THE UNIVERSITY OF MANITOBA

DUST CONTROL SYSTEM FOR FARM SEED CLEANING PLANTS

by

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A dissertation submitted to the Faculty of Graduate Studies of the University of Manitoba in partial fulfillment of the requirements of the degree of

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To my parents and wife Yashu

ABSTRACT

Dust control in farm seed cleaning plants is of paramount importance as harmful effects (health hazards and fire explosions) of grain dust are well know. Dust emission levels in eleven seed cleaning plants were measured and several particle size distribution analyses were conducted in order to study the grain dust properties. Approximately 40% of the farm seed cleaning plants that were surveyed in this dust monitoring study had dust levels larger than the acceptable limit of 10 mg/m^3.

Friction head losses in galvanized iron sheet metal pipes and flexible plastic pipes were determined at various air flow rates and then utilized to draw friction loss charts. Friction head losses in plastic flexible pipe bends were also measured and expressed as equivalent length of straight pipe. The frictional head losses in the plastic flexible pipes were 2 to 2.5 times larger than sheet metal pipes, whereas losses in pipe bends were almost equal to that from sheet metal elbows of the same size.

A typical exhaust system was balanced by using the friction head loss data. The fan static pressure, power, and fan rpm were established during operation of the system as well as from fan performance curves.

Two types of dumping hopper hoods were designed, fabricated and tested for their effectiveness in capturing dust. The corresponding pressure drops were also measured. The use of hoods at the dumping hopper reduced the dust concentrations considerably. Thus a partially

enclosed side draft type hood was recommended for collecting dust in a dumping hopper in farm seed cleaning plants.

Performance characteristics (collection efficiency and pressure drops) were determined for a cyclone separator and for three types of fabric filter bags. The cyclone separator was 82% efficient in the collection of grain dust while two of the three fabric filter bags tested were 99% efficient in removing grain dust from the work environment. Dust that penetrated through the filter bags followed a log-normal particle size distribution function.

Pressure drop in the cyclone separator varied with the air flow rate and was proportional to the 2.13 power of air flow rate. A linear relationship between bag pressure drop and collection efficiency was obtained in the fabric filter bags.

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1. INTRODUCTION

The seed processing industry is vital to Canadian Agriculture and is important to farmers and consumers. Seed grain should be free of other crop seeds, weed seeds, straw, chaff, and dust. The maximum percentage of pure crop seeds with maximum germination potential may be obtained through proper cleaning and grading of the seed grains. Cleaning seed grains provides a positive method of weed control. In Manitoba, a large percentage of the seed is cleaned in country elevators and on the farm. Most farmer-owned seed-cleaning plants were initially built by the registered seed growers for their own use. This was necessary in order to satisfy the seed quality standards required by Canada's seed regulatory agency - The Canadian Seed Growers Association. These plants may have developed into custom cleaning establishments. In the seed processing industry, the concern over the plant environmental dust levels and potential health hazards of the grain dust being emitted during seed processing has added a new dimension to the seed cleaning and grading operation.

The active handling and treating of seed grains generate dusts which can potentially pollute the work environment in the seed-cleaning plants. Dust is generated each time the grain is handled. The truck unloading station is the largest single source of dust (Sherman, 1973) but, however, dust is also generated at each transfer point such as bucket elevators, belt conveyors, screw conveyors, and bagging operations. Loading and unloading of bins create clouds of fine dust in the environment. During the cleaning process, dockage is removed from the grain and a considerable amount of fine dust is emitted.

The concentration of dust in the seed-cleaning plant environment will vary depending on the type of operation, field source of seed grains, type of seed, harvesting method, weeds present and chemical residual. For instance, the barley kernel generates a long fibre dust while wheat kernel produces a fine dust. Yoshida and Maybank (1974) observed maximum dust concentrations of 892 mg/m^3 and 81.9 mg/m^3 for barley and wheat respectively from a spout-penthouse while handling grains in elevators. Studies on generation of dust by repetative handling of corn indicate that the amount of dust removed per transfer was 0.088% of the corn mass (Norman et al, 1977). The level of emission will also depend on the type and design of the cleaning and grading machines. These machines may be open to the atmosphere or of the enclosed type having a cross-current or counter-current type of air flow. The finer the dust the more severe the atmospheric pollution problem would be because fine particles remain in suspension for a longer period of time. The respirable mass fraction of dust inside the work areas accounted for 50% and 80% of the total dust for wheat and barley, respectively (Yoshida et al, 1978).

The effect of grain dust on workers subjected to grain dust contaminated conditions is till controversial and is related to a number of contributing factors such as individual resistance and smoking habits. There is now active medical research involving toxic effects of grain dusts. Many individuals experience bronchial or allergic disturbances after exposure to feed and grain processing dust. Operators exposed to grain dust may develop acute symptoms of flu-like high-temperature condition that lasts for a week or more accompanied by shortness of breath, wheezing and coughing. In extreme cases they may suffer from inflamation of eyes, nose, ear and skin. Often, it leads to allergies that become more serious with increased exposure. Grain dust exposure produces discomfort or temporary physiological alteration due to dust accumulation in the bronchial tract prior to the development of chronic disorders. Workers are liable to develop a chronic respiratory condition commonly known as "Farmer's Lung" and is caused by the workers' exposure to spoiled grain dust (Dennis, 1973). A recent report shows that 75% of elevator agents in Manitoba had some respiratory symptoms, chronic cough and dyspnea associated with exposure to grain dust. An estimated 40% of the elevator operators leave the industry because they develop one of the acute conditions. According to Labour Canada (1977), the effects of grain dusts on health is a complicated one, to which definite answers are still being sought.

Extensive property damage and fatal accidents in grain elevator explosions and fires are well known. A few major explosions have occurred in Canadian grain elevators in the past. Grain dusts are capable of forming a mixture of an explosive nature. The major cause of a grain dust explosion is the accumulation of fine, dry dust on processing equipment and pipes which may be ignited by any heat source such as a flame or spark. Clouds of fine dust in the air actually create the greatest hazard. Generally two types of explosions; i.e., primary and secondary explosions occur in grain elevators. Secondary explosions are very severe and occur after the primary shock wave has dispersed dust deposits into the air, creating a massive explosive mixture. In the grain handling industry it is important to prevent any dust accumulation so as to avoid these secondary explosions (Canadian Grain Handling Association, 1979). A somewhat less dramatic but, deleterious effect of grain dust in a plant is the reduction in visibility and a photochemical reaction which produces smog.

Labour Canada and Health and Welfare Canada have adopted a provisional standard which provides an employee exposure to a maximum of 10 mg of total grain dust per m^3 of air averaged over any eight-hour daily period and a 40-hour work week (Labour Canada, 1977). Grain dust is of a complex nature and its characteristics affect the collection efficiency of dust control equipment. Furthermore, little research on dust control equipment specifically for seed cleaning plants has been conducted. However dust emission in seed cleaning plants may be kept within the proposed limits by using different available collectors such as cyclone separators, fabric filters, wet type collectors and electrostatic precipitators. Wet collectors and electrostatic precipitators are expensive and thus are not usually adopted on the other hand, cyclone separators are cheaper but have lower collection efficiencies. The use of fabric filters and/or a combination of cyclone separators and fabric filters seems feasible. A need now exists to develop, test and introduce dust control to grain handling systems on the farm.

The objectives of research reported in this thesis were as follows:

i) To monitor the dust concentration in various seed cleaning plants in Manitoba.

ii) To test and determine the characteristics of various types of pipes and elbows applicable to ducting in seed cleaning systems.

iii) To test and balance the flow rate and static pressure of a typical dust removal system.

iv) To test the characteristics (efficiency and pressure drop) of a cyclone separator and fabric filters.

v) To test the effectiveness of various types of suction hoods for use on truck dumping hoppers.

vi) Recommend suitable modifications to the typical existing systems to meet the threshold limit value of dust emission.

2. REVIEW OF LITERATURE

2.1 Grain Dust Defined

Grain dust is of a complex nature and its composition depends on a variety of factors such as: field source of the grains, type of seed, harvesting methods, weeds present and chemical residual. Thimsen and Aftan (1968), Sherman (1973) and Prosser (1975) defined grain dust as the particulate matter that becomes air-borne and varies in size from 1 to 100 μm. Martin and Sauer (1975) considered dust particles smaller than 125 μm to be of major concern in meeting air pollution standards. Labour Canada (1977) gave a more broad and detailed definition of grain dust as "dust present in the atmosphere during handling or processing of grains which may contain a mixture of many substances including ground up plant matter, insect parts and other containments which may have accumulated with the grain during the growing, harvesting and subsequent processing or storage periods. Any dust present during the handling or processing and dust generated in other operations is considered as grain dust". According to Martin (1978), grain dust is composed of solid particles that become air-borne during handling of grain. This includes all materials collected by the dust control system.

2.1.1 Dust Levels in the Seed Cleaning Plants

Dust is the most important problem of the work environment in farm seed cleaning plants. Unfortunately, no specific information is available on dust emissions in the work environment of farm seed cleaning plants. However, many researchers measured dust emissions from various operations during grain handling in the grain elevators. It was found that a considerable amount of dust is generated during the unloading of grains from trucks at the receiving hopper at grain elevators. According to Thimsen and Aften (1968), the amount of dust generated while unloading at the receiving hopper was 0.10% of the grain mass. Sherman (1973) noted that the grain unloading station was the largest single source of dust in the grain elevator handling system. In Saskatchewan, Yoshida and Maybank (1974) found that at the receiving hopper, the dust concentration varied from 20-40 mg/m^3 of air handled.

The mass concentration level of dust in the working environment depends on type of grain handled, according to a nationwide inventory of air pollutant emissions, where the dust emission from grain handling is estimated to be 8% of the total emission from industrial processing (Marier et al, 1974). Yoshida and Maybank (1974), in one of their studies on grain dust emission in elevators, found dust concentrations of 892 mg/m^3 and 81.9 mg/m^3 of air handled from the spout-penthouse for barley and wheat, respectively. Getchell et al (1977), while conducting tests on the use of additives for grain dust reduction during handling, observed dust concentrations as high as 2558 mg/m^3 in handling combine-harvested wheat. Studies by Norman et al (1977) on repetative handling of corn indicate that the amount of dust removal per transfer was 0.088% of the corn mass. The handling treatment affected the total amount of dust more than kernel breakage content (Martin and Stephens, 1977).

In order to limit the atmospheric pollution levels in the work environment in the grain industry, Labour Canada together with Health and Welfare Canada has adopted a threshold limit value of 10 mg/m^3 total dust for any eight-hour daily period (Labour Canada, 1977).

2.1.2 Dust Generation in Seed Cleaning

Dust is generated each time the grain is handled. The main sources of dust generation in seed cleaning plants are: grain receiving hoppers, transfer points such as bucket elevators, belt conveyors, screw conveyors, loading and unloading of bins and cleaning and grading machines. Thimsen and Aften (1968), William (1973), Martin and Sauer (1975), Norman et al (1977), and Yoshida et al (1978) discussed various sources of dust generation in the grain industry especially in grain elevators. They reported that the major dust emission problems in grain elevators were from the receiving hoppers, transfer points, loading and unloading of bins, and loading of cars.

2.1.3 Hygienic Significance of Grain Dusts

Significance of dust in respiratory disorders of humans has long been recognized. For instance, many aspects of industrial dust have been studied quite extensively. The effect of grain dust on workers' health, however, is still controversial as it is related to many contributing factors such as composition of grain dust, particle size distribution, individual resistance and smoking habits. Many researchers in the past have reported the allergic disturbances and discomforts caused by the grain dusts. According to Andersen (1966), air-borne dust is a hazard to health with respect to respiratory disorder, only to the extent that it is deposited in the respiratory system.

Henry and Zoerb (1967) reported that irritation to nose, throat and lungs results from breathing of an excess quantity of dust. This may cause inflamation of membranes which become vulnerable to infection and often lead to allergies that become more serious with increased exposure. Another common effect of grain dust exposure is irritation of the skin, especially around the wrists and back of the neck where clothing rubs the skin. This condition is known as "dermatitis" and may cause serious skin infections in extreme conditions.

A recent report shows that 75% of the elevator agents in Manitoba had some respiratory symptoms, chronic cough and sputum and dyspnea (shortness of breath) associated with grain dust exposure (Tse et al, 1973). Dennis (1973) described health problems from inhaling grain dust which usually resulted from handling moldy or heating grains. Workers are liable to develop a chronic respiratory condition commonly known as 'Farmers' Lung'. Martin and Sauer (1975) found that mold spores were concentrated in dust more than in grain, and the higher concentrations were in the dust that escaped from dust control cyclones into the atmosphere. Synnoniums (1976) studied the microbiological characteristics of grain dusts and analysed the concentrations of micro-organisms in the dust which may contribute to health hazards. Norman et al (1977) reported that the health effects of fungi and spores in the grain dust are still under investigation.

Prosser (1975) indicated that a health hazard to the lungs exists where particle size is between 0.5 μm and 6 μm but the pollen and other materials can give rise to allergic complaints in the range of 20 μm to 60 μm. The Andersen sampler's instruction manual indicates that dust particles between 3.3 μm to 7.0 μm are retained in the trachea and primary bronchi, 2 μm to 3.3 μm in the secondary bronchi, 1.1 μm to 2 μm in the terminal bronchi and less than 1.1 μm in the alveoli. Thus, particles below 7 μm in diameter are more dangerous to the health.

Khane (1977) reported that for those who work in grain elevators, the chances of developing chronic respiratory disorder are about two times as great as those of the general population. They are also liable to develop a chronic respiratory condition with symptoms of chronic cough and phlegm, obstruction of airways, emphysema and chronic bronchitis. According to Labour Canada (1977), an X-ray program for personnel having 15 or more years of employment in grain elevators was commenced in 1973 and their examination indicated that 19% of workers showed a higher than normal incidence of increased lung markings, a condition not necessarily associated with a person's occupation. A correlation between grain dust concentration and prevalence of health effects has been found by most investigators who have studied the effect of worker's exposure to grain dust. In fact, the effects of grain dust is a complicated one to which definite answers are still being sought.

Atliemo et al (1978) reported similar health effects of grain dust as reported by Dennis (1973). According to them the disease resulting from hypersensitivity appears to have caused most of the damage. Symptoms start with congestion in the throat and proceed to coughing accompanied by tightness in the chest. In some serious cases workers often awake breathless and wheezing with bursts of coughing.

2.1.4 Grain Dust Explosions

In the past forty years in Canada, there have been five major explosions at terminal elevators. Since 1925, grain elevators in the U.S.A. also assume the number one position in terms of deaths resulting from dust explosions. Thimsen and Aften (1968), Stevens and Schoeff

(1973), Canadian Grain Handling Association (1979) and many others explained that the major cause of grain dust explosion is the accumulation of fine dry dust on processing equipment and pipes to explosive concentrations, which may be ignited by any heat source such as a flame or a spark.

The two general types of explosions that occur in the grain industry are primary explosions and secondary explosions. Secondary explosions are more disasterous. Among the plant equipment items, the bucket elevator legs present the greatest hazard. It is therefore of paramount importance that secondary explosions should be prevented and limited (Stevens and Schoeff, 1973; CGHA, 1979). The most significant fuel for secondary explosions is the accumulation of dust in layers. Secondary explosions occur after the primary shock wave has lifted (primary explosion) and mixed heavy dust deposits with air creating a massive explosive mixture.

Fine dust particles that are suspended in the air will form a mixture of a highly explosive nature (Getchell et al, 1977). Gibson et al (1977) estimated the concentration of aerosol particles to be a function of temperature and dust layer thickness. This approach enabled him to study the response of ionization and combustible gas detectors to invisible aerosols and gases evolved from heated grain dusts at a temperature below the ignition temperature.

Stevens and Schoeff (1973), Honey and Mcquity (1976) and CGHA (1979) described that ignition temperature of grain dust, whether in the form of an aerosol or an aerogel, could be attained from heat sources such as exhaust systems, bearings or severely slipping V-belts. Static electricity was shown not to be normally present.

CGHA (1979) reported the explosive limits of suspended grain dusts. While upper limits are not always definite, there is a general agreement that the lower explosibility limit is 40000 to 55000 mg/m^3 with the lowest reported figure being 20000 mg/m^3. In addition to these limits, some grain dust properties such as minimum ignition temperature, minimum ignition energy, ignition sensitivity, explosive sensitivity, and explosibility index were also reported by CHGA (1979).

2.2 Dust Sampling

Two commonly used methods that are employed for determining the total dust concentrations in the work environment are the sampling jar and the high volume air sampler. Yoshida and Maybank (1974) used sampling jars to measure dust falls in the vicinity of grain elevators in the western Canadian prairies. However, the high volume air sampler is a more accurate and quicker method for sampling dusts. This sampler was used extensively in the past. For example, Annis (1972), Morrow et al (1972), Martin and Sauer (1975), Avant et al (1976), Kirk et al (1977), Norman et al (1977) and Parnell et al (1977) used the high volume air sampler for determining total mass concentrations of grain dusts in the working environment.

Environment Canada (1974) and Norman et al (1977) described the procedure for the determination of mass concentrations of suspended particles in the environment. Dust concentration levels are determined by obtaining the net mass of dust taken from the fibreglass filter after sampling with a high volume sampler. This is achieved by pre- and postweighing the filter in a constant environment. After determining the mass of dust on the filter, the concentration level can be determined for a given air flow rate and elapsed time of sampling.

The respirable dust monitor (GCA, 1976) is an advanced instrument designed for on-the-spot measurement of mass concentrations of respirable fractions or total mass loadings of dust particles in the air.

In order to evaluate the performance of dust control equipment and to characterize the dust precisely, the determination of the particle size distribution is most important. Jarett and Heywood (1954) conducted comparative tests on different methods of particle size analysis. They also recommend suitable methods for size analysis with proposed degree of error and range of particle size to which various methods are applicable. Methods of particle size analysis are varied to suit the nature of the particles, the needs of analyst or the sample collection method. Sargent (1971) classified various methods of particle size analysis as micro-classifier, elutriator, electronic counting by Coulter counter, cascade impactor, elutriatun, sieve shaker, microscopic examination and electro-static precipitators.

Avant et al (1976), Matlock and Parnell (1976) and Martin (1978) used a model TA Coulter counter to determine particle size distribution of dust emitted in the working environment of a cotton seed oil mill. The Andersen sampler simulates the human respiratory system. The human respiratory passage from the mouth to the lungs get progressively smaller. As a result, particles in the air we breathe penetrate to the lungs due to their size, shape, density and velocity imparted to each particle.

Inertial cascade impactors have been used extensively in both ambient air and emission source investigations. This instrument is used to obtain information on mass and chemical composition V_s aerodynamic particle size (Ondov et al, 1978).

Regardless of the sampling system used, it is essential that attempts be made to sample isokinetically for accurate determination of dust loadings. This means maintaining inlet velocity equal to the duct velocity at the sampler inlet point (Sargent, 1971; Morrow et al, 1972; Annis, 1972).

2.3 Grain Dust Properties

Grain dust may contain a mixture of many substances. The relative proportion of each sustance would depend upon type of grain, harvesting and storage methods and cleaning operations. Shannon et al (1973), Martin and Sauer (1975), Labour Canada (1977), and Yoshida and Maybank (1978) reported that the grain dust is composed of approximately 70% organic material and about 17% silicon dioxide. Specific materials in the dust include particles of grain kernels, spores from smuts and molds, insect debris, pollen, herbicides, fine fibrous dust or trichome particles and field dust. Corn dust had higher concentrations of fine, easily air-borne particles and mold spores than wheat.

Thimsen and Aften (1968), Sherman (1973) and Prosser (1975) considered grain dust as particulate matter with particle size between 1 µm and 100 µm. The geometric median diameters and geometric standard deviations are used to report the firneness of feed materials (ASAE, 1978). Avant et al (1976), Norman et al (1977), Martin (1978), Atiemo et al (1978) and Yoshida and Maybank (1978) analysed particle size distribution of grain dusts emitted during handling of different kinds of grains and found that the size distribution function represented a log-normal distribution. Avant et al (1976) reported that a fairly broad distribution centered near 12.5 μm for sorghum while Yoshida and Maybank (1978) found mass median diameters less than 10 μm for wheat and barley dusts.

Yoshida and Maybank (1978) determined the ratio of volume shape factor to the projected area diameter resistance shape factor with dust samples collected in Saskatchewan. They also reported that the respirable mass fraction of dust inside work areas was about 50% and 80% of the total dust for wheat and barley respectively.

2.4 Air Flow Measurement in Ducts

Flow measurements in the dust control system is necessary to monitor the air flow at each hood in order to balance the system properly and to maintain the required dust conveying velocities in various ducts. There are several methods for the measurement of air velocity and air flow rates in the ducts. Air flows are usually measured with differential head devices such as orifice plate, nozzle, venturimeter, and Pitot-static tubes. Spink (1958), Fan Engineering (1970), Industrial Ventilation (1974), and Strauss (1975) described these methods in detail with their merits and demerits. Although these methods are reliable for determining air velocity and are accepted in engineering practices, only the Pitot-static tube is suitable for field work (Industrial Ventilation 1974). Svistovski (1978) studied three different methods of air flow measurements

(Pitot-static tube, wind tunnel and velometer) for testing the local exhaust system for grain dust removal. He recommended that the Pitot-static tube and manometer may be used accurately for measuring static pressure, total pressure and velocity pressure.

The Pitot-static tube is one of the most accurate means of measuring air velocity in exhaust ducts. Industrial Ventilation (1974) indicated that the use of the Pitot tube in the field is limited to minimum velocities of 3 to 4 m/s with a percent error of 6-15% whereas, at 20.32 m/s, the error is only 0.25%. It is, however, a universally accepted method for measuring air velocity. The standard Pitot-static tube needs no calibration, if used with a calibrated manometer (Schuman, 1976). The Pitot-static tube consists of two concentric tubes, one of which measures the total or impact pressure while the other measures the static pressure. When the Pitot tube is connected to a U-tube manometer the velocity pressure is measured. This pressure is then used to compute the velocity of the air stream using the equation reported in Industrial Ventilation (1974). This modified equation is shown in equation 2.1:

$$V = 4.41 \sqrt{\frac{VP}{\rho}} \qquad (2.1)$$

where: V = velocity of air, m/s

VP = velocity head, mm water column (W.C.)

ρ = density of air, Kg/m^3.

