varepsilon_T = 0.5$ for a plane jet. The centreline velocity and profile of velocity is given by:

Plane jet: $U(x, y) = 12 \left( \frac{r_0}{x} \right) U_0 e^{-5\left( \frac{y}{x} \right)^2}$

(2.13)

Axisymmetric jet: $U(x, r) = 12 \left( \frac{r_0}{x} \right) U_0 e^{-9\left( \frac{r}{x} \right)^2}$

(2.14)

Equations 2.13 and 2.14 are for submerged turbulent jets.

Entrainment velocities are defined as follows:

Plane jet: $u_e = \frac{1}{2b} \frac{dQ}{dx}$

(2.15)
Axisymmetric jet: \[ u_e = \frac{1}{2\pi b} \frac{dQ}{dx} \] (2.16)

The entrainment velocities are the component of flow velocity across the intermittency surface toward the axis of the jet. \( \dot{Q} \) is the volume flow rate of the jet, i.e.

\[
\dot{Q}(x) = \int_A U(x, y) dA(y)
\] (2.17)

(volume flow rate per unit length of slot for the plane jet) which increases with the axial distance \( X \). The cross sectional plane area \( A \), corresponds to \( X = \text{constant} \); \( b^* = 2.5b \) is an estimate of the distance from the centreline of the axisymmetric jet to the edge of the intermittency surface. The required radius is \( b \), for the axial flow velocity \( U \) to fall to one half its value along the axis. Generally turbulent jets entrain more air than laminar jets.

2.1.2. Jets in Coflow

A jet in a coflow discharges into a fluid flowing in the same direction as the jet; as shown in Figure 2.3. A coflowing jet is also termed a compound jet or in the case of an axisymmetric jet, a coaxial jet. The ratio of the nozzle exit velocity to the velocity of the coflow characterizes a jet in coflow. As the velocity of the surrounding fluid is decreased to zero, the coflowing jet becomes a submerged jet issuing into still fluid. The jet disappears if the velocity of the surrounding fluid becomes equal to its own and if it exceeds the coflowing jet it becomes a wake.
Conservation of momentum in the axial direction for this case of a moving control surface in a uniform pressure field in the absence of any mechanically applied forces for a plane jet in coflow is given by the equation:

\[ U_0(U_0 - U_1)2b = \int_{-\infty}^{\infty} U(U - U_1)dy \]  \hspace{1cm} (2.18)

where \( b = \) jet width and \( b = 0.1X \) for a submerged jet. \( b = c_1X_1 \) for \( U_d/U_1 >> 1.0 \). For weak coflow jets

\[ b = c_1X^{c_2} \]  \hspace{1cm} (2.19)

### 2.1.3 Round Jets with Swirl

Swirl is the circumferential component of velocity which causes a round jet to rotate about its axis which may at the discharge end of jet fans unless they are fitted with flow straighteners. A form of ductless fans called vortex fans used in mine ventilation are used to provide a strong swirling jet and in most cases have high entrainment capabilities.

The dimensionless swirl number \( S \) is defined as the ratio of the angular momentum of the jet

\[ T = 2\pi \int_0^\infty r^2uwdr \]  \hspace{1cm} (2.20)

to the total axial momentum times the nozzle radius \( r_o \)
\[
W_{r_o} = \frac{2\pi r_o}{\rho} \int_0^r (p - p_{\infty} + \rho u^2) rdr = 2\pi r_o \int_0^r \left( u^2 - \frac{w^2}{2} \right) rdr \quad (2.21)
\]

\[
S = T / (W_{r_o}) \quad (2.22)
\]

P and \( P_{\infty} \) are the static pressures of the jet and the reservoir into which the swirling jet issues respectively and \( P < P_{\infty} \) in a swirling jet. The velocities \( u \) and \( w \) are time averaged components of axial and circumferential velocity, respectively; \( r \) is the radial distance from the centreline and \( r_o \) is the nozzle radius. Both axial momentum and angular momentum are conserved in a jet with swirl. The swirl number \( S \) is constant for a given nozzle flow.

In addition to swirl from \( S=0 \), the following changes take place in the jet: (i) the centreline velocity decreases more rapidly with axial distance (ii) the jet spreads more rapidly with axial distance (iii) The static pressure along the jet centreline decreases below reservoir pressure.

The presence of swirl increases the mixing in a round jet by increasing the jet width while decreasing the jet axial velocity. Profiles of axial velocity, radial velocity, circumferential velocity, and static pressure are axisymmetric about the jet axis.

While the maximum axial velocity and minimum static pressure fall on the jet centreline, the maximum circumferential velocity is located at approximately \( r / x = 0.12 \) where \( r \) is the radial distance from the nozzle. In summary swirl increases the spread of the jet. The swirl number \( S \) can vary from zero to a very strong value of 1.4.
2.1.5 Summary of Previous Work on Confined Jets

Figure 2.4 shows a ducted jet defining four flow regions. In Region I the inner and outer jet flows are separated by a shear layer. In Region II the shear layer has extended to the duct walls and the fluid is entrained from the surrounding stream rapidly enough to reduce the velocity of that stream. Region III is a region of possible eddy formation and recirculation. If the pressure gradient is large enough and the coflow velocity is small enough, the central jet will laterally entrain all the fluid in the coflow before the jet has spread to the wall. Thus, a recirculating eddy will be established. Region IV begins after the point of reattachment and marks the beginning of conventional duct flow. Jet fan flows can be considered as ducted flows and therefore can be approximately described by general jet flow relationships with some modifications.

Hill (1965) using the differential-integral technique was able to predict the mean flow field for jet mixing in the presence of fixed and varying area cross sections. Hill evaluated the velocity shear distribution and similarity profile in integration terms from free jet data. Confined jet mixing differs from free jets in that momentum is not conserved. Some results and techniques of the free jet investigations are readily extendible to confined flows. The presence of the confining walls causes a pressure gradient which modifies the rate of spread of the jet, rate of growth of the boundary layer, and velocity profile shape. The flow is more complicated than the boundary layer flow with an imposed adverse pressure gradient. The pressure gradient, while determined primarily by the mixing of the two streams is related to the boundary layer growth. Therefore, the flow in the early mixing region is much like a conical diffuser flow.

Curtet and Ricou (1964) conducted an experimental investigation of the axisymmetric confined jet to check the theoretical approach of Curtet (1958). Measured velocity profiles
were not strictly similar but the terms neglected by the assumption of similarity were found to be small. Razinsky and Brighton (1971) have also investigated the mixing of an air jet with a lower velocity air stream in a constant diameter pipe. The flow was investigated from the inlet where the jet and secondary velocities were uniform (but different to a location downstream where the flow is fully developed). Their measurements included wall static pressures, mean velocities, turbulence levels and Reynolds stresses.

2.2 Jet Fan Measurements in Mine Ventilation

Research in mine ventilation into jet fans has been very limited and the available previous work has been conducted by various workers with the U.S. Bureau of mines (USBM). Only a few articles have been published which are presented below.

Matta et al. (1978) evaluated jet fan effectiveness using SF6 tracer gas analysis in the ventilation of dead headings. Results of their study demonstrated the ability of jet fans to ventilate dead headings beyond maximum penetration distance indicated by smoke-tube detection of air movement and to redistribute available fresh air through larger working areas. They further commented that there must be sufficient fresh air for the jet fan to operate with in order to avoid the recirculation of contaminated air in the heading. Often the mine has sufficient fresh air flowing through it but directing the air where it is actually needed is always a problem. In working areas where mining is in progress, large quantities are necessary to dilute and remove contaminated gases.

Auxiliary fans redistribute this air and jet fans are in fact auxiliary fans that do not use bulkhead or tubing. Krause (1973) has shown that the penetration distance of a free jet is doubled by placing the fan outlet along a mine rib. The penetration was measured as the distance from the fan to where the velocity dropped to 0.33 m/s. The air is only entrained
from one side of the jet and as a result a "half jet" is formed. Computer studies and mine ventilation surveys show that jet ventilation systems can considerably improve the distribution of air currents in mines with large cross-sectional areas.

The Matta et al. (1978) study of the US.B.M. was reasonably successful in using sulfur hexafluoride (SF6) tracer gas to determine the effectiveness of a jet fan in providing fresh air. Studies by Drivas et al. (1972) using SF6 tracer gas in the evaluation of ventilation systems in buildings have enabled the effective purging of various rooms and provided the decay curve of the tracer gas. The Drivas method was used by the Matta et al. study (1978) in dead-end headings to evaluate the effectiveness of different fan sizes, effect of their positioning and the ability of the jet fans to redistribute fresh air. The effectiveness of jet fans in purging a fixed volume $V$, for SF6 released uniformly throughout the volume with concentration $C$, at time, $t$, can be determined in terms of a concentration decay.

$$C = C_o \exp(-Q/V)t \quad (2.23)$$

$Q$ is the amount of fresh air purging the volume, and $C_o$ is the initial SF6 concentration. A semi-log plot of the concentration versus time yields a straight line with a slope equal to $-Q/V$. The slope is given by the equation

$$-Q/V = \frac{\ln(C_1/C_2)}{t_1-t_2} \quad (2.24)$$

where $C_1$ and $C_2$ are any two concentrations along the straight line at time $t_1$ and $t_2$ respectively. The amount of fresh air $Q$ can be determined for a heading of volume $V$. 

Figure 2.5 shows the dead-heading test site used by Matta and coworkers for their investigations. Fan sizes of the following diameters: 635, 737, and 762 mm were used to ventilate the headings and to show the resulting SF6 decay curves for each size of fan. Figure 2.6 shows a test area with an inadequate source of fresh air, and Figure 2.7 shows a test area ventilated by a jet fan. SF6 measurements were used to determine the effectiveness of jet fan ventilation in the test site.

Typical SF6 decay curves for the 635 and 762 mm fan were determined in order to assess the rate of potential pollutant clearance by the jet fans. The 635 mm diameter fan had penetration distances of up to 27 m into the heading. The Matta study demonstrates the effectiveness of SF6 in evaluating ventilation studies of this kind.

A more detailed study was carried out by Thimons et al. (1986) in which they tried several strategies of face ventilation for oil shale mining to dilute and remove pollutants from the working face area. They considered all possible sources of air pollutants expected in oil shale mining and in particular the problem of methane and diesel pollutants was well addressed. For each of the pollutants including respirable dust, Thimons et al. quantified the amount of fresh air required to dilute these pollutants below the Threshold Limit Values (TLVs).

Thimons et al. favoured a jet fan system for non gassy oil shale mining and reversible fans with rigid ducts for gassy mines. Each system had a common design basis including (i) 154.8 m³/s capacity (ii) low power consumption (iii) components that must be handled with a minimum of special equipment (iv) two speed operation to conserve power consumption when full flow was not required.
In the characterization of the turbulent jet from the jet fan Thimons et al used a simplified method developed by McElroy (1945) even though theoretical analysis performed by Abramovich (1963) existed. McElroy's work was limited because it was solely based on empirical relationships. McElroy developed a group of equations to describe the decay in centreline velocity with increasing distance from the fan outlet. These equations are correlated with four phases of behaviour, as shown in Figure 2.8. For a round jet in phase 1 and 2 the centreline velocity $U_x$, at a distance $X$ from the fan outlet is characterised by

$$U_x = \frac{aU_o}{X} \quad (2.25)$$

$U_o$ = outlet discharge velocity, $a = \text{constant (between 1.0 and 1.2)}$

Equation 2.25 holds fairly well up to 5 outlet diameters. The constant $a$, decreases with decreasing velocity. In phase 3

$$U_x = \frac{KU_o D}{X} \quad (2.26)$$

where $K = \text{constant (between 3 and 10)}$, $D$ is outlet diameter, $X$ is distance from outlet. In the transition zone the centreline velocity $U_x$ decays rapidly to the range that is predicted using the flowrate and one half the area of the opening. After this zone $U_x$ is given by

$$U_x = \frac{k_f U_o D}{10Gx} \quad (2.27)$$
$k_f$ is a constant related to the ratio of the outlet diameter to opening dimension and $G = 0.026k_f$.

$$k_f = 12.2 \frac{DS_a}{W} \quad (2.28)$$

$S_a = \text{aspect ratio of the fan outlet} (= 1.0 \text{ for a round jet})$. $W$ is the large dimension of the opening. These equations developed by McElroy are for freely expanding jets. Walls restrict the growth of the jet and increase the distance in which phase 3 behaviour is observed.

The equations developed by McElroy are similar to the work reported by Krause (1972) and were tested by fitting the relationships to experimental data from Lewtas (1980). Values of $K$ (in equation 2.26) varied between 5.0 and 11.3 for fans located within three fan diameters from the wall. An approximate linear relationship was observed between the distance from the fan outlet to the transition zone and the outlet velocity divided by the discharge diameter.

Thimons et al. (1986) showed that the zone heading that is of critical importance in mining applications is the transition zone and beyond. The entrainment action increases rapidly causing a rapid decrease in flow velocity. In the design process it is important to ensure that the jet can force air to the face with sufficient velocity to provide good air quality and a complete face sweep. The design of the jet fan system in Thimons et al. work was based on the following criteria: (i) Flow capacity based upon expected rates of pollutant emissions at the face (ii) Fan diameter selected such that the jet reaches the face with 0.5 m/s velocity. Using $K = 5.0$ in equation (2.26) a flow rate of 47 m$^3$/s, and opening dimensions of 17 m wide by 9 m high, fans with diameters of between 1219 and 1524 mm
were predicted to project air in the range of 91 m with a minimum velocity of 0.5 m/s. A 1400 mm diameter fan was chosen for this system.

The jet fan tests by Thimons et al. showed superior performance in clearing out diesel exhaust and methane from muckpile tests. The duct system was more effective in clearing out blast gases and methane layering. The jet fan air recirculation value was 28.4 % and that for the ducted system was 23.8 %.

For a given room volume V, Thimons et al. (1986) gives the following equation to calculate the time to reach a TLV of a pollutant in a dead heading.

\[ T = \frac{V}{Q_{ef}} (\ln C_o - \ln TLV) \]  

(2.29)

where \( T \) is the time in seconds to reach a TLV, \( Q_{ef} \) is the effective flowrate of the fan equal to \( E_d Q_{fan} \) (m\(^3\)/s). The dilution efficiency \( E_d \) is equal to volume of fresh air delivered divided by volume flowrate of the fan. The peak pollutant concentration in ppm, is described by \( C_o \).

Thimons et al. concluded that the jet fan was more efficient at a flowrate of 28.32 m\(^3\)/s than at 41.72 m\(^3\)/s and delivered similar dilution rates suggesting some interaction between the turbulent jet and room dimensions that is still not well understood.

Dunn et al. (1983) carried out some extensive studies on testing of jet fans in metal and non metal mines with large cross-sectional airways. Their work tested various sizes of jet fans in open airways, dead headings and face areas, and they were able to develop some
basic guidelines on the positioning and sizing of jet fans. Fan tests were conducted in three mines with cross-sectional areas ranging from 88 to about 186 m² and mainly in room and pillar mining. It was concluded that for open airways large jet fans are less efficient in smaller airways but effective in larger open airways in achieving both uniformity and greater air entrainment when inclined about 10⁰ or elevated. Jet fans must be placed against the rib of a heading closest to the incoming airstream from the crosscut to reduce recirculation. This work also used tracer gas analysis to determine airflow recirculation and distribution.

The ventilation current propagated by a jet fan along a longwall face has been investigated by Radchenko et al. (1965). Stochinsky and Komarov (1969) reported the use of jet fans for increasing airflow in mine workings and provided some empirical formulas to compute jet fan air quantities.

Goodman et al. (1990) conclude that jet fans will be widely accepted in future for deep advance mining and as a general auxiliary method of ventilation compatible with computer assisted mining. Research along these lines is being carried out by the U.S.B.M.

Meets and Meyer (1993) conducted some ventilation tests in room and pillar headings in South African coal mines using two types of ductless fans. One was called a jet fan because it had a fitted nozzle and the other a vortex fan because its fan blade design was developed from vertical take-off aircraft. The latter fan produced a vortex or swirling jet and there was no nozzle to increase the outlet velocity. The purpose of these studies was to develop better methods of ductless ventilation and to replace the traditional ventilation system with its disadvantages. This study provided airflow patterns for the ductless fans situated at various positions with obstructions in the heading and determined the
percentages of recirculated air. This work was a significant contribution to the subject but it did not go far enough in providing fundamental ductless ventilation design data.

2.3 Jet Fan Investigations in Vehicle Tunnel Ventilation

Eck (1973) gives the following expression for excess pressure developed by the fan in a tunnel ventilated by momentum drive:

\[
\Delta P = \rho U_j^2 \left( \frac{A_i}{A_t} + \left( 1 - \frac{A_j}{A_t} \right) \left( \frac{U_t}{U_j} - \frac{A_j}{A_t} \right)^2 - \left( \frac{U_t}{U_j} \right)^2 \right)
\]

(2.30)

where \( U_t \) is the tunnel air velocity and \( U_j \) is the fan outlet velocity. \( A_t \) is the tunnel cross-sectional area and \( A_j \) is the fan outlet area. This equation applies to any tunnel situation where induction ventilation system is used but it does not account for wall friction and tunnel length. Induction ventilation systems for tunnels have been the subject of major research efforts (e.g., Kempf, 1965). The experience gained in tunnel ventilation has never been extended to auxiliary ventilation in underground mines and it is the author's belief that theories developed by early studies in tunnel ventilation can find some ground in mine ventilation.

Baumann (1973) evaluated friction coefficients and mean wall roughness of Swiss tunnels from pressure and volume flow measurements. Baumann gives the pressure drop in a tunnel ventilated by a jet fan to be

\[
\Delta P_j \pm \Delta P_e = (\zeta_i + \alpha + \beta) \rho \frac{U_j^2}{2} + \frac{A_e}{A_i} C_e \frac{P}{2} (N_+ (U_j - U_t)^2 - N_- (U_j + U_t)^2)
\]

(2.31)

and the pressure rise due to the jet fans to be given by
\[ \Delta P_j = \frac{n}{K A_t} \rho Q_j \left(U_j - U_t\right) \]  

(2.32)

where \( \Delta P_j \) is the pressure generated by the jet fans, \( \Delta P_e \) is the sum of the external forces (barometric, thermostatic and wind forces), \( f \) is the friction coefficient \( (f = 2 \Delta P / (\rho U^2)) \), \( \zeta_e \) is the pressure drop coefficient for entry into tunnel portal, \( \zeta_e \) is the pressure drop coefficient for tunnel exit, \( \rho \) is the air density, \( n \) is the number of jet fans in the tunnel, \( K \), is the reduction factor for jet fan thrust, \( C_c \) is vehicle drag coefficient, \( L \) is tunnel length, \( D \) is the diameter of the tunnel, \( U_j \) is the jet fan outlet velocity and \( U_t \) is the tunnel velocity. \( Q \) stands for volume flow of the jet fan, \( A_t \) is the tunnel cross sectional area, \( A_c \) is the cross sectional area occupied by vehicles, \( N_+ \), \( N_- \) is the number of vehicles in the direction of flow and against flow respectively. The friction coefficient can be estimated from measurements of velocity and pressure fields.

