AIR MOVEMENT AND ENERGY FLOWS IN AN AIR-CONDITIONED AND PARTITIONED INDUSTRIAL ENVIRONMENT

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TO MY PARENTS

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(ADRIAN C. PITTS)

SUMMARY

This study concerns an investigation into air movement and associated energy flows within the environment of a synthetic fibre producing factory. A multiplicity of air-conditioning and ventilation systems were operated within the factory to provide a suitable atmosphere for the yarn, and also to allow some degree of comfort in hot production areas. Potential for improved operation of these systems was anticipated.

Initial experiments showed certain anomalies and problems relating to air conditions and air movement; and an important facet of the production areas was identified as the regular partitioning created by the machine layout.

A review of previous studies of building air flows indicated a lack of information relating to industrial and partitioned areas. Mathematical relationships for air flows were studied and the interactions of similar, closely spaced partitions were considered.

A series of model scale tests using simple layouts supported a theory of interaction. The effect was substantial for wall type partitions and a considerable overestimation could result from the simple additive approach to determination of total resistance.

At the factory a computer based monitoring scheme was designed and installed in order to establish environmental conditions and energy flows. The concept of "total thermal efficiency" was developed as a means of evaluating the performance of some of the air-conditioning systems. Considerable variations were evident between seasons and between systems; improvements being possible and recommended.

Air flows were also investigated using Nitrous Oxide as a tracer gas. The effect of the internal partitioning combined with the high degree of ventilation and air-conditioning was to "compartmentalize" the spaces between the machines in the production areas, semi-isolating each from its neighbours. Thus, the results of the simplified model scale work could not be applied directly. However the isolation of the spaces offers potential for better systems operation by reducing air-conditioning requirements.

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CHAPTER 1

INTRODUCTION

1.1 SYNTHETIC FIBRES AND ICI

Man-made fibres, both regenerated and synthetic, are a relatively recent development, when compared with the thousands of years that the natural alternatives have been in use. All the significant industrial developments of man-made fibres have taken place this century and it was only in the late 1930's that nylon (said to be the "original" synthetic fibre) came to light, with the first commercial factory opening in the USA in 1940. Polyester fibres were developed slightly later than nylon, though techniques soon "caught up" due to the wide range of possible uses anticipated for polyester, despite the greater development work required.

Synthetic fibres have proved very useful in the modern world, often being harder wearing and more adaptable than natural fibres. Uses range from clothing and carpet weaving to tyre and rope manufacture. This usefullness produced a demand which could not be met in the early years of production, both during and following the Second World War. Expansion in production capacities, first in the United States and then Britain was followed by other countries around the world. This rapid expansion has led to some of the industry's current problems since plants were built and extended with more regard for immediate capital costs than future running costs.

Coal was the original source of the organic material required for synthetic fibre production, but this was soon replaced by an oil feedstock. Thus OPEC induced price rises of the early 1970's had an immediate effect on costs. Since the synthetic fibre production processes are themselves very energy intensive, the price rises had a doubly bad effect. With the occurrence in recent years of British and world economic recessions, synthetic fibre manufacturers have found themselves in a difficult position and operating in an increasingly competitive market. Many companies have sustained significant operating losses, whilst others have ceased production. The industry has recognised the need for improved efficiencies and it is in the field of energy comsumption that an opportunity for significant savings is seen.

One of this country's largest manufacturing groups is Imperial Chemical Industries, one division of which is ICI Fibres. ICI Fibres produce synthetic fibres both nylon and polyester types, brand names of which (such as "Bri-nylon" and "Terylene") are household names. Manufacture occurs at a number of sites within the United Kingdom and overseas, the Head Office being at Harrogate, North Yorkshire. Fibres are generally produced in one of two forms either as "continuous filament yarn" which after stretching can be used in "smooth" applications, eg, ropes, or in a crimped, cut, short length form known as "staple" which can be processed in a similar way to natural fibres for such products as clothing. The form and nature of the yarns produced are usually determined by the intended use. Table 1.1 indicates the production carried out at the main ICI plants.