The detailed procedure for measurement of air velocity and air flow rate with a Pitot-state tube has been described in Air Moving and Conditioning Association (1962), Simon et al (1973), Dimperio (1973), Industrial Ventilation (1974), Dorman (1974), Schuman (1976) and Field (1976).

The volume of air handled by an exhaust system is sometimes approximated by various types of field instruments such as rotary vane anemometer, swinging vane anemometer, heated wire anemometer, heated thermocouple anemometer, smoke tubes, and tracer-gas dilution. Industrial Ventilation (1974) discussed their advantages, disadvantages and suitability to certain specific processes.

2.4.1 Pitot-Static Tube Traversing

A single reading from an impact, static or Pitot tube will not be accurate because the velocity and pressure in a duct varies from point-to-point at any corss-section. The magnitude of such a reading will depend on its location and velocity profile in the duct. The velocity profile is a function of Reynolds number and roughness of the ducts. Fan Engineering (1970) proposed the following equation (equation 2.2) to approximate the mean velocity in a duct for a Reynolds number greater than 5000.

$$V_m = V_c \frac{1}{1 + 1.439\sqrt{f}} \qquad (2.2)$$

where: V_m = mean velocity, m/s

V_c = velocity corresponding to the velocity pressure at the center of the duct, m/s

f = friction factor, dimensionless.

It is therefore, necessary to obtain a more accurate average velocity by measuring the velocity pressure at numerous locations at the duct cross-section. AMCA (1962), Fan Engineering (1970), Simon et al (1973), Dorman (1974), Industrial Ventilation (1974), and Schuman (1976) suggested that, for more accurate determination, velocity pressure measurements should be made at points in a number of equal areas in the cross-section. With circular ducts, the approved method is to make two traverses across the diameter of the duct at right angle to each other. The cross-section is divided into a number of equal area concentric rings and the velocity pressure is measured at four points in each ring. For round ducts, 150 mm or smaller, at least 6 traverse points should be used and for ducts greater than 150 mm at least 10 traverse points should be employed. Twenty traverse points will increase the precision of air flow measurements. With rectangular or square ducts, the procedure is to divide the cross-section into a number of equal rectangular areas and measure the velocity pressure at the center of each area.

Whenever possible, the traverse should be made 7.5 times the duct diameter or more downstream from any major air disturbance such as an elbow, hood, branch entry, etc. Ducts smaller than 300 mm will require a Pitot-static tube smaller than the standard 7.94 mm O.D. (Industrial Ventilation, 1974).

2.4.2 Frictional Losses in Straight Galvanized Sheet Metal Pipes and Plastic Flexible Pipes

For satisfactory design of ductwork for dust control systems and its balancing, flow resistance must be known. Knowledge of flow resistance of the system provides a suitable guideline for selecting the fan and motor. Wright et al (1945) expressed the resistance to flow of a fluid in a closed conduit in nondimensional form as shown in equation 2.3:

$$h = f \frac{L}{D} \frac{V^2}{2g} \tag{2.3}$$

where: h = head loss due to friction, m

L = length of conduit, m

D = conduit inside diameter, m

V = fluid velocity, m/s

g = acceleration due to gravity, m/s^2

f = friction coefficient which depends on roughness of pipe, dimensionless.

Fan Engineering (1970) reported that the resistance to flow through any duct element or fitting may be considered to be the sum of the frictional loss and shock loss. Frictional losses are the losses in straight pipes and vary directly with the length of the pipe. Shock losses occur whenever there is an abrupt change in conduit and vary as the square of the velocity. Fox and Mcdonald (1973) stated that the head loss for flow in a constant cross-sectional area duct depends only on the details of the flow through the duct. It represents the amount of energy converted by frictional effects from mechanical to thermal energy.

Houghten et al (1939) estimated frictional resistance to flow of air in ducts and fittings and presented the relationship between the friction factor and the Reynolds number for round ducts. He also reported that the relation between the pressure loss and volume of air flowing through three different sizes of round ducts reported by earlier investigators showed a close agreement. Moody et al (1944) developed charts to estimate friction factors for clean new pipes and for closed conduits running full with steady flow. Wright et al (1945) surveyed available literature and using newest theoretical developments, produced a new frictional loss chart to provide basic data on the resistance of air flow in sheet metal ducts for the purpose of checking friction charts. Huckbsher (1948) estimated friction equivalents of round, square and rectangular ducts and found that, for most practical purposes, rectangular ducts of aspect ratios (depth to its width) not exceeding 8:1 had the same static friction pressure loss for equal length and mean velocities of flow as a circular duct of the same hydraulic diameter. Svistovski (1978) determined frictional losses in various types of circular ducts such as smooth steel pipes, corrugated plastic pipes and sewn vinyl ribbon pipes. Smooth steel pipes had the least frictional loss whereas, sewn vinyl ribbon pipe had the highest frictional resistance.

2.4.3 Frictional Losses in Plastic Flexible Pipe Bends

Flexible pipes are practical for dust control system as they can be adapted to alternations in the layout of the system. Generally, however, the flexible pipes have slack which create bends and increase the resistance of the piping system. A large percentage of the total pressure loss is caused by elbows and thus should be eliminated, if possible, in order to improve efficiency. Stuart et al (1942) considered pressure loss in an elbow by three phenomena: (1) the friction of air particles against the duct wall; (2) the loss due to turbulent flow; and, (3) the loss due to change in direction of flow. The first two phenomena contribute the loss expressed in the straight duct and also occur in an elbow because it has a definite length. The third loss is unique and was termed as an additional loss. This loss may be determined by finding the excess pressure drop for a section of straight pipe having an equal centerline length. Fan Engineering (1970) and Industrial Ventilation (1974) reported the simplest way to express the resistance of branch entries and elbows in equivalent length of straight duct of the same diameter that will have the same pressure loss at the fitting.

No specific information is available on frictional losses of flexible pipe bends, however, many investigators produced frictional loss data for different types of metallic elbows and bends. Madison et al (1936) determined pressure losses in rectangular elbows. They considered the effect of the size and shape of the elbow as well as surface friction for determining the pressure loss which previous researchers overlooked. In their investigation all the pressure losses were measured as the percentage of pressure corresponding to the mean velocity in the elbow. Stuart et al (1942) measured loss in metallic elbows and observed that the pressure loss caused by elbows was proportional to the 1.8 power of the velocity and that this loss at any velocity, may be expressed by a unique equivalent length of straight duct. They found, in another study involving the reduction of pressure loss in elbows, that the loss caused by an elbow may be reduced by increasing the radius until a minimum value is reached. This value occurs when an elbow has a radius ratio of 2.6. The radius ratio of an elbow is its centerline radius divided by its width in the plane of the bend.

An easy elbow had less loss than a mitre elbow equipped with the best vanes. According to Locklin (1950) he surveyed the available information and analysed data on energy losses in 90° duct elbows. He presented the results in a useful manner which can be used for practical engineering, with particular application to duct design. The equivalent resistance of straight pipe for elbows and branch entries for different diameters have been reported in Fan Engineering (1970) and Industrial Ventilation (1974).

2.5 Hoods and Dust Pick-up Systems

Control of emission requires proper design of the dust pick-up system to adequately collect the containments at the emission source. It is essential to catch the dust as near to the source as possible. If it is feasible, a hood or cover should be constructed to enclose the maximum possible area around the source. If the enclosure is not practicable, the exterior hood should be located near the source and shaped properly to control the blowing of the dust from the source. Fan Engineering (1970), Simon et al (1973), and Industrial Ventilation (1974) classified different hoods into three general types (enclosure hoods, receiving hoods and exterior hoods) on the basis of their suitability to various emission sources.

Dalla Valle (1932) studied the nature of air flow at suction hoods in an attempt to develop a fundamental relationship between shape size and type of hood, the velocity in front of the hood and the volume flow rate of air through the hood. Brandt et al (1947) conducted similar studies on suction openings to verify the relationship given by Dalla Valle (1932). Metzler (1960) developed nomographs for air flow rates in different types of exhaust hoods such as plain pipe, tapered hood, sharp edge, grinder wheel, half cylinder and quarter cylinder hoods.

The shape and size of hood, its position relative to the point of emission source and the nature and quantity of containment affects the quantity of air required to capture and convey the containments. Fan Engineering (1970), Simon et al (1973) and Industrial Ventilation (1974) reported the exhaust requirements for various operations and described the hood design procedure. Prosser (1975) stated that capture velocity can be related to particle size, the smaller the particle size the lower the capture velocity. The capture velocities may vary from 0.25 to 5.1 m/s. Overmyer (1976) recommended that the minimum face velocities should be from 1.27 to 2.54 m/s for normal process dusts and 10.16 m/s for design of slot hoods.

The hood should be especially designed for each process. Battista (1947) designed an effective semilateral tank ventilation hood for controlling containments. Sherman (1973) developed a swing type hood for shallow grain receiving pits equipped with a grating. Blossom (1976) used conical hoods for local oil mist control. Design details of different hoods applicable to different processes are given in Industrial Ventilation (1974).

2.6 Duct Sizing and Balancing of the System

Ducts that carry dust particles from the exhaust hoods must be properly sized to prevent dust settlements. Thimsen and Aften (1968), ASHRAE (1969), Fan Engineering (1970), Sherman (1973), Industrial Ventilation (1974) and Overmyer (1976) stated that conveying velocities of 15 to 20.32 m/s in the ducts are acceptable to keep the dust in suspension. Lower air velocities would cause the dust to settle in the ducts while higher velocities result in more static pressure losses in the system. Various methods of duct sizing are discussed in ASHRAE (1969), Fan Engineering (1970), and Industrial Ventilation (1974). These methods have different design levels of accuracy and complexity and should be selected to suit the application.

Vincent (1973), Field (1976) and Schuman (1976) emphasized that static pressure balancing is most important to insure adequate performance from an air exhaust system having multiple branches. The resistance of each branch must be adjusted so that static pressure balance, which exists at the junction of two branches, will give the desired air flow in each branch. Pressure losses in a duct system are carefully calculated for selection of a fan. Usually, the pressure drop for the branch of greatest resistance is calculated in detail. This resistance together with the total volume flow of the system, establishes the power requirements of the system.

Vincent (1973) and Industrial Ventilation (1974) described the methods and procedure for balancing an exhaust system. In general, two methods (static pressure balance method and blast gate adjustment method) were most commonly used. The first method is less flexible and more tedius to calculate but does not require frequent checking and adjustments. The second method is more flexible and simple but needs frequent checking and adjustments of dampers for proper air flow rates. Svistovski (1978) used the static pressure balance method for balancing a local exhaust vacuum system adapted for grain dust removal.

2.7 Performance of Dust Collectors

Dust control equipment may be classified into several general types such as filters, electrical precipitators, cyclones, mechanical collectors and scrubbers. The performance characteristics of this equipment are generally expressed according to collection efficiencies and pressure drops (Stern, 1968). Perhaps the most logical performance parameter is the one that relates collection efficiency to pressure drop through the collector (Adam, 1971):

$$r = \frac{\ln\ (1/1-n)}{\Delta Pi} \tag{2.4}$$

where: r = performance of collector

n = mass ratio of dust collected to dust entering the equipment (efficiency)

ΔPi = pressure drop through the dust collector, mm W.C.

Efficiency and pressure drop of a collector depends upon collector design, dust properties and loadings and properties of gas carrying dust. In general, a dust collector handles dust between 0.25 and 3% by mass of solid material being processed. According to Sargent (1971), to make a preliminary selection of suitable gas cleaning equipment, only four basic data are required; dust loading, particle size, gas flow, and allowable emission rates.

For testing the collection efficiency of the dust collector, American National Standard Institute (1972) developed a standard test method and recommended that equation 2.5 be used for expressing the overall mass collection efficiency:

$$E = 100\text{-}100\ \frac{DS}{US}$$

where: E = efficiency of the system, %

US = concentration of the aerosol in unfiltered air; that is, the sample removed from the duct upstream of the device, mg/m^3

DS = concentration of the aerosol in the filtered air; that is, the sample removed from the duct downstream of the device, mg/m^3.

Stern (1968) indicated that control techniques that result in progressively increasing pressure loss with time, for example in fabric filters accumulation of dust cake during the filtering cycle results in increased resistance to flow. With a resulting reduction in fan output. The power consumption may be predicted and if possible minimized by better choice of collector parameters (Strauss, 1975).

2.7.1 Cyclone Separator

The centrifugal collector in its simple form is the cyclone separator. Here the rate of dust precipitation is increased over that produced by gravity by applying radial acceleration from centrifugal motion (Silverman, 1953). It is a simple, inexpensive unit and has no moving parts. Cyclone separators have been in use for dust collection since 1885, but quantative design papers did not appear until the period 1951 to 1963. This remained the most useful dry collector for dust particles above 10 μm diameter (Martin, 1972). Silverman (1953), Caplan (1968), Stern (1968), Prosser (1975) and Doerchlag and Miczak (1977) explained the working principle of the cyclone separator. Centrifugal force is the primary mechanism of particle collection in the cyclone. The dust-laden air enters tangentially at the top, swirls around inside the cyclone separator and the clean air is discharged from the centre at the top. Due to centrifugal forces and reduced velocity the dust particles are directed toward the wall of the cyclone and collected in the inverted conical base. The performance of the cyclone separator is expressed in terms of its collection efficiency and pressure drop.

The collection efficiency of the cyclone separator depends on the cyclone physical parameters, particle size, dust loading and air flow rates. In general, the efficiency will increase with an increase in dust particle size or density, gas inlet velocity, cyclone body or cone length, and ratio of body diameter to gas outlet diameter. Conversely, efficiency will decrease with increase in gas viscosity or density, cyclone diameter, gas outlet diameter and inlet width or inlet area (Caplan, 1968; Shannon, 1973; Koch and Licht, 1977). When the cyclone diameter is reduced the efficiency is increased but this increase is at the cost of increased resistance (Silverman, 1953; Sargent, 1971; Sherman, 1973). The overall efficiency of the cyclone separator can be measured according to the ANSI (1972) standard. Many investigators used fractional efficiency (the efficiency with which a specified particle size range are collected) to indicate the cyclone efficiency. However, in most practical applications of cyclone separators the overall mass percentage efficiency for the dust in question will be the main consideration (Walton, 1974).

Wesley et al (1970) evaluated a cyclone for collecting cotton dusts and found that air volume, input feed rate and trash size statistically affected dust concentrations. Avant et al (1976) tested nine different types of cyclone separators with grain dust and compared his results with those obtained from theoretical models. They indicated that more empirical work is needed to predict cyclone efficiencies accurately. Doerchlag and Miczak (1977) developed the basis of comparision of data on cyclones for better selection of a cyclone dust collector. Yoshida et al (1978) proposed a modified formula for predicting collection efficiency (equation 2.6).

$$E = 1 - E_{xp}\ (-2\ (CP).(SP)) \qquad (2.6)$$

where: E = collection efficiency of cyclone, %

CP = modified geometry coefficient

SP = modified inertia parameter.

Values of CP and SP were empirically determined. They further indicated that cyclones can achieve collection efficiency of up to 97.5% of the total mass through improvement in operation and optimization of design. Cohn and Stack (1979) reported that cyclones can be used for removing particles greater than 5 μm in resource recovery.

Various investigators attempted to relate the design parameter of a cyclone to the pressure loss both theoretically and experimentally. Stairmand (1949) provided a mathematical model for calculating the loss from a consideration of the flow in a cyclone. The pressure drop in a cyclone is directly proportional to the dynamic pressure and is generally expressed in terms of inlet velocity heads. Dey et al, 1973) noted that Lapple (1963) developed a relationship between cyclone parameter and pressure drop. The pressure loss or gain in a cyclone depends on: the entrance pipe, expansion or compression at the entry, wall friction, kinetic energy loss in the cyclone, entrance to exit pipe and static head loss between inlet and exit pipes (Strauss, 1975). Avant et al (1976) meausred the pressure drops in 9 different types of cyclones. The loss was determined by subtracting the inlet and outlet static pressures of the cyclone. The pressure in the cyclones tended to be inversely proportional to the cyclone height. Koch and Licht (1977) presented a graphical method for determining the optimum inlet velocity and cyclone diameter for a desired separation. They reported that high inlet velocities not only cause re-entrainment but also excessive pressure drops. Browne and Strauss (1978) successfully designed a deswirler to recover an appreciable part of flow energy by placing it downstream of the cyclone in the outlet duct without reducing cyclone efficiency.

2.7.2 Fabric Filters

Fabric filters operate as high efficiency collectors of particulate matter, dust or fumes from the air or gas. These have been used to control essentially every kind of emission source involving grain handling and several grain processing sources (Shannon et al, 1973). The filter theory has been developed in depth and it is understood that various collection mechanisms such as direct interception, impingement, diffusion, electrostatics, gravity and centrifugal force are present (Davies, 1973; Shannon, 1973; Benson and Smith, 1976; Talty, 1978). According to Mckenna (1974), the removal of particles greater than 1 μm are considered to be controlled by impaction and interception while, in the sub-micrometer region diffusion and electrostatic attraction are considered the important factors. Dennis (1974) emphasized that in addition to collection efficiency, pressure drop, air to cloth ratio, the fabric aereal density as well as filter drag, free area and residual dust loadings must also be considered in predicting filter behaviour.

Shannon (1973), Sherman (1973), Dennis (1974), Reigel (1974), Mckenna et al (1974), Parnell et al (1978) and Cohn and Stack (1979) reported that fabric filters, when properly designed and operated trouble free, are 99.99% efficient for the average field application. Although the bags themselves are very efficient, the deposition of dust in the interstices of the fabric generally enhance their collection efficiency. In other words, dirty bags are much better than clean bags up to a point (Reigel, 1974). Dennis (1974) mentioned that the dust properties, operating parameters, filter cleaning methods, and their critical interdependence would significantly influence the collection characteristics of fabric filters. For optimization of filtration performance, Bakke (1974) considered the maximum filter rate at minimum pressure drop, collector size, fan horsepower, minimum outlet dust loading or maximum collection efficiency, and maximum bag life as main parameters.

Air to cloth ratio sometimes referred to as filter velocity is one of the key design parameters for fabric filter design (Shannon, 1973; and Sherman, 1973). Reigel (1974) defined air to cloth ratio as the ratio of actual volumetric gas flow rate to net on-line cloth areas (equation 2.7):

$$AC = \frac{Q}{A} \tag{2.7}$$

where: AC = air to cloth ratio, mm/s

Q = volumetric gas flow, mm^3

A = net area, mm^2.

AC is equal to the superficial face velocity of gas as it passes through the cloth in mm/s. Mckenna et al (1974) studied the effect of air to cloth ratio and found that increasing the air to cloth ratio from 3:1 to 6:1 increased the outlet dust loadings. Benson and Smith (1976) stated that air to cloth ratio is one of the important characteristics of fabric filters. A high filter velocity causes excess pressure drop, excessive wear of the bags, blending (clogging) of the bags and reduced collector efficiency. Low filter ratios result in an overdesigned baghouse and high cost.

The resistance of the fabric filters will vary directly with air flow and will depend on construction material, air to cloth ratio, dust feed rate, and method of cleaning the bags. Many workers and investigators have developed theoretical equations to predict the pressure drop across the filter and filter cakes but these are not adequate for designing a system (Shannon, 1973). Simon (1973) reported the following equation 2.8 for predicting the total pressure drop through the filter bags.

$$R = K_o V_f + K_d V_f W \qquad (2.8)$$

where: R = total pressure drop through filter cloth, mm W.C.

K_o = resistance factor, mm/m/s

V_f = filtering velocity, m/s

K_d = resistance coefficient, mm W.C./m/s/g of dust/m^2

W = dust loading, g/m^2.

Reigel (1974) reported the concept of drag as an important characteristic of filter (equation 2.9):

$$S = \frac{P}{V} \tag{2.9}$$

where: S = filter drag, mm W.C./m/s

P = pressure drop across filter, mm W.C.

V = superficial face velocity, m/s.

Drag is a measure of resistance to air flow of the pollution control devices and is directly proportional to fan power. Parnell et al (1978) developed an inexpensive bag filter system and measured pressure drops by using pressure taps. They observed that pressure drop increased from 12.5 to 100 mm W.C. when inlet dust loadings were increased from 4.68 to 27.78 g/m^3.

3. EXPERIMENTAL METHODS AND INSTRUMENTATION

3.1 Location

The research reported in this thesis was conducted in the summer of 1978 and 1979 in the Department of Agricultural Engineering, University of Manitoba. On-farm dust monitoring was conducted at various farm seed cleaning plants situated near Winnipeg, Manitoba.

