

CHARACTERIZATION OF AIR RECIRCULATION
IN MULTIPLE FAN VENTILATION SYSTEMS

by

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A thesis submitted to the faculty of
The University of Utah
in partial fulfillment of the requirements for the degree of

Master of Science

Department of Mining Engineering

The University of Utah

May 2012

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ABSTRACT

Booster fans, large underground fans, can increase the volumetric efficiency of ventilation systems by helping to balance the pressure and quantity distribution throughout a mine, reducing leakage and reducing the total power requirement. The increased volumetric efficiency and reduced system power consumption in ventilation systems utilizing booster fans is demonstrated using a VnetPC ventilation model of an underground room and pillar coal mine. In this mine, a booster fan has the potential to reduce the total flow volume by 58 m³/s and to reduce the air power requirement by 110 kW.

Though booster fans have the potential to increase the efficiency of coal mine ventilation systems, there are problems associated with their use, most notably recirculation, the leakage of contaminated return air into the intake air. Recirculation is a risk because it has the potential to increase the concentration of air contaminants, including methane, dust, and heat, in the intake air. Computational Fluid Dynamic (CFD) and Ventsim numerical models and a physical laboratory model are used to evaluate recirculation in ventilation systems with booster fans. The use of booster fans does increase the potential for recirculation in coal mine ventilation systems, but the potential for recirculation is strongly dependent on the size of the booster fan relative to the main surface fan, the location of the booster fan, and the quality of the ventilation

control devices. The potential for recirculation increases as the booster fan pressure increases relative to the main fan, as the booster fan is located closer to the development sections, and as the quality of the ventilation control devices decreases. In well-designed and well-maintained ventilation systems, recirculation can be managed.

In part because of the risk of recirculation, booster fans are prohibited in most underground coal mines in the United States. However, despite increased potential for recirculation, booster fans are used effectively in other major coal producing countries including Australia and the United Kingdom. Regulations specific to coal mine ventilation from Australia, the United Kingdom, and the United States are compared to help identify practices to reduce risks associated with the use of booster fans.

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ACKNOWLEDGMENTS

First, I would like to extend sincere thanks to my graduate committee, Dr. Felipe Calizaya, Dr. Mike Nelson, and Dr. Geoff Silcox, for their interest, guidance, and support. Each member of my committee played an important role in my education and in the completion of my graduate work. I would also like to extend thanks to the other faculty of the Mining Engineering Department at the University of Utah, including Professor Tom Hethmon, Dr. Bill Hustrulid, Dr. Kim McCarter, and Dr. Bill Pariseau for the role they played in my education and development. Each of these professors provided me with encouragement when it was most needed. They used every opportunity to challenge my engineering aptitude and, on occasion, my perspective.

In addition, I would like to thank Rob Byrnes for his time and patience constructing and reconstructing the ventilation laboratory model and for answering questions I didn't know to ask, Pam Hofmann for always knowing the answer and having the patience and fortitude to ensure everything runs smoothly, and the National Institute for Occupational Safety and Health (NIOSH) for funding my research and education.

Finally, I would like to extend my deepest thanks to my family: my parents, Kris and Stan, for their constant love, support, and guidance; my brother, Jake, for keeping me grounded; and Corey for being my heart, my conscience, and my best friend.

CHAPTER 1

INTRODUCTION

1.1 Introduction

Ventilation is a critical component of underground coal mining. Adequate airflow in underground mines is necessary to create a safe environment for mine workers and to create an environment that is conducive to the operation of heavy equipment. To create safe working conditions for workers and machinery, the quantity and quality of airflow in a ventilation system must be adequate to dilute mine gases and combustion products including gases and particles, to remove dust, and to control air temperature.

As a necessary component of underground mining, ventilation is also a source of significant costs. Capital costs of main fans and other ventilation control devices and the cost of power to operate a ventilation system can be substantial. As mines expand or develop deeper work areas, ventilation costs can increase dramatically due to increases in the necessary fan pressure and air quantity. Fan pressure increases as mines develop because the resistance of the system increases as the length of the airways grows. Leakage, or inefficient use of air, increases the total quantity of air required for adequate ventilation and as mines develop, more leakage paths are generated. Additionally, high

pressure differentials throughout a mine can increase the amount of leakage and therefore the total air quantity circulated by the main fans.

The use of underground booster fans is one method of increasing the effectiveness of a ventilation system. Booster fans can reduce ventilation costs and increase the system efficiency by reducing the required main fan pressure and by decreasing system leakage. Booster fan systems are used commonly in underground coal mines in the United Kingdom, Australia, and other coal producing countries and these systems are considered safe, reliable, and essential. Though booster fans are used regularly in other countries, they are prohibited in most underground coal mines in the United States (30 CFR § 75.302 2010).

Recirculation of contaminated mine air is a topic closely associated with the use of booster fans. In coal mines, intake airways that carry uncontaminated air to the work areas are isolated from return airways that carry contaminated air away from the work areas using seals, bulkheads, doors, and brattice cloth. These methods of isolating intake airways from return airways form paths of high resistance, but these pathways do not completely stop airflow. In most coal mine ventilation systems, to prevent contaminants in the return air from entering the intake air, intake airways are maintained at a higher pressure than return airways, causing air leakage from the intake to the return. However, in ventilation systems that use booster fans, regions of the return airways can have higher pressures than the intake airways, causing air to leak from the return to the intake; this practice is known as recirculation.

Recirculation is generally classified into two categories: controlled recirculation, where a limited and known quantity of air is deliberately passed from the return airways to the intake airways, and uncontrolled recirculation, where a quantity of air is leaked from the return airways to the intake airways unintentionally. Controlled recirculation can be helpful in increasing the velocity of air at the production face without increasing the demand on the ventilation system significantly. Controlled recirculation can also help manage costs associated with heating and cooling air by reusing a portion of the conditioned air. Although there are benefits associated with recirculation, there is a notable risk: recirculation increases the contaminant concentration in intake air. If recirculation is well controlled, the concentration of contaminants in intake air can be managed, but if recirculation is uncontrolled, there is the potential for contaminants to build up in the intake air, potentially forming a hazardous mine atmosphere.

1.2 Problem Statement

Booster fans have the potential to be a safe means of enhancing the capacity of a ventilation system and increasing the overall system efficiency. Since the prohibition of booster fans in the United States in the 1980s, there has been limited research about booster fan ventilation systems and controlled recirculation in underground coal mines. Additionally, because booster fans are accepted as a safe and effective means of ventilating coal mines in other developed mining countries including Australia and the United Kingdom, current research about the use of booster fan ventilation systems is limited. For booster fans to be considered for use in underground coal mines in the

United States, current research about the effects of booster fans on ventilation systems is needed.

Although there are similarities among the mining technologies and practices in the United States, the United Kingdom, and Australia, there are legal and practical dissimilarities that have caused each country to approach coal mine ventilation differently. Practices that contribute to the safe use of booster fans in the United Kingdom and Australia need to be identified and evaluated to determine the applicability of these practices to U.S. coal mines.

Increasing the capacity and efficiency of a ventilation system is one of the main motives for using booster fans, but as the efficiency of the system is increased by the use of booster fans, recirculation is more likely to occur. In fact, many ventilation systems that use booster fans experience a significant amount of recirculation. Most underground coal mines in the United Kingdom rely on booster fans and recirculation to provide adequate air quantities and velocities; however, in the United States, recirculation is not an accepted ventilation practice. Methods to limit recirculation in ventilation systems using booster fans need to be evaluated.

Research defining how system pressure, air power, and quantity are affected by booster fans; describing how system efficiency and recirculation are affected by the location, placement, and size of the booster fans; and identifying the relationships between booster fans and main surface fans in ventilation systems that are consistent with U.S. mining conventions is presented in this study.

1.3 Objective

The objective of this thesis is to quantify how the efficiency of ventilation systems can be increased with the addition of booster fans and to demonstrate that system recirculation, one of the primary risks associated with the use of booster fans, can be controlled by adequately sizing and position of booster fans. First, specific practices and regulations pertinent to the use of booster fans and recirculation in Australia, the United Kingdom, and the United States are identified and discussed. The effects of main fan pressure and booster fan pressure on system leakage and system recirculation based on studies of a scaled coal mine model are presented. Next, data from computer ventilation simulators are used to evaluate the effects of fan size, fan location, and resistance on leakage and recirculation. Finally, computational fluid dynamics (CFD) models are used to demonstrate how the system efficiency is affected by fan placement and installation parameters including the site geometry and bulkhead materials.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

The development of efficient mine ventilation systems is a complex engineering problem that affects the safety and cost of operating underground mines, and booster fans can play a role in managing ventilation costs. The following literature review presents a summary of the use of booster fans in coal mine ventilation systems as well as the benefits and problems associated with the use of air recirculation. First, some fundamental principles of mine ventilation are presented. Next, characteristics of single fan and multiple fan ventilation systems are discussed. Subsequently, air leakage and air recirculation, both of which can significantly impact efficiency and safety, are introduced. Finally, several topics related to the use of multiple fan systems in need to further research are identified.

2.2 Basic Equations

2.2.1 Atkinson's Equation

Dynamically, mine ventilation systems are treated almost exclusively as systems of incompressible fluid flow and are described most often through Atkinson's equation, commonly given by:

$$\Delta p = \left[\frac{KO(L+L_e)}{A^3} \right] Q^2 \quad (2.1)$$

where

Δp = pressure difference, Pa

K = friction factor, kg/m³

O = perimeter, m

L = length, m

L_e = equivalent length to account for shock losses, m

A = cross-sectional area, m²

Q = volumetric flow rate, m³/s

The parameters in brackets in Equation 2.1 are generally referred to jointly as resistance, R , with units of N s²/m⁸ (Hartman et al. 1997). The volumetric flow rate in Equation 2.1 is generally calculated using:

$$Q = VA \quad (2.2)$$

where

Q = volumetric flow rate, m³/s

V = velocity, m/s

A = cross-sectional area, m²

Additionally, in mine ventilation systems, the velocity of the air flowing through the airways is generally measured using vane anemometers and the velocity of air flowing through the fan inlet or outlet duct is calculated from velocity pressure measurements using:

$$p_v = \frac{\rho V^2}{2} \quad (2.3)$$

where

p_v = velocity pressure, Pa

ρ = fluid density, kg/m³

V = velocity, m/s

Based on Atkinson's equation, resistance is directly proportional to airway length and inversely proportional to airway cross-sectional area. Over 70 % of typical mine resistance is due to friction between air and rock that forms the airways (Ramani 1992). As mines develop, the airway lengths and therefore the resistance increase, which increases the pressure necessary to produce sufficient air quantity. Additionally, it is typical for coal mines to develop multiple parallel airways including intake, neutral, and

return airways. Most coal mines have a minimum of three parallel airways, one intake, one neutral, and one return; in the main development sections, multiple intake, neutral, and return airways are often used. However, deep coal mines, such as those in the western United States, experience high rock stresses. To manage the stresses and prevent ground failure, these mines often use only two parallel airways and the airways tend to have small cross-sectional areas (Agapito and Goodrich 2000). Because of a limited number of airways and airways with small areas, deep mines have high resistance. High resistance in extensive and deep mines is one factor that contributes to the need for substantial fan pressure to deliver adequate air quantity.

In addition to airway length and area, resistance is also dependent on the friction factor, K . Friction factors are related to the roughness of an airway and in practice they are determined experimentally based on pressure and quantity measurements. There has been a significant amount of research in the United States to determine standard friction factors for coal mines; for straight, rock-bolted coal mine airways, friction factors ranging from 0.00798 kg/m^3 to 0.0113 kg/m^3 have been suggested by Ramani and Kharkar, McPherson, and Prosser and Wallace (1973; 1993; 1999).

2.2.2 Kirchhoff's Circuit Laws

Kirchhoff's circuit laws are most commonly applied to engineering problems involving electrical circuits. Ventilation systems can also be viewed as closed loop circuits and Kirchhoff's laws are integral to the solution of most ventilation network problems. Kirchhoff's first law deals with the conservation of electrical charge and states

that the total flow of current into a node is equal to the total flow of currents away from a node, where a node is defined by the intersection of two branches of a circuit. In ventilation circuits, the concept of current is replaced by mass flow rate or volume flow rate given a constant air density. So, the flow into a ventilation node, where a node is defined by the intersection of two airways, is equal to the flow rate away from the node. For ventilation circuits, Kirchhoff's first law is generally summarized as:

$$\sum Q = 0 \quad (2.4)$$

where

$$Q = \text{volumetric flow rate, m}^3/\text{s}$$

Kirchhoff's second law states that the voltage drop around any closed electrical circuit is zero. In ventilation circuits, the concept of voltage is replaced by pressure difference, so the pressure change around any closed ventilation circuit must be zero. Kirchhoff's second law must account for pressure losses in the system, pressure gains in the system generated by a fan, and natural ventilation pressure caused by pressure and temperature differences in air as it passes through the ventilation circuit. Natural ventilation pressure can work with the ventilation system as a positive pressure source or work against the ventilation system as a negative pressure source (McPherson 1993; Hartman et al. 1997). For ventilation circuits, Kirchhoff's second law is summarized as:

$$\sum p_i - p_f \pm p_n = 0 \quad (2.5)$$

where

p_i = pressure difference in the i th branch of a closed circuit, kPa

p_f = pressure increase due to a fan, kPa

p_n = natural ventilation pressure, kPa

2.2.3 Fan Efficiency

In mine ventilation systems, fans are used to generate a pressure difference (Δp) that drives airflow (Q). The pressure difference and air quantity define the power consumed by the ventilation system through the following relationship:

$$P = Q \Delta p \quad (2.6)$$

where

P = power, kW

Q = volumetric flow rate, m³/s

Δp = pressure difference, Pa

In this form, power is generally referred to as the air power, because it identifies the power provided by the fan to the air, neglecting the overall fan efficiency where overall fan efficiency is given by:

$$\eta = \frac{P}{P_i} * 100 \quad (2.7)$$

where

η = overall fan efficiency, %

P = air power, kW

P_i = input power, kW

Power consumption is an important factor that directly affects the cost of operating a ventilation system (de Nevers 2000).

2.2.4 Volumetric Efficiency

Leakage is an important parameter in coal mine ventilation systems because it can significantly reduce the efficiency of air use. Leakage occurs because seals, door, bulkheads, and brattice cloth do not perfectly isolate return airways from intake or neutral airways so a quantity of air leaks through these structures and short circuits the system. Air that is leaked through these barriers decreases the quantity of air flowing to the development sections (McPherson 1993). The quantity of air leaked through ventilation control devices is dependent on the pressure difference across the structure and though some amount of leakage is expected, it decreases the volumetric efficiency of the system where volumetric efficiency is generally defined as:

$$\eta_v = \frac{\sum Q_w}{Q_f} * 100 \quad (2.8)$$

where

η_v = volumetric efficiency, %

Q_w = volumetric flow rate at each working area, m^3/s

Q_f = volumetric flow rate at the main fan(s), m^3/s

The volumetric efficiency is an important ventilation parameter because power increases proportionally to the cube of the volumetric flow rate. For example, if leakage increases the required volumetric flow rate by 5%, power requirements increase by close to 16%. Typical coal mine ventilation systems have volumetric efficiencies near 40% (Hairfield and Stinnette 2009).

2.3 Mine Fans

Coal mine ventilation systems generally contain single or multiple surface fans connected to underground mine airways. Fans are used to generate a pressure difference that induces airflow throughout the mine. Fans are either axial vane type in which air exits the fan in the same direction it enters the fan, or centrifugal type in which air exits the fan perpendicular to the direction it enters the fan (McPherson 1993). Figure 2.1 demonstrates a typical layout for an axial surface fan and Figure 2.2 demonstrates a typical layout for a centrifugal surface fan.

Ventilation systems can be blower type that use fans to generate a positive gage pressure at the intake forcing air into the system or exhaust type that use fans to create a negative gage pressure at the return drawing air out of the system (Hartman et al. 1997). There are advantages to both types of ventilation systems: blower systems generate pressures higher than atmospheric and this prevents gases trapped in the caved area (gob) from entering the mine atmosphere and exhaust systems form pressures lower than

atmospheric and this tends to draw gases into the mine atmosphere where they can be diluted (McPherson 1993). Blower systems have several disadvantages. Because they operate at high pressure near the intake, air lock doors are required and these doors can contribute to excessive air leakage and lower system efficiency. Also, in blower systems, if the fan fails, the relative pressure drops and large amounts of gas can be liberated (Hartman et al. 1997). Exhaust ventilation systems are used most commonly in coal mines in the United States.

2.3.1 Surface Fans

In ventilation systems with a single main fan, the system operating point is defined by the intersection of the mine characteristic curve, based on Atkinson's equation, and the fan characteristic curve. As mining progresses, the total resistance is increased, the mine characteristic curve becomes steeper, and the operating point moves up the fan curve, reducing the total air quantity and increasing the system pressure (Wallace and McPherson 1983). This concept is illustrated in Figure 2.3.

In single fan ventilation systems, the total pressure produced by the mine fan is equal to the total pressure consumed by friction and shock losses in the ventilation network and the velocity pressure at the fan discharge (Hartman et al. 1997). The total system pressure decreases continuously from the intake to the system exhaust. Figures 2.4 (a) and (b) illustrate this concept schematically for two ventilation systems: blower and exhaust. Mathematically, total pressure is the sum of static pressure and velocity pressure. Throughout a ventilation system, velocity pressure can be converted to static

pressure and static pressure can be converted to velocity pressure, but in general, the total pressure continuously decreases as the distance through the mine increases.

2.3.2 Booster Fans

In deep and extensive mines, it is often not feasible to generate adequate airflow without the use of high-pressure surface fans and multiple surface connections. High pressure fans can contribute to excessive air leakage and can significantly increase the fan operating cost (McPherson 1993). Booster fans, fans installed underground, represent an alternative to high-pressure surface fans. Unlike surface fans which significantly increase the pressure near the intake or exhaust, booster fans are designed to balance the pressures throughout the ventilation network, reducing leakage and increasing flow capacity and system efficiency (Robinson 1989).

Booster fans are generally installed in series with the main surface fan and affect the pressure and air quantity in a section of a mine (McPherson 1993). Although a booster fan is generally installed in series with a main surface fan, the quantity of air passing through a booster fan is usually less than the quantity of air passing through the main fan. A booster fan operating pressure can be significant, up to several kilopascals, but this pressure is generally less than the operating pressure of a main surface fan (Robinson 1989).

Like surface fans, booster fans can be either axial or centrifugal fans. Axial type booster fans can be installed as custom built single units or as cluster units, groups of fans installed in a parallel and series arrangement. Figures 2.5 and 2.6 demonstrate plan views

of a single axial fan installation and a single centrifugal fan installation, respectively. Both of these installations include bulkheads, which are necessary to limit local recirculation around the fan, and an airlock system, which allow the fan to be accessed readily and bypassed in the event of a fan failure to prevent the complete loss of airflow in a section of the mine. Figure 2.7 (a) and (b) show schematics of a cluster fan installation. Clusters of booster fans often have relative low efficiency, less than 40 %, but they are relatively easy to install and maintain. To increase fan efficiency, custom built fans are used and though these fans tend to be expensive and require comprehensive designs, fan efficiencies of greater than 80% offset the high costs (Cory 1982; Burrell and Bennett 1995).

Additionally, jet fans, free-standing fans that blow air at high velocity, have been suggested for use as booster fans (Wolski 1995). Though jet fans may increase air quantity, the pressure produced by a jet fan is much less than the pressure produced by custom built and cluster booster fans, and because jet fans are free-standing and not generally installed in bulkheads, the risk for air recirculation around the fan is quite high (Burrell and Bennett 1995; Wolski 1995). There is little literature supporting the use of jet fans as booster fans in practice.

Booster fans are used in ventilation systems in metal/nonmetal mines in many countries including the United States, the United Kingdom, Canada, South Africa, and Australia (McPherson et al. 1985; De Souza et al. 2003). Booster fans have also been used in coal mines in India, the United Kingdom, and Australia (Kumar et al. 1991; Jobling et al. 2001; Martikainen and Taylor 2010). However, in the United States,

booster fans are prohibited from use in most underground coal mines, except for anthracite mines (30 CFR § 75.302 2010). Despite the prohibition of booster fans, several mine operators have expressed interested in the use of booster fans: both Jim Walter Resource's No. 7 Mine and Consolidation Coal Company's Loveridge No. 22 Mine have submitted petitions to the Mine Safety and Health Administration (MSHA) to permit the use of booster fans in their ventilation systems; both petitions have been denied (Sartain and Stevenson 1989; Langton 2003; Martikainen and Taylor 2010).

2.3.2.1 Neutral point. Booster fan ventilation systems are often designed around the concept of a neutral point. A neutral point occurs where the pressure in the intake is equal to the pressure in the return so there is no tendency for air to leak from the intake to the return or to recirculate from the return to the intake. In single fan ventilation systems (blower or exhaust type), the pressure in the intake airways is always greater than the pressure in the return airways because pressure decreases continuously throughout the system, so there is no potential for the development of a neutral point. Unlike single fan ventilation systems, in systems with booster fans, pressure is not a strictly decreasing function. Figures 2.8 and 2.9 demonstrates the pressure profile of main and booster fan systems; in these systems, pressure decreases from the main fan to the booster fan, pressure increases at the booster fans, and finally, pressure decreases to the systems' exhaust. Because pressure is not a strictly decreasing function, there is a potential for the creation of a neutral point. The location of a neutral point is noted in Figure 2.9. In ventilation systems that have multiple booster fan installations located in multiple sections of a mine, there is potential for the occurrence of multiple neutral points.

The concept of a neutral point is important because, installing booster fans at the neutral point minimizes system leakage and maximizes system efficiency without inducing recirculation. If a booster fan is located too far out by the neutral point, system efficiency is lost; however, if a booster fan is located too far in by a ventilation system, past the neutral point, recirculation will occur. Although the concept of neutral point is useful, in practice the position of the neutral point is not fixed and changes as the system parameters, including resistance and fan performance, change (Robinson 1989).

2.3.2.2 Recirculation. Recirculation is an issue that is closely related to the concept of a neutral point. Recirculation occurs when the pressures in the return airways are higher than the pressures in the intake airways causing air to leak from the returns to the intakes. In single fan ventilation systems, intake airways are always maintained at higher pressures than the return airways and these pressure differences cause air to leak from the intakes to the returns through bulkheads, doors, and other ventilation control devices. In systems with multiple fans, if the booster fan is located in by the neutral point, recirculation will occur in the region of the mine between the neutral point and the booster fan. Figure 2.9 illustrates the pressure profile of a booster fan system with recirculation; the region where recirculation occurs is shown in red and the neutral point is noted.

Recirculation is discussed primarily in two forms: controlled recirculation and uncontrolled recirculation. In controlled recirculation systems, a portion of the return air is purposefully directed into the intake air and transmitted to the production areas and the quantity of recirculated air is closely monitored and managed (Calizaya 2009). Controlled

recirculation can increase the capacity of a ventilation system by increasing the air quantity and air velocity near the production areas (Marks 1989). Uncontrolled recirculation occurs when air is leaked from the return airway to the intake airway but the leakage is not expected or planned and the quantity of recirculating air is not managed, so there is the potential for a buildup of harmful air contaminants including gasses, dust, and heat in a section of the mine.

Controlled recirculation systems are used commonly in deep mines and mines that require heating or cooling of air. In these mines, recirculation is used to increase face velocities and to help manage cost associated with conditioning the air (Aldred et al. 1984; Fleetwood et al. 1984; Rose 1992; Hardcastle and Kocsis 2004). In addition, although air recirculation is prohibited in coal mines in the United States, controlled recirculation has been used successfully in coal mines in the United Kingdom, South Africa, and India (Robinson and Harrison 1987; Kumar et al. 1991; Meyer 1993).

Controlled recirculation has been fairly well researched. Calizaya, McPherson, and Mousset- Jones presented result of an experimental study of two controlled recirculation systems: one system with booster fans placed in-line with the main fans and one system with booster fans located in a crosscut (1991). Both in-line and cross-cut systems were used to induce system recirculation and increase the volume of air at the face and both systems were effective at managing gas concentrations; however, the in-line recirculation system was found to be more effective at limiting the gas concentration in intake air.

Cecala et al. studied a controlled recirculation system used to enhance the ventilation system at the Texasgulf Trona Mine (1991). In this system, recirculation was induced by fans located in crosscuts. Although the use of a controlled recirculation system at the Texasgulf mine was ultimately denied (Teaster 1999), results from this study indicated that controlled district recirculation could have been an effective method of increasing the capacity of the ventilation system and that the recirculation system could maintain acceptable concentrations of carbon monoxide, carbon dioxide, methane, and respirable dust.

Hardcastle, Kolada, and Stokes have demonstrated the need for booster fans and for controlled recirculation in high resistance mines with large production and large flow requirements (1984). These researchers noted that controlled recirculation does increase the concentration of contaminants including heat, gases, and dust in intake air; however, they note that when the quantity of fresh air is maintained, recirculation does not increase the maximum contaminant concentration in the return air. In addition, Lee and Longson observed that recirculation often increases the velocity of the airflow in areas where methane production can be high, resulting in better air mixing and improved dilution (1987).

The concept that the maximum concentration of contaminants in the return air is independent of the amount of air recirculation has also been demonstrated by McPherson (1988). Although McPherson's study deals specifically with the effect of controlled recirculation on methane concentrations in auxiliary ventilation systems, most of the

results are applicable to system recirculation. In general, the concentration of contaminants in the return is given by:

$$C = \frac{G_i + G_h}{Q} * 100 \quad (2.9)$$

where

C = concentration of contaminant, %

G_i = volumetric flow rate of contaminant into intake air, m³/s

G_h = volumetric flow rate of contaminant at the production heading, m³/s

Q = volumetric flow rate of return air, m³/s

G_i , G_h , and Q are illustrated schematically in Figure 2.10. In a system with no recirculation, if the flow rate of air into a production section is 10 m³/s and the methane concentration is 0.1%, then G_i is 0.01 m³/s and if the gas emission rate at the production heading is 0.05 m³/s, then the concentration of methane in the return air is 0.6 %.

Considering a system with 10 % recirculation, in this case the flow rate of air into the production section is 11 m³/s. The methane concentration of the intake air is 0.15%, so G_i is 0.016 m³/s. The concentration of methane in the intake air is higher in the system with recirculation than in the system with no recirculation because 10% of the air has a methane concentration of 0.6%. Again, assuming the 0.05 m³/s gas inflow at the production heading, the concentration of methane in the return air in a system with 10% recirculation is 0.6 %. This example demonstrates that recirculation increases the concentration of contaminants in the intake air but, as long as the initial quantity of intake

airflow is maintained, recirculation does not affect the concentration of contaminants in the return.