Rohne (1964) assumes the fan thrust to be

\[ F = \rho Q_j (U_j - U_t) = \Delta P A_t \]  

(2.33)

from which the pressure head developed by a single fan, \( \Delta P \) results.

\[ \Delta p = \rho U_j^2 \frac{A_j}{A_t} \left(1 - \frac{U_t}{U_j}\right) \]  

(2.34)

and in terms of velocity and area ratios

\[ \Delta P = \rho U_j^2 \Omega (1 - \Phi) = \rho U_t \Omega (1 - \Phi) / \Phi^2 \]  

(2.35)
Meidinger (1964) applied continuity and mechanical energy equations to obtain

$$\Delta p = \frac{\rho}{2} U_j^2 \left( \frac{2\Omega - 2\Omega^3 - 2\Omega^2 - 2\Omega \Phi + \Phi^2 \Omega^2}{(1 - \Omega)^2} \right)$$  \hspace{1cm} (2.36)

where $\Phi$ is velocity ratio, and $\Omega$ is area ratio $A_j/A_i$. Reale (1968) obtained

$$\Delta p = \rho u_j^2 \frac{\Omega}{1 - \Omega} (1 - \Phi)^2 = \rho u_i^2 \frac{\Omega}{1 - \Omega} \left( \frac{1 - \Phi}{\Phi} \right)^2$$  \hspace{1cm} (2.37)

by applying momentum and continuity equations.

Reale (1973) defines an induction efficiency $\eta_i$ as the ratio of the ventilation power output to the power transmitted to the fluid by the fan.

$$\eta_i = \frac{2\Delta P Q_i}{\rho U_j (u_j^2 - u_i^2)} = \left( \frac{2\Phi}{1 - \Omega} \right) \left( \frac{1 - \Phi}{\Omega^2} \right)$$  \hspace{1cm} (2.38)

and total efficiency of the ventilation system is given by

$$\eta_s = \eta_i \eta_f$$  \hspace{1cm} (2.39)

where $\eta_f$ is the overall fan efficiency. Reale (1973) promotes the use of experimental tests on ventilation models because a theoretical analysis alone is insufficient in giving design data useful for modern systems.
In tunnel ventilation, the losses due to vehicle motion and the effect of winds must be considered when evaluating the performance of jet fans. Possibilities for the reduction of energy consumption in ventilation systems using jet fans are considered by Pinter (1982) particularly for tunnels with lengths of up to 2 km. Pinter divides the possibilities of saving energy in jet fan ventilation systems into three categories (i) reduction of the tunnel resistance (ii) increase of efficiency of the jet fan ventilation and (iii) reduction of operating hours a year. Tunnel resistance is comprised of entrance, exit and friction losses. Entrance loss coefficients for well designed tunnel portals are around $\zeta_e = 0.3$ and can be as low as 0.1. Traffic signs and the piston effect of moving vehicles inside the tunnel increase its resistance. In the mining situation mine vehicles and other obstructions contribute to resistance.

Fan efficiencies of up to 70 % are possible with axial-flow fans. A number of factors affect the efficiency of the fan arrangement and include the reduction of thrust due to wall friction and uneven diffusion of the jet if the fan is installed near the wall. Factors have to be incorporated in the equations to account for losses. Pinter suggests fitting conically extended transition pieces to the impeller stage of the fan, to improve efficiency of the jet and to better utilise the free space in the tunnel cross section. This results in lower jet speeds and higher jet efficiencies and with significant lower energy consumption of up to 55 %. The reduced jet speeds result in lower thrust and require more jet fans for a given length of tunnel, which increases capital costs.

Noon and Smith (1990) compared the performance and sound levels of jet fans in laboratory and road tunnel tests. The main parameters measured were volume flow, axial thrust, input power and sound level using the BS848: Part 1:1980 Type Chamber A test method. Noon and Smith conclude that more work needs to be carried out in order to find
methods of improving jet fan system efficiency, and in particular to establish a reliable method of measuring the pressure rise in a tunnel due to a fan.

Mizuno and Araie (1989) carried out some measurements of pressure rise performance of a jet fan in a tunnel by the use of a 1/35 scale model experiment. Their model tunnel was 8m long and used a 230 mm diameter acrylic circular tube. The size of the jet pipe was 46 mm in diameter. They used \( U_i / U_j \) values of 0.2, 0.4, and 0.66 and traversed the jet from the centre of the duct to the wall. In this case \( U_i / U_j \) is the ratio of mean velocity of the duct to the jet velocity \( U_j \). In order to predict the pressure rise coefficient \( C_{p_{th}} \) they used the expression

\[
C_{p_{th}} = 2\Omega(1 - \Phi)\left[1 + \frac{\Omega}{2}(1 + \Phi)\right]
\]  

(2.40)

which is close to Meidinger's (1964) equation

\[
C_{p_{th}} = 2\Omega(1 - \Phi) + \frac{\Omega^2 + 2\Omega^2\Phi - \Omega^2\Phi^2 - 2\Phi^3}{(1 - \Omega)^2}
\]  

(2.41)

Equations 2.40 and 2.41 reduce to

\[
C_{p_{th}} = 2\Omega(1 - \Phi)
\]  

(2.42)

because the second terms can be ignored since they are negligible. In the above equation \( \Omega \) is the area \( A_j/A_t \) and \( \Phi \) is the velocity ratio of the tunnel and jet discharge. Mizuno and
Araie observed that 15% of the momentum is lost by friction when the jet is in contact with the wall.

Hayward (1973) used model road tunnels to evaluate a longitudinal tunnel ventilation system. Smith (1982) investigated the design aspects of high reaction fans from both an aerodynamic and mechanical design perspective in order to increase jet fan performance. Other jet fan studies in road tunnel ventilation were performed by Baba and Ishida (1985) to determine economic considerations. Like Pinter they favoured low jet fan outlet velocity as opposed to high velocities of up to 30 m/s. They showed that the input energy of a 30 m/s velocity jet fan is about 3.4 times that of a 20 m/s velocity jet fan so that for a 20 year period, the total cost of a low velocity fan application will be lower than that of a high velocity fan application. Fudger and Lowndes (1985), and Ohashi et al. (1976) also provided some useful information on vehicle tunnels ventilated by jet fans.

Previous studies have addressed various aspects of jet fan ventilation but they have not answered all questions pertaining to system effects of jet fan aerodynamics. More research is needed in this area. The nature and extent of work proposed in the present study will have far reaching results. The present research investigates the most important and fundamental aspects of jet fan ventilation.
Fig. 2.1 A jet issuing into a fluid reservoir
(Ue is the entrainment velocity)
[bo=ro and y=r for axisymmetric jet]

Fig. 2.2 Submerged turbulent jet (not to scale)
Fig. 2.3 Submerged free jet in a coflowing stream
Fig. 2.4 Ducted jet showing regions of development
(not to scale)
Figure 2.5 Mine heading test site (Matta et al. 1978 study)
Fig. 2.6 Mine test area with an inadequate source of fresh air
Figure 2.7 Mine test area ventilated by a jet fan (Matta et al. 1978 study)
Fig. 2.8 Illustration of McElroy's different phases of velocity decay of a freely expanding turbulent jet
CHAPTER THREE

EXPERIMENTAL PROGRAM

3.1 Design and Construction of Experimental Apparatus

3.1.1 Wind Tunnel

In order to investigate jet fan aerodynamics in mine ventilation a wind tunnel was used to simulate a mine environment or road tunnel arrangement. It was the philosophy of this work that jet fans can be simulated by carefully using laboratory test facilities. The use of models in experimental work provides many advantages in both cost and convenience. In order to predict the performance of a full scale industrial jet fan from the small scale model, complete similarity of the flow pattern is required.

This requires both kinematic and dynamic similarity as well as geometric similarity. Kinematic similarity requires that velocities and velocity gradients be exactly proportional and dynamic similarity requires that various force ratios be equal in each case. It is not always possible to obtain complete similarity. Generally it is important to maintain similar flow conditions by keeping the same Reynolds number. Dynamic dissimilarity arises principally in the boundary layer flow due to frictional and shear forces and shock losses.

Common design methods for sizing ducts may be listed as (a) constant velocity, (b) velocity reduction, (c) equal friction and (d) static regain. The theory of the design of low speed wind tunnels is given by Bradshaw and Pankhurst (1964), and Pope and Harper (1966) and many others.
The studies to be carried out in this research demanded a unique wind tunnel to be solely available all the time. Wind tunnel facilities in the Mechanical Engineering Department (UBC) could not be used due to the fact that they were being used all the time for other studies and that they were not quite suitable for mine ventilation measurements. There are two main factors which influenced the present wind tunnel design (i) a 7.5 kW variable pitch axial flow fan of diameter 630 mm was already available for use in the wind tunnel design. The fan was manufactured by Woods of Colchester (England) and can handle an airflow of more than 10 m$^3$/s if required. (ii) A rectangular shaped wind tunnel was required to simulate the general geometry of mine drifts and vehicular tunnels. The cross sectional dimensions had to be large enough for the insertion of objects and allow easy access to personnel working in the tunnel.

The space available for the layout of the tunnel also limited the maximum length that could be achieved. The current design is unique because the size of the wind tunnel that was required in relation to the area of the room in which it was to be situated, made the task a challenging one. There are many standing structures at the walls of the room and near the end, the roof is abruptly lower to accommodate a concrete explosion duct. The most important objective, despite the unfavourable conditions of the room was to achieve an inlet airflow to the wind tunnel that was as smooth as possible. Budgetary constraints also limited the size and material that could be used for the construction of the wind tunnel. After consideration of all the factors it was decided that the wind tunnel cross sectional dimensions should be 900 by 900 mm. The length of the wind tunnel depended on the flow conditions to be achieved in the working section. The flow conditions should be as smooth as possible over the test section especially if accurate aerodynamic studies were to be conducted to reduce air fluctuations.
Most flows that exist in mine ventilation are turbulent. For a fully developed turbulent flow the length \( L \), to diameter \( D \), ratio is given by

\[
\frac{L}{D} = 14.2 \log_{10} \text{Re} - 46
\]  

(3.1)

for \( \text{Re} > 10000 \). \( L \) is the inlet development length of the flow, and \( D \) is the diameter of the duct. For a tunnel diameter of 0.9 metres the above equation would require a length of at least 10 m for a Re=10000. The entrance length \( L \), cannot really be treated mathematically since there is a downstream reduction of the "potential core" because of the boundary layer growth profile. The wind tunnel is a multi-purpose one and the entire length is used for measurements especially for jet fan aerodynamic studies. The main body of the tunnel of 8 metres was deemed a sufficient test length for the various studies to be carried out and when required honeycombs and screens can be fitted at the inlet to straighten and reduce the turbulence of the flow.

A contraction piece was designed to join the square cross section wind tunnel to the 630 mm diameter axial flow fan. The contraction piece design was critical for uniform smooth flow from the wind tunnel to the axial flow fan. A good design was essential in order to promote good fan performance. Poorly designed contraction pieces lead to very high pressure losses. The contraction piece is square at the tunnel side and circular at the fan side. The contraction area ratio is 2.61:1 and according to Figure 3.1 (Chmielewski, 1974) gives a length to diameter ratio \( l_c / D_t \) of 0.7. The contraction length is \( l_c \) and diameter of the wind tunnel \( D_t \) is equal to 900mm. This \( l_c / D_t \) ratio avoids flow separation which should be prevented in the contraction section because this would bring flow irregularities to the fan and eventually cause it to stall. The length of the contraction piece worked out to be 0.63 metres.
In order to produce smooth flow conditions in the tunnel an inlet section was designed which ensured maximum aerodynamic performance. A bellmouth entrance piece gives the lowest flow loss coefficient. The minimisation of entry losses to any wind tunnel is critical to obtaining good measurements. The pressure loss coefficient $K_x$ for bell mouth inlet pieces is dependent on the ratio $r/D$ where $r$, is the radius of curvature of the bell mouth piece and $D$, is the diameter of the tunnel. For values of $K_x$ approaching zero, the ratio $r/D$ gradually approaches unity. The radius of curvature $r$, for the wind tunnel bell mouth inlet piece was chosen to be 900 mm to give an $r/d$ ratio of 1. It was determined that the bell mouth piece should be 540 mm long and the inlet dimensions are 1264 by 1264 mm. It has the same dimensions as the wind tunnel where the two are joined by flanges.

3.1.2 Construction of the Wind tunnel

The bell mouth entrance piece is made out of aluminum and it has a smooth profile to reduce inlet turbulence. A plywood transition piece of 900 by 900 mm cross section and 800 mm long is connected to the inlet piece and the main body of the wind tunnel. The main body of the wind tunnel is 7314 mm long and is constructed in three equal parts. The roof and the floor of the wind tunnel are made from wood board whose surface was specially treated with Danish oil finish an oil resin sealer and varathan diamond finish an interior penetrating polymer coating with twice the abrasion resistance of polyurethane to give it a smooth texture when it was cured. The wind tunnel sides are made from plexiglass 9.5 mm thick. The advantage of plexiglass is that, it enables flow visualization tests to be made since it is transparent. The plexiglass pieces are inserted into 9.5 mm wide grooves of the same depth along the entire length of the roof and floor of the tunnel. Special gaskets cushion the joints of the plexiglass and the wood making the whole structure completely air tight. At every joint, wooden ribs of 50 by 75 mm cross section are fastened to the structure one each side of the tunnel as shown in Figure 3.2. All legs
are made of wood and are distributed evenly in order to anchor the structure properly. A contraction piece joins the wind tunnel to the fan through flanges and rubber gaskets. The contraction piece is made from aluminum.

Figure 3.3 shows static pressure holes drilled on one side of the tunnel and spaced at 180 mm apart in each of the six sections of the tunnel. On the other side of the tunnel six sets of 5 holes are located in each section for the insertion of velocity measurement instruments as shown in Figure 3.4. Complete velocity traverses can be made at each station covering the entire cross section and length of the wind tunnel. Figures 3.5 and 3.6 show the layout of the wind tunnel as viewed from the bell mouth inlet piece and the axial fan discharge outlet respectively. The layout of the wind tunnel in relation to the room can also be seen clearly in these figures. Mutama and Hall (1993) give a detailed description of design, construction and testing of the wind tunnel.

3.1.3 Wind Tunnel Instrumentation

Measurement techniques for airflow are described by Ower and Pankhurst (1977) among others. The instrumentation available included an array of low pressure transducers which were connected to the static pressure holes to give the axial pressure distribution in the tunnel. The pressure transducers were connected to an analog-digital data acquisition board system linked to a computer. These pressure measurements were necessary when assessing jet fan aerodynamics.

The instrumentation available for the measurement of velocity comprised an Airflow Developments MED 500 digital micromanometer which worked in conjunction with a pitot static tube and was capable of displaying velocity, pressure and volumetric flow readings in both S.I. and English units. This instrument had extra features which gave the
user reliability in assessing the various aspects of airflow. The instrument could also be connected to an analog data logger so that the readings could be recorded for further analysis on a spreadsheet. Various sizes of pitot static tubes and a hot wire anemometer were available for the measurement of velocity and these were adapted so that they could be used with the electronic traversing ruler. This ruler could determine the position of a velocity traverse to within 1/100 of a millimeter. An Omega Thermo-electronic vane anemometer with analog outputs was used mainly to measure the inlet velocity profiles of the wind tunnel.

3.1.4 Wind Tunnel Testing

The test program for the wind tunnel was aimed at assessing flow distributions and uniformity at two different flow settings. Flow patterns at the inlet bell mouth piece were determined. Axial pressure variations were measured for both a high and low Reynolds number regime. Pressure drops across the contraction piece between the fan and the wind tunnel were determined. The above mentioned instruments were used to measure the static pressure and velocity. The instrumentation was arranged as shown in Figure 3.7. Results of the test program are presented and discussed in the following section.

3.1.4.1 Test Results

In Figure 3.8 axial static pressure variation is shown for the two flow settings investigated in the test program. The wind tunnel was tested at two Reynolds numbers of 219560 and 427253, both of them being high turbulent airflows. For the Re=219560 the static pressure can be said to be constant throughout the main body of the wind tunnel i.e from the end of the inlet piece to the start of the contraction section which joins the fan. For the high flow setting the pressure changes relatively little throughout the tunnel but fluctuations are
noticeable because of the higher turbulence that is present. In fact for this type of wind tunnel the static pressure is not expected to change by any significant percentage throughout the constant area portion of the tunnel.

Figure 3.9 shows the centreline entrance velocity time plots of the wind tunnel for the two flow settings investigated. The fluctuations shown in Figure 3.9 for the Reynolds number of 219560 are not as large as those for Re = 427253. The inlet flow disturbances increase significantly as the volume flow is increased. A large proportion of this fluctuation results from the wind tunnel room. The structure occupies a third of the space available in the laboratory and at high Reynolds numbers air currents exist throughout the whole room. In fact turbulence around the wind tunnel room was deliberately increased in order to establish its effect on the inlet velocity and Figure 3.9 shows this effect. The nature of the aerodynamic tests performed inside this tunnel are not affected by the existence of background turbulence in the room as long as circulating air currents at the inlet are reduced by a significant proportion. Turbulence could be reduced by eliminating some of the return currents from the fan discharge to the wind tunnel inlet.

Results of the velocity traverses can be seen in Figures 3.10 and 3.11 for three stations in the tunnel. In the Figures the length of the tunnel is presented in non-dimensionalized form as X/Dt where X is the axial distance from the inlet and Dt is the wind tunnel diameter. Z/Dt represents non-dimensionalized distance from the wind tunnel roof to the floor. The first station was at X/Dt = 2.21 (1.99 m), the second at X/Dt = 4.93 (4.44 m) and finally the third was established at X/Dt = 7.65 (6.89 m) from the entrance of the tunnel. These three stations were carefully chosen from the six traversing stations available in the tunnel. As can be seen from the graphs five traverses were performed at each station at positions of Z/Dt of 1/6, 1/3, 1/2, 2/3 and 5/6. The traverses were performed from one side to the
other side wall of the wind tunnel. The velocity profiles are of a turbulent nature and the distributions are quite flat. In Figure 3.10(a) to 3.10(b) it can be seen that the velocities of the five traverses of each station are all within a close magnitude. The profiles show an even distribution of the flow with very thin turbulent boundary layers.

At the last measuring station at the lower flow setting the velocities exhibit a very smooth profile showing good flow symmetry although in Figure 3.11(b) unlike 3.10(b) the maximum velocities were obtained at the bottom of the tunnel. One would expect the velocities at Z/Dt of 2/3 and 5/6 to be lower than those at Z/Dt = 1/2 since this is the centre of the tunnel. In an ideal situation the profile at Z/Dt of 1/6 and 5/6 are similar and those at Z/Dt of 1/3 and at 2/3 are also expected to be the same since they are taken at the same distances relative the top and bottom walls respectively.

One other aspect of the wind tunnel testing was to establish the pressure drop variation across the contraction piece which joins the tunnel to the fan as a function of Reynolds number or flowrate. The result of this test was quite satisfactory as shown in Figure 3.12 and it conforms to the square law relationship i.e. the pressure drop is proportional to the square of velocity.

