MODELLING OF DUCTED VENTILATION SYSTEM IN AGRICULTURAL STRUCTURES

by

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ABSTRACT

Ventilation is a necessary part of the environmental control system for all structures which are used for housing livestocks and storing plant materials. The object of ventilation is to remove the stale air of a building, and replace it with fresh air thereby removing the danger of toxic gases.

Air distribution ducts are used in the environmental control of livestock and poultry building as well as the conditioning of most agricultural produce.

Balanced flow conditions for ventilation ducts have been well defined, but very little attention has been given to unbalanced flow condition. ASHRAE (1985) stipulates that calculations in terms of non-equal flow rates are impractical and merit no attention. Actually, some agricultural applications work better with unbalanced flow conditions. Unbalanced air distribution, from ventilation ducts, can save energy and provide for a more effective use of the system.

The uniformity of air distribution relies heavily upon the design of the duct in consideration of its length, cross-sectional area, total outlet area and total air displacement. In order to simplify the approach to the design of ventilation ducts, mathematical equation has been derived to describe the average air velocity of duct.

The primary objective of the research work was to test goodness of fit of an equation describing the average air velocity of perforated ventilation ducts,

under balanced as well as unbalanced air distribution:

$$V = H_o \frac{X}{L} + (V_L - H_o) \frac{X^2}{L^2}$$

This equation was successfully tested using data measured from 14 ducts of constant cross sectional area, built of wood or polyethylene with outlet of various shape, and aperture ratio. Results indicated that aperture ratio and distance along the duct are the two most significant factors influencing the average duct air velocity values, but material and outlet shape had little effect.

RESUME

La ventilation constitue une partie importante du système de systeme de contrôle environnemental de toutes les structures qui sont utilisees pour abriter les animaux et entreposer les plantes. Le but de la ventilation est d'expulser l'air malsain du batiment et de le remplacer par de l'air frais, contribuant ainsi à améliorer le confort et à éliminer le danger d'intoxication.

Les conduits de distribution d'air sont utilisés pour le contrôle de l'environment des batiments d'élevage et des poulaillers aussi bien que pour le conditionnement de tous les produits agricoles. Des conditions d'écoulement uniforme, pour les conduits de ventilation, ont été très bien définies, mais très peu d'attention a été donnée aux conditions d'écoulement non-uniforme. ASHRAE (1985) stipule que les calculs concernant les débits non-égaux ne sont pas pratiques et par conséquent ne méritent pas d'être pris en considération. Cependant certaines applications agricoles donnent de meilleurs resultats avec des conditions de débit non-uniforme.

Une distribution d'air non-uniforme, dans des conduits de ventilation, permet d'économiser de l'énergie et contribue à une utilisation efficace du système.

L'uniformité de la distribution de l'air dépend énormément du design du conduit de ventilation tenant compte de sa longueur, de sa surface, de la surface totale des sorties et du déplacement total de l'air.

Dans le but de simplifier le design des conduits de ventilation, une

équation mathématique a été derivée pour décrire la vitesse moyenne de l'air dans un conduit.

L'objectif principal de cette recherche a été de vérifier l'exactitude d'une équation décrivant la vitesse moyenne de l'air des conduits de ventilation perforés en condition de distribution d'air uniforme et non-uniforme.

$$V = Ho\frac{X}{L} + (V_L - Ho)\frac{X^2}{L^2}$$

Cette équation a été testée avec succès, utilisant des données mesurées à partir de 14 conduits de ventilation, construits avec du bois ou du polyéthylène avec des sorties de différentes formes et des ratio d'ouverture différents. Les resultats indiquent que le ratio d'ouverture et la distance le long du conduit sont les deux facteurs significatifs influençant les valeurs de la vitesse moyenne de l'air dans le conduit, alors que le matériau et la forme des orifices ont très peu d'effet.

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A - cross-sectional area of the ventilation duct, m^2

D_H - hydraulic diameter of the ventilation duct, 4A/R, m

e - average variation in duct vadis - a measure of duct roughness, mm

F - Darcy-Weisbach friction coefficient, dimensionless

h - height of the outlet, m

Δh - drop in static pressure, Pa

Ho - equivalent duct kinetic potential at $x = 0$, m/s

L - length of the duct, m

n - number of outlet openings along a ventilation duct, dimensionless

P - static pressure inside the duct, measured above that of the atmosphere, Pa

Po - static pressure, P, measured at the back of the duct, Pa

q - air flow over a partial surface, m^3/s

Qo - outlet air flow over a length dx, m^3

Q - fan volumetric displacement, m^3

R - hydraulic perimeyer of the ventilation duct, m

Re - Reynold number, dimensionless

u - outlet air velocity parallel outlet area, m/s

v - duct velocity acting over the duct's portion of its cross-sectional area, m/s

V - duct average cross-sectional velocity, m/s

V_L - duct average cross-section velocity at $X = L$, m/s

Vo - outlet jet velocity, m/s

X - distance along the duct measured from the back, m

θ - aperture ratio, Lh/A, dimensionless

ρ - air density, 1.2 kg/m^3

μ - the coefficient of kinematic viscosity, m^2/s

μ_o - the coefficient of absolute viscosity, m^2/s

Chapter I

INTRODUCTION

" It is unnecessary then to argue at length the importance of a study of ventilation, that may be taken as admitted . . ."

Rafter (1896), a civil engineer.

1.1. Applications of Ventilation System.

Aside from shelter and absence of disease, all living matter depends upon an adequate supply of fresh air. Ventilation is defined as the science or process of maintaining fresh air conditions within a space. (The Engineering Staff of American Blower Corporation and Canadian Cirocco Company LTD, 1935). It consists of supplying or exhausting air to or from any space by natural or mechanical means in order to maintain proper temperatures, and a satisfactory level of purity as well as freshness.

Ventilation is a necessary part of the environmental control system of all livestock and plant structure. The purpose of the ventilation system is to provide an exchange of fresh environmental air based on climatic conditions and the environmental requirements of the biological units in the structure. In any seriously under-ventilated building, the stagnant air gradually becomes warmer and more humid. There also is a rising concentration of dust and particulate matter, ammonia and other gases as well as pathogenic microorganisms the livestock carries. The object of ventilation is, therefore, to remove the stale air in a building, and replace it with fresh air thereby removing the danger of toxic gases. Similarly for greenhouses and storages, the ventilation system is a prerequisite to either the well-being of the plots or to the quality control of the produce (Hellicks and Walker, 1983).

The effective environmental control of most plant and animal facilities involves the introduction of outside air into the space occupied by the plants or animals. In the design of ventilation systems for this purpose, both the quantity and the distribution of the introduced air are important. Ventilation ducts provide convenient means of distributing air uniformly along the length of a building. In practice, inflatable ducts made of plastic film with holes for air discharge, distributed along their length, help improve air distribution. These plastic ducts are usually inflated by a fan of a cross-section similar to that of the duct.

Balanced flow conditions for ventilation ducts have been well defined, but very little attention has been given to unbalanced flow condition. ASHRAE (1985) stipulates that calculations in terms of non-equal flow rates are impractical and merit no attention. Actually, some agricultural applications work better with unbalanced flow conditions. Such conditions are required in greenhouse heating where hot air is distributed by means of a perforated duct extending over the length of the structure. Because of the heat lost over the length of the duct, uniform heat distribution requires the outlet of more air at the end of the duct opposing that of the fan and heating system. In the design of a composting system for manures in storage, gradually added to a pile over a period of several months, unbalanced air distribution provides maturing conditions adjusted to the age of the waste. For produce conditioning and drying, air distribution from ducts could be reduced with time of treatment and along the length of the duct as a function of the degree of conditioning. Unbalanced air distribution, from ventilation ducts, can therefore save energy and provide for a more effective use of the system.

1.2. Research Objectives.

The primary objective of the research work was to test goodness of fit of an equation describing the average air velocity of perforated ventilation ducts, under balanced as well as unbalanced air flow distribution. In order to achieve this primary objective, the duct systems used in the project had to be characterized. This made up the secondary objective of the project.

1.3. Scope.

This research project will be limited to the study of the following duct systems:

i) rectangular, constant cross sectional area wooden ducts.

ii) round polyethylene ducts of constant cross sectional area.

This study will also be limited to the testing of one equation, that describes the average air velocity inside the duct.

But this study will pertain to balanced as well as unbalanced air flow distribution.

Chapter II

LITERATURE REVIEW

The importance of proper ventilation was emphasized by Reid (1844) who, a century ago, wrote :" Mental anxiety may, perhaps, he considered the most powerful enemy to the duration of human life, and, next to it, defective nutriment, whether in quantity or quality. But after these, no other cause, at least in modern times, appears to have inflicted so great an amount of evil upon the human race as defective ventilation..."

2.1.Introduction.

Ventilation system are an essential component of all agricultural building. This system is used to maintain fresh air conditions in livestock shelters, to condition agricultural produce in storage, to maintain adequate ambient conditions in greenhouse facilities and to even treat agricultural wastes to compost them. Perforated ducts can make up part of the ventilation system. When they do, perforated ducts often bring about better distribution of fresh air and improved control of temperatures. Furthermore, their unbalanced (uneven) air distribution could bring about improved ventilation conditions over those produced by balanced air flow. The following sections will provide further insight into agricultural ventilation systems, the lack of information pertaining to uneven flow distribution and the development of an equation describing the average air velocity of ventilation ducts under balanced and unbalanced flow conditions.

2.1.1. Ventilation Systems for Livestock Structures.

Farm livestock are homeotherms, that is, they must keep their body temperature within a moderately narrow range to work efficiently. To do this they maintain a thermal balance between the heat they produce, and the heat they lose to their environment. Heat escapes from the body by a number of routes. A small amount is lost with the faeces and urine but the main losses are by radiation, convection, conduction and evaporation. Radiation heat losses occur because a warm body emits heat when it is at a temperature higher than that of its surrounding surfaces. Convection losses are governed by the surface area of the animal, its temperature and that of the surrounding air, as well as the movement of air over the surface. Evaporation from the skin is dependent, so far as external factors are concerned, on the temperature, humidity and movement of the air. From the aspect of the physiology of the animal, the air humidity has a number of very important influences. The amount of water vapour in the air controls the rate of evaporation of moisture from the internal surfaces of the animal, especially from the lungs and respiratory tract. Therefore, the temperature and humidity are two major factors affecting the animal's health and which can be controlled by the heating and ventilation system (Sainsbury and Sainbury, 1988).

The object of ventilation is also to remove the stale air in livestock building, and replace it with fresh air thereby removing the danger of toxic gases. For human subjects, ventilation is mainly concerned with comfort, for livestock it is also concerned with comfort interpreted through welfare, behaviour and health. The ability to maintain desired environmental conditions in livestock buildings is dependent on the design and performance of the ventilation systems. These systems must provide the correct quantities of air flow in the proper distribution patterns to meet the needs of each application. Selection of a correct ventilation system requires an understanding of the principles involving air flow, physiological responses of livestock, recommendations for ventilation system control and management as well as economic considerations.

Ventilation systems for livestock structures are of two types, natural or mechanical. Natural ventilation systems may be considered for the following types of livestock housing (Esmay, 1978): (a) freestall housing for dairy cattle, (b) bedded pack barns for dairy cattle, beef or sheep, (c) swine growing-finishing building, (d) calf barns and (e) slotted floor beef barns. Mechanical ventilation is generally preferred when it is desirable to maintain a warm environment and in general to produce the following conditions (ASHRAE, 1978): (a) dry floors, dry litter or both, (b) uniform temperature at all animal locations, (c) a minimum of rapid changes and wide fluctuations in environmental temperature and (d) prevention of drafts over the livestock, especially when the incoming air is colder than the existing environment.

Ducts provide a convenient means of distributing air uniformly along the length of a building. In practice, inflatable ducts made of plastic film, with holes for air discharge distributed along their length, provide a cheap and convenient means of air distribution.

There are many ways to ventilate livestock buildings. The ventilation system without ducts may causes excessive air movement in the region which nears the fan or the air inlets but weak air movement in the regions far away from these two components (Fig. 2.1a).

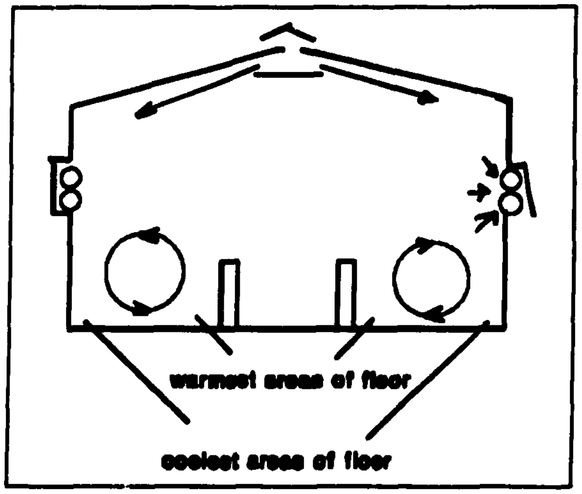

Figure 2.1a Ventilation Condition in Livestock Building without Duct

The perforated ventilation ducts can solve this kind of problems (Fig. 2.1b and Fig. 2.1c). (Clark, 1981).

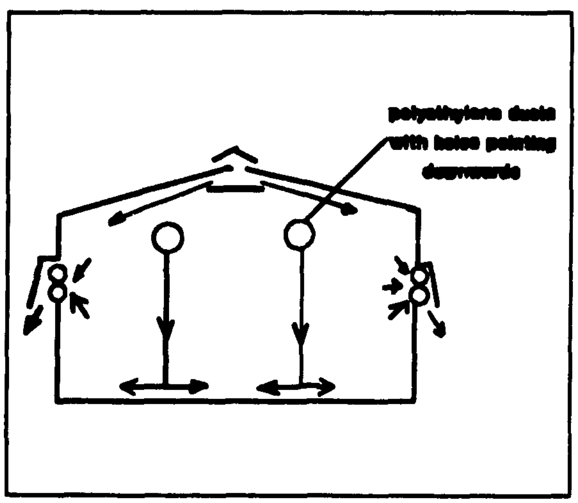

Figure 2.1b Ducted Ventilation System

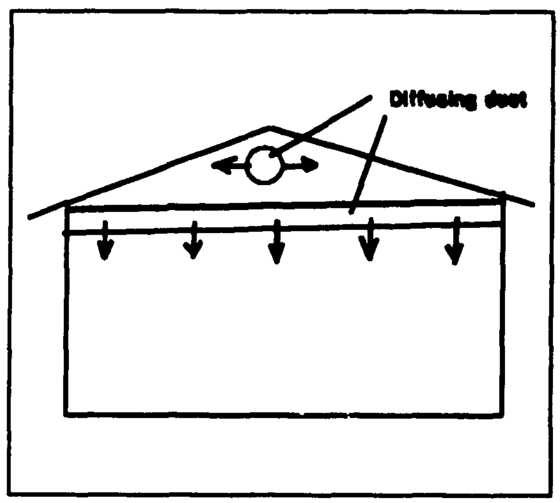

Figure 2.1c Diffusing Ventilation Duct

2.1.2. Ventilation Systems for Greenhouse Structures.

Flowers and vegetables are grown wherever humans have established themselves. In the tropical climates, plants are grown out of doors. Limited quantities grown by amateurs and businessmen alike enter the local sales channels there. The compulsion to purchase floral and green products are greater in temperate and frigid climates, where natural plants are not so abundant and where there is a need to establish ties with nature during the dormant winter season. A few tropical regions have recognized this need and have developed impressive export businesses. In nontropical areas, vast quantities of agricultural products are produced under protected environments (Nelson, 1985).

There are more than 300,000 species and approximately 1,000,000 cultivars of plants known to man. They grow under diverse habitats ranging from the arctic to the tropics and from below sea level to high mountain tops. Day length and radiant energy vary with region and season. Soils vary in origin, composition, and fertility. Temperature varies from one locality to another. Atmospheric composition of CO_2 and air pollutants may vary greatly between urban and rural areas.

During the last three decades we have seen the development of air-conditioned greenhouse, growth chambers, phytotron, and many other types of controlled-environment facilities for plants (Downs and Hellmers, 1975). All these conditions can be controlled artificially within a greenhouse by heating, ventilation and CO_2 producing systems.

The control of high temperatures in greenhouses due to solar heat gain is normally accomplished by ventilating with outside air. When temperatures are desired which are below those that can be achieved by ventilation alone, evaporative cooling pads can be used to reduce the incoming air temperature. Plant water stress is also reduced by increasing the air humidity. Both natural and forced ventilation systems are common and where sufficient air exchange occurs, both perform well. The forced systems are necessary if evaporative cooling is to be used.

The first thought in ventilation is temperature control, and this is proper within limits. It is true that when it is warm in the greenhouse and cold outdoors, warm air will be exchanged for cold air if the vents are opened. The temperature inside will then be reduced. The exchange of air also has an effect on the oxygen, carbon dioxide, and water vapour content of the greenhouse air. These are the critical gaseous components of the air. During the period of the year when heating is required, the grower must regulate heating lines and ventilators so that the correct temperature is maintained uniformly throughout the house. This requires some thought and planning. The grower needs a concept of the temperature trend outdoors. If the outdoor temperature is falling, he will activate heating lines while continuing to vent and then gradually close the ventilators. This practice is used very successfully each evening in the transition period when the heating system takes over from the sun as the source of heat. Not only is it an effective way to keep the greenhouse up to temperature, but some moist air in the greenhouse is replaced with drier air from the exterior. The radiation from the heating lines also keeps the plants a little warmer than the surrounding air. The combined effect is that the plant surfaces remain dry and less susceptible to diseases (Nelson, 1977).

Ventilation systems for plant growing structures, (primarily greenhouses) may be natural or mechanical types. Natural types were used predominantly in glass greenhouses before the advent of low cost electrically powered fans and shutters. With the natural systems, manual or motor operated hinged ventilator panels at the ridge and foundation provide openings for air movement and exchange. Natural ventilation function primate by convective air currants during hot calm periods. It also functions by air movement due to pressure gradients when wind currants against the side wall and over the roof create pressure differentials.

Mechanical ventilation by fans and air control vents, shutters, and louvres is widely used in present day plant growing structures. Fans are used to provide forced air flow through the structure with sufficient internal mixing and circulation to provide uniform environmental conditions.

Ventilation can be handled in the greenhouse with exhaust fans, ventilation tubes and wet pads. The ventilation tubes that are used during cold weather are attached to a heating system or furmance at one end, and are closed the other end. The ventilation tubes are installed overhead, beneath the ridge of the greenhouse, and they are punched or perforated evenly so that the air distributed from them will escape uniformly over the length of the greenhouse. If the shutter at the furmance is open, the tube will fill with hot air. In greenhouse, exhaust fans are only used for humidity control during cold weather. If the heating system is controlled by thermostats, it is possible to handle the temperature level during the winter with a minimum supervision from the grower. During warm weather,the ventilation tubes cannot move sufficient air and for this time of the year, more exhaust fans are used, and the air is introduced into the greenhouse through moist pads. In warm weather it is possible that the exhaust fans may be operated continuously, day and night (Nelson, 1977).

The temperature at which winter ventilation is desired is set on a thermostat which in turn activates two events simultaneously (Fig. 2.2a).

A louvre is opened in a gable end through which cold air enters the plastic tube when an exhaust fan is simultaneously activated. A more popular alternate winter cooling system makes use of the exhaust fan, inlet louvre, and distribution tube. But its distribution tube is separated from the inlet louvre and has a pressurizing fan located in the inlet end of the distribution tube. Cooling

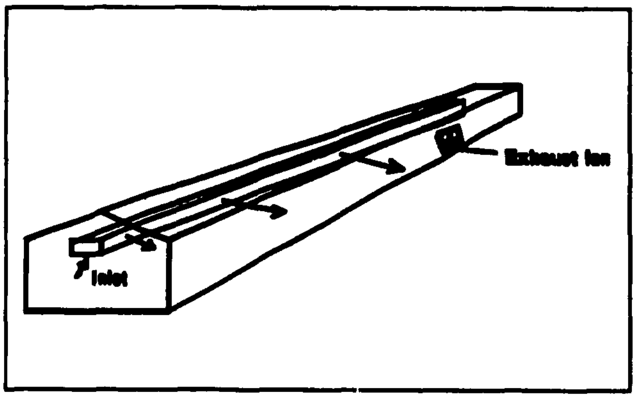

Figure 2.2a A Fan-Tube Winter Cooling System in Greenhouse

and heating are often required during late winter when days are bright and evenings are cold. In either case the same polyethylene tubes are used to distribute the cold air from the outside or the warm air from the heater. One design calls for the unit heater to be attached to the inlet end of the tube, as illustrated in figure 2.2b (Nelson, 1985).

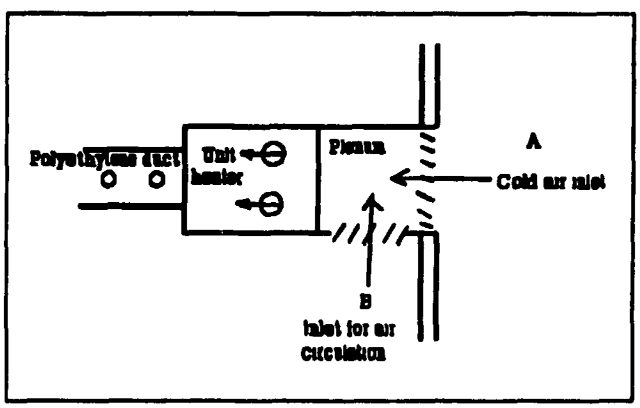

Figure 2.2b An Integrate Winter Coolingand Heating System

Another type of system uses a pressurizing fan in the tube inlet and places a unit heater away from and perpendicular to the inlet of the tube. Completely integrated evaporative cooling-fan tube cooling-heating system are also used for 4 season temperature control (Nelson, 1985).