TABLE 1.1

ICI FIBRES SITE PRODUCTION

LOCATION	TYPE OF FIBRE PRODUCED
Pontypool	Continuous Filament Nylon Continuous Filament Polyester
Doncaster	Continuous Filament Nylon Staple Nylon
Gloucester	Continuous Filament Polyester Continuous Filament Nylon
Wilton	Staple Polyester
Oestringen (West Germany)	Continuous Filament Nylon Staple Nylon Continuous Filament Polyester

1.2 DONCASTER SITE

One of the factories in the ICI Fibres group is at Doncaster in South Yorkshire. It now produces nylon, though the original building was erected by British Bemberg company in 1929 to produce cuprammonium viscose rayon. In 1955 the site was taken over by British Nylon Spinners (a company partly owned by ICI) and major extensions were added in the late 1950's and early 1960's. In 1964 ICI took full ownership of British Nylon Spinners and it was incorporated into the Fibres Division in 1966.

The basic processes carried out at Doncaster use nylon polymer "chips" as feedstock. These nylon chips are melted by "Thermex" heat transfer medium at about 300°C and are extruded to form filaments. The filaments are quench cooled to allow solidification and are then collected, either wound-up onto a "cake" for continuous filament yarn, or in bulk in large bins for staple yarn. (The above sections of the process take place in a part of the factory known as the "Spinning Tower"). The nylon yarn is then taken to other areas of the factory to undergo further processing (which depends on the type of fibre and its intended use) and storage before despatch to customers.

Because the factory was not purpose built and because the layout is not ideal, rationalization of production and changes in production machinery seem to suggest opportunities for improved operation.

1.3 ENERGY

It was recognised that the Doncaster plant consumed significant amounts of energy in the forms of natural gas and electricity. Between 1973 and 1979 this amounted to an average of 233,000 MWh (electricity and electricity equivalent) per year[1]. However during this period though energy use remained relatively constant, the cost of providing it increased from £644,000 to £2,240,000 and the energy cost of each tonne of Grade 1 nylon produced rose from £13.70 to £55.06. Figure 1.1 shows energy use, production and price index for the given period.

These price increases and the prevailing economic climate prompted the appointment of a full time Energy Utilisation Engineer and the initiation of energy monitoring and conservation measures. The main areas of energy use can be identified as follows. Almost all of the total gas consumption is accounted for by two main systems: the first being high pressure hot water boilers which provide steam and space heating across the plant; the second being the gas fired boilers used to heat the Thermex fluid used in the nylon melting process. The largest proportion of the electricity consumption is used in the provision of air conditioning and mechanical ventilation in the various process areas. Power for the machines and lighting in these areas also takes a significant fraction of the total. In 1979 the budget allocation of energy costs, area by area within the factory, showed that when all ancillary and service equipment was taken into account, the Spinning Area was responsible for 62.2% of the site total[2].

Improvements in this area with regard to efficient energy utilisation would undoubtedly show significant cost advantages.

The Spinning process at Doncaster is associated with the Spinning Tower building and the attached Boilerhouse. There are three main aspects to the spinning process energy consumption. The first arises from the need to heat and melt the polymer; three Thermex circuits exist to supply the spinning machines with the required heat. In order to melt the nylon, the Thermex is kept up to a temperature of approximately 300^{o}C by a number of gas fired furnaces. Because of the long pipe runs and the spread out nature of the spinning machines, which allow heat losses, this melting process has a very low thermal efficiency. Secondly the spinning machines require power in order to deliver the polymer chips to the melting grid and to collect the "spun" filaments after extrusion, as well as to run the spinning machines themselves. Lighting for the production areas could also be classified under this heading. Thirdly there is a need for various air conditioning and ventilation systems around the spinning process for three reasons:-

i) The melting of the polymer releases a great deal of heat into the environment from exposed hot metallic surfaces. To prevent overheating of personnel and machinery heat must be removed and cooling carried out. This requirement is met by a number of ventilation systems using many fans to supply fresh air and exhaust warm air.

ii) In the yarn collection areas, environmental control of temperature and humidity is needed to prevent degradation of the fibre and preserve its properties(made use of in subsequent processing). A number of large air conditioning plants operate to provide the necessary control; these incorporate fans to supply and extract considerable volumes of air.

iii) Another air movement system exists to provide air to quench cool the extruded and solidifying nylon filaments. This air is supplied to the spinning machines by a multiple fan system.