Although controlled recirculation can be beneficial for increasing the velocity and quantity of air near the production areas, there are risks associated with both controlled and uncontrolled recirculation. Controlled recirculation can increase contaminants to unsafe levels in ventilation systems with innately high heat, gas, or dust (Alderred et al. 1984; Hall et al. 1987). Controlled recirculation has the potential to recirculate high concentrations of contaminants generated by transient events such as the stopping or starting of a booster fan or the release of a large quantity of gases from the production heading or the longwall gob (Longson et al. 1987; Lowndes 1987; Calizaya et al. 1991). Perhaps the most significant risk in controlled recirculation is the recirculation of contaminants produced by mine fires (Burton et al. 1986; Marks 1989; Cecala et al. 1991). Fires and uncontrolled recirculation have been associated with several notable mine catastrophes. Recirculation was noted as the cause of 63 deaths in the Mauricewood Disaster in 1889 in the UK (Gracie and Job 1984). Booster fans and uncontrolled recirculation contributed to the severity of the Sunshine Mine fire in Idaho and the Auchengeich Colliery fire in Scotland (Jarrett 1972; Rogers 1960).

2.3.2.3 Safety and monitoring. In mines using booster fans and controlled recirculation, robust atmospheric monitoring systems (AMS) are almost always used to ensure adequate air quality is maintained (Robinson 1989). Because fires in underground ventilation systems are a significant risk, monitoring systems usually include methane monitors, to limit the potential for the formation of an explosive

atmosphere, and carbon monoxide monitors, to detect the early stages of combustion (Allan 1983; Middleton et al. 1985; Challinor 1987). Although underground coal ventilation systems in the United States do not use booster fans or controlled recirculation, many U.S. mines use AMS to monitor carbon monoxide and methane (Francart 2005). AMS in mines using booster fans and controlled recirculation also often include fan pressure and vibration sensors, motor and bearing temperature monitors, air velocity monitors, and gas, dust, and smoke sensors, including tube bundle air sampling systems; additionally, the components of the monitoring system are physically inspected regularly (Burton et al. 1986; Pearce 1987; Calizaya 1989). In some cases, where the risk of fire from spontaneous coal combustion is unusually high, nitrogen is injected into the ventilation air to help stabilize the atmosphere (Leeming et al. 2008).

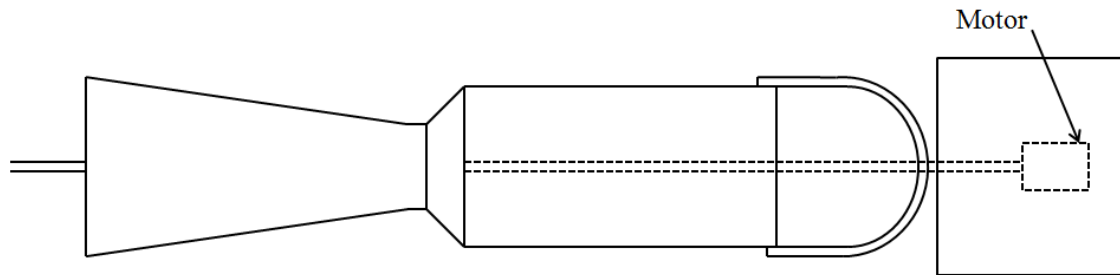
In an effort to address the risks associated with the use of booster fans and controlled recirculation, numerous guidelines have been proposed in addition to the use of AMS. Booster fans should not be oversized and the fan locations should be chosen carefully so that the fans do not contribute to excessive system leakage and uncontrolled air recirculation (Calizaya et al. 1990; Krog 2002; Detharet 2006). To minimize the risk of uncontrolled recirculation, ventilation planning should include efforts to identify potential recirculation paths (Calizaya et al. 1990). It has been suggested that surface fans and booster fans should be powered by an independent power supply and that the fans should be electrically interlocked so that a fan failure does not cause the remaining fans to stall (Calizaya et al. 1990). In some instances, to prevent high concentrations of gas from passing through the fan power system, the motors are ventilated with intake air

(Brake and Nixon 2006). Some mines use electrical interlocking with the AMS so that if high concentrations of gas are detected, electrical equipment downstream is de-energized and the booster fans are disabled to allow flow through ventilation and prevent the recirculation of contaminants (Burton et al. 1986). Procedures for powering fans are all intended to address risks associated with booster fans and recirculation; unfortunately, these practices are not consistent.

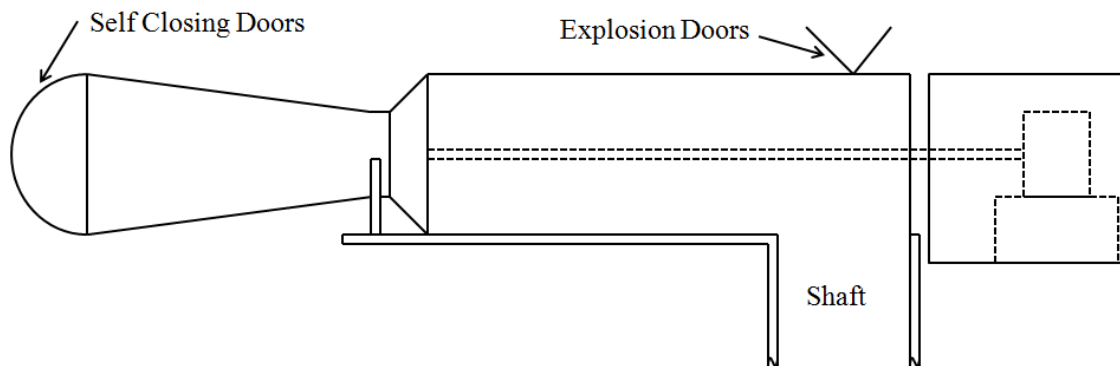
2.4 Conclusion

Current mine ventilation research is primarily focused on procedures for reducing mine hazards, optimizing ventilation systems, and reducing operating costs. Controlling total pressure and total quantity of a ventilation system is critical to ensuring safe and efficient operation of the system. Booster fan ventilation systems represent a potentially practical option for increasing the efficiency of coal mine ventilation systems by balancing the total pressure and reducing leakage, thus reducing the total required air quantity.

Because booster fans have been expressly prohibited in underground coal mines in the United States, current research related to the use of these fans is limited. Studies defining the relationships among recirculation, leakage, pressure, quantity, and power that are consistent with mining practices in the United States need to be undertaken. Booster fan ventilation systems in countries where the safety and health standards are similar to the standards in the United States need to be analyzed to better define hazards associated with the use of booster fans and to help establish practices to minimize risks.



(a) Plan view of a typical axial vane fan system



(b) Side view of a typical axial vane fan system

FIGURE 2.1 Plan view (a) and side view (b) of a typical axial surface fan installation

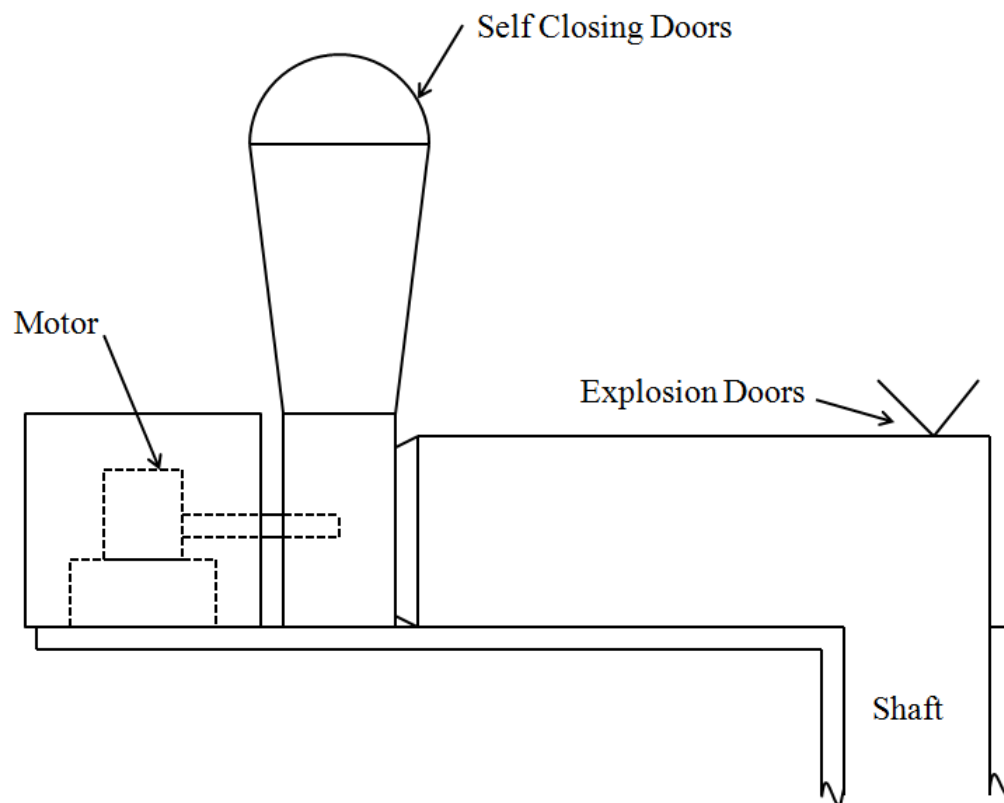
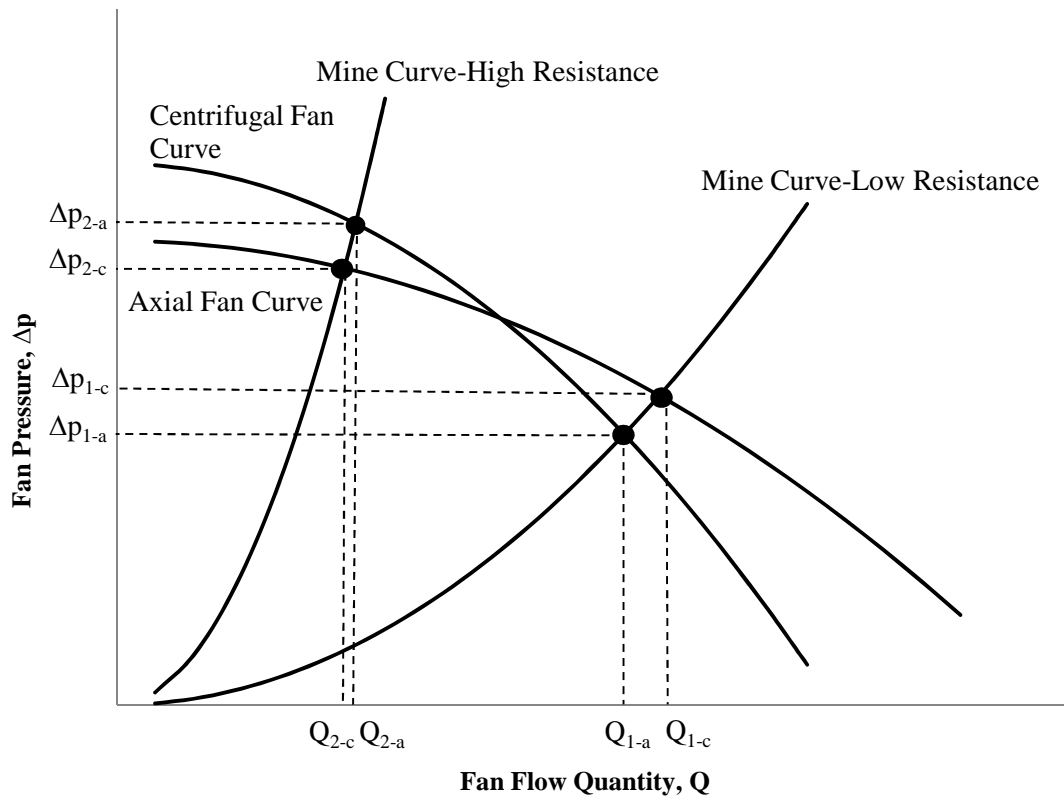
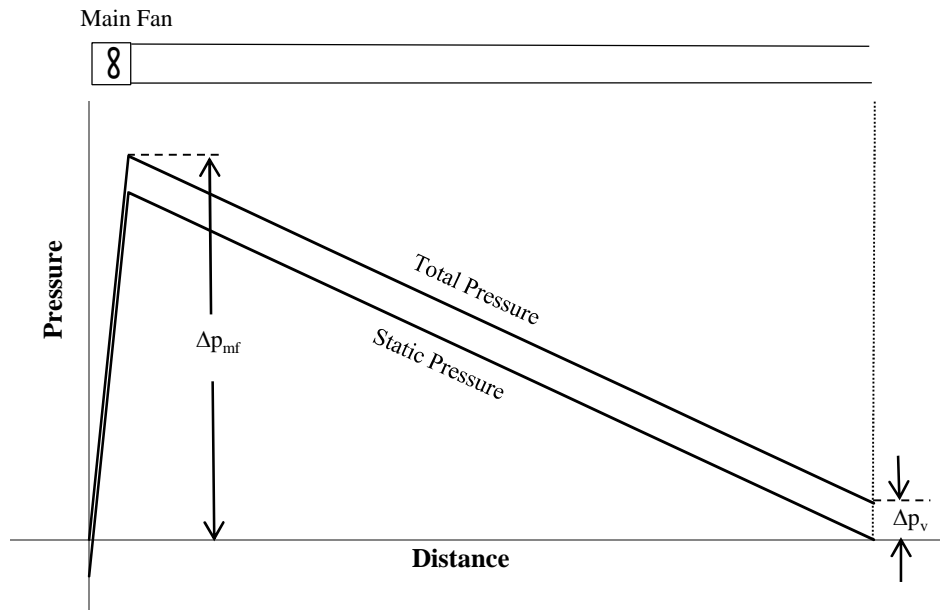


FIGURE 2.2 Side view of a typical centrifugal surface fan installation

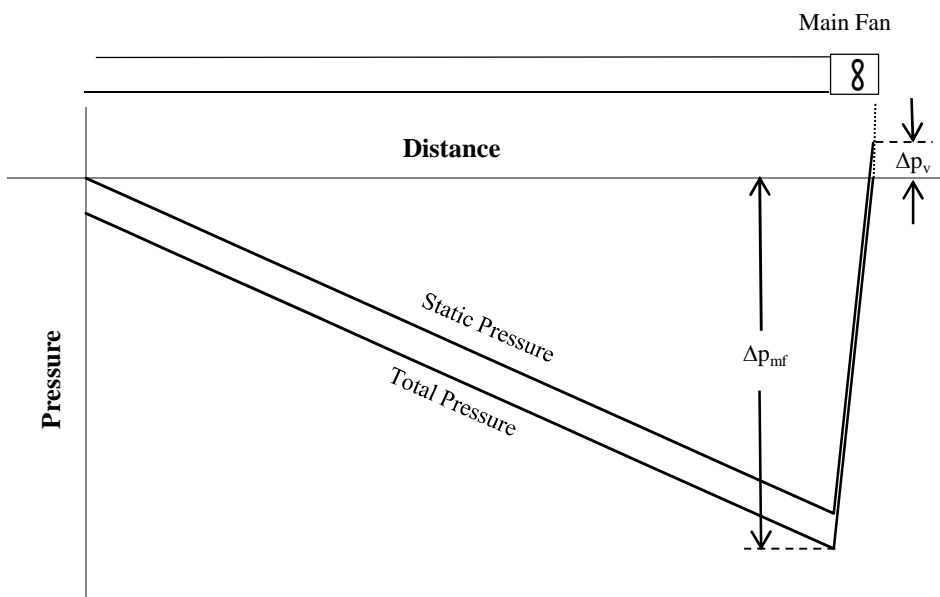


Adapted from: Wallace and McPherson 1983

FIGURE 2.3 Interaction between fan operating curves and mine characteristic curves



(a) Pressure profile of a blower ventilation system



(b) Pressure profile of an exhaust ventilation system

Adapted from: Hartman et al. 1997

FIGURE 2.4 Pressure profiles of (a) blower and (b) exhaust single fan ventilation systems

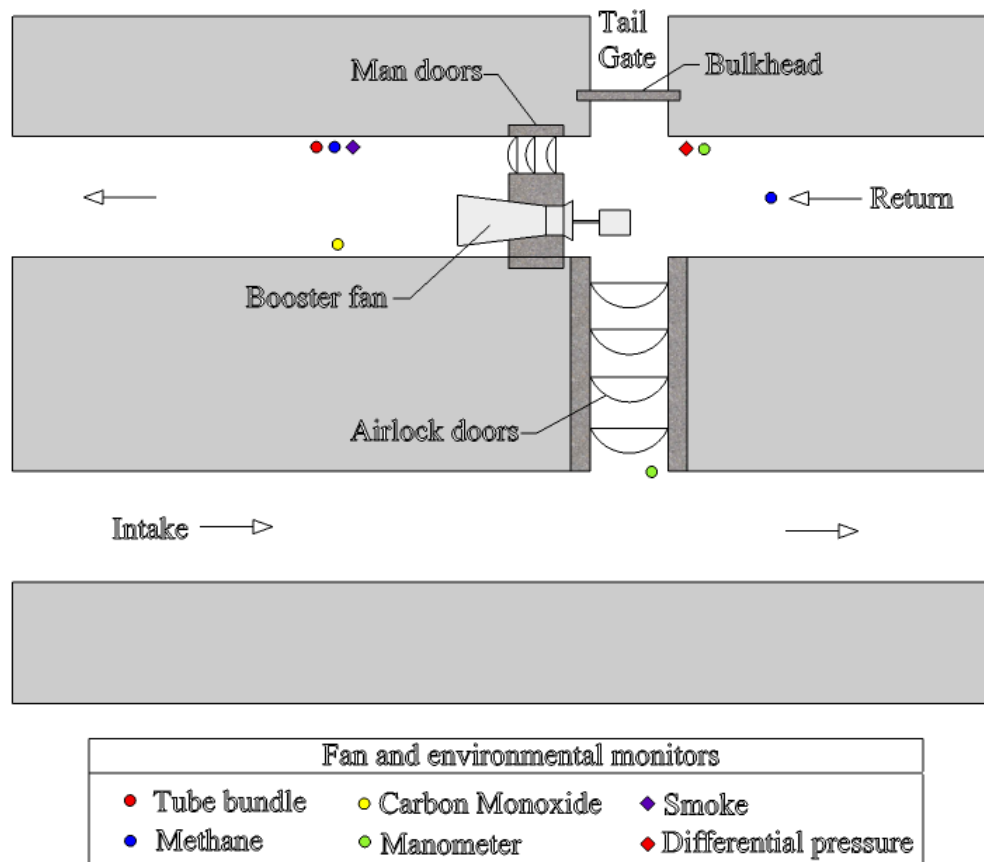


FIGURE 2.5 Plan view of a single axial booster fan installation

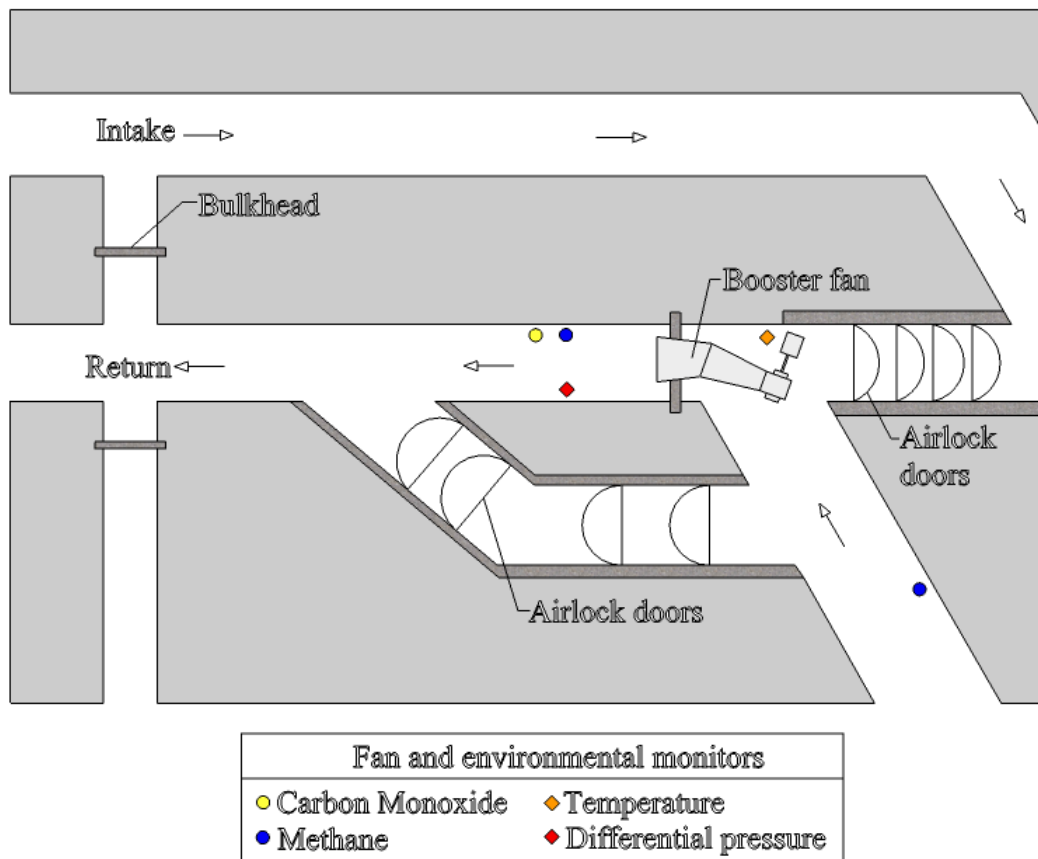
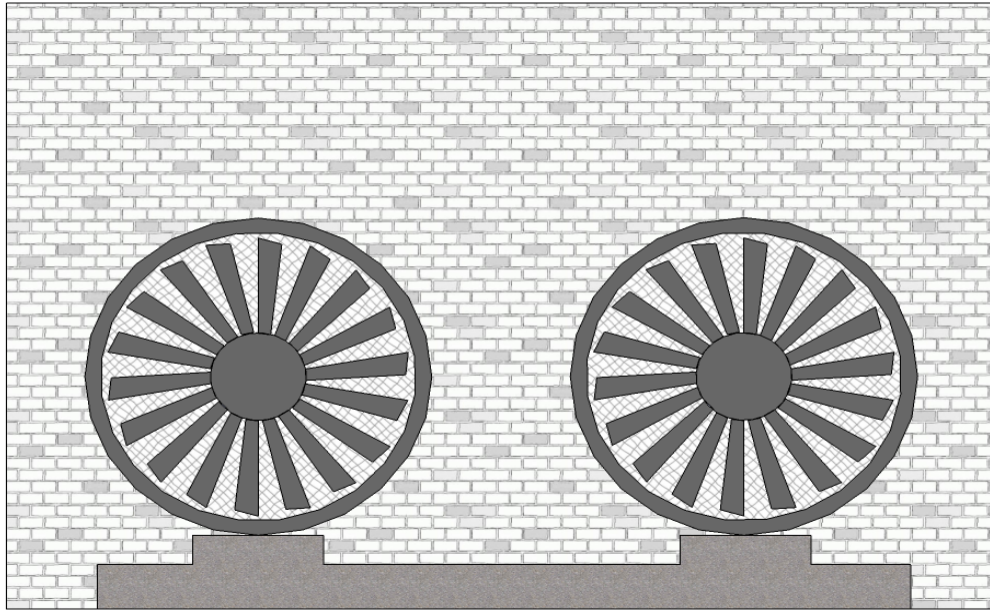
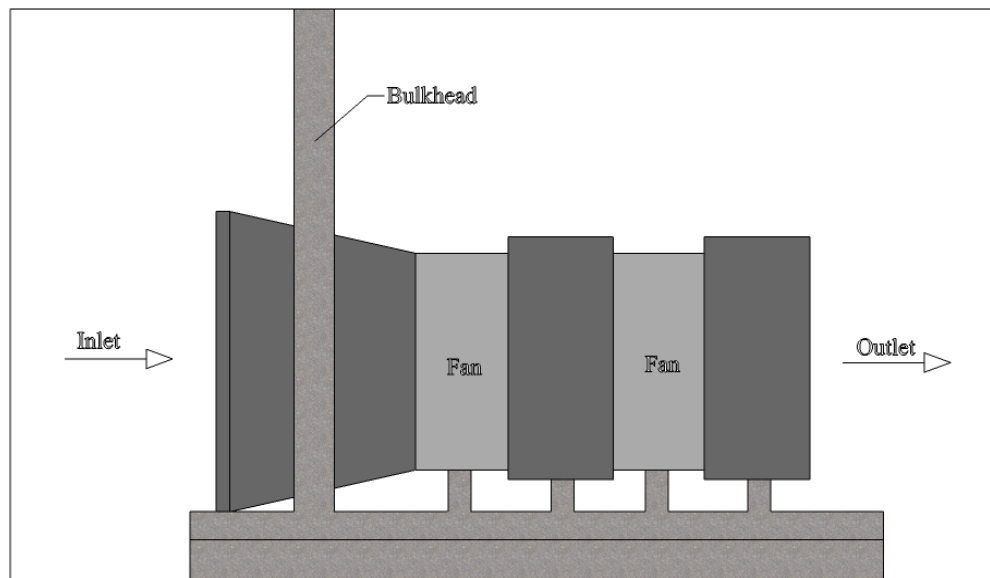


FIGURE 2.6 Plan view of a single centrifugal booster fan installation

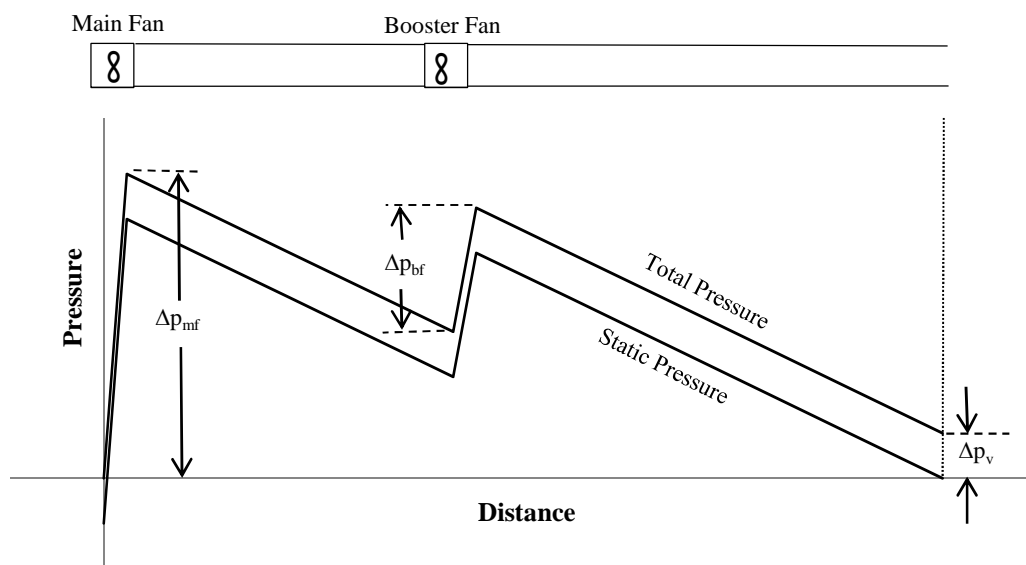


(a) Front view



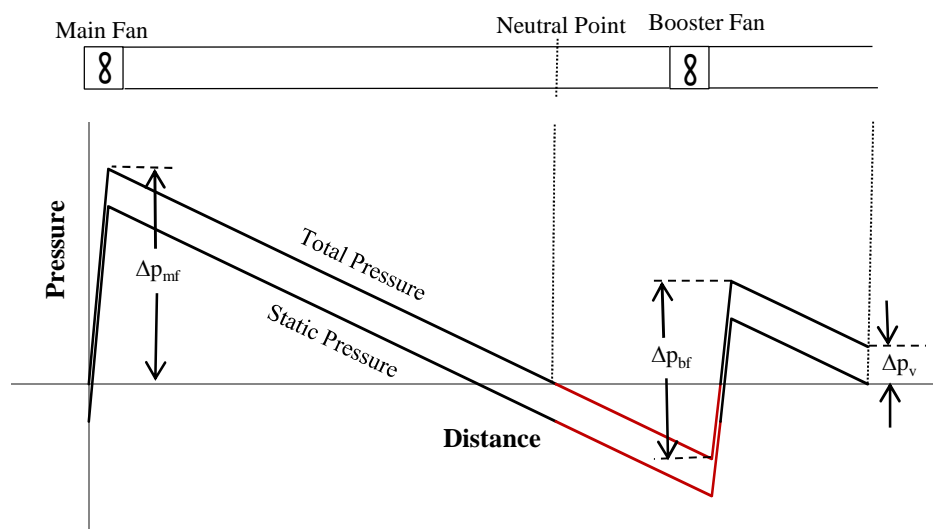
(b) Side view

FIGURE 2.7 Schematics of a cluster booster fan installation showing two sets of two axial fans