2.1.3. Ventilation for Horticultural Crop Storage.

Long distance distribution and modern marketing of fresh produce, depends on successful storage and preservation techniques. Storage is viewed here in a broad sense. It may be interim storage before packaging (with or without precooling), long-term cold storage, storage in transportation vehicles or retailing storage on supermarket shelves. No matter where or how the produce is stored, the main objective is invariably to extend the useful life of the product, as determined by its suitability for fresh consumption. Without modern refrigeration and produce preservation techniques, most fruits and vegetables would have to be consumed within a few weeks, or sometimes even days, after harvest. This is unthinkable in today's society of large industrialized urban centres, far removed from agricultural produce growing areas. The refrigerated produce shelves of a modern supermarket may display produce varieties out of season or grown in tropical countries thousands of kilometres away.

Under Canadian weather conditions, vegetable production is limited to one crop per year. For most vegetables, the harvest takes place from August to the end of October, and the produce is usually sold readily after the harvest or stored for short periods of time in conventional refrigerated room. The ideal long-term storage system should have the ability to provide low temperature, high relative humidity and optimum gas composition in order to retard the product's senescence and decay. A product stored under these conditions is expected to maintain its freshness and quality for a much longer period of time than those stored in a conventional refrigerated room. Among the high relative humidity storage systems operating under regular atmosphere, RA, the jacked system is commercially used in Quebec (Gariepy and Raghavan, 1983).

There are many kinds of storages such as naturally ventilated storages, refrigerated storages and controlled atmosphere storages (CA). Ventilated storages at ambient temperatures are particularly suitable for temperate climates. If the storage structure is insulated, appropriate air ducts and automatically controlled fans may be used to pass cool night air through the stored produce, while during the hot part of the day they are shut off. Such a regime is quite often adequate to keep many types of produce at optimal temperatures and relative humilities. Refrigerated storages have closely controlled temperatures and are, by far, the most commonly used method for prolonging the shelf life, of many fruits and vegetables. Suitable fans and ducts circulate the air in the storage room through the evaporator and the stacked produce in the cold room. Thus heat is removed from the produce, passed to the refrigerant in the evaporator coils and expelled to the outside atmosphere or water coolant of the condenser. Controlled atmosphere (CA) storage refers to a system holding the produce in an atmosphere substantially different from regular air. Controlled atmosphere storage is based on removal and/or addition of different gases into the storage room, effectively creating an artificial atmosphere most favourable for significantly prolonging the shelf life of the commodity (Peleg, 1985).

The air distribution system should provide an equal and proportionate amount of air to products in all areas of the building. A fan moving below-freezing outside air into the building through an inlet opening without the benefit of an air distribution duct can cause product freezing near the air inlets and still leave other areas in the building too warm. The air distribution duct may be located on the lower chord of a truss used to support a gable roof, or attached to ceiling joists. An open insulated ceiling allows free air movement from the duct outlets to the sides of the building. Ducts may also be located above a ceiling and discharge below the ceiling using appropriate deflectors. An open ceiling truss building provides the clearance space (Hellickson and Walker, 1983).

To prolong the quality of stored fruits and vegetables it becomes essential to provide a proper storage environment to maintain the life processes. These life processes gradually consume the commodity and eventually all physiological functions will cease. The most feasible solution is to provide an environment to support the minimum rate of chemical reactions to maintain the living commodity, without causing damage or accelerated deterioration to the commodity. This includes the control of temperature, humidity, pressure, carbon dioxide, oxygen and ethylene. To remove the heat from stored fruits and vegetables, it is important that the ventilating air in the storage structure be able to flow through the mass of the produce.

2.1.4. Types of Ventilation system.

The previous systems have clearly identified two main types of ventilation systems. These systems are classified according to the physical forces creating air movement into the building: natural system and mechanical system.

2.1.4.1.Natural Ventilation System.

Natural ventilation is the movement of air through specific building openings resulting from the natural forces produced by temperature differences and wind. The ventilation rate depends on the wind speed and direction, interference of nearby obstructions such as hills or buildings and the size, design and location of outlet and inlet openings. Natural ventilation differs from forced ventilation in that the latter requires a mechanical energy input to produce the pressure differential necessary to cause air flow.

Natural ventilation is the oldest form of ventilation and has been used for as long as housing has been provided for animals. Simplicity, low initial cost and low energy cost are primary factors that make it the most common type of ventilation. However, ventilation that is dependent on natural forces is inherently variable and consequently has numerous limitations. These include such factors as the nature of the climate, geographic area, terrain, obstructions to the wind, environmental requirements and others that must be considered in the design of a natural ventilation system and its subsequent management (Hellickson and Walker, 1983).

2.1.4.2. Mechanical Ventilation System

Mechanical Systems are system in which air is positively moved by means of fans acting as prime movers. In large buildings, ventilation by means of windows is not always practical, and it is necessary to use mechanical methods. A system of ventilation in which the air is impelled into a room is known as a plenum or propulsion system, and when air is withdrawn from the room the installation is known as an extraction or exhaust system. Either system may be used alone, or a ventilation scheme may incorporate both impulsion and extraction (Bedford, 1966).

Experience has convinced most architects and engineers that the natural or gravity system is unreliable because it is entirely dependent upon outside atmospheric conditions. The mechanical system is essential to meet the exact requirements of ventilation because it offers the following advantages:

(1) it provides positive circulation of air at desired temperature and volume.

(2) it is entirely independent of outside atmospheric conditions, where as

natural ventilation system are not.

(3) it can be easily controlled to meet possible variations in requirements.

Perforated ducts are a component of mechanical ventilation system because they are generally fed air by means of a motorized fan. Because of this, natural ventilation systems will be omitted from future discussion.

2.2. Air Distribution Ducts.

The effective environmental control of most agricultural building involves the introduction of outside air into the space occupied by the plants or animals. In the design of ventilating systems for this purpose, both the quantity and the distribution of the introduced air are important. The engineer can use a wide variety of components to achieve the ventilation of the building, such as gravity inlets, perforated ducts, centre roof baffles, and motorized fans.

Since the turn of the century, fan fed ducts have been used for the air-conditioning of public and industrial buildings. These ducts were introduced for the ventilation of agricultural buildings during the 1950's and have gained popularity ever since. Today, duct systems are used in livestock and poultry barns, fruit and vegetable storages, greenhouses as well as grain facilities.

In livestock and poultry buildings, perforated ventilation ducts are found to provide a fresh air distribution superior to that of gravity inlet. This ability is particularly important in eliminating stagnant zones reducing ventilation efficiency (Barber and Ogilvie, 1982). The emitted fresh air jets are also more stable, this stability resulting from the duct's ability to recirculate warm inside air along with the cool fresh air and to maintain an adequate air outlet velocity. The duct jets benefit from stronger buoyancy forces preventing their fall at their entry.

When a duct system is used, the ventilation air is driven into the building, generally by means of a fan, and is distributed through ducts. The system has various advantages. The slight positive pressure set up within the building helps to prevent the leakage of air inwards, and thus cold draughts near doors and windows are to a large extent avoided. If a plenum is used at the inlet of the duct system, the incoming air can be readily warmed by the incorporation of a heating battery. At this point the humidity can be controlled and the entering air can be filtered to remove any suspended matter. The air can be distributed throughout the building as required.

In designing air ducts systems with a plenum, the air cannot be converged as directly as possible as well as at suitable velocities. This serves the purpose of power and material economy. Ducts should be smooth, (ie of metal sheeting,) and as straight as possible. Sharp bends and sudden enlargements or contractions should be avoided. To minimize frictional losses, the ducts should have a minimal surface area and if they are rectangular in shape they should tend towards a square cross section.

2.2.1. The Performance of Ventilation Ducts under balanced conditions.

Uniformity of air distribution is one of the major design preoccupations for perforated ventilation ducts. Research work pertaining to this aspect has used two different approaches: tapering the duct to maintain a constant average duct air velocity; and reducing the aperture ratio while maintaining the duct's cross-sectional area constant. Aperture ratio pertains to the ratio of total outlet area to the cross sectional area of the duct. Parameters affecting the performance of ventilation ducts will be reviewed according to these two approaches.

The tapering of ducts was the first method used by researchers in an attempt to balance air flow. Koestel and Young (1951) explained the tapering effect as one eliminating the static regain phenomenon. Static regain is a process pertaining to the positive static pressure gradient developing inside ventilation ducts with distance away from the fan. Koestel equated the static regain to the average duct velocity head losses in consideration of the Bernoulli's law. By tapering the duct, the average air velocity was maintained constant, thus producing no static regain. Allen (1974) introduced a simulation model built of mathematical series and designed to determine the performance of tapering ducts. Using aperture ratios of 1.0 and 0.5, he obtained balanced conditions by tapering the duct from 450 mm at the fan, to 75 mm at the opposite end.

Balanced condition parameters have also been defined for ducts of constant cross sectional area. These ducts have been preferred over the tapered conduits for manufacturing as well as installation reasons. Haerter (1963) summarized the state of the art on duct flow performance. He recommended the use of a pressure regain coefficient calculated from the ratio of the air velocity within the duct near the outlet to that of the mean duct velocity. He also demonstrated that, for balanced conditions, total pressure differentials ranged from one third to one tenth of the friction losses experienced by their tapered counterparts. Davis et al. (1980) modelled the performance of perforated corrugated metal ducts through the use of mathematical series calculated by computer. Balanced flow conditions were obtained with an aperture ratio between 0.25 and 1.25, for a duct length to hydraulic diameter of 60 to 80, as well as a number of outlets of 80. Perforated plastic ducts were investigated by Carpenter (1972) who demonstrated that balance conditions required an aperture ratio of 1.0 and a minimum duct length of 90 m. Very unstable conditions (negative duct static pressures) prevailed under aperture ratios of 2.0 and more. Also, once balanced, the duct gave uniform flow distribution for a wide range of fan air flow levels. Plastic duct flow was further investigated by Saunders and Albright (1984) who used mathematical series to compute air distribution performance under several conditions. Pressure differentials were assumed to result from friction losses and average air velocity head reductions. Duct stability was found to require an aperture ratio of 1.6 or less. Comparing the performance of these to that of a real duct, Saunders and Albright approximated actual flow distribution within an error level of 10%.

Although Saunders and Albright (1984) did not disscuss the parameters influencing the duct's air distribution, their data does not indicate any significant combination producing even flow distribution over the length of the tube. Nevertheless, Carpenter (1972)'s data consistently show balanced conditions for aperture ratio under 1.06; at this aperture ratio, discharge velocity varies by 15% from one end of the duct to the other.

Brundett and Vermes (1987) further tested perforated polyethylene ventilation tubes by evaluating the swirl effects produced by the fan. They found that higher swirl angles increased the friction factor by 7 to 10%, depending upon the Reynold's number associated with the duct air flow. Furthermore, antiswirl mechanisms helped maintain a high fan flow level in cases where duct diameters has been reduced. Brundett and Vermes obtained uniform flow distribution for both their 457 mm tube (aperture ratio of 1.9) and 610 mm tube (aperture ratio of 1.2) but used a fan which maintained a pressure above 30 Pa for all experimental set ups.

2.2.2. The performance of ventilation ducts under unbalanced condition.

Often, agricultural applications of the air duct require unbalanced air distribution. Such conditions are required in greenhouse heating where hot air is distributed by means of a perforated duct extending over the length of the structure. Because of the losses occurings over the length of the duct, uniform heat distribution requires the outlet of more air at the end of the duct opposing that of the fan and heater. In the design of a composting system for manures in storage, where the waste is gradually added to a pile over a period of several months, unbalanced air distribution can provide maturing conditions adjusted to the age of the waste. For produce conditioning and drying, air distribution from ducts could be reduced with time of treatment along the length of the duct as a function of the degree of conditioning. In livestock building, a more uniform temperature could be developed over the length of the building. Unbalanced air distribution, from ventilation ducts, can therefore save energy and provide for a more effective use of the system.

Unbalanced air distribution from ventilation ducts is a condition which has received very little attention. ASHRAE (1985) stipulates that calculations in terms of non-equal flow rates are impractical and merit no attention. Furthermore, the concept of balanced flow conditions simplifies the analysis of the flow performance of ventilation ducts. As opposed to balanced flow conditions where the air velocity gradient is constant inside the duct, unbalanced conditions produce a non linear gradient over the length of the duct. The solution therefore becomes more complicated.

2.3. Mathematical Modelling of Ventilation Duct Performance.

Air distribution ducts are used in the environmental control of livestock and poultry building as well as the conditioning of most agricultural produce. The uniformity of air distribution relies heavily upon the design of the duct in consideration of its length, cross-sectional area, total outlet area and total air displacement. Although some design criteria have been established through duct performance experimentation, the conception of ducts is still based on balanced flow distribution (ie. a constant velocity gradient for the air inside the duct).

As early as the 1950's, models have been introduced to predict uniform air distribution of ventilation ducts. Koestel and Young (1951) used tapered ducts to eliminate static regain and to improve the uniformities of air distribution. They developed a model accurately predicting static head distribution along the conduit by assuming that the regain in pressure was equal to the loss in duct velocity.

Similar to that of Koestel and Young, but while accounting for friction losses, Shove and Hukill (1963) developed an equation describing the static pressure of the air as a function of length of the duct. Static regain along the duct was equated to the change in velocity pressure head less the friction losses. Although design concepts were presented by Shove and Hukill, the method was not tested against the performance of experimental ducts.

Considering balanced conditions (conditions of uniform duct flow) as well as all outlet momentum losses, Haerter (1963) used a dimensionless pressure recovery factor to model duct static pressure as a function of distance from the fan. This pressure recovery factor was equated to the ratio of the duct's air velocity at the outlet divided by the average duct air velocity. Although Haerter presented a thorough summary of all mathematical models developed to predict ventilation duct flow under balanced conditions, very few design engineers have used these concepts because of difficulties associated with the measurement of this pressure recovery factor.

Purser and Greig (1967) introduced a velocity head coefficient to correct the duct momentum calculated from its average air velocity. Thus, they recognized that the air velocity profile over the cross section of the duct was not constant. They also assumed that this coefficient did not change over the entire duct length even if Haerter (1963) had demonstrated that the duct velocity profile varies with distance away from the fan.

Several researchers used mathematical series, calculated through computer program, to model duct flow conditions. Allen (1974) applied such series to describe the air flow performance of a tapered duct. He considered a regain coefficient of 1.5 to correct the duct air momentum calculated from the average velocity. Allen obtained good correlation between his model and the performance of his experimental ducts although he used aperture ratios of 0.5 and 1.0, conditions generally producing balanced flow. Davis et al. (1980) have also produced an iterative model based on the following assumption: 1) the discharge coefficient is constant over the length of the duct; 2) the outlet flow is solely a function of the static pressure head. The model was successfully tested under conditions of high static pressure with respect to the velocity head or conditions, which tend to produce uniform flow along the length of the duct. Saunders and Albright (1984) produced a model similar to that of Davis et al (1980). but applied to perforated polyethylene ducts and assumed to exhibit a constant outlet discharge coefficient over the length of the system. This model equated static regain to velocity head losses less that of friction. Although this model predicted air flow performances with a 10% error level, it was tested under unusually high static pressure levels.

Chapter III

THE THEORETICAL ANALYSIS

The average air velocity of the duct expressed in terms of its length,x, can be modelled considering an equation applicable to balanced and unbalanced conditions. Balanced conditions are those conditions occurring when the air distribution from the long slot outlet is perfectly uniform over the length, L. This equation is:

$$V = H_o \frac{X}{L} + (V_L - H_o) \frac{X^2}{L^2} \qquad [1]$$

where $Ho = \theta Co\sqrt{2Po/\rho}$

θ	-- aperture ratio, dimensionless
Co	-- discharge coefficient at X=0
Po	-- static pressure at X=0, Pa
ρ	-- air density, Kg/m^3
L	-- length of duct, m
V_L	-- duct average cross-section velocity at X=L, m/s

The following discussion will develop this equation which will be tested for goodness of fit by this research project.

THE THEORETICAL ANALYSIS

Fluid mechanic theories will be used to analyze and describe the flow performance of ventilation ducts fed by a fan. Equation will be developed for a ventilation duct of constant cross sectional area,A, and of length, L. A single long slot allows for the outlet of air over the full length of the duct. All mathematical expressions will be developed as a function of x, the distance along the duct measured from the back end or end opposite to that of the fan. The following boundary conditions must be respected:

at $x=0$, the air velocity of the duct, V_x is zero.

at $x=L$, the average air velocity of the duct, V_x is V_L or the total volumetric fan flow divided by the duct's cross sectional area, A.

The air flowing through the duct system must respect the laws of mass conservation (continuity) and momentum conservation (Bernoulli's Law).

Furthermore, air flow distribution will be analyzed under steady state conditions and static regain will be assumed to govern the duct's pressure profile.

3.1. Mass Conservation

The flow of air through the duct and out by the long slot must respect the law of mass conservation. Considering that the flow at point x is equal to the flow at point (x+dx) less that lost through the long slot over distance dx,(Figure 3.1):

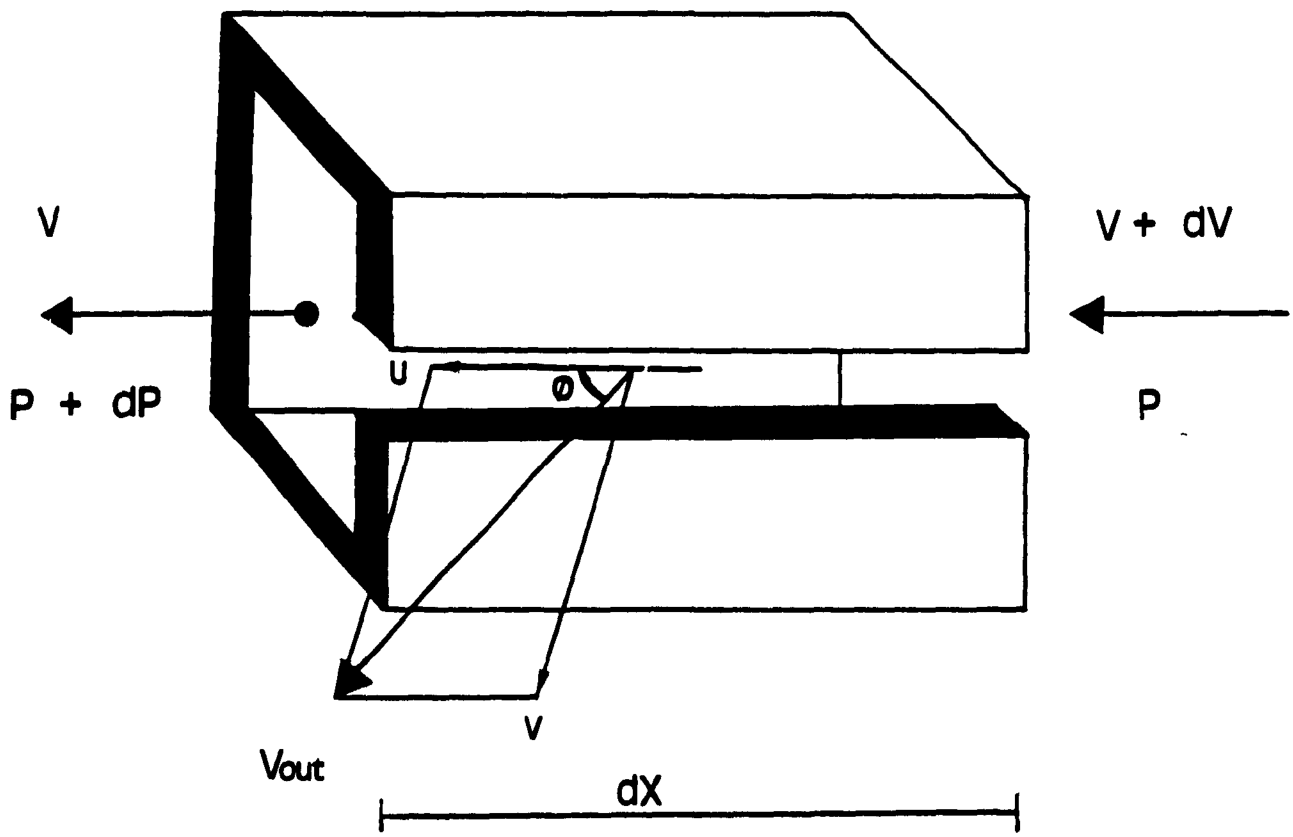

Figure 3.1 Duct Section Air Flow Parameters

$A(V+dV)=AV+ dq$ or $dq = AdV$ [2]

and $dq = v(Chdx)$ [3]

Where A -- cross-sectional area of the ventilation duct, m^2

V -- average air velocity inside the duct, m/s

h -- the height of the long slot outlet.