To recap then, the major energy demands for the spinning process are:

a) Gas for the boilers supplying Thermex/nylon melting process

b) Electricity to power spinning machines, ancillary equipment and area lighting

c) Electricity to power the fans, etc, of the air conditioning and ventilation systems.

In 1980, a period of low demand and hence low production allowed a major reorganisation of the spinning area to be carried out. This effectively made one section of the plant surplus to requirements by concentrating production into a more compact area using higher throughput machines. This resulted in an immediate reduction in gas consumption since it became possible to operate the process using two Thermex heating circuits instead of three. The concentration of production also made more efficient use of spinning machines and reduced electricity demand. "Good housekeeping" measures instigated throughout the plant have allowed further savings. It was recognized however that if improvements were to be made with the third major energy use, that of air conditioning and ventilation, fairly detailed investigations would be required.

Discussions with the Department of Building Science at the University of Sheffield, were already taking place concerning energy related matters. This link provided the basis for the setting up of the current research work, for which it would be necessary to gain a working knowledge of the plant and processes involved in general and in particular of the Spinning Tower and spinning process. Familiarity with the air conditioning and ventilation systems and their operation would also be required. Environmental conditions within the process area would be of great importance as would the energy transfers either through the building fabric or due to the mechanical ventilation systems. Air movement in general in such an industrial environment would also have to be investigated.

The major British and American design guides are sadly lacking in information regarding retrospective (and also to a large degree, initial) optimisation of air movement systems in industrial situations. Information gained from work carried out at ICI Fibres, Doncaster would therefore not only be useful in similar fibre producing environments, but also in a wider range of industrial and similar process industries.

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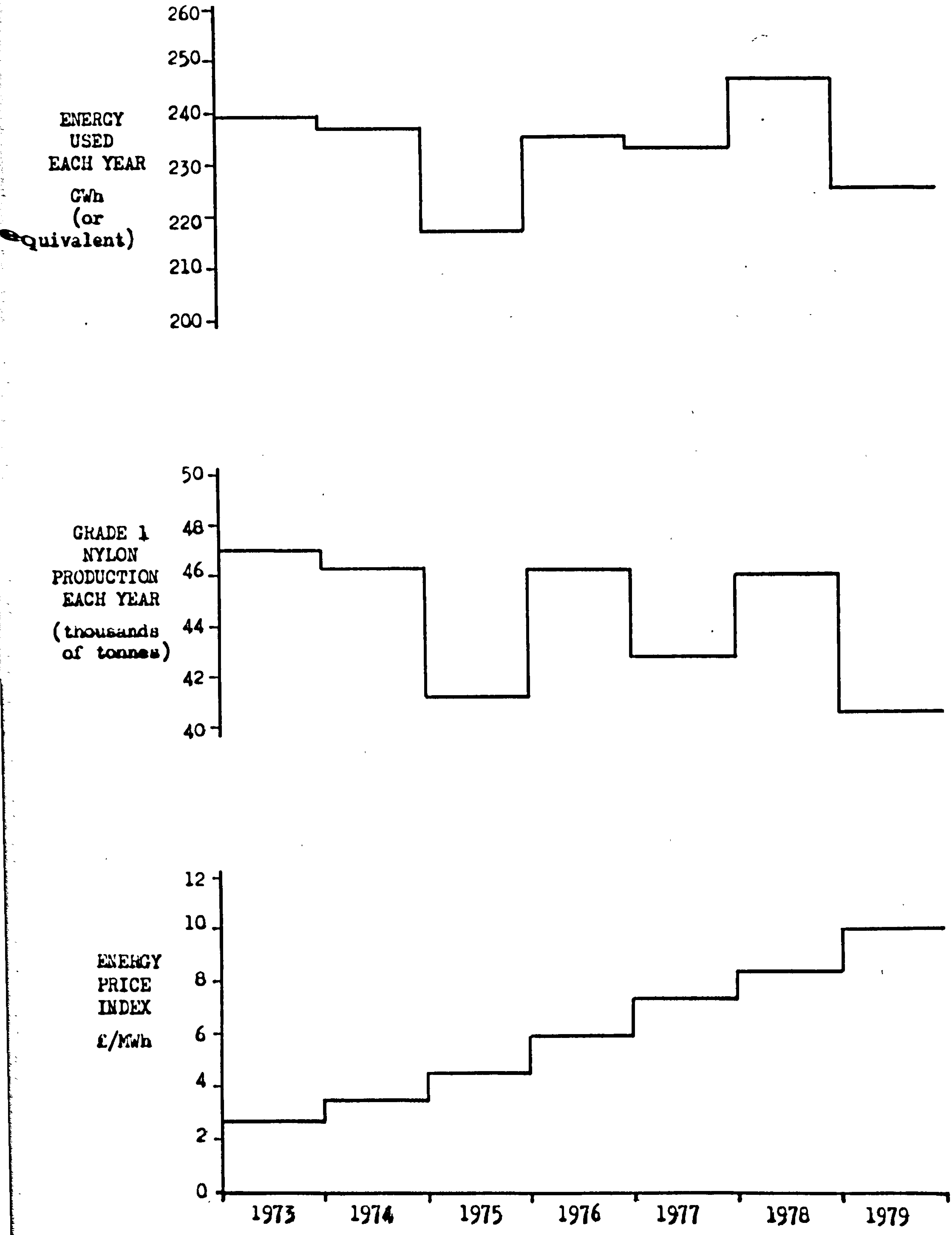