Adapted from: Hartman et al. 1997

FIGURE 2.8 Pressure profiles of a booster fan ventilation system



Adapted from: Liu 1985

FIGURE 2.9 Pressure profiles of a booster fan ventilation system with recirculation

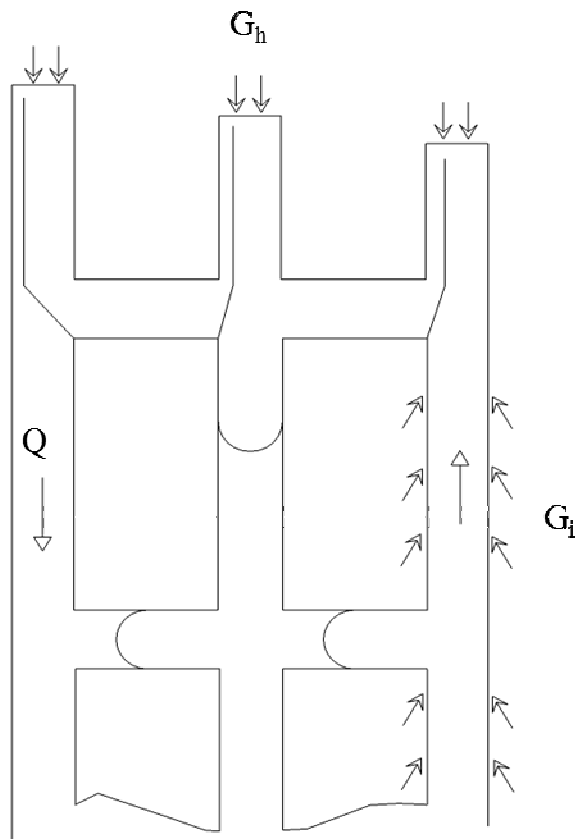


FIGURE 2.10 Schematic illustrating the variables Q , G_i , and G_h from Equation 2.7, where Q has a gas concentration of C

CHAPTER 3

VENTILATION PRACTICES

3.1 Introduction

Underground coal mines in the United States, the United Kingdom, and Australia face similar ventilation challenges. These countries have similar mine safety and health performance, and comparable production methodologies and technologies are employed in underground coal mines. Each country has developed legislation and regulations with the objective of increasing the safety of underground coal mines; however, the approach to mine regulation and the regulations in place are frequently dissimilar. In this chapter, the underground coal production rates in each country are identified; methods of coal production, typical mine geometries, and ventilation practices are discussed; and finally, regulations pertaining to face ventilation requirements, gas concentrations, booster fans, and air recirculation are compared.

3.2 Coal Production

In the United States, Australia, and the United Kingdom, coal remains an important source of energy. In 2009 in the United States, there were 1,375 operating coal mines, including 835 surface mines and 540 underground mines. The majority of coal

mines in the United States are small mines located east of the Mississippi River. However, most coal production occurs in large mines in states west of the Mississippi River including Arizona, Colorado, Montana, New Mexico, North Dakota, Texas, Utah, and Wyoming. The 50 largest mines account for about 60% of the total coal production per year. In 2009, the United States produced 972,000,000 tonnes with approximately 70% of the total production from surface mines and the remaining 30% produced by underground mines. Room (bord) and pillar mining accounts for approximately 50% of underground coal production and longwall mining accounts for the remaining 50% (U.S. Energy Information Administration 2009).

Like the United States, Australia is a major world producer and exporter of coal. The number of coal mines (collieries) in Australia is less than the number of mines in the United States, numbering in the hundreds rather than the thousands, and additionally, mines in Australia tend to be operated by global mining companies including BHP Billiton, Rio Tinto, Vale, and Xstrata (Australian Coal 2010). In 2002, Australia produced close to 349,000,000 tonnes of coal. About 75% of the production was from surface mines with the remaining 25% produced by underground mines. Longwall mines accounted for the majority of underground coal production while room and pillar mines accounted for only about 16% of underground coal production and 4% of total coal production (Moreby 2009).

Though coal is an important energy source in the United Kingdom, coal production is relatively low compared to the production in the United States and in Australia. In 2010, there were 10 underground mines which produced a total of

7,390,000 tonnes and 28 surface mines which produced a total of 10,426,000 tonnes. Total coal production in the United Kingdom was 17,816,000 tonnes. Nearly all of the underground coal mines in the United Kingdom use the longwall method as the primary means of production (MacLeay et al. 2011).

3.3 Mine Geometry and Ventilation Practices

In the United States, approximately half of the underground coal production comes from mines using the room and pillar method and half of the production comes from mines that use the longwall method. In the room and pillar method, coal is extracted from entries and crosscuts using continuous miners, leaving pillars of coal as the primary support structures. There are two common methods for ventilating the production sections in room and pillar mines: unidirectional (single split) systems, where the intake airways and the return airways are on opposite sides of the development panel, and bidirectional (double split) systems, where the intake air flows up the middle of the panel and the return air flows through the airways on both sides of the panel (Hartman et al. 1997). These methods are illustrated schematically in Figure 3.1 (a) and (b) for a five-entry room and pillar panel. In both unidirectional and bidirectional ventilation methods, the airway containing the conveyor belt is maintained with neutral airflow and ventilation control devices are used to separate the intake, neutral, and return airways. In unidirectional systems, there is only one leakage path from intake to return so these systems handle air more efficiently than bidirectional systems. Unidirectional systems also require fewer stoppings to segregate the intake, neutral, and return airways.

However, unidirectional systems can expose intake air to large amounts of gas from the side walls (ribs), so room and pillar mines that liberate a large amount of methane gas often use bidirectional systems to limit the exposure of intake air to methane.

In the longwall mining method, development entries are driven around a longwall panel using continuous miners, and a longwall mining machine is used to extract the coal between the headgate and the tailgate, allowing the roof behind the longwall to cave. In the United States, regulations prescribe that longwall panels are developed with at least two-entries. Systems with at least two-entries are required because belt air courses, entries containing the conveyor belt used to transport coal, cannot be used as the only air intake and because two separate and distinct travel ways must be maintained to act as escape ways during mine emergencies (30 CFR § 75.350 2010; 30 CFR § 75.380 2010). However, longwall panels are most often developed using three-entry systems. In three-entry systems on the headgate side of the longwall panel, adjacent to undeveloped coal, two entries are used as clean air intakes and one entry is maintained as a neutral airway and used to transport coal away from the development section. On the tailgate side of the longwall panel, adjacent to the gob, all three entries typically serve as return airways. While three-entry systems are more common in mines with large airflow requirements, two-entry systems are common in deep mines where additional airways can contribute to strata instabilities (Agapito and Goodrich 2000). In two-entry systems, the headgate generally includes one intake airway and one neutral airway and the tailgate consists of two return airways.

In longwall mines that are highly resistive or gassy, the entry containing the conveyor belt, which is typically neutral, is often used as an additional intake airway. Utilizing the belt entry as an addition intake reduces the resistance of the ventilation system, increasing the volumetric flow rate at the development section without increasing the demand on the main fan (Calizaya and Tien 2009). In order to use belt air to ventilate development sections, the belt entry must be equipped with an AMS, the miners must be trained in the operating of the AMS, the concentration of respirable dust in the belt entry must be maintained at or below 1 mg/m^3 , the primary escapeways must be monitored for carbon monoxide, and the areas ventilated with belt air must be developed with three or more entries (30 CFR § 75.350 2010). Belt air can be used to increase the ventilation capacity in two-entry systems, but only if a petition for modification for the use of belt air is granted. Although the use of belt air to ventilate workings can increase the effectiveness of ventilation systems, it does not significantly reduce leakage throughout the ventilation system. Because belt air ventilation does not significantly reduce leakage, as ventilation demands increase, booster fans may be a necessary alternative to belt air ventilation.

Most longwall mines in the United States use bleeder systems, where a quantity of the return air is forced past the gob to dilute or bleed off excess methane. If ventilation through the bleeders is not adequate, there is potential for explosive levels of methane to build up in the system which can perpetuate explosions underground and contribute to mining disasters such as the Upper Big Branch disaster in 2010 (Page et al. 2011). Figure 3.2 shows a typical two-entry retreating longwall ventilation system with bleeders. When

bleederless systems are used, the return entries are sealed as the longwall advances to limit airflow through the gob. Figure 3.3 shows a typical bleederless longwall ventilation system.

In Australia and the United Kingdom, the longwall method accounts for a much larger portion of underground coal production than in the United States. Both Australia and the United Kingdom use longwall ventilation systems that are similar to the ventilation systems in the United States, but there are several notable differences. In Australia, bleederless, multiple entry systems with intake and return airways are typical (Figure 3.3). Rather than passing a large quantity of ventilation air through a bleeder system, many Australian mines use boreholes to help degasify the gob (goaf) (Moreby 2009). In the United Kingdom, longwall ventilation systems typically use single entries, with one intake airway and one return airway also used to transport coal. In American and Australian mines, the headgate for one panel becomes the tailgate for the next panel as mining advances, but because U.K. mines use a single entry system, new intake and return airways are developed for each longwall panel with a barrier pillar of coal separating the longwall panels, and bleeder systems are not used. Additionally, ventilation systems in Australia and the United Kingdom do not use neutral airways. Figure 3.4 illustrates the typical longwall ventilation system for mines in the United Kingdom.

3.4 Regulatory Structure

In the United States, Title 30 Part 75 Subpart D of the Code of Federal Regulations (CFR) defines the requirements for underground coal mine ventilation (Federal Register 2010). In the United Kingdom, mine ventilation requirements are addressed in the Coal and Other Mines (Ventilation) Order 1956 and the Mines and Quarries Act 1954 (National Archives 1999; United Kingdom Statutory Instruments 1956). In Australia, regulations specific to coal mining vary by state: New South Wales (NSW) is regulated by the Occupational Health and Safety Act of 2000, the Coal Mine Regulation Act of 1982, Coal Mines (General) Regulation 1999, Mines Inspection General Rule 2000, and the Coal Mines (Underground) Regulations 1999; and Queensland is regulated by the Coal Mining Safety and Health Act 1999 and the Coal Mining Safety and Health Regulation 2001 (Moreby 2009).

In the United Kingdom, and more so in Australia, a performance-based approach to regulation that emphasizes risk management is practiced. In these systems, regulations are based on a duty of care principle: those who create risk have the primary responsibility for the safety of all parties exposed to the risk. Within this context of regulation, mine operators have the obligation to establish safe and healthy working conditions and practices, and to manage risks; however, regulations are broadly phrased and do not legislate precise and detailed practices. The structure of the U.K. and Australian regulatory systems contrasts sharply with the prescriptive approach to regulation used in the United States where procedures to mitigate risks are rigidly defined and stricter enforcement of regulations is viewed as a means of increasing the level of

mine safety (Lauriski and Yang 2011). The distinct difference between the Australian approach to mine regulation and the American approach to mine regulation is relatively new: prior to 1999, regulation in Australia was mostly prescriptive. However, in 1994 an explosion at the Moura No. 2 mine, an underground coal mine in Queensland, led to the deaths of 11 miners and this disaster prompted a comprehensive review of mine regulation in Queensland (Everson 2008). In Queensland in 1999, new legislation was enacted based on goal setting and risk management rather than prescribed standards (Gunningham 2006).

3.5 Ventilation Regulations

In the United States, coal mines are required to use surface fans for ventilation, and with the exception of anthracite mines, the use of booster fans is prohibited. There are fewer than 64 surface and underground anthracite coal mines in the United States and only 160,000 tonnes of coal are produced from underground anthracite mines, which accounts for less than 1% of the total underground coal production. Underground anthracite mines make up a very small sector of the underground coal industry and because booster fans are prohibited in all underground coal mines except anthracite mines, booster fans are illegal in the vast majority of underground coal mines in the United States.

In contrast, booster fans are permitted in coal mines in both Australia and the United Kingdom. In Queensland, Australia, mine operators must have a principal hazard management plan that defines procedures for using booster fans and action required if

methane levels reach 1.25% or higher. In the Australian state of NSW, which contains a high concentration of underground coal mines, booster fans can only be installed if the installation and operation of the fan have been approved by the mine inspectorate.

Booster fans are not referenced specifically in regulations in the United Kingdom, but they are classified as nonauxiliary, underground mine fans. For booster fans to be permitted in the United Kingdom, an extensive ventilation survey must be conducted to identify how the ventilation system is affected by booster fans, the need for booster fans to provide adequate mine ventilation must be demonstrated, and the mine inspectorate must be notified of the operator's intent to use booster fans so the installations can be approved and inspected. Interestingly, although recirculation is a topic closely associated with the use of booster fans, neither Australia nor the United Kingdom have regulations pertaining specifically to recirculation in relation to booster fans.

In 2009, two Australian coal mines were using booster fans. Figure 3.5 demonstrates an approved Australian booster fan system. In this system, the fans are located in a return airway, but ducting is used to ventilate the fan motors with intake air. The system also includes extensive environmental and fan monitors. In 2011, there were at least three coal mines in the United Kingdom using booster fans. Figures 2.4 and 2.5 demonstrate approved booster fan systems in the United Kingdom. Like the Australian system, the booster fans are located in the return airways and include extensive environmental and fan monitors. Unlike the Australian system, the fan motors in the United Kingdom are not ventilated with intake air but the motor housings are flame proof.

Although booster fans are allowed in the United Kingdom and in Australia, main surface fans are the primary source of ventilation. As such, given their critical life safety function, failure of these fans is given important consideration including fail safe power systems, failure warning systems, and backup fans. Main fan failure is particularly significant in systems using booster fans because if the main fan is disabled, a large amount of recirculation is likely to occur. The United States, the United Kingdom, and Australia all have regulations related to main fan failures. In the United States, if a surface fan stops, all the electrically powered and mechanized equipment must be disabled, and if the fan is stopped for more than 15 minutes and ventilation is not restored, the mine must be evacuated. Backup fans can be used to provide adequate ventilation in the event of a fan failure, but they are not required, and few U.S. mines maintain backup fans. In Australia, requirements in the event of a main fan failure are similar to the requirements in the United States but fan failure does not necessarily require mine evacuation. In the United Kingdom, mines are required to maintain standby surface fans that are capable of providing adequate airflow to allow a mine to be safely evacuated in the event of a main fan failure. Although it is not required, most mines in the United Kingdom maintain a standby fan that is capable of producing full volume flow. Like Australia, the failure of a main fan does not immediately require mine evacuation.

In addition to regulations regarding fans in underground coal mines, regulations defining adequate airflow in development areas and limiting contaminant concentrations throughout a mine are important because they help to define the necessary air capacity of

ventilation systems. Because the method of coal production can affect the amount of dust produced and the volume of gas liberated, the United States specifies different air quantities for continuous miner and longwall mining methods: at a minimum, 4.25 m³/s of airflow is required at the last open crosscut of each set of entries in continuous miner sections and 14.16 m³/s of airflow is required at a longwall working face. In New South Wales, continuous miner sections must maintain 0.3 m³/s for each square meter of roadway cross-section and longwalls must maintain 4 m³/s for each meter of extraction height. Queensland requires a velocity of 0.3 m/s in all areas that are identified as explosion risk zones (ERZ), including the working sections, where the concentration on methane is between 0.5% and 2% (Moreby 2009).

In contrast, the United Kingdom does not have a prescribed face quantity or velocity criteria; rather, it is the duty of mine operators to dilute gases and provide air containing sufficient oxygen. Flow volumes in the United Kingdom are dictated by limiting air contaminants to accepted levels: carbon dioxide must be maintained at a level less than 1.25% by volume, oxygen must be maintained at a level greater than 19% by volume, and methane must be maintained at a level less than 2% by volume in the return. Notably, as long as specified air quality is maintained, series ventilation, when air that has passed one working face is allowed to bypass a second face, is allowed in coal mines in the United Kingdom. Regardless of the air quality, ventilating multiple working sections with the same intake air is prohibited in coal mines in the United States and in Australia.

In addition to the use of prescribed flow rates, the United States and Australia also require that contaminants be limited to acceptable levels. Australia's limits for methane, oxygen, and carbon dioxide are identical to the limits required in the United Kingdom. In the United States, the eight-hour time weighted average (TWA) exposure to carbon dioxide is 0.5% with a 15-minute short term exposure limit (STEL) of 3% by volume. Oxygen must be maintained at 19.5% by volume. Methane concentrations in the intake airways must be maintained at less than 1% by volume, return airways must be maintained at less than 1.5% methane by volume, and bleeders must be maintained at less than 2% methane by volume. Maximum allowable levels of gas concentrations in the United Kingdom, Australia, and the United States are shown in Table 3.1.

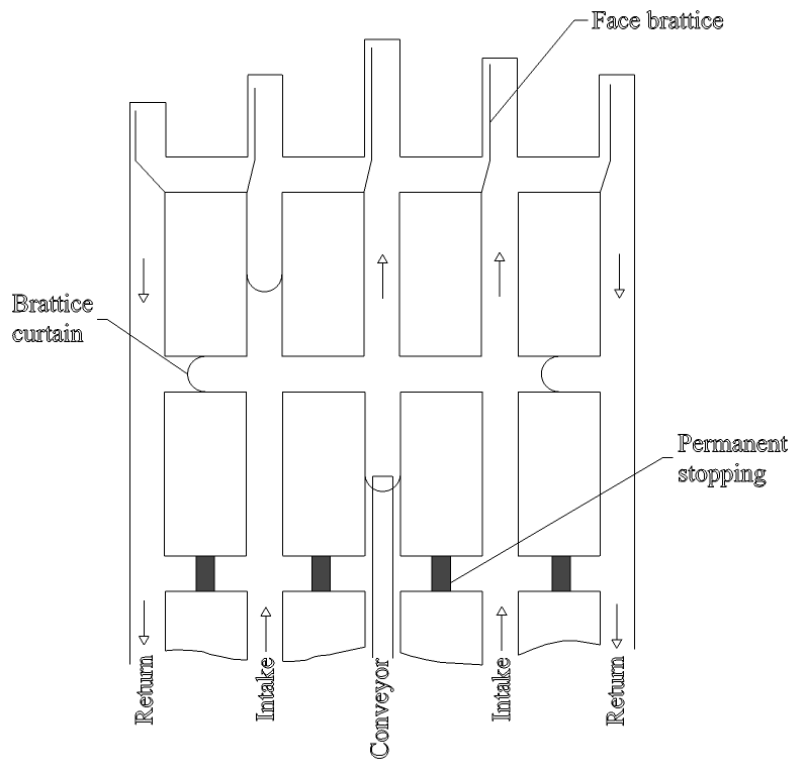
3.5 Conclusion

The United States, the United Kingdom, and Australia employ similar mining methods and technologies but ventilation practices and requirements vary significantly. Booster fans have been proven as an effective means of increasing ventilation capacity in mines in the United Kingdom and Australia; however, because the regulatory approach is performance-based with a primary focus on risk management, there are no strict procedures for safely installing booster fans or addressing recirculation in either country. Legalizing booster fans in the United States has the potential to increase the efficiency of hundreds of ventilation systems; however, because the United States has a prescriptive regulatory system, methods to limit recirculation and install the fans safely need to be addressed.

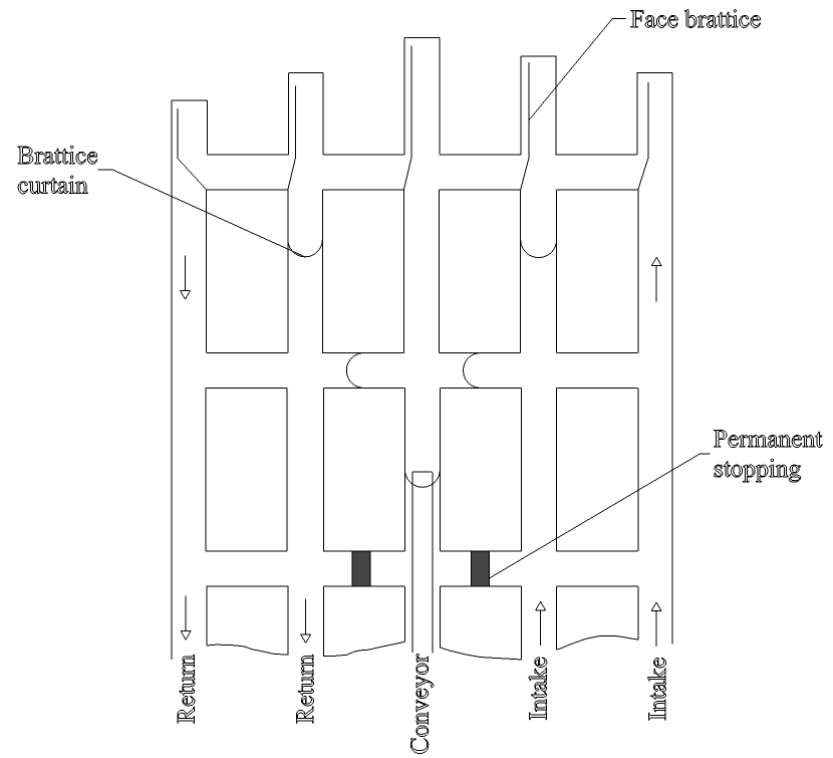
TABLE 3.1 Gas concentration limits as a percentage by volume

	CO ₂ %	O ₂ %	CH ₄ %	
			Face	Return
United Kingdom	1.25	19.0	1.25	2.00
Australia	1.25	19.0	1.25	2.00
United States	0.50	19.5	1.00	1.50

Adapted from: National Archives 1999; Moreby 2009; 30 CFR § 75.321 2010; 30 CFR § 75.323 2010



(a) Bidirectional system (double split)



(b) Unidirectional system (single split)

Adapted from: Hartman et al. 1997

FIGURE 3.1 Bidirectional (a) and unidirectional (b) ventilation systems for a five-entry room and pillar development panel