3.2 Dust Monitoring in Various Farm Seed Cleaning Plants

Dust emission levels were measured in 11 farm seed cleaning plants situated near Winnipeg, Manitoba in order to assess the polution problems in the work environments. A high volume air sampler was used to monitor total dust emission in the plant's environment. The sampler was calibrated in the laboratory according to the instruction manual. A calibration chart (Figure 3.1) was drawn to correct the observed air flow rates. Fibreglass filters were conditioned for 24 hours in a desicator prior to use. The tare mass of the filter was obtained with a balance scale and recorded along with filter identification number. These filters were then used at the plants for collecting dust samples. For sampling, the high volume sampler was located near the largest emission source in the work environment. The filer was installed in the filter holder and the sampler was operated for one hour. Rotameter readings were taken at the beginning and at the end of the test, to measure the average air flow rate through the filter paper. Temperature of the ambient air in the plant was also recorded. The filters were carefully put in the manila envelopes and transported

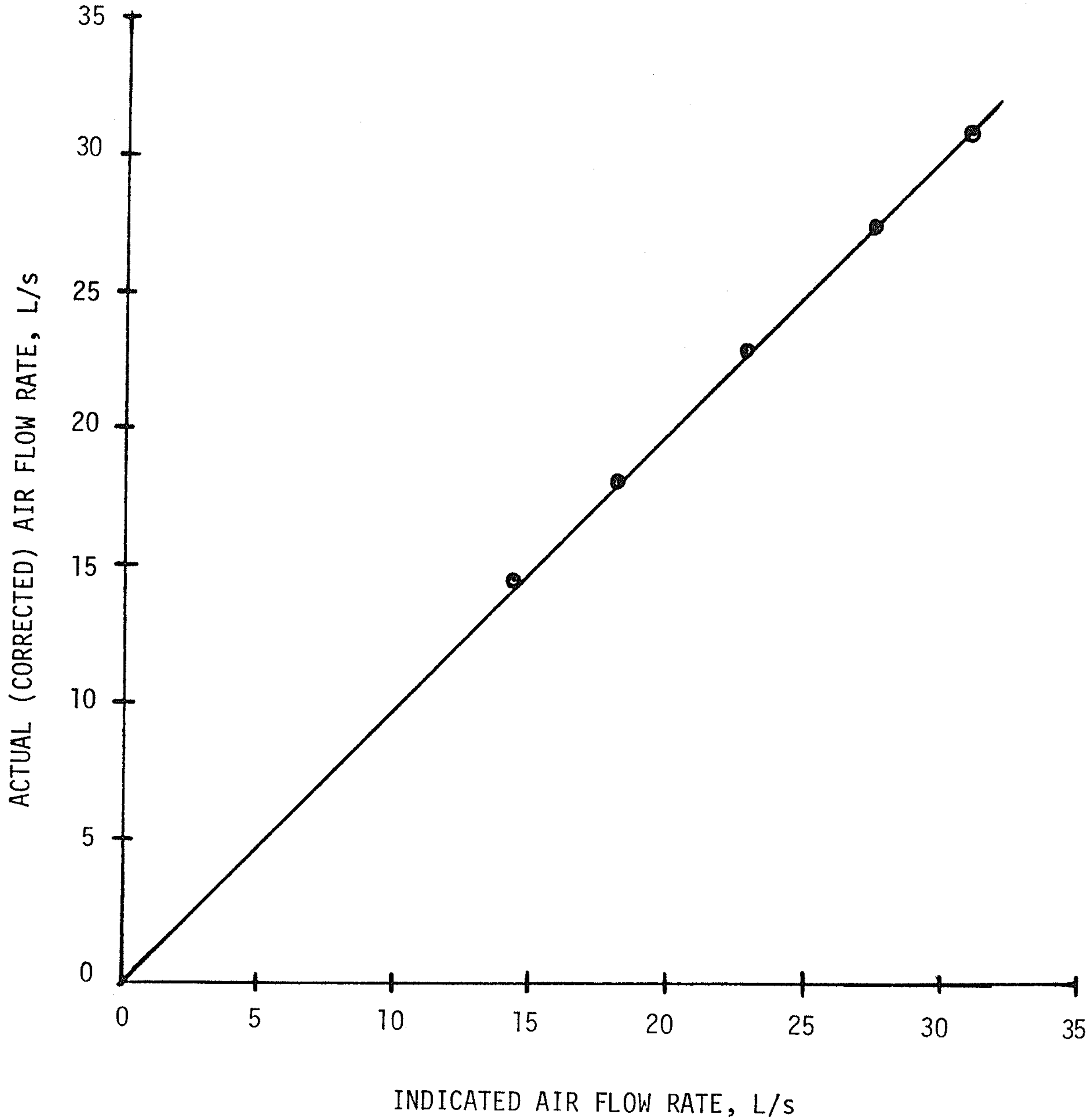


Figure 3.1 Calibration of high volume air sampler

back to the laboratory and allowed to equilibrate for 24 hours in the desicator prior to weighing. The exposed filter mass minus its tare mass yielded the total mass of the dust sample. The total volume of air that passed through the filter was calculated from the average air flow rate and sampling time. The total dust mass divided by the total volume of air that passed through the filter yielded the dust concentration in mg/m^3.

For particle size analysis the Anderson head (Figure 3.2) was used on the high volume air sampler. Preconditioned and preweighed

Figure 3.2 Andersen particle sizing head

filters were installed in the Andersen particle sizing head. The sampler was put near the emission source in the plant environment and the Andersen head was fitted on to the sampler. The sampler unit was started and the air flow rate through the filters was adjusted to

9.44 L/s with the help of a variable output transformer. The unit was allowed to operate for about two hours. The sizing head was then removed from the sampler and transported to the laboratory, where the filters from each stage were removed carefully and put in the desicator for conditioning for 24 hours. The filters were weighed again and the mass of dust on each stage was determined from the gross mass minus the tare mass of each filter. The total mass in mg, divided by the volume of the air drawn through the filter, yielded the dust concentration in mg/m^3 of air for each of five stages of the sizing head.

Using the net dust mass on each stage, the cumulative mass percentages were calculated for each stage. A linear regression representing the particle size distribution with percent cumulative mass as the independent variable and the natural log of particle diameter as the dependent variable was determined for each sample. The mass median diameter and geometric standard deviation were calculated to describe the particle size distribution pattern.

3.3 Air Flow Measurements

A standard Pitot-static tube with an 8 mm tube diameter (specifications given in Appendix I) was used to measure the velocity of air in the pipes and ducts of experimental dust control system. The Pitot-static tube station (Figure 3.3) was located in a 228.6 mm diameter duct, 1714-5 mm downstream of an elbow in the system. Based upon the AMCA (1962) standard test code, 20 traverse points were established for measuring the average velocity in the duct; that is, two series of ten Pitot-static tube traverse points were made at 90-degree to

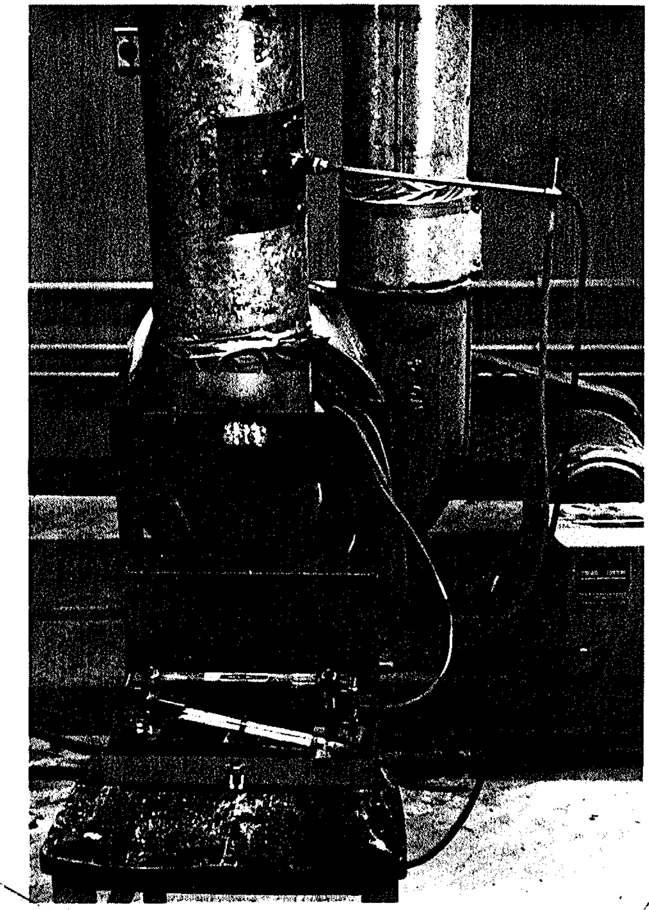


Figure 3.3 Pitot-static tube traverse station.

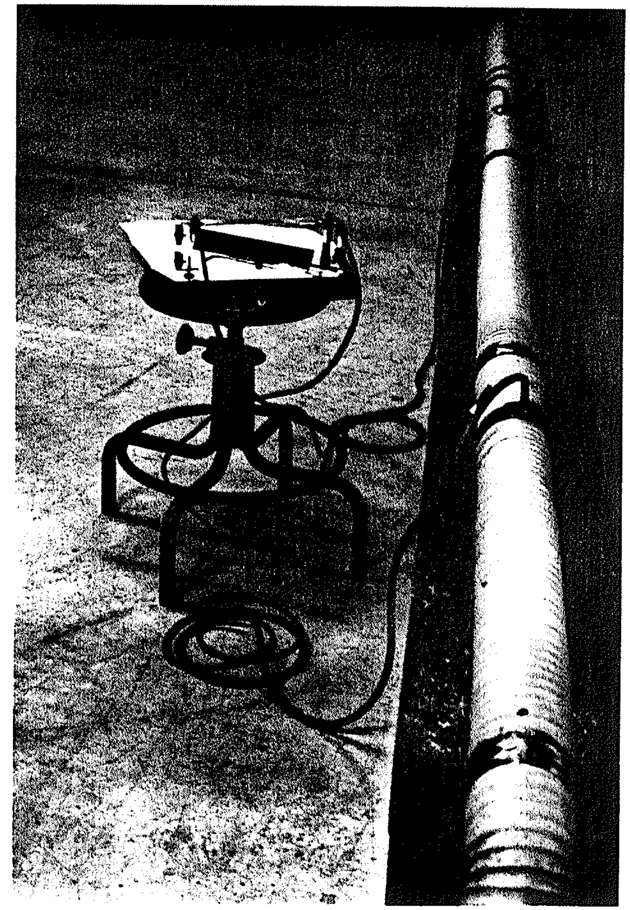


Figure 3.4 Frictional loss measurement in plastic flexible pipe.

one another in the cross-section of the duct. The Pitot-static tube was connected to an inclined manometer and a velocity pressure reading was taken while maintaining the Pitot-static tube perpendicular to the duct and parallel to, but in the opposing direction of, the air flow. The velocity at each traverse point was calculated using equation 2.1. The average air velocity in the duct was obtained by averaging the velocities of all the 20 traverse points. Diameter measurement was used to determine duct area. The air flow rate in the duct was calculated by equation 3.1:

$$Q = V\,A \tag{3.1}$$

where: Q = air flow rate in duct, m^3/s

V = velocity of air, m/s

A = area of cross-section of duct, m^2.

3.4 Pressure Drop Measurements

For the measurement of pressure drops, velocity pressure, static pressure, and total pressure, the inclined tube manometer (Dwyer manufacturing company, U.S.A.) was selected. This manometer had an expanded scale to permit readings within 0.51 mm W.C. and a 0 to 50 mm W.C. range. Coloured oil (specific gravity 0.826) was used as an indicating fluid. For measuring pressure drop greater than 25 mm W.C. a U-tube manometer filled with water was used.

3.5 Frictional Losses Measurement in Galvanized Sheet Metal and Plastic Flexible Pipes

Tests were conducted to measure the frictional losses in 76.2, 101.6, 127, and 203.2 mm diameter sheet metal ducts due to air flow. Approximately 10 m of straight run was used for each test. The pipes were joined together with aluminum painted adhesive tape. The pipes were then connected to the centrifugal fan by means of a 228.6 mm diameter traverse duct. At a sufficient distance downstream from the entrance to the duct, a 2.7 mm I.D. static probe was inserted into the duct. Another pressure probe of the same size was used 3 m downstream from the first probe. These tubes were connected to the manometer and the pressure drop in a 3 m length of straight duct was measured in mm W.C. for different air flow rates. Air flow rates were also measured with a Pitot-static tube. The air flow rate was varied by changing the fan speed. This was accomplished by using a variable-speed pulley.

Frictional losses in the 76.2, 101.6, 152.4 and 203.2 mm diameter plastic flexible pipes were also measured. The flexible pipe was fixed on wooden planks (Figure 3.4) by adhesive tape and then connected to the fan through the traverse duct. Two static probes were inserted 3 m apart and at a sufficient distance downstream of the duct entrance. The frictional loss was measured by the manometer connected to these probes, in mm W.C. as a function of air flow rate. The measured frictional loss was expressed in Pa/m duct length.

3.6 Measurement of Frictional Losses in Plastic Flexible Pipe Bends

Frictional losses in 45° and 90° bends in different diameter plastic pipes were measured. A 90° bend (Figure 3.5) was made and held in position by adhesive tape. One static probe was inserted into the pipe upstream and close to the bend. Another probe was inserted about 3 m downstream of the bend as, according to the AMCA (1962), the pressure taps should be 7.5 times the pipe diameter downstream from bends for accurate measurements.

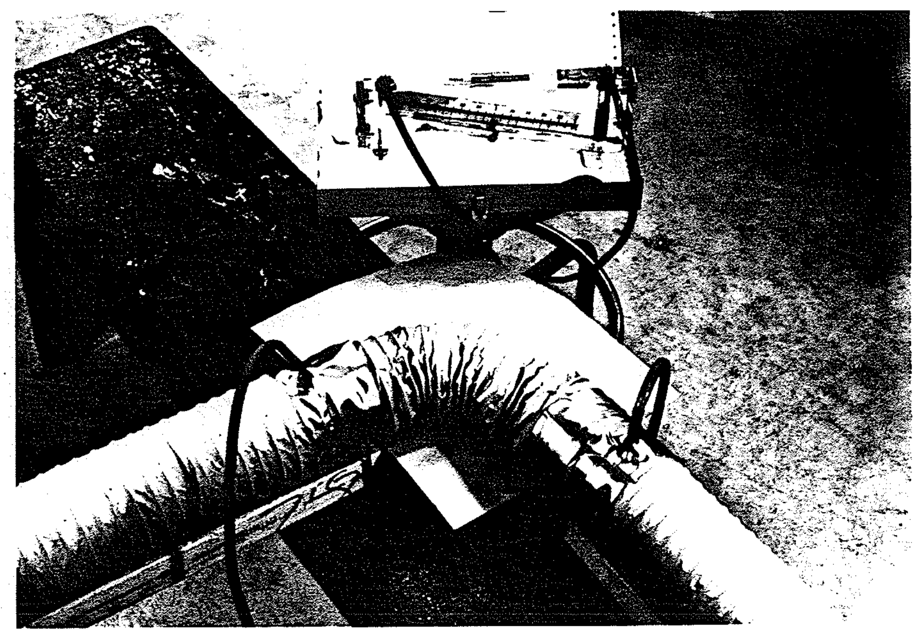


Figure 3.5 A 90° bend in flexible plastic pipe.

The pressure probes were connected to the manometer. The indicated pressure drop would be due to loss in the bend plus the frictional loss in 3 m of straight pipe. This measured value minus the pressure loss in the 3 m long pipe at the same air flow rate yielded the loss due to the pipe bend. The loss was expressed in terms of additional equivalent length of straight pipe in m.

3.7 Dumping Hopper Hood Design and Testing

A semilateral hood (Figure 3.6) and partial enclosure side-draft hood (Figure 3.7) for use as a dumping hopper and were designed (detailed dimensions in Appendix II) on the basis of 2 m/s face velocity and 165 L/s air flow rate through the hood. These values were selected from Industrial Ventilation (1974). Three prototypes of partial enclosure hoods were fabricated from galvanized sheet metal, wood, and polyethylene, while, one prototype of a semilateral hood was fabricated from galvanized sheet metal.

For testing these hoods, the space around the dumping hopper was covered with polyethylene sheet (Figure 3.8). The grain was dumped in the hopper from an overhead hopper bin and the dust generated during this operation was measured by a high volume air sampler. To compare the effectiveness of the hoods, dust concentration (Figure 3.9) in the enclosed space was measured when no hood was used. In order to maintain a constant amount of dust in the recirculated grain, about 2 g of grain dust was added to the grain at the inlet of screw conveyor loading the hopper, after every five minutes. The hood was installed on the dumping hopper the centrifugal fan was started to maintain the designed air flow rate through the hood and the dust concentration in the enclosed environment was measured. In addition, the pressure drop at the hood opening, air velocity in the duct, and air temperature (dry bulb and wet bulb temperature) were also recorded.

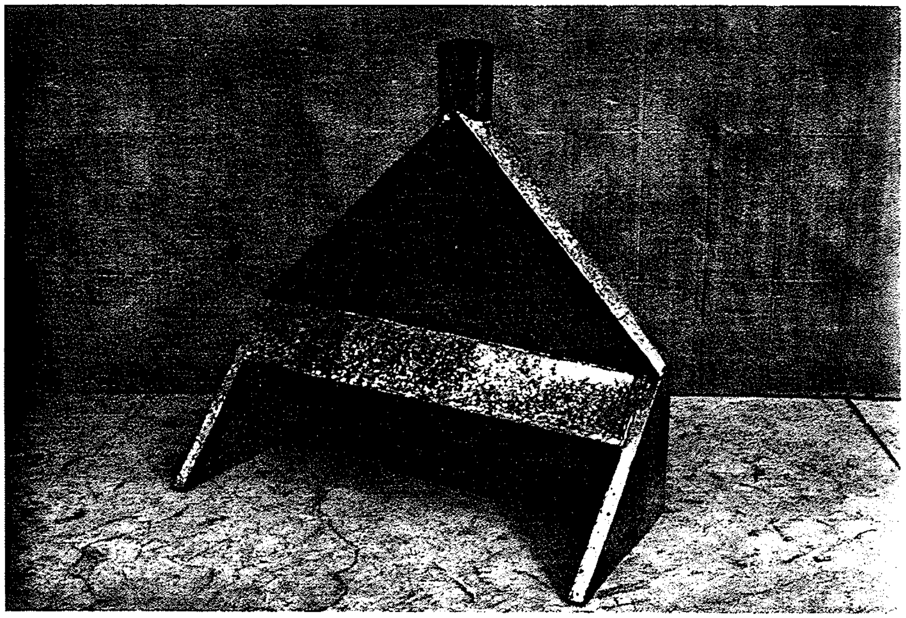


Figure 3.6 Semilateral hood for dumping hopper.

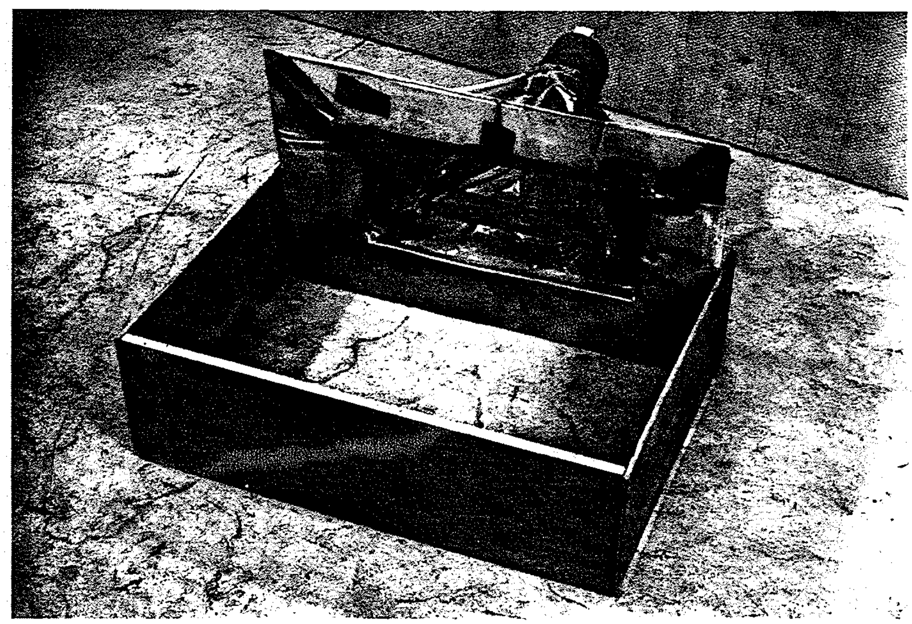


Figure 3.7 Partial enclosure side draft hood for dumping hopper.

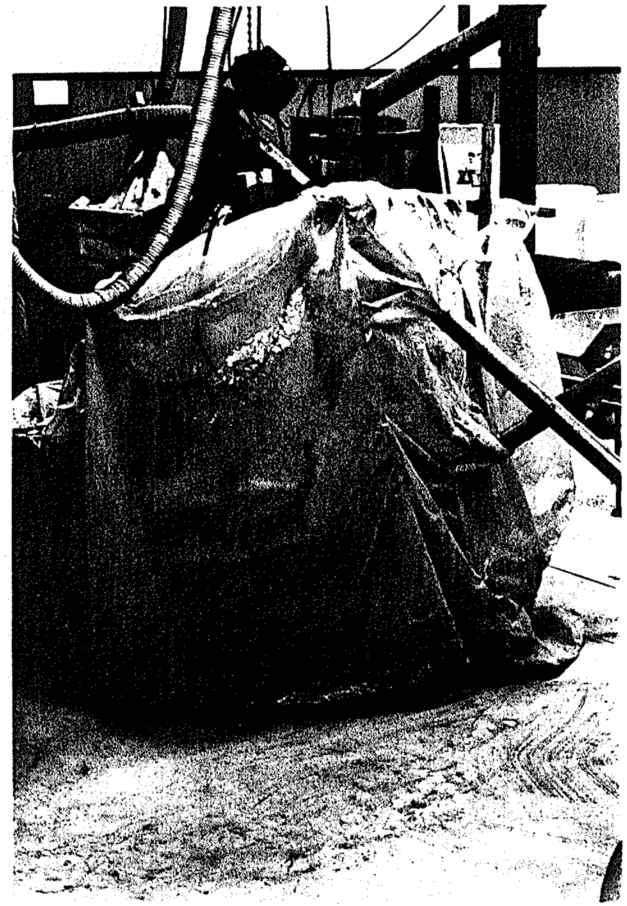


Figure 3.8 Enclosure for hood testing.

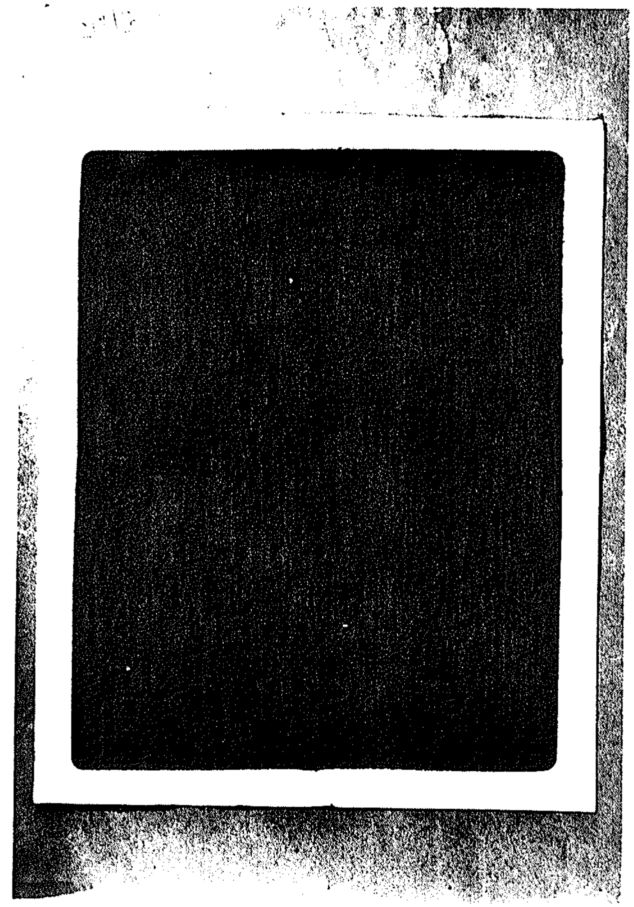


Figure 3.9 Dust sampled when no hood was used.