### 3.2 Jet Fan Simulation and Arrangement

The sizes of the fans available on the market that could be fitted in the wind tunnel for jet fan tests were too large to make an effective simulation. The smaller fans were not able to produce a jet outlet velocity sufficient to enable effective jet fan assessment unless they were fitted with extension pipes of smaller diameter than the fan in order to increase the fan discharge velocities.
A compressed air jet was a good option for jet fan simulation and it offered the advantage that the primary flow can be varied and measured easily but it could not be installed in the wind tunnel. A centrifugal fan was selected and adapted for the jet production. This was achieved by redesigning its discharge side. A plenum air chamber measuring 295 by 262 mm and 300 mm long is fitted to the discharge side of the fan. A conical contraction accelerates the flow to 100 mm diameter aluminum pipe. The aluminum pipe is 200 mm long and can be fitted with an orifice plate between its flange and that of another 100 mm diameter aluminum pipe. The second aluminum pipe extends into the tunnel for a distance of 1400 mm. The end of the aluminum pipe can be fitted with nozzles of different diameters ranging from 50 to 100 mm. The nozzles can be either straight, or converging. Air jet velocities ranging from 1 to 20 m/s can be achieved with this centrifugal fan which has a 1/3 hp motor on a 100 mm diameter pipe without the orifice plate. When the orifice meter is installed it causes so much resistance such that there is hardly any flow.

A much more powerful centrifugal blower with a 1 hp motor capable of discharging 0.378 m$^3$/s (800 cfm) was purchased in order to provide higher outlet jet velocities. The orifice meter was used to determine the air jet mass flow rate. A detailed arrangement of the jet mechanism in relation to the wind tunnel is shown in Figures 3.13 and 3.14.

A range of jet outlet velocities could be produced by adjusting the suction area of the fan. The jet producing centrifugal fan is mounted on traversing rails or grooves such that the discharge pipe or jet is situated on the centreplane of the tunnel i.e. 450 mm from the floor of the tunnel. Therefore the jet can be traversed from wall to wall of the tunnel. This enabled jet wall proximity to be investigated. The secondary flow in the wind tunnel could be varied by adjusting the pitch of the axial flow fan blades if required.
3.3 Experimental Description of Jet Fan Performance Measurements

3.3.1 Jet Fan Velocity Field Measurements

The instrumentation used to assess the jet fan performance is as described in section 3.1.3 and shown diagrammatically in Figure 3.7. Velocity measurement was carried out using a hot wire anemometer which can determine one velocity component and its fluctuation. An Airflow Developments MED500 digital micromanometer working in conjunction with a pitot static tube was also used as a backup for some of the measurements. The velocity readings were recorded on a data logger or an IBM PC. An electronic vane anemometer of 25 mm diameter specially adapted for the present study was used to determine the magnitude and direction of the flow whenever required and could be traversed across the wind tunnel. All the velocity and pressure traversing probes were coupled to a digital traversing ruler described earlier.

Velocity traverses were performed at a height of 0.5D_1 from the tunnel floor although at first they were performed at each of the five vertical holes of the six stations in order to assess the complete velocity distribution of the flow at a particular cross section. After the initial measurements were completed it was sufficient to determine velocity profiles at the height of 0.5D_1 from the tunnel floor for all the six stations and at each fan position. At the 5th and 6th measuring stations complete velocity grids were determined in order to obtain the total volumetric flow rate. The amount of air entrained by the fan could be calculated by subtracting the initial fan discharge volume flow from the total flow out of the wind tunnel. The purpose of velocity measurements were twofold; (i) to determine the jet diffusion process from the fan and (ii) to determine the entrainment characteristics of the confined jet fan. The axial velocity fluctuation pattern could also be determined giving a good indication of the turbulence in the tunnel.
3.3.2 Jet Fan Pressure Measurements

Pressure measurements were carried out using an array of transducers which were connected to the static pressure holes to give the axial pressure distribution of the tunnel. The pressure transducers were connected to an analog-digital data acquisition board system coupled to a computer. In addition a digital micromanometer was used quite successfully to record the axial variation of static pressure. It proved to be a reliable source of pressure measurements. The purpose of the pressure measurements was to determine how the pressure field varied axially and obtain pertinent data from it. This was a useful exercise because very interesting results were obtained.

The fan was set to deliver jets with 20.8 and 40 m/s outlet velocities from a jet fan diameter of 100 mm i.e. $D_R (D_j/D_t)$ of 0.11. A second jet fan to tunnel diameter ratio of $D_R = 0.17$ was also used with a jet outlet velocity of 21 m/s. The jet fan was fitted with straight nozzles in each case. The range of velocities of between 20 and 40 m/s is the usual one used in jet fan ventilation. In this study it was necessary to use the two velocity extremes in order make a good comparison. The two chosen jet fan to tunnel diameters $D_R$ of 0.11 and 0.17 were a good starting point for the study of jet fan-tunnel geometry which is an important parameter.

The jet fan was traversed at positions $F_p$ ($Y/D_t$) of 0.06, 0.11, 0.17, 0.22, 0.28, 0.33, 0.39, 0.44 and 0.5. from one tunnel wall. These positions corresponded to $Y/D_t$ steps of 0.055 from one wall to the centre of the tunnel. The reason for doing this was to study the effect of the tunnel walls on jet fan performance. The first task was to record axial static pressure distributions on both sides of the wind tunnel. The first measurements were taken with the jet fan in contact with the wall and subsequent measurements were taken at varying $Y/D_t$ positions until the jet fan was at the centre of the tunnel.
corresponding to $Y/D_t = 0.5$. For each jet position 36 axial wall static pressure readings were taken. It was necessary to record a large sample of readings for each static pressure position so that a representative average could be determined.
Figure 3.1 Minimum length for contractions, without separation
Figure 3.2 View of wind tunnel showing support structure
Fig. 3.3 Wind Tunnel South Wall showing static pressure holes
Fig. 3.4 Wind Tunnel north wall showing hot wire probe access holes
Figure 3.5 Wind tunnel layout
Figure 3.6 View of Wind tunnel from the axial fan discharge end
array of pressure transducers
wind tunnel

airflow direction

pitot static tube
or hot wire anemometer

from pressure transducers
interfacing computer
with data acquisition card

digital micromanometer

electronic computerized traversing ruler

Fig. 3.7 Arrangement of instrumentation
Fig. 3.8 Axial static pressure variation for two flow settings for the working section of the wind tunnel.

Fig. 3.9 Wind tunnel inlet velocity variation with time.
Fig. 3.10 Velocity profiles at Re = 219560
Fig. 3.11 Velocity profiles at Re = 427 253
Fig. 3.12 Pressure drop-Reynolds number plot for wind tunnel contraction piece
Fig. 3.13 Jet fan simulation mechanism
Figure 3.14 Photograph showing jet fan simulation arrangement.
CHAPTER FOUR

DATA REDUCTION AND ANALYSIS

An explanation of the analysis and processing of experimental results is given in this chapter. The primary data is essentially pressure and velocity measurements which are used to determine secondary parameters of interest in the analysis of the jet fan jet flow field.

4.1 Analysis of Pressure Results

Axial static pressure measurements have been plotted as a dimensionless pressure or in normalised form in order to make comparison easier from one set of measurements to another. The axial static pressures are presented as \((P - P_e)/(0.5pU_j^2)\) and are plotted against distance in jet nozzle diameters \(X/D_j\). \(P_e\) is the pressure of the secondary stream at entry in the tunnel. The pattern of static pressure variation is compared for both tunnel walls in order to assess differences. A few static pressure results are also presented where the tunnel had a coflow velocity which was obtained by running the main axial flow fan at the discharge end. The jet fan then discharged into an already moving tunnel air stream. The purpose of this analysis was to determine the pattern of pressure variation in a strong tunnel air stream which is not initially entrained by the jet fan from outside. Therefore the effect of different jet fan to tunnel Reynolds number ratio can be examined.

The data presented is for two sizes of jet fan to tunnel diameter ratio \(D_R\) and three jet discharge Reynolds numbers or three jet outlet velocities \(U_j\). The jet position from one tunnel wall is characterised by a parameter \(F_p\), which can be defined as the distance from
the tunnel wall divided by the tunnel diameter. This parameter is very important in determining the performance of the jet fan in this study and has been plotted against the pressure ratio \( \frac{P - P_e}{0.5 \rho U_j^2} \). This pressure ratio is used to define a performance parameter also plotted against \( F_p \). The measured pressure rise \( \frac{P_{in} - P_e}{th} \) was plotted as a ratio of the theoretical pressure rise \( \frac{P_{in} - P_e}{th} \). This was necessary in order to assess the momentum losses of the system.

4.2 Analysis of Velocity Readings

The distribution of radial velocity of the jet fan - tunnel system at various axial locations was necessary in order to determine the development of the flow throughout the entire tunnel. For each jet fan diameter ratio \( D_R \) the velocity \( U/U_j \) was plotted against the distance across the tunnel \( Y/D_t \), i.e. the velocity was normalised using the jet discharge velocity and the distance across the tunnel \( Y \) was normalised using the tunnel diameter \( D_t \) for all the profiles at various axial locations. Several distribution plots were made for the jet position parameter \( F_p \).

Since the flow is confined the jet fan flow field falls into the category of confined jets where self similarity of the flow is not usually achieved. It was necessary to check this by plotting the velocity \( U/U_{max} \) against the distance across the tunnel \( Y/D_t \) for selected values. When the profiles are self similar they fall under the same curve when plotted this way.

A complete set of velocity profiles at \( X/D_j=45.3 \) for \( D_R=0.11 \) and \( X/D_j=30.2 \) for the \( D_R=0.17 \) is presented in order to (i) assess the flow symmetry of the tunnel and (ii) determine the tunnel mean velocity which can then be used to determine the volume flow.
rate. The velocity profiles are normalised using the tunnel bulk velocity $U_B$ and are plotted against $Y/D_i$ at various tunnel heights $Z/D_i$.

### 4.2.1 Jet Axis Velocity Decay Profile

These velocities are taken at the jet axis and plotted as $U_m / U_j$ against dimensionless axial distance $X/D_j$. These velocities show how the jet discharge velocity declines from its initial high value to approximately the average tunnel mean velocity at some point at the end of mixing between the primary and secondary stream. The jet axis velocity decline is compared for the two jet fan diameters. The theoretical jet axis decay curve

$$\frac{U_m}{U_j} = 6 \frac{D_j}{X}$$  \hspace{1cm} (4.1)

for an axisymmetric jet is also plotted on the same graph as the measured values in order to compare the two data sets.

### 4.3 Backflow Analysis and Jet Expansion Angle Determination

It was necessary to plot the backflow velocity distribution independently for the various jet position parameters $F_p$ and diameter ratios $D_R$. Complete cross sectional profiles of the backflow velocity distribution have been obtained in order to provide a full assessment tunnel height velocity distribution. The backflow $U_R$ is normalised by the backflow bulk velocity $U_{Br}$.

The backflow quantity $Q_R \ (m^3/s)$ has been determined from the velocity measurements for each cross section and at all the jet fan position parameters $F_p$. The quantity $Q_R/Q_T$ has been plotted against axial distance $X/D_j$. The average backflow quantity $Q_R$ normalised by
Q_T and Q_j for each jet fan position has been plotted against the parameter F_p. It was very useful to be able to compare the amount of air recirculated as a fraction of the total tunnel flow or jet discharge for the assessment of system energy losses.

For each jet fan - tunnel arrangement the backflow field has been characterised completely. One variable of interest was the width Y_w/D_t of the backflow at each axial location of the tunnel which has also been presented graphically as a function of X/D_j. The backflow length L_R/D_j has been plotted against the jet fan position parameter F_p.

4.3.1 Jet Expansion Angle Determination

The jet expansion angle of the jet fan has been estimated from the measured results and primarily from knowledge of the backflow length L_R. An assumption is made that the jet from the jet fan expands like a free jet until it reaches the walls of the tunnel. At this point some of the air stream is recirculated back to the entry region of the tunnel. By estimating this length from the jet nozzle discharge the jet expansion angle can be determined with a reasonable accuracy. A complete account of how the jet expansion angles have been determined in this study is detailed in Chapter Seven dealing with theoretical considerations. The half jet expansion angle is plotted against the jet fan position parameter F_p for both diameter ratios D_R=0.11 and 0.17.

4.4 Entrainment Results

The entrainment results are derived from velocity measurements across the entire tunnel cross section where backflow was totally absent. The total tunnel airflow quantity Q_T (m3/s) is determined from knowledge of the mean velocity. Q_T/Q_j has been plotted against the jet fan position parameter F_p in order to compare the entrainment characteristics of
the jet fan at various positions from the wall. The amount entrained flow $Q_e$ has been plotted as a fraction of the jet discharge quantity $Q_j$. This parameter is generally regarded as the flow ratio $n$ in this study since it expresses the ratio between the secondary and the primary flow.

**4.5 Jet Fan Performance**

The jet fan performance has been plotted as a performance parameter $\eta$ derived from pressure results by multiplying the pressure ratio $(P_m-P_e)/(P_j-P_m)$ by the flow ratio $n$. The $\eta$ parameter is plotted against the jet fan position parameter $F_p$. Another performance parameter termed the induction efficiency

$$\eta_i = \left( \frac{2\Phi}{1-\Omega} \right) \left( \frac{1-\Phi}{1+\Phi} \right)$$

(4.2)

has also been plotted for comparison purposes. The symbols $\Phi$ and $\Omega$ are the velocity and area ratios $U_j/U_i$ and $A_j/A_i$ respectively.

**4.7 Longitudinal Velocity Fluctuations**

The velocity fluctuations measured at the tunnel axis are plotted and normalised by the tunnel axis velocity or local centreline velocity. The tunnel axis is not necessarily the jet axis in this case. The purpose of these measurements was to give an indication of the longitudinal turbulence in the tunnel with the aim of gaining a deeper understanding of the flow structure and its mechanisms.
4.8 Uncertainty analysis in measured and derived quantities

Detailed guidelines to the determination of experimental design and uncertainty are described by Coleman and Steele (1989), Holman (1984), Moffat (1988) and are also given in the ANSI/ASME PTC.

The methods recommended are that the precision and the bias limit be determined in order to determine the overall uncertainty of a result. The bias error is the fixed, systematic or constant entity of the total error. When the true bias error is defined as $\beta$, the quantity $B$ is the experiment's 95 percent confidence estimate such that $|\beta| \leq B$. The precision error $\pm \varepsilon$ is the random component of the total error and is the lack of repeatability of the result. The $\pm \varepsilon$ interval about a result (single or averaged) is the experiment's 95 percent confidence estimate of the band within which the mean of many such results would fall, if the experiment was repeated many times under the same conditions and using the same equipment.

The $\pm \sigma$ interval about the result is the band within which the experimental result is at 95 percent confidence level of the true value. The 95 percent confidence uncertainty $\sigma$ is calculated from the root-sum square (RSS)

$$\sigma = \left[ B^2 + \varepsilon^2 \right]^{0.5} \quad (4.3)$$

Bias limits from $M$ elemental error sources can be combined using RSS as

$$B_J = \left[ \sum_{k=1}^{M} (B_{j_k})^2 \right]^{0.5} \quad (4.4)$$

For the data reduction equation

$$r = r(X_1, X_2, \ldots, X_J) \quad (4.5)$$
the bias limit for the experimental result is found from the uncertainty analysis expression

\[ B_r = \left[ \sum_{i=1}^{J} \theta_i^2 B_i^2 \right]^{0.5} \]  \hspace{1cm} (4.5)

\[ \theta_i = \frac{\partial r}{\partial X_i} \]  \hspace{1cm} (4.6)

Similar expressions can be written for the precision error \( \varepsilon \) and the overall estimate of the uncertain \( \sigma \) can be calculated from the uncertainty equation which combines both the bias and precision limit for each result.

\[ \sigma_r = \left[ \left( \frac{\partial r}{\partial X_1} - \sigma_{x_1} \right)^2 + \left( \frac{\partial r}{\partial X_2} - \sigma_{x_2} \right)^2 + \ldots + \left( \frac{\partial r}{\partial X_j} - \sigma_{x_j} \right)^2 \right]^{0.5} \]  \hspace{1cm} (4.7)

Tabulated experimental uncertainties based on the above analysis have been given in the appendix for a representative portion of the data.
CHAPTER FIVE

DISCUSSION OF PRESSURE FIELD RESULTS

This chapter and the next present a detailed discussion of the results obtained in this investigation. Jet fan ventilation results are described by the pressure and flow field measurements and their subsequent analysis. The importance of the pressure results is twofold. (i) The pressure field results help in determining the entrainment rates of the jet flow field of the fan. (ii) From the pressure rise characteristic of the jet fan, power calculations can be performed and the effectiveness of the system determined. The main variables are (i) the diameter of the jet fan nozzle in relation to the tunnel diameter ($D_j/D_t$ or $D_R$), (ii) the jet discharge velocity ($U_j$) and (iii) the influence of the jet fan positioning from the walls ($Y/D_t$ or $F_P$) on the measured quantities. It will be shown that these variables play a significant role in determining jet fan performance.

5.1 Effect of Jet Fan Position $F_p$ on Axial Pressure Development

5.1.1 Pressure results for $D_R=0.11$ Without Tunnel Coflow

The axial static pressure results have been normalised by the dynamic pressure of the jet fan in most cases. Two values of jet fan to tunnel diameter ratio $D_R$ were used: 0.11 and 0.17 with jet fan discharge velocities of 40 and 21.4 m/s respectively. In this section the axial static pressure results are discussed for the diameter ratio of 0.11 ($U_j = 40$m/s) and these are presented in Figures 5.1 to 5.8.

Figure 5.1 shows the axial static variation for three fan positions ($F_p$) of 0.06, 0.11 and 0.17. The position $F_p=0.06$ is closest to the wall. The axial static pressure varies in the
same way at these three jet fan positions. At the point of the jet discharge the static pressure is zero i.e. it is at the atmospheric or ambient level and at this stage the walls of the tunnel do not affect the jet pressure field. Initially the pressure falls slightly and then rises within eight to ten jet fan nozzle diameters downstream, due to the influence of the tunnel wall, when the jet flow is expanding. When the jet reaches the wall the pressure once again falls due to inward deflection of the jet flow away from the wall. Minimum pressure which is below atmospheric is attained at just over 25 nozzle diameters downstream in all three cases. The respective minimum pressures are -0.005, -0.007 and -0.009 of the jet discharge dynamic pressure for the three positions $F_p=0.06$, 0.11 and 0.17.

After the minimum is reached the jet flow pressure rises monotonically until a maximum of about 0.016 jet dynamic pressure is achieved, after which it remains unchanged over the entire test section. At this point the jet is thought to have reached both tunnel walls and the flow develops in the same way as ordinary pipe flow.