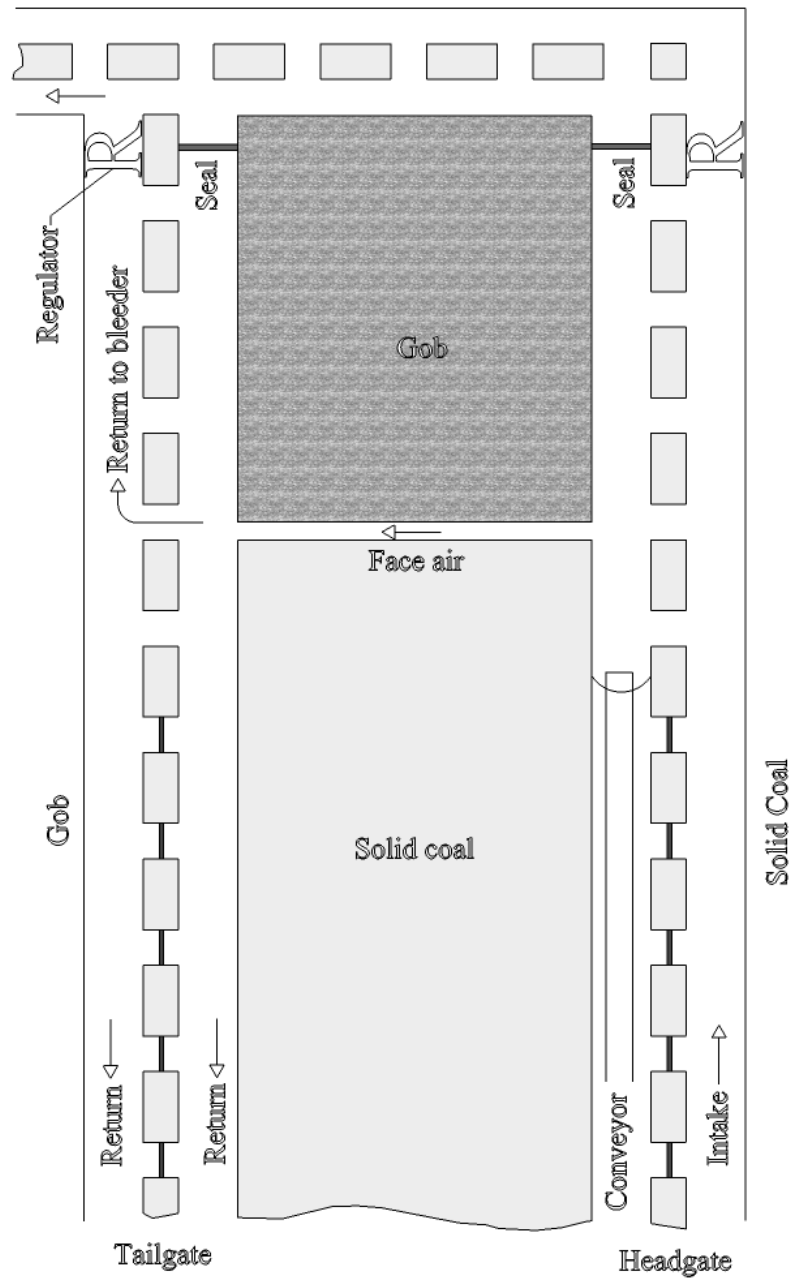


FIGURE 3.2 A typical two-entry, retreating longwall system with a bleeder

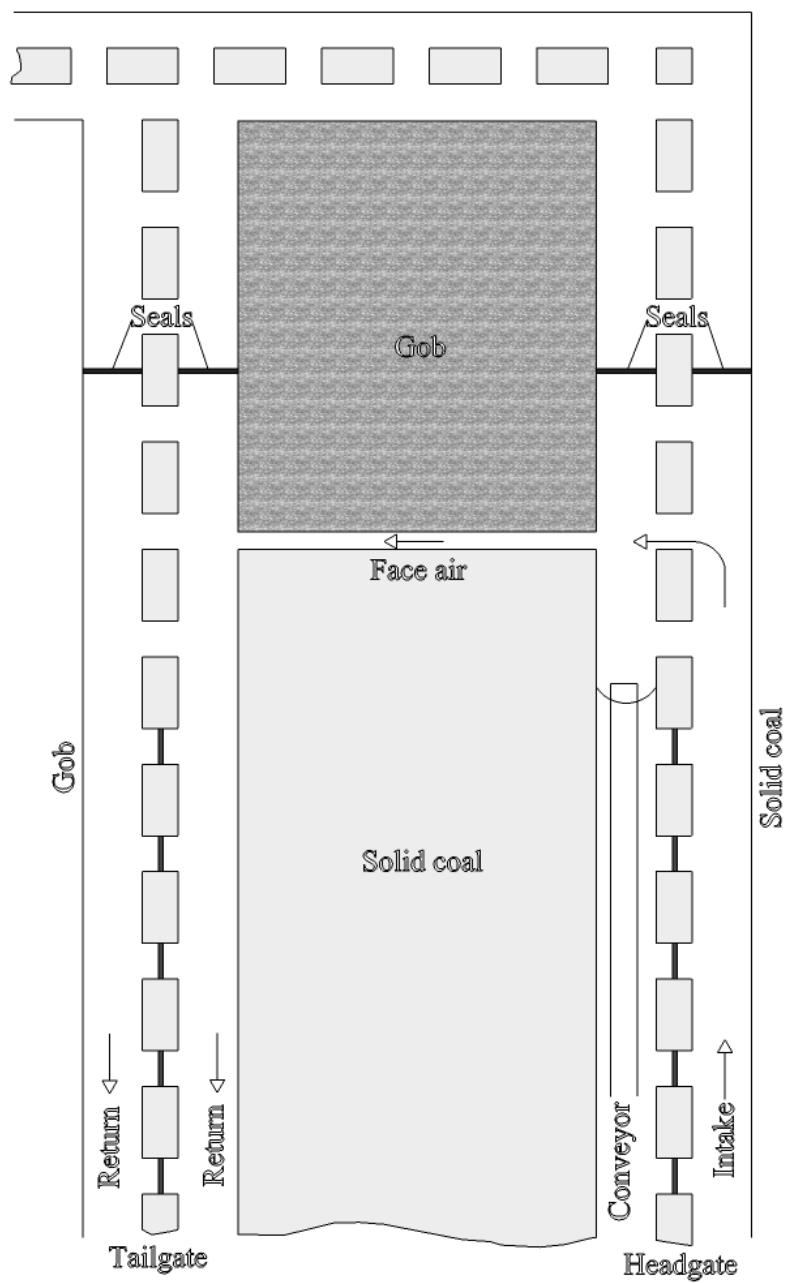


FIGURE 3.3 A typical two-entry, bleederless longwall system

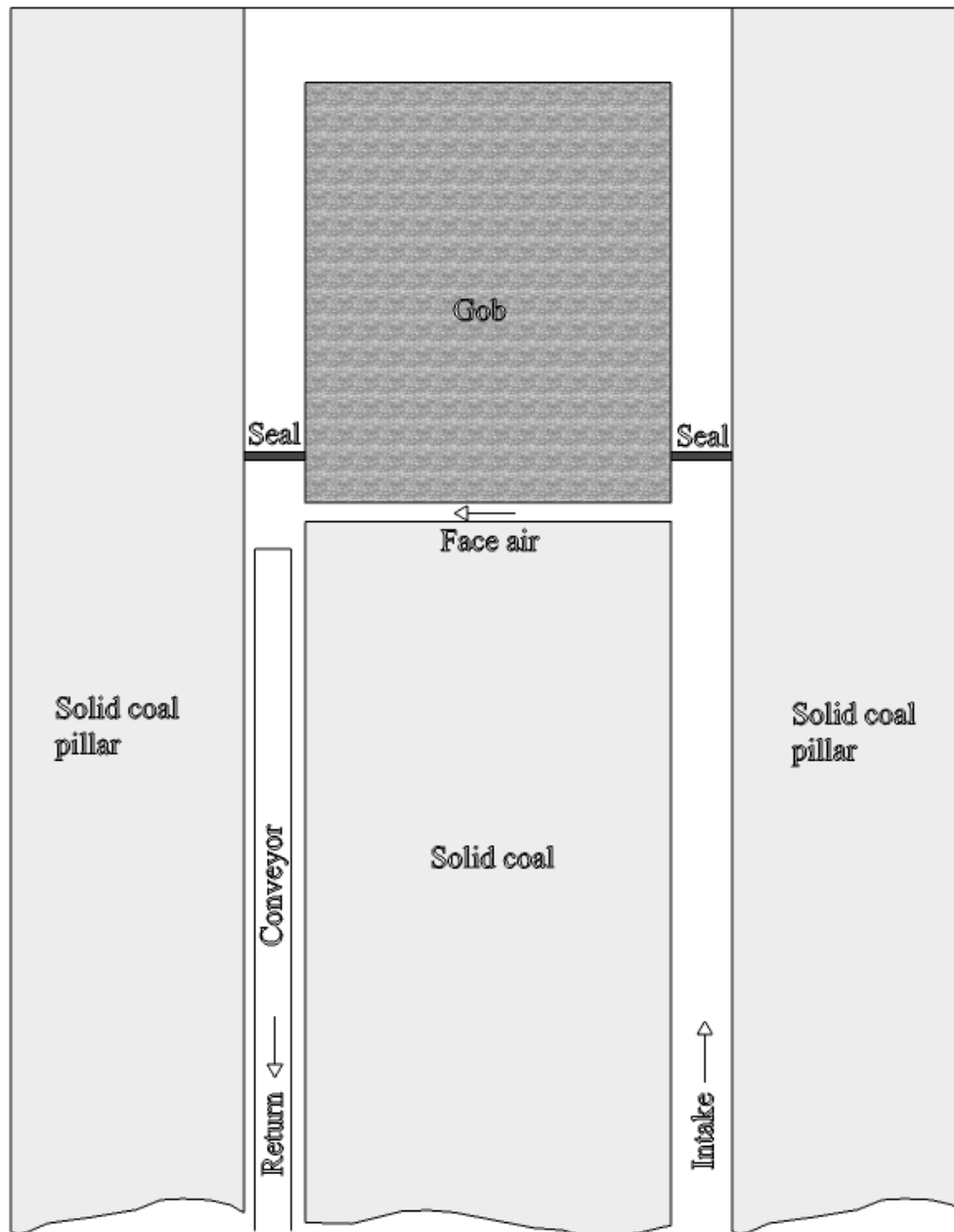
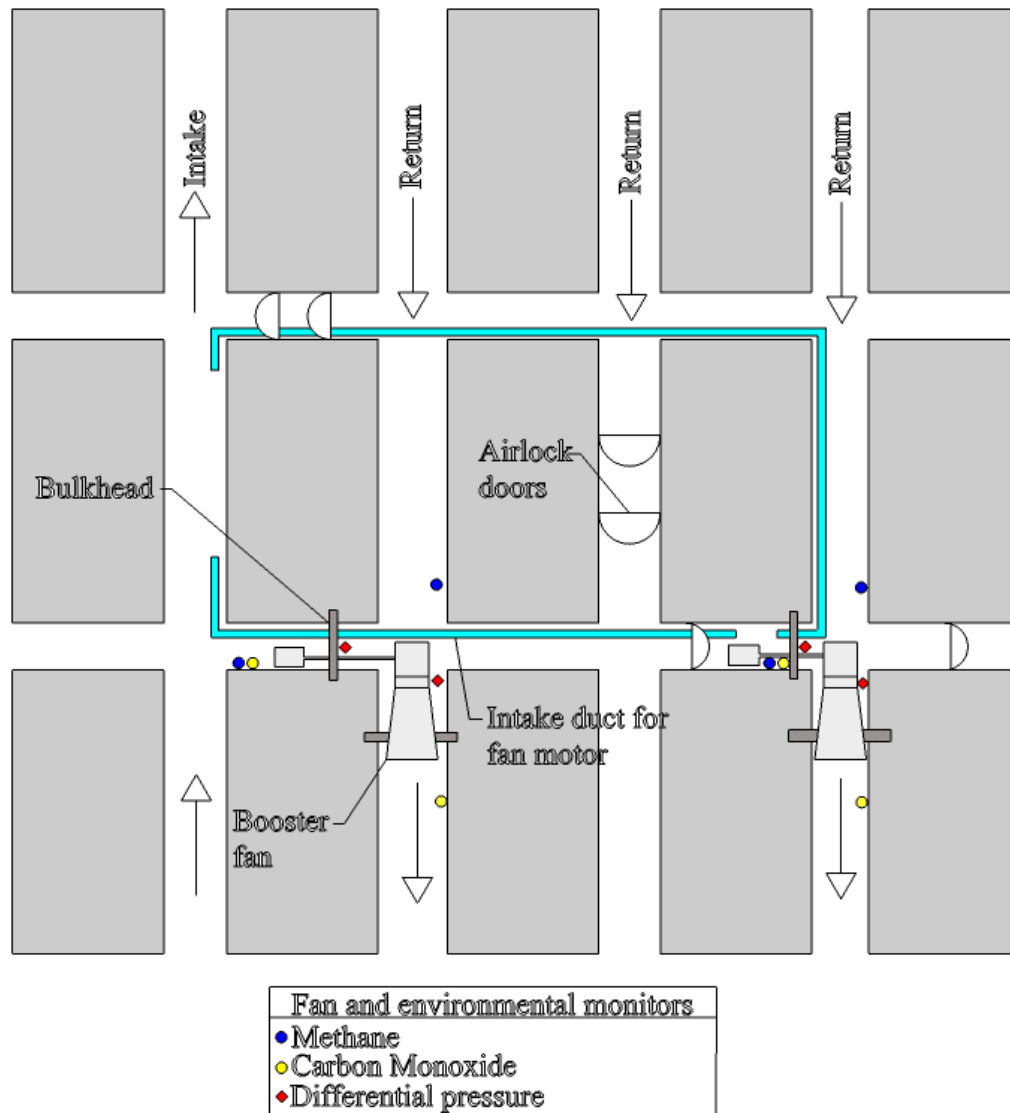


FIGURE 3.4 Single-entry, retreating longwall system typical in the United Kingdom



Adapted from: Moreby 2009

FIGURE 3.5 Approved Australian booster fan system

CHAPTER 4

LABORATORY MODELING

4.1 Introduction

Active coal mine ventilation systems are complex and dynamic and for these reasons, accurate and detailed field experiments are difficult to conduct. In addition, because booster fans are not used in U.S. coal mines, field experiments to study the characteristics of recirculation in an active coal mining environment are impractical. A scaled ventilation model of a coal mine including a booster fan was developed so that experiments could be conducted in an environment where conditions could be well controlled and variables could be systematically modified. This chapter summarizes the research method, parameters, data, and results for the laboratory model. Included is an analysis focusing on how recirculation and leakage are affected by the size of the main fan, the size of the booster fan, and the resistance of the ventilation circuit.

4.2 Model Similitude

The laboratory coal mine ventilation model was designed based on the fluid mechanics principle of similitude. For flows that are dynamically similar, the ratios of the forces acting on the corresponding fluid particles and the boundary surfaces are constant.

Dynamic similarity requires that the boundaries of the flows are geometrically similar so the characteristic lengths are proportional and that the flows are kinematically similar so the stream lines for the flows are similar. When flows are dynamically similar, they can be characterized by the same nondimensional parameters. For flow through a pipe, which is analogous to the flow in a mine ventilation system, the Reynolds number, the ratio of inertial forces to viscous forces, is most often used to characterize flow (Kundu and Cohen 2008). The Reynolds number is defined as:

$$Re = \frac{UD}{\nu} = \frac{\rho UD}{\mu} \quad (4.1)$$

where

Re = Reynolds number

U = relative fluid velocity, m/s

D = diameter, m

ν = kinematic viscosity, m²/s

ρ = fluid density, kg/m³

μ = dynamic viscosity, Pa s

In pipe flow systems, the hydraulic diameter is most often used to define the diameter of the conduit, where the hydraulic diameter is defined as:

$$D_h = \frac{4A}{o} \quad (4.2)$$

where

D_h = hydraulic diameter, m

A = cross-sectional area, m²

O = perimeter, m

In coal mine ventilation systems, the Reynolds number can vary dramatically depending on the airways velocity; however, turbulent flow is prevalent. For pipe flow systems, the transition from laminar to turbulent flow generally occurs at a Reynolds number of 3,000 (Kundu and Cohen 2008). For a coal mine ventilation system with a hydraulic diameter of 3.65 m, 25 times the hydraulic diameter of the laboratory model, Reynolds numbers range from 58,000 for a room and pillar development section with a volumetric flow rate of 4.25 m³/s and a velocity of 0.232 m/s to 194,000 for a longwall face with a volumetric flow rate of 14.16 m³/s and a velocity of 0.774 m/s. In the laboratory model (Figure 4.1) experiments for the flow at both faces, the Reynolds numbers ranged from a minimum of 10,000 at Face 1 to a to a maximum of 116,000 at Face 2. The Reynolds numbers calculated for the experimental model overlap with values that are realistic for an actual ventilation system indicating that the flows exhibit dynamic similarity and that all of the flows are turbulent. Fluid and physical characteristics for the lab model and for a realistic mine ventilation system are summarized in Table 4.1.

4.3 Model Parameters

The main branches of the laboratory coal mine ventilation model are constructed of 0.146 m diameter pipe. A schematic of the model is shown in Figure 4.1 and images of the models are presented in Figures 4.2 and 4.3. The pipe is configured in a standard U-shaped ventilation network with one intake and one return joined by three crosscuts, referred to as A, B, and C. The crosscuts are constructed of 0.056 m diameter pipe and act as leakage paths between the intake and the return. The magnitude of leakage through these crosscuts is restricted using perforated valves with variable open areas. The valves were designed with multiple small diameter holes so that leakage flow through the stoppings would be laminar because, although turbulent flow is prevalent in the main airways in mine ventilation systems, leakage flow is most often assumed to be laminar (Hartman et al. 1997). The number of holes and the diameter of the holes in the valves were used to control the open area of the valves. Table 4.2 summarizes the characteristics of the leakage valves used to regulate crosscuts A, B, and C. The percentage open area of the valves can be related to the quality of stopping construction in an actual ventilation system (Stephens 2011).

Two additional crosscuts join the intake and the return airways. These crosscuts, identified as Face 1 and Face 2 in Figure 4.1, are designed to represent two active mining sections. Face 1 and Face 2 are both constructed of 0.146 m diameter pipe. Face 1, the free split in the ventilation system, is completely open so the quantity of air that passes Face 1 is dictated by the system pressures and is not limited by a regulator. To limit the flow across Face 2 and prevent the majority of air from short circuiting the

system, Face 2 was regulated using a valve similar to the values used to regulate crosscuts A, B, and C. The Face 2 regulation valve is identified as V3 in Figure 4.1. This valve has a diameter of 0.146 m, and during all of the experiments, V3 was blocked using a regulator with an open area of 15.9%.

In addition, the ventilation model includes three valves, V1, V2, and V4, which are used to direct and restrict the airflow at various points in the system. These valves are identified in Figure 4.1. Valve V1 is located in the branch connected to the booster fan. For all of the experiments where the booster fan was enabled, this valve was not regulated; however, several experiments were conducted with the booster fan disabled and in these experiments, V1 was fully blocked. Valve V2 is installed in the bypass duct, below the booster fan. This valve allows the booster fan to be completely bypassed and for the experiments with the booster fan disabled, V2 was unregulated. When the booster fan was enabled, this valve was regulated to simulate a bulkhead with airlock doors and the open area of the valve was varied to simulate various bulkhead constructions. For the experiments with the booster fan enabled, the percentage open area of V2 was varied to simulate a range of bulkhead constructions. Table 4.3 summarizes the properties of the valves used to regulate V2. Valve V4, located at the system exhaust, was used to increase the resistance of the system and to increase the pressure losses in the return. In all of the experiments, V4 was regulated with a valve with a diameter of 0.146 m and an open area of 28%.

Finally, the laboratory model uses two centrifugal blower fans with variable frequency drive (VFD) motors. The 2.5 kW main fan is located at the system intake and

the 2.0 kW booster fan is located in the duct between crosscut A and Face 2 as illustrated in Figure 4.1. Both fans are capable of operating between 0 and 60 Hz and, for the given experimental set up, each fan produces a flow rate near $0.2 \text{ m}^3/\text{s}$ at a pressure of about 2.3 kPa.

4.4 Experimental Method

Experiments were carried out by measuring the velocity and static pressures throughout the coal mine ventilation model. There are 18 pressure stations in the model; static pressure was measured at each of the pressure taps along the duct and velocity pressures were measured at stations where changes in air velocity were expected due to leakage or recirculation. A calibrated digital manometer and pitot static tubes were used to measure the pressures. Static pressures were measured directly and velocity pressures were measured using a six-point equal area traverse and the air velocities and volumetric flow rates were calculated using Equations 2.2 and 2.3. Pressure differentials across crosscut valves A, B, and C were also measured. For each experiment, first the frequency of the main fan and the frequency of the booster fan were set. The main fan was then energized and once the fan was running at full speed, the booster fan was energized. The airflow through the system was given time to stabilize before any pressures were recorded.

Regulator and frequency settings for each experiment were selected based on Box-Behnken experimental designs (Mathworks 2012). Nine experiments were carried out with the booster fan disabled and bypassed. In these experiments, the frequency of the

main fan was varied among 30, 45, and 60 Hz and the regulation of crosscut valves A, B, and C was varied among 3.9, 15.4, and 27.2% open area. Twenty-five experiments were carried out using the main fan and the booster fan. These experiments were designed around four variables: main fan and booster fan frequencies which were varied among 30, 45, and 60 Hz; regulator settings for crosscut valves A, B, and C which varied among 3.9, 15.4, and 27.2% open area; and regulator settings for valve V2 which varied among 0, 15.9, and 28% open area.

4.5 Experimental Results

For each experiment, fan pressure and flow rates, and system leakage and recirculation quantities were determined. Complete experimental results are summarized in Appendix A. Section 4.5.1 presents data demonstrating how system leakage, volumetric efficiency, and recirculation in the laboratory model are affected by the booster fan pressure and the quality of the stopping construction. Section 4.5.2 presents data that demonstrate the relationship among main fan pressure, booster fan pressure, and system recirculation. Finally, Section 4.5.3 presents data demonstrating how local recirculation, around the booster fan, is affected by the quality of construction of the bulkhead and the pressure of the booster fan.

4.5.1 Leakage and Volumetric Efficiency

In the 25 experiments which used both the booster fan and the main fan, system leakage ranged from 2% to 26% of the flow quantity at the main fan, where the leakage

quantity was calculated by taking the difference between the flow rates upstream and downstream of each stopping and summing these values. In all of the experimental systems, leakage was strongly dependent on the percentage open area of the valves used to block crosscuts A, B, and C: leakage increased as the open area of the valves increased. This dependence is demonstrated by the experiments summarized in Table 4.4. In these two experiments, all of the variables were constant except for the open area of the crosscut valves, A, B, and C. The open area of the crosscut valves was varied between 3.9 % open, which represents a very well-constructed stopping, and 27.2% open, which represent a moderately well-constructed stopping and leakage varied between 5% and 21%, respectively.

Additionally, the use of a booster fan in the ventilation model decreased the overall leakage and increased the volumetric efficiency, which is defined in Equation 2.8. In the experiments where the booster fan was bypassed, the percentage of system leakage ranged from 5% to 38%. So, in general, leakage was higher in these systems than the systems where both the main fan and the booster fan were enabled. Figure 4.4 demonstrates typical pressure profiles of the laboratory model experiment where the booster fan was bypassed and the main fan was operating at a pressure of 2.35 kPa and a flow rate of $0.223 \text{ m}^3/\text{s}$. This experiment had a leakage value of 23% and a volumetric efficiency of 77%. In comparison, Figure 4.5 demonstrates the pressure profiles of the laboratory model for an experiment where the main fan was operating at a pressure of 2.33 kPa and a flow rate of $0.236 \text{ m}^3/\text{s}$ and the booster fan was operating at a pressure of 0.42 kPa and a flow rate of $0.264 \text{ m}^3/\text{s}$. This experiment has a leakage value of 18% and

a volumetric efficiency of 82%. Neither of these experiments exhibited system recirculation and the use of the booster fan in the system decreased leakage by 5%.

4.5.2 System Recirculation

Operating the booster fan in the coal mine ventilation model decreased the leakage measured in the system; however, in many of the experiments, the leakage in crosscut A was reduced to the point that airflow reversed and recirculation was induced. Because of the location of the booster fan relative to the main fan, crosscut A is the only location where there is potential for recirculation. Sixteen of the 25 experiments that utilized the booster fan exhibited measureable system recirculation through crosscut A and seven of these experiments at least 10% system recirculation. In the majority of the experiments that exhibited large percentages of recirculation, the booster fan pressure was notably higher than the main fan pressure. System characteristics for these seven experiments are summarized in Table 4.5.

It is evident that in addition to fan pressures, the magnitude of recirculation was strongly dependent on the open area of crosscut valves A, B, and C. In general, higher quantities of recirculation occurred as the open area of the valves increased. In three of the seven experiments that exhibited 10% or more recirculation, the open area of crosscut valves A, B, and C was 27.2% and these three experiments had an average of 18% recirculation. In the remaining four experiments, the open area of crosscut valves A, B, and C was 15.4% and these experiments had an average of 12% recirculation. Notably,

none of the experiments with a crosscut valve open area of 3.9 % experienced more than 6% recirculation.

4.5.3 Local Recirculation

In addition to system recirculation, local recirculation around the fan, controlled by valve V2, was significant. In experiments where V2 was completely blocked, no measureable recirculation around the fan occurred. These experimental systems can be correlated to a full-scale booster fan system with an ideally constructed bulkhead and air lock doors. As the open area of the valve increased, the local recirculation increased substantially. In the experiments where V2 was 15.9% open, which represents a relatively poor airlock condition, local recirculation ranged from 19.5 to 61.5%. When valve V2 was 28% open, local recirculation ranged from 45.5 to 97%. This result reflects that the recirculation is likely to occur in a ventilation system with a damaged bulkhead and airlock doors.

The percentage of local recirculation was also strongly dependent on the percentage of system recirculation and on the relative fan pressure. High system recirculation also contributed to high local recirculation. This relationship is demonstrated by Figures 4.6, 4.7, and 4.8. For the experiments where V2 had an open area of 15.9% and crosscuts A, B, and C had an open area of 15.4%, the local recirculation was on average 6 times higher than the system recirculation; these data are summarized in Figure 4.6. For the experiments where V2 had an open area of 15.9% and crosscuts A, B, and C had an open area of 27.2%, the local recirculation was on average

11 times higher than the system recirculation, these data are summarized in Figure 4.7. For the experiments where V2 had an open area of 28.0% and crosscuts A, B, and C had an open area of 15.4%, the local recirculation was on average 3 times higher than the system recirculation; these data are summarized in Figure 4.8.

4.6 Discussion

The laboratory model experiments demonstrate that booster fans can effectively increase the volumetric efficiency of ventilation systems by reducing leakage and directing the airflow to areas where it is needed. In general, leakage was less in the experiments where the booster fan was used. The amount of leakage in the ventilation systems both with and without the booster fan was highly dependent on the leakage path resistance; as the stopping area increased and thus the resistance decreased, leakage increased and the volumetric efficiency decreased. System recirculation was also found to be highly dependent on the quality of the stoppings: the larger the open areas of the valves, the more likely the systems were to experience recirculation regardless of fan pressure. In addition to the quality of the stoppings, the pressure of the booster fan relative to the main fan had a substantial effect on system recirculation. As the booster fan pressure increased relative to the main fan pressure, the potential for system recirculation increased; however, the experiments demonstrate that system recirculation can be limited as long as the booster fan pressure is less than the main fan pressure.

Local recirculation around the booster fan is also an important consideration in booster fan systems. When local recirculation occurs, energy is consumed as air

recirculates around the fan, but there is no increase in airflow through the system. The laboratory model experiments showed that local recirculation is highly dependent on the quality of the bypass system, including the bulkheads and airlock doors. When V2 was completely blocked, no local recirculation occurred. This system simulates an ideally constructed bulkhead and with well-designed airlock doors. However, the experiments in which V2 was partially open demonstrate that even relatively well-constructed bulkheads can experience high quantities of local recirculation. Because no bulkhead installations are ideal and airlock doors, which are an important safety feature in booster fan systems, allow some leakage, some quantity of local recirculation is inevitable. Installing booster fans in the return airways, where the airlock doors are used less frequently, may help to reduce damage to the airlock system and thus reduce local recirculation. Additionally, the experiments demonstrated that relative fan pressures and system recirculation have a significant influence on local recirculation: as the booster fan pressure increases relative to the main fan, so does the percentage of local recirculation and, additionally, high system recirculation contributes to high local recirculation. In summary, both local and system recirculation increase as the relative booster fan pressures increase and the quality of the ventilation control devices throughout the ventilation system decrease.