C -- discharge coefficient

v -- outlet air velocity perpendicular outlet area, m/s

dq -- air flow from the outlet over a partial surface, m^3/s

ρ -- air density, Kg/m^3

thus $AdV = vChdx$ [4]

The aperture ratio,θ, is the ratio of the total outlet area to the cross-section of the duct: $\theta=hL/A$ [5]

Therefore, the law of mass conservation is respected in terms of equation [4] and [5] if:

$dV=\theta Cv(dx/L)$ [6]

3.2. Outlet Air Momentum

The flow of air inside the duct can be considered a stream line which separates at the outlet. Then the momentum of the air in the duct equals that of the outlet. From Bernoulli's equation,

$P + \rho V^2/2 = \rho v^2/2 + \rho u^2/2$ [7]

where u is outlet air velocity parallel outlet area, m/s

At the end of the duct, x=0, the average air velocity of the duct is zero. Also, Horlock (1956) demonstrated that u is a multiple of V. Therefore, at x=0, if V=0, so is u=0. It follows, then, that

$$Po = \rho v_o^2/2$$

and $v_o^2 = 2Po/\rho$ at x=0 [8]

3.3. Definition of C.

The discharge coefficient of an air outlet, C, can be defined as:

$$C = \frac{dq}{[h\sqrt{2\frac{P}{\rho} + v^2}]\ dx} \qquad [9]$$

(Hellickman and Walker, 1983)

According to boundary conditions, at x=0, V=0 and P=Po. From equation [9], and the fact that $\rho dq = \rho AdV$, at x = 0,

$$dV/dx = (C\theta/L)\sqrt{2Po/\rho} \qquad [10]$$

3.4. Head Loss in the Ventilation Duct

It is known that head loss, Δh, along a ventilation duct in turbulent flow depends on the following parameters:

1. D_H, duct hydraulic diameter, m
2. L, length of duct over which the Δh occurs, m
3. μ, the coefficient of kinematic viscosity, m^2/s

4. V, the mean-time-average velocity over the duct's cross section, which is equivalent to q/A, m/s
5. ρ, mass density of the air, Kg/m^3
6. e, average variation in duct radius - a measure of duct roughness, mm

The pressure loss, Δh, along a system as a result of the flow of a fluid, can therefore be expressed in terms of dimensionless groups of parameters (Shames, 1982):

$$\Delta h/(\rho V^2) = f(L/D_H, Re, e/D_H) \qquad [11]$$

In order to simplify this equation, a term, F, is introduced such that: $\Delta h/(\rho V^2/2) = (L/D_H)*F$. The term F is called the "Darcy-Weisbach friction factor" because it is not a constant. Rather, it is a function of the Reynold's number, Re, and the ratio of the roughness coefficient of the conduit to its hydraulic diameter, (e/D_H). The measurement of the Darcy-Weisbach friction factor therefore requires that the flow conditions $(Re, e/D_H)$ be stated (Shames, 1982).

3.5. The Equation of Air Velocity

The average duct air velocity, expressed in terms of length along the duct,x, can be developed considering the equation expressing balanced conditions. Balanced conditions are those conditions occurring when the air distribution from the long slot outlet is perfectly uniform over the length, L.

Under balanced conditions, equation [6] is solved as follows:

$$\frac{dV}{dx} = \frac{V_L}{L} \qquad [12]$$

Considering now unbalanced conditions, the velocity solution must be a subset of that for balanced conditions. It will be assumed that, unbalanced conditions are described by:

$$V = G + M\frac{X}{L} + N\frac{X^2}{L^2} + S\frac{X^3}{L^3} + \ldots + Z\frac{X^n}{L^n} \qquad [13]$$

where G, M, N, S, ... Z are constant.

Furthermore, the constants associated with higher powers of X are determined by boundary conditions. From boundary conditions at X=0, it therefore follows that:

$G = 0 \quad M = C_o\theta\sqrt{2Po/\rho} = H_o$

Similarly at X=L, $V = V_L$. In consequence, $V_L = \theta C_o\sqrt{2Po/\rho} + N$ where:

$N = V_L - C_o\theta\sqrt{2Po/\rho} = V_L - H_o$

The solution for G, M and N, provide a definition for V:

$$V = H_o\frac{X}{L} + (V_L - H_o)\frac{X^2}{L^2} \qquad [1]$$

where $Ho = \theta Co\sqrt{2Po/\rho}$

θ -- aperture ratio, dimensionless

Co -- discharge coefficient at X=0

Po -- static pressure at X=0, Pa

ρ -- air density, kg/m^3

L -- length of duct, m

V_L -- duct average cross-section velocity at X=L, m/s

The goodness of fit of this equation will be verified by experimentation.

Chapter IV

MATERIALS AND METHODS

Outlet air flow and air duct static pressure were measured using wooden and polyethylene ventilation conduits. These ducts differed in length, aperture ratio and outlet shape as well as in spacing. The measured average duct air velocity was compared to that obtained from the velocity equation:

$$V = H_o \frac{X}{L} + (V_L - H_o) \frac{X^2}{L^2} \qquad [1]$$

where $Ho = \theta Co\sqrt{2Po/p}$

θ -- aperture ratio, dimensionless

Co -- discharge coefficient at X=0

Po -- static pressure at X=0, Pa

ρ -- air density, kg/m^3

L -- length of duct, m

V_L -- duct average cross-section velocity at X=L,m/s

4.1. Materials and Instrumentation

Measurements were taken to determine the air flow performance of nine (9) wooden and five (5) polyethylene ventilation ducts. These measurements pertained to duct air static pressure and outlet velocity as well as direction.

4.1.1. Wooden ventilation ducts

The experiment was carried out using nine (9) wooden ducts (Fig 4.1). Their frame was made of wooden members, 39 mm by 39 mm, and this frame was covered by 10 mm thick particle board panels. Particle board consists of wood particles of a wide variety of shapes and sizes, bonded together under heat and pressure using an adhesive resin (Keenan, 1986). These ducts offered a constant and gross cross-sectional area of 597 mm by 292 mm, or a net cross-sectional area of 0.17 m^2. Three ducts had a length of 16.8 m and six had a length of 8.54 m. Air was forced into these duct by a 450mm diameter axial fan with a 0.25kw variable speed motor. The fan was equipped with an air straightener and a 1.8 m long tapered section to fit it onto the experimental ducts. Total fan displacement was measured using a 510mm by 510mm plywood box fitted tightly over the frame of the fan at its inlet. Pairs of outlets were simply cut out from both sides of the duct panelling. Because this construction method is typically used by agricultural producers, no attempt was made to smooth out the outlet sides. The number, size and shape of the outlets are varied (Table 4.1).

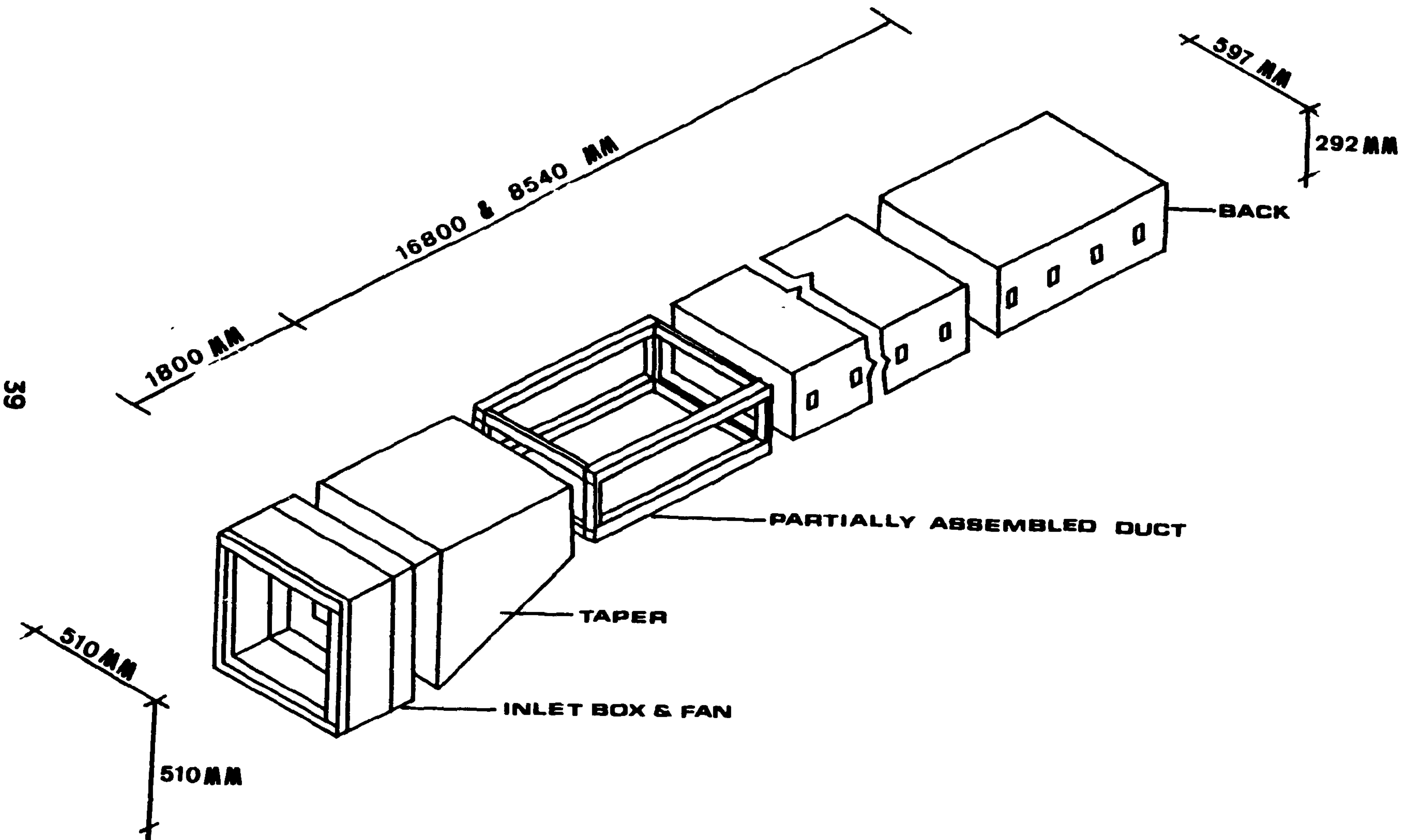

Figure 4.1 The Wooden Experimental Duct

Table 4.1 The Nine Wooden Experimental Ducts

Duct Number	Length m	Outlet Size mm*mm	Outlet Spacing mm	Outlet Number Pairs	Aperture Ratio
1	16.80	125*25	1200	14	0.50
2	16.80	125*25	600	28	1.00
3	16.80	125*50	600	28	2.00
4	8.54	125*25	600	14	0.50
5	8.54	D=63 mm	600	14	0.50
6	8.54	125*50	600	14	1.00
7	8.54	D=89 mm	600	14	1.00
8	8.54	125*75	600	14	1.50
9	8.54	D= 109 mm	600	14	1.50

4.1.2. Polyethylene Ventilation Ducts.

The experiment was carried out using five (5) polyethylene tubes, 457 mm in diameter, and of constant cross-sectional area(Fig. 4.2).For the purpose of the experiment, these polyethylene ducts of identical diameter were tested by

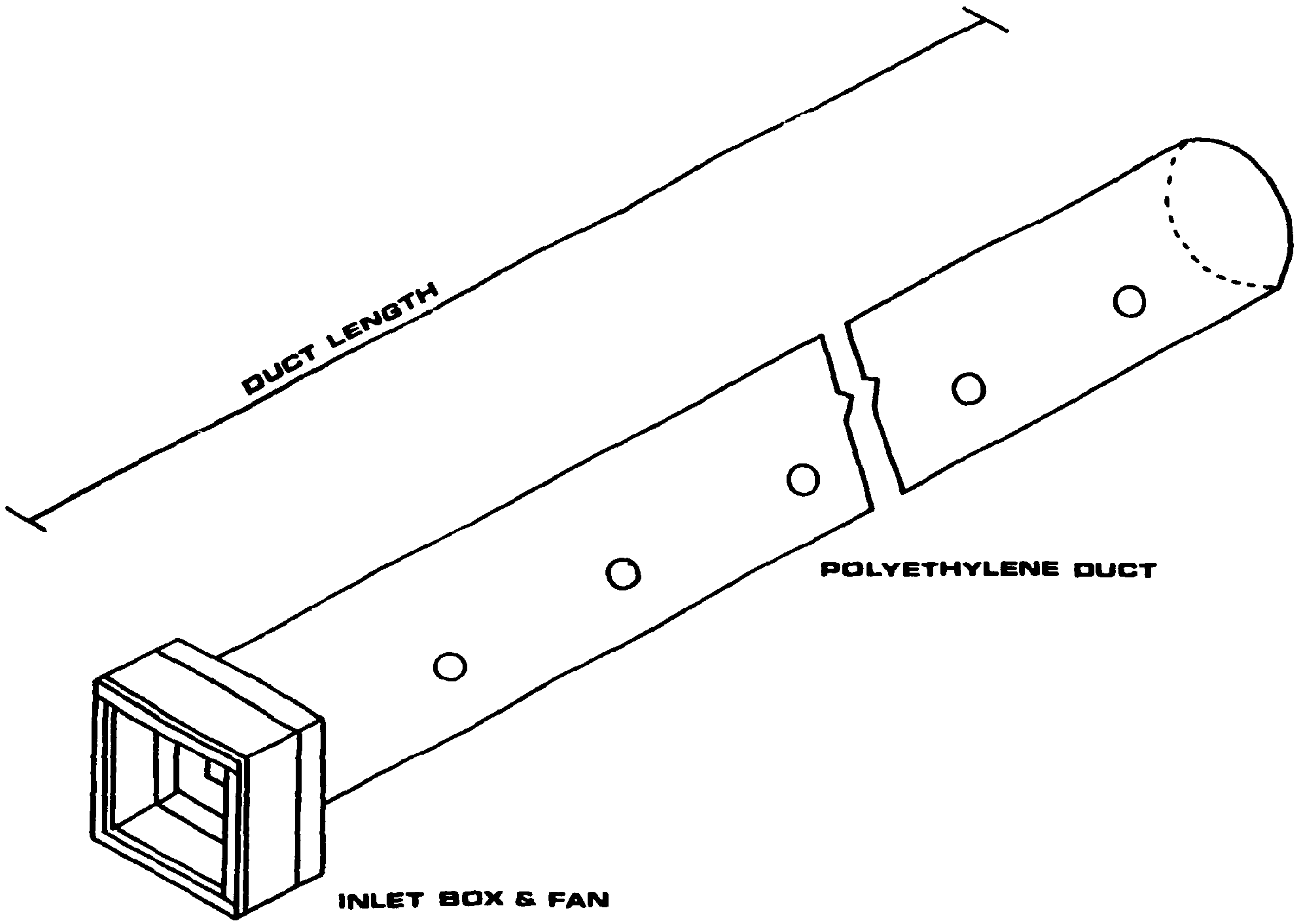

Figure 4.2 The Polyethylene Experiment Duct

varying their aperture ratio, their length as well as their outlet size and spacing. A 0.25 kw, 457 mm diameter axial fan was used to push air into the ducts. This fan was equipped with an air straightener and was directly connected to the tubing. Circular perforations were punched out of the sides of the tube in pairs. The diameter and spacing of these outlets were varied according to the length of the duct and the required aperture ratio (Table 4.2).

Table 4.2 The Five Polyethylene Wooden Experimental Ducts

Duct Number	Length m	Outlet Diameter mm	Outlet Spacing mm	Outlet Number Pairs	Aperture Ratio
1	14.30	47	715	20	0.40
2	15.02	69	1365	11	0.50
3	14.26	51	460	31	0.80
4	24.80	47	620	40	0.90
5	13.95	69	450	31	1.50

4.1.3. Instrumentation.

Static pressure was measured using an ALNOR microtector micromanometer with an accuracy of ± 0.10 Pa (± 0.4 thousandths of an inch, water gauge) over a range of 0 to 500 Pa (0-2 inch, water gauge). All air velocities were measured using an ALNOR compuflow thermo-anemometer with an accuracy of ± 3% of the indicated reading over a range of 0.1 m/s to 15 m/s. This thermo-anemometer was further calibrated using a calibration drum apparatus.

4.2. Methodology.

The nine (9) wooden ducts and five (5) polyethylene ducts were initially characterized by measuring their friction coefficient. Then, their outlet air flow performance was monitored in order to determine their average air velocity.

4.2.1. Roughness Coefficient Measurement.

The duct was characterized by obtaining its roughness coefficient, e, from the measurement of its Darcy-Weisback friction coefficient, F, under a known Re value and specific air condition.

The Darcy-Weisback friction coefficient, F, was calculated from the measured head loss, Δh, using a modified equation [11]:

$$F = (\Delta h / L) \times (2D_H / \rho V^{2)} \qquad [14]$$

Where F = Darcy-Weisback friction coefficient, dimensionless

Δh = drop in static pressure, Pa

D_H = duct's hydraulic diameter, m

L = length of duct over which Δh occurs,m

V = average duct air velocity, m/s

ρ = air density, kg/m^3

(Shames, 1982).

The parameter ($\Delta h/L$) was measured by obtaining static pressure readings along the length of the duct. A linear regression analysis was carried out between all static pressure readings and their distance along the duct's length. All pressure drop measurements were used for this analysis except for those located within the unable air movement area immediately after the fan over a distance of 2.4m. This method was used to define the average ($\Delta h/L$) value (Steele and Torrie, 1980).

The Reynold number can be calculated from the air flow conditions inside the ventilation duct:

$$Re = \rho V D_H/\mu. \qquad [15]$$

where ρ - air density at a given temperature and atmospheric pressure, Kg/m^3

V - average air velocity in the duct, m/s

D_H - hydraulic radius for the ventilation duct, m

μ. - absolute viscosity of air at a given temperature and atmospheric pressure, m^2/s

The duct's hydraulic radius was obtained using the relation:

$$D_H = 4A/R \qquad [16]$$

where A - cross-section area of the ventilation duct, m^2

R - hydraulic perimeter of the ventilation duct, m

(Hearter, 1963).

The experimental ventilation ducts can therefore be characterized by measuring their head loss under a specific Re value. Knowing the mean-time average air velocity in the duct and the properties of this air under the specific experimental conditions (air temperature and atmospheric pressure), the Darcy-Weisback friction coefficient,F, can be calculated along with the Re value. With the knowledge of F, Re and D_H, the experimental ducts can be characterized by obtaining the skin roughness coefficient, e.

4.2.2. Measurement of Static Pressure, Outlet flow and Outlet air jet angle.

The performance of the nine (9) wooden experimental ducts and 5 polyethylene experimental ducts were then obtained by measuring the interior static pressure along their length and the flow, the angle as well as the average velocity of the air jet escaping from each pair of outlet.

Static pressure was measured at half an outlet spacing after the duct's reducing section, at every other outlet and directly at the back of the duct. All static pressure readings were measured three times and averaged.

Outlet flow measurement was measured using the grid method (ASHRAE, 1985). The readings were repeated five (5) times and averaged to establish the actual velocity of the perforation.

It is necessary to measure the outlet air jet angle since the outlet air velocity perpendicular outlet area were used to determine the air velocity inside duct. To measure the outlet air jet angle, with respect to the duct longitudinal wall, a three (3) tube pitot instrument was used. This instrument was mounted in such a way as to measure the air jet angle at the centre of each outlet. The jet angle was measured only at the outlet centre because this angle was not found to vary by more than 1° for all the grid points used for the flow measurement of the outlet.

Total fan throughput was measured using a 510 mm by 510 mm box at its inlet and grid method (ASHRAE, 1985). Total outlet flow and fan throughput agreed within an error of 5%, error level exceeded by Brundett and Vermes (1987).

4.2.3. Testing the Mathematical Model.

In order to test the mathematical model introduced earlier, the measurements were analyzed to obtain actual values for V_L, Co and Ho.

The average cross sectional duct velocity was obtained by summing up the flow from each outlets starting at the back and away from the fan. This measured velocity was compared to that obtained from mathematical model, equation [1]. For this mathematical model, V_L was obtained from the average duct velocity at the fan and Ho was obtained from θV_{out}, at x=0 (Appendix A).

All the experimental data was compared to the values obtained from the mathematical model by analyzing their respective standard deviation using the student t method (Steel and Torrie, 1980). Linear regression was used to test if X^3 and X^4 term, would improve the performance of equation [1].

Chapter V

RESULTS

The air flow performances obtained from fourteen (14) ducts were used to test the mathematical equation,

The results will be presented in three (3) sections,

1). three (3) wooden ducts, 16.8 m in length but with variable aperture ratios;

2). six (6) wooden ducts, 8.54 m in length but with two (2) different outlet shapes and three (3) different aperture ratios;

3). five (5) polyethylene ducts of various length, aperture ratio and outlet spacing.

5.1. The Wooden Ducts.

The measurements will be presented in two (2) sections according to different duct length.

5.1.1. Roughness Coefficient.

The characterization of the wooden ducts consisted in measuring their skin friction coefficient, F. With the outlets closed but the back of the duct open, static pressure was measured along the length of the duct (Figure 5.1.1.1). The average air velocity in the duct was obtained by measuring the air flowing into the fan.