Figure 5.2 shows pressure results for the jet fan position parameter $F_p=0.22$, 0.28, 0.33, 0.39, 0.44 and 0.5. These are presented together because they follow similar trends which differ from the positions close to the wall shown in Figure 5.1. In these positions the pressure continues to drop from the jet discharge until a minimum of about $-0.003\frac{1}{2}\rho U_j^2$ is reached in all cases at twenty jet nozzle diameters. After the minimum is reached the pressure rises monotonically until a maximum is reached as in the previous cases. It is interesting that in the first twenty nozzle diameters the pressure variation for all these positions is very close both qualitatively and quantitatively.

Considering the pressure variation in both Figures 5.1 and 5.2 it is observed that the pressure drops below ambient level before rising to a maximum value. The initial pressure
drop is responsible for inducing secondary flow into the tunnel. The secondary flow then mixes with the jet primary stream and causes the static pressure to rise to a certain level. The closer the jet fan is to the wall the greater the initial pressure drop on one side of the tunnel.

In Figures 5.3 to 5.8 the same pressure results are presented but this time showing a comparison of the variation from both sides of the tunnel. The main feature of this comparison is that although the variation pattern is very similar on both sides of the tunnel, the pressure rise is much steeper on the side of the tunnel for which $F_p \geq 0.5$. Figures 5.3 to 5.8 also show that the static pressure is not symmetrical on both sides of the tunnel. In the first twenty diameters in Figures 5.3 and 5.5 for $F_p \geq 0.5$ the pressure drop variation does not strictly follow that for $F_p \leq 0.17$. This is to be expected in this case because of the differences of the jet fan location from either wall. The fan position parameter $F_p$ is defined from both walls in Figures 5.3 to 5.12. For example the position $F_p=0.11$ from one wall is $F_p=0.89$ from the other wall. When the jet fan is moved towards the axis of the tunnel the pressure magnitude is almost the same in the first twenty nozzle diameters downstream as can be seen in Figures 5.7 and 5.8.

**5.1.2 Pressure Results for $D_R=0.17$ and $U_j=21.4$ (Re=21165)**

The results for the jet fan to tunnel diameter ratio $D_R=0.17$ are presented in Figure 5.9 for four jet fan positions $F_p$ of 0.083, 0.17, 0.33 and 0.5. The purpose of these results was to establish the effect of having a larger fan discharge diameter with a lower discharge outlet velocity but keeping the outlet mass flow rate approximately the same. The axial static pressure variation is similar to the variation for the $D_R=0.11$ ($U_j=40\text{m/s}$) case for similar values of jet fan position from the wall $F_p$. When the larger diameter jet fan is nearest to the wall the pressure initially rises above ambient and declines to negative values before it
begins to rise but the pressure drops are much less than those reported in Figures 5.1 and 5.2. The farther the jet fan is moved from the wall the faster the static pressure rises to reach a constant value over the measuring test section.

Comparing Figures 5.1 and 5.9 it can be seen that the larger the jet fan diameter the more rapid the pressure field develops in the tunnel. For $F_p \geq 0.33$ the pressure variation is effectively the same, and it reaches a minimum in this case at about 10 jet fan nozzle diameters compared to 15 and 18 nozzle diameters for the positions $F_p$ of 0.08 and 0.17 respectively. An important feature of Figure 5.9 is that the normalised static pressures have a higher magnitude than in Figure 5.1 where the jet fan diameter is smaller and the outlet velocity is higher. It can be deduced from the two figures that for approximately the same jet discharge mass flow rate, larger diameter jet fans have a higher jet pressure utilisation factor than smaller diameter fans.

5.2 Comparison of Pressure Variation for Two Differing Jet Fan Discharge Velocities

Figures 5.10 and 5.11 compare axial pressure variation for a jet fan of the same diameter positioned at the same distance from one tunnel wall with two outlet discharge velocities. In the first case the jet fan is positioned close to the wall $F_p = 0.06$ and in the other case it is situated further away from the wall at $F_p = 0.33$. The most important point to make is that the normalised pressure values stay almost the same up to the point where the jet with a discharge Reynolds number of 13714 reaches its maximum value. The fully developed normalised pressure for the Re=26374 case ($U_j=40\text{m/s}$) is about one and half times higher than that for Re=13714 ($U_j=20.8\text{m/s}$). In Figure 5.10 results are also presented for jet fan position $F_p = 0.83$ which compares the pressure variation from the other side of the
tunnel with that for the position $F_p$ of 0.06 for the two jet discharge Reynolds numbers. It can be seen that the pressure variation is not symmetrical for both tunnel side walls.

For any jet fan discharge mass flow rate the parameter which controls the manner in which the axial static pressure varies is the jet fan position relative to the confining tunnel walls rather than the discharge velocity or Reynolds number. Figures 5.10 and 5.11 clearly show this fact.

5.3 Comparison of Pressure Variation for Two Jet Fan to Tunnel Diameter Ratios

In Figure 5.12 pressure results for two jet fan to tunnel diameter ratios of $D_R=0.11$ and 0.17 are presented. The average jet discharge flow is approximately the same in both cases but the outlet velocities are 21.4 and 40 m/s for the larger and smaller diameter jet fan respectively. Qualitatively the pressure varies in the same way. For the larger diameter jet fan the pressure develops rapidly axially and reaches a peak at shorter distances. The magnitude of the normalised pressure is also higher in this case. The minimum normalised pressures are lower than for the smaller diameter jet fan. For the smaller diameter jet fan the magnitude of the pressure change from jet discharge to minimum and then to maximum is more gradual; and lower peaks are reached at larger downstream distances than for the larger diameter jet fan. After the maximum values are reached in the tunnel the pressure gradually falls downstream to ambient level again. In both situations the pressure develops more rapidly when the jet fan is positioned at the tunnel axis ($F_p=0.5$) and than when it located near the wall ($F_p=0.06$ and 0.08)
5.4 Axial Static Pressure Variation in the Presence of a Strong Tunnel Coflow

In a number of situations jet fans are used to boost the pressure of an existing flow in a tunnel or mine airway. The pressure rise characteristic in this case is of interest in the ventilation design of the jet fan and airway or tunnel system because in this case the fan derives its airflow from the airway and gives it a momentum boost.

Figure 5.13 shows axial static pressure plots for a jet fan to tunnel diameter ratio $D_r=0.11$ and initial velocity ratio $U_i/U_j=0.09$. The tunnel mean velocity was 3.6 m/s before the jet flow was introduced. In this case the coflow velocity is considered to be very strong. The pressure varies in a very interesting way for both jet fan positions $F_p=0.06$ and 0.5. In the situation when the jet fan is located close to the wall ($F_p=0.06$) the pressure rises and falls in the first 10 jet nozzle diameters and then remains below ambient for most of the tunnel. At about 50 nozzle diameters it starts to rise slowly and at $X/D_j=70$ it is still rising and this time it is well above ambient with a value of about 0.3% of the jet discharge dynamic pressure. When the jet fan is located at the tunnel axis ($F_p=0.5$) the pressure starts from well below ambient with a value of -0.0075 of the jet discharge pressure. The pressure gradient is positive right from the start. The pressure rises above ambient at about $X/D_j=35$ and at $X/D_j=70$ the pressure is still rising with a value of 0.065% of the jet discharge diameter. When Figures 5.1, 5.2 and 5.13 are compared it can be seen that the effect of tunnel coflow is to stretch the pressure variation and therefore reduce the steepness of the pressure gradients in all cases. In Figures 5.1 and 5.2 there is no initial tunnel flow i.e. zero coflow velocity and it is observed that the pressure rise is much steeper and the pressure reaches a maximum value much earlier than in Figure 5.13 where there is strong tunnel coflow. Therefore the effect of tunnel coflow is to retard the development of the jet flow which explains why the pressure develops more slowly.
In Figure 5.14 results are presented for a stronger coflow of velocity ratio \(U/U_j = 0.15\) and for the jet diameter ratio \(D_R = 0.17\) for four jet fan positions. In this case the pressure rises more slowly than for the case in Figure 5.13. The tunnel pressure variation for an average velocity of 3 m/s, without a jet fan is also shown and this shows no change over the entire test section. At some point the jet fan will have no marked effect on the secondary (tunnel) stream for a sufficiently high coflow velocity. Thus jet fans are useful where there is weak or no coflow in the confining space. In Figure 5.15 a comparison is made for two sets of pressure results for a jet fan with and without coflow for a diameter ratio of \(D_R = 0.17\). This graph shows that the rate of pressure rise is steeper with no coflow than when there is a coflow. The importance of these results is that they are a useful guideline in ventilation design for a jet fan used to boost airway pressure in a mine or tunnel.

5.5 Pressure Ratio as a Function of Jet Fan position \((P_p)\) Inside the Tunnel

Figures 5.16 and 5.17 show the measured to theoretical tunnel end pressure ratio \((P_{in} - P_e)_{exp} / (P_{in} - P_e)\) as a function of jet fan position from the tunnel wall. This pressure is the maximum pressure rise achieved over the tunnel test section. The theoretical pressure \((P_{in} - P_e)_{th}\) is determined from the following equation:

\[
P_{in} - P_e = \alpha \rho U_j^2 (1 - \alpha n)^2
\]

Equation 5.1 is derived from momentum balance considerations of the jet flow \(m_j U_j\), secondary or entrained flow \(m_e U_e\) and total tunnel flow momentum \(m_f U_f\) and expressed as:
\[ m_j U_j + m_e U_e = m_T U_T + \Delta p A_t \quad (5.2) \]

The symbols \( U_j, \alpha, n \) and \( \rho \) represent the jet fan discharge velocity, jet fan outlet to secondary flow entry area ratio, flow ratio and air density respectively. The last term in equation 5.2 \( \Delta p A_t \) is the momentum due to tunnel pressure forces. The assumption made in the derivation of the above equations is that there are no wall friction losses, thus equation 5.1 represent ideal conditions. From the ratio \( \frac{(P_m - P_e)_{\text{exp}}}{(P_m - P_e)} \), the losses due to wall friction can be estimated. In Figure 5.16 it is very clear that walls have marked effect on the pressure rise due to excessive friction. When the jet fan is situated close to the wall it is observed that 15\% of the momentum is lost to friction in this investigation. Mizuno and Araie (1990) also observed that as much as 15\% of the momentum is lost due to friction when the jet flow is close to the wall. The percentage momentum loss when the jet fan is close to the wall can vary depending on the roughness of the walls and the jet flow Reynolds number. The results shown in Figure 5.16 are for the jet fan-tunnel diameter ratio \( D_R = 0.11 \) and a jet fan discharge velocity of 40 m/s.

As the jet fan is moved towards the tunnel centre the momentum loss due to friction loss is reduced. At the tunnel centre \( (F_p = 0.5) \) the difference between measured and theoretical pressure rise is about 9\%. In Figure 5.17 for the larger jet fan-tunnel diameter ratio \( D_R = 0.17 \) and lower jet discharge velocity \( U_j = 21.4 \) m/s the percentage loss due to friction when the jet fan is close to the wall is about 9.5\%. As the jet fan is traversed towards the tunnel centre, the pressure loss due to friction diminishes much faster than in the previous case. At the jet fan location \( F_p = 0.25 \) the measured pressure rise is almost the same at the theoretical one derived from equation 5.1. However at jet fan position \( F_p = 0.33 \) the measured pressure falls sharply to only about 94\% of the theoretical one.
i.e. the losses due to friction increase to 6%. This value is reduced to about 1% as the jet fan is moved closer to the tunnel centre position.

From Figures 5.16 and 5.17 it can be said that pressure losses are less for larger diameter jet fans with lower outlet velocities. Thus the selection of an optimum position for an effective system depends on the physical parameters of the jet fan such as outlet area and velocity. The width and height of the opening or tunnel should be of a certain size which reflect the jet fan flow and geometrical conditions.

In Figures 5.16 and 5.17 a second pressure ratio is also plotted. This pressure ratio defines the pressure rise of the tunnel airstream to the pressure drop by the jet flow \((P_m - P_e)/(P_j - P_m)\). The symbols \(P_m\), \(P_e\) and \(P_j\) are the tunnel end, secondary stream and jet fan discharge pressure. Jet fan positions closer to the wall have lower pressure ratios mainly because of friction. In Figure 5.16 at the position closest to the wall \((F_p = 0.06)\) the pressure rise is 1.7% of the jet discharge pressure drop and when the jet fan is at the tunnel centre \((F_p = 0.5)\) the pressure rise is 1.9%. The variation follows that of experimental to theoretical pressure rise plotted on the same figure. In Figure 5.17 the ratio of the pressure rise to jet total pressure drop is higher than that shown in Figure 5.16 for \(DR=0.11\) and \(U_j=40m/s\). In Figure 5.17 the diameter ratio \(D_R=0.17\) and jet discharge velocity is 21.4 m/s. The deduction that larger diameter jet fans with a lower discharge velocity have a higher pressure utilisation capability is a reasonable one.
Fig. 5.1 Axial pressure variation (Fp=0.06 to 0.17)

Fig. 5.2 Axial pressure variation (Fp=0.22 to 0.5)
Fig. 5.3 Axial static pressure variation on tunnel side walls (Fp=0.06 and 0.94)
(DR=0.11 Re=26374 Uj=40m/s)

Fig. 5.4 Axial static pressure variation on tunnel side walls (Fp=0.11 and 0.89)
(DR=0.11 Re=26374 Uj=40m/s)
Fig. 5.5 Axial static pressure variation on the tunnel walls (Fp=0.17 and 0.83)

Fig. 5.6 Axial static pressure variation on the tunnel walls (Fp=0.22 and 0.78)
Fig. 5.7 Axial static pressure variation on the tunnel walls (Fp=0.44 and 0.56)

Fp=0.44 △ Fp=0.56 DR=0.11 Re=26374 Uj=40 m/s

Fig. 5.8 Axial static pressure variation on both sides of the tunnel

Fp=0.5 - North wall ○ South wall DR=0.11 Re=26374 Uj=40 m/s
Fig. 5.9 Axial static pressure variation vs axial distance for jet fan DR=0.17
Fig. 5.10 Static pressure variation of jet fan for two jet Reynolds numbers
(Fp=0.17)

Fig. 5.11 Static pressure variation of jet fan for two jet Reynolds numbers
(Fp=0.33)
Fig. 5.12 Comparison of pressure variation for two jet fan diameter ratios

Fig. 5.13 Pressure variation for jet fan with tunnel coflow
Tunnel at $U_t=3\text{m/s}$ without jet fan

Fig. 5.14 Pressure variation with tunnel coflow $U_t=3\text{m/s}$

Fig. 5.15 Comparison of pressure variation with and without tunnel coflow
Fig. 5.16 Plot of various pressure ratios vs jet fan position Fp

DR=0.11 Uj=40m/s

Fig. 5.17 Plot of various pressure ratios vs jet fan position Fp

DR=0.17 Uj=21.4m/s
CHAPTER SIX

DISCUSSION OF THE JET FAN VELOCITY FIELD DEVELOPMENT

Velocity results are presented in this chapter. The velocity field development is important in evaluating the airflow or aerodynamic characteristics of the jet fan inside the tunnel. The velocity profiles were taken at half the distance from the floor of the wind tunnel to the roof i.e. at half the tunnel diameter and measured from one side wall to the other. The jet fan was traversed across the cross section of the tunnel and the results presented are for four jet fan positions for two diameter ratios and jet discharge velocities.

6.1 Velocity Distribution for Jet Fan to Tunnel Diameter Ratio DR=0.11 and \(U_j=40\) m/s

Figure 6.1 shows the normalised velocity distribution \((U/U_j)\) for a jet fan (\(D_R=0.11\), \(U_j=40\) m/s) at four tunnel positions \(F_p=0.06,0.17,0.33\) and 0.5. \(F_p\) is the distance of the jet fan from the wall normalised by the tunnel diameter. The profiles presented start from \(X/D_j=6.35\) up to 32.7. The velocity profile development is affected by the presence of the tunnel walls. At the position \(F_p=0.06\) (Figure 6.1a) the velocity profiles have a common feature in that the velocities are positive for a part of the tunnel cross section and negative on the other. The jet flow field is close to one tunnel wall and shows lack of symmetry. Its closeness to one tunnel wall limits the way it spreads with axial distance. On the other side of the tunnel the backflow covers a large area and by 45.3 jet fan outlet diameters the reverse flow has disappeared. When all the velocity profiles are compared from Figure 6.1a to 6.1d it is found that the jet flow spreads less rapidly when the jet fan is close to the tunnel wall than away from it. As the jet fan is moved towards the tunnel axis the size of the backflow decreases. A much more detailed analysis of the backflow will be
given in chapter eight. When the jet fan is at the tunnel axis the flow is symmetrical and the flow field is observed to oscillate with periods varying between two to four seconds. This takes places in wall regions up to 0.17 tunnel diameters from the wall. The flow changes in direction in this region from positive to negative and this extends from a short distance downstream of the jet discharge to nearly 32.7 jet fan outlet diameters.

In Figure 6.3 the same velocity profiles are nominalised using $U_{\text{max}}$ and it is clear that the flow does not vary smoothly across the tunnel at most of the measurement stations mainly due to the high turbulence intensity inside the tunnel. Even at $X / D_j = 45.3$ for the jet fan position $F_p = 0.06$ the flow is still skewed on one side of the tunnel. It seems it would take a longer distance from the jet fan for the flow to be uniform. The backflow is shown clearly in this Figure. It is apparent that the flow in this setting does not approach self similarity which is not surprising for a confined jet, unlike a free jet.

Figure 6.5 shows a complete velocity distribution for the entire tunnel cross section at $X / D_j = 45.3$ for three jet fan positions $F_p$ of 0.11, 0.33 and 0.5. The velocity profiles are normalised by the local bulk velocity $U_B$ and are measured at fractional height of the wind tunnel, $Z/D_t$ of 0.17, 0.33, 0.5, 0.67 and 0.83. At this axial location there is no reverse flow. However the velocity distribution is not evenly distributed for the jet fan position away from the tunnel axis as shown in Figures 6.5a and 6.6b. The turbulence of the flow field at this point contributes to the large scatter observed in Figures 6.5a and 6.5b. In Figure 6.5c the flow is effectively symmetrical.
6.2 Velocity Distribution for Jet Fan to Tunnel Diameter Ratio $D_{r}=0.17$ and $U_{j}=21.4$ m/s

Velocity distribution ($U / U_{j}$) for the larger jet fan shown in Figure 6.2 is very similar to the previous case (Figure 6.1). When the jet fan is not located at the axis of the tunnel the flow develops on one side with a significant reverse flow covering a significant part of the tunnel. The reverse flow area is not as large as in the smaller diameter, higher velocity jet fan. The flow develops faster than in the smaller diameter jet fan and by $X / D_{j} = 21.8$ the jet flow has spread to cover the entire cross section of the tunnel. When the jet fan is located at the axis of the tunnel the velocity distribution is symmetrical. The oscillation observed for the smaller diameter, and higher velocity jet fan is not as strong in this case.