TABLE 4.1 System characteristics

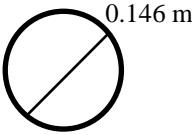
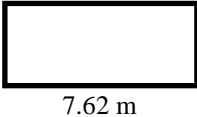
Characteristics	Model	Mine
Airway Shape and Dimensions		
Characteristic Length, m	0.146	3.651
Kinematic Viscosity, m ² /s (t = 10°C)	1.46 x 10 ⁻⁵	1.46 x 10 ⁻⁵
Volumetric Flow Rate, m ³ /s	0.098	4.25
Velocity, m/s	5.86	0.23
Reynolds Number	5.86 x 10 ⁴	5.81 x 10 ⁴

TABLE 4.2 Characteristics of crosscut A, B, and C regulation valves

Label	Dia., m	No. of Holes	Dia. of Holes, m	Open Area, %
1	0.056	37	0.0064	27.2
2	0.056	21	0.0064	15.4
3	0.056	21	0.0032	3.9

TABLE 4.3 Characteristics of V2 regulation valves

Valve Labels	Valve Dia., m	No. of Holes	Dia. of Holes, m	Open Area, %
1	0.146	0	--	0.0
2	0.146	21	0.0127	15.9
3	0.146	37	0.0127	28.0

TABLE 4.4 Leakage dependence on percentage open area

Experiment No.	Crosscut A, B, and C Open Area, %	Leakage, %	Main Fan		Booster Fan	
			Δp , kPa	Q, m ³ /s	Δp , kPa	Q, m ³ /s
19	3.9	5	2.27	0.254	1.08	0.331
21	27.2	21	2.28	0.256	1.07	0.335

TABLE 4.5 Recirculation dependence on fan pressures

Experiment No.	Crosscut A, B, and C Open Area, %	Leakage, %	Recirculation, %	Main Fan		Booster Fan	
				Δp , kPa	Q, m ³ /s	Δp , kPa	Q, m ³ /s
2	27.2	23	15	1.27	0.21	1.28	0.24
3	15.4	12	12	0.52	0.18	1.31	0.20
7	15.4	13	13	0.50	0.21	2.16	0.36
12	15.4	12	10	0.54	0.16	1.07	0.32
17	27.2	26	18	1.24	0.23	2.05	0.39
18	27.2	26	21	0.54	0.16	1.18	0.29
24	15.4	14	12	1.22	0.25	2.32	0.28

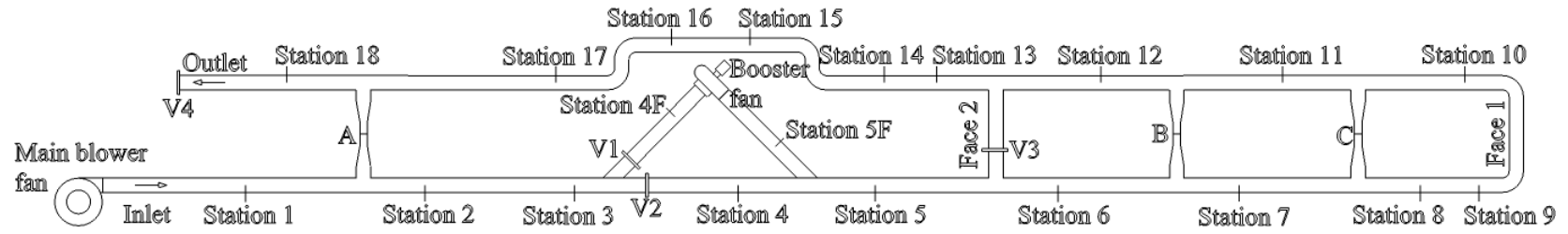


FIGURE 4.1 Schematic of the coal mine ventilation laboratory model



FIGURE 4.2 Photograph of the coal mine ventilation model

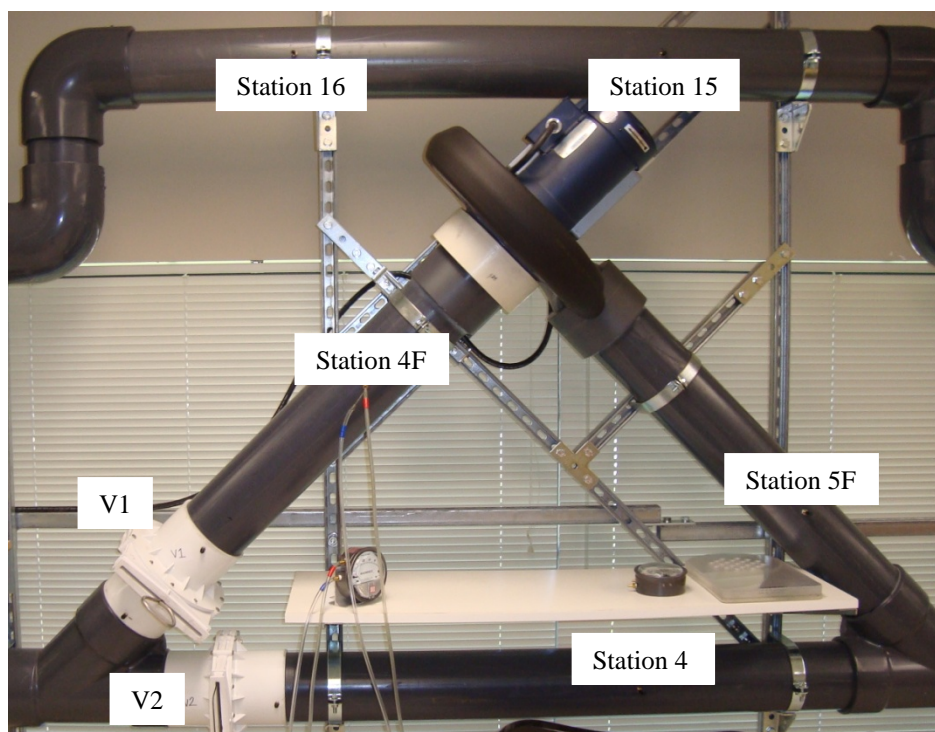


FIGURE 4.3 Photograph of the ventilation model booster fan

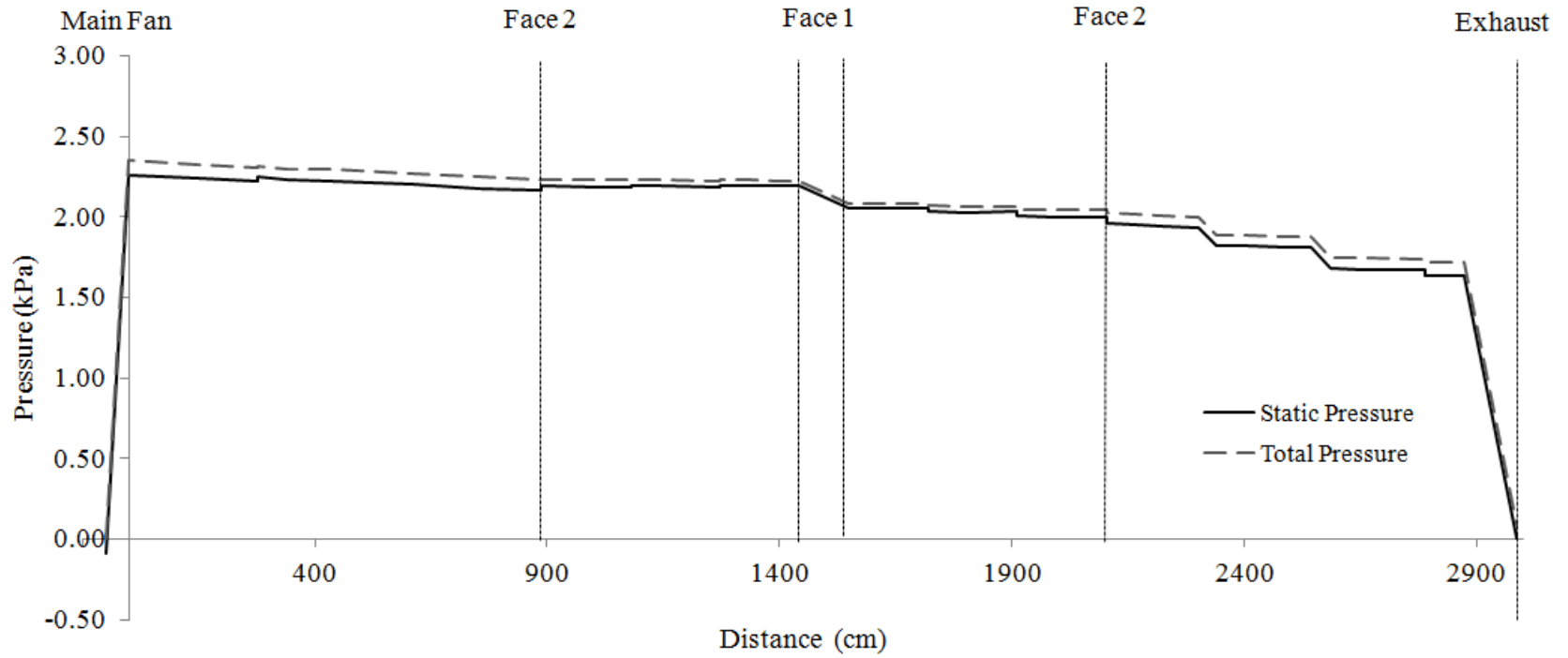


FIGURE 4.4 Pressure profiles of the coal mine ventilation system with the booster fan disabled and bypassed

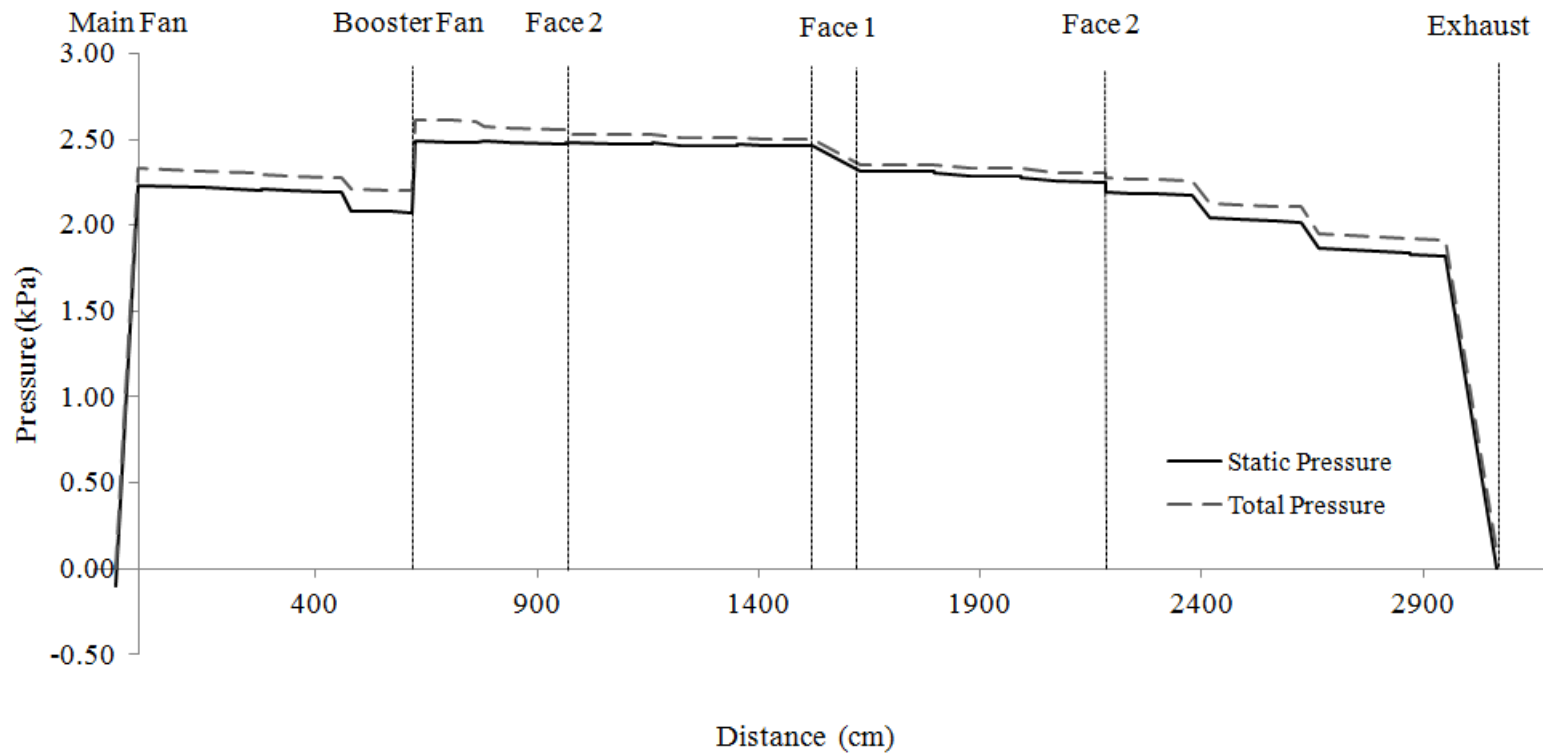


FIGURE 4.5 Pressure profiles of the coal mine ventilation system with the booster fan enabled

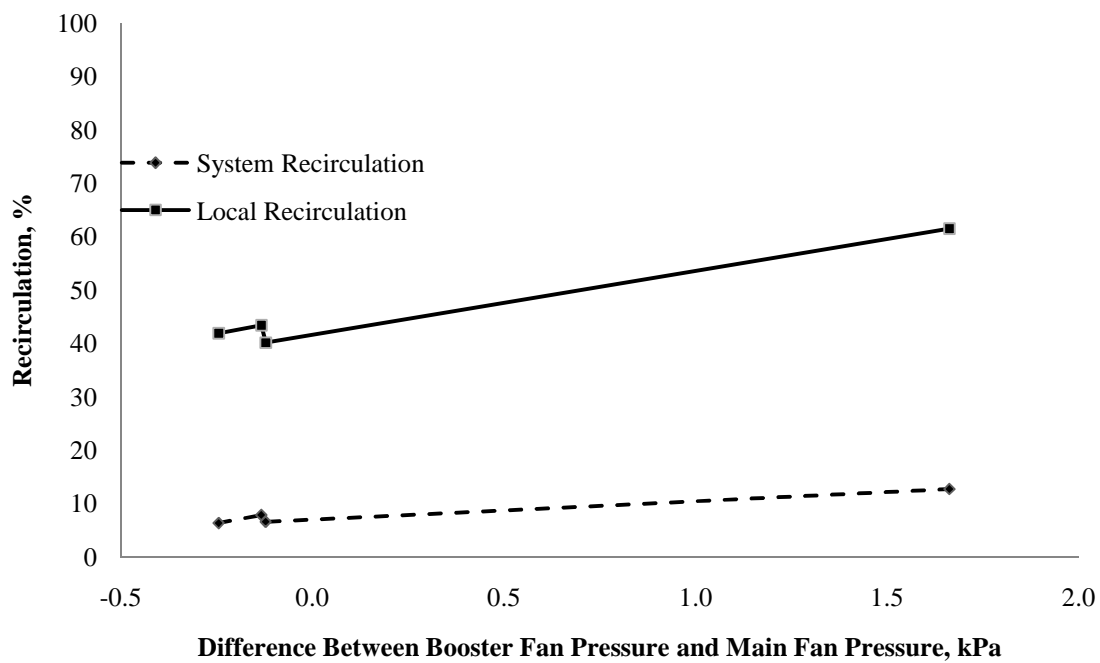


FIGURE 4.6 Relationship between local and system recirculation for experiments where V2 was 15.9% open and crosscuts A, B, and C were 15.4% open

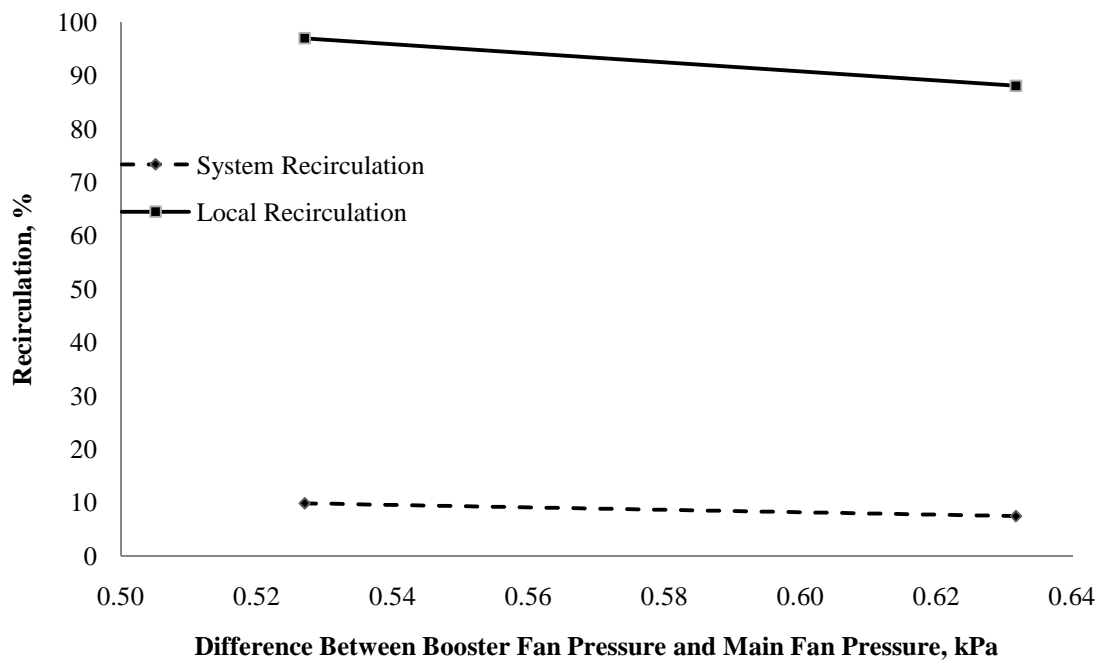


FIGURE 4.7 Relationship between local and system recirculation for experiments where V2 was 15.9% open and crosscuts A, B, and C were 27.2% open

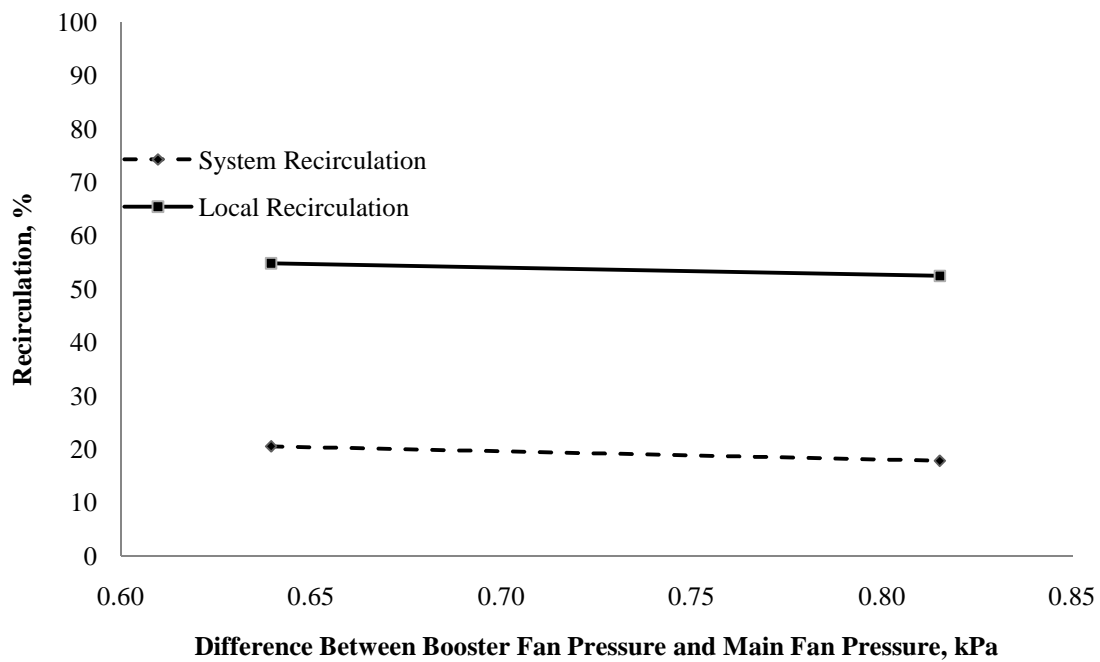


FIGURE 4.8 Relationship between local and system recirculation for experiments where V2 was 28.0% open and crosscuts A, B, and C were 15.4% open

CHAPTER 5

NUMERICAL MODELING USING VENTSIM

5.1 Introduction

The laboratory coal mine ventilation model was useful to help identify relationships among the main fan pressure, the booster fan pressure, and the stopping resistance but its application for research is somewhat limited. Because of size constraints, the model cannot be readily lengthened to study the effects of increasing resistance on recirculation and the booster cannot be easily relocated so it is difficult to assess the neutral point in the ventilation system and the system recirculation is affected by the location of the booster fan. For reasons like these, numerical ventilation simulators play a critical role in ventilation research and planning. VnetPC and Ventsim are two ventilation simulators used commonly throughout the mining industry. Both programs can be used to evaluate fan operating conditions, flow rates, and pressure losses in ventilation systems and both programs were used to develop numerical models to further study the effects of booster fans on coal mine ventilation systems. Ventsim was used to develop a numerical model of the laboratory model. This model was used to evaluate the effects of booster fan location, booster fan pressure, and increased resistance on the location of the neutral point and the magnitude of the system recirculation.

5.2 Laboratory Model Characteristics

To evaluate how changes in the booster fan location, booster fan pressure, and system resistance affect the location of the neutral point and the magnitude of recirculation in a ventilation system, a numerical model based on the laboratory model was developed using Ventsim. This program was chosen because, unlike VnetPC, it includes an algorithm to detect recirculation. Although the laboratory model consists of three crosscuts and two working faces, to allow for more variation in the location of the booster fan, the numerical simulation was initially designed to include ten crosscuts and one working face. A schematic of the numeric model is presented in Figure 5.1. Resistance values for the intake, return, and the crosscuts were calculated from the experimental data presented in Chapter 4 using Equation 2.1. The characteristics of the branches are summarized in Table 5.1. Additionally, fan curves were developed for the main fan in the laboratory model and used in the numerical simulation for both the main fan and the booster fan. The fan curves are presented in Figure 5.2. Finally, the ventilation laboratory model was designed as a blowing ventilation system, but because exhaust ventilation systems are more common in coal mines in the United States, the numerical model was laid out as an exhausting ventilation system. It is important to note that the dynamics of recirculation are the same in both blowing and exhausting ventilation systems.

5.3 Laboratory Model Data

A base case Ventsim model was developed with the main fan operating at 2.43 kPa and 0.427 m³/s. Figure 5.3 demonstrates the pressure profile for this system. For the resistance values used in the Ventsim model, the pressure drop is approximately linear for a stopping resistance of 800,000 Ns²/m⁸. Additionally, the Ventsim model does not account for additional resistance from shock losses. Because the system does not include a booster fan, the pressure in the return airway is always less than the pressure in the intake airway and there is no potential for the development of a neutral point, where the pressure in the intake and the return are balanced, or recirculation.

Next, a booster fan was added to the system at branch 18 to evaluate how the fan pressure is related to the neutral point and the system recirculation. The booster fan was first operated at 0.51 kPa and then at 1.66 kPa. For these two scenarios, the main fan operating conditions were relatively constant near 2.4 kPa. When the booster fan was operating at 0.51 kPa, the pressure profile did not reach a neutral point and no recirculation occurred. When the pressure at the booster fan was increased to 1.66 kPa, the pressure profile crossed a neutral point and recirculation was induced. Figures 5.4 and 5.5 show the pressure profiles for these two scenarios. In addition, Figure 5.6 shows the Ventsim model for the scenario where the booster fan was operating at 0.51 kPa. Again the profiles are approximately linear.

The effect of increasing the distance between the main fan and the booster fan on the tendency of the system to recirculate was evaluated by placing a booster fan to the model at branch 16. Again the main fan pressure was held relatively constant at near

2.4 kPa and the booster fan was operated at 0.54 and 1.67 kPa, respectively. In both of these scenarios, the pressure from the booster fan led to the formation of a neutral point and induced recirculation. The quantity of recirculation was minimal at $0.001 \text{ m}^3/\text{s}$ when the booster fan pressure was 0.54 kPa, but when the booster fan pressure was 1.67 kPa, the quantity of recirculation was more significant at $0.048 \text{ m}^3/\text{s}$ (10%).

Finally, the effect of increasing the airway resistance was evaluated. Three crosscuts were added to the model, increasing the length of the system by 23%. The main fan was operated at 2.4 kPa with a flow rate of $0.4 \text{ m}^3/\text{s}$. The booster fan, placed in the model at branch 16, was first operated at 0.56 kPa and then the pressure was increased to 1.70 kPa. At a booster fan pressure of 0.56 kPa, no recirculation occurred. At a pressure of 1.70 kPa, the system crossed a neutral point and a quantity of $0.044 \text{ m}^3/\text{s}$ (10%) was recirculated. In the scenarios with increased resistance, the magnitude of recirculation was less when compared to the lower resistance scenarios. Table 5.2 summarizes the model characteristics, including fan operating points and system recirculation for each experimental scenario.

5.4 Discussion

When evaluating booster fan ventilation systems, the concept of a neutral point is beneficial because it can help to identify the most efficient fan placement and also regions where recirculation is likely to occur. In relatively simple ventilation systems, the formation of a neutral point is strongly dependent on the distance between the main fan and the booster fan. In general, as the booster fan is located closer to the work areas and

the distance between the main fan and the booster fan increases, the potential for recirculation increases. Comparing the results of ten crosscut systems, when the booster fan was placed at branch 18, recirculation did not occur, but when the fan was placed at branch 16, the pressure profile crossed a neutral point and recirculation was induced. In both cases, the booster fan was operating near 0.5 kPa

In addition to the relative distance between the main fan and the booster fan, the size of the booster fan has a significant effect on the tendency for recirculation. Systems with high booster fan pressure are more likely to form a neutral point and induce system recirculation. This is demonstrated by comparing the pressure profiles in Figures 5.4 and 5.5. In both of these systems, the booster fan was located at branch 18 and the main fan was operating near 2.4 kPa. In the scenario represented by Figure 5.4, the booster fan was operating at a pressure of 0.51 kPa and the system did not cross a neutral point and no recirculation occurred. In the scenario represented by Figure 5.5, a booster fan with a higher frequency fan curve was used, increasing the pressure to 1.66 kPa. In this scenario, a neutral point was formed and recirculation was induced by the higher booster fan pressure.

Finally, in simple ventilation systems, increasing system resistance tends to decrease the potential for recirculation. In the Ventsim scenarios, recirculation was decreased by about 8% in the high resistance ventilation system with a booster fan pressure near 1.7 kPa when compared to the low resistance scenario with a booster fan pressure of 1.67 kPa.

In general, recirculation is highly dependent on the location of the booster fan: the closer the fan is located to the work areas, the more likely it is to cause recirculation. Booster fan pressure is also an important consideration: higher fan pressures are more likely to cause recirculation. In systems with multiple booster fans, there may be multiple neutral points and as the capacities of the booster fans increase, recirculation is more likely to occur. In any ventilation system utilizing booster fans, the potential recirculation can be managed by thoroughly evaluating the effect of the booster fan pressure and location on the ventilation system. In general, as long as the booster fan is located far enough from the working areas and the booster fan pressure is less than the main fan pressure, recirculation can be limited. Additionally, because the recirculation quantity is dependent on the mine resistance, it is important to consider how changes in the system resistance impact the potential for recirculation through the life of the ventilation system.