3.8 Balancing of the Experimental Duct System

A typical experimental dust control system (Figure 3.10) was balanced by using the static pressure balance method. At each junction, the static pressure in the main and the branch was maintained within five percent variation. If the difference in static pressure in two branches was greater than 20 percent, then a smaller diameter duct was used. When the pressure difference was between 5 and 20 percent, the balance was obtained by increasing the air flow in the branch with the lower loss. The increased volume flow was calculated as:

$$\text{Corrected flow (L/s)} = \sqrt{\frac{h_s \text{ larger}}{h_s \text{ lower}}} * \text{Original flow (L/s)} \qquad (3.2)$$

This procedure was utilized until the entire system was balanced. The minimum dust conveying velocity of 15 m/s was maintained in order to avoid dust settlement in the ducts. The calculated values of air flow rate and pressure drop were used to determine the system resistance at different flow rates and to select the optimum size of fan and motor.

3.9 Dust Feeding Mechanism

The dust dispenser used in this study is shown in Figure 3.11. The dust dispenser was fabricated so as to feed the dust in suspended form to the dust collecting equipment being tested. It was made by connecting a 25 mm diameter pipe to which another pipe of 12.5 mm diameter pipe at 90°. A funnel with a 60° angle to the vertical was provided for feeding dust manually into the dispenser. One end of the larger diameter pipe was kept in the centre of the dust duct of the

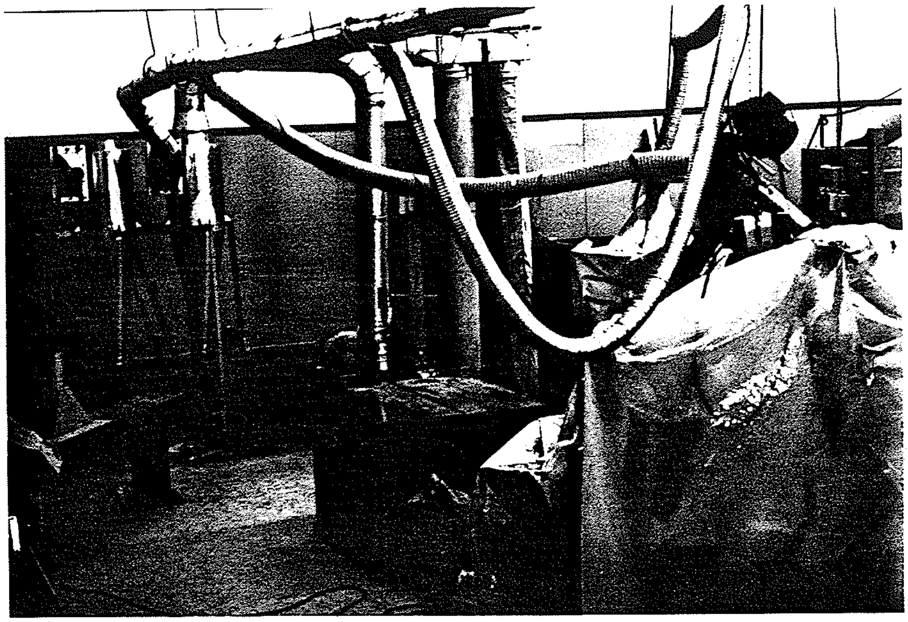


Figure 3.10 Typical experimental dust control system.

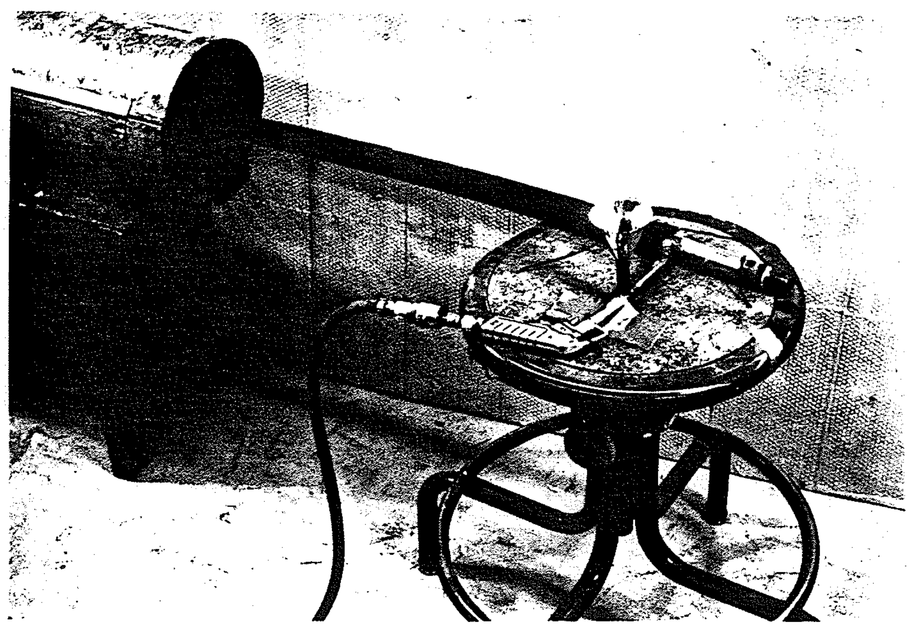


Figure 3.11 Dust dispenser for testing of dust control equipment.

control system. Compressed air was blown through a small diameter pipe at a pressure of 137.9 kPa. The pressure of the compressed air was selected according to the ANSI (1972). The dispenser was capable of feeding dust from 2 to 10 g/min. by varying the measuring spoon size.

3.10 Measurement of Collection Efficiencies of Cyclone Separator and Fabric Filters

The cyclone separator (dimensions in Appendix III) collection efficiency was measured by feeding 4 g/min. of grain dust to the cyclone separator through the dust dispenser. The test dust was brought from a farm seed cleaning plant and screened through a 100 μm size sieve. Grain dust below 100 μm size was used to test the cyclone separator. The air flow rate through the cyclone was measured with a Pitot-static tube. The test was conducted for one hour. The amount of dust retained in the cyclone divided by the mass of dust fed yielded the cyclone collection efficiency. A series of tests were conducted at different air flow rates with constant dust loadings.

The filter bag efficiency was determined according to the ANSI (1972) standard test code. The space around the bag was enclosed with polyethylene sheet for sampling the dust penetration through the fibre. The grain dust was introduced into the system upstream of the fan by means of the dust dispenser. The dust feed rate was 2g/min. selected on the basis of information reported by Poynting (1976). The high volume air sampler was placed in the enclosed area for gathering samples of dust emitted from the filter bag. The air flow rate through the filter was measured by a Pitot-static tube installed upstream of the fan. The total dust fed in one hour divided by air flow yielded the dust concentration in the unfiltered air. The dust concentration in the air downstream of filter bag was measured by the high volume air sampler. Collection efficiency was determined using equation 2.5. Three filter bags were tested for collection efficiencies and pressure drops. Other characteristics such as thread count, mass, weave count, thickness, and air permiability were also determined according to Canadian standard test methods for textiles. Combined collection efficiencies of the cyclone and filter bag were also measured for different air flow rates.

Pressure drops across the dust collectors were measured by connecting the manometer to pressure probes provided upstream and downstream of the collectors.

4. RESULTS AND DISCUSSION

4.1 Dust Emission in Various Seed Cleaning Plants

Dust emissions from various farm seed cleaning plants is presented in Table 4.1. It is shown in Table 4.1 that four out of eleven seed cleaning plants surveyed do not meet the allowable threshold limit value (10 mg/m^3) of dust emission in the grain industry set by Environment Canada (Labour Canada, 1977). The remainder of the plants are within the allowable limits and there seems to be no danger of pollution in the near future with the exception of Plant 9. Plant 9 was very near to the allowable limits of dust emission and the existing dust control system in the plant will need improvements in order to lower the dust emission level in the work environment. Total dust emission was lowest in plant 7 which is 1.04 mg/m^3 of air handled whereas a maximum of 114.23 mg/m^3 was observed in plant 10.

The dust emission level in a seed cleaning plant depends on the type of crop seed cleaned and the sampling location. In plant 4 more dust was generated in the work environment while cleaning barley than wheat. The dust concentration near the dump hopper was 114.23 mg/m^3 in plant 10 while, during the cleaning operation in the same plant, the respirable dust level was 6.99 mg/m^3. This value was much higher than that reported by Yoshida et al (1978) for the receiving hoppers in grain elevators. They reported a mass concentration of 40 mg/m^3 of air handled for wheat and barley.

The respirable mass fraction of grain dust was determined in only three plants (4, 8 and 10) only by the Andersen head (Table 4.1).

Table 4.1 Dust emissions from various farm seed cleaning plants

Date Sampled	Site code or plant number	Crop Seed	Dust Concentration mg/m^3	
			Total	Respirable
27.4.78	1	wheat	7.46	-
9.5.78	1	wheat	20.65	-
28.4.78	2	wheat	2.78	-
1.5.78	3	wheat	1.07	-
2.5.78	4	barley	60.44	-
27.4.79	4	wheat	55.06	47.0
4.5.78	5	wheat	2.64	-
4.5.78	6	barley	3.4	-
15.5.78	7	flax	1.04	-
19.4.79	8	fababeans	-	7.39
4.5.79	9	oats	7.7	-
22.5.79	10 (Dumping)	peas	114.23	-
22.5.79	10 (Near Cleaner)	peas	-	6.99
22.5.79	11	barley	37.17	-

Wheat seed generated more fine dust than fababeans and peas but the respirable mass fraction in fababeans and peas was almost equal. These levels are very close to total allowable limits and may pose serious health hazards as respirable dust penetrates to the lungs.

Figures 4.1 a and 4.1 b represent particle size mass distribution curves for wheat, fababeans and peas in the three plants. The particle size distribution function follows a log-normal distribution wheat dust. The mass median diameter and geometric standard deviation values in wheat dust were 78.75 μm and 20, respectively. The probable reasons for these large values may be due to: i. a large percentage of dust particles emitted in the plant were greater than 7 μm; and, ii. the physical characteristics of wheat dust and the heavy concentration of particles could have resulted in larger particles forcing the small size particles out of the jet stream thus impacting on the filter surface. Due to these factors a high percentage of dust was collected on the first stage. Matlock and Parnell (1976) also observed similar problems with the Andersen particle sizing head when the dust concentrations in the environment being monitored were high. It is, however, evident from Figure 4.1 b that mass median diameters for fababeans and peas were 15.56 and 13.06 μm, respectively. These values are relatively close to the results obtained by Norman et al (1977) for sorghum dusts.

4.2 Frictional Losses in Straight Sheet Metal Pipes and Plastic Flexible Pipes

Frictional loss versus air flow rate charts plotted on a log-log scale were developed for 76.2, 101.6, 127, and 203.2 mm diameter galvanized iron sheet metal pipes (Figure 4.2). The regression equations

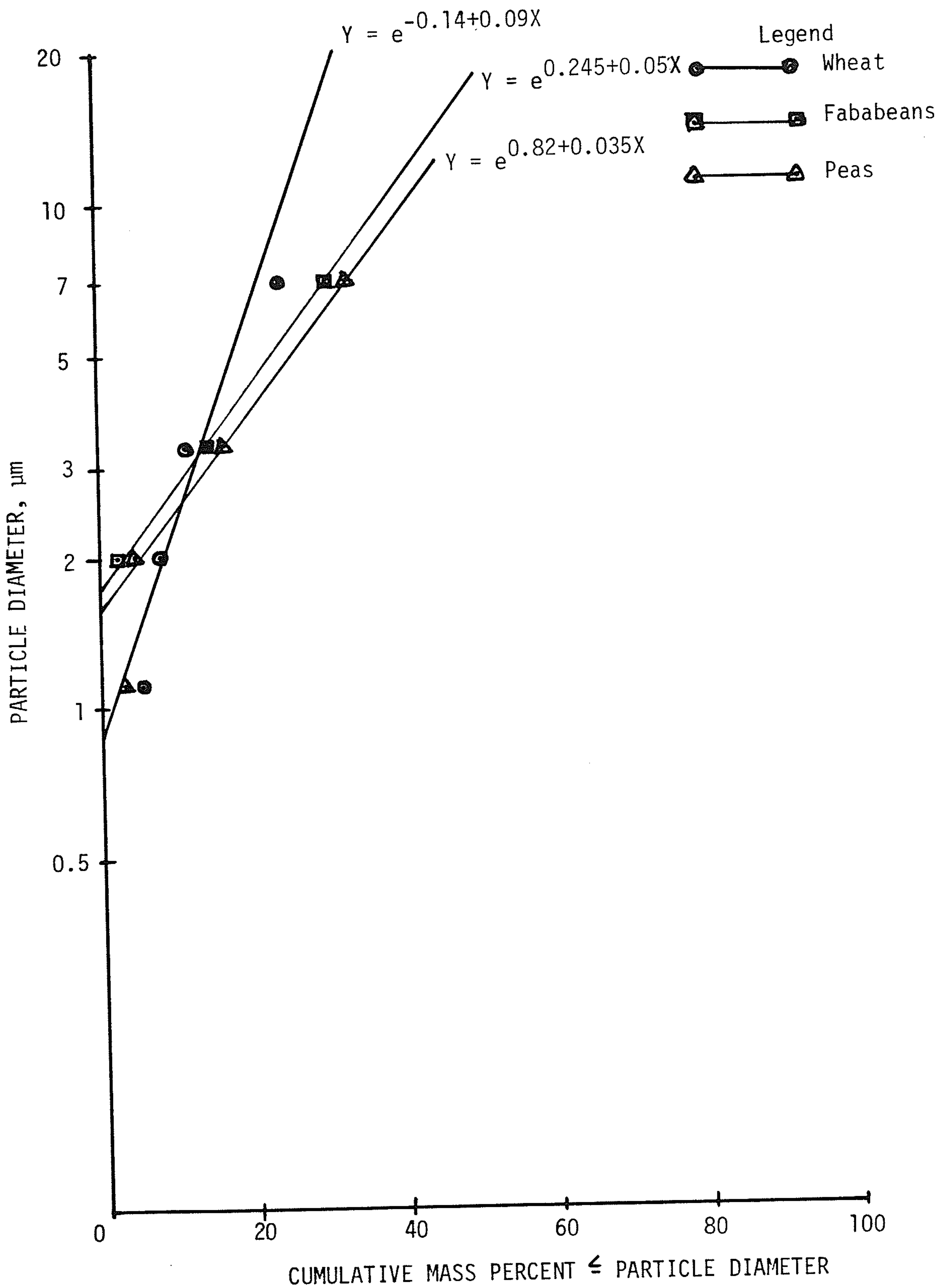


Figure 4.1 a Particle size distribution of respirable dust in seed cleaning plants

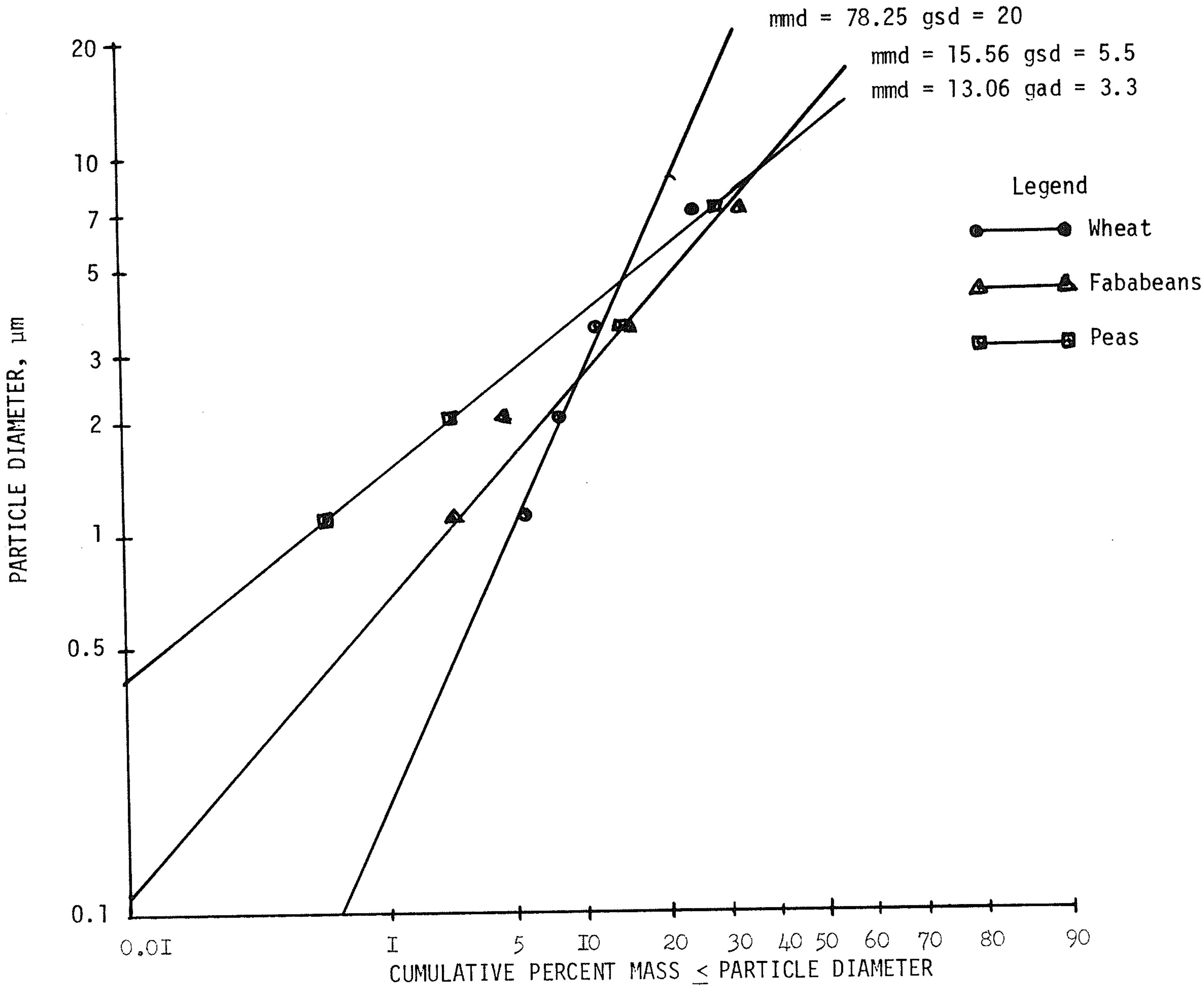


Figure 4.1b Particle size distribution of respirable dust in seed cleaning plants

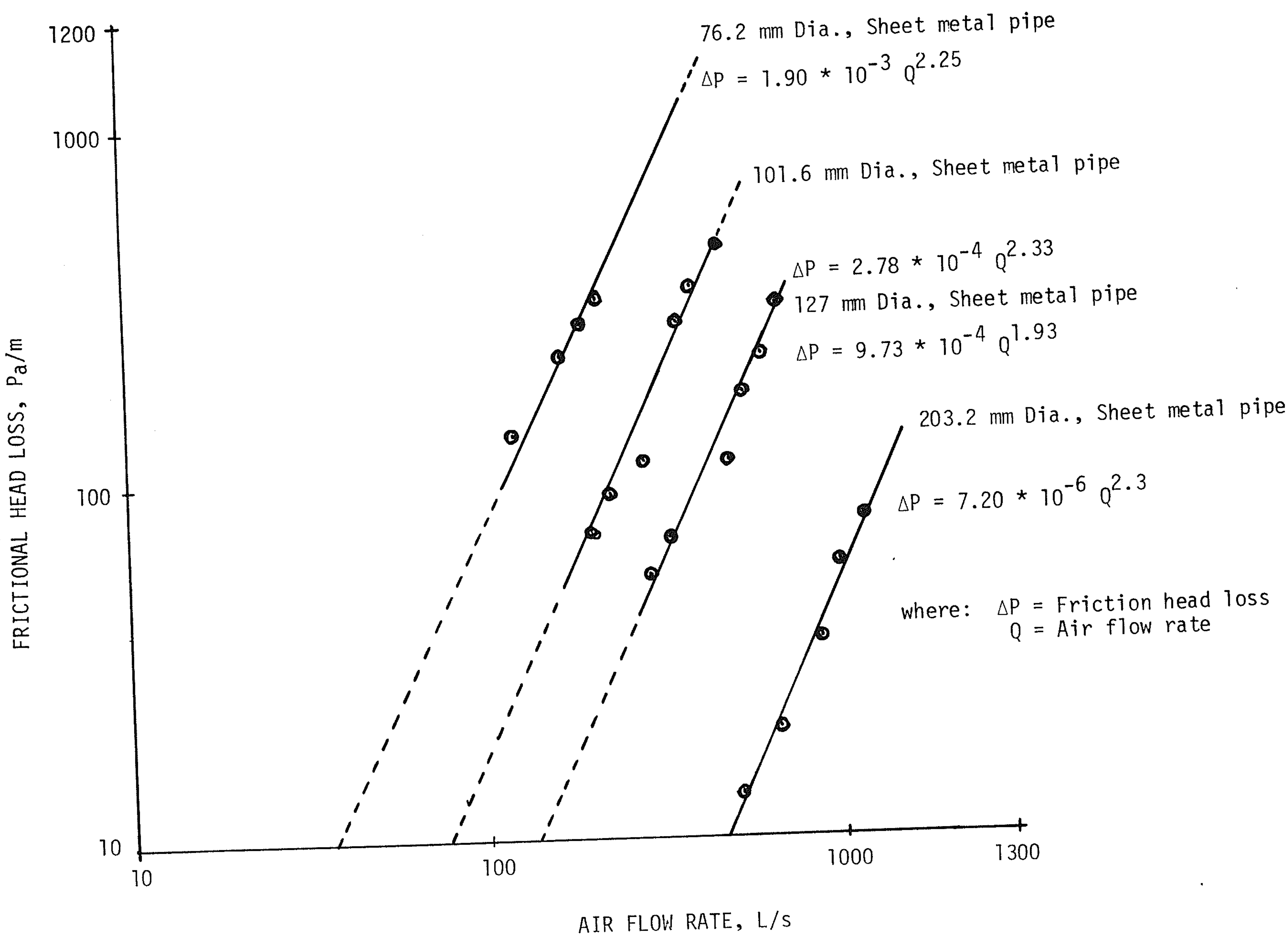


Figure 4.2 Frictional losses in straight galvanized iron sheet metal pipes

indicate that the frictional losses in Pa/m in straight sheet metal pipes were approximately proportional to the 2.2 power of the air flow rate (L/s). Such a result is slightly higher than the theoretical friction head loss in straight ducts which is proportional to the square of the air velocity or flow rates. The probable reason for the higher pressure loss could be due to the turbulence loss caused by the joints in the pipes. Houghtan et al (1939) also observed higher frictional losses in ducts having crude joints. They also found that for pipes without joints, the head loss was proportional to the 1.84 power of the air velocity. The slopes of the regression equations were slightly different. It is, however, evident from the graphs and theoretical analysis that the average regression coefficient for all the four pipes was 2.2. The pressure head loss is dependent on pipe diameter, as the pipe diameter is reduced from 203.2 mm to 76.2 mm, the head loss increased considerably.