Static pressure as a function of duct length was analyzed through linear regression:

$$h = 6.35 - 1.5X \qquad [17]$$

$$r = 0.975$$

the slope of this linear regression equation represents the pressure gradient, $\Delta h/\Delta L$ and can therefore be used to calculate F where:

$$F = (\Delta h/\Delta L)*D*(2/\rho V^2)$$

$$= 0.01706$$

This air velocity of the duct for such F value was 7.28 m/s. Duct's hydraulic diameter was 0.392 m. μ, was $1.6*10^{-5}$ m^2/s (T = 25°C) (Streeter and Wylie, 1981). The Reynold number, Re, was $2.14*10^5$. From standard factor coefficient curves for flow in pipes (Shames, 1982), the relative

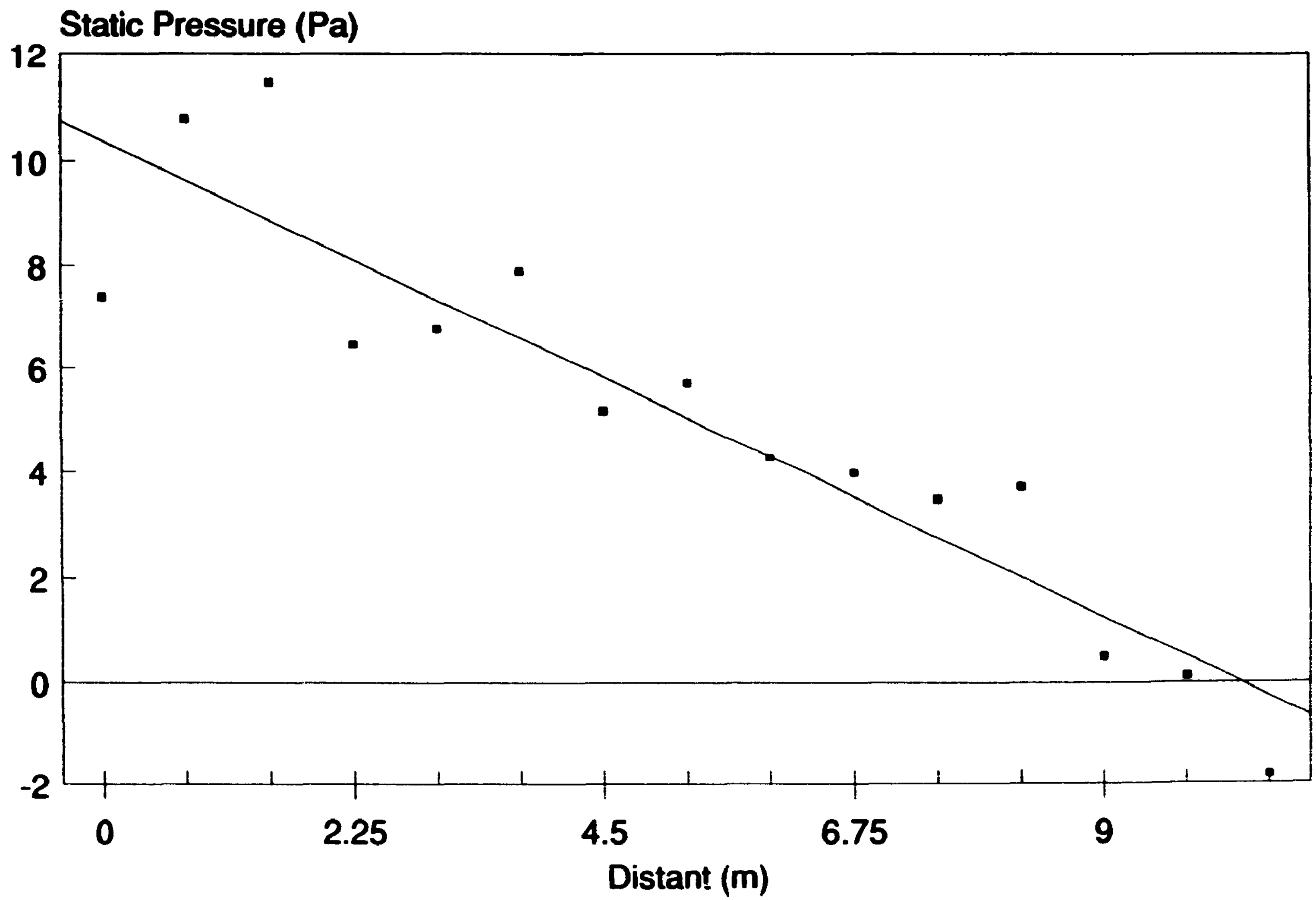

Figure 5.1.1.1. Linear Regression For F of Wooden Duct

roughness,e/D, was $6.0*10^{-4}$. Therefore, the average variation in duct radius -- a measure of duct roughness, e, was 0.235 mm. This value is slightly corresponds to that reported by Shames (1982) of 0.18 mm to 0.90 mm for wooden stave conduit.

5.1.2. The Performance of Ducts.

5.1.2.1. Three (3) Wooden Ducts, 16.8 m in length but with variable aperture ratio.

Air distribution parameters were measured using three (3) wooden ventilation ducts with a different outlet aperture ratios.

The effect of the aperture ratio on flow distribution was determined by varying the outlet area and outlet spacing. Outlet velocity, duct static pressure and outlet air jet angle were measured as shown on figure 5.1.1.2, 5.1.1.3 and 5.1.1.4. Outlet velocity became more uniform over duct length as the aperture ratio was reduced from 2.0 to 0.5. For all aperture ratios under 1.0, uniformity in outlet velocity was maintained at the expense of a loss in volume displacement. The optimum aperture ratio therefore appeared to be in the vicinity of 1.0, this ratio providing for uniform flow distribution under maximum fan throughput. Davis et al. (1980) and Barrington and Mackinnon (1989) obtained similar results with a perforated corrugated metal duct and a 8.54 m length wooden duct. The static pressures as measured from one end of the duct to the other, varied in the aperture ratio and distance along the duct. For the

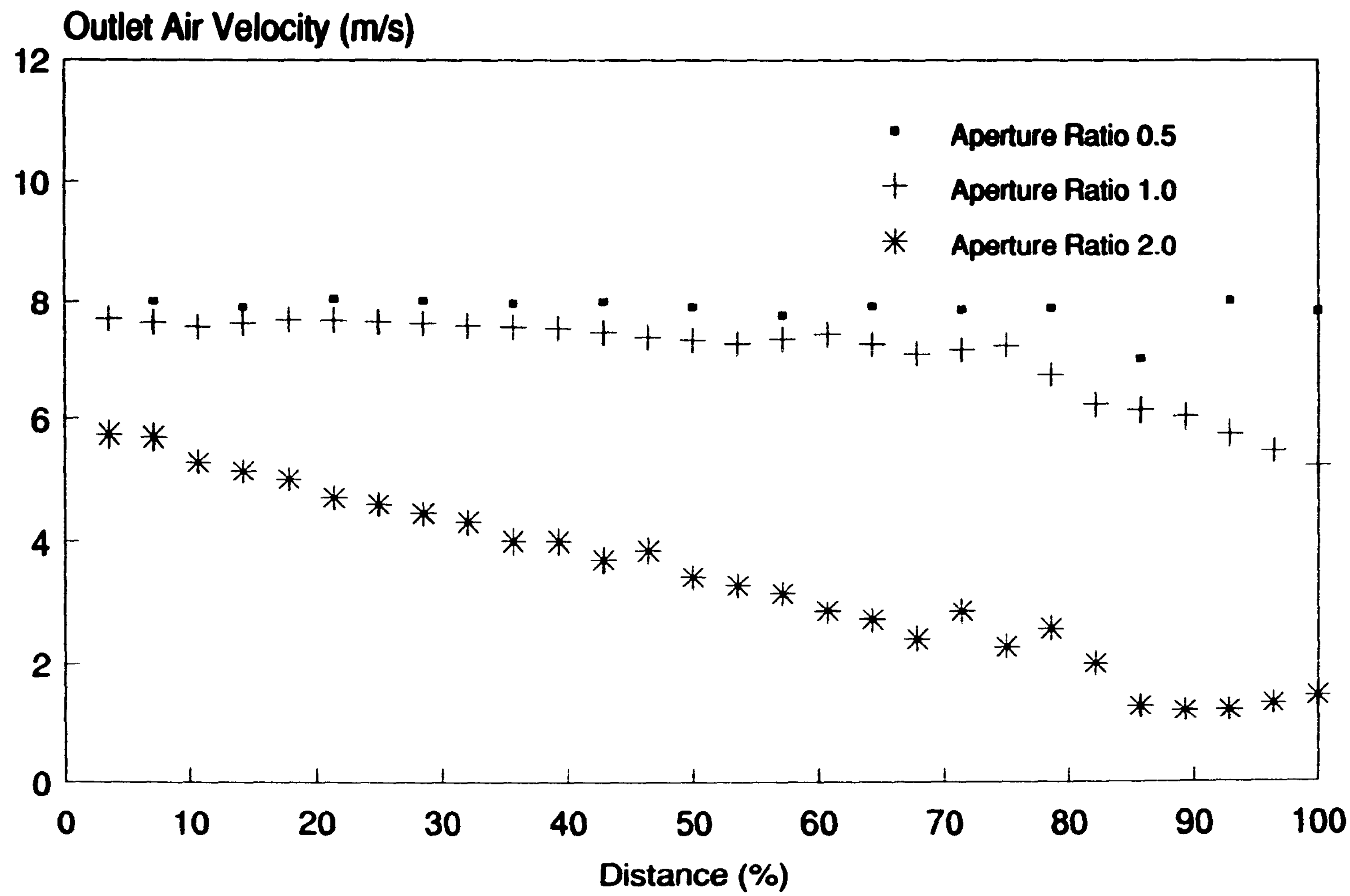

Figure 5.1.1.2. Duct Outlet Velocity with Distance along the Duct and as Function of Aperture Ratio

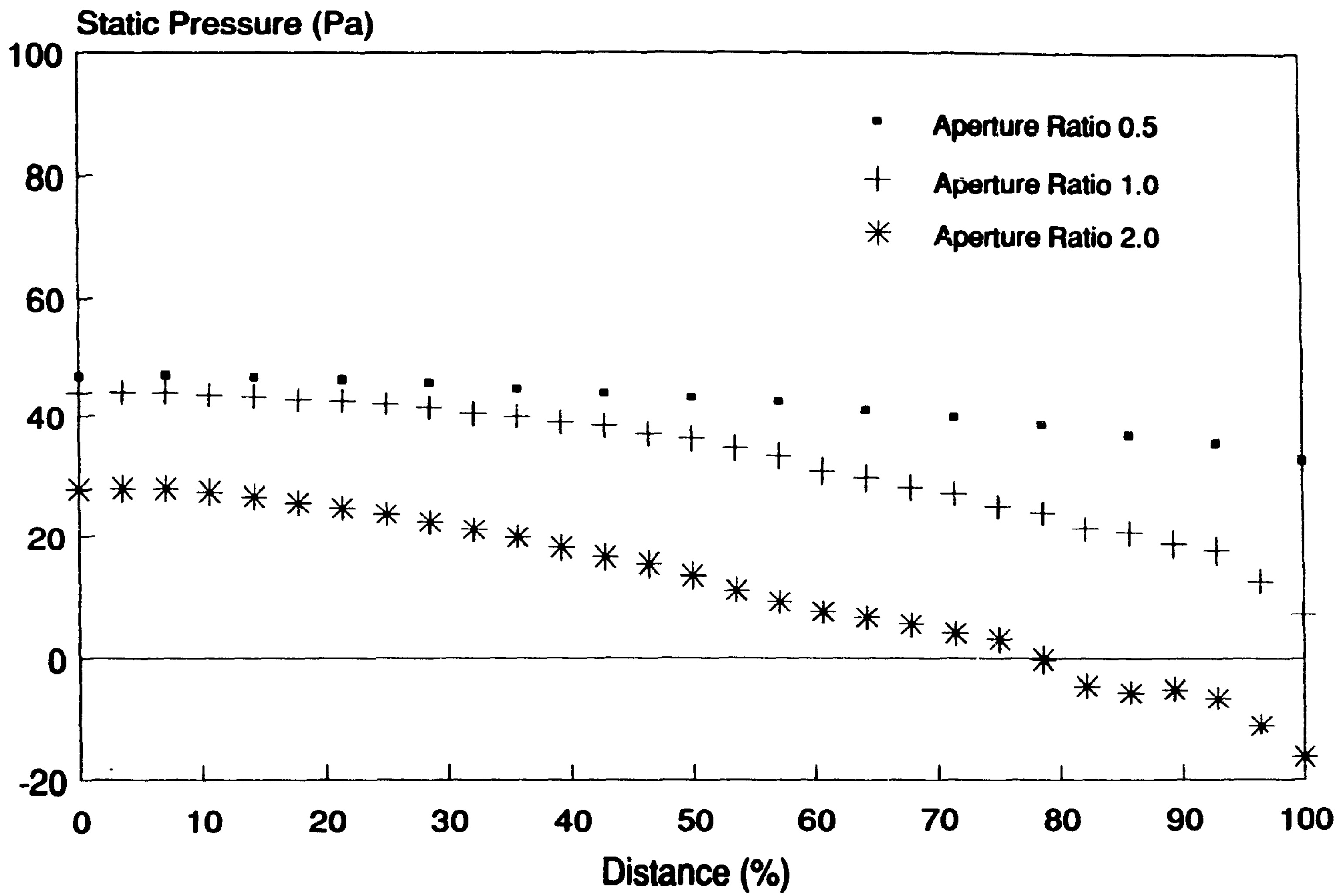

Figure 5.1.1.3. Duct Static Pressure with Distance along the Duct and as Function of Aperture Ratio

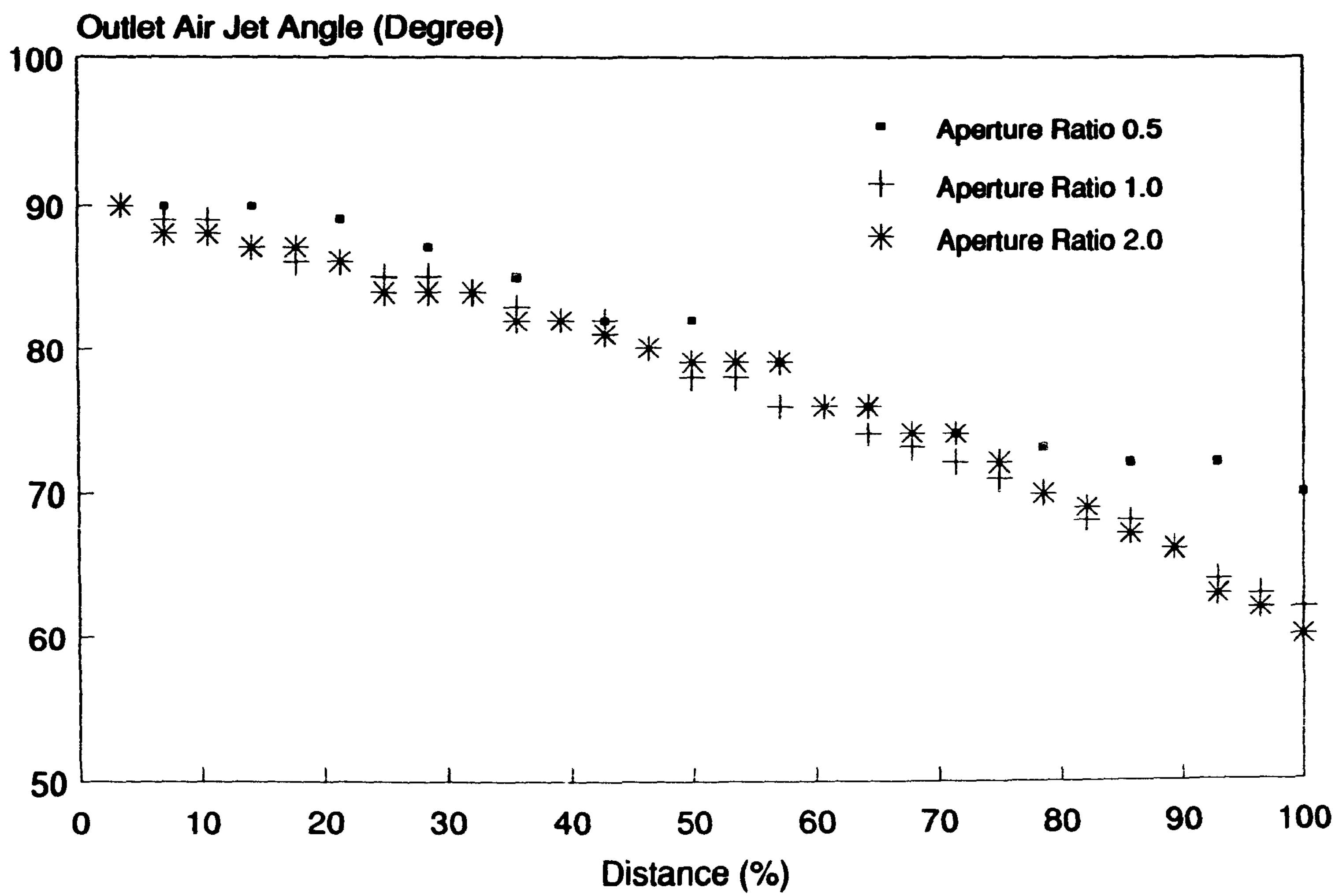

Figure 5.1.1.4. Outlet Air Jet Angle with Distance along the Duct and as Function of Aperture Ratio

aperture ratio of 2.0, the static pressure fell to a negative value at a distance of 3.2m from the fan. The outlet air jet angle varied from 90° at end of the duct to 70° at first outlet near the fan, for the aperture ratio of 0.5; for the aperture ratio of 1.0 and 2.0, the angle varied from 90° to 60°.

5.1.2.2. Six (6) Wooden Ducts, 8.54 m in length but with two (2) different outlet shapes and three (3) different aperture ratios.

Air distribution parameters were measured using six (6) wooden ventilation ducts with two (2) different outlet shapes and three (3) different aperture ratios.

Air distribution parameters, outlet air velocity, duct static pressure and outlet air jet angle were obtained from two outlet shapes, rectangular and round, at the three aperture ratio are illustrated by figure 5.1.2.1a.b.c, 5.1.2.2a.b.c and 5.1.2.3a.b.c. The two different outlets resulted in different air static pressures, outlet air velocity as well as air jet angle. This difference increased with aperture ratio. The round apertures consistently gave lower outlet air velocities as compared to the square outlets, except near the fan. Similarly, the round outlets produced higher air static pressures inside the duct except for the aperture ratio of 0.5. Differences in air jet angle were not consistently different.

5.2.Polyethylene Ducts.

Air distribution parameters were measured using five (5) polyethylene ventilation ducts with different duct length, aperture ratio and outlet spacing.