TABLE 5.1 Characteristics of Ventsim model

Characteristics	Intake	Return	Face	Crosscuts
Branch Number	1 - 10	13 - 23	12	24 - 33
Length, m	21.70	20.90	1.10	
Area, m ²	0.02	0.02	1.07	
Perimeter, m	0.46	0.46	0.46	
Resistance, Ns ² /m ⁸	1.36 x 10 ⁴	1.31 x 10 ⁴	0.69 x 10 ⁴	8.03 x 10 ⁵

TABLE 5.2 Ventsim model experimental summary

Booster Fan Location	No. of Crosscuts	Recirculation Q, m ³ /s	Main Fan		Booster Fan	
			Δp , kPa	Q, m ³ /s	Δp , kPa	Q, m ³ /s
--	10	0.000	2.43	0.427	--	--
Branch 18	10	0.000	2.41	0.449	0.51	0.294
Branch 18	10	0.018	2.37	0.483	1.66	0.400
Branch 16	10	0.001	2.41	0.444	0.54	0.269
Branch 16	10	0.048	2.39	0.467	1.67	0.395
Branch 16	13	0.000	2.44	0.414	0.56	0.247
Branch 16	13	0.044	2.42	0.433	1.7	0.365

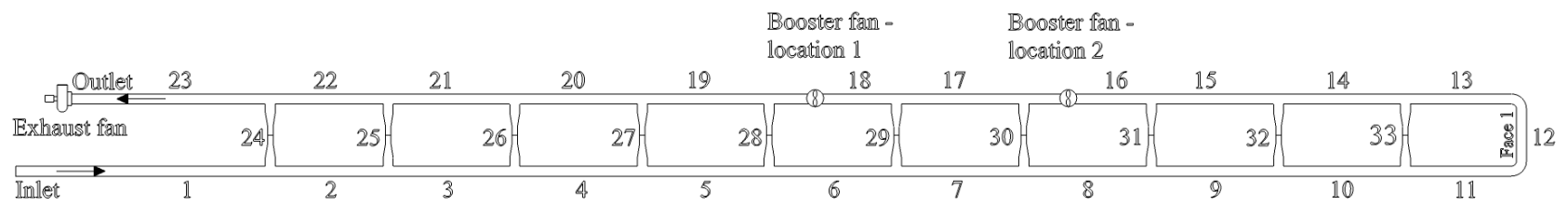


Figure 5.1 Schematic of the numeric coal mine ventilation model

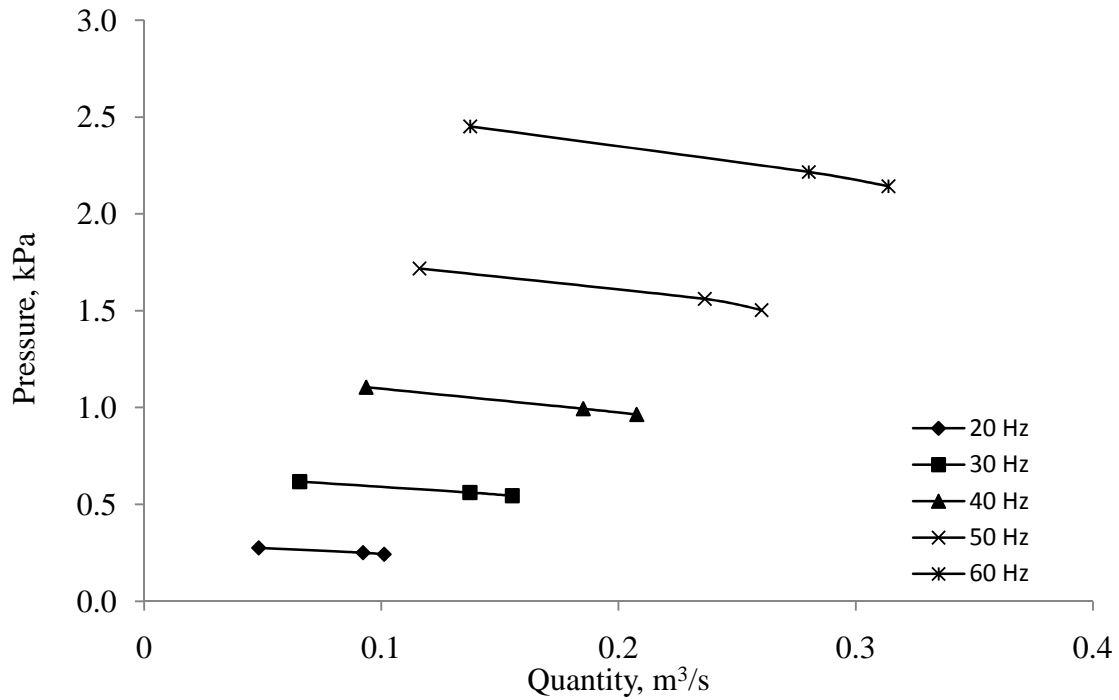


Figure 5.2 Fan characteristic curves used for the main fan and the booster fan in the numeric coal mine ventilation model

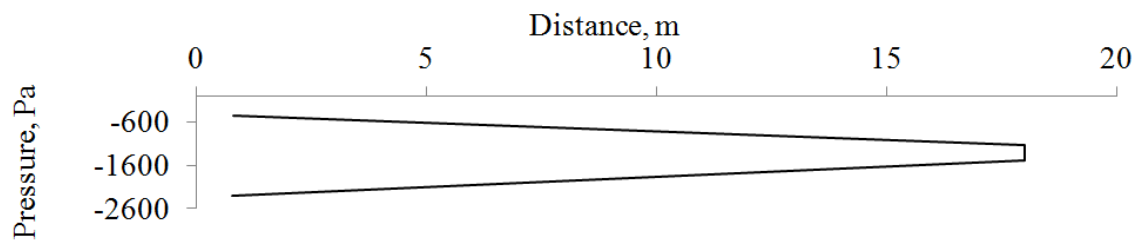


Figure 5.3 Pressure profile of the ten crosscut scenario with no booster fan where the stopping resistance is $800,000 \text{ N s}^2/\text{m}^8$

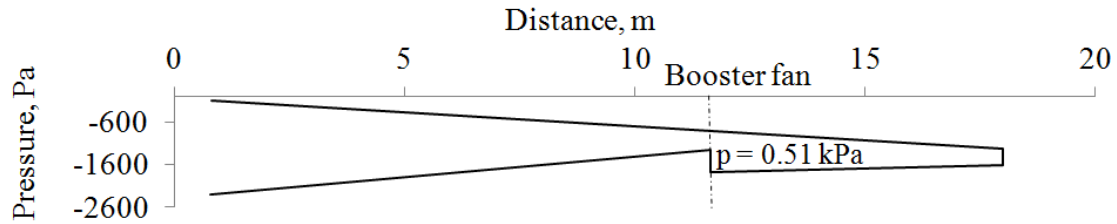


Figure 5.4 Pressure profile of the ten crosscut system with the booster fan located at branch 18 operating at a pressure of 0.51 kPa

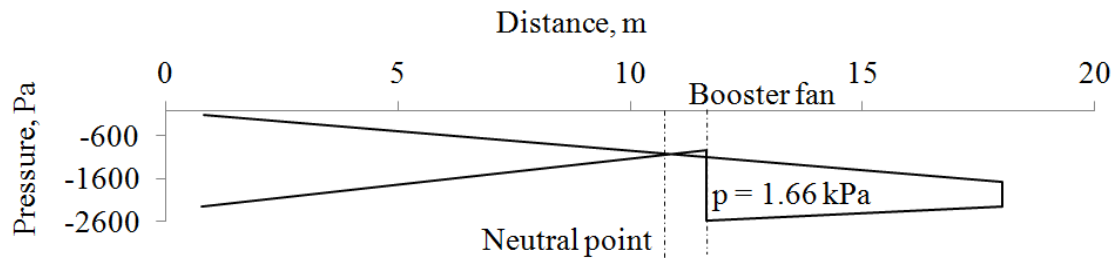


Figure 5.5 Pressure profile of the ten crosscut system with the booster fan located at branch 18 operating at a pressure of 1.66 kPa

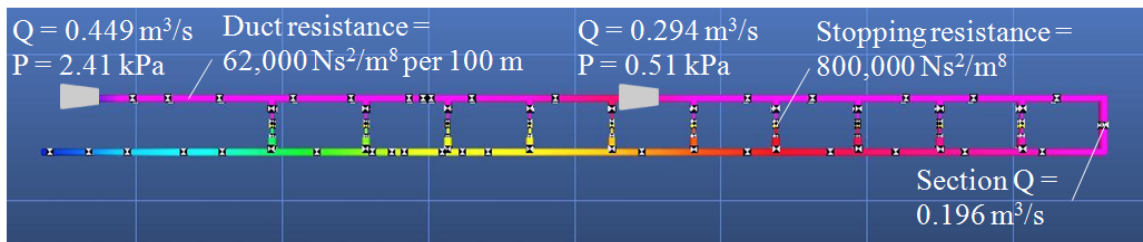


Figure 5.6 Venstim model showing system characteristics for the ten crosscut system

CHAPTER 6

COAL MINE VENTILATION MODELING

6.1 Introduction

To evaluate the potential impact of a booster fan on an active coal mine ventilation system, ventilation data were collected from an underground coal mine in the Eastern United States referred to as Mine A. Using these data, the ventilation modeling software VnetPC was used to develop a baseline ventilation model. The baseline model was expanded to define the future ventilation requirements and the expanded system was then modeled with a booster fan. These models were compared to evaluate the effect of the booster fan on the efficiency of the ventilation system.

6.2 Mine Characteristics

Mine A uses the room and pillar method as the primary means of coal production. At this mine, the distance from the surface to the farther working face is 6.1 km. The coal seam is horizontal, with an average thickness of 1.8 m, and the maximum depth of cover is 107 m. The mine operates near sea level with an average air density of 1.2 kg/m³. The ventilation system at Mine A consists of one intake slope, one intake shaft, five development sections, and one return shaft. Development mains are typically driven with four intake, four return, and three neutral airways. Development submains are

typically driven with two intake, two return, and three neutral airways. Intake entries have an average resistance of $0.0010 \text{ N s}^2/\text{m}^8$ per 100 m, neutral entries have an average resistance of $0.0039 \text{ N s}^2/\text{m}^8$ per 100 m, and return entries have an average resistance of $0.0011 \text{ N s}^2/\text{m}^8$ per 100 m. The system is powered by a 670 kW axial fan. The main fan exhausts $238 \text{ m}^3/\text{s}$ of air at a static pressure of 1.95 kPa with an air power of 464 kW. Figure 6.1 presents a characteristic curve for the main fan. Table 6.1 lists the quantities measure in the return of each development heading as well as the quantities in the main intakes and the main return. These values are reflected in the baseline ventilation model which is represented schematically in Figure 6.2.

6.3 Projected Ventilation Model

The baseline ventilation model of Mine A was expanded to include five years of additional development. Typical resistance values for the mine were used in the projected model. A quantity of $9.4 \text{ m}^3/\text{s}$ was assumed to be adequate to ventilate each of the development headings. In the expanded model, the fan operating pressure was increased when compared to the base ventilation model and the power requirements were higher. With no additional shafts or surface fans, the main fan would need to exhaust $200 \text{ m}^3/\text{s}$ at a static pressure of 2.24 kPa with an air power of 440 kW. At a fan efficiency of 65%, this system would require an input power of 676 kW, which is near the limits of the existing main fan.

To help balance the ventilation circuit and assist the main fan in overcoming the mine's resistance, a booster fan was added to the ventilation model in series with the main fan. A schematic of the expanded ventilation model showing the location of the

booster fan is presented in Figure 6.3. Using the algorithm presented by Calizaya et al. (1987), the booster fan was positioned to minimize the power consumed by the ventilation system. Initially, several reasonable booster fan location were selected. Simulations were run varying the size of the main fan and the booster fan and the optimum solution was identified as the model with the lowest combined power requirement. Because this method of optimization does not account for potential recirculation and VnetPC does not have an algorithm to detect recirculation, the simulations were visually inspected for recirculation. Models experiencing recirculation were not considered acceptable solutions. In the optimum simulation, the booster fan exhausts $150 \text{ m}^3/\text{s}$ at a static pressure of 0.57 kPa and the main fan exhausts $180 \text{ m}^3/\text{s}$ at a static pressure of 1.37 kPa . This is within the capability of the existing surface fan. For this optimum two-fan scenario, the total air power is 330 kW with a power savings of 110 kW when compared to the single-fan scenario (Wempen et al. 2011)

6.4 Discussion

The simulations demonstrate that as mines advance, ventilation requirements tend to increase fairly dramatically. As Mine A develops, the pressure at the main fan is increased from 1.95 kPa to 2.24 kPa . Though this increase in pressure is not excessive, it is enough to stretch the capacity of the current ventilation system. One way of increasing the capacity of the ventilation system would be to develop a new intake shaft, and at Mine A, this would likely be good option because, although extensive, the mine is relatively shallow, but land ownership may pose a problem for shaft development. Booster fans represent an alternate to developing a new intake shaft and, although a new

intake shaft may be feasible at Mine A, a booster fan is still a good alternative. Based on the simulation results, the expanded ventilation system utilizing a booster fan develops an air power of 330 kW, while the expanded ventilation system with a booster fan develops an air power of 440 kW. The booster fan generates a reduction of 110 kW. In addition to reducing the system power requirements, booster fans can balance the system pressure distribution and, consequently, reduce the overall system leakage. In Mine A, the addition of a booster fan to the expanded system reduced the total quantity by 20 m³/s. Based on the simulation results, installing a booster fan in a ventilation system has the potential to positively impact the system, reducing the main fan pressure, lowering the system power requirement, and limiting the amount of leakage. Though booster fans can be beneficial, they may not be the most appropriate or economic solution to increase the ventilation capacity at all underground coal mines.

TABLE 6.1 Ventilation survey data for Mine A

Fan duty	
Quantity (m ³ /s)	238
Pressure (kPa)	1.95
Airflow rates (m ³ /s)	
Intake Slope	98.87
Intake Shaft	137.53
Return Shaft	237.77
Face A	9.07
Face B	19.95
Face C	10.92
Face D	11.15
Face E	10.34

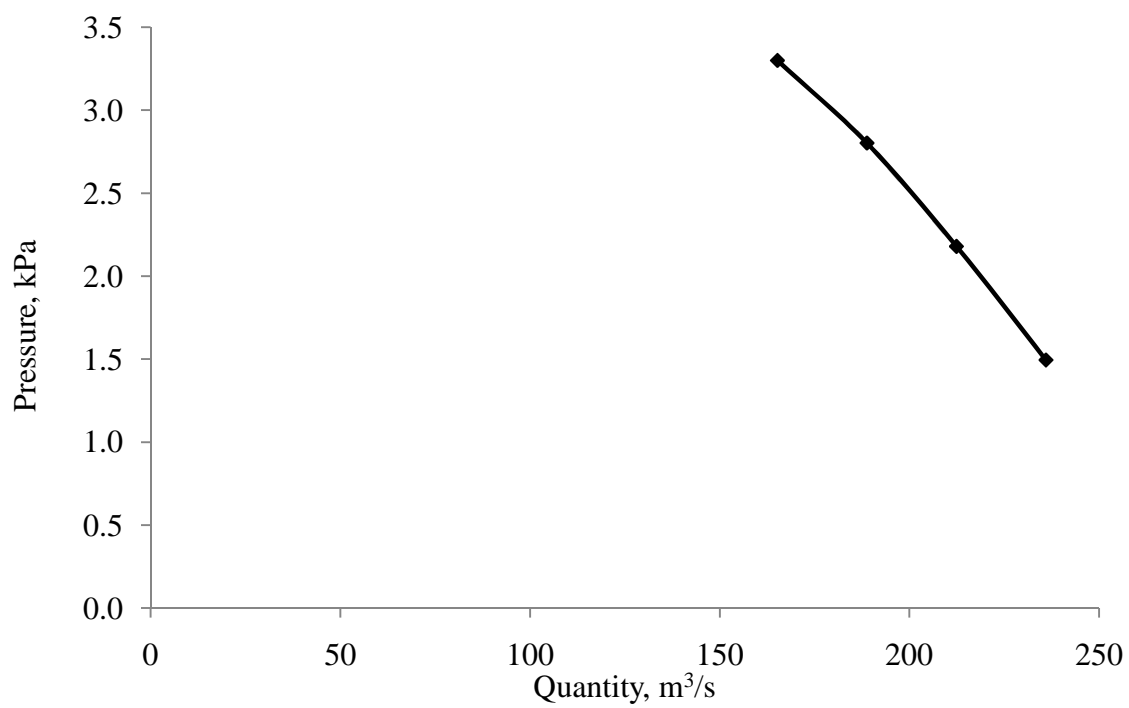


FIGURE 6.1 Mine fan characteristic curve

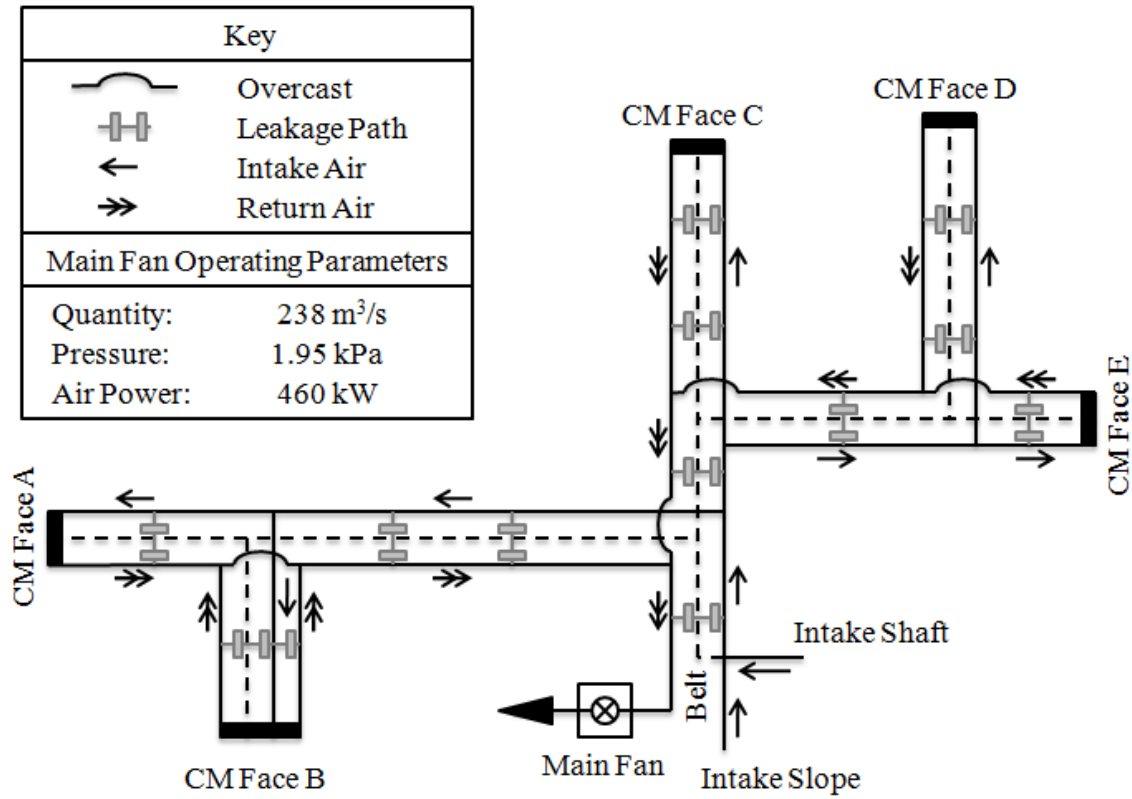


FIGURE 6.2 Base ventilation schematic

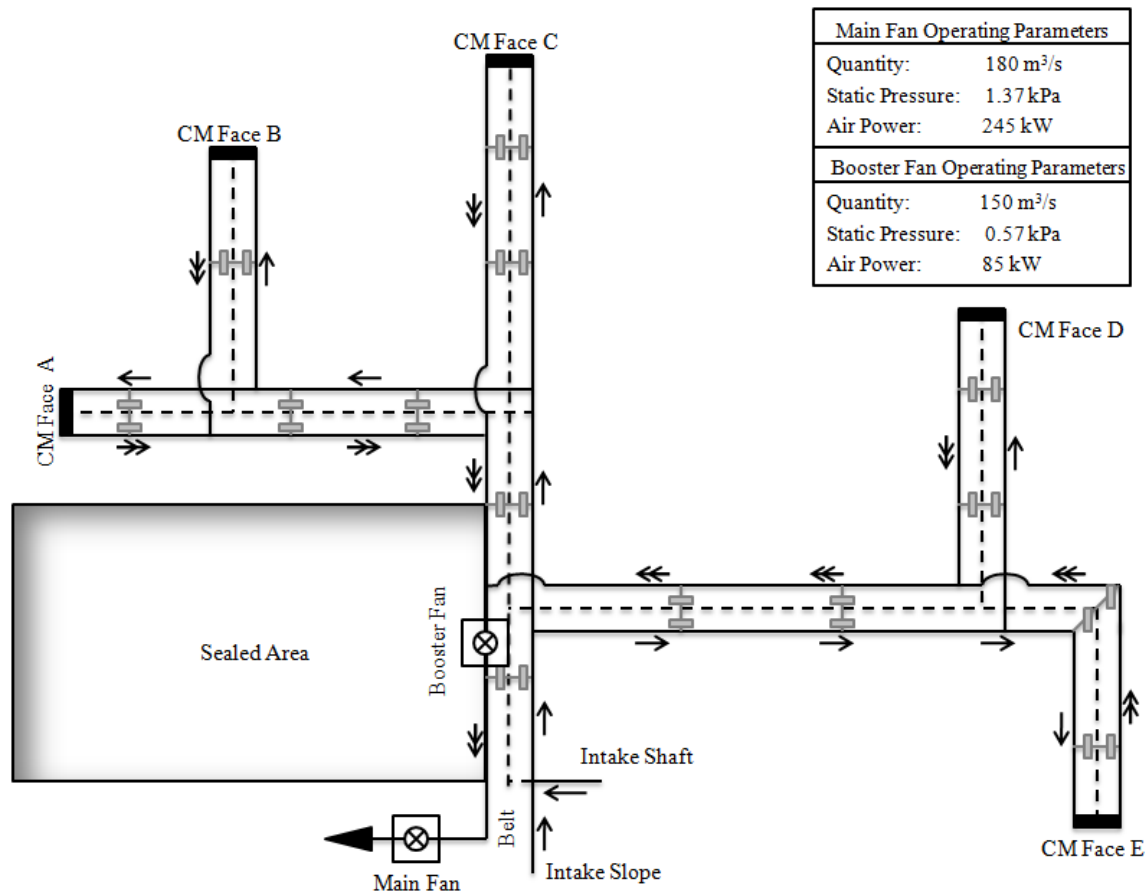


Figure 6.3 Expanded ventilation schematic

CHAPTER 7

CFD VENTILATION MODELING

7.1 Introduction

The effect of booster fans on coal mine ventilation systems has been evaluated using a scaled physical model (Chapter 4), a scaled numeric model (Chapter 5), and a full-scale numeric model (Chapter 6). All of these models are useful and help to demonstrate how booster fans can be used to increase the efficiency of coal mine ventilation systems. These models are also useful in identifying characteristics of recirculation in booster fans systems and they highlight the importance of fan size and placement in reducing the potential for recirculation. Though these models provide valuable data regarding flow quantity and pressure losses in the ventilation systems, they do not provide data detailing the flow characteristics, including the streamlines, pressure profile, and velocity distribution, within the airways and therefore they do not represent the full complexity of flow in ventilation systems. To understand the detailed flow characteristics of a ventilation system with a booster fan, CFD models were developed using ANSYS FLUENT. Two-dimensional models were developed because full-scale, three-dimensional mine ventilation models are too computationally intensive for the available resources. Though two-dimensional models do not represent the full complexity of the flow dynamics, they do provide a good representation of the flow

characteristics and they allow for qualitative analysis of how the flow dynamics are affected by the system characteristics. In this chapter, the CFD models were used to evaluate how the number of booster fans, the booster fan placement, the location and geometry of the fan installation, and the construction of the airlock system affect the flow characteristics and the localized efficiency of the ventilation system.

7.2 CFD Model Parameters

Eight CFD models were developed to evaluate the effects of the fan installation parameters, including the quality of the airlock construction, the number of booster fans, and the geometry of the fan installation on the localized efficiency. These models were developed to include only the region immediately around the booster fan, including two return entries and an airlock system. The models are represented schematically in Appendix B. In addition, Appendix C includes a detailed description of the geometry, input parameters, and results for scenario 2, as an example.

In all eight of the experimental scenarios, the input parameters including the intake pressure, the intake velocity, the fan pressures, and the pressure at the outlets were consistent. To evaluate the effect of the quality of the ventilation control devices on the flow distribution, the parameters used to characterize the resistance of the bulkhead and airlock system were varied. The bulkhead and airlock system were modeled as porous media defined by a permeability, α in m^2 , a pressure jump coefficient, C_2 in m^{-1} , and a thickness, Δm in m. These parameters were calculated based on a typical coal mine's stopping resistance values and the calculation is shown in Appendix D. Scenario 1

represents a high resistance bulkhead and airlock system, scenario 2 represents a low resistance bulkhead and airlock, and scenario 3 represents a system with no airlock.

Additionally, the number of booster fans in the models and the dimension of the fan installation area were varied. Scenario 2 represents the base case with a single fan and a typical entry width of 6 m. In scenario 4, the bulkhead was moved closer to the intake entries; in scenario 5, the angle of the crosscuts was changed from 90° to 45°; and in scenario 6, the entry width was increased from 6 m to 12 m. Finally, two booster fans in parallel were used in scenarios 7 and 8. In scenario 7, the entry width was 6 m with 1 m separating the fans and in scenario 8, the entry width was 12 m with 3 m separating the fans. The model input parameters, including the bulkhead and airlock parameters, the width of the fan installation, and the initial pressures and velocities are summarized in Table 7.1. All of the models were run using a realizable k-epsilon turbulence method and the models were run until continuity, x-velocity, y-velocity, k, and epsilon converged to values less than 0.001.

7.3 CFD Results

For each CFD model, the average total gauge pressure and the average velocity magnitude were calculated for critical locations in the model including the intakes, outlets, airlock, bulkhead, and fans. Table 7.2 summarizes these data for the inlets and outlets and Table 7.3 summarizes these data for the fans, bulkhead, and airlock. In Table 7.2, three velocities are presented for each of the inlets and outlets: V_x , which represents the average velocity of the flow in the x-direction, V_y , which represents the average velocity of the flow in the y-direction, and V , which represents the average magnitude of

the velocity with no regard for the flow direction. All of these velocities were calculated using FLUENT. In ventilation models in VnetPC and Ventsim, where the cross-sectional areas of the airways do not change, it is typical to sum the velocities at the inlets and compare them to the velocities at the outlets. As long as the cross-sectional areas are consistent, the velocity of the flow into the systems should be equal to the velocity of the flow out of the system; however, this assumes that the flow is perpendicular to the cross-sectional area of the airway. FLUENT does not operate under this constraint. From Table 7.2, comparing the average velocity magnitudes (V) at the inlets to the average velocity magnitudes at the outlets, the flow velocities do not appear to balance; however, there is balance when only the average x-velocities (V_x) of the inlets and outlets are compared. This highlights the fact that even in relatively simple CFD models, the results can be complex and confusing and care must be taken to interpret the data so that the results are correct and meaningful. In Table 7.3, only the average velocity magnitudes (V) are reported; however, it is important to note that in the bulkhead and the airlock, the flow direction is from the high pressure (outlet) side to the low pressure (inlet) side. Perhaps one of the best ways to interpret CFD data is through flow visualization. As an example, Figure 7.1 shows the velocity vectors at the intakes and outlets for scenario 2 with the calculated pressures and velocities noted and Figure 7.2 shows the total pressure contours with the calculated average pressures and average velocity magnitudes noted. Plots of the velocity vectors for scenarios 2, 3, 5, 6, and 8 are shown in Figures 7.3 through 7.7. In these images, the color and density of the data points indicates the velocity: high-velocity regions are represented by dense red data points and low-velocity regions are represented by widely spaced blue data points.