FIGURE 1.1 VARIATIONS IN ENERGY USE, NYLON PRODUCTION AND ENERGY PRICE INDEX (1973-1979) (1)

CHAPTER 2

INVESTIGATION AND ASSESSMENT OF PLANT AND PROCESS AT ICI FIBRES, DONCASTER

2.1 INTRODUCTION

Before more detailed studies could be undertaken it was decided that a knowledge of plant layout, building construction and production process at the Doncaster factory, would be required. Energy consumption figures were to be examined and some experimental investigations carried out at the site. A description of the results of this work is given in this chapter.

2.2 THE NYLON PROCESS

Dr Wallace H Carothers is credited with the discovery of nylon whilst working with the rapidly expanding American chemicals company, EI DuPont de Nemours, in the 1930's. The first commercially viable form was produced in 1935, but the announcement of its synthesis was not made until 1938. Nylon is the generic name for a group of substances known as synthetic polyamides. Many different types of nylon can be produced, but only four are suitable for commercial manufacture, these being Nylon 6, Nylon 6.6, Nylon 6.10 and Nylon 11. (The numbers referring to the number of carbon atoms in the basic chains making up the polymer). Nylon 6, and 6.6 are the most common types in use, with

ICI Fibres producing mostly Nylon 6.6.

Du Pont granted ICI a licence to manufacture nylon in 1939 and in 1940 ICI and Courtaulds set up the company of British Nylon Spinners to deal in fibres and yarns derived from nylon polymer. At that time, there was immediate demand for nylon for military purposes, such as parachutes and aeroplane tyres, and the first plant was rapidly build, in Coventry. Expansion was quick to follow and a plant at Pontypool was soon opened. Besides the war-time uses for nylon, there was a large civil market especially for clothing and other fabrics. During the 1950's and 1960's further plants were opened including the take over of the Doncaster site in 1955.

The production of Nylon 6.6 is quite complex and uses the basic raw material, benzene (obtained from coal or crude oil) which has a molecule containing six carbon atoms. Benzene is converted to adipic acid and hexamethylenediamine, which are reacted together to form the basic Nylon 6.6 salt. The salt is then polymerized by reaction with itself at high temperature and pressure in an autoclave. The polymer is extruded from the autoclave in the form of a liquid ribbon. This is quench cooled by water and is then broken into "chips" by special cutters. It is in this form that the polymer arrives on site at Doncaster in containers known as "totebins".