Figure 4.3 reveals the effect of air flow rates in L/s on the frictional head loss in Pa/m for sewn vinyl ribbon pipes and corrugated plastic pipes. The regression analysis of these curves indicated that the slopes of regression equations of all the four pipes varied. This difference could be due to a different type of reinforcement in the pipes which results in a change in pipe diameter when the pipe is not fully extended. The average regression coefficient for all the four pipes was 1.9. Thus, the frictional head loss was approximately proportional to the 1.9 power of the air flow rate or air velocity. The values of the intercepts for the regression equations in Figures 4.2 and 4.3 are larger for flexible pipes than the sheet metal pipes of the same diameter. For example the intercept for 76.2 mm sewn vinyl ribbon pipe

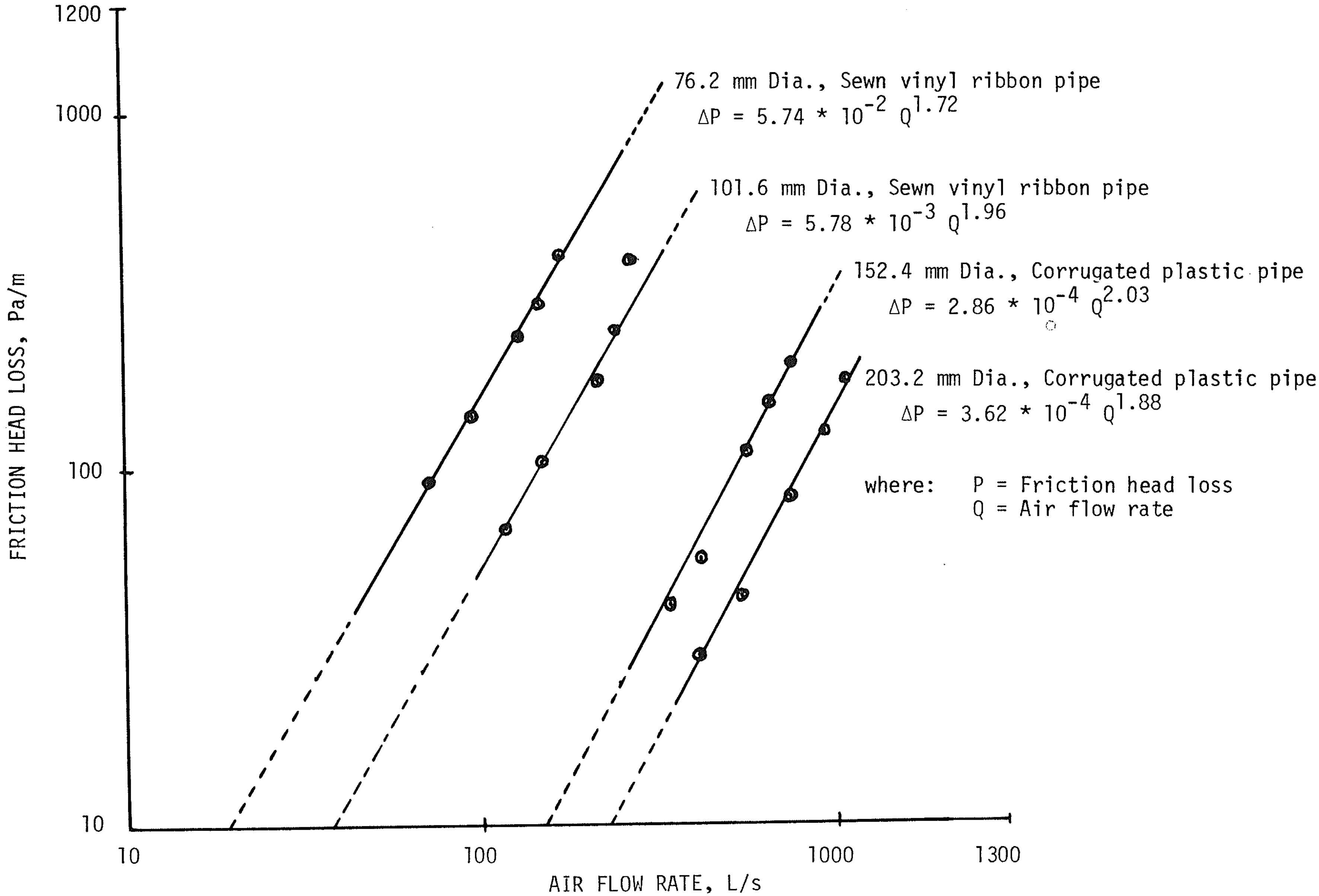


Figure 4.3 Frictional head losses in plastic flexible pipes

is $5.74 * 10^{-2}$ while for the same diameter sheet metal pipe is $1.9 * 10^{-3}$. Figures 4.2 and 4.3 show that the frictional head losses in flexible pipes are 2 to 2.5 times the losses in sheet metal pipes due to the fact that the diameter of flexible pipe cannot be maintained uniform throughout its length. When the pipe is not fully extended the diameter cannot be maintained uniform and hence, increases the frictional head loss as shown in Figure 4.4. The sewn vinyl ribbon pipes resulted in the highest pressure drops whereas sheet metal pipes offered the least resistance. These results are in agreement with those obtained by Svistovski (1978).

The frictional losses in Pa/m in these pipes can be predicted/ calculated at different air flow rates (L/s) by using regression equations or with the help of frictional charts.

4.3 Frictional Losses in Plastic Flexible Pipe Bends

The head losses in pipe bends depend on the centerline radius of the bend, diameter of the duct, and angle of the bend. The head losses in terms of equivalent length of straight pipe in the plastic flexible pipe bends are given in Table 4.2. These results indicate that the losses are strongly dependent on the centreline radius of the bend. For a 90° bend in 101.6 mm diameter pipe when the centreline radius was increased from 1 D to 1.5 D the equivalent length of the straight duct decreased from 3.02 m to 1.9 m. Thus, the loss is a function of the centreline radius of the bend. The bend loss in terms of equivalent length of pipe remained the same for different flow rates. Friction losses in bends are larger when the duct diameter is increased. For

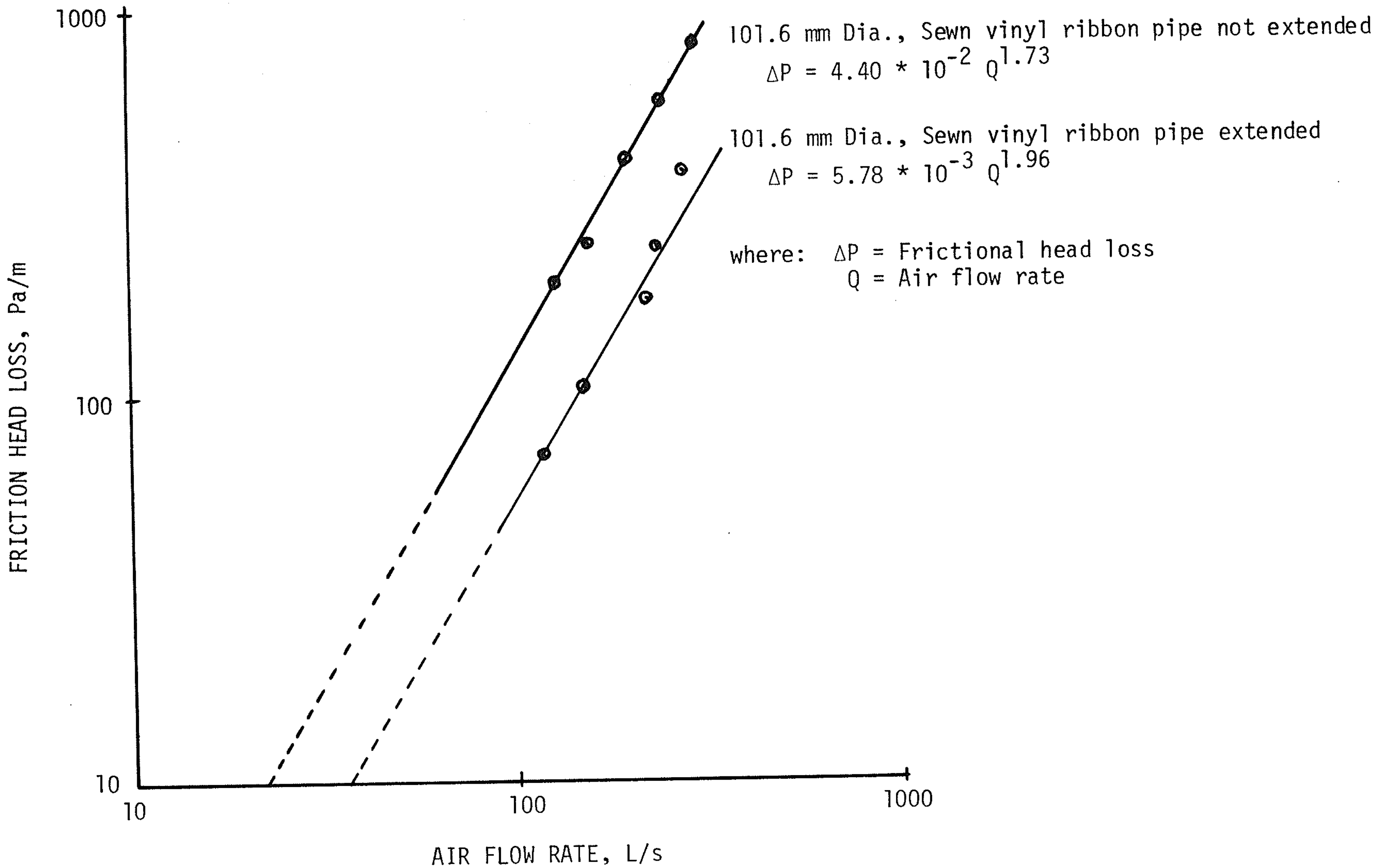


Figure 4.4 Frictional losses in plastic flexible pipes

Table 4.2 Frictional head loss in flexible plastic pipe bends

Pipe diameter D mm	Equivalent length of straight pipes, m							
	90-degree bend centreline radius				45-degree bend centreline radius			
	1 D*	1.5 D	2 D	2.5 D	1 D	1.5 D	3 D	3.5 D
76.2	-	-	0.91	-	0.8	-	-	-
101.6	3.02	1.9	-	-	1.98	0.69	-	-
152.4	-	-	-	1.48	-	-	-	0.47
203.2	-	-	-	1.71	-	-	1.00	-

*Diameter

example, a 90° bend changes in duct diameter from 152.4 to 203.2 m increased the equivalent length of straight pipe from 1.48 to 1.71 m. The losses in 45° bends are about 50 to 60% of the losses in 90° bends for the same size of duct and centreline radius. These losses are fairly the results to those reported in Industrial Ventilation (1974) for elbows made of galvanized iron sheet metal.

4.4 Dumping Hopper Hood Performance

The dust pick-up at the emission source depends on the type of hood employed and its location and quantity of air flowing through the hood. Table 4.3 depicts the performance of the four hoods tested in comparison with no hood at the dumping hopper. An average dust concentration of 250.9 mg/m^3 of air handled was measured when no hood was employed at the dumping hopper. This concentration level was lowered to nearly 1 mg/m^3 by all four hoods which is well below the allowable emission rate; i.e., 10 mg/m^3 of air handled. A semilateral galvanized sheet metal hood allowed the minimum amount of dust 0.85 mg/m^3 to escape in the work environment while partially enclosed polyethylene hood allowed 1.24 mg/m^3. Nevertheless, no significant difference in dust concentrations resulted between the two hoods. The quantity of air passing through all the hoods was maintained constant. A minimum hood entry loss of 199.2 Pa was observed in a semilateral galvanized sheet metal hood while the maximum loss of 261.45 Pa occurred in the partially enclosed wooden hood. In the later case, the rough wooden surface offered greater resistance to air entering the hood.

Table 4.3 Hood performance at receiving hopper in seed cleaning plant

Type of hood	Dust concentration mg/m^3					Hood entry loss Pa				
	Test Number					Test Number				
	1	2	3	Mean	S.D.	1	2	3	Mean	S.D*
Semi-lateral sheet metal hood	0.76	0.78	1.00	0.85	0.13	199.2	199.2	199.2	199.2	0.0
Partially enclosed sheet metal hood	1.15	0.99	0.8	0.98	0.17	205.42	202.31	199.2	202.31	3.11
Partially enclosed wooden hood	0.95	0.91	0.9	0.92	0.03	261.45	261.45	261.45	261.45	0.0
Partially enclosed polyethylene hood	1.55	1.04	1.14	1.24	0.27	232.81	230.32	226.60	229.90	3.12
No Hood	256.15	247.5	249.1	250.9	4.59	-	-	-	-	-

*Standard deviation

The effect of hood location with regards to dust concentration levels at the dumping hopper is shown in Table 4.4. An average dust level of 0.98 mg/m^3 was measured when the hood was 280 mm away from the dumping source with three sides of hood covered. This concentration increased to 2.07 mg/m^3 when the distance of the hood was increased to 560 mm. The dust level increased to 13.78 mg/m^3 when the sides of the hood were not covered at a distance of 560 mm from dumping source. Such an increase in dust concentration is due to a decrease in capture velocity at the emission source since most of the air would enter from the sides of the hood without capturing any dust particles. Consequently, insufficient dust pick-up allowed more dust to escape into the work environment. The proper location of the hood is of utmost importance for complete capture of dust particles at the dumping hopper.

4.5 Flow Rate and Static Pressure Balancing of a Typical Exhaust System

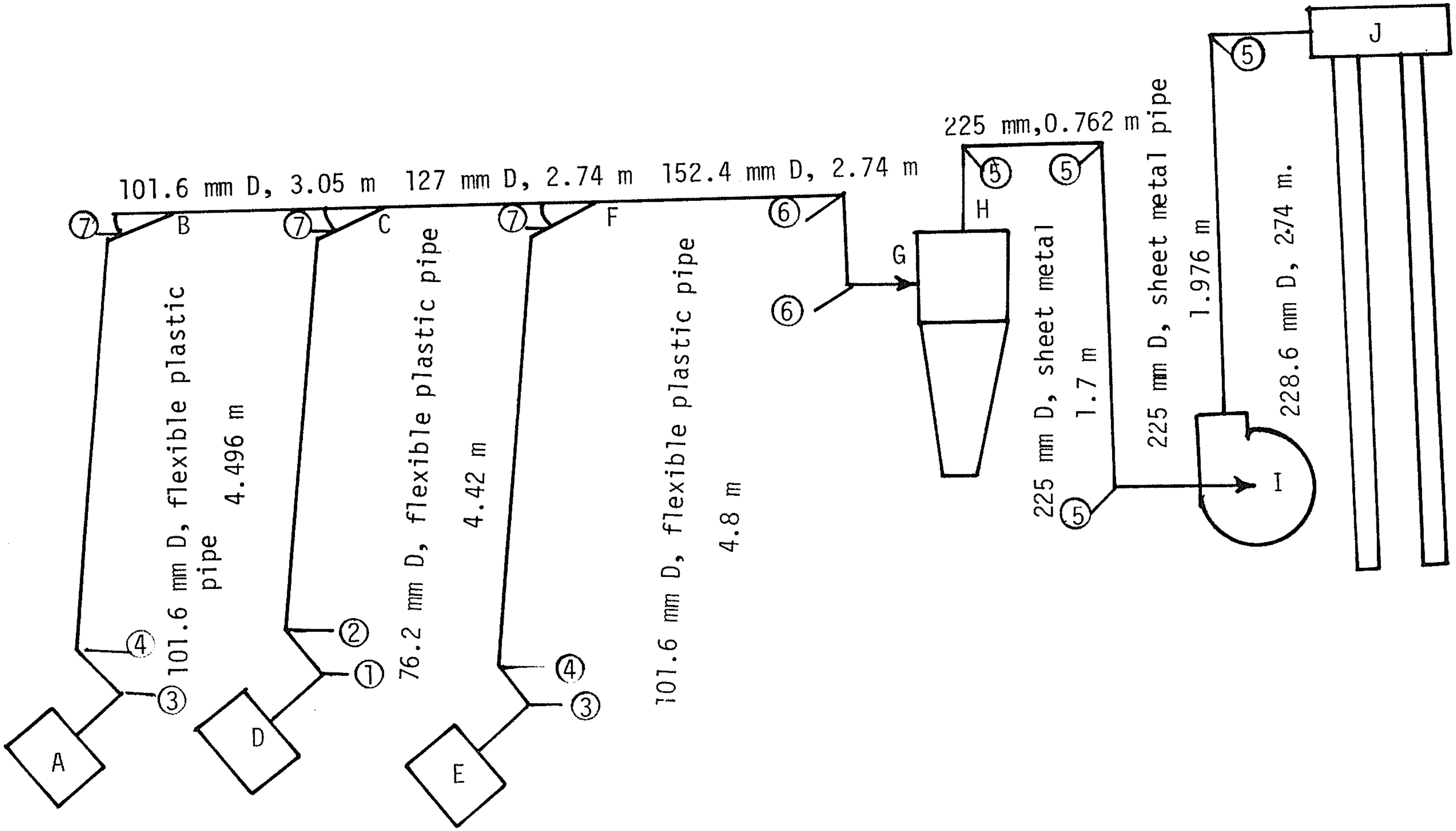
The layout of the duct work for experimental dust removal system is shown in Figure 4.5. Air flow rate and static pressure balancing of this system was accomplished using the balanced duct method. Air flow rates of 144.5, 79.23 and 163.3 L/s were maintained in the hoods at the bin, transfer point, and dumping hopper, respectively in order to capture and convey the dust to the cyclone and filter bags. The duct velocity was maintained higher than 17.8 m/s to avoid settling of dust in the ducts. The calculations necessary to balance the system are shown in Table 4.5. The calculation of the system resistance was started at the line of greatest resistance (A-B) and continued until the entire system was balanced. The pressure loss data was taken from the frictional loss

Table 4.4 Effect of hood location on dust concentration at dump hopper

Hood location	Pressure drop Pa	Dust concentration mg/m^3 Test Number				
		1	2	3	Mean	S.D.*
No Hood	-	256.15	247.5	249.1	250.9	4.59
280 mm from dumping source with three sides covered	202.31	1.15	0.99	0.8	0.98	0.17
560 mm from dumping source with three sides covered	202.31	2.14	2.01	2.05	2.07	0.06
560 mm from dumping source with three sides open	202.31	13.4	13.55	14.4	13.78	0.54

*Standard Deviation

1. Throat radius for 90° bend in 76.2 mm D* flexible pipe = 2D
2. Throat radius for 45° bend in 76.2 mm D flexible pipe = 1D
3. Throat radius for 90° bend in 101.6 mm D flexible pipe = 1.5D
4. Throat radius for 45° bend in 101.6 mm D flexible pipe = 2.5D
5. Throat radius for 90° elbow in 225 mm D sheet metal pipe = 1D
6. Throat radius for 90° elbow in 152 mm D sheet metal pipe = 1.5D
7. All branch entries = 30°



*Diameter

Figure 4.5 Layout of duct work for experimental dust removal system charts for galvanized sheet metal pipes (Figure 4.2) and plastic flexible pipes (Figure 4.3) and Table 4.2 for flexible pipe bends. The static pressures at the junction of main C-F and branch E-F are not equal in Table 4.5 so the air velocity in the branch was increased to 20.19 m/s and accordingly, the air flow rate was calculated in the branch. This procedure resulted in a pressure drop of 1.71 kPa at 386.6 L/s in the system. These values were then used to calculate the system resistance curve as shown in Table 4.6. The fan data was taken from fan performance table supplied by the manufacturer for a flow of 386.6 L/s and corrected to 1800 rpm as shown in Table 4.7. These datas were used to plot system characteristic curves (Figure 4.6). For satisfactory performance, the system should be operated where the system curve and fan curve meet. The power requirement at this operating point was 1.16 kW.

4.6 Cyclone Separator

4.6.1 Collection Efficiency

The collection efficiency of the cyclone separator varied with the air volume flow rate (Figure 4.7). Efficiency increased sharply with flow rate up to 150 L/s but for air flow rates greater than this value, the increase in the collection efficiency was gradual. At an air flow rate of 275 L/s the cyclone separator attained collection efficiency of about 82% and no improvement in the efficiency resulted as the air flow rate increased.