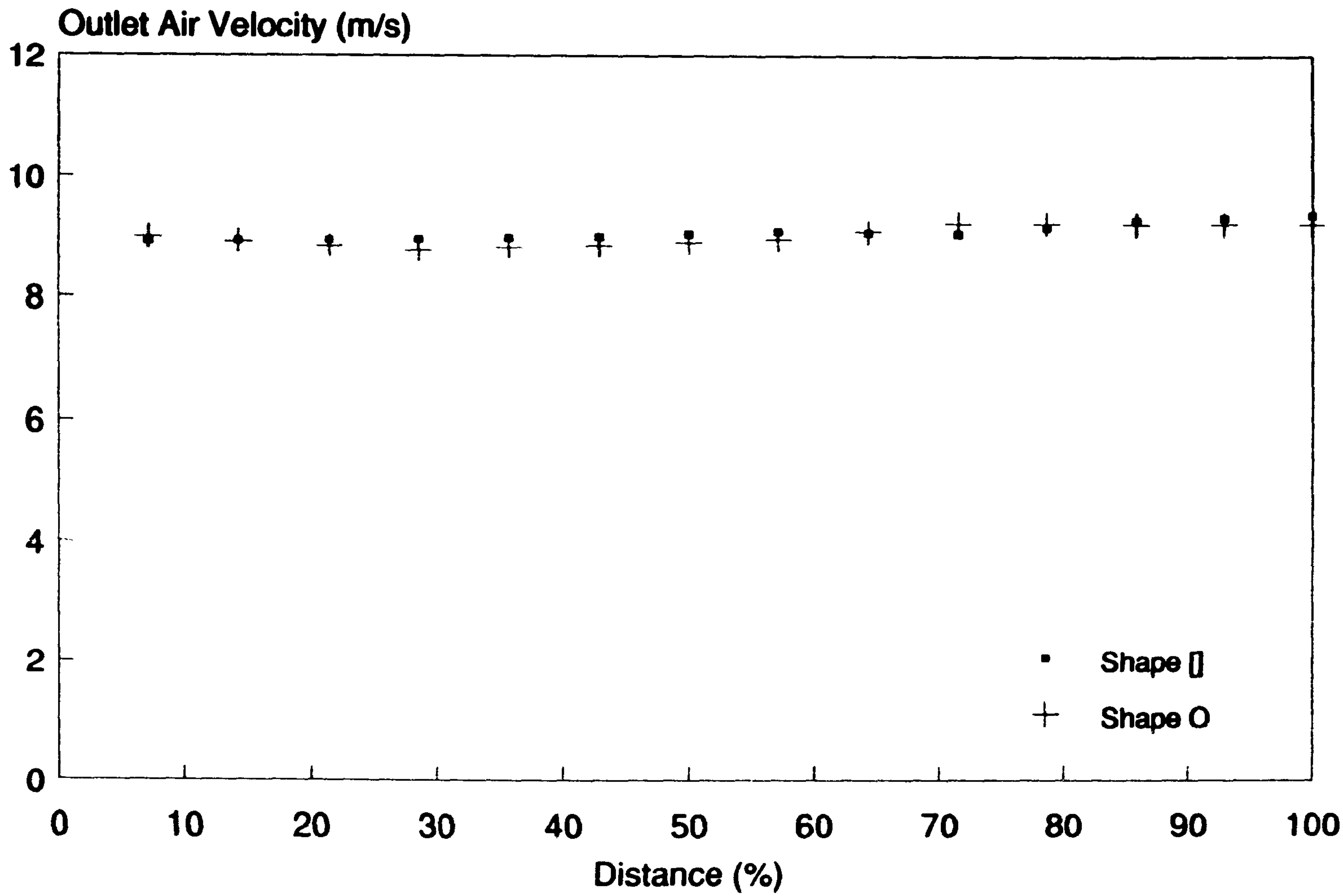

Figure 5.1.2.1a Duct Outlet Velocity as Function of Distance
for Aperture Ratio of 0.5

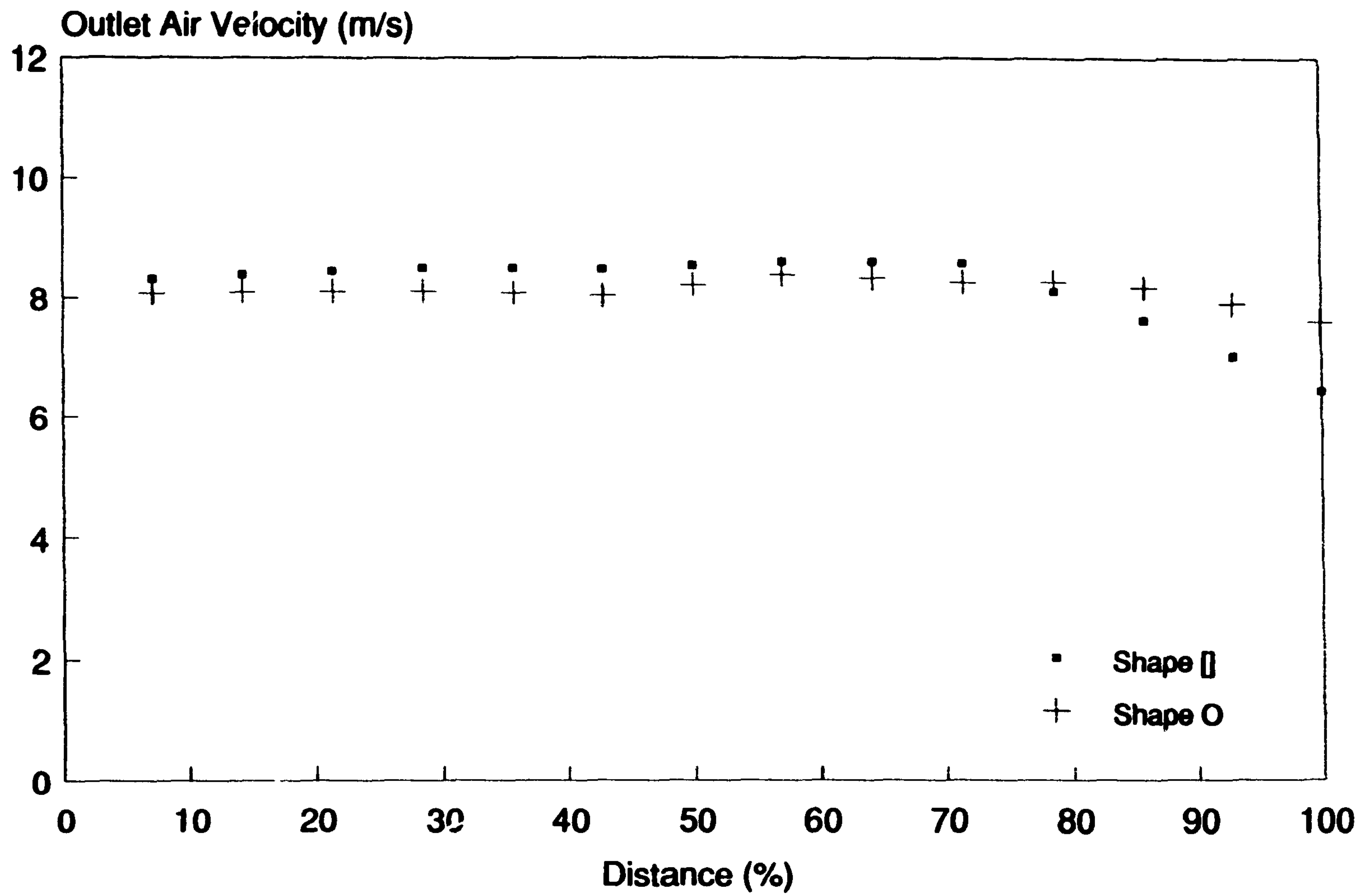

Figure 5.1.2.1b Duct Outlet Velocity as Function of Distance
for Aperture Ratio of 1.0

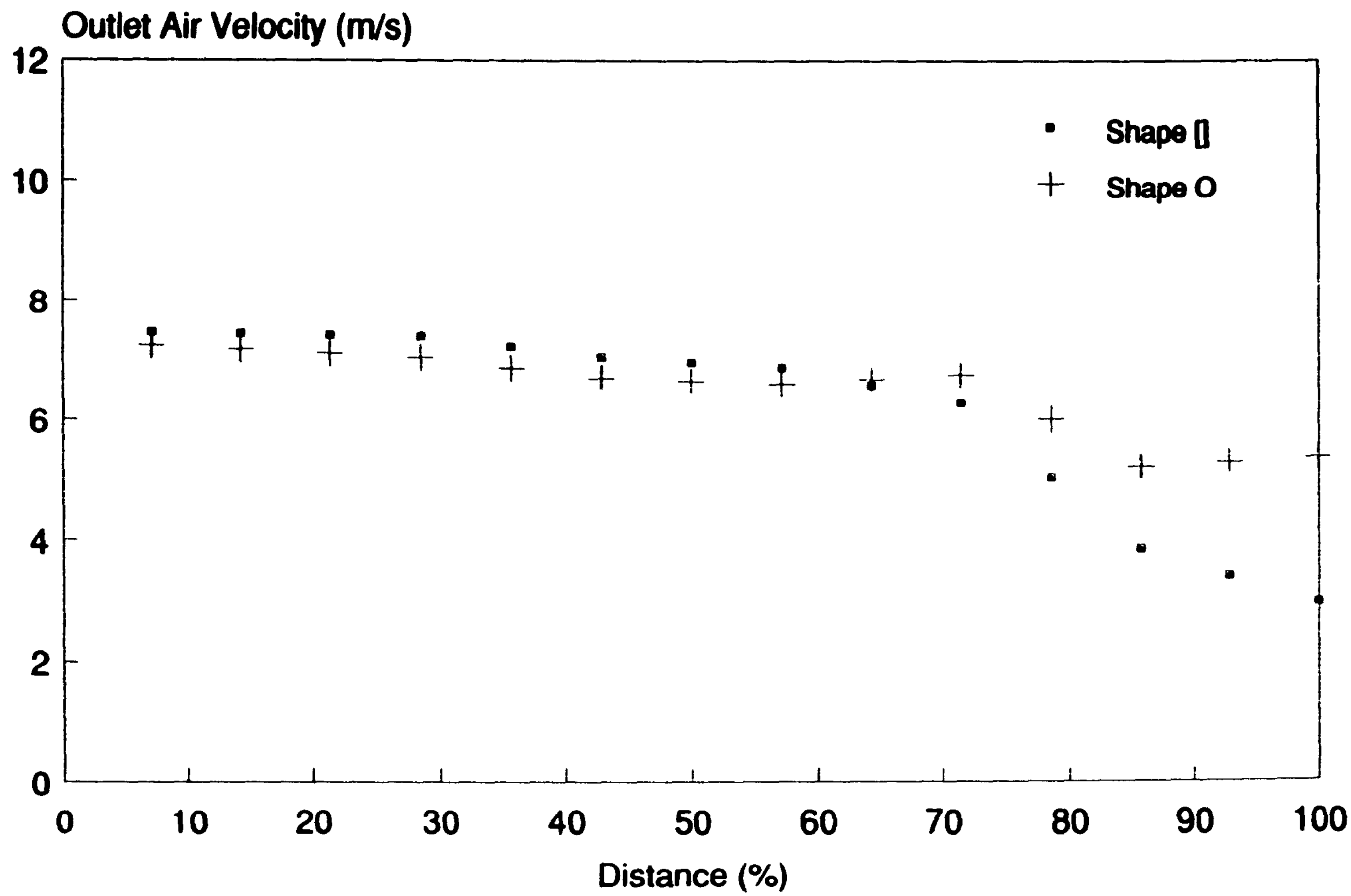

Figure 5.1.2.1c Duct Outlet Velocity as Function of Distance
for Aperture Ratio of 1 5

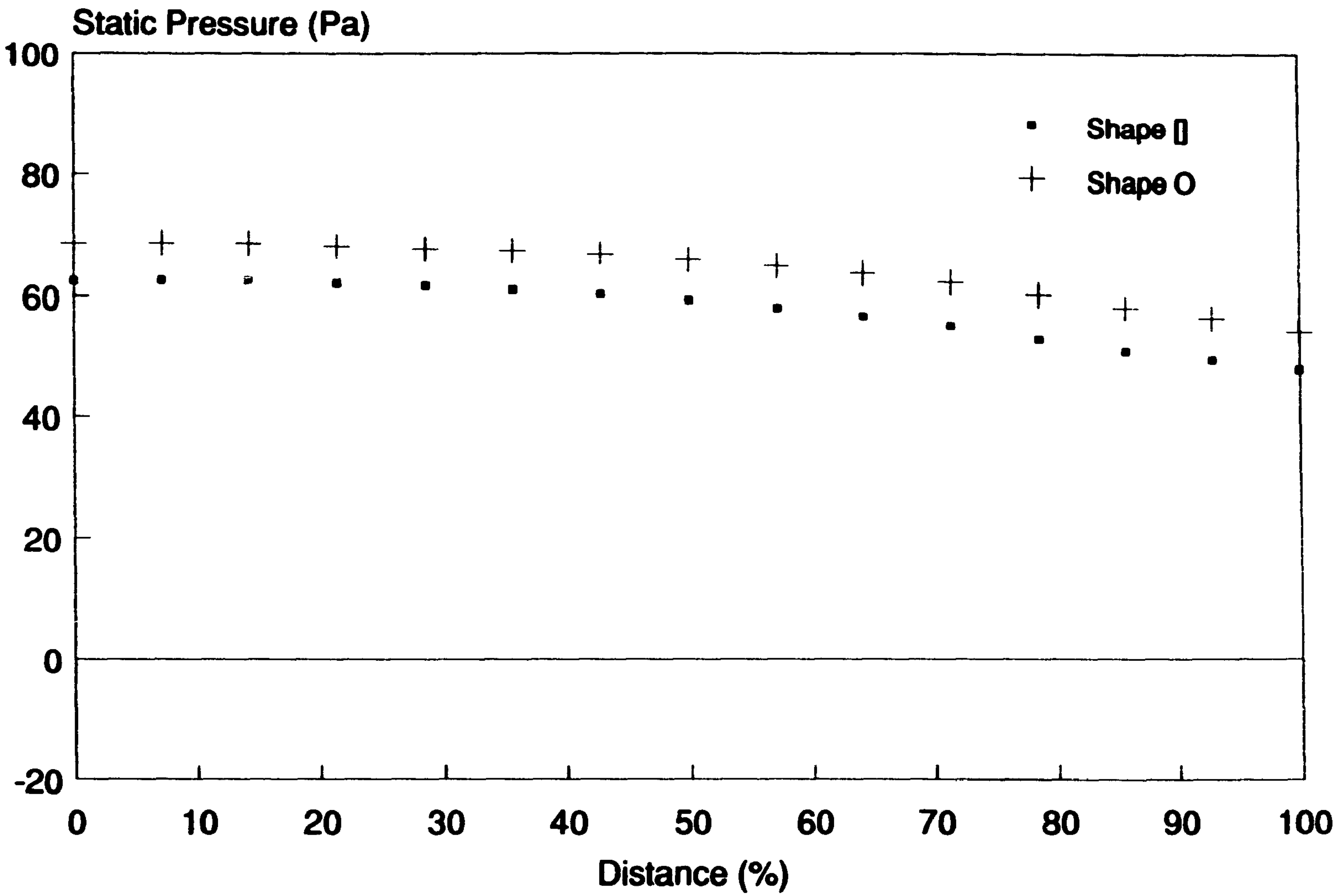

Figure 5.1.2.2a Duct Static Pressure as Function of Distance for Aperture Ratio of 0.5

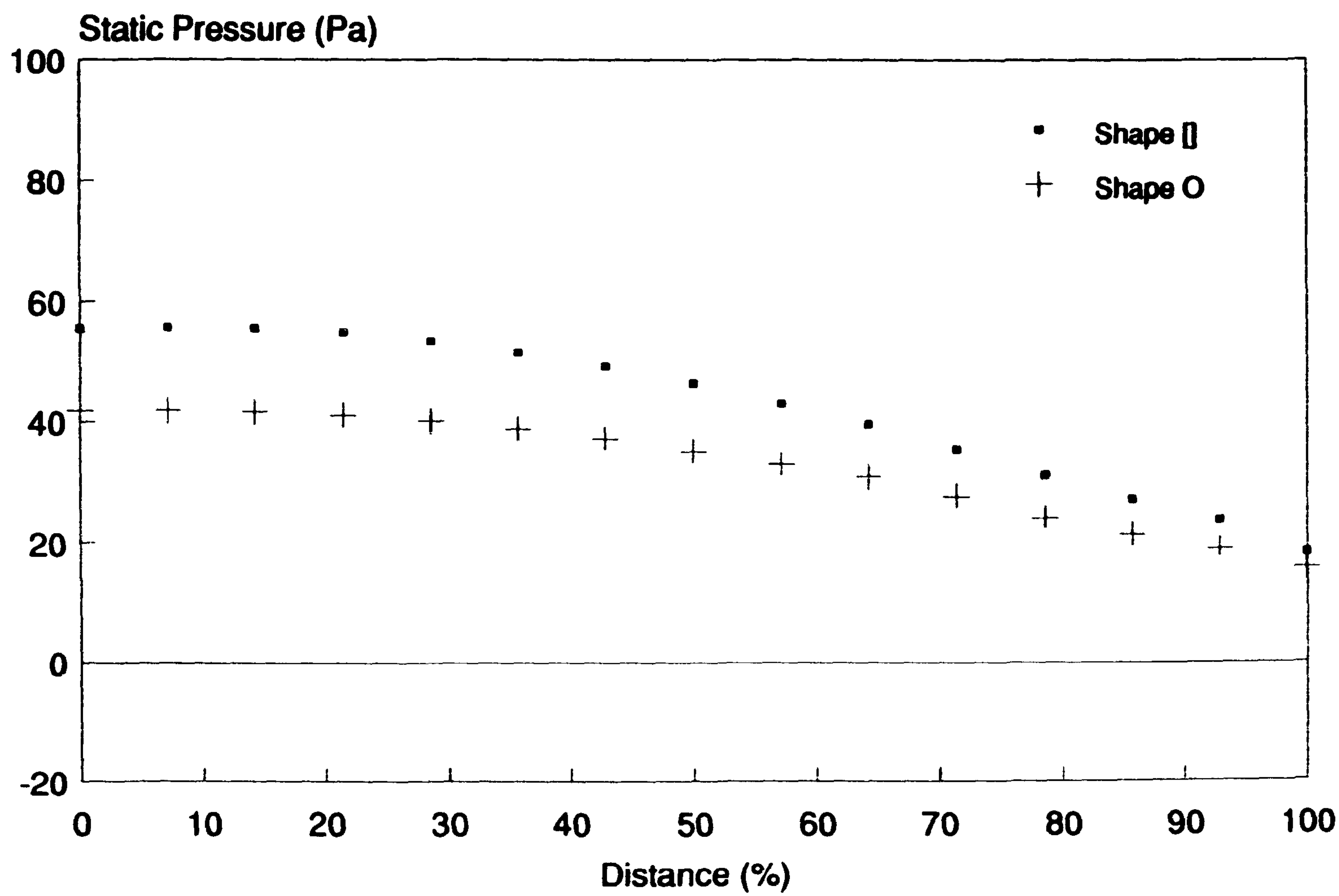

Figure 5.1.2 2b Duct Static Pressure as Function of Distance for Aperture Ratio of 1 0

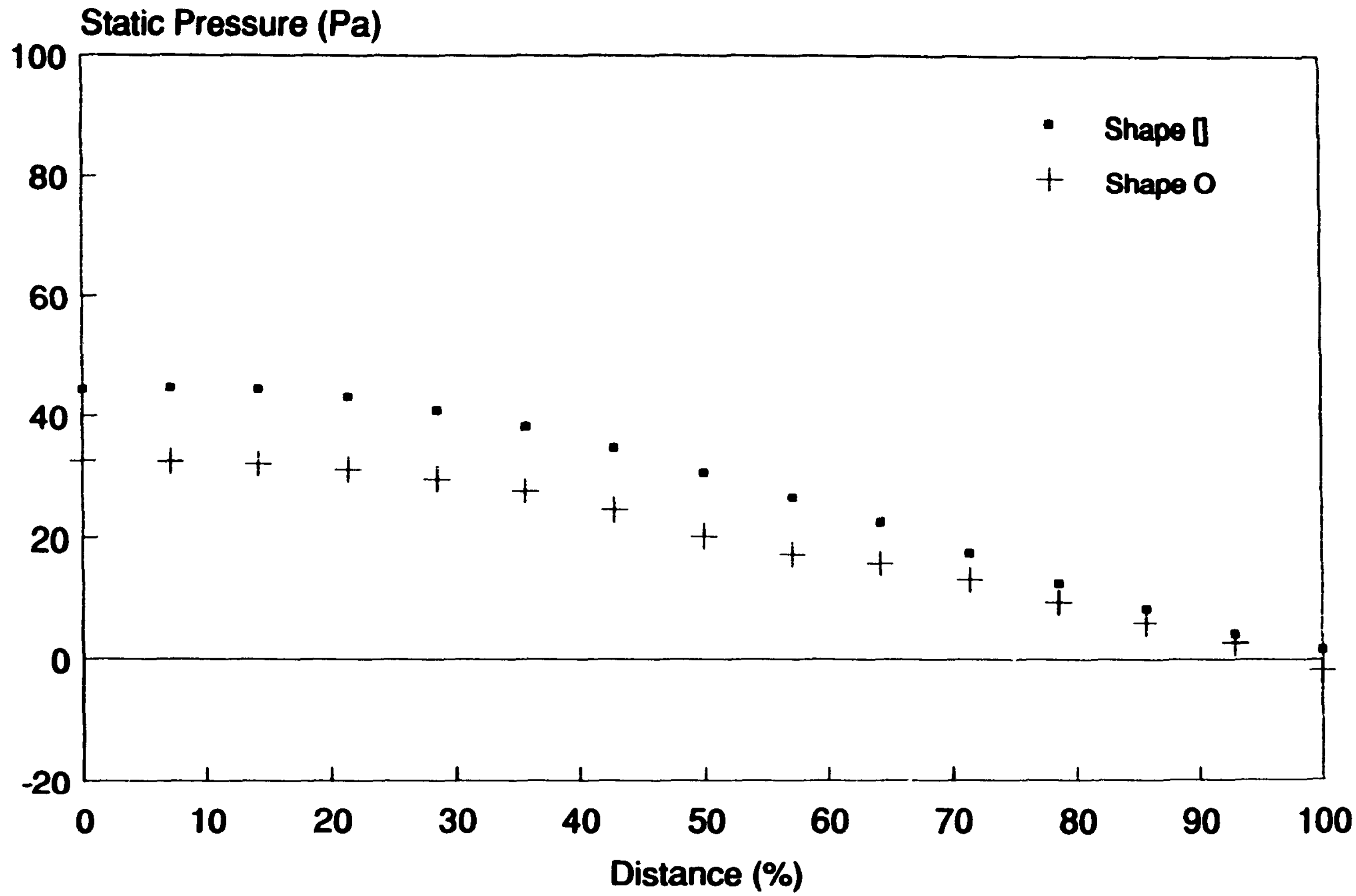

Figure 5.1.2.2c Duct Static Pressure as Function of Distance for Aperture Ratio of 1.5