2.3 THE SPINNING PROCESS

The nylon spinning process as carried out at the Doncaster site can be divided into three main stages:

2.3.1 MELTING

Polymer chips, contained within hoppers above individual spinning units, are fed by gravity to the top of the unit. The size and moisture content of the chips must be within certain limits to allow an even flow of polymer to the melter. A screw mechanism feeds the polymer to the melting grid which is kept at a temperature of 289^{o}C using a Thermex vapour heating medium. Since molten nylon degrades rapidly in air, air must be excluded from the spinning unit. This is achieved by allowing moisture within the chips to vaporize and so form a steam atmosphere. The polymer melts and flows into the melt pool which is provided with a steam blanket. The level in the melt pool is kept at a reasonably constant level by an electronic sensing system which controls the polymer chip feeding mechanism. The residence time in the melt pool is also kept reasonably constant. The melt pool is stirred and drains into a booster pump which ensures that polymer is available at sufficient pressure for the extrusion process.

2.3.2 EXTRUSION

After the booster pump, metering pumps are used to deliver a set amount of polymer into a "pack".

A pack consists of mineral filter media and a spinneret; it is used to filter impurities from the molten nylon and give an homogenous mixture. The nylon is forced through the holes of the spinneret to form filaments and a blanket of steam is provided at the spinneret face. The number of separate filaments depends upon the number of holes in the spinneret. The filaments are cooled in what is termed a "chimney" by air ("blower air") being blown across them in a controlled manner. Too fast or too slow a cooling will produce yarn that is unsuitable for subsequent processing end use. The bundle of filaments is converged at the bottom of the chimney and passes into the conditioner tube.

2.3.3 WIND-UP/YARN COLLECTION

Nylon yarn is prone to static electricity production, so to help prevent this between the convergence guide and wind-up point, a conditioner tube is provided. This basically contains a column of steam. The steam is prevented from being drawn through with the yarn by the maintenance of a slightly higher air pressure in the wind-up area (also called the Spin Doff Area) than the Extrusion Area.

The yarn leaves the conditioner tube, passes through guides and over a rotating glass cylinder where it is given a coating of chemicals called "spin finish". This provides (i) anti-static properties (ii) cohesion (for the binding of filaments together) and (iii) the correct bulk frictional properties for the yarn. The type of spin finish applied varies according to the subsequent processing to be carried out.

The Tex of a yarn is a measure of its linear density and is the weight in grams of 10,000 metres. It is determined by the quantity of molten nylon pumped through the spinneret and the speed at which it is collected. An increase in molten nylon (other things being equal) will increase the Tex, whilst an increase in collection or wind-up speed will slightly draw the nylon and reduce the Tex.

2.4 FORM OF YARN

The two forms of yarn, staple and continuous filament (as mentioned in Chapter 1) are quite different.

2.4.1 STAPLE

Nylon Staple is a form of nylon that can be used with equipment designed for natural fibres and for this the nylon is required in short lengths. The filaments of nylon, produced in the spinning process, are all gathered together (from a row of perhaps ten spinning units) and collected as "Tow" in a large metal bin known as a Tow can. Full cans are then taken to the Staple Area of the factory where the yarn is drawn (stretched) to about four times its original length. Next it is crimped and steam set, then cut into short lengths and baled prior to despatch to customers.

2.4.2 CONTINUOUS FILAMENT

The nylon yarn produced in the spinning process is wound onto a "cake", at the bottom of the spinning unit, in a similar manner to string wound onto a bobbin. This yarn must undergo a draw-twisting process before it can be used. The original spun yarn is drawn to about four times its initial length and as it is wound up a twist is given which holds the filaments together in a more compact and easier-to-use form.

2.5 PLANT LAYOUT DESCRIPTION

The ICI Fibres plant at Doncaster, covers quite a large site. A plan of the site showing the main areas is given in Figure 2.1.

The spinning process takes place in that area of the factory known as the "Spinning Tower". Basically, this building is of rectangular plan, with its longest axis running almost due East-West When referred to, it is usually split into two areas known as the "Type 8" area and the "Type 14" area. These names derive from the main type of spinning machines found in each area; however it also distinguishes the age of the parts of the building. The Type 8 area consists of the original 1929 building together with changes and extensions made up to 1958. It rises to a height of approximately 33 metres over six storeys. However it is only in the lower three storeys where the spinning process itself occurs. The major extensions of 1963-4 form the Type 14 area, which rises only to three storeys.