Comparing scenarios 1 and 2 where the porous characteristics of the bulkhead and the airlock system were varied, the flow characteristics of these scenarios are similar, but when the resistance of the bulkhead and airlock system was decreased, the velocity of the flow through these regions increased. In scenario 1, which had a high resistance bulkhead and airlock, the average velocity of the flow through the bulkhead was 0.17 m/s and the average velocity of the flow through the airlock was 0.02 m/s, indicating a small amount of recirculation. In scenario 2, which had a low resistance bulkhead and airlock, the average velocity of the flow through the bulkhead was 0.34 m/s and the average velocity of the flow through the airlock was 0.04 m/s. Recirculation in this scenario was slightly higher. In both scenarios, the average pressure in the regions near the bulkhead and the airlock were similar.

In scenario 3, the airlock was removed to simulate a system with a maximum potential for recirculation. Physically, this scenario is improbable because even with the doors in the airlock completely open there would be some resistance from the construction surrounding the airlock doors. However, this resistance is hard to quantify and removing the airlock completely from the model creates a system with a maximum potential for recirculation and represents a worst case scenario. When compared to scenarios 1 and 2, removing the airlock from the first entry drastically affected the flow characteristics of the system. Using the same initial conditions as scenario 2 in scenario 3, the average pressure at the inlets increased by about 486 Pa, the average pressure in the bulkhead decreased by 179 Pa, the average velocity of flow through the bulkhead increased by 0.07 m/s, and the average velocity of flow through the fan increased by 9.49 m/s, indicating significantly more local recirculation.

Asymmetry of flow through the fan is notable in most of the experimental scenarios. The asymmetry of the systems and the pressure imbalances in the intakes and the returns contribute to the curvature of the flow profiles. Because this flow imbalance has the tendency to increase the wear on the fans and decrease system efficiency, several approaches were taken to increase the symmetry of the flow. In scenario 4, the bulkhead and fan were moved closer to the intake, in line with the airlock system. In this system, the flow near the inlets is more symmetric, but the asymmetry of the flow through the fan and the outlets is increased. The pressure at the inlets is also higher than the pressure in scenario 2. In scenario 5, decreasing the angle of the crosscuts from 90° to 45° increased the symmetry of the flow at the inlets and the outlets but it did not increase the symmetry of the flow exhausting from the fan. Additionally, the width of the fan installation was increased in scenario 6. In this system, there is increased flow separation on the inlet and outlet sides of the bulkhead; however, there is good symmetry to the flow exhausting from the fan.

Finally, the effect of increasing the entry width on flow symmetry was evaluated for systems with 2 fans in parallel. Scenario 7 illustrates the flow characteristics of a system with 2 fans and an entry width of 6 m. In this scenario, the average velocity of the flow through fan 1 is 20.71 m/s and the average velocity of the flow through fan 2 is 23.61 m/s. Scenario 8 illustrates the flow characteristics of a system with 2 fans and an entry width of 12 m. In this scenario, the average velocity of the flow through fan 1 is 20.25 m/s and the average velocity of the flow through fan 2 is 24.00 m/s. In contrast to the single fan scenarios, in systems with 2 fans, increasing the entry width does not significantly increase the flow symmetry.

7.4 CFD Discussion

Based on the experimental results from the laboratory model (Chapter 4), significant local recirculation was anticipated in the CFD models as the resistance of the ventilation control devices was decreased. In scenario 1, the flow velocity through the bulkhead was about 3% of the total flow velocity at the intakes and in scenario 2, the flow velocity through the bulkhead was about 6% of the total flow velocity at the intakes. In both scenarios, the flow through the bulkhead was from the high pressure (outlet) side to the low pressure (inlet) side. Though there is a notable difference between the high resistance system (scenario 1) and the low resistance system (scenario 2), the recirculation was not as large as anticipated based on the laboratory model experiments (Chapter 4). This difference can be attributed to several factors. First, the laboratory model has one outlet while the CFD models have two outlets, and the addition of a second outlet may have substantially changed the dynamics of local recirculation. Second, the bypass valve in the laboratory model may not have been representative of the leakage potential in a full-scale airlock system. And third, the porosity characteristics of the airlock and bulkhead were calculated based on the resistance of individual stoppings; because bulkheads and airlock systems generally include doors, using resistance of stoppings may have been too conservative. The CFD data do demonstrate that with a robust airlock system and bulkhead, local recirculation can be limited to very low levels. In scenario 3, with no airlock, significant uncontrolled recirculation occurred, and this is a situation that must be avoided. However, the scenarios with airlocks showed relatively little recirculation.

In addition to the construction of the bulkhead and airlock system, the efficiencies of booster fans installations are affected by the installation parameters. As demonstrated by scenario 4, decreasing the distance between the bulkhead and the inlets increases the efficiency of the system by reducing backflow and turbulence in the intake near the bulkhead but it does not substantially affect the flow symmetry. Reducing the angle of the crosscuts, as demonstrated in scenario 5, helps to balance the inlet and outlet pressures and increases the symmetry of the flow into the fan, but it does not significantly reduce the asymmetry of the flow at the fan exhaust. In systems with a single fan, increasing the width of the installation has the potential to increase the flow symmetry at the fan intake and exhaust; however, in systems with two fans, increasing the width of the installations may not increase the symmetry of the flow. In systems with two parallel booster fans, the symmetry of the flow is a particularly important consideration because multiple fan installations are likely to include multiple fans with the same characteristics. If the flows through the fans in these systems are not balanced, the pressure and quantity demands on one of the fans is likely to be significantly higher, leading to lower efficiency, increased wear, and higher maintenance costs. To increase the symmetry of the flow in the exhaust, increasing the distance between the bulkhead and the pressure outlets by increasing the size of the pillar, changing the geometry of the airways, reducing the booster fan pressure, and in two-fan systems, modifying the distance between the fans may be helpful. Finally, the efficiency of the fans in the booster fan system is an important consideration: as long as the bulkhead and airlock system are well built and well maintained, the efficiency of the booster fan system will likely be dictated

by the efficiencies of the fans. For this reason, the use of custom built booster fans with high efficiencies may be most practical.

TABLE 7.1 CFD model input parameters

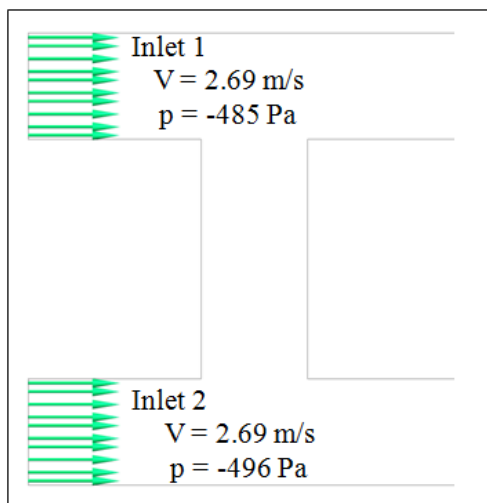
Scenario	Fan 1	Fan 2	Outlet 1	Outlet 2	Inlet 1		Inlet 2		Dimension	Bulkhead		Airlock	
	Δp , Pa	Δp , Pa	Δp , Pa	Δp , Pa	Δp , Pa	V, m/s	Δp , Pa	V, m/s	Width, m	α , m ²	C ₂ , m ⁻¹	α , m ²	C ₂ , m ⁻¹
1	1000	--	0	50	0	2.69	0	2.69	6	1.7984	3,003,303	1.7984	3,003,303
2	1000	--	0	50	0	2.69	0	2.69	6	1.7984	669,764	1.7984	669,764
3	1000	--	0	50	0	2.69	0	2.69	6	1.7984	669,764	--	--
4	1000	--	0	50	0	2.69	0	2.69	6	1.7984	669,764	1.7984	669,764
5	1000	--	0	20	0	2.69	0	2.69	6	1.7984	669,764	1.7984	669,764
6	1000	--	0	50	0	2.69	0	2.69	12	1.7984	669,764	1.7984	669,764
7	1000	1000	0	50	0	5.38	0	5.38	6	1.7984	669,764	1.7984	669,764
8	1000	1000	0	50	0	5.38	0	5.38	12	1.7984	669,764	1.7984	669,764

TABLE 7.2 CFD model results for the inlets and outlets

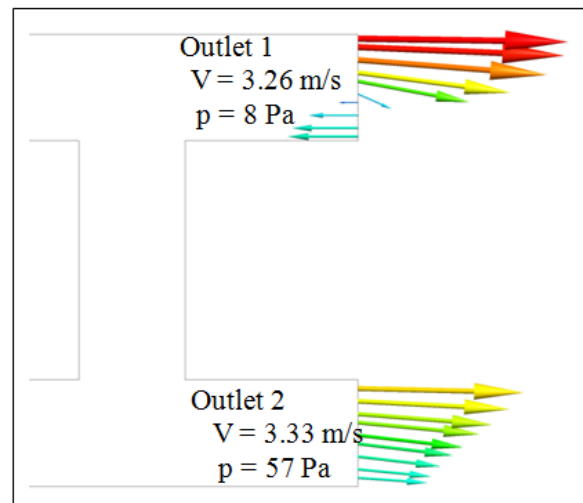
Scenario	Inlet 1				Inlet 2				Outlet 1				Outlet 2			
	p, Pa	V _x , m/s	V _y , m/s	V, m/s	p, Pa	V _x , m/s	V _y , m/s	V, m/s	p, Pa	V _x , m/s	V _y , m/s	V, m/s	p, Pa	V _x , m/s	V _y , m/s	V, m/s
1	-492	2.69	0.00	2.69	-503	2.69	0.00	2.69	8	2.03	-0.25	3.24	58	3.34	-0.25	3.35
2	-485	2.69	0.00	2.69	-496	2.69	0.00	2.69	8	2.04	-0.25	3.26	57	3.32	-0.24	3.33
3	10	2.69	0.00	2.69	-20	2.69	0.00	2.69	1	0.04	-0.11	1.26	72	5.33	-0.30	5.34
4	-504	2.69	0.00	2.69	-516	2.69	0.00	2.69	13	2.69	-0.42	4.13	55	2.67	-0.13	2.68
5	-555	2.69	0.00	2.69	-557	2.69	0.00	2.69	7	3.38	0.09	3.38	27	2.00	-0.08	2.00
6	-550	2.69	0.00	2.69	-562	2.69	0.00	2.69	12	3.27	-0.26	3.74	53	2.10	-0.06	2.10
7	-537	5.38	0.00	5.38	-582	5.38	0.00	5.38	8	1.86	-0.10	2.99	99	8.89	0.10	8.89
8	-398	5.38	0.00	5.38	-443	5.38	0.00	5.38	7	1.99	-0.18	2.95	97	8.76	0.09	8.76

TABLE 7.3 CFD model results for the fans, bulkhead, and airlock

Scenario	Fan 1		Fan 2		Bulkhead		Airlock	
	p, Pa	V, m/s	p, Pa	V, m/s	p, Pa	V, m/s	p, Pa	V, m/s
1	-180	22.08	--	--	-608	0.17	-236	0.02
2	-179	22.23	--	--	-611	0.34	-232	0.04
3	96	31.72	--	--	-790	0.41	--	--
4	-179	22.26	--	--	-653	0.35	-238	0.04
5	-226	22.19	--	--	-652	0.34	-277	0.04
6	-233	22.21	--	--	-660	0.39	-264	0.04
7	-300	20.71	-227	23.61	-665	0.62	-261	0.04
8	-242	20.25	-154	24.00	-565	0.33	-189	0.03



(a) Inlet velocity vectors



(b) Outlet velocity vectors

FIGURE 7.1 Scenario 2 (a) inlet and (b) outlet velocity vectors, average velocities, and average pressures

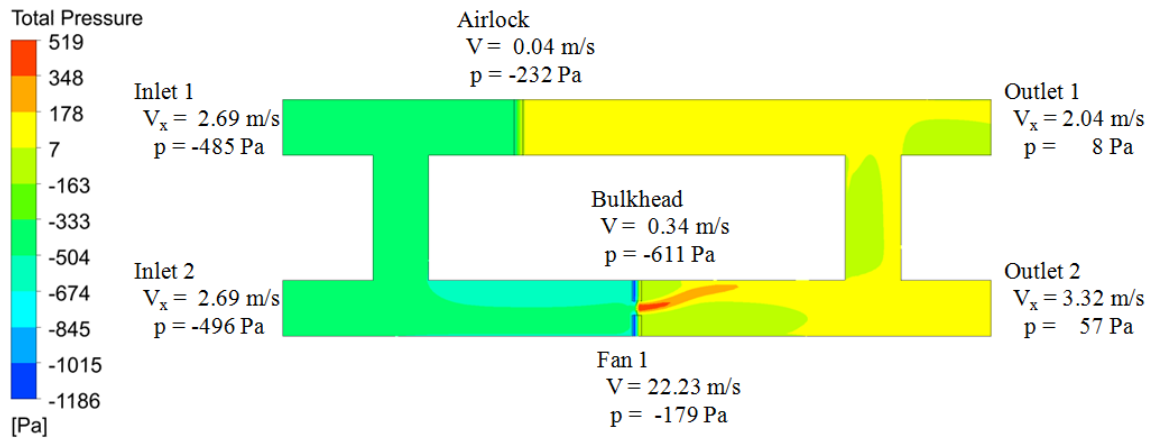


FIGURE 7.2 Scenario 2 pressure contours

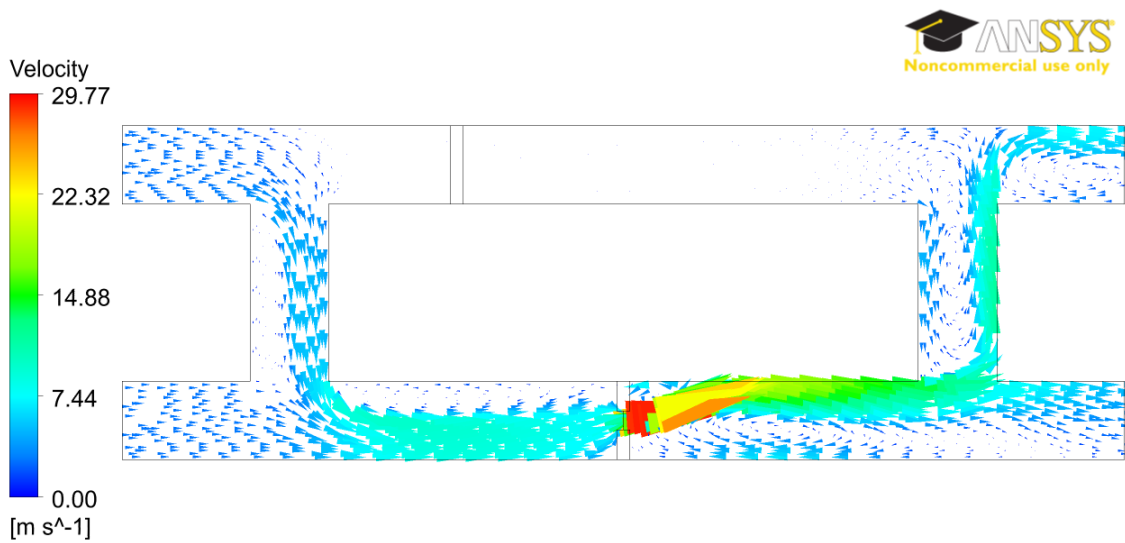


FIGURE 7.3 Plot of the velocity vectors for scenario 2

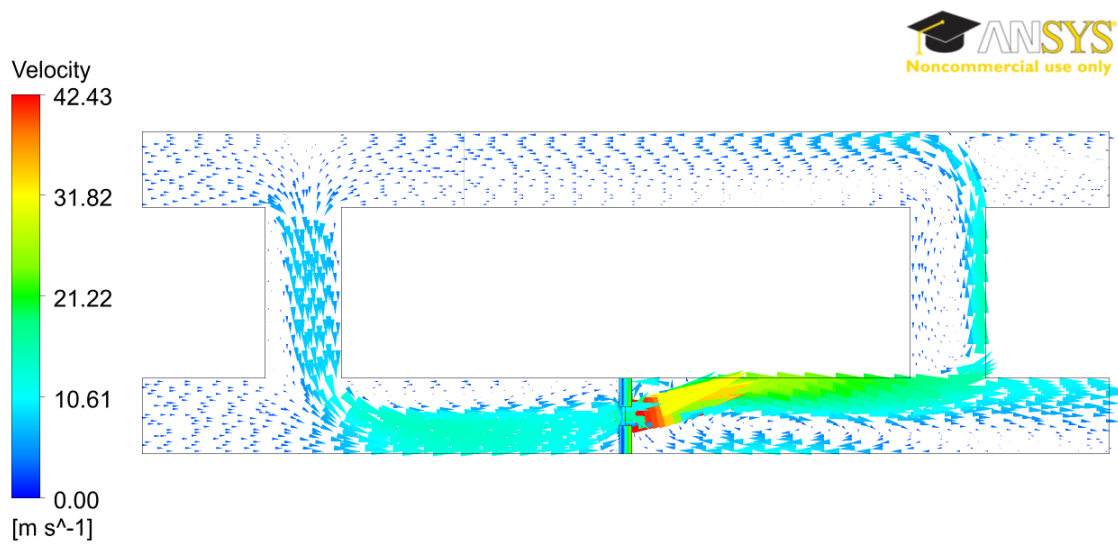


FIGURE 7.4 Plot of the velocity vectors for scenario 3

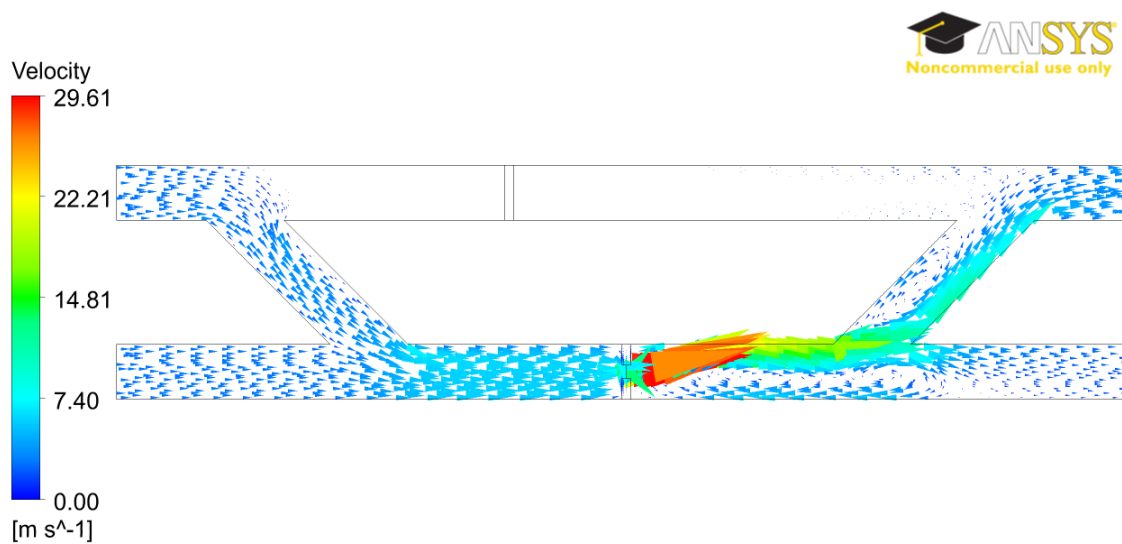


FIGURE 7.5 Plot of the velocity vectors for scenario 5

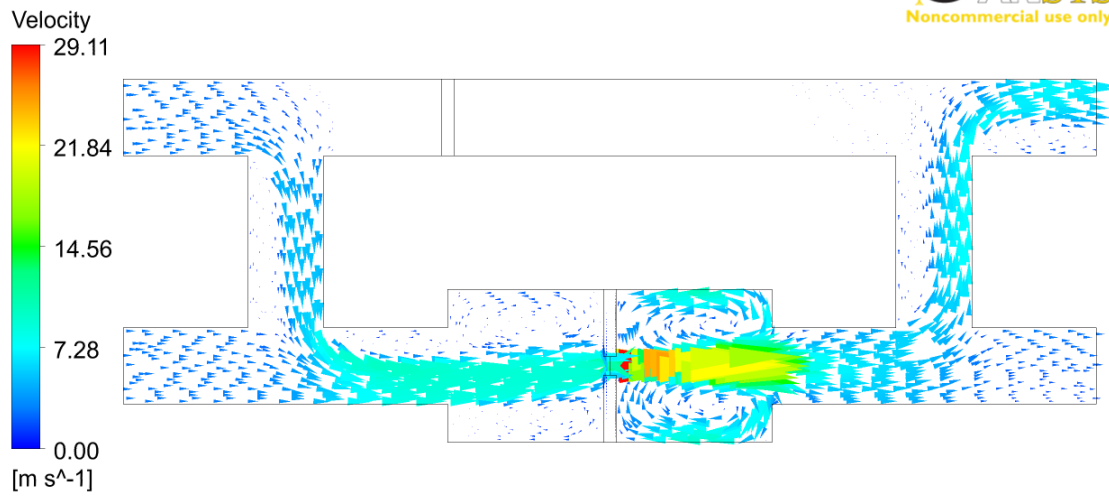


FIGURE 7.6 Plot of the velocity vectors for scenario 6

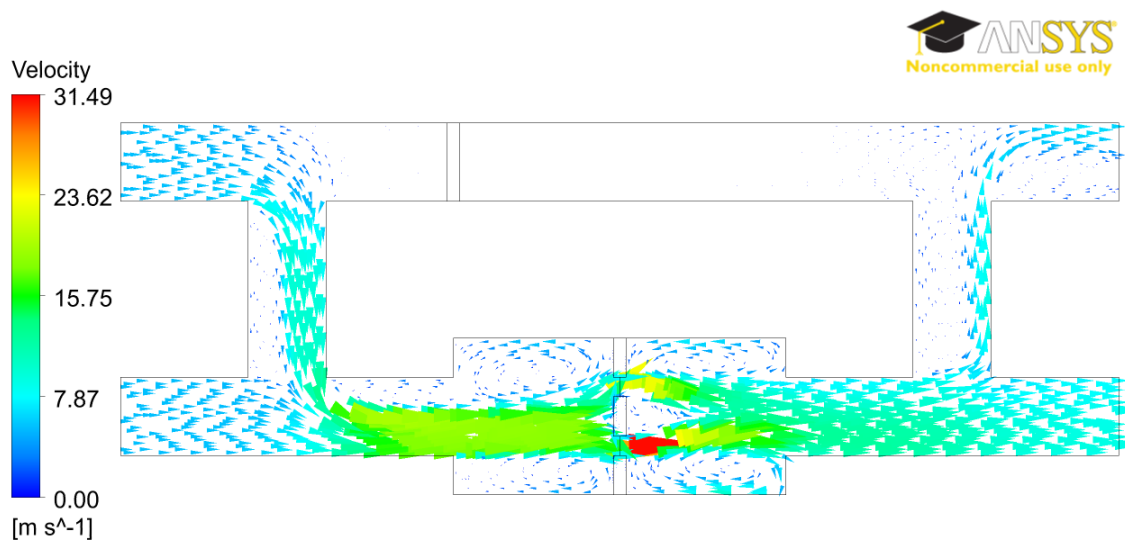


FIGURE 7.7 Plot of the velocity vectors for scenario 8

CHAPTER 8

CONCLUSIONS AND RECOMMENDATIONS

8.1 Conclusions

As underground coal mines in the United States are developed in deeper deposits and as the resistance of active mines increases due to more extensive development, adequate mine ventilation becomes difficult. In these mines, the volumetric efficiency of the ventilation systems is often low and high pressure main fans with high volumetric flow capacity are necessary to achieve adequate airflow. The use of booster fans is one method of increasing the efficiency of ventilation system, reducing the required main fan pressure and reducing the total flow rate by reducing leakage. The efficiency gain is demonstrated by the data presented in Chapter 6. In Mine A, which is relatively shallow and has a relatively high volumetric efficiency, a booster fan has the potential to decrease the air power required by the ventilation system by 110 kW. This level of energy savings is significant and in deeper or more extensive mines with lower ventilation efficiencies and higher airflow requirements, the power saving due to booster fans is likely to be notably higher. In addition, by reducing leakage, installing booster fans can reduce the pressure of the main surface fans and this is an important safety consideration.

Booster fans have the potential to increase the efficiency of coal mine ventilation systems, but there are risks associated with their use, most notably recirculation, and as the efficiency of ventilation systems with booster fans increases, so does the potential for recirculation. However, the potential for recirculation is a risk that can be managed. As the laboratory model data demonstrate, the size of the booster fan relative to the main fan is an important consideration. If the booster fan operates at a pressure as large as or larger than the main fan pressure, there is significant potential for recirculation. In addition, the location of a booster fan in a ventilation system has a significant impact on recirculation. The closer the booster fan is to the development sections, the greater the likelihood of recirculation regardless of the booster fan pressure. To limit the potential for system recirculation, the location of a booster fan in a ventilation system should be thoroughly evaluated. The fan should be located so that pressures in the intake airways are higher than pressures in the return airways and the formation of neutral points should be avoided. It is important to note, as the mine develops further from a booster fan, the section resistance increases and the potential for recirculation decreases. Though booster fans can increase ventilation efficiency, this benefit must be balanced with the risk of system recirculation.

In addition to the size and location of a booster fan, system recirculation is strongly dependent on the quality of the ventilation control devices and this is best demonstrated by the laboratory model data. In these experiments, system recirculation increased substantially as the resistance of the crosscut valves decreased. Notably, the largest percentages of recirculation occurred in the system with the lowest resistance

valves and the systems with highest resistance valves experience very low percentages of recirculation.

Local recirculation is also an important consideration in booster fan systems. When local recirculation occurs, energy is consumed as air is recirculated around the fan but there is no increase in airflow through the system. As the laboratory model data indicate, local recirculation is dependent on the quality of the airlock system and the pressure across the system. If the bulkhead and airlock doors are not well built and well maintained, the potential for local recirculation is significant; however, as the CFD data demonstrate, high resistance bulkhead and airlock systems will limit local recirculation. Additionally, installing booster fans in the return airways, where airlock doors are used less frequently, may help to reduce damage to the airlock system and thus reduce local recirculation.