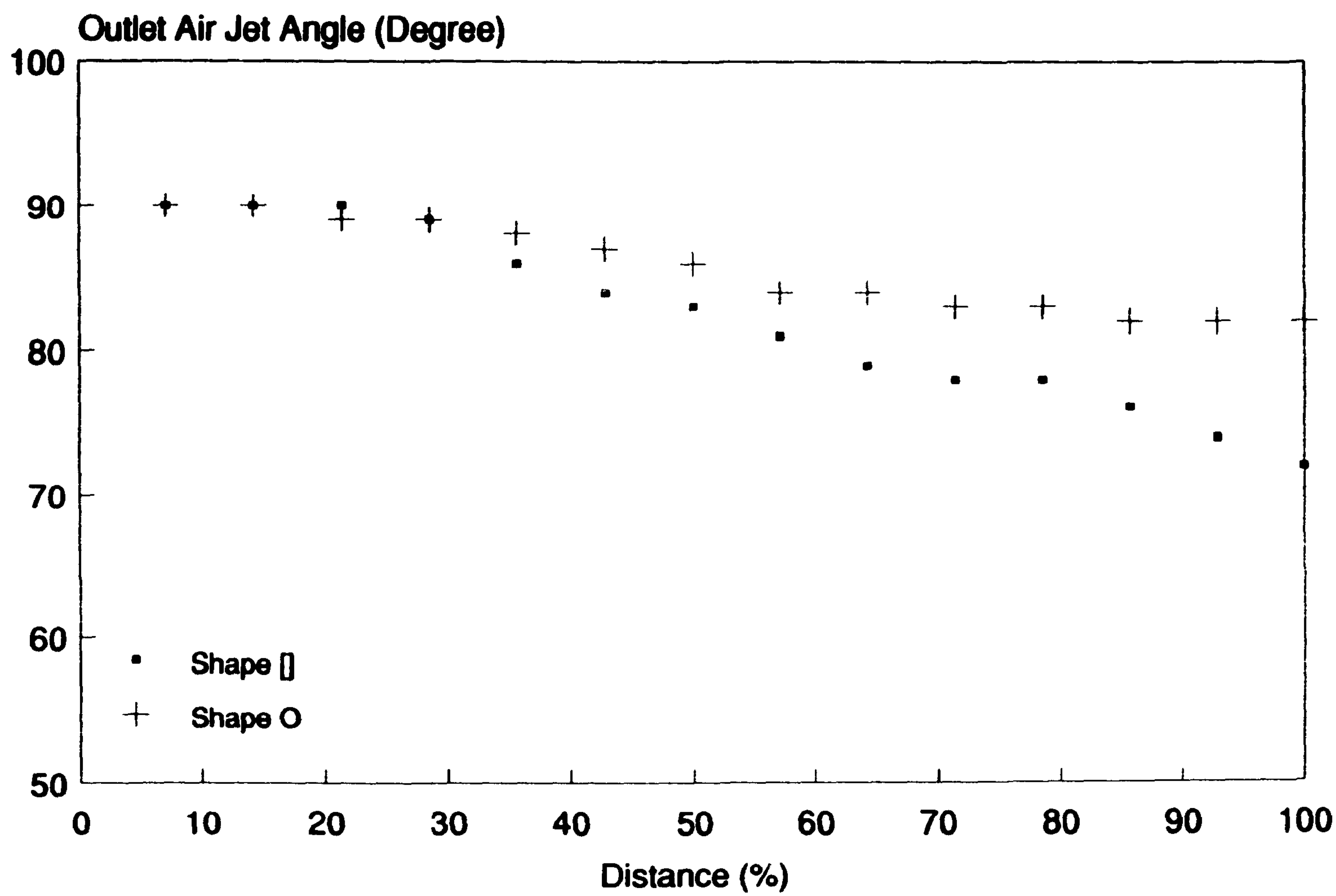

Figure 5.1.2.3a Outlet Air Jet Angle as Function of Distance
for Aperture Ratio of 0.5

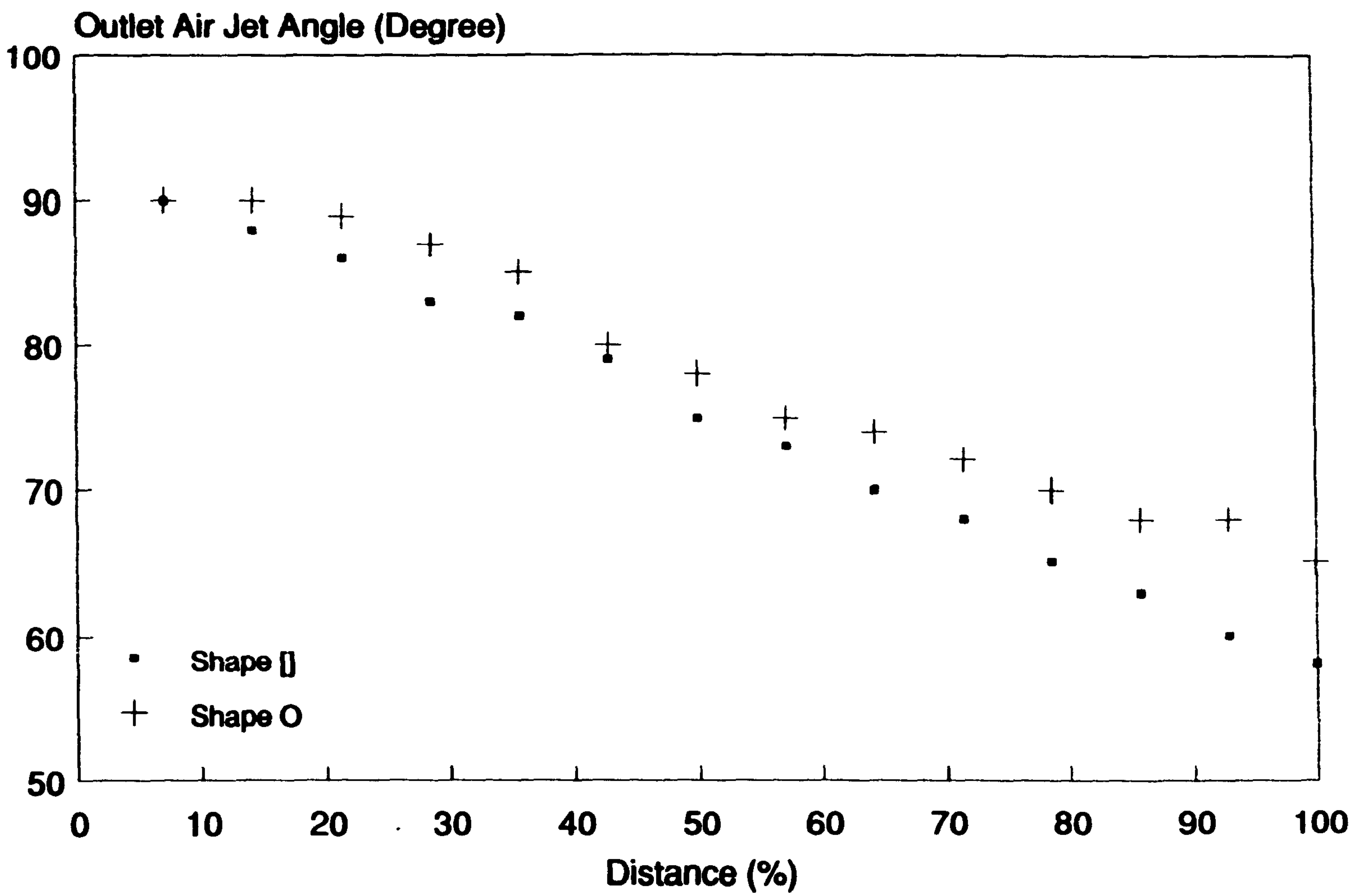

Figure 5.1.2.3b Outlet Air Jet Angle as Function of Distance for Aperture Ratio of 1.0

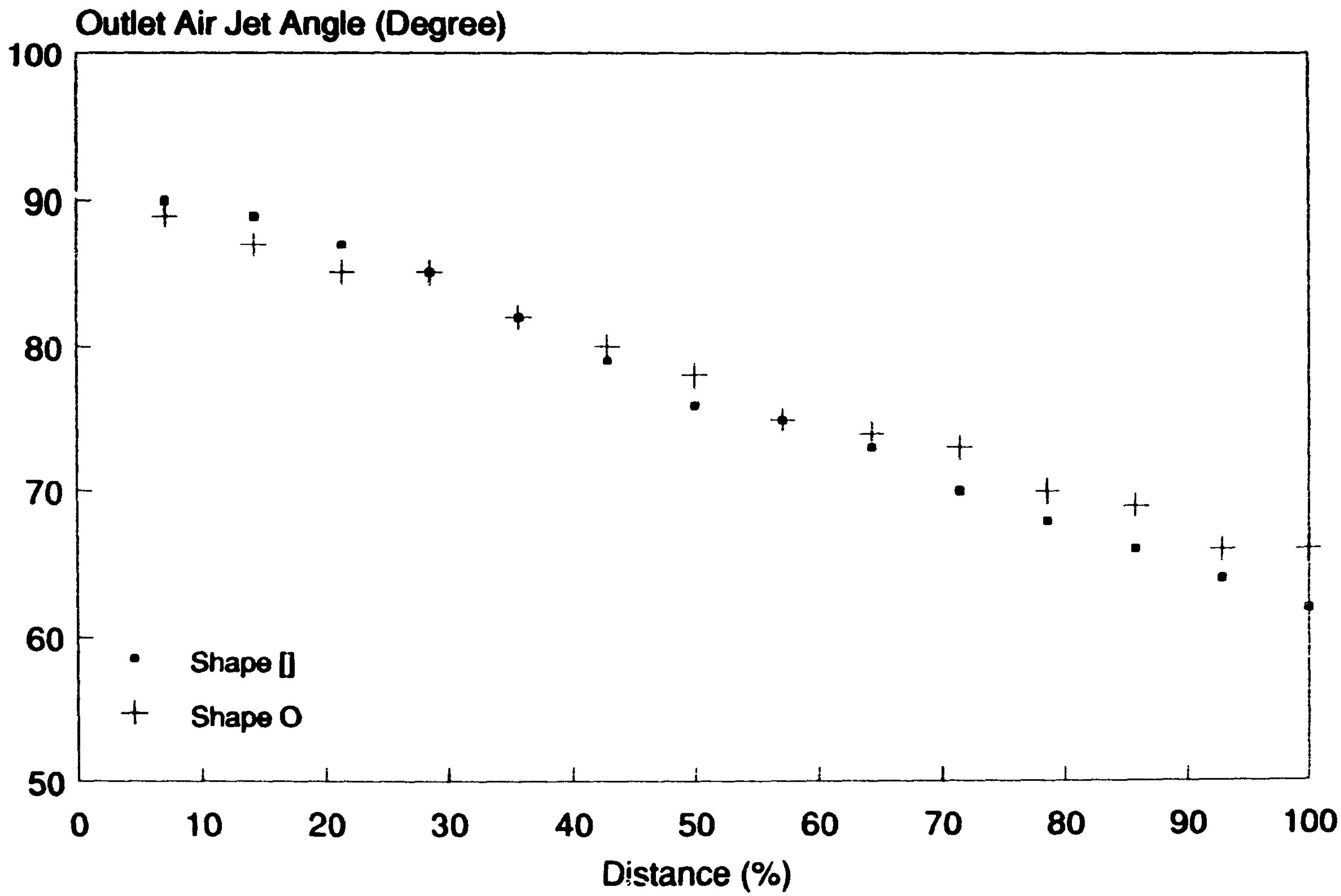

Figure 5.1.2.3c Outlet Air Jet Angle as Function of Distance
for Aperture Ratio of 1.5

5.2.1. Roughness Coefficient.

The characterization of the wooden ducts consisted in measuring their skin friction coefficient, F. With the outlets closed but the back of the duct open, static pressure was measured along the length of the duct (Figure 5.2.1.1). The average air velocity in the duct was obtained by measuring the air flowing into the fan.

Static pressure as a function of duct length was analyzed through linear regression:

$$h = 54.81 - 1.23X \qquad [18]$$

$$r^2=0.956$$

the slope of this linear regression equation represents the pressure gradient, $\Delta h/\Delta L$ and can therefore be used to calculate F where:

$$F = (\Delta h/\Delta L)*D*(2/\rho V^2)$$

$$= 0.0131$$

This air velocity of the duct for such F value was 8.54 m/s. Duct's hydraulic diameter was 0.465 m. The coefficient of viscosity, μ, was $1.6*10^{-5}$ m^2/s (Streeter and Wylie, 1981). The Reynold number, Re, was $2.98*10^5$. From the standard factor coefficient curves for flow in pipes (Shames, 1982), the relative roughness,e/D, was $2.0*10^{-6}$. Therefore, the average variation in duct radius -- a measure of duct roughness, e, was 0.0009 mm. This value is slightly corresponds to that reported by Shames (1982) of 0.0003 mm (drawn tubing) to 0.0015 mm

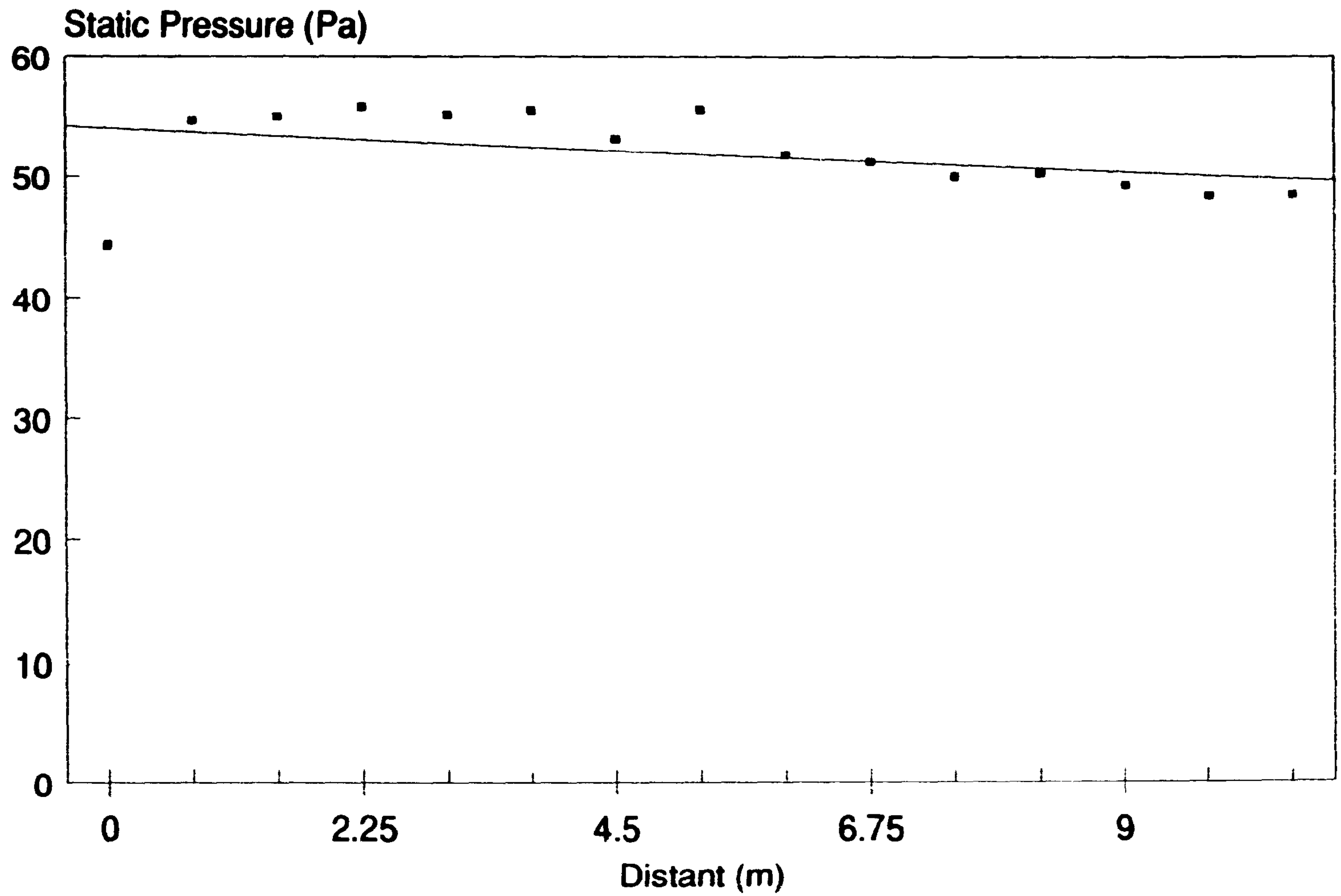

Figure 5 2 1.1. Linear Regression for F of Polyethylene Duct (glass pipes). Also, the relationship between Reynold number, Re, and friction coefficient, F, is slightly corresponds to Brundrett and Vermes's reports (1987).

5.2.2. The Performance of Ducts.

The experimental method was designed to characterize the system in a first instance and then measure the performance of five different polyethylene tubes, all of 457 mm in constant diameter.

The outlet flow, static pressure and outlet air jet angle, as a function of duct distance away from the fan, are illustrated by figure 5.2.2.1, 5.2.2.2 and 5.2.2.3. Outlet velocity became more uniform over duct length as the aperture ratio was reduced from 1.5 to 0.5. The static pressures measured from the end of the duct to fan, varied from 85 Pa to 10 Pa. For all five ducts, static pressures never fell below zero even near the fan. The outlet air jet angle varied from 90° to 65°, from the back end to the fan.

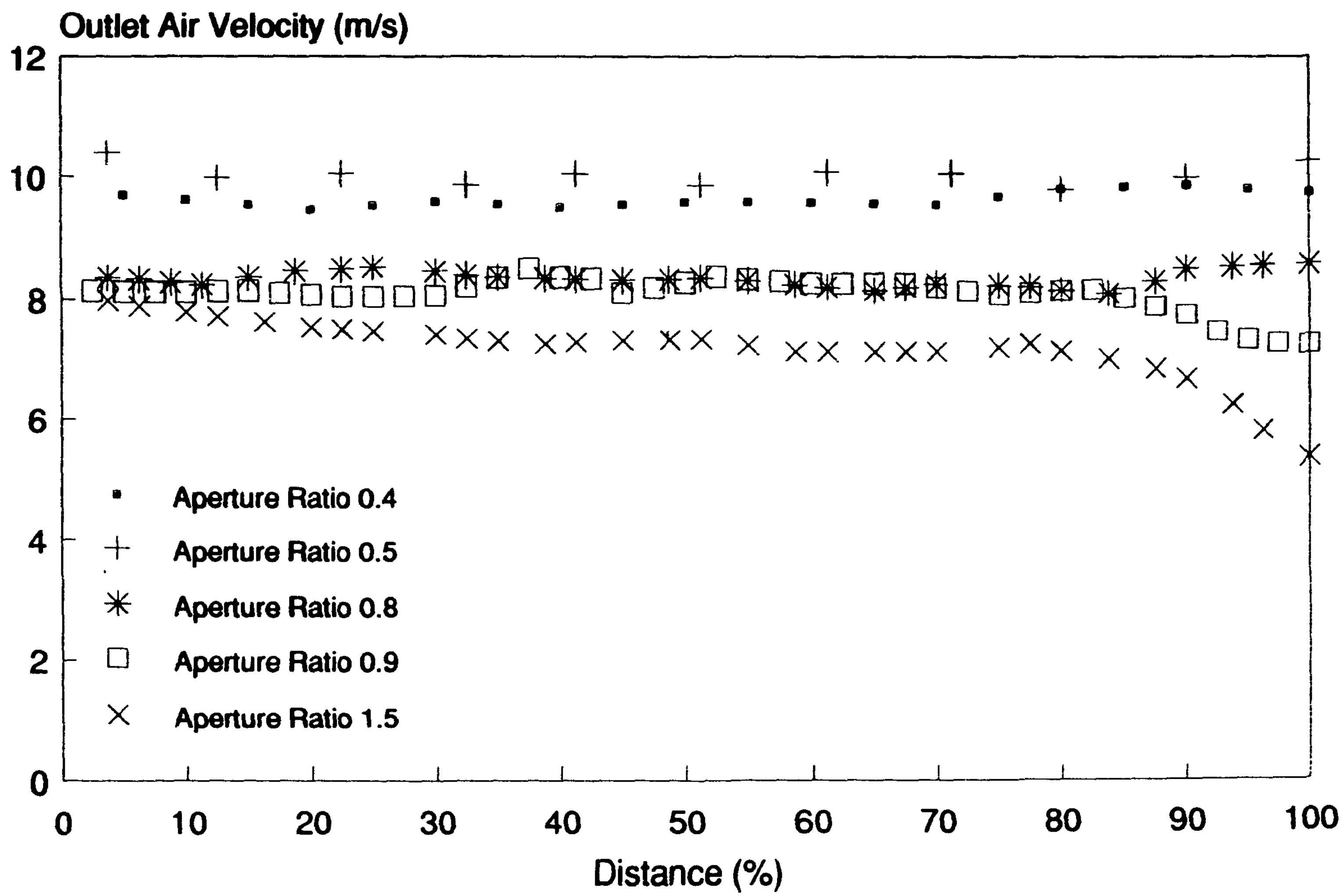

Figure 5.2.2.1. Duct Outlet Velocity with Distance along the Duct and as Function of Aperture Ratio