Since the reorganisation of the factory during the last few years, production at the western end of the spinning area has been curtailed. Machines 3 and 4 have been removed and machines 5 to 10 are now idle. Machines which continue to run are 11 to 22 in the Type 8 area, and 25, 27 to 42 and 47 to 52 in the Type 14 area.

The spinning machines themselves are aligned in a North-South direction, with each machine consisting of a number of spinning units.

In order to give an idea of the plant layout, each floor level will be described below, noting main features. Reference to Figure 2.2 may also be made since this shows a diagramatic sketch of the Spinning Tower.

2.5.1 FIFTH FLOOR

This is the top floor level, it extends only over the Type 8 part of the building. Part is inside and part outside, on the roof over the fourth floor.

Inside are the tops to the ten main polymer chip storage bunkers, each of capacity 100 tonnes.

The polymer chips are pneumatically transferred up from ground level, each bunker being served by its own delivery tube. The air used in the delivery is cleaned after depositing the polymer and before being exhausted to the outside.

On the roof outside there are a number of flues and small extract duct outlets. The largest flue carries the combustion gases from the boilers used to heat the Thermex (which melts the nylon). These boilers are contained within a boilerhouse adjacent to the Spinning Tower. Other flues dispense the waste gases from the steam-air furnaces (situated three floors below).

There also used to be on the roof, five large air intakes for fans located at the 3rd floor level. Each intake contained an integral water spray section for evaporative cooling during the summer months. The ducts descended along the outside of the building and then each split to serve two fans. These intakes and ducts are being removed and where required, replaced by a water spray unit situated on the outside wall at a lower level, adjacent to each fan.

Various water storage tanks are also located, both inside and outside, on this floor.

2.5.2 FOURTH FLOOR

The main activity which takes place on this floor is the production of "Spin Finishes". The plant for the production of these chemical applications is located along the northern side of the floor area. Chemicals required for the process are also kept at this level; the highly viscous components being stored in the vicinity of the Thermex flue. The plant connects directly to the spinning machines at ground floor level, to provide the correct finish for the type of yarn being produced.

Along the southern side of the floor area are the ten main polymer chip bunkers, which also extend down to the floor below.

Part of a waste-heat recovery system is positioned next to the Thermex flue. This system was never commissioned because it could have reduced the chimney draught on the boilers, and possible caused an automated boiler shutdown. The costs and penalties of such a shutdown were deemed too great, by comparison with the benefits of heat recovery, and the plant has never been completed. As with the fifth floor, this floor exists only over the Type 8 area.

2.5.3 THIRD FLOOR

Along the northern side of the third floor are eleven "Gallery" fans. Ten of these fans used to draw air from intakes (previously mentioned) located at fifth floor level, but only fans numbered 6 to 11 remain in use and their intakes have been replaced by inlets which protrude through the outside wall only a short way. These fans deliver air to the first and second floor levels.

The tapering bottom sections of the large polymer chip bunkers are located along the southern side of the area. These feed down through the floor. There are also four smaller storage bunkers and four sets of "Tipplers". The Tipplers allow the contents of a specific, nylon containing, "totebin" to be delivered to the floor below.

Between two of the smaller bunkers a "Sortex" machine operates. This sorts low grade or reject polymer chips into usable and unusable fractions.

Adjacent to the Thermex flue, which "zig-zags" its way up through this floor, is the major portion of the unused waste heat recovery system.

Amongst the bunkers are a number of small ventilation systems which serve specific offices and other small areas on floors below. Since their capacities are small, and their use not directly involved with the main production areas, they will be disregarded in subsequent references to air conditioning and mechanical ventilation plant.

There is access from the Type 8 area to the roof over the Type 14 area. On the southern side of this roof are twenty-one fans drawing air from the first, and to a lesser extent from the second, floor areas below. On the northern side of the Type 14 roof are inlets leading down to the main 8 plants which provide the air conditioning of ground floor area.