Finally, the CFD data demonstrate that the efficiency of a booster fan system is affected by the size of the fan installation area and the number of booster fans. In systems with a single fan, widening the entries near the booster fans may help to produce more symmetric flows. Though wider entries produce more symmetric flow, some energy is consumed by additional turbulence and back flow in these systems. The location of the bulkhead relative to the inlet and the angle of the crosscuts also affects the flow symmetry. Reducing the angle of the crosscuts and reducing the distance between the bulkhead and the inlet helps to balance the flow and reduce shock losses. Flow symmetry is probably most critical in multiple fan installations, so that the fan duties are similar. Ultimately, the efficiency of a booster fan system is likely to be most dependent

on the efficiencies of the fans themselves. Custom built fans tend to have high efficiencies and, although expensive, they may be best suited for use as booster fans.

8.2 Recommendations

Though recirculation is an important risk associated with the use of booster fans, it is not the only topic that needs to be addressed. Optimization of ventilation systems with booster fans is one area in need of further research. Several methods of optimization have been researched and proposed, but few of these methods are being used in industry. In practice, the optimization of ventilation systems is often based on experience. A robust and practical method of optimization that includes optimization over the entire life of a mine cycle would be valuable. A method like this would help to address issues associated with the sizing and location of multiple booster fans in complex ventilation systems.

Next, though booster fans have the potential to increase the ventilation system efficiency, and thus to reduce the operating costs, the capital costs to install a booster fan are likely to be quite high. It would be helpful to perform economic evaluations of a booster fan installation in comparison with traditional methods of increasing ventilation capacity including slashing exiting entries, developing new entries, driving additional shafts, and upgrading the surface fans. Though booster fans do increase ventilation efficiency and capacity, they may not be the most cost effective alternative for all circumstances. Without an economic evaluation, it is difficult to estimate the true cost saving associated with booster fans.

Additionally, it would be beneficial to develop large-scale, three-dimensional CFD models of a booster fan ventilation system that is consistent with mining practices in the United States and include multiple intake, return, and neutral airways. Models like these would be useful in identifying the detailed flow characteristics of booster fan systems including the effect of the neutral airways on recirculation. Models like these could also be used to evaluate the effects of gravity, gas layering, and dust and gas transport on mine ventilation systems. Large-scale, three-dimensional models were not included as part of this work because they were too computationally intensive for the available resources. In addition, in order for models like these to provide meaningful data, the porous characteristics of the ventilation control devices should be well defined based on experimental measurements.

Finally, before booster fans are used practically in the United States, methods for controlling the risks associated with their use need to be identified and defined. Booster fans have been proven as an effective means of increasing ventilation capacity in mines in the United Kingdom and Australia and in these countries, the risks associated with the use of booster fans, including recirculation, are well-defined and managed. In these countries, before booster fans are installed, the hazards associated with their use are identified, the risk associated with each hazard is assessed, and methods to control the hazards are defined along with levels of acceptable risk. Though recirculation is allowed in ventilation systems in the United Kingdom and Australia, the use of extensive and robust atmospheric monitoring systems helps to minimize the risks associated with the use of booster fans and recirculation. Electrical interlocking of the fans has also been suggested as a valid control method. In the United States, specific requirements for the

safe and effective use of booster fans in underground coal mine ventilation systems need to be defined.

APPENDIX A

LABORATORY MODEL EXPERIMENTAL PARAMETERS AND RESULTS

TABLE A.1 Laboratory model experimental parameters

Experiment No.	V1 Open Area, %	V2 Open Area, %	V3 Open Area, %	V4 Open Area, %	Crosscut A, B, and C Open Area, %
1	100	0.0	15.9	28.0	3.9
2	100	0.0	15.9	28.0	27.2
3	100	0.0	15.9	28.0	15.4
4	100	0.0	15.9	28.0	15.4
5	100	0.0	15.9	28.0	15.4
6	100	15.9	15.9	28.0	15.4
7	100	15.9	15.9	28.0	15.4
8	100	15.9	15.9	28.0	15.4
9	100	15.9	15.9	28.0	15.4
10	100	28.0	15.9	28.0	3.9
11	100	28.0	15.9	28.0	27.2
12	100	28.0	15.9	28.0	15.4
13	100	28.0	15.9	28.0	15.4
14	100	15.9	15.9	28.0	3.9
15	100	15.9	15.9	28.0	3.9
16	100	15.9	15.9	28.0	27.2
17	100	15.9	15.9	28.0	27.2
18	100	15.9	15.9	28.0	27.2
19	100	15.9	15.9	28.0	3.9
20	100	28.0	15.9	28.0	15.4
21	100	15.9	15.9	28.0	27.2
22	100	15.9	15.9	28.0	3.9
23	100	28.0	15.9	28.0	15.4
24	100	0.0	15.9	28.0	15.4
25	100	15.9	15.9	28.0	15.4
26	0	100.0	15.9	28.0	15.4
27	0	100.0	15.9	28.0	15.4
28	0	100.0	15.9	28.0	15.4
29	0	100.0	15.9	28.0	3.9
30	0	100.0	15.9	28.0	3.9
31	0	100.0	15.9	28.0	3.9
32	0	100.0	15.9	28.0	27.2
33	0	100.0	15.9	28.0	27.2
34	0	100.0	15.9	28.0	27.2

TABLE A.2 Summary of fan parameters and flow volumes

Experiment No.	Main Fan		Booster Fan		Face 1 Q, m ³ /s	Face 2 Q, m ³ /s
	P, kPa	Q, m ³ /s	P, kPa	Q, m ³ /s		
1	1.26	0.218	1.30	0.220	0.164	0.050
2	1.27	0.209	1.28	0.240	0.145	0.046
3	0.52	0.176	1.31	0.198	0.135	0.041
4	2.30	0.262	1.29	0.264	0.178	0.055
5	1.34	0.193	0.56	0.188	0.123	0.042
6	0.62	0.141	0.50	0.207	0.101	0.034
7	0.50	0.207	2.16	0.361	0.161	0.046
8	2.33	0.236	0.42	0.264	0.144	0.049
9	2.24	0.275	1.99	0.407	0.195	0.063
10	1.28	0.200	1.06	0.343	0.149	0.048
11	1.30	0.198	1.03	0.357	0.130	0.041
12	0.54	0.157	1.07	0.324	0.117	0.036
13	2.31	0.250	1.01	0.367	0.159	0.052
14	1.29	0.185	0.46	0.231	0.132	0.042
15	1.20	0.243	2.15	0.370	0.184	0.055
16	1.29	0.189	0.47	0.227	0.107	0.036
17	1.24	0.231	2.05	0.394	0.166	0.047
18	0.54	0.165	1.18	0.289	0.120	0.035
19	2.27	0.254	1.08	0.331	0.184	0.058
20	1.31	0.183	0.43	0.253	0.115	0.038
21	2.28	0.256	1.07	0.335	0.153	0.049
22	0.52	0.173	1.21	0.270	0.133	0.040
23	1.25	0.223	1.88	0.436	0.164	0.050
24	1.22	0.248	2.32	0.278	0.188	0.055
25	1.26	0.203	1.13	0.306	0.147	0.045
26	0.58	0.111	--	--	0.065	0.020
27	1.32	0.169	--	--	0.098	0.032
28	2.35	0.223	--	--	0.131	0.040
29	0.59	0.108	--	--	0.077	0.023
30	1.34	0.164	--	--	0.117	0.039
31	2.33	0.218	--	--	0.156	0.024
32	2.32	0.227	--	--	0.111	0.034
33	1.31	0.171	--	--	0.082	0.029
34	0.58	0.114	--	--	0.055	0.017

TABLE A.3 Summary of leakage and recirculation

Experiment No.	Leakage		System Recirculation		Local Recirculation	
	Q, m ³ /s	%	Q, m ³ /s	%	Q, m ³ /s	%
1	0.005	2	0.002	1	0.000	0
2	0.048	23	0.031	15	0.000	0
3	0.021	12	0.021	12	0.000	0
4	0.032	12	0.002	1	0.000	0
5	0.027	14	0.000	0	0.000	0
6	0.016	11	0.009	7	0.056	40
7	0.026	13	0.026	13	0.127	61
8	0.044	18	0.000	0	0.046	19
9	0.034	13	0.017	6	0.115	42
10	0.004	2	0.000	0	0.144	72
11	0.039	20	0.012	6	0.147	74
12	0.019	12	0.015	10	0.152	97
13	0.038	15	0.000	0	0.130	52
14	0.011	6	0.000	0	0.052	28
15	0.007	3	0.002	1	0.125	51
16	0.046	24	0.000	0	0.050	26
17	0.059	26	0.041	18	0.121	52
18	0.043	26	0.034	21	0.090	55
19	0.012	5	0.000	0	0.084	33
20	0.031	17	0.000	0	0.083	45
21	0.054	21	0.000	0	0.082	32
22	0.004	2	0.005	3	0.093	54
23	0.025	11	0.017	7	0.196	88
24	0.035	14	0.030	12	0.000	0
25	0.027	13	0.016	8	0.088	43
26	0.027	24	0.000	0	0.000	0
27	0.039	23	0.000	0	0.000	0
28	0.052	23	0.000	0	0.000	0
29	0.008	7	0.000	0	0.000	0
30	0.008	5	0.000	0	0.000	0
31	0.082	38	0.000	0	0.000	0
32	0.082	36	0.000	0	0.000	0
33	0.060	35	0.000	0	0.000	0
34	0.042	37	0.000	0	0.000	0

APPENDIX B

CFD MODEL SCHEMATICS

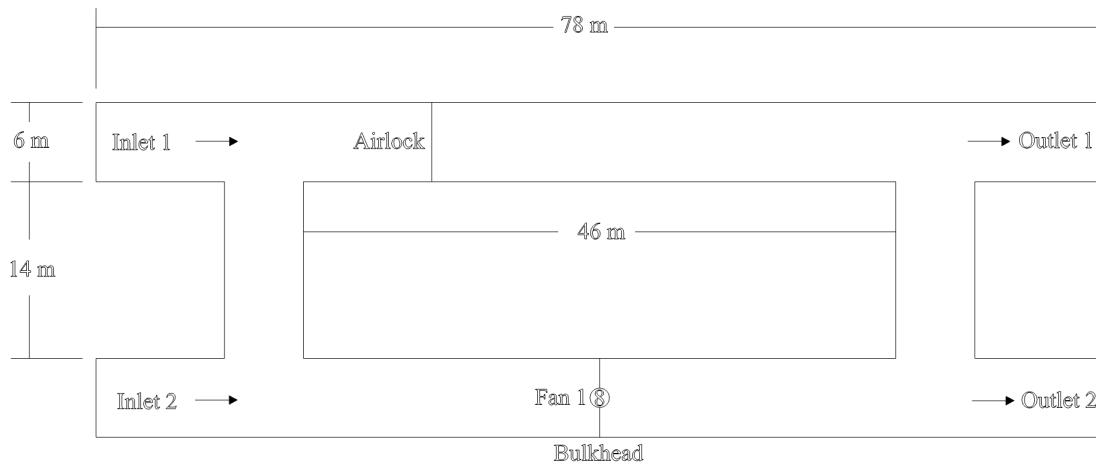


FIGURE B.1 Schematic of CFD model for scenarios 1 and 2

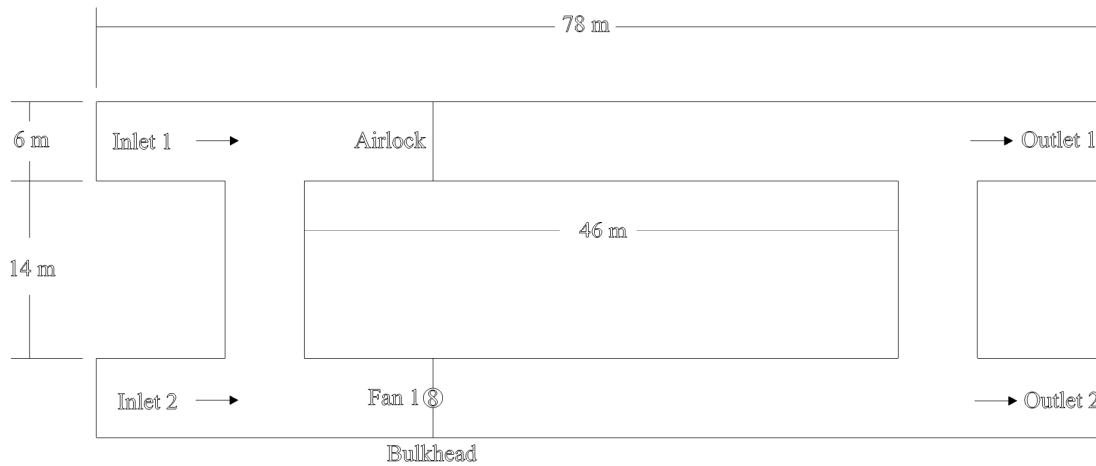


FIGURE B.2 Schematic of CFD model for scenario 4

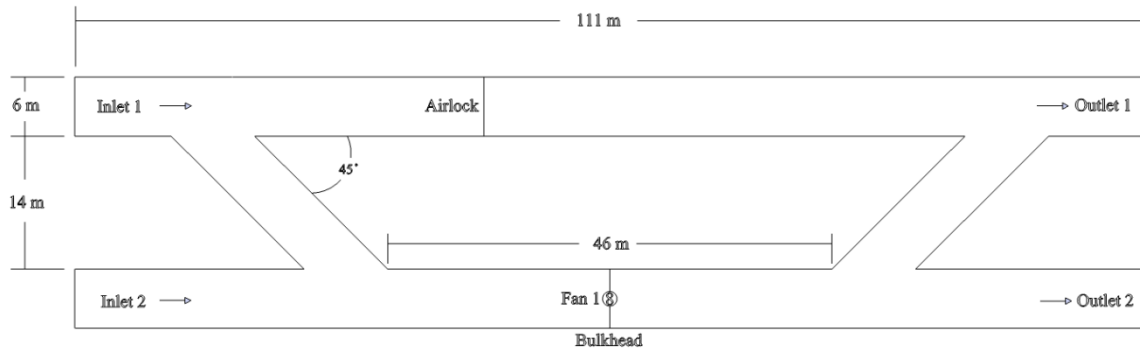


FIGURE B.3 Schematic of CFD model for scenario 5

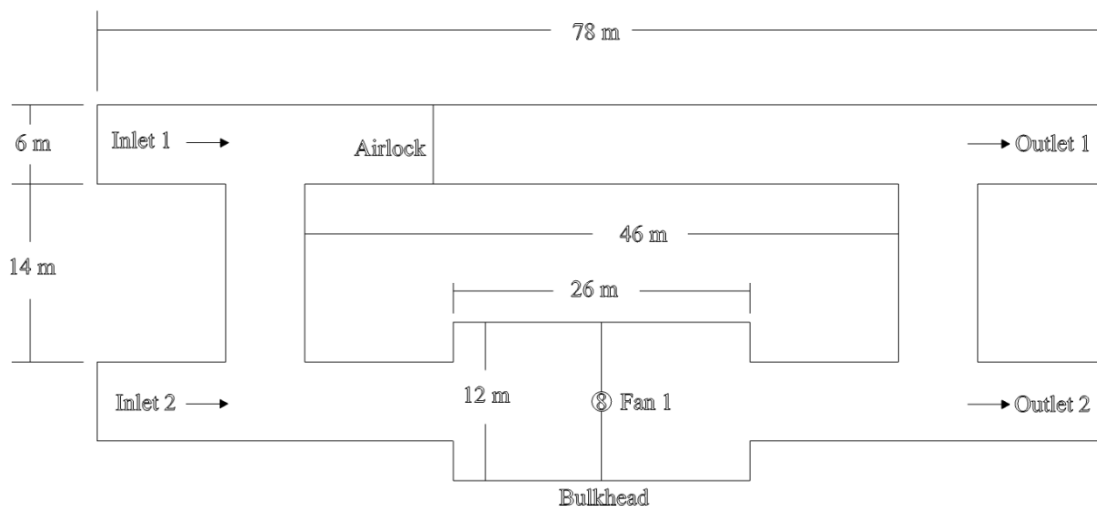


FIGURE B.4 Schematic of CFD model for scenario 6

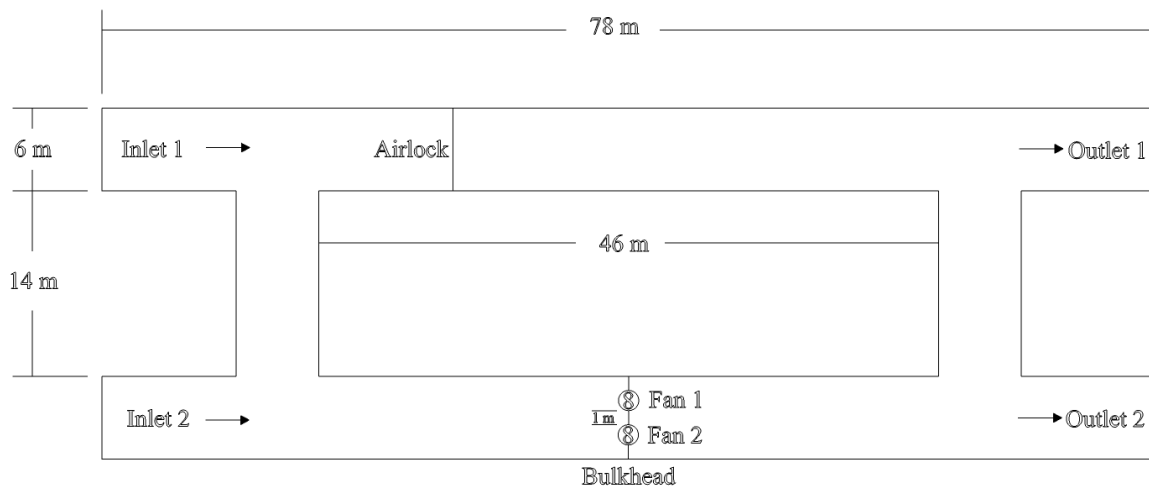


FIGURE B.5 Schematic of CFD model for scenario 7

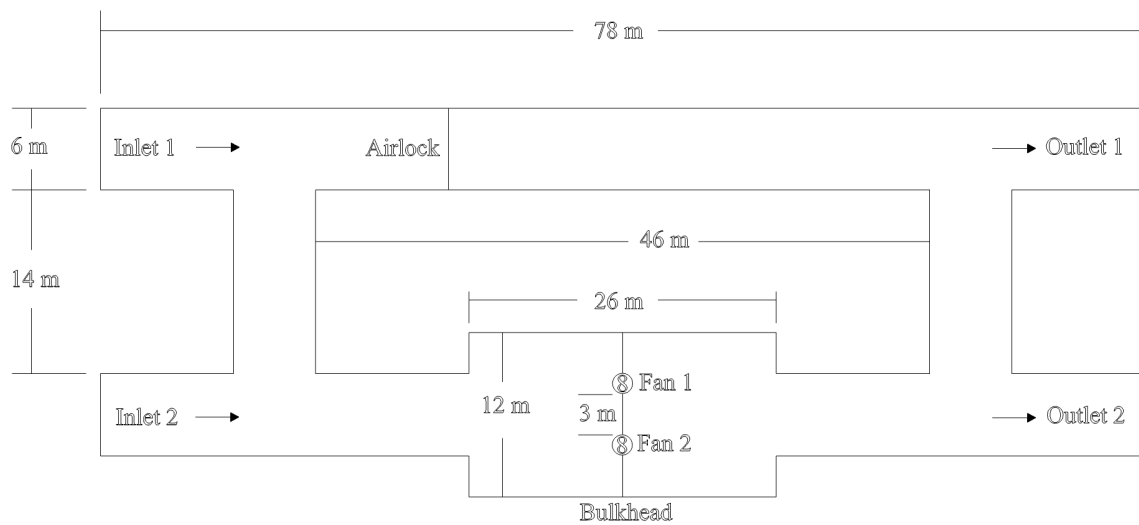


FIGURE B.6 Schematic of CFD model for scenario 8

APPENDIX C

CFD SCENARIO 2 INPUT PARAMETERS AND RESULTS

Scenario 2 represents a booster fan installation with two parallel airways. Each airway is 6 m wide and the airways are separated by a pillar that is 46 m long by 14 m wide. The airlock is 1 m wide and was placed 36 m from the inlet of the first airway. In FLUENT, the airlock porosity parameters C_2 and α were assigned values of $669,764 \text{ m}^{-1}$ and 1.7984 m^2 , respectively. The bulkhead is 1 m wide and was placed 59 m from the inlet of the second airway. For the bulkhead, C_2 was given a value of $669,764 \text{ m}^{-1}$ and α was assigned a value of 1.7984 m^2 . The fan was represented as a 1.5 m line in the middle of the bulkhead and it was assigned a pressure of 1000 Pa. A schematic of scenario 2 is shown in Figure B.1. Initially, both of the inlets were assigned pressures of 0 Pa. Outlet 1 was assigned a pressure of 0 Pa and outlet 2 was assigned a pressure of 50 Pa. As an initial condition, a velocity of 2.69 m/s was specified at each of the inlets. Figure C.1 shows a diagram of scenario 2 with the initial conditions and the input parameters noted.

After the model converged, the average pressure and average velocity magnitudes were calculated for the critical locations in the model, including the inlets, outlets, bulkhead, airlock, and fan. The average pressure at inlet 1 was -485 Pa, the average velocity magnitude was 2.69 m/s, the average x-velocity was 2.69 m/s, and the average y-velocity was 0.00 m/s; the average pressure at inlet 2 was -496 Pa, the average velocity magnitude was 2.69 m/s, the average x-velocity was 2.69 m/s, and the average y-velocity was 0.00 m/s; at outlet 1, the average pressure was 8 Pa and the average velocity magnitude was 3.26 m/s, the average x-velocity was 2.04 m/s, and the average y-velocity was -0.25 m/s; and at outlet 2, the average pressure was 57 Pa and the average velocity magnitude was 3.33 m/s, the average x-velocity was 3.32 m/s, and the average y-velocity was -0.24 m/s;. Additionally, the average pressure in the 1 m region of the bulkhead was

-611 Pa and the average velocity the flow through the bulkhead was 0.34 m/s from the high pressure (outlet) side to the low pressure (inlet) side. For the 1 m region representing the airlock system, the average pressure was -232 Pa and the average velocity of the flow through the airlock was 0.04 m/s from the high pressure (outlet) side to the low pressure (inlet) side. Finally, the average pressure at the fan was -179 Pa and the average velocity of the flow through the fan was 22.23 m/s. Figure C.2 shows the flow characteristic of scenario 2 with the average pressures and velocities for each of the critical locations noted.

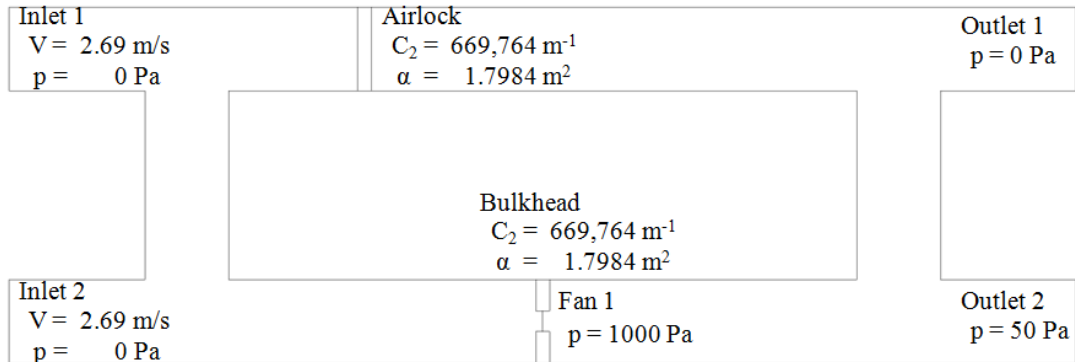


FIGURE C.1 Initial conditions and input parameters for scenario 2

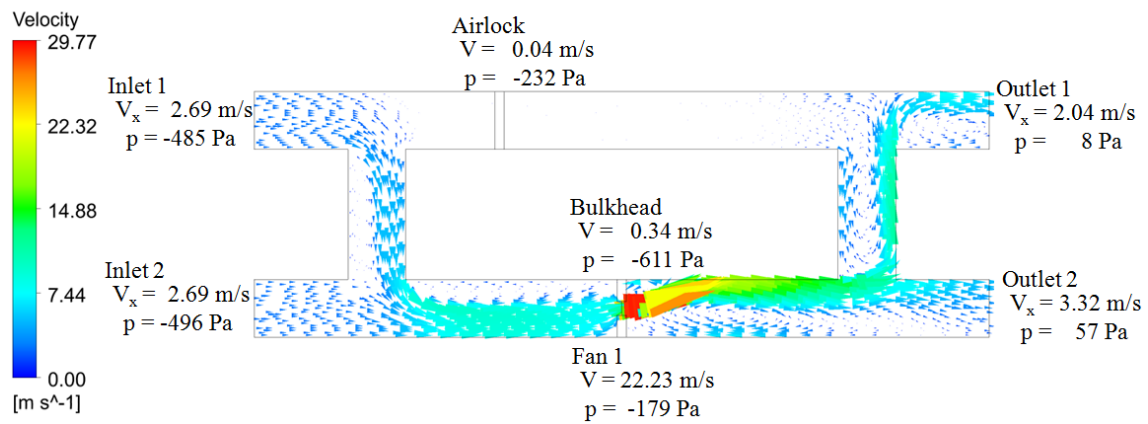


FIGURE C.2 Scenario 2 flow profile with average velocities and pressures noted

APPENDIX D

DERIVATION OF POROUS MEDIA

CHARACTERISTICS

Because FLUENT has application to a broad range of fluid flow problems, not specific to mine ventilation, there are a number of methods that can be used to model ventilation control devices such as stoppings, bulkheads, and airlock doors. In the models presents in Chapter 7, the bulkhead and the airlock system were modeled as 1 m thick porous zones. This method of modeling requires three variables to define the porosity characteristics: permeability, α in m^2 , pressure jump coefficient, C_2 in m^{-1} , and thickness, Δm in m. The relationship among these variables is based on Darcy's law and an additional term due to inertial losses (ANSYS 2010):

$$\Delta p = \frac{\mu}{\alpha} V \Delta m - C_2 \frac{1}{2} \rho V^2 \Delta m \quad (\text{D.1})$$

where

Δp = pressure difference, Pa

μ = viscosity, Pa s

α = permeability, m^2

V = velocity, m/s

Δm = thickness, m

C_2 = pressure-jump coefficient, m^{-1}

ρ = density, kg/m^3

In mine ventilation systems, the second term, due to inertial losses, dictates the flow behavior and this is reflected in Atkinson's Equation 2.1. The values C_2 were calculated based on typical resistance values for coal mine stopping as reported by Ostwald et al. (2008). The effect of the first term in Equation D.1 is relatively negligible in mine

ventilation systems and the value for α was calculated to minimize the effect of this term.

The results of these calculations are presented in Table D.1.

TABLE D.1 Derivation of C_2 and α

R, Ns^2/m^8	A, m^2	Δm , m	ρ , kg/m^3	μ , Pa s	C_2 , m^{-1}	α , m^2
2,425	13	1	1.225	1.7894×10^{-5}	669,764	1.7984
10,874	13	1	1.225	1.7894×10^{-5}	3,003,303	1.7984

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