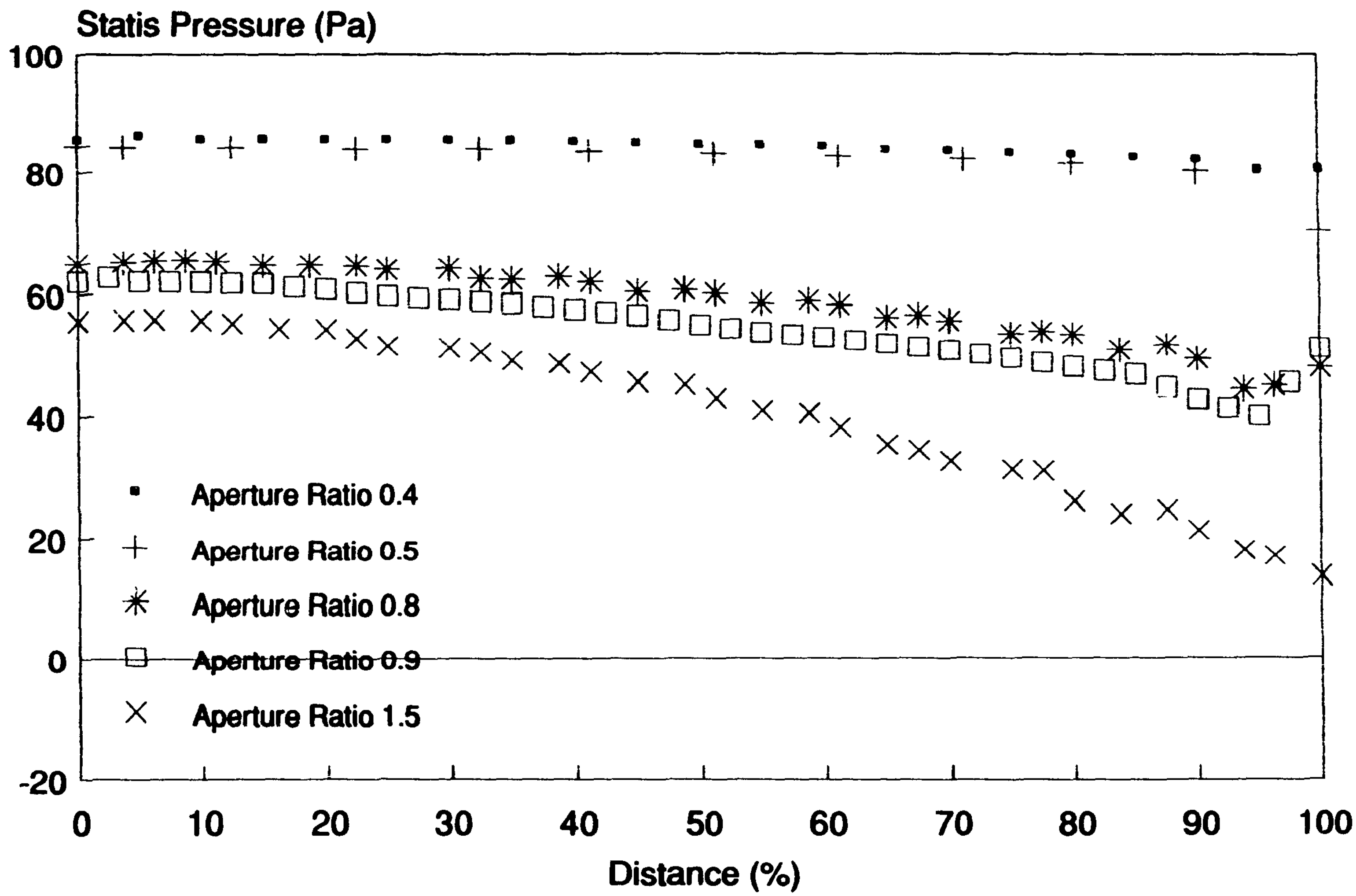

Figure 5.2.2.2. Duct Static Pressure with Distance along the Duct and as Function of Aperture Ratio

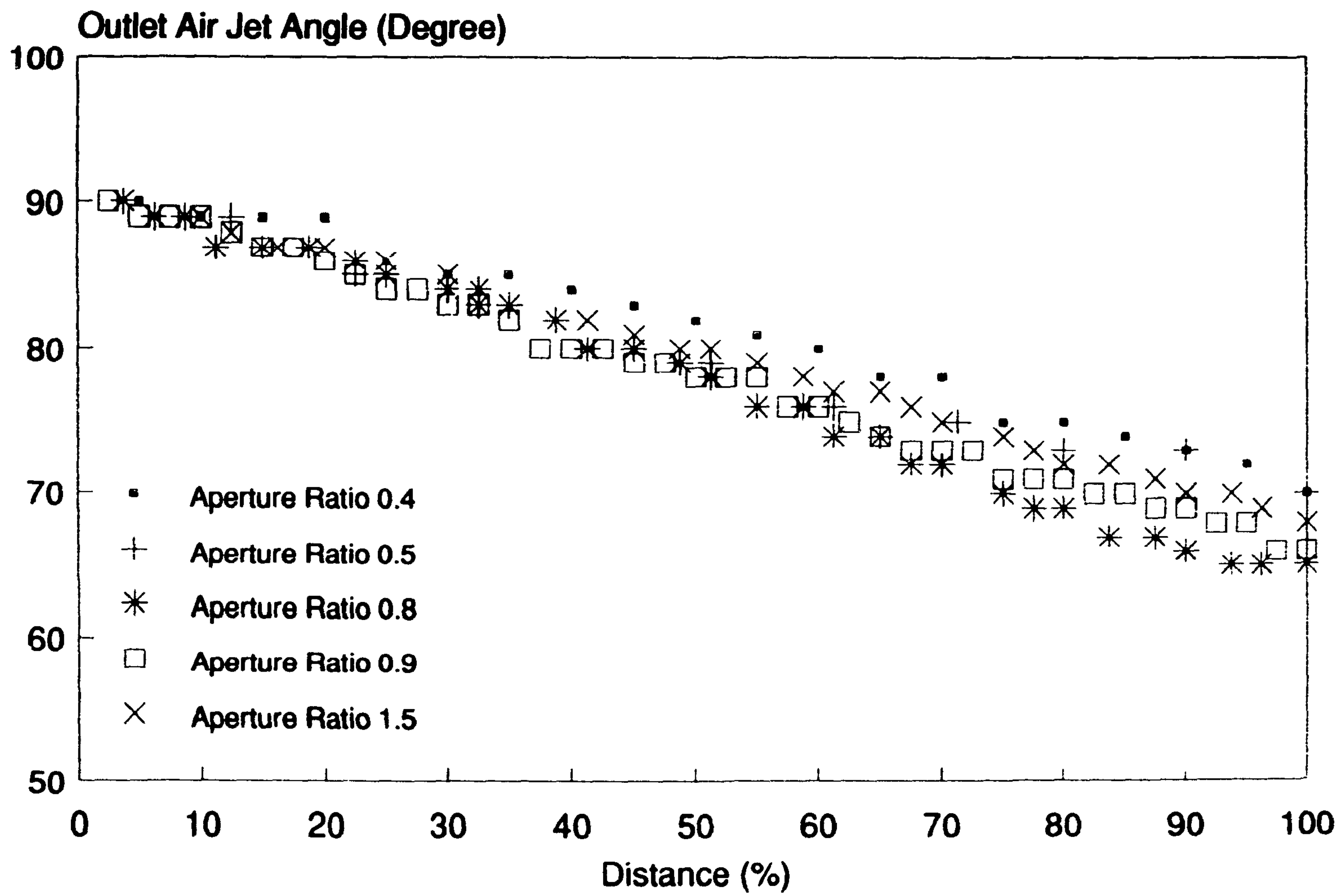

Figure 5 2 2.3 Outlet Air Jet Angle with Distance along the Duct and as Function of Aperture Ratio

Chapter VI

DISCUSSION

Analysis of the results lead to the following conclusions:

i). There is no significant difference in V value among duct material.

ii). There is no significant difference in V value among outlet shape.

iii). There is a significant influence on the V values from both distance along the length of the duct and aperture ratio.

iv). There is no significant difference between the measured data's regression equation of V_d and predicted equation of V_x, as obtained from equation [1] expressing the average duct air velocity.

V_d -- measured average duct air velocity, m/s

V_x -- predicted average duct air velocity, m/s

DISCUSSION

6.1. Testing the Mathematical Equation.

All the data measured from the fourteen (14) ventilation ducts was used to evaluate the mathematical equations developed to predict the average duct velocity, Vx.

6.1.1. Wooden Ducts.

The results will be presented in two (2) sections according to different duct length.

6.1.1.1. Three (3) Wooden Ducts, 16.8 m in length but with variable aperture ratios.

Air distribution parameters were measured using three (3) wooden ventilation ducts with a different aperture ratio. All predicted average duct velocity equations were developed using V_L, (the total outlet flow divided by the duct's cross sectional area), θ, the aperture ratio, Po, the static pressure at x=0, and Vo, the back outlet average velocity (Table 6.1). All measured average duct velocities were developed using SAS (Statistical Analysis System) program (Appendix B) to obtain the regression equations (Table 6.1). Figure 6.1 illustrates the measured and predicted average duct air velocity. The fit is excellent since only slight differences of the order of 0.0% to 1.2% occur at the centre length of the duct. This error remain within the error range of the experimental instruments. The statistical analysis results shows there is no significantly different between these two equation at 0.05 level (Appendix C).

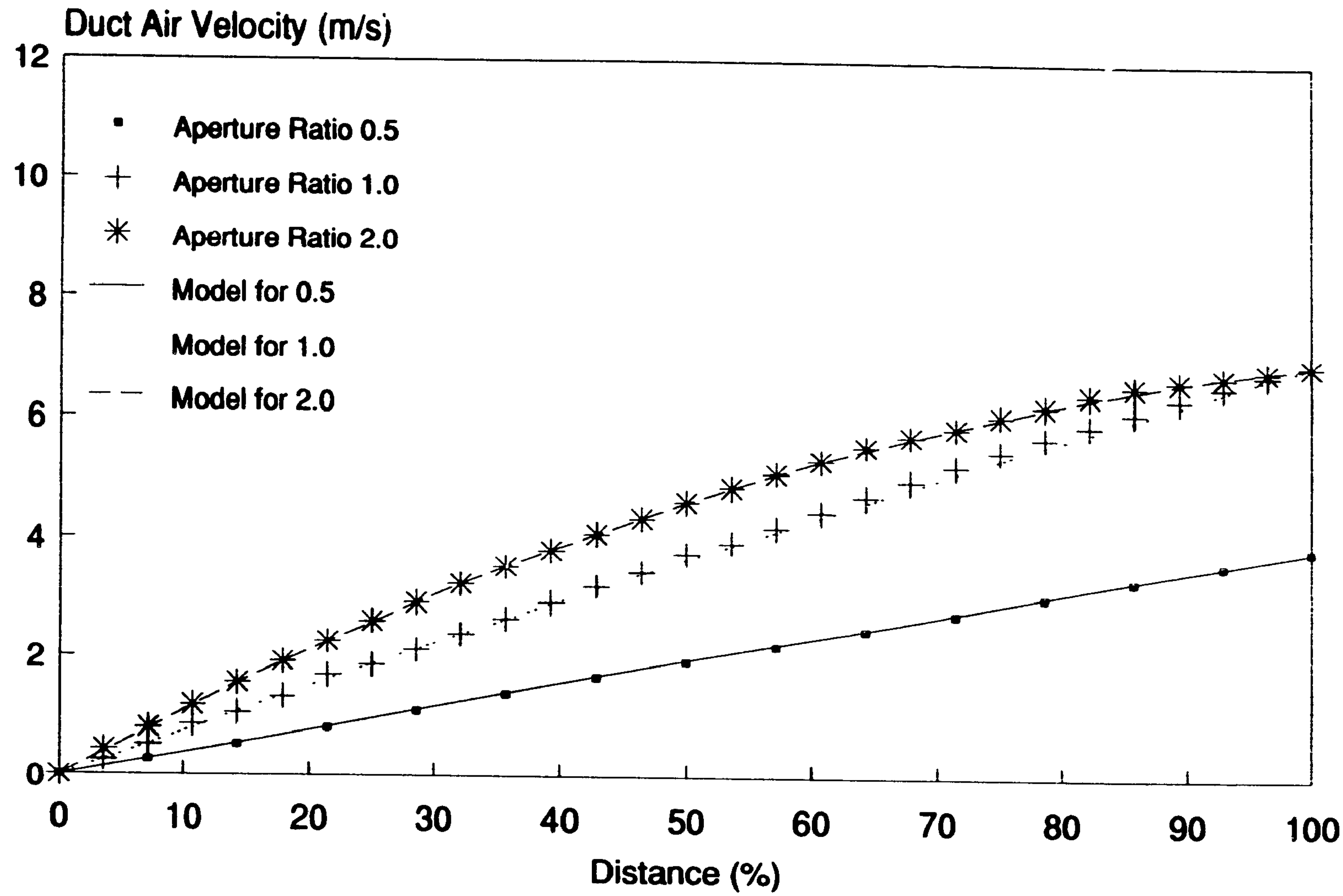

Figure 6.1. Duct Air Velocity with Distance along the Duct
and as Function of Aperture Ratio

Table 6.1 Regression Equation and Mathematical Model

Duct#	Regression equation	Mathematical model
1	$0.2320X - 0.00013X^2$	$0.2387X - 0.0005X^2$
2	$0.4747X - 0.0033X^2$	$0.4589X - 0.0028X^2$
3	$0.6868X - 0.0163X^2$	$0.6857X - 0.0163X^2$

6.1.1.2. Six (6) Wooden Ducts, 8.54 m in length but with two (2) different outlet shapes and three (3) different aperture ratios.

Air distribution parameters were measured using six (6) wooden ventilation ducts with a different outlet shapes and different aperture ratios.

All predicted average duct velocity equations were developed using V_L, (the total outlet flow divided by the duct's cross sectional area), θ, (the aperture ratio), Po, (the static pressure at x=0), and Vo, the back outlet average velocity (Table 6.2). All measured average duct velocities were developed using SAS (Statistical Analysis System) program (Appendix B) to obtain the regression equations (Table 6.2). Figure 6.2a.b.c. illustrate the measured and predicted average duct air velocity. All two shapes responded similarly for each aperture ratio by demonstrating almost indentical Vx values as a function of distance along the duct's length. The equation for Vx was found to give very good fit except for maximum error level of 0.85% at the duct's middle length. This error