2.5.4 SECOND FLOOR

The main activity taking place on this level is the transfer of polymer chips from the various bunkers on the floor above. The polymer enters through chutes, which can be opened or closed, in the ceiling. Travelling "hoppers" or trucks take chips from these chutes and deposit them into hoppers which serve individual spinning units on the floor below. These unit hoppers are accessed by lifting covers set in the floor. This floor level is usually referred to as the Hopper Floor and is the top floor of the Type 14 area.

On the southern side of the Type 8 area are the Steam-Air furnaces which are used to clean spinning units and dirty/blocked spinneret packs which have been removed. Doors on the northern side of the Type 8 area lead to an outside area and a number of ventilation plants. There are housings for the supply and extract ducts for the S plants (which are located on the floor below). Also present are three ventilation plants (A, B and C) which supply additional air to the Type 8 part of the first floor area. On a slightly higher level are 10 axial flow extract fans (some of which can be operated at variable speeds) which draw air from the first floor area.

At the northern side of the Type 14 area are located six fans, drawing air from a common chamber, to be supplied to the extrusion area around the Type 14 machines.

2.5.5 FIRST FLOOR

This floor has two levels; the second is formed by a large number of catwalks which surround the spinning machines at a height about 2.5 metres above the general floor level. These catwalks allow access to the machines for maintenance and other purposes. The general floor area is known as the Extrusion Area (referring to the main activity performed) and the raised access level is referred to as the Extrusion Catwalk or Mezzanine level.

At the Catwalk level, the nylon chips are melted and then are extruded downwards (as previously described). Both the higher and lower levels are served by extensive ventilation systems, which provide some cooling for the area which is very warm due to the heat liberated by the Thermex pipework and spinning units.

On the northern side of this area, set apart in their own corridor are the S plants (S1, S2, S3, S4, S5). These provide air conditioning for the ground floor area. Also along the northern side are the majority of the "Blower Air" fans which supply air to the spinning machines for quench cooling of the extruded nylon filaments. The air conditioning and ventilation systems are described in more detail in a later section.

2.5.6 GROUND FLOOR

This floor area is referred to as the "Spin Doff" area. The main operations which take place are the application of spin finish to the yarn, and yarn collection.

In order to maintain the quality of the nylon yarn the environmental conditions must be controlled. The level of moisture contained in the yarn is critical (since this affects both static build-up and subsequent processing). It is the S plants which provide the air conditioning to meet this requirement.

The yarn is drawn down from the extrusion area over rollers and then collected either on a "cake" at the bottom of each spinning unit, or in a "tow can" at the end of a spinning machine. The method of collection is usually determined by the type of yarn being produced. When the yarn has been collected it is moved into either the "staple" or "drawtwist" areas which lie adjacent to the Spin Doff area along its northern side. In these areas, which are also air conditioned, further processing of the yarn is carried out.

2.6 AIR CONDITIONING AND VENTILATION SYSTEMS

Since the factory was first built, the yarn produced at the site, the processing carried out and the size of the operation have changed greatly, and consequently, so have the requirements for air conditioning and ventilation.

A number of areas within the factory are air conditioned or mechanically ventilated, but this study is concerned only with those parts associated with the spinning process. For this process there are three main needs - first to control the environment in the areas where spun yarn is collected; second to supply air to quench cool the extruded nylon filaments; and third to provide suitable ventilation in the hot areas (mainly caused by the melting process).

The systems which operate at present can be split into those serving the older (Type 8) areas of the factory, and those serving the newer (Type 14) areas of the factory. Figure 2.3 shows a schematic diagram of the main air flows taken at a typical cross section of each of the areas.

Figures 2.4 and 2.5 show the air conditioning and ventilation around a spinning machine in the Type 8 and Type 14 areas. Each of the systems are briefly discussed below.

2.6.1 S PLANT SUPPLY - TYPE 8 AREA

There are three fans in this system each of which feeds into a common header duct and each of which can be operated independently (but together with its associated extract fan). The air flow is marked as (1) in Figures 2.3 and 2.4.