Velocity profiles ($U / U_{max}$) are presented in Figure 6.4. They are similar to the smaller diameter jet fan velocity profiles and they show no self similarity. These results also show the presence of backflow and the rapid development of the jet flow at positions closer to the tunnel axis. The velocity distribution at $X / D_{j} = 21.8$ for the jet fan located at the tunnel axis ($F_p = 0.5$) shown in Figure 6.4d is completely evenly distributed suggesting the jet flow is totally mixed with the secondary stream.

Velocity profiles at $X / D_{j} = 30.2$ normalised by the local bulk velocity presented in Figure 6.6 are similar to the profiles shown in Figure 6.5 (for the smaller diameter jet fan). At jet fan positions closer to the wall the velocity distribution remains skewed at downstream points. The profile in Figure 6.6c is very similar to that shown in Figure 6.5c since they are for the same jet fan position i.e. the tunnel axis location and they are reasonably symmetrical.
6.3 Jet Axis Velocity Decay Inside Tunnel for a Jet Fan at Various Positions

The normalised centreline velocity $\frac{U_m}{U_j}$ decay of a confined jet fan in the tunnel is shown in Figures 6.7 to 6.9. These results are for two jet fan to tunnel diameter ratios and for various tunnel jet fan locations. Figure 6.7 shows that the centreline velocity of a jet fan located closer to the wall is slower in its decrease than when the jet fan is positioned at the tunnel centre after distances $X/D_j \geq 5$. Results shown in this figure are for the diameter ratio $D_R = 0.17$ and $U_j = 21.4$ m/s. At the jet fan position $F_p=0.08$ the centreline velocity after 22 jet outlet diameters is greater than at the other positions shown on the figure. At this distance the centreline velocity is less than 15% of the initial jet fan discharge velocity for all the jet fan positions in the tunnel. The velocity decays rapidly in the first 15 jet fan outlet diameters to less than 50% of the initial jet discharge velocity.

For the purpose of comparison the axial centreline velocity decay profile of an axisymmetric free jet derived from the formula

$$\frac{U_m}{U_j} = 6 \frac{D_j}{X}$$  \hspace{1cm} (6.1)

is plotted on the same figure. In the above equation $D_j$ is the free jet outlet diameter equal to the jet fan diameter and $X$ is the axial distance from the jet outlet. The centreline velocity decline for a free jet is lower and more gradual than that for a confined jet fan. At about $X/D_j = 22$, the velocity for an axisymmetric jet is about six times higher than that for a jet fan located at the tunnel axis. The difference is caused by fluid friction, mixing and reverse flow losses in the main tunnel flow.
Figure 6.8 shows the centreline velocity decline for a lower diameter ratio $D_r = 0.11$ and higher jet fan discharge velocity $U_j = 40$ m/s. The same trends are observed as for the larger diameter ratio jet fan but in this case the velocity decline is slower. Generally the decline of centreline velocity is slower as the jet is moved closer to the wall. In both cases the axisymmetric jet declines more gradually and smoothly than for a confined jet fan. Results for the two jet fan diameter ratios are plotted together in Figure 6.9 where it is clear that the centreline velocity decline is faster for the larger diameter jet fan. A steep decline of the jet fan flow means that the flow is spreading and mixing very rapidly. This causes the jet flow to slow down and at distances much further downstream the final flow velocity is less than 2.5 % of the initial jet discharge velocity $U_j$. Most of the energy lost by the jet fan to mixing is within the first 20 jet outlet diameters.

6.4 Jet Expansion Angle and Reverse Flow Phenomena

6.4.1 Jet Expansion Half Angles

The jet expansion angle of the jet fan flow is a useful property which enables the distance of confining space covered by the jet during its expansion before it impinges on the wall to be determined. Information concerning reverse flow or recirculation of some of the flow stream can be estimated from this distance. The estimation of jet expansion angle has been described in chapter four. The assumption made was that the jet from the fan expands as in a free jet until it reaches the wall. In this study the distance it takes for the jet to reach the wall has been estimated by assuming that the axial static pressure at that point reaches a maximum and therefore the distance can be determined by measurement. The other method used in this study was to assume that the point where reverse flow begins downstream in the tunnel is where approximately the jet reaches the wall. Experience from this study has shown these to be reasonable assumptions.
Figure 6.10 shows jet expansion half angle for the jet fan with $D_R = 0.11$ and $U_j = 40$ m/s for various positions in the tunnel. The average jet expansion half angle $\theta$ determined for this flow setting is $12.43^\circ$. The minimum half angle determined is $10.5^\circ$ when the jet fan is located at the tunnel axis. The maximum half angle obtained is about $14^\circ$ when the jet fan is at position $F_p = 0.33$. For every flow condition i.e. jet fan Reynolds number or outlet velocity, jet fan outlet diameter and configuration, the jet expansion angle should be independent of jet fan position inside the tunnel. Thus a jet fan expansion half angle of $12.43^\circ \pm 1.5^\circ$ for this flow condition is a reasonable estimate. The differences in jet expansion half angle shown in Figure 6.10 for the jet fan at various positions inside the tunnel are due to the uncertainty in determining the point where the jet reaches the wall. However the jet expansion angle estimation gives reasonable results because the differences at the various positions are not as large.

Figure 6.11 also shows jet expansion half angles for a jet fan with discharge velocity $U_j = 21.4$ m/s and diameter ratio $D_R = 0.17$ for various tunnel positions. The average jet expansion half angle was determined to be $12.9^\circ$ with a maximum and minimum value of about $16.5^\circ$ and $11^\circ$ respectively. Although the average value of $12.9^\circ$ in this case is higher than that shown in Figure 6.10 of $12.43^\circ$ for the smaller diameter jet fan with a higher velocity ratio these values can be considered to be quite close. Large diameter jet fans with a low outlet velocity should have slightly larger jet expansion angles due to the fact that the development length of the resulting jet is shorter than in a high velocity case. The expansion angle of the jet is larger for divergent nozzles than for straight and convergent nozzles.

The results obtained in this study for jet expansion half angles are in reasonable agreement with those reported in the literature. Abramovich (1963) reports a value of $\theta = 13.5^\circ$ for a
circular free jet. Wesely (1984) measured a jet propagation half angle of $14^\circ$ in the uninhibited horizontal direction for a jet fan inside an arch-supported mining roadway. Thimons et al. (1986) obtained values between $12^\circ$ and $14^\circ$ for the jet expansion half angle of a jet fan ventilating a heading 8.5 m wide by 50 m long. The values of these half angles encountered in many practical situations will tend to vary between 9 and $16^\circ$ depending on jet fan configurations and conditions. The results from this study of 12.43$^\circ$ and 12.9$^\circ$ fall within this range. When the jet fan is close to the wall the expansion of the jet is restricted on the side that is close to the wall.

6.4.2 Description of Reverse Flow

Reverse or backflow is a feature of jet fan ventilation which needs special attention in mine ventilation due to the potential hazards that exist in some underground situations where the level of pollutants might be very high. Recirculation of dangerous gases or dust is an undesirable feature in mine ventilation, therefore its onset can best be treated by careful studies.

In this investigation reverse flow was observed when the jet fan was not located at the tunnel axis. In most situations auxiliary fans are placed near side walls of openings particularly for the ventilation of dead end headings. In a heading a well placed jet fan can cause air to flow all the space with an average velocity of 1 to 2 m/s. The jet flow reaches the face of the heading and returns to the last open crosscut using the remainder of the space. Possible recirculation takes place around the intake of the jet fan or at the face if the air supply to the fan intake is inadequate. In this investigation the flow of the jet fan is allowed to discharge to a straight open end of the tunnel. The reverse flow analysis is useful particularly in some mining situations where jet fans are used as pressure boosters.
and in headings to see the jet effect and the interaction of the jet flow with the opening walls.

Figure 6.12 shows typical reverse flow velocity profiles at the jet fan position of \( F_p = 0.11 \) at 6.35 nozzles diameters from the jet fan for a jet fan/tunnel diameter ratio \( D_R = 0.11 \). The profiles are at various heights from the tunnel floor. In Figure 6.12 the profiles are effectively flat and uniform regardless of height from the tunnel floor signifying thin turbulent boundary layers even right from the edge of the interface between the axial flow and the reverse floor. Figure 6.12 is representative of all reverse flow profiles measured at all the axial distances from the jet fan in this study.

Figure 6.13 shows normalised reverse flow velocity profiles \( U_R / U_{BR} \) at various axial distances \( (X / D_j) \) at the normalised height from the tunnel floor \( H \) of 0.5 for the jet fan diameter ratio \( D_R = 0.17 \). \( U_R \) and \( U_{BR} \) are the reverse and bulk reverse flow velocity respectively. The reverse flow covers about two thirds of the tunnel cross section at the position \( F_p = 0.08 \) and 0.17 at \( X / D_j = 4.23 \). At larger axial distances from the jet fan nozzle the cross sectional area occupied by the reverse flow diminishes. The reverse flow originates at the point where the jet flow reaches the wall and extends to the jet fan nozzle. Comparing Figures 6.13 (a) to (c) it can be seen that reverse flow diminishes as the jet fan is moved towards the tunnel axis i.e. at larger values of \( F_p \leq 0.5 \). Thus the magnitude of the reverse flow at \( F_p = 0.33 \) is less than that at \( F_p = 0.08 \) for both sizes of the jet fan diameters investigated.

Results of backflow or reverse flow are also plotted in Figures 6.14 to 6.16 in the form of the width of the reverse flow \( (W_R / D_l) \) at selected stations for both jet fan sizes investigated at various positions from the tunnel walls. \( W_R \) is the width of the backflow
and $D_t$ is the diameter of the tunnel. It is easier to visualize the extent of the backflow by observing Figures 6.14 to 6.16. For each jet fan position from the tunnel wall the backflow width is zero near the jet fan nozzle and reaches a maximum value at 6.35 diameters in the case shown in Figure 6.14. After this maximum is reached the width decreases with axial distance until it is zero at some downstream location. This data confirms that the size of the backflow decreases with jet fan distance from the wall. The trends are similar for the two jet fan sizes. The normalised data in Figure 6.15 for the larger diameter jet fan indicates that the backflow lengths are shorter. In Figure 6.16 data is presented for backflow width obtained for a weak tunnel coflow velocity of 0.5 m/s. Even with a initial tunnel velocity of 0.5 m/s reverse flow is established by the jet flow. The initial tunnel flow has to be strong enough to prevent reverse flow from being established. If this is the case it would not be necessary to use a jet fan. Jet fans should only be used to move air where the surrounding flow is weak or stagnant.

Flow visualization of the backflow was achieved by attaching strings of ribbons to a common thread and this was fixed across the wind tunnel at half the tunnel diameter from the floor. The ribbon arrangement was moved from one axial location to another each time observing and photographing the flow. It worked very well and illustrated the existence of reverse flow qualitatively. Results of this exercise can be seen in Figure 6.17 for various jet fan positions from the wall. Figure 6.17 reinforces the data presented in Figures 6.13 to 6.16. During the testing the ribbons were lifted by the flow in both directions i.e. (i) by the forward moving jet flow and in the other by (ii) reverse flow. This flow phenomena was also recorded by a video camera and was an effective way of analysing the backflow. The flow visualization provided a very convincing way of addressing the backflow issue.
At each jet fan position the length of the reverse flow was estimated. This was plotted in normalised form $L_{R}/D_{j}$ against jet fan position and is shown in Figure 6.18 and 6.19 for the two jet fan sizes respectively. As has already been described the length of the reverse flow decreases with the distance of the jet fan away from the tunnel wall. At the tunnel axis position of the jet fan i.e $F_{p} = 0.5$ the reverse length could not be determined but it could be assumed to be zero. The jet flow for the tunnel axis position for the smaller size jet fan was found to produce both positive and negative velocities intermittently within 0.17 tunnel diameters from both side walls in periods of 2 to 4 seconds. This could been caused by the jet failing to attach to the walls upon reaching them and combined with an oscillation of the jet flow field itself. The jet flow oscillation was observed by attaching cotton threads at the jet fan discharge nozzle circumference. In Figure 6.19 the backflow length is largest for jet fan positions close to the wall and decreases sharply for positions away from the wall. It reaches a minimum at the position $F_{p} = 0.33$ and increases slightly at the remaining jet fan position for which backflow was observed. The backflow length was important in the calculation of jet expansion angles and it was also a necessary part of the tunnel flow analysis.

6.4.3 The Quantity of Reverse Flow as Fraction of Jet Discharge and Total Tunnel Flow

Results of the reverse flow as a fraction of the total tunnel flow for a few jet fan positions are presented in Figures 6.20 to 6.22. The amount of recirculated fluid at some specified axial locations is estimated for a few selected jet fan positions as shown in Figures 6.20(a) to (c). In Figure 6.20(a) the normalised reverse flow $Q_{R} / Q_{T}$ is zero at $X / D_{j} = 0$ and then rises to a maximum level of 1.5 for positions $F_{p} = 0.08$ and 0.17, after which it drops to zero at some downstream point for all the jet fan positions $F_{p}$. $Q_{R}$ and $Q_{T}$ are the recirculated quantity and total tunnel flow (m$^{3}$/s) respectively. Results in Figure 6.20(a)
are for a jet fan with 21.4 m/s discharge velocity and diameter ratio $D_R = 0.17$. In Figure 6.20(b) the recirculated fluid fraction is presented for the smaller diameter jet fan with the higher discharge velocity. The quantity $Q_R / Q_T$ varies in the same way as that shown in Figure 6.20(a) with sharper peaks at the maximum level. At the peak of recirculation there is almost twice the amount of tunnel flow in reverse flow. The tunnel jet fan positions of $F_p$ of 0.06, 0.08, 0.17 and 0.33 are representative of the axial variation of the recirculated fluid for both jet fan sizes.

Figure 6.20(c) compares the recirculated quantity $Q_R / Q_T$ for the two jet fan sizes at two positions from the wall. From Figure 6.20 it is clear that there is more recirculated fluid for the smaller diameter jet fan than the larger jet fan for the same discharge mass flow. Figure 6.20 can be better understood with the aid of Figure 6.13 to 6.19 described earlier.

The amount of recirculated fluid as a fraction of the total tunnel flow $Q_R / Q_T$ is plotted against jet fan position $F_p$ in Figure 6.21 for the two jet fan sizes. The quantity $Q_R / Q_T$ varies relatively little with jet fan position where reverse flow exists and is around 0.72 for the smaller diameter jet fan $D_R = 0.11$, and $U_j = 40$ m/s. For the larger diameter ratio jet fan plotted on the same Figure the quantity $Q_R / Q_T$ is around a value of 0.55. There is a drop in $Q_R / Q_T$ from 0.58 to 0.53 from the position $F_p$ of 0.08 to 0.17 before a constant value of 0.55 is reached for jet fan positions further from the tunnel wall.

Figure 6.22 gives the recirculated quantity normalised by the jet fan discharge flow $Q_R / Q_j$ and plotted against jet fan position in the tunnel for both fans. The results from Figure 6.21 are also plotted in Figure 6.22 for the purpose of comparison. The quantity $Q_R / Q_j$ varies from 1.4 to about 1.1 for the fan positions $F_p$ of 0.06 to 0.33 for the smaller diameter setting ($D_R = 0.11$ and $U_j = 40$ m/s). In the larger diameter fan
the quantity \( Q_R / Q_j \) varies from 0.85 down to 0.63 for the jet fan positions \( F_p \) of 0.08 to 0.33. Again this fraction is lower than in the other case. Figure 6.22 shows that when the reverse flow \( Q_R \) is normalised by the total tunnel flow \( Q_T \) it seems to vary relatively little regardless of jet fan position but when it is normalised by \( Q_j \) (the jet fan discharge volume flow) the change is quite noticeable. This observation results from the amount of recirculated air which is higher when the jet fan is close to the wall than away from it. In a larger diameter jet fan with a lower discharge velocity the amount of recirculated air is less than in a smaller jet fan with a higher discharge velocity.

When the total tunnel flow \( Q_T \) is used to normalise the reverse flow \( Q_R \) it is observed that the fraction of recirculated fluid \( Q_R / Q_T \) varies very little with jet fan position in the tunnel for both jet fan sizes. The higher velocity jet fan has a higher proportion of recirculated air than the lower outlet velocity fan for the same mass flow (and different diameters). The reason for this is that the total tunnel flow is different for different jet fan positions. The jet fan entrains more air at some positions closer to the wall than when moved towards the tunnel axis. The amount of recirculated fluid also goes up in proportion to the total tunnel flow. Thus the more secondary air is entrained in the tunnel the more air is recirculated but the ratio \( Q_R / Q_T \) remains almost invariant for a specified jet fan discharge condition. The amount of recirculated air however is greater than the initial jet fan discharge flow and this varies for each jet fan position in the tunnel as shown in Figure 6.22. This recirculation phenomena obtained in this study has never been reported in previous work to the best of the author's knowledge. No previous work has ever produced a flow field for a jet fan traversed from near walls to tunnel axis conditions for through flow situations. A few studies in the literature (e.g. Curtet, 1958) have described confined jet flow and recirculation for centrally positioned ducted jets.
Curtet (1958) developed an approximate theory on confined jets and recirculation phenomena in an attempt to give a full account of the effects of furnace walls and the surrounding environment on turbulent diffusion flames for a jet located on the duct axis \((F_p = 0.5)\). Higher velocities were used (up to 140 m/s) and the amount of secondary flow was controlled. Curtet (1958) observed backflow under these conditions on both walls of the confining duct. The resulting velocity profiles resemble those in the present studies for the centrally positioned jet fan with \(U_j = 40\) m/s and \(D_R = 0.11\) shown in Figure 6.1(d).