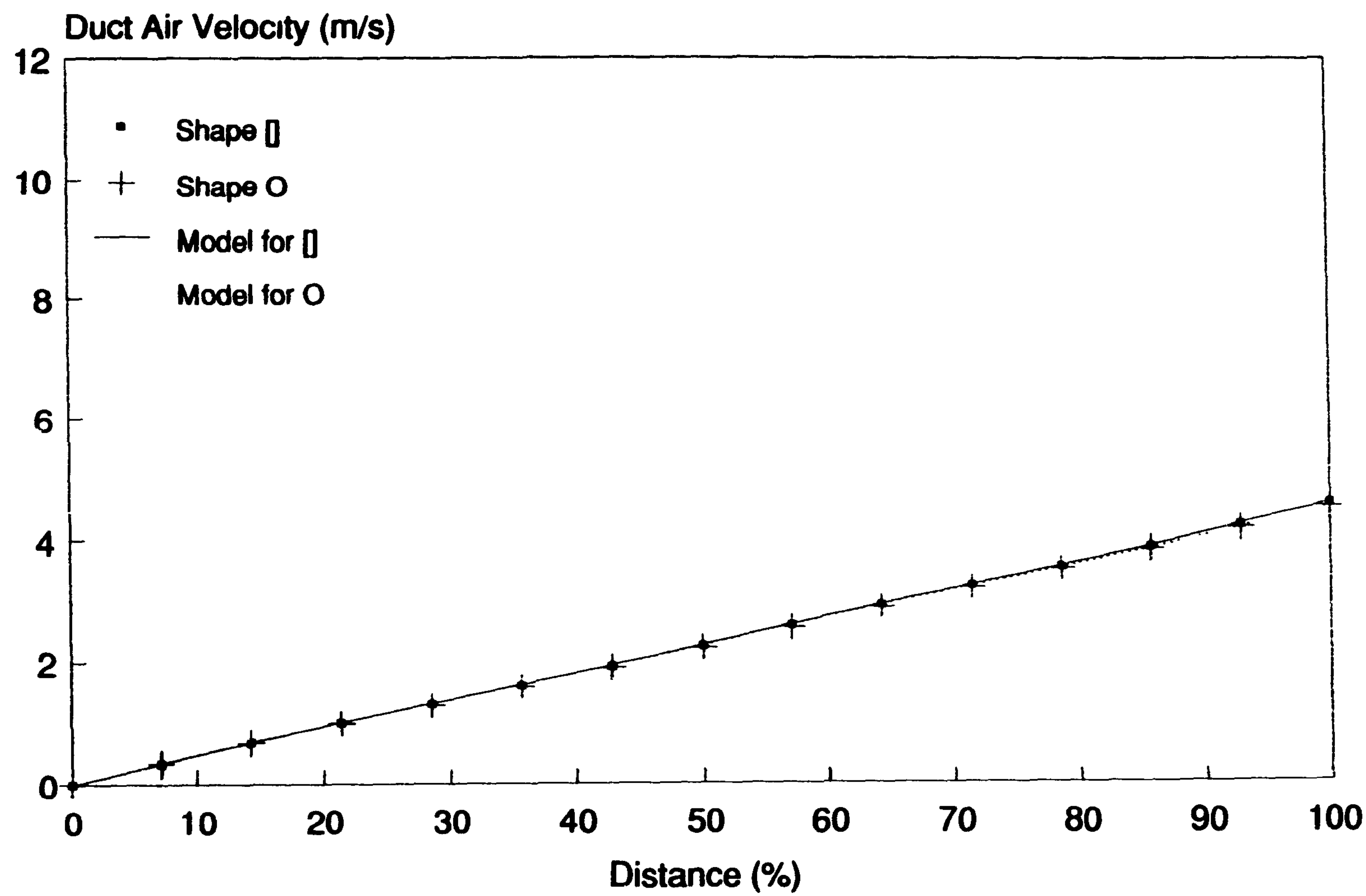

Figure 6.2a Duct Air Velocity as Function of Distance for Aperture Ratio of 0.5

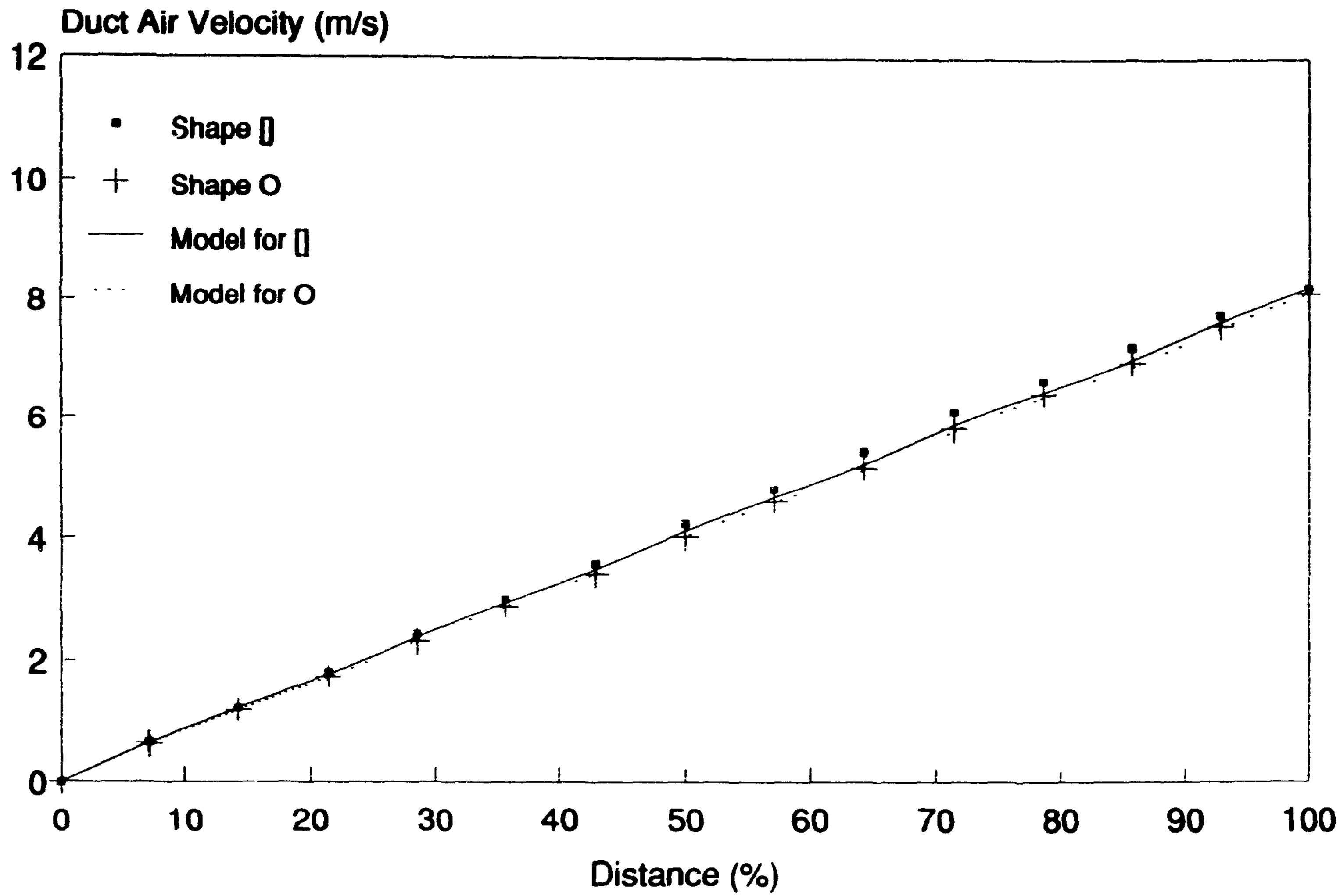

Figure 6.2b Duct Air Velocity as Function of Distance
for Aperture Ratio of 1.0

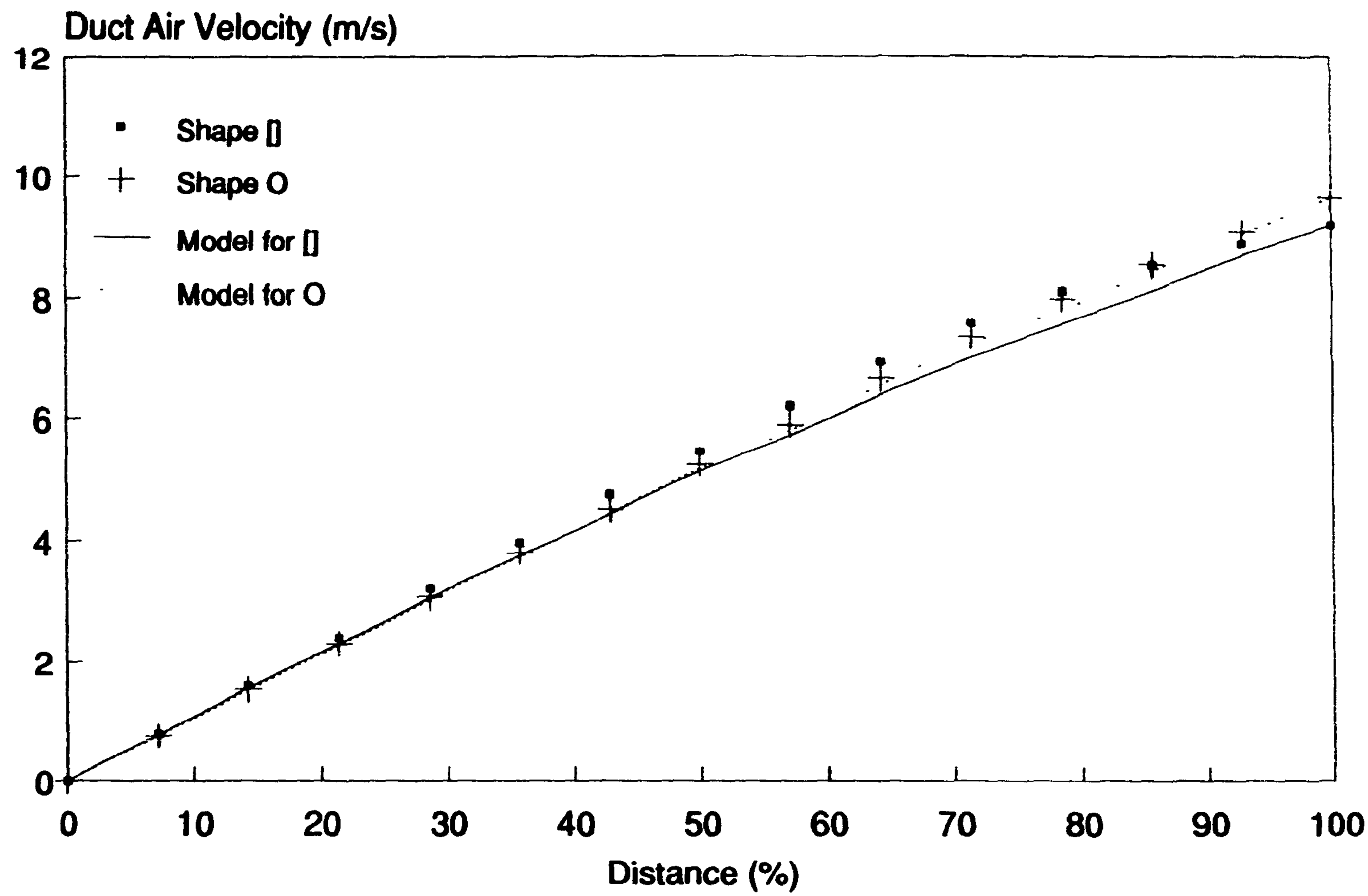

Figure 6.2c Duct Air Velocity as Function of Distance
for Aperture Ratio of 1.5

also remain within the error range of the experimental instruments. The statistical analysis results shows there is no significant difference between these two equations at 0.01 level (Appendix C).

Table 6.2 Regression Equation and Mathematical Model

Duct#	Regression equation	Mathematical model
1	$0.5172X - 0.0021X^2$	$0.5252X + 0.0013X^2$
2	$0.5089X - 0.0023X^2$	$0.5294X - 0.000013X^2$
3	$1.0493X - 0.0086X^2$	$0.9754X - 0.0021X^2$
4	$0.9531X + 0.00002X^2$	$0.9461X + 0.00013X^2$
5	$1.5369X - 0.0482X^2$	$1.3085X - 0.0263X^2$
6	$1.3313X - 0.0212X^2$	$1.2717X - 0.0155X^2$

6.1.2. Polyethylene Ducts.

Air distribution parameters were measured using five (5) polyethylene ventilation ducts with different duct length, aperture ratio and outlet spacing.

All predicted average duct velocity equations were developed using V_L, the total outlet flow divided by the duct's cross sectional area, θ, the aperture ratio, Po, the static pressure at x=0, and Vo, at x=0, the back outlet average velocity

(Table 6.3). All measured average duct velocities were developed using SAS (Statistical Analysis System) program (Appendix B) to obtain the regression equations (Table 6.3). The equations are illustrated against the measured data in figure 6.3. The average duct air velocity equation corresponded exactly with that measured. Differences between the measured and the theoretical values ranged between 0.5% to 1.25%, and this for all five polyethylene tubes. The statistical analyses results show there is no significant difference between these two equations at 0.01 level (Appendix C).

Table 6.3 Regression Equation and Mathematical Model

Duct#	Regression equation	Mathematical model
1	$0.2808X + 0.0004X^2$	$0.2909X - 0.00024X^2$
2	$0.3362X - 0.0002X^2$	$0.3451X - 0.00075X^2$
3	$0.4775X - 0.00074X^2$	$0.4691X - 0.000098X^2$
4	$0.2837X + 0.0001X^2$	$0.2793X - 0.000076X^2$
5	$0.7894X - 0.0044X^2$	$0.8566X - 0.0097X^2$

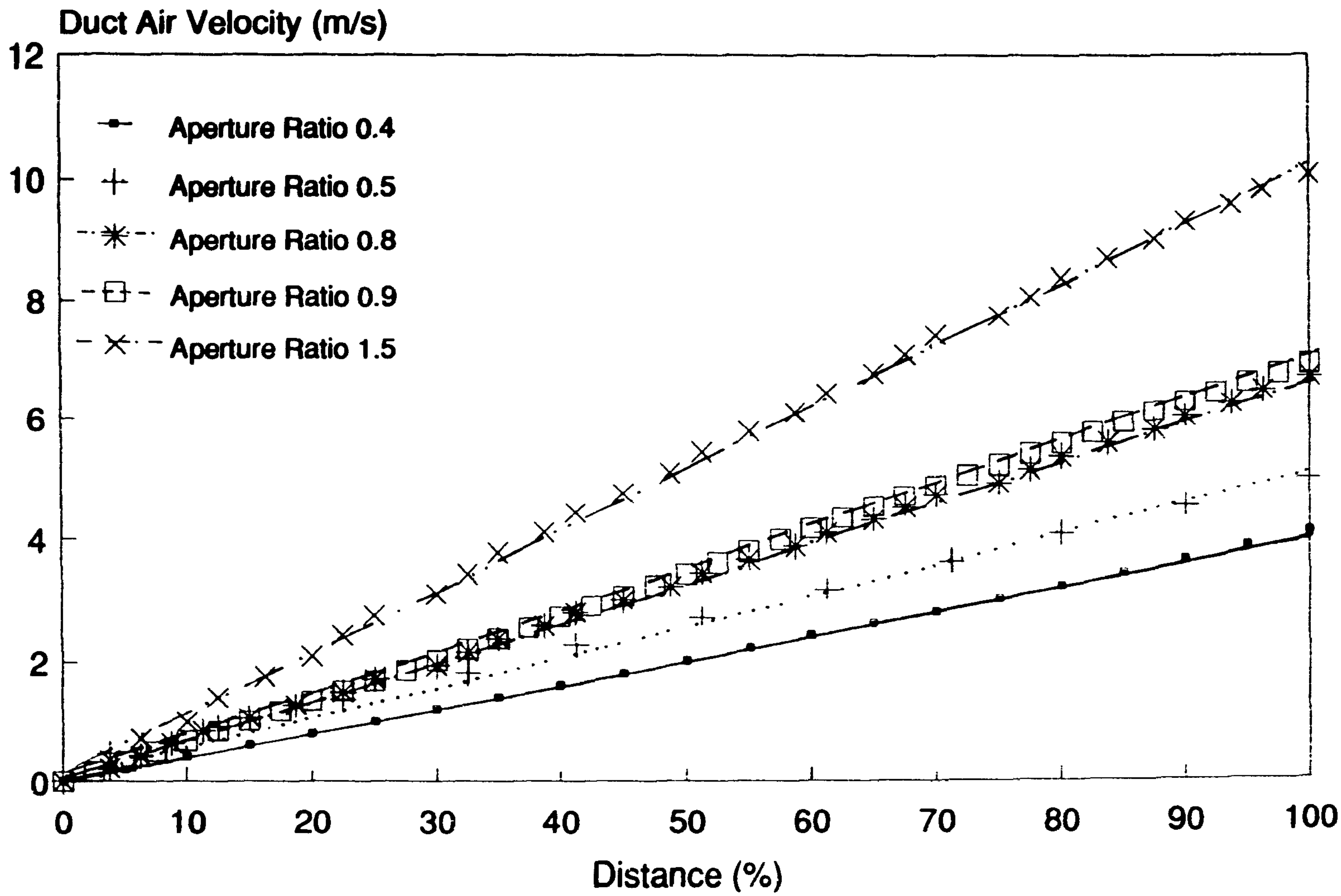

Figure 6.3 Duct Air Velocity with Distance along the Duct and as Function of Aperture Ratio

6.2. Comparison.

Air distribution parameters were measured using fourteen (14) ducts with different material, wooden and polyethylene, different outlet shapes, round and rectangular, different aperture ratios 0.5, 1.0, 1.5 and 2.0, different outlet spacing and at different points along the duct's length.

Statistical comparison of V values with respect to material, outlet shape, distance along the duct's length and aperture ratio using a split plot design (Appendix D) demonstrated that:

i). There is no significant difference in V value among material.

ii). There is no significant difference in V value among outlet shape.

iii). There is a significant influence on the V values from both distance along the length of the duct and aperture ratio.

iv). There is no significant difference between measured data's regression equation of V and predicted equation of V, ie. the mathematical model for the average duct air velocity fits the data.

6.3. Conclusions

Theoretical consideration of the fluid mechanics of air movement within a ventilation duct with a long slot outlet suggests the following model, for the average duct air velocity,

$$V = H_o \frac{X}{L} + (V_L - H_o) \frac{X^2}{L^2} \qquad [1]$$

where $Ho = \theta Co\sqrt{2Po/p}$

θ -- aperture ratio, dimensionless

Co -- discharge coefficient at X=0

Po -- static pressure at X=0, Pa

ρ -- air density, kg/m^3

L -- length of duct, m

V_L -- duct average cross-section velocity at X=L, m/s

This equation was successfully tested using data measured from 14 ducts with different material, outlet shape, aperture ration and length.

The average duct air velocity measured along ducts using two kinds of material (wooden and polyethylene), two outlet shapes (round and rectangular) and four size of outlets (aperture ratio are 0.5, 1.0, 1.5 and 2.0) indicated that aperture ratio and distance along the duct are the two most significant factors influencing V values. Material and outlet shape had no effect on the average air velocity of the duct.

6.4. Recommendations for Future Research and Modification of the Model

To make the regression equation more useful for predictive purposes, we should include in the regression equation all possible independent variables that may affect the dependent variable. However, due to the cost involved in obtaining information about all independent variables, only those independent variables that have significant effect on the dependent variable are included in the regression equation. Therefore, information is first collected about all possible independent variables and then only those variables that have significant effect on the dependent variable are selected.

Regression equations were obtained from measured data, for the fourteen (14) ducts, using SAS program with independent variables X, X^2 and X^3 (Appendix E). The results are shown in Table 6.4, and indicate that:

i). The nature of the response curves are likely to be linear on the ducts which have higher outlet spacing (lower aperture ratio such as 0.5). Therefore, for small aperture ratio (wide outlet spacing) balanced condition are defined. The average duct air velocity equation becomes:

$$V = N + M \frac{X}{L} \qquad [16]$$

The constants N and M can be defined by considering the boundary conditions.

ii). The variable X^3 had significant effect on the variable V where outlet spacing is short (higher aperture ratio such as 1.0, 1.5 and 2.0). The assumption made in order to develop the theoretical equation is that a single long slot allows for the outlet of air over the full length of the duct. The ducts with short outlet spacing (high aperture ratio such as 1.0, 1.5 and 2.0) produce condition where the X^3 coefficient is significant. Under this condition, the cubic of the distance along the duct may has an effect on the duct air velocity. Recall the equation [12] in chapter III:

$$V = G + M\frac{X}{L} + N\frac{X^2}{L^2} + S\frac{X^3}{L^3} + \ldots + \frac{X^n}{L^n} \qquad [12]$$

Considering the boundary conditions, the constants G, M, N and S can be defined (The four constants of the V equation can be fully defined from the four boundary conditions):

- at $x = 0$, $v = 0$

- at $x = L$, $v = V$

- at $x = 0$, $dV/dx = C(\theta/L)(2P/\rho)^{1/2}$

- there exists an x where $dV/dx = 0$; this condition can be used to define N and S.

TABLE 6.4 Nature of the Response Curve

Duct#	Outlet size mm*mm	Outlet spacing mm	Aperture Ratio	Nature of the response curve
1W1*	125*25	1200	0.5	Linear
P2**	D=69 MM	1365	0.5	Linear
1W2	125*25	600	1.0	Quadratic
2W1	125*25	600	0.5	Quadratic
2W2	D= 63 MM	600	0.5	Quadratic
2W3	125*50	600	1.0	Quadratic
2W4	D= 89 MM	600	1.0	Quadratic
P1	D= 47 MM	715	0.4	Quadratic
P3	D= 51 MM	460	0.8	Quadratic
1W3	125*50	600	2.0	Cubic
2W5	125*75	600	1.5	Cubic
2W6	D= 109 MM	600	1.5	Cubic
P4	D= 47 MM	620	0.9	Cubic
P5	D= 69 MM	450	1.5	Cubic

* 1W1 -- Experiment I (16.8m in length), Wooden Duct # 1
2W1 -- Experiment II(8.54m in length), Wooden Duct # 1

** P2 -- Polyethylene Duct # 2

Chapter VII

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Chapter VIII

APPENDICES

Appendix A

```
10 CLS:PRINT:PRINT:PRINT

20  A$="###.###   "

30  C$=" ##.    "

40  B$="##.##  "

50  F=##

55  SP=1:OPT$="Y:

60  INPUT " INPUT # OF OUTLETS = "; B

80  INPUT"INPUT OUTLET SPACING (IN METERS)";SP

81  INPUT "DATA FILENAME";F$:GOTO 100

83  INPUT :SIZE = :;S$

84  INPUT "SHAPE = ";SH$

85  INPUT "REP. # =";R$

87  F$=S$+SH$=R$

100 B=B+1

110 DIM P(B),v(B+2),O(B),C(B+2),K(B+2),E(B+2),G(B),T(B)

120 OPEN "b:"+F$+".prn" FOR INPUT AS #1

130 FOR Q=1 TO B:INPUT #1,O(Q),P(Q),V(Q):NEXT Q

140 FOR Q=B TO 1 STEP -1:PRINT O(Q),V(Q),P(Q):NEXT Q

143 FOR Q=1 TO B+1:VSUM=VSUM+V(Q)^2:NEXT Q :PRINT VSUM :S=VSUM

150 GOTO 230
```

```
160 D=1
170 INPUT"S=";S
180 INPUT"P=";P(D)
190 INPUT"V=";V(D)
200 INPUT:"O=:;O(D)
210 D=D+1
220 IF D<B+1 GOTO 180
230 D=D+1
240 K(1)=0
250 C(D+1)=1
260 V(B+1)=(V(D+1)+V(D))/2
270 G(D+1)=(P(D+1)+P(D))/2
280 T(D+1)=(O(D+1)+O(D))/2
290 W=V(D+1)-V(D)
300 U=P(D)-P(D+1)
310 LOCATE 20,10
320 PRINT:d=";D
330K(D+1)=(0.833*U+1.076*SP*F*V(B+1)^2+T(D+1)*W/C(D+1)+K(D)*V(D)^2)/
(V(D+1)^2)
340 E(B+1)=G(D+1)/1.2+(K(D)*V(D)^2+K(D+1)*V(D+1)^2)/2
350 C(B+1)=T(D+1)/(E(B+1)^0.5)
```

```
360 R=(C(D+1)-C(B+1))*(C(D+1)-C(B+1))

370 R=R^0.5

380 IF R<0.001 GOTO 410

390 C(D+1)=C(B+1)

400 GOTO 330

410 E(D+1)=P(D+1)/1.2+K(D+1)*V(D+1)^2

415 C(D+1)=O(D+1)/E(D+1)^0.5

420 D=D+1

430 IF D<B GOTO 250

440 CLS

450 PRINT"      ";F$

470 PRINT

480 PRINT "OUTLET  Vo  Vd  Pd  C  E  K "

500 D=2:BB=B-1

504 GOTO 520

505 IF P(D)<0 THEN GOTO 520

510 Z=O(D)*O(D)/E(D)^0.5/(P(D)/1.2)^0.5

520 PRINT USING C$;BB;

530 PRINT USING B$;O(D),V(D),P(D);

540 PRINT USING A$;C(D),E(D),K(D)

550 D=D+1:BB=BB-1
```

```
560 IF D<B+1 GOTO 520

570 FOR P=1 TO 2:PRINT:NEXT P

575 IF OPT$="N"THEN GOTO 590 ELSE GOTO 600

590 END

600 OPEN "b:"+F$+"OUT"+".prn" FOR OUTPUT AS #2

675 FOR Q=2 TO B

685 PRINT #2,O(Q),V(Q),P(Q),C(Q),E(Q),K(Q)

695 NEXT Q

725 CLOSE #2

727 LPRINT"  Vo   Vd   P   C   E  K

729 FOR Q=1 TO B:LPRINT O(Q),V(Q),P(Q),C(Q),E(Q),K(Q):NEXT Q

735 RETURN
```

Appendix B

SAS STATEMENT:

```
DATA;
INPUT DISTANCE VELOCITY;
DSQ=DISTANCE**2;
CARDS;
*DATA LINES
PROC GLM;
MODEL VELOCITY=DISTANCE DSQ;
RUN;
QUIT;
```

Appendix C

Interrelationship between regression and experimental design:

Comparison of fitted models (Berenson et al., 1983, 326-364)

In cases, where all responses are linear in nature, a T-test of the intercepts and slopes of two lines will do the job.

However, if the responses are not linear in nature, all regression parameters need to be tested to determine the similarity of all regression curves involved.

The easiest way to compare fitted models is to use analysis of covariance.

SAS STATEMENT:

```
DATA;
INPUT T X Y;
XSQ = X**2;
** TREATMENT T HAS 2 LEVELS:1, MEASURED DATA 2,DATA FROM
MODEL;
PROC SORT;
BY T;
PROC GLM; BY T;
MODEL Y=X XSQ / SOLUTION;
```

```
PROC REG; BY T;
MODEL Y=X XSQ / SELECTION=STEPWISE SLS=0.05;
** THE ABOVE STATEMENTS ARE TO GET THE APPROPRIATE REGRESSION RESPONSE FOR EACH TREATMENT;
PROC GLM;
CLASSES T;
MODEL Y=X XSQ X*T XSQ*T / SOLUTION;
** THE ABOVE STATEMENTS ARE TO TEST THE DIFFERENCES AMONG THE REGRESSION RESPONSES OF EACH TREATMENT. IF TREATMENT EFFECT HAS NO EFFECT, THEN THE FITTED MODEL OF EACH TREATMENT ARE NOT DIFFERENT FROM EACH OTHER.
RUN;
QUIT;
```

Appendix D

SAS STATEMENT:

```
DATA;
INPUT SHAPE RATIO LENGTH V;
CARDS;
*DATA LINES
PROC PRINT;
TITLE '9 WOODEN DUCTS';
PROC GLM;
CLASSES SHAPE RATIO LENGTH;
MODEL V=SHAPE RATIO LENGTH;
TITLE 'ANALYSIS OF DIFFERENT SHAPE AND LENGTH OF WOODEN
DUCTS';
DATA;
INPUT MATERIAL RATIO LENGTH V;
CARDS;
*DATA LINES
PROC PRINT;
TITLE '3 WOODEN AND 5 POLYETHYLENE DUCTS';
PROC GLM;
CLASSES MATERIAL RATIO LENGTH;
```

```
MODEL V=MATERIAL RATIO LENGTH;
RUN;
QUIT;
```

APPENDIX E

SAS STATEMENT

```
DATA;
INPUT X Y;
XSQ = X**2;
SCU = X**3;
PROC REG;
MODEL Y = X XSQ XCU /SELECTION = STEPWISE;
TITLE' SELECTING THE BEST REGRESSION EQUATION BY STEPWISE
TECHNIQUE';
RUN;
QUIT;
```