Inlet dust particle size affected the collection efficiency of

Table 4.5 Balancing calculations for exhaust system

Branch or main	Duct Diameter (mm)	Air velocity (m/s)	h_v (Pa)	Hood % h_v	losses (Pa)	Length straight duct (m)	Elbows No. & angle	equivalent length (m)
A-B	101.6	17.78	190.0	25	54	4.5	2,45° 1,90° 1,U	1.37 1.91 0.91
B-C	101.6	17.78	190.0	-	-	3.05	-	-
D-C	76.2	17.78	190.0	25	54	4.42	2,45° 1,90° 1,U	1.56 0.91 0.31
C-F	127.0	17.63	-	-	-	2.74	-	-
E-F	101.6	17.78	190.0	40	76	4.80	2,45° 1,90° 1,U	1.37 1.91 0.91
F-G	152.4	21.17	271.25	-	-	2.74	2,90°	4.27
G-H (cyclone)	304.8* 76.2	16.65	167.0	150	250	-	-	-
H-I	228.6	9.4	-	-	-	2.44	3,90°	17.37
I-J	228.6	9.4	-	-	-	1.84	1,90°	5.80
Bags (two)	-	-	-	-	-	-	-	-

(continued)

Table 4.5 (continued)

Branch No. & angle	entry equiv. length (m)	Total equiv. length (m)	Resistance H_r (Pa/m)	(Pa)	Static Press. h_s branch	(Pa) main	Correct. branch veloc.	Air Flow (L/s) branch	main
1,30°	0.91	9.60	63.49	609.55	853.55	853.55	-	144.15	-
-	-	3.05	33.46	102.0	-	955.55	-	-	144.15
1,30°	0.61	7.81	93.99	734.39	978.3	955.55	17.37	79.23	223.38
-	-	2.74	24.61	67.52	-	1023.07	-	-	223.38
1,30°	0.91	9.90	63.49	628.94	894.94	1023.07	20.19	163.3	386.68
-	-	7.01	42.62	298.72	-	1321.79	-	-	386.68
-	-	-	-	250.5	-	1572.29	-	-	386.68
-	-	19.8	5.75	113.78	-	1686.07	-	-	386.68
-	-	7.62	5.75	43.70	-	1729.77	-	-	386.68
-	-	-	-	250.0	-	1979.77	-	-	386.68

Fan static pressure $= h_s - h_v$

$= 1979.77 - 271.25$

$= 1708.52$ Pa.

Table 4.6 Calculations for system resistance curve

Air Flow L/s	Multiplying factor	New h_s* kPa
386.67	-	1.71
300.0	$(300/386.67)^2 * 1.71$	1.02
200.0	$(200/386.67)^2 * 1.71$	0.46
100.0	$(100/386.67)^2 * 1.71$	0.11
400.0	$(400/386.67)^2 * 1.71$	1.83
500.0	$(500/386.67)^2 * 1.71$	2.84
600.0	$(600/386.67)^2 * 1.71$	4.09

*Static pressure

Table 4.7 Fan capacity at various static pressure at 1800 rpm

From fan data				corrected to 1800 rpm		
Fan speed rpm	Air flow L/s	Static press. h_s (kPa)	Power kW	Air flow L/s	Static press. h_s (kPa)	Power kW
1772	1098.33	0.75	2.46	1113.67	0.77	2.58
1792	1000.53	1.0	2.36	1005.0	1.01	2.39
1768	837.71	1.24	1.95	853.33	1.30	2.06
1795	671.11	1.49	1.70	673.0	1.51	1.72
1795	387.94	1.74	1.15	389.0	1.75	1.16

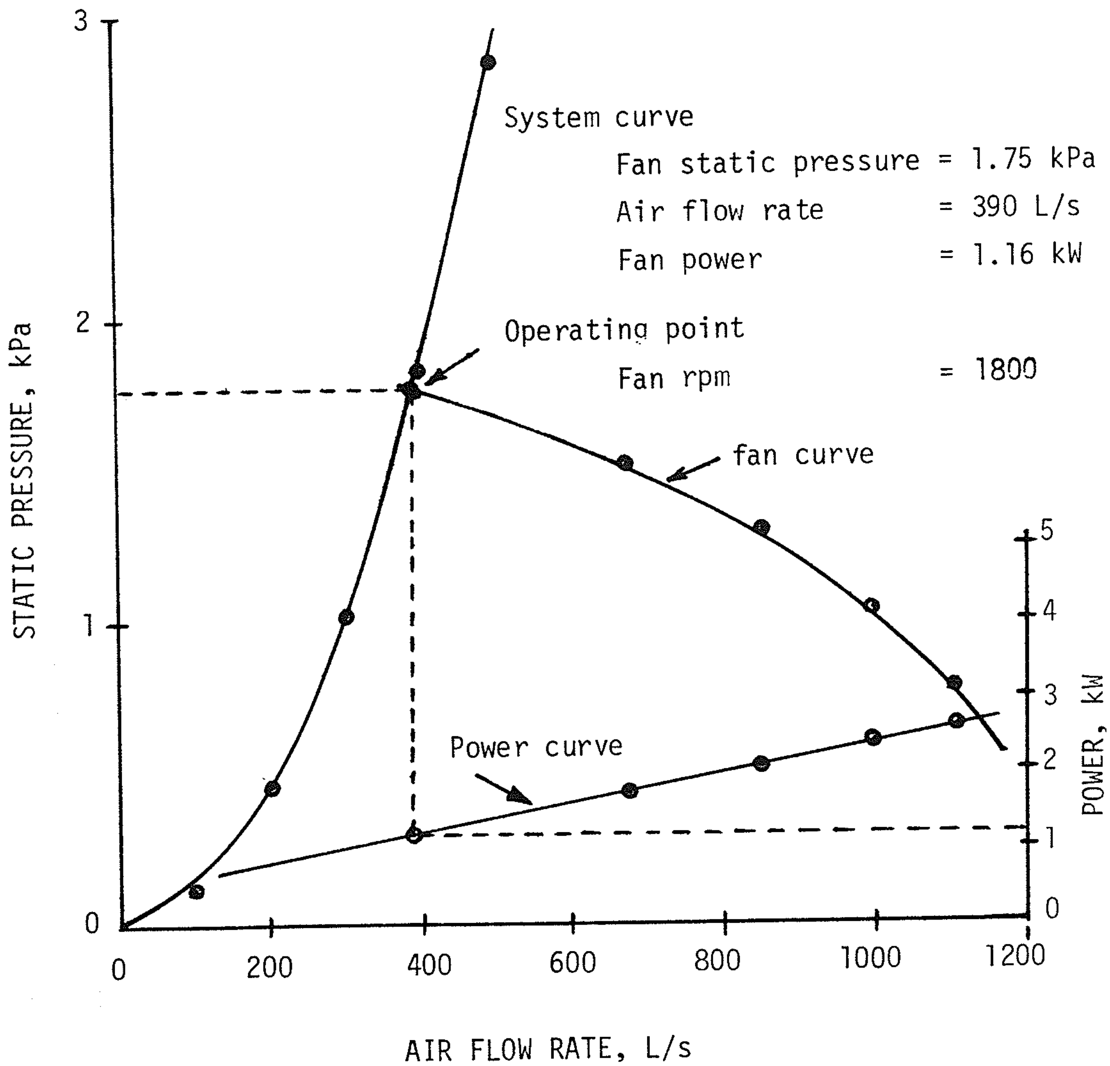


Figure 4.6 Characteristic curves for dust control system

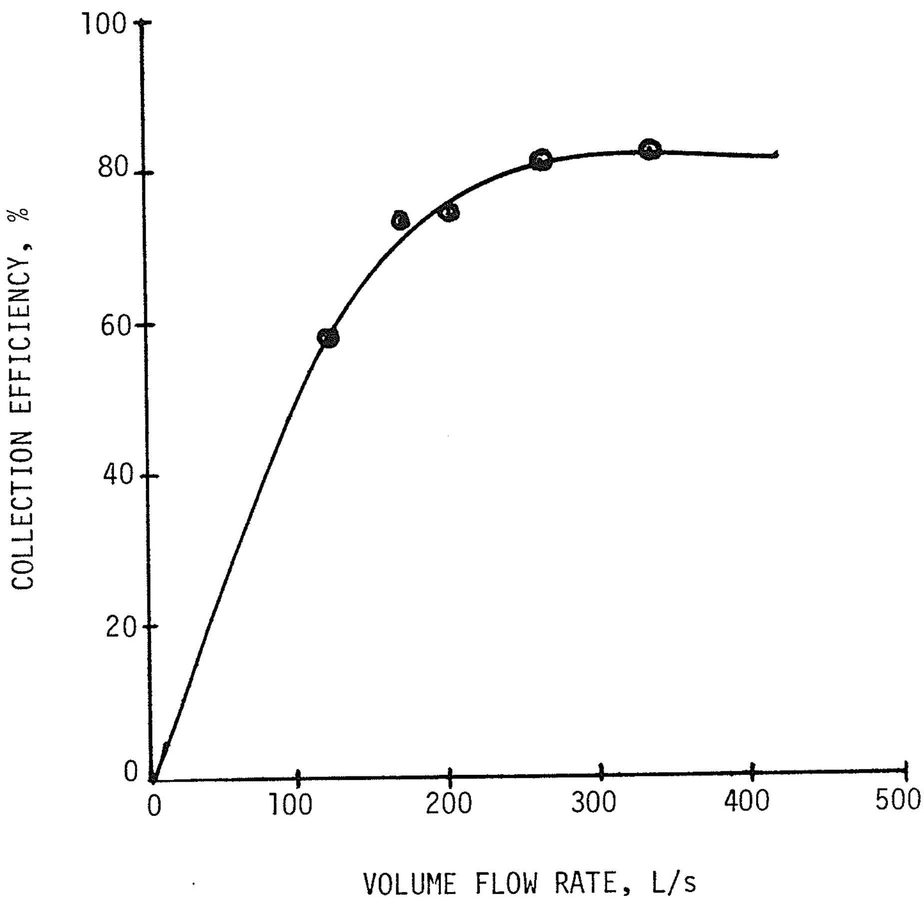


Figure 4.7 Dust collection efficiency of cyclone separater

the cyclone separator. However, increased inlet velocity or air flow rate improved the collection efficiency but also increased the pressure drop in the cyclone separator. In this study the collection efficiency increased until the inlet air velocity reached 7.5 m/s but remained constant for the inlet air velocity range of 7.5 m/s to 16.8 m/s. Thus, it was hypothesized that the effect of inlet air velocity on collection efficiency was insignificant beyond 7.5 m/s because of the large percentage of dust particles less than 10-20 μm in size that were escaping the cyclone separator. The cyclone separator was 100% efficient on particles greater than 20 μm in diameter in collecting cotton dust (Wesley et al, 1970). Furthermore, the collection efficiency of the cyclone separator was smaller than the efficiencies of large diameter cyclones reported by Silverman (1953). He observed cyclone efficiencies between 78 to 88% for inlet air velocities of 8.5 to 10.2 m/s in controlling different industrial dusts.

4.6.2 Pressure Drop

The pressure drop in the cyclone separator increased with an increase in air flow rate (Figure 4.8). The regression equation of the curve indicated that the pressure drop in the cyclone separator was proportional to the 2.13 power of the air flow rate (Figure 4.8) the result was slightly larger than the theoretical value whereby, the pressure drop was proportional to the inlet air velocity or air flow rate. The higher pressure drop probably occurred due to a baffle connected the modified inlet of the cyclone which could have caused re-entrainment at the cyclone inlet and hence, an increased pressure drop. Moreover, the

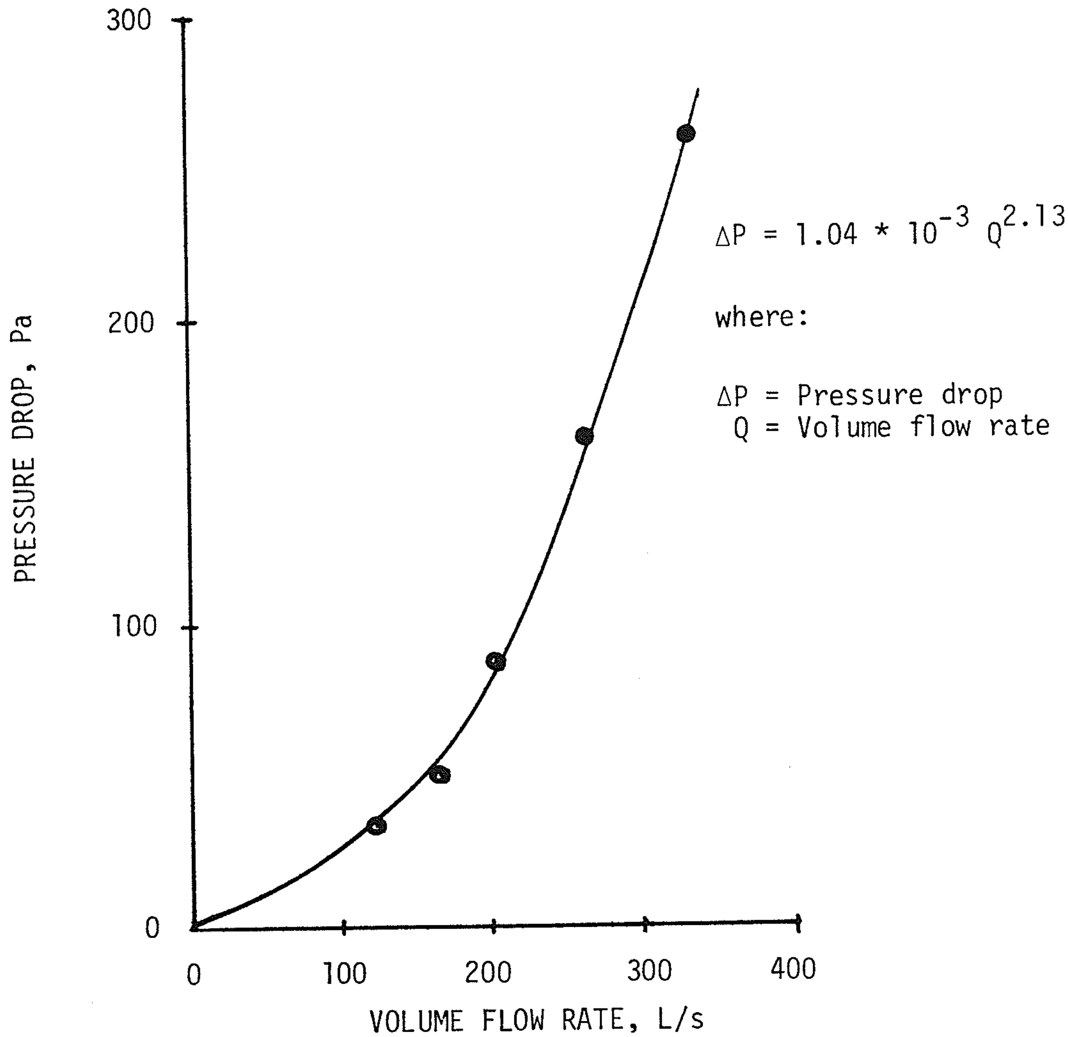


Figure 4.8 Pressure drop in cyclone separater

rough surface of the inside walls of the cyclone also could have contributed to the increased pressure drop. Stairmand (1949) also found that measured pressure drops were higher than those calculated theoretically in the cyclone separators.

The static inlet pressure of the cyclone separator also varied with the air flow rate (Figure 4.9). The regression equation for the inlet pressure (Figure 4.9) indicated that the cyclone inlet pressure was proportional to the 1.84 power of the air flow rate. This finding agrees with the results obtained on frictional losses in plastic flexible pipes reported in section 4.2. The cyclone inlet pressure curve can be used for balancing the system and adjusting the air flow rates through the cyclone separator. By measuring the inlet pressure of the cyclone in a system the air flow rate can be predicted and from the efficiency curve collection efficiency of the unit can be obtained.

From the Figures 4.7 and 4.8, the cyclone separator attained highest efficiency at 275 L/s with a pressure drop of 175 Pa. As noted in section 4.5, for a typical exhaust system the air flow rate should be 390 L/s and the pressure drop in the cyclone separator would be 350 Pa. This means that the pressure drop in the cyclone would have doubled without any increased in collection efficiency. Under these conditions the abrasion and wear of the unit as well as fan power requirement increased. Thus, this unit was not representative of a typical exhaust system.

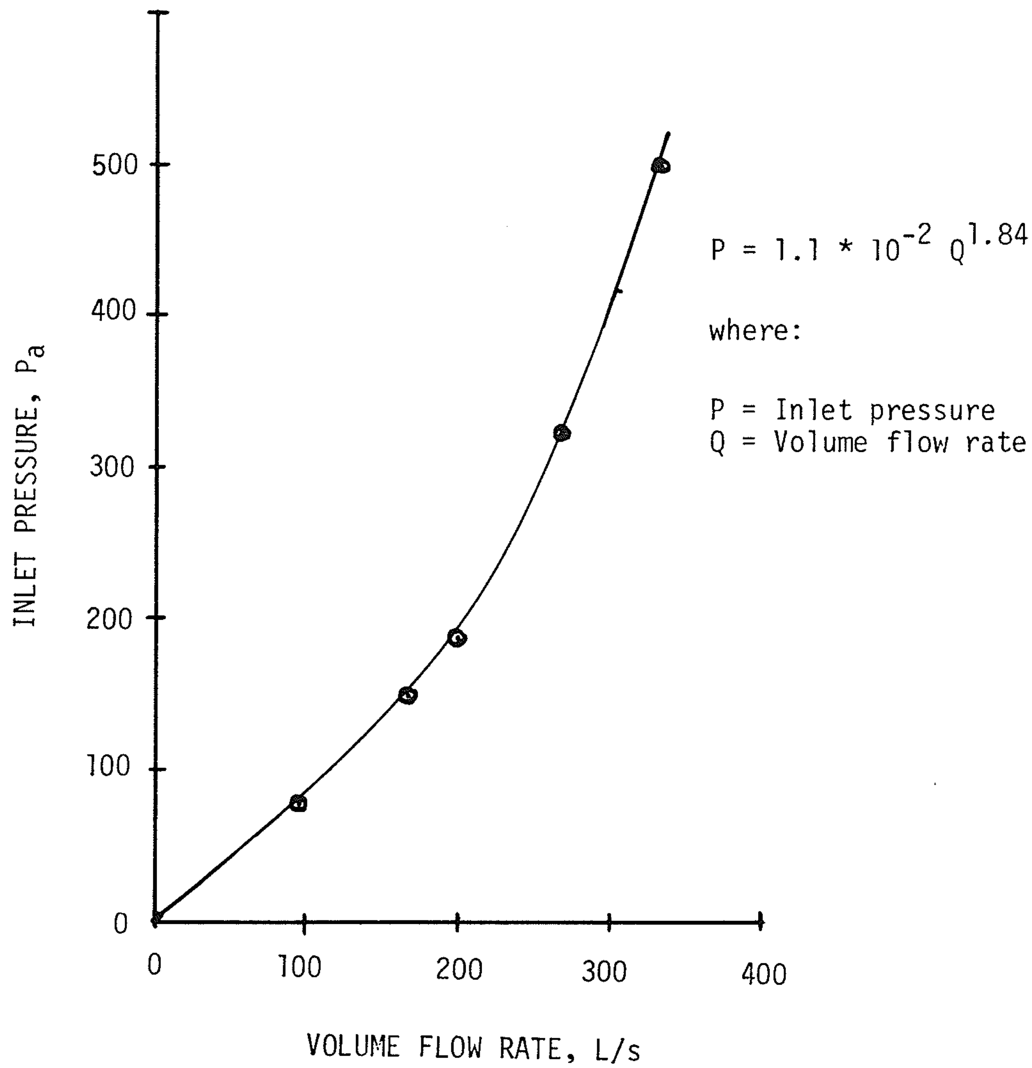


Figure 4.9 Inlet pressure in the cyclone separater

4.7 Fabric Filters

4.7.1 Specifications of Fabric Filters

The characteristics of the filter cloth depend on material and construction of the yarn such as weave, count and finishes. Table 4.8 indicates the characteristics of fabric filter bags tested for grain dust control. Bag I had a greatest number of threads per cm^2 (i.e., 40 * 26) lowest air permiability (2.17 $cm^3/cm^2/s$) and largest fabric mass (338.39 g/m^2) compared to bag II and II. Bag III was thicker , had higher permiability and lower mass than bags I and II. High air permiability is not necessarily good for dust control as dust penetration through the bag would be higher. Air permiability was lower in bags I and II but higher in bag III. Bag II had a thread count of 22 * 20/cm^2 and a fabric mass of 298.91 g/m^2.

4.7.2 Pressure Drop

Bag pressure drop varied with the air-to-cloth ratio (Figure 4.10). This figure shows a nonlinear relationship between the bag pressure drop and the air-to-cloth ratio. When the air-to-cloth ratio was increased the curves straightened out. As the curve eventually turns vertical no gain in air flow can be achieved by increasing the pressure drop. In other words, if the bags were undersized, increasing the fan speed would not result in increase in flow because the bags will be overloaded with the dust.