The Craya-Curtet parameter \(C_t\) which is similar to Thring and Newby (1953) similitude parameter \(\theta = (Q_j + Q_s)(D_j/2)/\left(Q_j D_t\right)\) defines the possibility of recirculation. The terms \(Q_j\), \(Q_s\), \(D_j\) and \(D_t\) are the jet discharge, secondary stream flow, jet diameter and tunnel diameter respectively. For values of \(C_t = 0.075\) the extent of backflow is significant and decreasing with increase of \(C_t\). According to Barchilon and Curtet (1964) at \(C_t > 0.9\) there is no backflow. In this study values of \(C_t\) are less than 0.25 and recirculation is observed at most jet fan positions \(F_p < 0.44\). The controlling factors for the onset of backflow are (i) the jet fan to tunnel diameter ratio (ii) jet fan position from the wall and the tunnel to jet fan outlet velocity ratio \(U_t / U_j\).
Fig. 6.1 Velocity profiles of jet fan inside tunnel at various positions
Fig. 6.2 Velocity profiles for jet fan inside tunnel DR=0.17

- X/Dj=4.23
- X/Dj=13.87
- X/Dj=21.8

- X/Dj=12.13
- X/Dj=13.87
- X/Dj=21.8

Fig. 6.2 Velocity profiles for jet fan inside tunnel DR=0.17
Fig. 6.3 Plot of $U/U_{\text{max}}$ at various axial locations at different fan positions.
Fig. 6.4 Velocity plot $U/U_{\text{max}}$ at various axial locations

(a) $F_p=0.17$

(b) $F_p=0.5$

(c) $F_p=0.08$

(d) $F_p=0.33$
Fig. 6.5 Tunnel Velocity profiles at X/Dj = 45.3 (jet fan DR=0.11)
Fig. 6.6 Tunnel Velocity profiles at X/Dj = 30.2 (DR=0.17)
Fig. 6.7 Jet axis velocity decay (DR = 0.17, Uj = 21.4 m/s)

Fig. 6.8 Jet axis velocity decay (DR = 0.11, Uj = 40 m/s)
Fig. 6.9 Jet axis velocity decay for two jet fan sizes
DR = 0.11, Uj = 40 m/s
Fig. 6.10 Plot of jet expansion angle vs jet fan position

DR = 0.17, Uj = 21.4 m/s
Fig. 6.11 Plot of jet expansion angle vs jet fan position
Fig. 6.12 Tunnel cross-sectional backflow velocity profile for jet at $X/D_j=18.2$
Fig. 6.13 Plot of backflow velocity profile vs distance across tunnel Y/Dt
Fig. 6.14 Width of backflow at various jet fan positions for $DR=0.11$, $U_j=40$ m/s

Fig. 6.15 Width of backflow at various jet positions for $DR=0.17$, $U_j=21.4$ m/s
Fig. 6.16 Backflow width for jet fan with tunnel coflow velocity of 0.5 m/s
Figure 6.17(a) Flow visualization showing reverse flow ($U_j = 40$ m/s, $D_R = 0.11$)
Figure 6.17(b) Flow visualisation showing reverse flow ($U_j = 21.4 \text{ m/s}, D_R = 0.17$)
Fig. 6.18 Extent of backflow length vs jet fan position Fp

DR=0.11, Uj=40 m/s

Fig. 6.19 Extent of backflow length vs jet fan position Fp

DR=0.17, Uj=21.4 m/s
Fig. 6.20 Qr/QT vs X/Dj for jet fan diameter ratios DR=0.11 and 0.17
Fig. 6.21 Plot of backflow fraction Vs jet fan position

Fig. 6.22 Backflow fraction $\frac{Q_r}{Q_j}$ and $\frac{Q_r}{Q_T}$ Vs jet fan position
CHAPTER SEVEN

DISCUSSION OF JET FAN PERFORMANCE ANALYSIS

7.1 Tunnel Axis Longitudinal Turbulence Levels

Axial turbulence levels \( \sqrt{u'^2 / U_c} \) were measured at the tunnel axis at various downstream locations, for different jet fan positions. The turbulence levels for the jet fan positions other than on the tunnel axis (i.e. \( F_p = 0.5 \)) do not coincide with the axis of the jet flow. These velocity fluctuations were normalised using the local tunnel axis or centreline velocity \( U_c \) and these are presented in Figure 7.1 for the jet fan of diameter ratio \( D_R = 0.11 \) and outlet velocity \( U_j = 40 \) m/s. The importance of the longitudinal turbulence levels was to establish their relationship with entrainment rates in the tunnel and to serve as an aid in the understanding of the structure of the tunnel flow field.

The data presented in Figure 7.1 includes Curtet (1958) and free jet axis turbulence levels. These data are included for comparison purposes. When the jet fan is not positioned at the tunnel axis the turbulence levels obtained at the tunnel axis are greater than 50 % at \( X / D_j = 6.35 \). The turbulence level then reduces to minimum levels between \( X / D_j \) of 18.2 and 20.8, after which it rises to a peak and begin to decrease again. A peak of 55 % is reached at \( X / D_j = 32.7 \) for the jet fan positions of \( F_p \) of 0.22 and 0.33. At the position \( F_p = 0.06 \) the turbulence levels continue to rise monotonically and at \( X / D_j = 57.2 \) the turbulence level is nearly 65%. High entrance values occur because close to the tunnel entrance there are high velocity fluctuations observed in the entrained flow which reduce downstream and then increase to a peak again.
The turbulence levels for the jet fan at the tunnel axis position are somewhat different from the other positions in the first twenty jet fan nozzle diameters mainly because measurements are taken directly on the jet axis. From the jet nozzle up to the end of the potential core of the jet flow at $X/D_j < 20$, the turbulence levels resemble that of a free jet i.e. initially very low and rising gradually. The free jet turbulence levels gradually reach a constant level of around $25\%$ at greater distances away from the nozzle and those of the jet fan rise significantly at $X/D_j = 18.2$ and reach a peak at $43.5$ nozzle diameters before dropping to a value of about $30\%$. This value is almost half that of the turbulence level when the jet fan is located at near the wall. Curtet's turbulence data for a centrally positioned jet varies in the same manner in the first twenty jet nozzle outlet diameters as in the present case when the jet fan is located at the tunnel axis. Further downstream the turbulence levels in Curtet's case rise continuously up to a value of $80\%$ as shown in Figure 7.1. The presence of the confining walls causes a larger increase in the turbulence level than would be obtained in a free jet.

### 7.2 Entrainment Rate as Function of Jet Fan Position

The pressure and flow results including the turbulence levels are needed to understand the jet fan entrainment data. The entrainment results are a direct measure of jet fan performance. The results presented are for a wind tunnel without an initial flow when the jet fan flow was introduced. The total tunnel flow is the sum of the induced secondary flow and primary jet flow for two conditions. (i) In Figure 7.2 the flow ratio $Q_e/Q_j$ and $Q_T/Q_j$ are plotted against jet fan position $F_p$ for the jet fan diameter to tunnel diameter ratio of $D_R = 0.11$ and $U_j = 40$ m/s. (ii) Figure 7.3 presents the same quantities but for the diameter ratio $D_R = 0.17$ and $U_j = 21.4$ m/s. In both situations the same Reynolds
number is maintained. The quantities $Q_j$, $Q_e$ and $Q_T$ (m$^3$/s) are the jet fan discharge, entrained (secondary) and total tunnel flow ($Q_T = Q_e + Q_j$).

Figures 7.2 and 7.3 show that the amount of secondary air entrained is higher when the jet fan is closer to the tunnel wall. The entrained air reduces when the jet fan is moved away from the tunnel walls and reaches a minimum level at $F_p = 0.39$ for the smaller diameter jet fan (Figure 7.2) and at $F_p = 0.42$ for the larger diameter jet fan. The amount of entrained air increases significantly at the tunnel axis position but does not reach the same level as when the jet fan is located at the wall as shown in Figure 7.2 for the smaller diameter jet fan. In the larger diameter jet fan (Figure 7.3) there is a small increase in the entrained air at the tunnel axis position after the minimum entrainment level is attained. In Figure 7.2 it is clear that the jet fan can entrain an amount of secondary flow equivalent to its own outlet discharge volume flow when situated near the tunnel wall. The tunnel total flow thus varies from about twice to one and a half times the jet fan flow for the higher velocity jet fan (Figure 7.2). The larger diameter jet fan (with lower outlet velocity) does not entrain as much air as the higher velocity, lower diameter fan. At the peak value in Figure 7.3 the jet fan entrains secondary air equal to half its discharge volume flow (i.e. $Q_e / Q_j = 0.5$). The total tunnel flow varies between one and a half times and $1.1Q_j$.

An explanation for the entrainment results presented in Figures 7.2 and 7.3 can be accomplished with the aid of the axial pressure results described in Chapter 5. Because the jet fan flow is confined a pressure gradient is obtained as shown in Figures 5.1 to 5.11. The pressure initially is negative i.e. below ambient in the first twenty jet fan nozzle diameters from the outlet and this causes secondary air to be entrained into the tunnel as in a jet pump or ejector. When the jet fan is situated close to one tunnel wall it is observed that the initial pressure drops are quite considerable compared to the other positions and thus suction of secondary air is enhanced even though strong reverse flow is present in this
situations. The slight recovery in entrained air seen at the tunnel axis position is probably due to a combination of high turbulence mixing between the jet flow and secondary flow and the symmetrical positioning of the jet fan in the tunnel, which might favour the entrainment of secondary air.

7.3 Jet Fan Performance Assessment

Jet fan performance criteria can be derived from the pressure and flow measurement. This enables jet fans of different outlet diameters, discharge outlet velocity and Reynolds numbers to be compared directly.

In Figures 7.4 and 7.5 two major performance parameters are plotted for two jet fans of different outlet velocities and diameter but with the same discharge mass flow or Reynolds number. The first parameter defining jet fan performance $\xi_{jf}$ is obtained by the product of the pressure ratio $(P_{in} - P_e) / (P_j - P_{in})$ and the flow ratio $Q_T / Q_j$ as

$$
\xi_{jf} = \frac{Q_T}{Q_j} \frac{(P_{in} - P_e)}{(P_j - P_{in})}
$$

(7.1)

The pressure ratio in equation (7.1) results from the pressure rise of the total tunnel flow $(P_{in} - P_e)$ and the total pressure of the jet fan minus the total pressure in the tunnel $(P_j - P_{in})$. The above equation is an energy ratio of the tunnel flow to the jet fan discharge. In jet pumps it is common to use the flow ratio $Q_e / Q_j$ instead $Q_T / Q_j$ used in equation (7.1). This is also presented in Figure 7.4 but yields lower performance values. Equation (7.1) agrees reasonably well with another performance parameter defined by Reale (1973) as induction efficiency $\eta_{i}$ and given by equation (7.2) as
\[ \eta_i = \left( \frac{2\Phi}{1+\Omega} \right) \left( \frac{1-\Phi}{1+\Omega} \right) \]  

(7.2)

In the above equation \( \Phi \) is the velocity ratio of the tunnel to the jet outlet flow \( U_t / U_j \) and \( \Omega \) is the area ratio of the jet fan to the tunnel \( A_j / A_t \). The induction efficiency can be defined as the ratio of the ventilation power output to the power transmitted to the fluid by the jet fan. The induction efficiency is plotted in Figures 7.4 and 7.5 for various jet fan positions in the tunnel. The values of performance parameters defined by both equations 7.1 and 7.2 vary from 3.5% to a minimum of about 3%. It can be seen that the jet fan performance is better for positions closer to the tunnel wall than away from it. When the jet fan is at the tunnel axis the performance also improves but is still lower than that at the wall positions. The fact that equations 7.1 and 7.2 yield similar results plotted in Figure 7.4, is very encouraging, and gives confidence in the current measurements.

Figure 7.5 also presents the performance parameter \( \xi_j \) and the induction efficiency \( \eta_i \) for the jet fan with outlet velocity 21.4 m/s and \( D_R = 0.17 \). Both performance values decrease when the jet fan is moved away from the tunnel wall and again after a minimum value is reached there is some recovery at the tunnel axis position. The results of jet fan performance presented in Figure 7.5 range from about 6% down to 4.6%. At jet fan positions \( F_p < 0.2 \) the induction efficiency values are slightly higher than the results derived from the measured pressure ratios in both Figures 7.4 and 7.5. At jet fan positions closer to the tunnel axis the induction efficiency results are slightly lower.

Although the results shown in Figures 7.4 and 7.5 show low performance parameters, jet fans are still considered a convenient way of ventilation in vehicle tunnels and some underground applications. When multiple jet fans are used to create high ratios of flow,
areas and pressure, the efficiency of the system is improved significantly. Figures 7.4 and 7.5 give a very good indication of the effect of position on the performance system. The performance variation follows that of flow ratio against jet fan position shown in Figures 7.2 and 7.3. It is worth mentioning that in the test facility room the amount of space was limited and this could reduce the amount of entrained air considerably. In a real facility there are fewer limitations than in a wind tunnel facility.

The results presented in Figures 7.4 and 7.5 show that larger diameter jet fans with low outlet velocities give better performance than low diameter ones with higher discharge velocities. The performance results presented in Figure 7.5 are almost double that shown in Figure 7.4 for the smaller diameter jet fan simulation even though in the latter case more secondary air is entrained. Therefore most of the energy in the jet fan flow is lost through a number of ways. An overall energy balance of the system can be formulated as follows:

\[
E_{\text{in}} = E_{\text{out}} + E_{\text{\textsuperscript{\textast}}} + E_{R} + E_{\text{ml}} + E_{\text{\textbar}}
\]  \hspace{1cm} (7.3)

where \( E_{\text{in}} \) is the initial amount of energy in the jet flow. The other energy terms \( E_{\text{out}}, E_{\text{\textsuperscript{\textast}}}, E_{R}, E_{\text{ml}} \) and \( E_{\text{\textbar}} \) are respectively the work output of the secondary flow, energy loss due to friction, reverse flow, mixing, and jet loss. The reverse flow, mixing and friction losses account for a significant amount of the losses incurred by the system. A special treatment of this topic is given in Chapter 8 where an attempt is made to give theoretical formulations of jet fan tunnel system.
Fig. 7.1 Tunnel centreline longitudinal turbulence levels
Fig. 7.2 Flow ratio vs jet fan position inside wind tunnel

\[ QT = Q_e + Q_j \]
\[ (1+n) \]
\[ QT/Q_j \]
\[ n \]
\[ Q_e/Q_j \]

\( DR = 0.11 \) \( U_j = 40 \) m/s

Fig. 7.3 Flow ratio vs jet fan position inside wind tunnel

\[ DR = 0.17 \] \( U_j = 21.4 \) m/s
Fig. 7.4 Jet fan performance vs position inside tunnel (DR = 0.11)

Fig. 7.5 Jet fan performance vs position inside tunnel (DR = 0.17)
Theoretical considerations are necessary in analysing a flow system fully and these considerations are required to understand the mechanisms of jet fan performance. One way of analysing jet fans is to perform a momentum and energy analysis by considering the losses that are encountered when two fluid streams of dissimilar velocities are mixed. Jet pump performance is often analysed in this way e.g. McClintock and Hood (1946), Cunningham (1957), and Cunningham (1976). However the analysis for jet fans is much more complicated because of the three dimensional nature of the flow but only a simplified theoretical approach will be developed for this study. It is reasonable to assume that density differences between the primary jet flow and the secondary induced stream are negligible.

Figure 8.1 shows a schematic diagram of the jet fan-tunnel system. The area of the cross section occupied by the fan in the tunnel can be defined as follows:

\[ A_s = A_r - A_j \]

and from continuity

\[ m_t = m_s + m_j \]
A_j, A_s and A_t are the areas of the jet fan outlet or nozzle, the area through which the secondary flow is introduced to the tunnel and the tunnel cross sectional areas respectively. The mass flows m_j, m_s and m_t represent the jet flow, secondary stream and total tunnel mass flow respectively.

Let \( \frac{A_j}{A_s} = \alpha \), then (i) \( \frac{A_j}{A_t} = \frac{\alpha}{1 + \alpha} \)

(ii) Let \( \frac{m_s}{m_j} = n \) (iii) then \( \frac{m_t}{m_j} = \frac{m_j + m_s}{m_j} = 1 + n \) (iv)

The velocity of the secondary stream \( u_s \) and tunnel flow \( u_t \) can be expressed in terms of the jet discharge velocity \( u_j \) as follows:

\[ u_s = \alpha nu_j, \quad (v) \]

\[ u_t = \frac{\alpha(1 + n)}{1 + \alpha} u_j, \quad (vi) \]

The axial static pressure of the flow at the initial mixing of the primary jet and secondary stream and at the final mixed stage is given by the following equation if zero friction loss is assumed,

\[ (p_t - p_s)A_t = m_j u_j + m_s u_s - m_t u_t \]

(8.1)

By using equations (i) to (vi) the above equation reduces to
\[ p_t - p_e = \alpha \rho u_j^2 (1 - 2\alpha n + \alpha^2 n^2) \]

\[ p_t - p_e = \alpha \rho u_j^2 (1 - \alpha n)^2 \quad (8.2) \]

The above equation is valid only when there are no losses in the system which is not the case in reality. A number of equations can be written for each part of the jet fan ventilation system.

8.1 Jet Fan Nozzle Energy Equation

\[
\frac{p_i + \frac{u_{ni}^2}{2}}{\rho} = \frac{p_e + \frac{u_j^2}{2} + P_{fl}}{\rho} \quad (8.1)
\]

\[ p_{fl}/\rho \] represents the nozzle energy loss. Let \( P_j = p_i + \rho u_{ni}^2/2 \) the total pressure of the jet and \( p_{fl} = \xi_n \rho \frac{u_j^2}{2} \)

The jet fan nozzle equation is then

\[ P_j - p_e = (1 + \xi_n) \frac{\rho u_j^2}{2} \quad (8.3) \]

Secondary or entrained flow energy equation
An equation similar to (8.3) can be written for the secondary stream.

\[ P_s - p_e = (1 + \zeta_s) \frac{\rho u_j^2}{2}, \]

\( P_s \) and \( p_e \) are the suction pressure at entry and start of mixing point respectively. \( \zeta_s \) is a suction loss coefficient. The above equation can now be expressed as

\[ P_s - p_e = (1 + \zeta_s) \alpha^2 n^2 \rho u_j^2 \] (8.4)

8.2 Momentum Balance in the Tunnel

Momentum balance in the tunnel is the form of equation (8.1) but all possible losses are considered. The momentum equation can be written as follows,

\[ m_j \dot{u}_j + m_i \dot{u}_i - m_i u_i - \zeta_i A_i \frac{\rho u_i^2}{2} = (p_i - p_e) A_i \] (8.5)

If significant obstructions are present which might contribute to the overall momentum loss then equation (8.5) can be expressed as follows;

\[ m_j \dot{u}_j + m_s u_s - m_i u_i - \zeta_i A_i \frac{\rho u_i^2}{2} - \zeta_{ab} f_{ab} A_i \frac{\rho u_i^2}{2} = (p_i - p_e) A_i \] (8.6)
\( \zeta_t \) is the friction loss due to tunnel or confining walls, \( \zeta_{ob} \) is drag or a resistance coefficient due to the presence of obstructions and \( f_{ob} \) is the fraction of tunnel area occupied by obstructing objects giving \( f_{ob}A_t \) to be the frontal area of these objects.

The tunnel experiences a significant amount of recirculation in the mixing section but momentum is not lost. This momentum is equivalent to \( m_r u_r \) where \( m_r \) and \( u_r \) are the back or recirculating mass flow and average backflow velocity throughout the tunnel respectively.

By using equations (i) to (vi) equation (8.6) becomes

\[
    m_j u_j + \alpha n^2 m j u_j - \frac{\alpha(1+n)^2}{1+\alpha} m_j u_j - (\zeta_t + \zeta_{ob} f_{ob}) A_t \frac{\alpha^2(1+n)^2 \rho u_j^2}{2(1+\alpha)^2} = (p_t - p_e) A_t \quad (8.7)
\]

By making the substitution \( m_j u_j = A_j \rho u_j^2 = \frac{\alpha A_t}{1+\alpha} \rho u_j^2 \) in the above equation the pressure drop in the tunnel is given by

\[
    p_t - p_e = \frac{1}{A_t} \left( \frac{\alpha A_t}{1+\alpha} \rho u_j^2 + \frac{\alpha^2 A_t}{1+\alpha} \rho u_j^2 - \frac{\alpha^2(1+n)^2}{(1+\alpha)^2} A_t \rho u_j^2 - (\zeta_t + \zeta_{ob} f_{ob}) A_t \frac{\alpha^2(1+n)^2 \rho u_j^2}{2(1+\alpha)^2} \right)
\]

(8.8)

\[
    p_t - p_e = \rho u_j^2 \left( \frac{\alpha}{1+\alpha} + \frac{\alpha^2 n^2}{1+\alpha} - \frac{\alpha^2(1+n)^2}{(1+\alpha)^2} - (\zeta_t + \zeta_{ob} f_{ob}) \frac{\alpha^2(1+n)^2}{2(1+\alpha)^2} \right) \quad (8.9)
\]

\[
    p_t - p_e = \frac{2\alpha}{1+\alpha} \frac{\rho u_j^2}{2} \left( 1 + \alpha n^2 - \frac{\alpha(1+n)^2}{(1+\alpha)} - (\zeta_t + \zeta_{ob} f_{ob}) \frac{\alpha(1+n)^2}{2(1+\alpha)} \right) \quad (8.10)
\]

\[
    p_t - p_e = \frac{\rho u_j^2}{(1+\alpha)^2} \left( \alpha(1+\alpha) + \alpha^2 n^2(1+\alpha) - \alpha^2(1+n)^2 - (\zeta_t + \zeta_{ob} f_{ob}) \frac{\alpha^2(1+n)^2}{2} \right) \quad (8.11)
\]
The pressure drop of the jet flow is given by subtracting equation (12) from (3) and obtaining

\[ P_j - P_i = (1 + \zeta_n) \frac{\rho u_j^2}{2(1 + \alpha)^2} - \frac{\rho u_i^2}{2(1 + \alpha)^2} \left(2\alpha(1 - \alpha n)^2 - \alpha^2(1 + n)^2(\zeta_t + \zeta_{ob,f_{ob}})\right) \]  

(8.13)

and can be further simplified to

\[ P_j - P_i = \frac{\rho u_j^2}{2(1 + \alpha)^2} \left((1 + \alpha)^2(1 + \zeta_n) - 2\alpha(1 - \alpha n)^2 + \alpha^2(1 + n)^2(\zeta_t + \zeta_{ob,f_{ob}})\right) \]  

(8.14)

### 8.4 Jet Fan Performance Efficiency

A jet fan performance parameter can now be described as the ratio of energy output to energy input

\[ E_{out} = \frac{m_J}{\rho} (P_i - P_e) = \frac{n_m}{\rho} (P_i - P_e) \]  

(8.15)

\[ E_{in} = \frac{m_J}{\rho} (P_j - P_i) \]  

(8.16)

\[ \eta = \frac{E_{out}}{E_{in}} = n(P_i - P_e)/(P_j - P_i) \]  

(8.17)
\[ \eta = n \frac{2\alpha(1 - \alpha n)^2 - \alpha^2(1 + n)^2(\zeta_r + \zeta_{ob} f_{ob})}{(1 + \alpha)^2(1 + \zeta_{n}) - 2\alpha(1 - \alpha n)^2 + \alpha^2(1 + n)^2(\zeta_r + \zeta_{ob} f_{ob})} \] (8.18)

Equation (18) shows that the efficiency of the jet fan depends on the entrainment or the mass flow ratio \( n \) i.e. the secondary stream to the primary jet fan mass flow ratio \( m_s/m_j \) and the area ratio \( \alpha \) which is the ratio of the area occupied by the jet fan at the tunnel entry to the area available for the secondary flow to enter the tunnel \( A_s/A_j \). The loss coefficients \( \zeta_r \) and \( \zeta_{ob} \) can be determined directly or indirectly. The friction coefficient \( \zeta_r \) can be obtained from wall shear stress measurements. The resistance caused by obstructions \( \zeta_{ob} \) can be estimated fairly accurately. If the pressure drop for a clear tunnel is known, the differences can be attributed to the presence of obstructions and therefore the coefficient \( \zeta_{ob} \) can be found. If the pressure drop in the tunnel or opening and the flow ratio \( n \) are known then the loss coefficients can be estimated by solving equation (14). Both \( n \) and the recirculation or backflow fraction \( \zeta_r \) are a function of the jet fan position \( Y/D_t \) from the confining walls.

### 8.5 Theoretical Estimation of the Backflow Fraction \( \zeta_r \)

A jet fan of diameter \( D_j \) is situated at a distance \( Y \) metres from the tunnel wall to its axis. The tunnel is of hydraulic diameter \( D_t \). Since from experimental observation it is known that recirculation takes place on one side of the tunnel when the jet fan is situated at distance \( \frac{Y}{D_t} < 0.5 \) an estimation of the volume occupied by the body of the backflow can be carried out by assuming that the jet develops like a free jet until it reaches the tunnel walls. The jet will expand with angle \( \theta \) from the nozzle and the distance it takes for it to reach one side of the tunnel wall is \( L_r \), the recirculation length.
\[ \tan \frac{\theta}{2} = \frac{D_t - Y - 0.5D_j}{L_r} \quad (8.19) \]

the backflow length is then given by

\[ L_r = \left( \frac{D_t - Y - 0.5D_j}{\tan \frac{\theta}{2}} \right) \quad (8.20) \]

The area occupied by the backflow eddy if assumed to be approximated by triangular shape can be expressed as

\[ A_r = 0.5(D_t - Y - 0.5D_j) \left( \frac{D_t - Y - 0.5D_j}{\tan \frac{\theta}{2}} \right) = 0.5(D_t - Y - 0.5D_j)L_r \quad (8.21) \]

If an average backflow velocity \( u_1 \) is assumed within this area then the recirculation fraction can be calculated as

\[ \zeta_r = \frac{0.5(D_t - Y - 0.5D_j)u_1L_r}{L_r u_1 D_t} = \frac{0.5(D_t - Y - 0.5D_j)u_1}{u_1 D_t} \quad (8.22) \]

and in terms of the jet discharge velocity

\[ \zeta_r = \frac{0.5(1+\alpha)(D_t - Y - 0.5D_j)\alpha}{\alpha(1+\alpha)u_1 D_t} \quad (8.23) \]

### 8.6 Jet Fan Analysis from Energy Considerations

The jet fan performance can be analysed from energy considerations by carrying out an energy balance in the tunnel. This is done by first assessing all possible losses and taking into account the energy input and output. The losses are accounted as follows.
1. Friction losses

(i) Secondary stream energy loss = $\zeta_s m_s \frac{u_s^2}{2}$  \hspace{1cm} (8.24)

(ii) Jet fan nozzle loss = $\zeta_n m_j \frac{u_j^2}{2}$  \hspace{1cm} (8.25)

(iii) Tunnel friction loss = $\zeta_r m_r \frac{u_r^2}{2}$  \hspace{1cm} (8.26)

2. Backflow (recirculation) energy loss $E_{rl} = m_r \frac{u_r^2}{2} = \zeta_r m_r \frac{u_r^2}{2}$  \hspace{1cm} (8.27)

3. Jet loss which the jet experiences a loss during its discharge from the fan. This loss in energy is

$$E_{ji} = \frac{m_j}{\rho} (p_s - p_a) = \frac{m_j}{\rho} D \frac{u_j^2}{2} = m_j \frac{u_j^2}{2}$$ \hspace{1cm} (8.28)

(4) Mixing Energy Losses

In addition to the other losses mixing energy losses occur because two streams of different velocities are brought together. The mixing loss is

$$E_{ml} = m_i \frac{u_{mix}^2}{2}$$ \hspace{1cm} (8.29)

In the above equation $m_i$ is the tunnel mass flow and $u_{mix}$ is the mixing velocity which will be dealt with later.
An overall energy balance can be written as follows

\[ E_{in} = \dot{W}_t + E_{rl} + E_{md} + E_{fl} + E_{fl} \]  

(8.30)

\[ \dot{W}_t = E_{out}, \text{i.e. energy out of the tunnel.} \]

The total energy loss is therefore \( E_{loss} = E_{rl} + E_{md} + E_{fl} + E_{fl} \)

\( E_n \) is the sum of all frictional energy losses and is

\[ E_{fl} = \zeta_s m_s \frac{u_s^2}{2} + \zeta_n m_j \frac{u_j^2}{2} + \zeta_s m_s \frac{u_j^2}{2} \]  

(8.31)

\[ E_{fl} = \zeta_s \alpha^2 n^3 m_j \frac{u_j^2}{2} + \zeta_n m_j \frac{u_j^2}{2} + \alpha^2 (\frac{1+n}{(1+\alpha)^3}) \zeta_s m_j \frac{u_j^2}{2} \]  

(8.32)

\[ E_{fl} = m_j \frac{u_j^2}{2} \left( \zeta_s \frac{u_s^2}{2} + \zeta_n + \alpha^2 \frac{(1+n)^3}{(1+\alpha)^3} \right) \]  

(8.33)

The recirculation loss \( E_{rl} \) is

\[ E_{rl} = \zeta_r m_r \frac{u_r^2}{2} = \zeta_r \alpha^2 \frac{(1+n)^3}{(1+\alpha)^3} m_j \frac{u_j^2}{2} \]  

(8.34)

Equation (29) describes a mixing energy loss. The mixing velocity has to be defined and determined. The following reasoning can be applied. First it is necessary to assume that the jet once past the nozzle of the fan develops like an axisymmetric free jet until it reaches the confining walls. The centreline velocity of axisymmetric jet is given by

\[ u_m = 12 \frac{a}{x} u_j \]  

(8.35)
The mixing velocity is given by \( u_{\text{mix}} = \frac{1}{(x_i - x_0)} \int_{x_0}^{x_i} (u_m - u_r) \, dx \) \hspace{1cm} (8.36)

The limits of integration \( x_0 \) and \( x_i \) are the end of the jet potential core and point where jet centreline velocity \( u_m = u_r \) respectively. At this point the body of the tunnel airflow is assumed to be fully mixed (see Figure 8.2).

\[
\begin{align*}
    u_{\text{mix}} &= \frac{12 \rho u_j}{(x_i - x_0)} \int_{x_0}^{x_i} \left( \frac{dx}{x} \right) - \int_{x_0}^{x_i} u_r \, dx = \frac{12 \rho u_j}{(x_i - x_0)} u_j \ln \left( \frac{x_i}{x_0} \right) - u_r \\
    &\hspace{1cm} (8.37)
\end{align*}
\]

The mixing energy loss is then

\[
E_{m1} = \frac{m_r}{2} \left( \frac{12 \rho}{(x_i - x_0)} u_j \ln \left( \frac{x_i}{x_0} \right) - u_r \right)^2 = (1 + n) \frac{m_j}{2} \left( \frac{12 \rho}{(x_i - x_0)} u_j \ln \left( \frac{x_i}{x_0} \right) - \frac{\alpha(1+n)}{(1+\alpha)} u_j \right)^2
\]

\[
E_{m1} = (1 + n) m_j \frac{u_j^2}{2} \left( \frac{12 \rho}{(x_i - x_0)} \ln \left( \frac{x_i}{x_0} \right) - \frac{\alpha(1+n)}{(1+\alpha)} \right)^2
\hspace{1cm} (8.38)
\]

The total energy loss can be determined by summing all the losses

\[
E_{\text{loss}} = \zeta_s \alpha^2 n^3 m_j \frac{u_j^2}{2} + \zeta_n n m_j \frac{u_j^2}{2} + \alpha^2 \frac{(1+n)^3}{(1+\alpha)^2} \zeta_s m_j \frac{u_j^2}{2} + (1+n)m_j \frac{u_j^2}{2} \left( \frac{12 \rho}{x_i - x_0} \ln \left( \frac{x_i}{x_0} \right) - \frac{\alpha(1+n)}{(1+\alpha)} \right)^2 + \zeta_r \frac{\alpha^2(1+n)^3}{(1+\alpha)^2} m_j \frac{u_j^2}{2}
\]

\[
E_{\text{loss}} = m_j \frac{u_j^2}{2} \left( \zeta_s \alpha^2 n^3 + \zeta_n + \alpha^2 \frac{(1+n)^3}{(1+\alpha)^2} \zeta_s + (1+n) \left( \frac{12 \rho}{x_i - x_0} \ln \left( \frac{x_i}{x_0} \right) - \frac{\alpha(1+n)}{(1+\alpha)} \right)^2 + \zeta_r \frac{\alpha^2(1+n)^3}{(1+\alpha)^2} \right)
\hspace{1cm} (8.39)
\]
\[ E_{in} = \frac{m_j}{\rho} (P_j - p_t) \]  
\hspace{1cm} (8.40)

\[ E_{out} = \frac{m_j}{\rho} (p_t - p_e) = \frac{m_j}{\rho} (p_t - p_e) \]  
\hspace{1cm} (8.41)

\[ E_{in} - E_{out} = E_{loss} \]  
\hspace{1cm} (8.42)

\[ E_{loss} = m_j \frac{u_j^2}{2} (\zeta_s \alpha^2 n^3 + \zeta_n + \alpha^2 \frac{(1+n)^3}{(1+\alpha)^2} \zeta_t + (1+n) \left( \frac{12r_0}{x_{t-x_0}} \ln \left( \frac{x_t}{x_0} \right) - \frac{\alpha(1+n)}{(1+\alpha)} \right)^2 \alpha \alpha^2 (1+n)^3) + \zeta_r \]  
\hspace{1cm} (8.43)

Efficiency can now be defined as

\[ \eta = 1 - \frac{E_{loss}}{E_{in}}, \quad \eta = 1 - \frac{m_j \frac{u_j^2}{2}}{\frac{m_j}{\rho} (P_j - p_t)} \]  
\hspace{1cm} (8.44)

\[ \eta = 1 - \left( \frac{m_j \frac{u_j^2}{2}}{\frac{m_j}{\rho} (P_j - p_t)} \right) (\zeta_s \alpha^2 n^3 + \zeta_n + \alpha^2 \frac{(1+n)^3}{(1+\alpha)^2} \zeta_t + (1+n) \left( \frac{12r_0}{x_{t-x_0}} \ln \left( \frac{x_t}{x_0} \right) - \frac{\alpha(1+n)}{(1+\alpha)} \right)^2 \alpha \alpha^2 (1+n)^3) + \zeta_r \]  
\hspace{1cm} (8.45)
8.7 Analysis of the Performance \( \eta \)

The dependency of the performance parameter \( \eta \) on flow ratio \( (n) \) and area ratio \( (\alpha) \) from momentum considerations can be evaluated by plotting equation (8.18) graphically. Equation (8.18) is a product of flow ratio \( (n) \) and the pressure ratio \( (p_t - p_e) / (p_j - p_t) \). The numerator \( (p_t - p_e) \) is the tunnel pressure rise due to the jet fan and \( (p_j - p_t) \) is the pressure drop of the jet fan. For ease of understanding, equation (8.18) is repeated below as equation (8.46)

\[
\eta = n \frac{2\alpha(1 - \alpha n)^2 - \alpha^2(1 + n)^2(\zeta_t + \zeta_{ob}f_{ob})}{(1 + \alpha)^2(1 + \zeta_n) - 2\alpha(1 - \alpha n)^2 + \alpha^2(1 + n)^2(\zeta_t + \zeta_{ob}f_{ob})} \tag{8.46}
\]

To simplify the problem, it is assumed that there are no obstructions in the tunnel and therefore the quantity \( \varsigma_{ob}f_{ob} = 0 \). Equation 8.18 becomes

\[
\eta = n \frac{2\alpha(1 - \alpha n)^2 - \alpha^2(1 + n)^2}{(1 + \alpha)^2(1 + \zeta_n) - 2\alpha(1 - \alpha n)^2 + \alpha^2(1 + n)^2} \tag{8.47}
\]

Equation (8.47) is depicted in Figure 8.3 as a plot of \( \eta \) vs. flow ratio \( n = Q_0/Q_j \) for constant values of the friction loss factor \( \varsigma_\alpha \). The area ratio \( (\alpha) \) is the area that is not occupied by the fan (and available for the secondary stream to enter the airway), to that of the airway. A value of \( \alpha = 0.99 \) obtained from the experimental work is used. A friction loss term \( \varsigma = 0.009 \) is used in this example. This value is obtained by using the Blasius friction formula for smooth surfaces i.e. \( \zeta_t = 0.079 \text{Re}^{-0.25} \) for a tunnel Reynolds number of 6000 for the purpose of this example. A jet fan nozzle loss coefficient of \( \varsigma_n \) of 0.01 is used. In Figure 8.3 the value of \( n \) discounting nozzle losses peaks at about 0.1 for a flow
ratio (n) of 0.28. For flow ratios at this set of conditions, when the entrainment flow exceeds 90% of the jet flow (n > 0.9), the performance (η) becomes meaningless (i.e. its value drops below zero).

When the nozzle losses are included, one observes that the peak value of (η) is reduced to 0.084 and occurs at a lower flow ratio (n) of 0.22. As well the practical range of meaningful flow ratios declines to the upper limit of 0.61. Figure 8.3 shows that η = 0, at n = 0, and η will increase with n, until a maximum is reached at an optimum value of n between 0.2 and 0.3 for the two plots. The higher the nozzle loss factor and friction in the passage the lower the performance value η. It is important to note that the jet fan performance values will always be low (less than 0.2 in all cases).

Figure 8.4 compares theoretical and experimental performance parameter (η) values. This shows that each performance curve represents a frictional factor value $\zeta_f$ which corresponds to an experimental value of (η). Experimental data fit the theoretical curve at its lower end, at low performance (η) and high flow ratio (n). To obtain optimum performance values from measured data, high pressure ratio values $(p_t - p_e)/(p_f - p_e)$ are required. Figure 8.5 shows optimum pressure ratio at optimum performance (η) and flow ratio (n). On average the optimum pressure ratio is 0.27. The experimental values are a factor of 10 less than the optimum and at a large flow ratio (n) along the performance curve.

An increase in the frictional factor loss term results in a decrease of performance (η) values and a narrowing of the flow ratio (n) range. Each curve in Figure 8.4 represents a particular fan position parameter $F_p$ and therefore a friction factor loss $\zeta_f$ value which covers a particular range of flow ratio (n) on the performance curve. In Figure 8.6, the
calculated frictional loss term $\zeta$ is plotted against fan position parameter $F_p$. The friction loss factor is high for the small $(n)$ value range which occur at positions farther from the wall. This small $n$ range corresponds to a low tunnel Reynolds number at jet fan positions farther from the wall.

In Figure 8.7 the performance parameter ($\eta$) is plotted against area ratio ($\alpha$) for a family of flow ratio $n$ values of 0.1, 0.3 and 0.7. At each flow ratio, there is an optimum area ratio ($\alpha$). The larger the flow ratio e.g. $n = 0.7$ the higher the performance parameter ($\eta = 0.19$) and the smaller the optimum area ratio ($\alpha = 0.3$). The optimum area ratios for smaller $n$ values are larger e.g. $\alpha = 0.48$ for $n = 0.3$, $\alpha = 0.78$ for $n = 0.1$. Figure 8.8 shows the optimum area ratio plot for various flow ratios $n$. The value of optimum area ratio declines rapidly with increasing flow ratio. In most mining applications the optimum value of area ratio is not used because the sizes of the passages are determined primarily by the mining method and not by ventilation needs. Fans are of limited range in diameter and therefore on average, area ratios encountered in mining are above 0.8. Only in a few situations is a smaller area ratio encountered and it would be normally greater than 0.5.