The filter bag pressure drop depends on dust loadings, fabric properties like weave, count and air permiability. In Figure 4.10 the

Table 4.8 Specifications of fabric filter bags

Characteristics	Bag No.		
	I	II	III
Fabric count:			
warp (no./cm)	40	22	16
weft (no./cm)	26	20	28
Weave count:			
type	twill	twill	twill
angle of weave	75°	45°	45°
direction of diagonal	right	right	left
count	4/1	2/2	3/1
Thickness at 0.69 kPa (mm)	0.87	0.73	1.32
Air permiability ($cm^3/cm^2/s$)	2.17	7.63	28.82
Mass of fabrics (g/m^2)	338.39	298.91	278.36

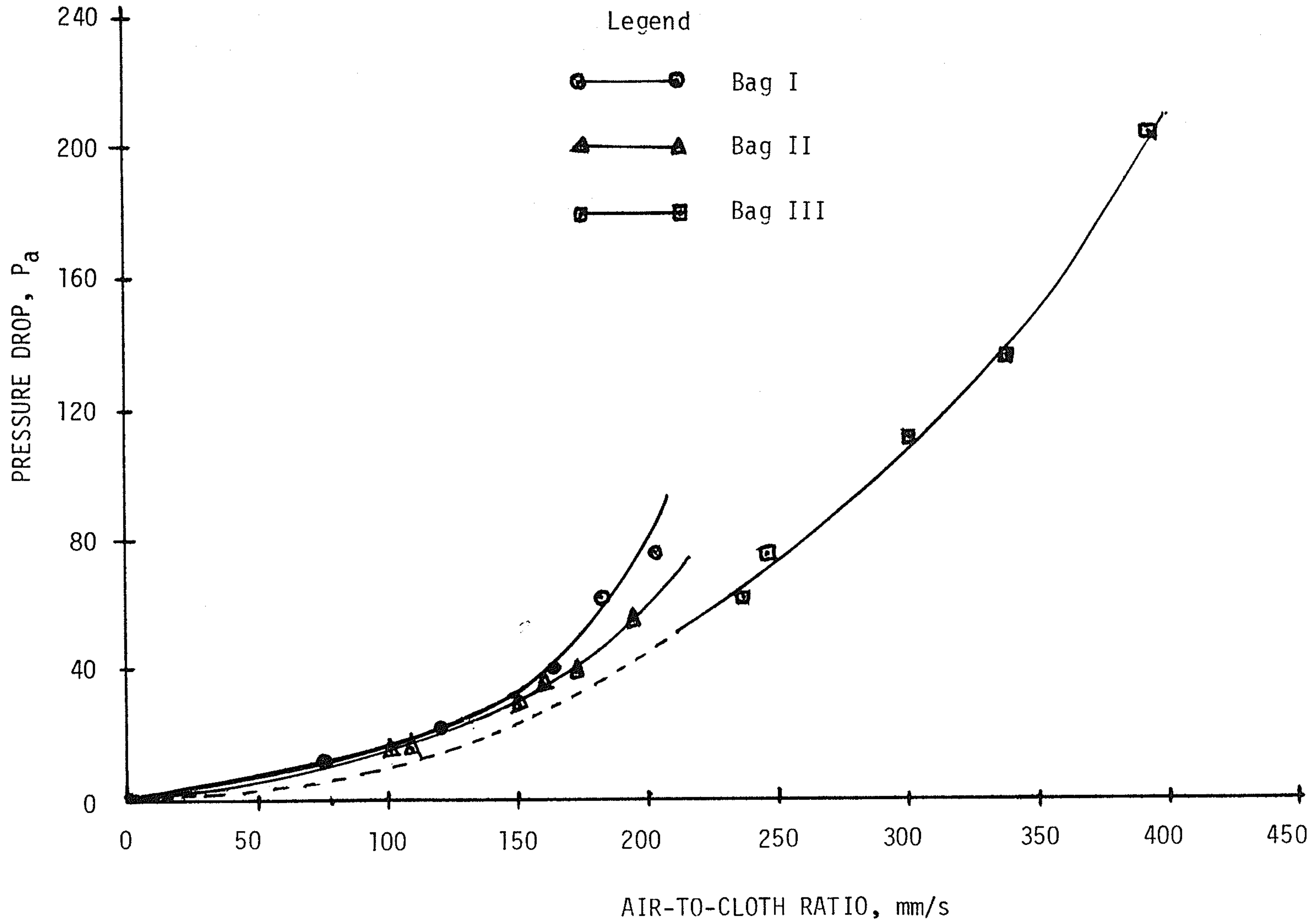


Figure 4.10 Bag pressure drop versus air-to-cloth ratio

pressure drop was highest in bag I, for example at air-to-cloth ratio of 200 mm/s the pressure drops of 80, 60 and 45 Pa were observed in bags I, II and III respectively. The pressure drop curve in bag I straightened out when the air-to-cloth ratio reached 225 mm/s, beyond that, increase in pressure drop would not allow an increase in the air-to-cloth to ratio. In bag II it is feasible to go up to 250 mm/s and because of high air permiability in bag III, air-to-cloth ratio up to 400 mm/s can be achieved. However, this will result in poor efficiency due to larger pore spaces as fine dust can penetrate through the filter material easily.

Figure 4.11 shows a linear relationship between bag pressure drop and collection efficiency for bag I and II while Figure 4.12 displays a similar relationship for bag III. These figures indicate that, as the bag pressure was increased, the collection efficiency decreased. The efficiency decreased because dust penetration would be greater at the greater pressure. Bakke (1979) observed a similer relationship while testing wool felt with a pulse-jet test and using talc powder.

4.7.3 Collection Efficiency

Dust collection efficiencies of filter bags are illustrated in Table 4.9. Bags I and II were essentially greater than 99% efficient. There was little effect of the air-to-cloth ratio on dust collection efficiencies in bags I and II. Only finer fractions of dust penetrated through the bags. However, the air-to-cloth ratio greater than 237.10 mm/s resulted in dust collection efficiencies between 72.8 to 90.8% in bag III. The reason for this low collection efficiency was due to the

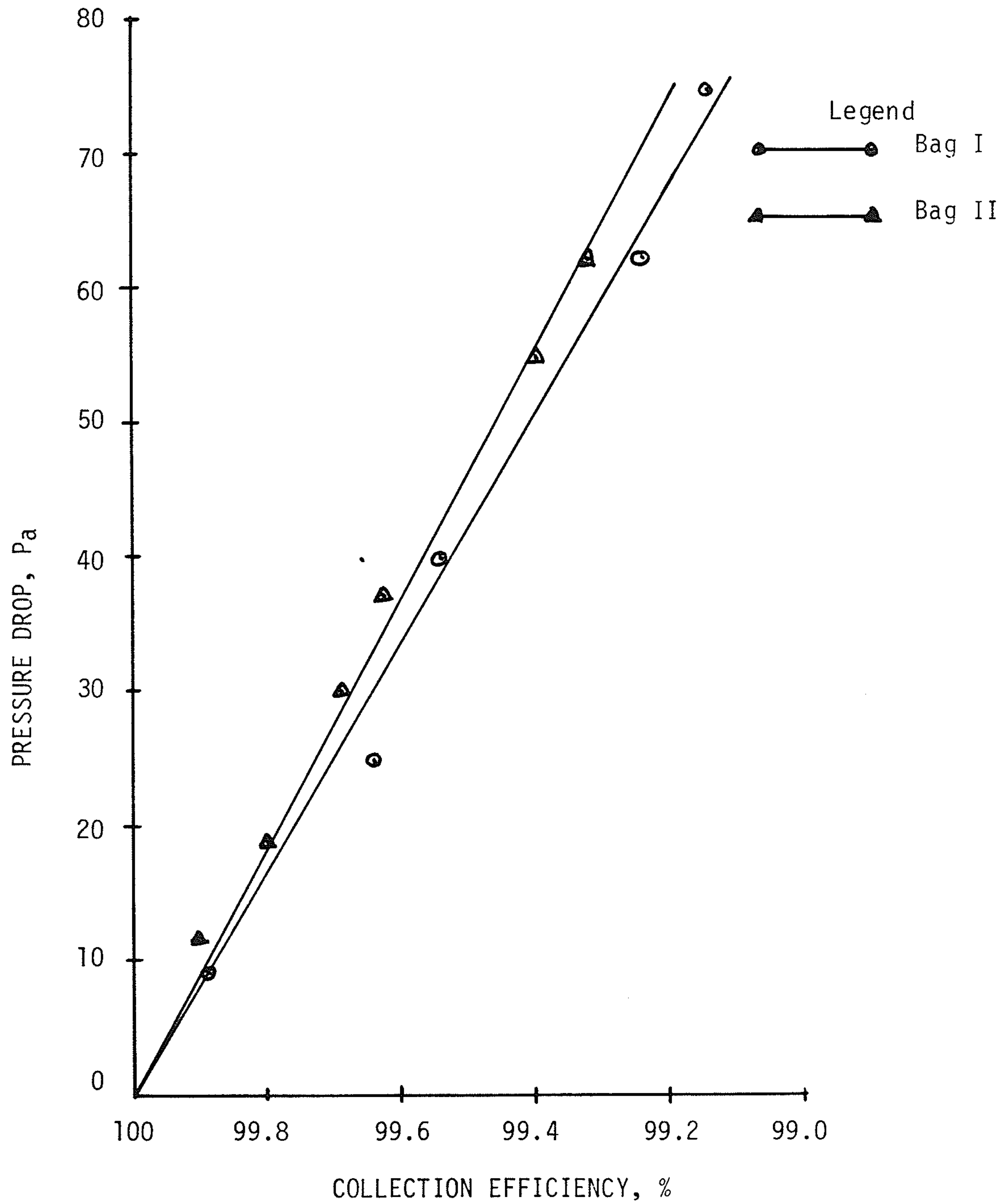


Figure 4.11 Bag pressure drop versus dust collection efficiency

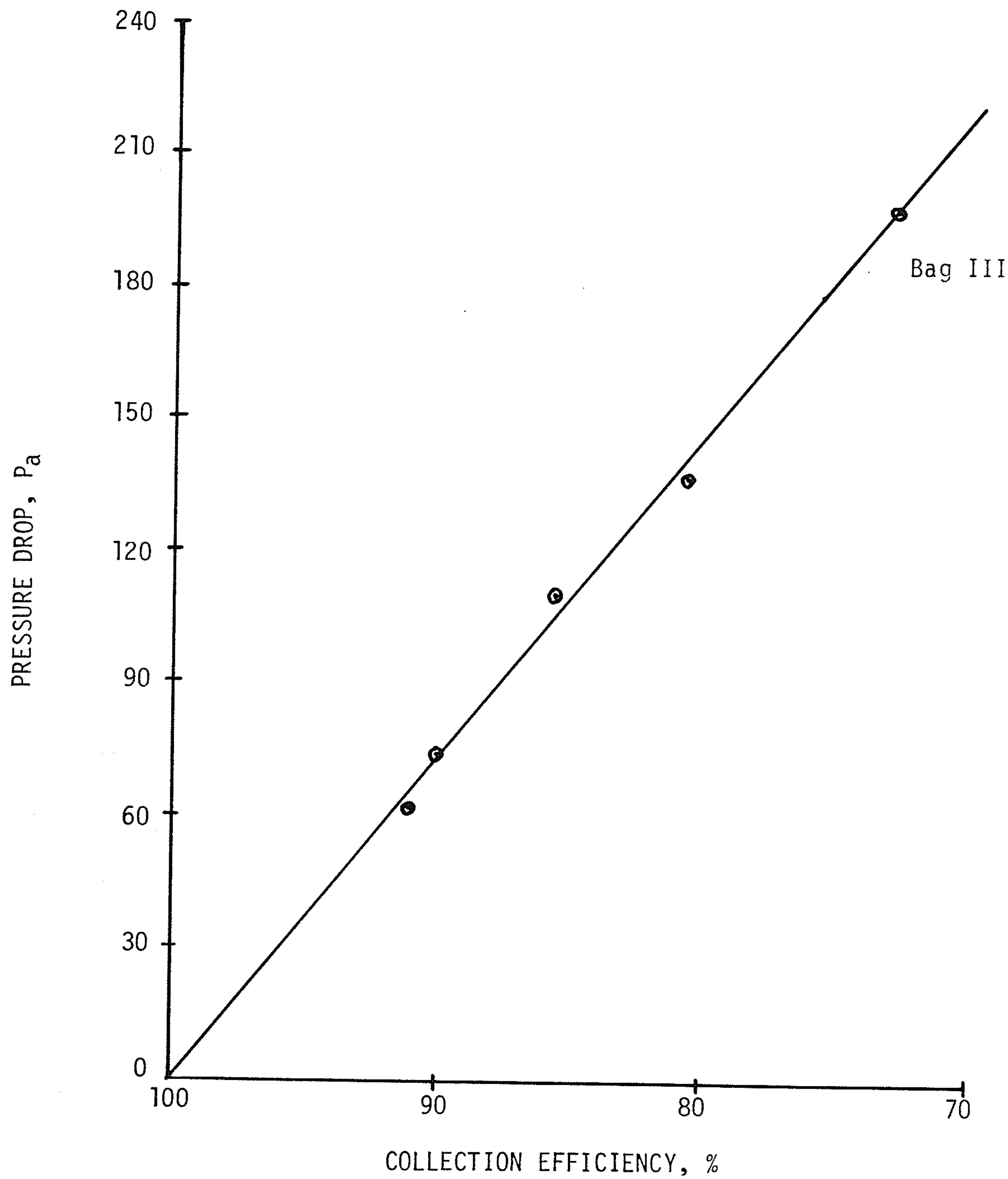


Figure 4.12 Bag pressure drop versus collection efficiency

Table 4.9 Collection efficiencies of fabric filter bags

Bag I		Bag II		Bag III	
Air-to-cloth ratio mm/s	Efficiency %	Air-to-cloth ratio mm/s	Efficiency %	Air-to-cloth ratio mm/s	Efficiency %
74.17	99.88	100.08	99.81	237.10	90.80
134.62	99.80	150.37	99.69	246.38	90.00
164.50	99.55	172.72	99.66	301.85	85.50
183.34	99.34	194.54	99.40	338.58	80.60
203.70	99.15	203.20	99.32	392.33	72.80

Table 4.10 Dust emission concentrations from fabric filter bags

Air-to-cloth ratio mm/s	Dust concent. mg/m^3	Air-to-cloth ratio mm/s	Dust concent. mg/m^3	Air-to-cloth ratio mm/s	Dust concent. mg/m^3
74.17	0.21	100.08	0.24	237.10	5.49
134.62	0.27	150.37	0.24	246.38	5.54
164.50	0.28	172.72	0.25	301.85	6.16
183.34	0.50	194.54	0.43	338.58	8.08
203.70	0.50	203.20	0.48	392.33	9.84

larger porosity in the filter material. The larger porosity reduced re-entrainment of dust particles in the bag and hence resulted in higher dust penetration or lower collection efficiency. The collection efficiencies reported in Table 4.10 were measured when the filters were relatively new. Furthermore, the dust removal efficiency will further improve after the bags are used as entrainment of larger dust particles in the interstices will prevent penetration of finer dust particles.

4.7.4 Air-to-Cloth Ratio

The air-to-cloth ratio or surface velocity is an important design variable in fabric filters. The bag pressure drop was highly dependent on air-to-cloth ratio (Figure 4.11) and increased with the increase in air-to-cloth ratio; however the relationship was nonlinear. The air-to-cloth ratio also affected the dust collection efficiencies of the bags especially in bag III. The recommended air-to-cloth ratio or face velocities for the fabric should not be exceeded otherwise higher pressure drops and lower collection efficiencies would result. Air permiabilities of the bags are reported in Table 4.9 and the recommended face velocities for bag I, II and III would be 21.8, 76.8 and 288.2 mm/s, respectively.

4.7.5 Exhaust Dust Concentration and Particle Size Distribution

The dust concentrations from bags I and II were far below the acceptable limit of 10 mg/m^3 (Table 4.10). Although bag III was also within the acceptable limit, it was not advisable to operate the bag near the threshold limit value of dust emission. Bags I and II were very effective in the control of dust, and the emissions from each bag were approximately equal even though bag I operated at an air-to-cloth ratio higher than the recommended value of 21.8 mm/s. The air-to-cloth ratio also affected the dust penetration through the bags. For instance, in bag III the dust concentration increased from 5.49 to 9.84 mg/m^3 when the air-to-cloth ratio increased from 237.1 to 392.33 mm/s.

Figures 4.13 a and 4.13 b display particle size mass distribution curves of grain dust emitted through the filter bags. These figures show that the particle size distribution function followed a log-normal distribution. The mass median diameter and geometric standard deviation were highest in bag III (Figure 4.13 b) followed by bag I and II. This indicates that the dust particles larger than 5 μm also penetrated through bag III. In bag II 50% of particles penetrating through the filter were less than 2.4 μm in size. The mass median diameter from bag I was slightly larger than bag II, and this may be because bag I was operated at an air-to-cloth ratio higher than its recommended value. However, the dust concentrations in both the bags were less than 0.5 mg/m^3.

4.8 Air Recirculating System

4.8.1 Collection Efficiency of the Cyclone and Filter Bag

The combined dust collection efficiencies of a cyclone separator and filter bag I were greater than 98.6% (Table 4.11). The cyclone efficiency improved as the air flow rate increased however, the combined efficiency decreased. This occurrence should happen because the large diameter particles were separated by the cyclone separator while small diameter particles penetrated through the filter material. Nevertheless, this combination still maintained the dust concentration well below the acceptable limits of 10 mg/m^3.

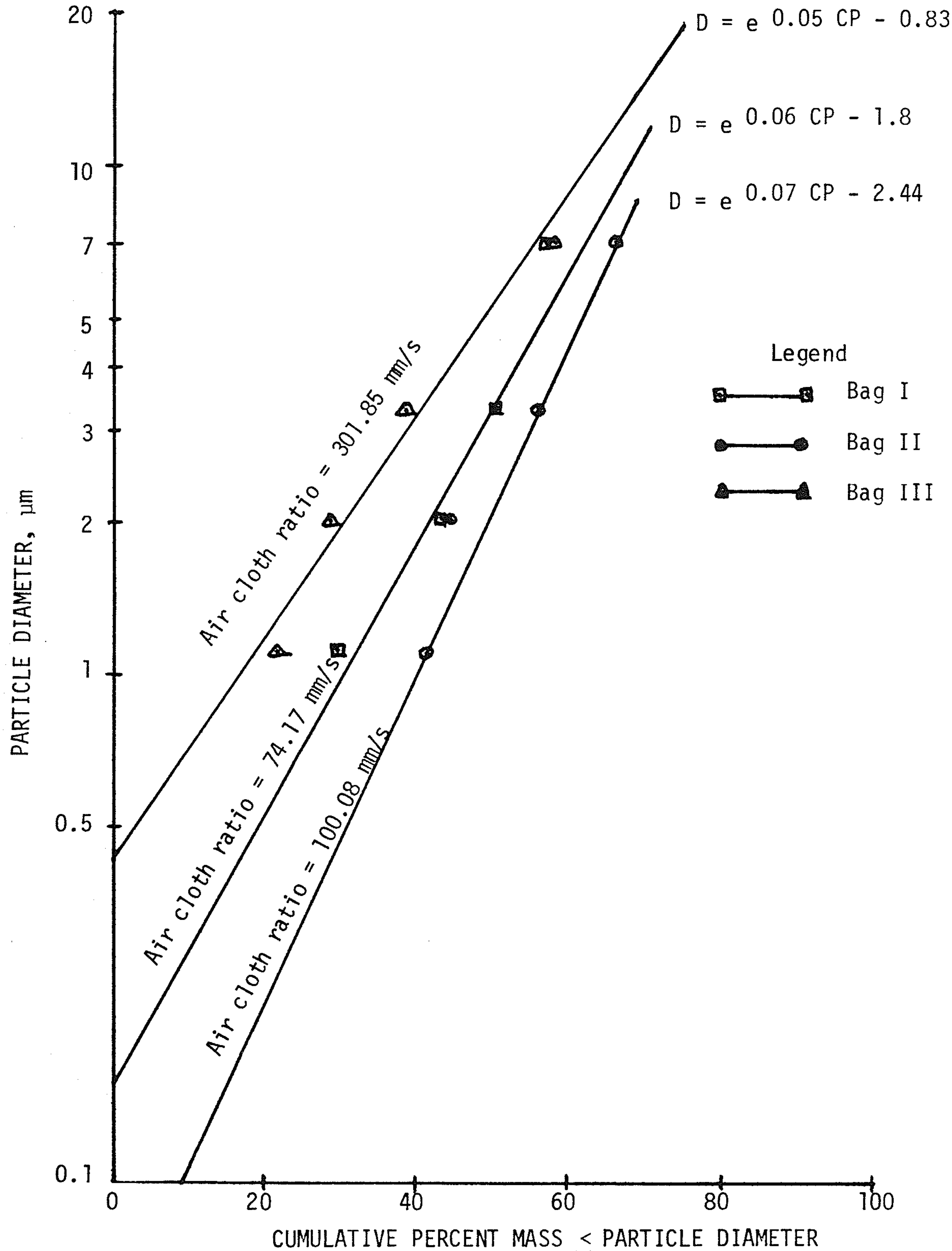


Figure 4.13 a Particle size distribution of dust emitted through the fabric filter bags

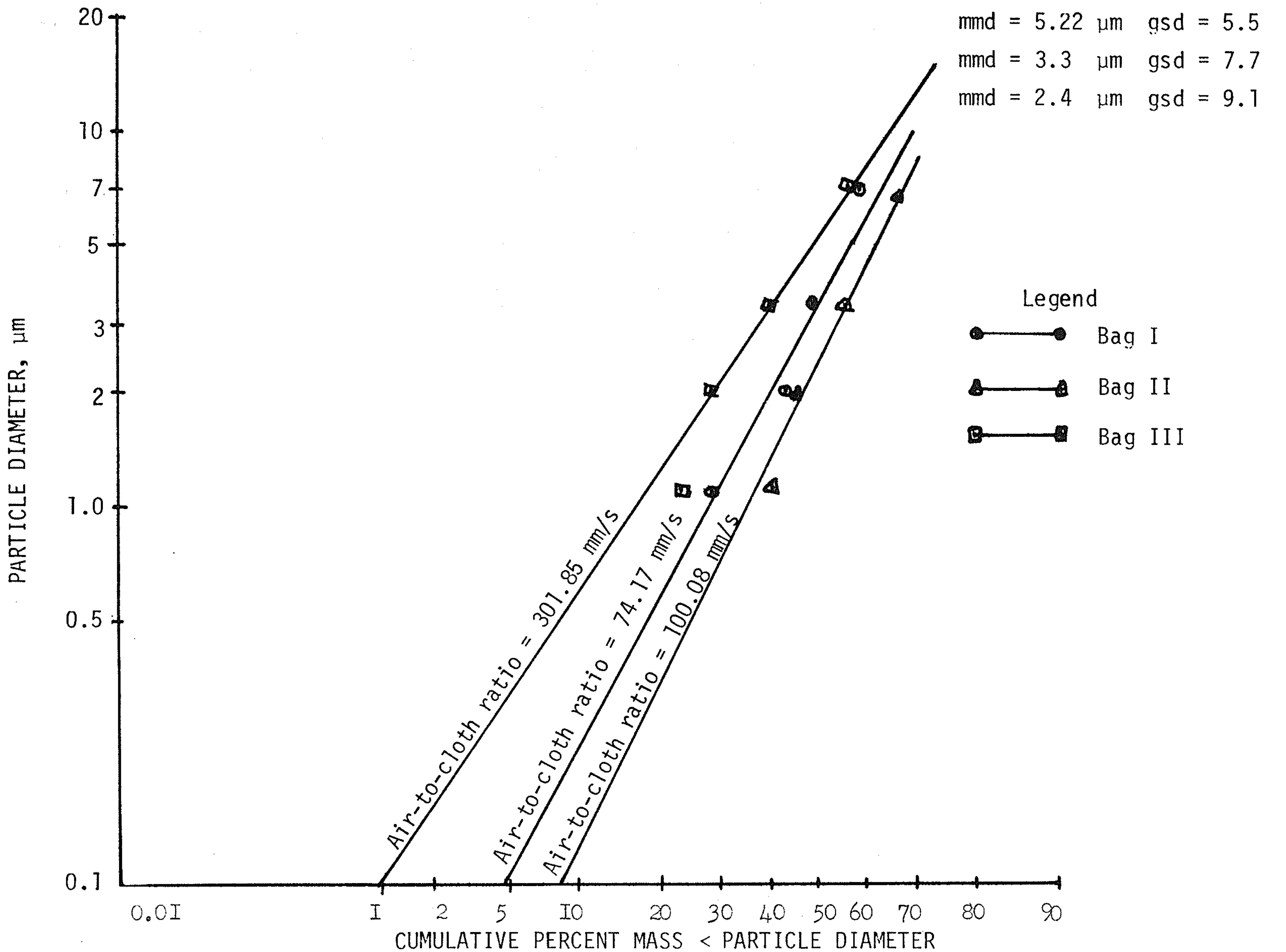


Figure 4.13 b Particle size distribution of dust particles emitted through the fabric filter bags

Table 4.11 Collection efficiencies of cyclone separater and fabric filter

Air flow rate L/s	Collection efficiency %	
	Cyclone Separater	Cyclone and Filter Bag I
167.5	73.72	99.84
204.15	74.22	99.86
261.8	81.10	99.89
293.61	81.63	98.71
331.46	82.74	98.65

Table 4.12 Dust concentrations in air recirculating type system

Location	Dust Concentration mg/m^3 Test Number				
	1	2	3	Mean	S.D.*
Filter bags	0.68	0.28	0.25	0.40	0.24
Cyclone and bags	0.65	0.43	0.27	0.45	0.19
Dumping hopper	0.95	0.91	0.90	0.92	0.03
In building environment	0.24	0.21	0.18	0.21	0.03

*Standard deviation

4.8.2 Exhaust Dust Concentration

The exhaust dust concentrations of the air recirculating system were measured (Table 4.12) at four different locations. The largest dust concentration of 0.92 mg/m^3 was measured near the dumping hopper while the smallest concentration of 0.21 mg/m^3 was obtained in the building environment. Dust concentrations were slightly greater than the cyclone and bag I were used in combination than when the filter bag was used alone. The reason was because the larger particles were collected by the cyclone separator and the re-entrainment of the smaller dust particles in the filter material was reduced. Based on Table 4.12 the air recirculating system using a cyclone separator and filter bags could be adopted for dust control provided that the filter is monitored and serviced frequently.