The values of flow ratio plotted in Figure 8.4 specify a range of $n$ which can be varied by controlling the amount of secondary air entrained. In the experiment the amount of secondary fluid entrained measured by the ratio $n$ is not controlled and is found to depend on jet fan position $F_p$. The foregoing analysis has demonstrated that equation 8.46 can be used in jet fan performance assessment. For each set of conditions defined by the friction loss term or jet fan position there is an optimum performance value and an optimum flow ratio $(n)$ for a given entrained flow area ratio $\alpha$. 
Table 8.1 Description of jet fan - tunnel system

- **Ae**: secondary flow inlet area
- **Aj**: jet fan outlet area
- **At**: tunnel cross sectional area
- **Pt**: tunnel pressure
- **Pe**: sec. flow entry pressure
- **Pj**: jet total pressure
- **mj**: jet fan mass flow
- **mt**: tunnel total mass flow
- **Y**: distance of jet fan from tunnel wall
- **α**: jet expansion angle
- **Lr**: backflow length

*Figure 8.1 Schematic description of jet fan - tunnel system*
Figure 8.2 Velocity decay of an axisymmetric jet and mixing velocity concept
Figure 8.3 Plot of theoretical performance vs flow ratio $n$

$S_t = 0.009, \alpha = 0.99$

Figure 8.4 Plot of theoretical performance vs flow ratio $n$ for various jet fan positions (friction loss factors)

$S_t = 0.01, \alpha = 0.99$

- Experiment DR=0.11, $U_j=40$ m/s
- Experiment DR=0.17, $U_j=21.4$ m/s
Figure 8.5 Optimum flow ratio and performance vs optimum pressure ratio

Figure 8.6 Friction loss factor $\xi$ vs jet fan position $F_p$
Figure 8.7 Performance $\eta$ vs area ratio for various flow ratio $n$

Figure 8.8 Flow ratio $n$ vs optimum area ratio $\alpha$
CHAPTER NINE

PRACTICAL APPLICATIONS OF JET FAN WIND TUNNEL STUDIES

The results of the present study were obtained using an open ended wind tunnel investigation and it is necessary to give a practical application of the results. There are many potential applications of jet fans in mine and industrial ventilation. In mine ventilation, the present results directly provide information for a jet fan used in an open ended airway such as that shown in Figure 9.1.

9.1 Jet Fan Application Case Study 1

It is desired to supply airflow to airway B, D and E following a disruption of airflow due to the collapse of airway C which used to supply airflow to D and E. The ventilation district is arranged as in Figure 9.1. The airflow should all be supplied from airway A which has an airflow of 60 m$^3$/s. We need to supply D and E with approximately 10 and 20 m$^3$/s respectively and this requires an airflow quantity of 30 m$^3$/s in airway B. This can be done by installing a mobile jet fan in airway B of capacity 20 m$^3$/s, with a diameter of 800 mm and about 40 m/s discharge velocity. For operational reasons the jet fan must be placed at least 12 meters from the intersection of airway A and B. For an effective ventilation system airway B should be no more than 150 m in length in order to maintain the pressure of the system. The cross sectional hydraulic diameter of airway B should be no more than ten times that of the jet fan. In order to determine the amount of air the jet fan entrains, Figure 7.2 is used. For a jet fan positioned at $F_p = 0.05$ the entrainment ratio $Q_e/Q_j = 0.9$ (or a total flow of 1.9 times the original jet fan discharge volume flow) for a discharge velocity of $U_j = 40$ m/s. Therefore on average, about 38 m$^3$/s will flow in
airway B and for a cross sectional area of 16 m² the average velocity will be 2.4 m/s. If other jet fan discharge velocities are used the entrainment or flow ratio will differ. The positioning of the fan from the walls will also affect the flow ratio.

To determine the pressure rise in the airway caused by the jet fan, equation 8.12 is used i.e.

\[ p_t - p_e = \frac{\frac{\rho U_j^2}{2(1+\alpha)^2} (2\alpha(1-\alpha n)^2 - \alpha^2(1+n)^2(\zeta_f + \zeta_{ob}f_{ob}))}{2} \] (8.12)

If it is assumed that there are no obstructions in airway B then the drag resistance \( \zeta_{ob} \) is zero. The airway friction factor \( \zeta_f \) can be estimated fairly reasonably. The area ratio \( \alpha \) is the area not occupied by the fan to that of the total airway. Using \( n = 0.9 \), \( U_j = 40 \text{ m/s} \), \( \alpha = 0.969 \), \( \rho = 1.2 \text{ kg/m}^3 \), the dimensionless friction factor of the airway \( \zeta_f = 0.001 \) (for 10% pressure drop). The pressure rise of the airway \( (p_t - p_e) \) according to equation 8.12 will be 7.01 N/m² and this should be constant along the length of the airway to maintain the flowrate of 38 m³/s. The pressure \( p_e \) and \( p_t \) are the pressure upstream of the jet fan and maximum pressure attained respectively.

To calculate the induction efficiency of the system, equation 4.2 is used

\[ \eta_i = 2\Phi(1-\Phi)/((1-\Omega)(1+\Phi)) \]

Since the airway to jet fan velocity ratio \( \Phi = 0.06 \), and the jet fan to airway area ratio \( \Omega = 0.0314 \), the above equation gives an induction efficiency of about 11% i.e. \( \eta = 2\times 0.06(1-0.06)/((1-0.0314)(1+0.06)) = 0.1099 \)
9.2 Jet Fan Application Case Study 2

A closed end mining heading is to be ventilated by a jet fan. The velocity at the production face is to be at least 0.5 m/s for the effective removal of pollutants. The heading is 30 metres long and has cross sectional dimensions of 10 metres wide by 5 metres high. The jet fan can be positioned as shown in Figure 9.2 with a curtain running along part of the heading to reduce re-entrainment of the return air from the working face and allowing the jet fan to entrain as much fresh air as possible from the main airway. In order to dilute pollutants below their TLVs at least 10 m³/s is required to reach the working face. A jet fan of 10 m³/s capacity is chosen and with a discharge velocity of 30 m/s. The fan diameter D_j is therefore 0.65 metres. To calculate what the centreline velocity will be at the face 30 metres away, the free jet centreline velocity decay equation is used as a guideline (i.e. equation 6.1). This yields a centreline velocity of 3.9 m/s. From Figures 6.7 to 6.9, it is clear that the actual value lies between 1 and 3.9 m/s since the jet is not totally confined as in the wind tunnel case. The total flow can be obtained using Figure 9.3 for a 10 m³/s jet fan if the entrainment ratio n is known or estimated. For any entrainment ratio n the amount of total flow will increase linearly. If an entrainment ratio n = (Q_e/Q_j) = 0.5 is assumed the total amount of fresh air reaching the working face area will be Q_j+Q_e and Q_e = 0.5 x 10 m³/s. Therefore the total amount of air will be 15 m³/s. This flow takes place in half of the heading area and the average velocity throughout the volume is 15 m³/s/25 m² = 0.6 m/s. This is a good velocity for the removal of pollutants in a mining area.

According to Thimons et al (1986) the recirculation of contaminated air can be up to 28 % which is not a problem considering the amount of excess fresh air entering the heading. In this case, a curtain of 5-10 metres in length can be used to separate the fresh air from the return air as shown in Figure 9.2. It is over this length that there is a possibility of return air being entrained and therefore the curtain can reduce this likelihood.
The advantage of a jet fan in this application is (i) more air at a higher average velocity is supplied to the working face and (ii) the need for ventilation tubing has been removed therefore cutting ventilation costs.

Jet fans can be fitted with entrainment tubes as shown in Figure 9.4 to control the amount air entrained from the return air in a closed airway. The use of a tube in this application would significantly reduce the recirculation fraction. There are other applications of jet fans in mine ventilation. Jet fans can be used for cooling purposes in hot working climates by increasing local air velocities. They can also be used to dilute gas emissions in large excavations by directing flow to the required point of application. Ventilation engineers have been hesitant to use jet fans in mines because of the lack of knowledge regarding their effects. The data contained in this work can assist the ventilation engineer in understanding the best way to employ these fans.
Figure 9.1 Example of jet fan used in through flow to increase airflow in other mine ventilation districts.
Figure 9.2 Jet fan used in a closed heading with curtain to reduce return air entrainment

\[ n = 0.5, Q_j = 10 \text{ m}^3/\text{s}, \quad Q_T = (1+n)Q_j = 15 \text{ m}^3/\text{s} \]
Figure 9.3 Illustration of flow ratio vs total flow for a 5 and 10 m3/s jet fan

Figure 9.4 Jet fan fitted with entrainment tube as in an ejector
CHAPTER TEN

CONCLUSIONS

This study has sought to provide data concerning jet fan ventilation in mines and tunnels. The studies have examined the pressure rise characteristics, the velocity development and the overall air entrainment of a jet fan simulated in a wind tunnel. Two jet fan sizes were used with different outlet velocities. By varying the jet fan position relative to the wind tunnel walls the optimum conditions and overall aerodynamics have been investigated. The conclusions drawn from the results of this investigation are presented in this chapter.

(1) A unique wind tunnel - jet fan test facility was designed, constructed and tested at the University of British Columbia, Department of Mining and Mineral Process Engineering for mine ventilation jet fan performance analysis in an open end case. The wind tunnel facility can be used for other mine ventilation studies in the future. This is a major contribution of this work.

(2) The pressure and flow conditions of jet fans applied to geometry appropriate to mine openings with respect to wall interactions was investigated.
(a) For the same discharge velocity, a jet fan situated at near wall positions ($F_p \leq 0.17$) produced larger pressure drops in the first thirty jet nozzle diameters downstream in the range of magnitudes between -0.005 and -0.006 jet discharge dynamic pressure. At the other positions ($F_p \geq 0.22$) the corresponding pressure drop values were less than -0.003 of the jet dynamic pressure; for the fan with a discharge velocity of 40 m/s and diameter ratio $D_R=0.11$. The pressure recovery for jet fans at positions $F_p > 0.22$ is higher by as much as 7 % than that at the closest jet fan position to the wall ($F_p = 0.06$). The larger
diameter fan ($D_r=0.17$) with a discharge outlet velocity of 21.4 m/s shows similar pressure variations to those of the smaller fan for the same discharge Reynolds number. However the pressure recovery factors expressed as a ratio of the jet discharge dynamic pressure are on average 2.5 times larger than for the smaller fan. When the experimental pressure recovery is compared to the theoretical one, it is found that there is a 15% difference for near wall jet fan position at a discharge velocity of 40 m/s ($D_r=0.11$) and 9% for the discharge velocity of 21.4 m/s ($D_r=0.17$). This indicates that the discharge velocity is important in determining the frictional pressure loss for a fan positioned close to the wall of an airway. This must be balanced by the requirement for the air to be moved. The wall friction losses for the smaller jet fan situated at the tunnel axis ($D_r = 0.11$) is about 8% and that for the larger fan ($D_r = 0.17$) is less than 2%. This shows that for a given air quantity, it is better to use a larger diameter fan and an optimum discharge velocity. The pressure rise for the smaller diameter fan reaches a peak at about 70 jet fan nozzle diameters and that for larger fan (and lower discharge velocity) peaks at about 45 nozzle diameters.

(b) Comparing the velocity profiles of the smaller and larger diameter fan, it can be concluded that the flow develops in the same manner qualitatively at similar jet fan positions from the wall. Generally for fan positions $F_p < 0.4$ there is a region of reverse flow on one side of the tunnel which extended for 20 and 40 jet nozzle diameters for the near wall jet fan position for the larger and smaller diameter fan respectively. Reverse flow reduces as the fan is traversed towards the axis of the tunnel. On average the magnitude of the reverse flow expressed as a fraction of the total flow at the tunnel cross sections is found to be about 0.72 and 0.55 for the smaller and larger fan respectively. Jet fan jet expansion angles ranged between $24^\circ$ and $26^\circ$ for both fan sizes and velocity discharges.

(c) From the velocity profiles results, it can be concluded that the jet fan can provide a ventilation airflow velocity greater than 0.5 m/s for distances up to 70 fan diameters from
discharge. The distances are slightly higher for fans positioned near walls. In a real mine application, the distances of ventilation can be even larger because fans of up to 50 m$^3$/s discharge volume can be used.

(d) Entrainment ratios for the smaller jet fan (diameter ratio, $D_R = 0.11$, $U_j = 40$ m/s) ranged from 0.92 when the fan was located near the wall $F_p = 0.06$ to about 0.6 when the fan is at position $F_p = 0.39$. The entrainment ratio $n$ for the larger jet fan ($D_R = 0.17$, $U_j = 20$ m/s) ranged between 0.5 for the fan wall position to a minimum of 0.1 at position $F_p = 0.44$.

(e) The measured performance parameter $\xi_{ij}$, defined by equation 7.1 as the product of the flow ratio $Q_T / Q_j$ and the pressure ratio $(P_m - P_e) / (P_j - P_m)$, agrees fairly well with the calculated induction efficiency $\eta_i$ (equation 7.2) and is higher for near wall positions than at positions away from the wall. The values obtained with the smaller fan at the near wall positions were $\xi_{ij} = 0.034$ and $\eta_i = 0.036$. Values of $\xi_{ij} = 0.058$ and $\eta_i = 0.062$ were obtained for the same position with the larger fan at same Reynolds number. These values decreased as the jet fan was moved away from the wall with some recovery for the tunnel axis position. The performance of a jet fan can be evaluated using both entrainment and pressure considerations.

(3) Theoretical formulations have shown that performance $\eta$ is dependent on the flow ratio $n$. For zero flow ratio, performance is also zero and it will rise to reach a maximum at a particular $n$ value and drop as $n$ is further increased. The useful range of flow ratio $n$ is $0 < n \leq 1$. This flow ratio range ensures a total flow in any ventilation passage greater than the initial jet fan discharge volume flow. The range of flow ratios $n$ decreases as the value of the friction factor loss increases. The optimum performance parameter ($\eta$) decreases from a value of about 0.1 to 0.024 as the friction loss factor $\zeta_f$ is increased from zero to 0.6. This friction loss factor corresponds to jet fan position in the tunnel.
Fan positions with a low range of flow ratio \( n \) therefore low tunnel Reynolds number have higher friction loss factors and lower performance values. When performance is plotted against area ratio \( \alpha \) for each flow ratio value \( n \) there is an optimum \( \alpha \) for each \( n \) (e.g. \( n = 0.1, \alpha_{\text{opt}} = 0.78 \), \( n = 0.3, \alpha_{\text{opt}} = 0.48 \), \( n = 0.7, \alpha_{\text{opt}} = 0.3 \)). The values will change a little depending on the friction loss factors or coefficient terms used in the equations.

Agreement between theoretical performance values and experimental ones was obtained on the lower range of the performance curve corresponding to the upper limit of the flow ratio \( n \) for each jet fan position. Apart from the theoretical analysis of Chapter 8 which uses the flow ratio \( n \) the parameter \( \xi_{\text{jet}} \) and the induction efficiency \( \eta_i \) provide an alternative way of analysing jet fan performance which is based on the tunnel total flow.

(4) This study has demonstrated that jet fans can be used to solve ventilation problems in open air passages and also in closed end situations by careful selection of the right capacity jet fan with a discharge velocity ranging between 20 and 40 m/s. Entrainment ratio \( 0 < n \leq 1 \) will provide excess mine ventilation air in production areas and therefore increase mine productivity. This makes it possible to mine out difficult areas which were not accessible because of ventilation problems.
CHAPTER ELEVEN

RECOMMENDATIONS

The results described in this thesis identified major parameters in the assessment of jet fan performance. These are size ratios, outlet velocities, flow ratios or entrainment and pressure rise characteristics.

(1) Jet fans should be used where there is a plentiful supply of fresh air so that enough air for the fan and entrainment can be available. In open passages jet fans positioned close to the wall \( F_p \leq 0.17 \) with an adequate source of air can move a larger mass of air than their own initial discharge volume for distances greater than 50 jet nozzle diameters. The range of discharge velocity to be used is 20 to 40 m/s. The optimum probably lies between these values and should be researched further. In closed end headings, jet fans can be used to increase airflow requirements to the working areas by selecting a fan of the right capacity \( Q_j \) and size and then positioning it close to the wall while extending the suction well into the fresh air. An entrainment ratio of 0.5 can be assumed to calculate the total amount of air reaching the face i.e. \( Q_j + 0.5Q_j = 1.5Q_j \). To reduce any possibility of entrainment of return air from the heading end a restricting curtain can be installed in the first 10 - 15 m of the heading. The use of jet fans therefore eliminate the need for ventilation tubing. Jet fan ventilation is very effective in clearing out pollutants because of good mixing of the jet flow and contaminants.

(2) Jet fans used in open tunnels should be used to boost the pressure of the flow in order to overcome friction losses of the passage walls so that air can be moved over large distances. Jet fans are also ideal for balancing flows in short interconnecting airways (e.g.}
up to 150 metres) as shown in Chapter 9.1. They can be placed near a major airway with a large airflow. The fan will move its own inlet volume of air plus secondary air moved by entrainment. The result is that a larger amount of air can be moved at a high pressure. This study shows that if the fan is close to the wall, the amount of air moved is much larger than when the fan is closer to the airway axis. For a jet fan situated at any position in the airway, the present results show that these fans can be used without any partition or long ventilation tubing. Jet fans always increase the pressure of the secondary stream by momentum and energy exchange and also provide more air than ducted fans for any flow (entrainment) ratio $n > 0$. 
CHAPTER TWELVE

CLAIMS TO ORIGINAL RESEARCH

I claim the following as original research from this work:

(1) The design and construction of a wind tunnel facility for studying jet fan applications in tunnels and mines.

(2) The establishment of position dependency of jet fan performance (pressure and flow ratio characteristics) in the ventilation of open passages i.e. wall interaction effects.

(3) The establishment of mathematical formulations to predict performance and friction losses based on flow ratio of the entrained and jet stream.
REFERENCES


Kempf, J., 1965, "Wall Effect on the Efficiency of Booster Fan," (In German), Schweizer Bauzeitung, 83 Jg, Heft 4, S47.


APPENDIX

The error or uncertainty analysis presented in Chapter four was used to estimate the errors in the present study. These errors represent the worst possible cases that are present in the measurements.

Table A1 Experimental uncertainty estimates of the data

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<tr>
<th>Quantity</th>
<th>Error estimate %</th>
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<td>$(P - P_e) / \frac{1}{2} (PU_j^2)$</td>
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</tr>
<tr>
<td>$U / U_j$</td>
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</tr>
<tr>
<td>$W_R / D_t$</td>
<td>8.6</td>
</tr>
<tr>
<td>$Q_R / Q_j$</td>
<td>8.74</td>
</tr>
<tr>
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<tr>
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<tr>
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</tr>
<tr>
<td>$(P_{in} - P_e) / (P_j - P_e)$</td>
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<tr>
<td>$(P_{in} - P_e) Q_T / Q_j (P_j - P_e)$</td>
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