5. CONCLUSIONS

The following conclusions are drawn from the data results:

1. The amount of dust generated in the seed cleaning plants depended on the type of seed grain cleaned and the sampling location.

2. Four out of eleven farm seed cleaning plants surveyed had air pollution problems with dust levels reaching up to 114.23 mg/m^3.

3. The particle size distribution of grain dust followed a log-normal distribution.

4. The frictional head losses (Pa/m) in galvanized sheet metal and flexible plastic pipes were approximately proportional to 2.2 and 1.9 power of the air flow rates (L/s) respectively.

5. The highest frictional head losses were obtained in sewn vinyl ribbon pipes followed by the corrugated plastic pipe and galvanized sheet metal pipes.

6. The frictional head losses in flexible pipes were 2 to 2.5 times higher than the losses in galvanized sheet metal pipes.

7. The frictional head losses in terms of equivalent length of straight pipe, in the flexible pipe bends which depend on the centreline radius of the bend and diameter of the duct and were almost equal to the losses in sheet metal elbows of the same size.

8. Both the semilateral and the partially enclosed hoods reduced the dust concentrations from 250.9 mg/m^3 to about 1 mg/m^3 at the dumping hopper. These values were well below the allowable limit.

9. The type of construction material and style of hood affected the hood entry loss. The largest losses resulted from the wooden hood, followed by the polyethylene and galvanized sheet metal hoods, repsectively.

10. The dust pick-up at the dumping hopper was reduced considerably when the hood distance from the dumping source was increased from 280 to 560 mm with three sides of the hood open.

11. During the operation of the experimental dust removal system, the air flow rate, fan power, and fan rpm were 1.75 kPa, 390 L/s, 1.16 kW and 1800 rpm, respectively.

12. Pressure drop across the cyclone separator varied with the air flow rate and was proportional to the 2.13 power of the air flow rate.

13. A maximum dust collection efficiency of 82.6% was achieved with the test cyclone separator. It increased with the increased air flow rate.

14. The dust collection efficiencies for filter bags I and II were larger than 99% and depended on the air to cloth ratios. The collection efficiency bas III ranged from 72.8 to 90.8%.

15. Bag pressure drop varied non-linearly with the air to cloth ratio. That is, bag pressure drop increased with an increase in the air-to-cloth ratio. For undersized bags, an increase in fan speed would not result in an increase in air flow.

16. The mass median diameters of grain dust that penetrated through bags I, II and III were 3.3, 2.4 and 5.22 μm, respectively.

17. Dust collection efficiencies for the cyclone separator and filter bag I combination were slightly lower than filter bag I and II.

6. RECOMMENDATIONS

The following recommendations are made on the basis of this study.

1. About 60% of the farm seed cleaning plants visited used a cyclone separator as a dust control device, while the remaining plants did not have any dust control facilities. The outlet of the cyclone separator in all the plants was outside the building so that the dust, which is not collected by the cyclone was exhausted to the outside air. In order to reduce the air pollution problems in the farm seed cleaning plants, either a cyclone separator in conjunction with a fabric filter or filters should be used.

2. At the higher dust concentrations, the dust collected on the first stage of the Andersen head tended not to follow the log-normal distribution. Therefore, for particle size analysis under these conditions the results of the Andersen head should be compared with other methods.

3. Although the frictional head losses in the flexible pipes were higher than sheet metal pipes their use as dust collection ducts in the farm seed cleaning plants could be used in awkward installations.

4. Partially enclosed hood types should be utilized to control dust emissions at dumping hoppers.

5. An air recirculating dust control system may be used for energy conservation provided that the filter is monitored and serviced frequently.

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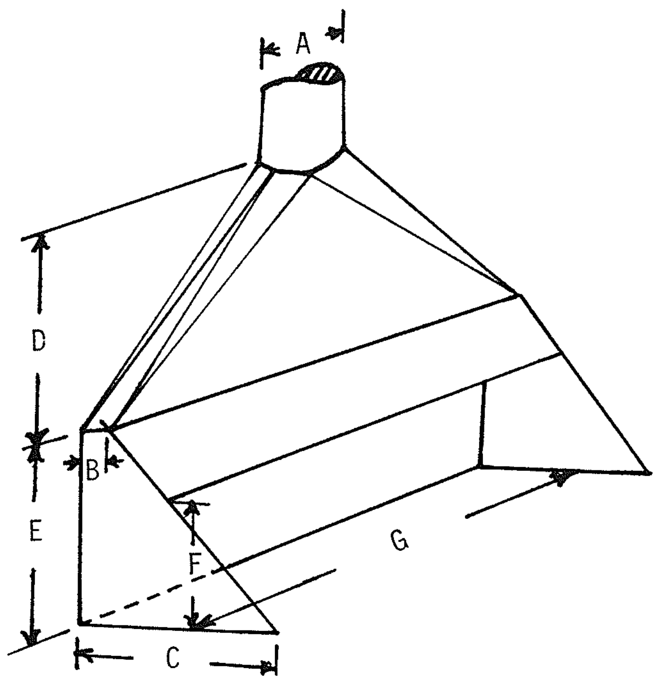
APPENDIX I

Specifications of the Pitot-Static Tube

Make:	Air Flow Developments Ltd. High Wycombe, Bucks, England,	
Model:	New N.P.L. modified ellipsoidal nose form.	
Size:	Overall length	= 0.483 m
	Tube diameter	= 8 mm
	Head hole diameter	= 1 mm
	Static hole diameter	= 1 mm
	No. of static holes	= 6
	Catalogue No.	= 7013501
Material of Construction:	Stainless steel with silver brazed joints.	
Other features:	Rounded heel for conveniently passing through a hole in a duct wall and is provided with a direction pointer at the connecting end. The Pitot-Static tube can be used for flow and pressure measurements in gases up to 690°C. Glands can be provided for permanently installing the Pitot-Static tube in the duct.	

APPENDIX II

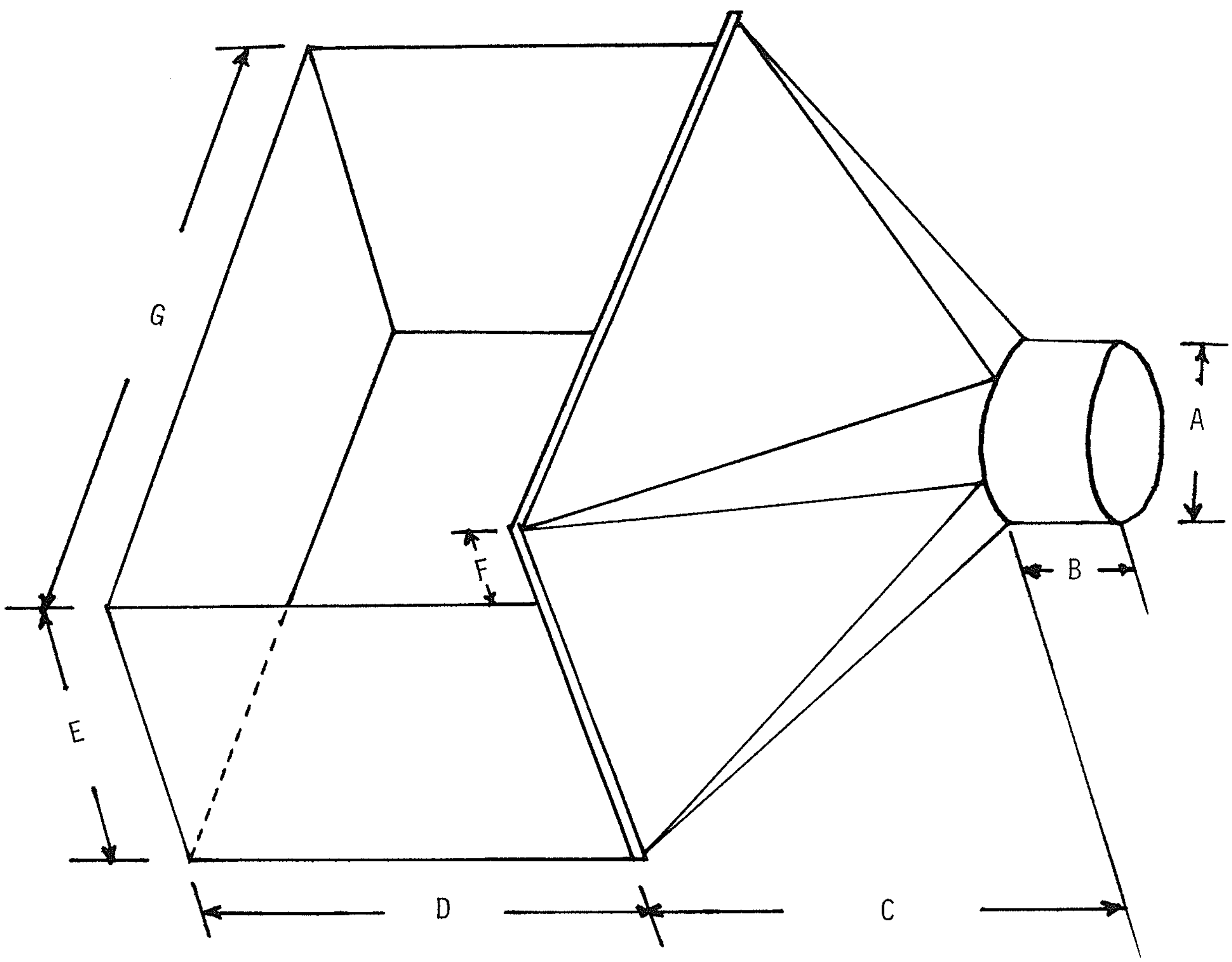
Detailed dimensions of hoods for dumping hopper



Key to the dimensions:

A = 101.6 mm
B = 50.8 mm
C = 390.57 mm
D = 457.20 mm
E = 304.80 mm
F = 203.20 mm
G = 914.40 mm

Semi-lateral type of exhaust hood



Key to the dimensions: A = 101.60 mm

B = 152.40 mm

C = 254.00 mm

D = 457.20 mm

E = 304.80 mm

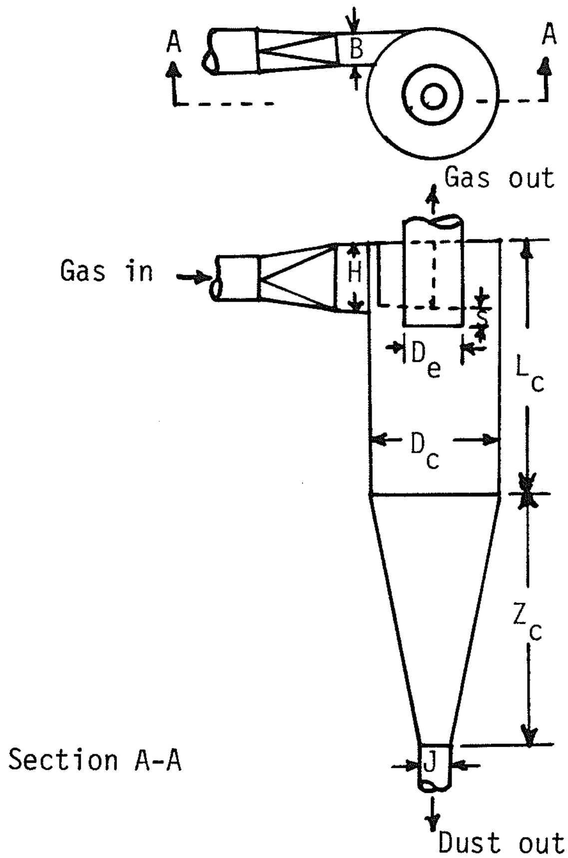
F = 101.60 mm

G = 685.80 mm

Partially enclosed side-draft hood

APPENDIX III

Dimensions of a typical test cyclone separater



Key to the dimensions:

B = 152.40 mm modified to 76.20 mm

D_c = 428.50 mm

D_e = 304.80 mm

H = 304.80 mm

L_c = 660.40 mm

S = 50.80 mm

Z_c = 990.60 mm

J = 101.60 mm

APPENDIX IV

Data for frictional head losses in sheet metal pipes and flexible plastic pipes

Table IV.1 Friction loss data in 203.2 mm ϕ^{*}, 3.05 m sheet metal pipe.

Test No.	Fan Static Pressure mm W.C.	Temperature °C dry bulb	wet bulb	Barometric Pressure kPa	Average Velocity head mm W.C.	Average Head loss mm W.C.
1.	172.72	22.5	17.5	98.0	46.27	24.76
2.	134.62	22.0	15.55	97.99	31.42	18.41
3.	109.22	22.22	15.55	98.0	26.20	11.43
4.	68.58	21.40	17.78	97.96	14.93	6.35
5.	50.80	23.33	20.55	97.72	9.32	4.06

Table IV.2 Friction loss data in 127 mm ϕ^{*}, 2.62 m sheet metal pipe.

Test No.	Fan Static Pressure mm W.C.	Temperature °C dry bulb	wet bulb	Barometric Pressure kPa	Average Velocity head mm W.C.	Average Head loss mm W.C.
1.	73.66	20.55	15.0	98.29	2.92	15.24
2.	88.90	21.10	15.55	98.29	3.95	18.41
3.	177.80	20.0	16.67	98.31	8.40	33.02
4.	205.74	19.44	15.0	98.29	10.34	46.99
5.	254.0	20.0	15.55	98.3	12.88	60.96
6.	325.12	20.55	16.67	98.3	16.66	81.28

*Diameter

Table IV.3 Friction loss data in 101.6 mm ϕ^*,1.75 m sheet metal pipe.

Test No.	Fan Static Pressure mm W.C.	Temperature °C dry bulb	wet bulb	Barometric Pressure kPa	Average Velocity Head mm W.C.	Average Head loss mm W.C.
1.	355.60	20.55	16.67	98.03	8.27	85.72
2.	269.24	22.22	17.22	98.03	5.15	64.13
3.	208.28	23.89	18.90	97.99	4.30	49.53
4.	149.86	22.78	18.90	98.00	2.90	20.32
5.	93.98	24.44	19.44	97.99	1.83	16.51
6.	71.12	25.0	19.44	97.96	1.38	12.70

Table IV.4 Friction loss data in 76.2 mm ϕ^*,3.25 m sheet metal pipe.

Test No.	Fan Static Pressure mm W.C.	Temperature °C dry bulb	wet bulb	Barometric Pressure kPa	Average Velocity Head mm W.C.	Average Head loss mm W.C.
1.	71.12	20.0	17.78	97.32	0.67	40.00
2.	96.52	22.22	17.78	97.26	0.66	49.53
3.	142.24	21.10	18.33	97.30	0.95	59.69
4.	162.56	22.5	18.90	97.26	0.95	75.18
5.	210.82	22.22	16.67	97.30	1.27	91.44
6.	294.64	20.55	17.78	97.87	1.58	106.68

*Diameter

Table IV.5 Friction loss data in 203.2 mm ϕ^*, 5.03 m, corrugated plastic pipe

Test No.	Fan Static Pressure mm W.C.	Temperature °C dry bulb	wet bulb	Barometric Pressure kPa	Average Velocity Head mm W. C.	Average Head loss mm W.C.
1.	58.42	23.00	15.55	98.62	5.64	15.24
2.	78.74	23.00	16.10	98.62	9.9	22.22
3.	132.08	22.22	16.10	98.62	18.73	41.91
4.	185.82	22.78	15.55	98.62	28.64	64.77
5.	231.14	23.33	16.10	98.60	38.28	90.17

Table IV.6 Friction loss data in 152.4 mm ϕ^*, 5.4 m corrugated plastic pipe

Test No.	Fan Static Pressure mm W.C.	Temperature °C dry bulb	wet bulb	Barometric Pressure kPa	Average Velocity Head mm W. C.	Average Head loss mm W.C.
1.	55.88	23.0	18.6	97.56	3.96	22.22
2.	91.44	21.67	17.2	98.1	5.82	32.38
3.	160.02	21.94	17.78	98.1	10.97	60.96
4.	195.58	22.5	17.78	98.12	14.37	79.50
5.	256.54	22.5	18.33	98.12	19.37	107.95

*Diameter

Table IV.7 Friction loss data in 101.6 mm ϕ^*, 5.385 m sewn vinyl ribbon pipe.

Test No.	Fan Static Pressure mm W.C.	Temperature °C dry bulb	wet bulb	Barometric Pressure kPa	Average Velocity Head mm W.C.	Average Head loss mm W.C.
1.	360.68	23.0	21.1	97.5	2.37	208.91
2.	236.22	23.33	21.1	97.51	1.85	132.10
3.	172.72	23.89	21.4	97.52	1.60	93.98
4.	107.95	22.78	21.1	97.51	0.78	56.51
5.	71.12	23.89	21.67	97.49	0.47	38.10

Table IV.8 Friction loss data in 76.2 mm ϕ^*, 3.84 m sewn vinyl ribbon pipe.

Test No.	Fan Static Pressure mm W.C.	Temperature °C dry bulb	wet bulb	Barometric Pressure kPa	Average Velocity Head mm W. C.	Average Head loss mm W. C.
1.	297.18	23.33	20.00	97.93	0.97	152.4
2.	222.25	23.89	20.00	97.93	0.73	114.30
3.	179.07	24.17	20.3	97.98	0.56	91.44
4.	109.22	23.89	20.55	98.00	0.32	54.61
5.	74.93	23.89	20.55	98.04	0.15	35.98

*Diameter

APPENDIX V

Data for dust particle size analysis

Table V.1 Dust particle size distribution data for plant 4.

Stage	ECD µm	Mass of dust mg	Percentage %	Cumulat. %	Cumulative % Particle Diameter
1	7.0	1212.0	75.77	75.77	24.23
2	3.3	199.5	12.47	88.24	11.76
3	2.0	60.0	3.75	91.99	8.01
4	1.1	38.0	2.38	94.37	5.63
5	-1.1	90.0	5.63	100.0	-

Table V.2 Dust particle size distribution data for plant 8.

Stage	EDC µm	Mass of dust mg	Percentage %	Cumulat. %	Cumulative % Particle Diameter
1	7.0	294.00	66.84	66.84	33.11
2	3.3	78.0	17.75	84.64	15.36
3	2.0	49.50	11.26	95.90	4.10
4	1.1	6.0	1.366	97.27	2.73
5	-1.1	12.0	2.73	100.00	-

Table V.3 Dust particle size distribution data for plant 10.

Stage	ECD µm	Mass of dust mg	Percentage %	Cumulat. %	Cumulative % Particle Diameter
1	7.0	191.50	70.67	70.67	29.33
2	3.3	34.5	14.53	85.20	14.80
3	2.0	26.0	12.53	97.73	2.27
4	1.1	4.5	1.89	99.62	0.38
5	-1.1	0.9	0.38	100.00	-

Table V.4 Dust particle size distribution data for Bag I at air-to-cloth ratio of 74.17 mm/s.

Stage	ECD µm	Mass of dust mg	Percentage %	Cumulat. %	Cumulative % Particle Diameter
1	7.0	3.0	42.85	42.85	57.15
2	3.3	0.5	7.14	49.99	50.01
3	2.0	0.5	7.14	57.13	42.87
4	1.1	1.0	14.29	71.42	28.58
5	-1.1	2.0	28.58	100.00	-

Table V.5 Dust particle size distribution data for Bag II at air-to-cloth ratio of 100.08 mm/s.

Stage	ECD µm	Mass of dust mg	Percentage %	Cumulat. %	Cumulative % Particle Diameter
1	7.0	10.0	34.01	34.01	65.99
2	3.3	3.0	10.20	44.21	55.79
3	2.0	3.5	11.90	56.11	43.89
4	1.1	0.9	3.06	59.17	40.83
5	-1.1	12.0	40.82	99.99	-

Table V.6 Dust particle size distribution data for Bag III at air-to-cloth ratio of 301.85 mm/s.

Stage	ECD µm	Mass of dust mg	Percentage %	Cumulat. %	Cumulative % Particle Diameter
1	7.0	40.5	42.63	42.63	57.37
2	3.3	18.0	18.95	61.58	38.42
3	2.0	9.5	10.0	71.58	28.42
4	1.1	6.5	6.84	78.42	21.58
5	-1.1	20.5	21.58	100.00	-