This system supplies conditioned air to the area between the spinning machines at Spin Doff level. The air is drawn through dampers which can be moved to vary the proportions of fresh outside air, and recirculated air from the associated S Plant extracts. This mixed air is saturated by spray humidifiers, using borehole water, to an approximate dew point of 61^{o}F (16^{o}C), the air then passes through the fan and into the header duct. At the beginning of each duct supplying the machine areas, there is a heater battery which is controlled, in most cases, by a thermostat in the extract duct. This operates to try to maintain a return air temperature of 72.5^{o}F (22.5^{o}C) and an expected relative humidity of 67%.

Each of the fans has a rating of 125000 c.f.m. (cubic feet per minute) = 59 m^3/s; but it is usual to have only two, or perhaps even one, system operating, and evidence suggests that the fans operate below their quoted ratings. [2]

2.6.2 S PLANT EXTRACT - TYPE 8 AREA

The S Plant extract system is closely related to the supply system. There are three fans drawing air through a series of extract ducts between the spinning machines via a common header. These flows are shown as (2) on Figures 2.3 and 2.4.

As mentioned above, the temperature of the extracted air is used to control the heater in the supply air stream. The three extract fans each have a rated capacity (which is again in doubt) of some 111000 c.f.m. (52.4 m^3/s). As with the supply, only one or two are usually operated.

2.6.3 BLOWER AIR - TYPE 8 AREA

Blower air is that air used to quench cool the extruded nylon filaments as they pass down through so called "chimneys" at Extrusion level. For machines in the Type 8 area, air was drawn, in the past, from a number of sources - Staple area, Drawtwist area and Spin Doff area. Now, however, most of the air is drawn from the Spin Doff area, as can be seen from Figure 2.6. This figure also shows the connections which exist between some of the fans. (The Blower Air system is indicated by (3) in the figures).

If a group of spinning machines is not in production, then some of the fans may be switched off. The amount of air transferred by the Blower Air fans is very difficult to measure; a previous study [2] gives a value of approximately 1.6 m^3/s for each machine, indicating a total rate of about 20 m^3/s, for the Type 8 area.

2.6.4 EXTRUSION SUPPLY - TYPE 8

There are two systems which supply cooling air to the extrusion floor and also, to a much lesser extent, to the hopper floor. The first consists of eleven "Gallery" fans, of which only six are now used. These fans are situated on the third floor of the Spinning Tower. When extra cooling is required (during summer months), water sprays are used to evaporatively cool the air at the inlet to each fan. One fan serves the area around one pair of spinning machines. This system is augmented by three further fans (A, B and C fans) located on the roof outside, at Hopper Floor level. These also have water sprays for eveporative cooling and feed into a common header duct.

Each of the Gallery fans has a rated capacity of 38-40,000 c.f.m. (17.9-18.9 m^3/s) whilst each of the A, B, C fans is rated at 50,000 c.f.m.

($23.6\ m^3/s$). The two systems are shown as (4) and (6) in Figures 2.3. and 2.4. Most of the air is used for cooling the areas at extrusion level near to the hot spinning machines. Some air is used to provide a background supply to the Hopper Floor. Some extra control is allowed for the Gallery fans which can draw some recirculation air from the Hopper Floor ceiling.

2.6.5 EXTRUSION EXTRACT - TYPE 8 AREA

There are ten axial flow fans which extract air from the extrusion area, though only six of these are still in use. (Corresponding, as on the supply side, to the production area still in use). The rated flow of each fan is 65,000 c.f.m. ($30.7\ m^3/s$).

This system is designated (5) on the diagrams in Figures 2.3 and 2.4. Air is drawn mainly from above the Extrusion catwalk, but some extract is also taken from the Hopper Floor area. The fans themselves are located on the roof over part of the Hopper Floor.

2.6.6 S PLANT SUPPLY - TYPE 14 AREA

This system consists of two fans connected to a common header. The principles of operation are as for the Type 8 area S Plant supply. The conditions required in the area are $72.5^oF \pm 2.5^oF$ ($22.5^oC \pm 1.5^oC$) and 67% relative humidity $\pm$ 4%.

Some air is supplied by an underfloor duct directly to the areas where the yarn is wound up, and this air will have a higher humidity and lower temperature than the general environment. The main supply of air to the area, is through ducts between the lines of spinning machines. These flows are indicated in Figures 2.3 and 2.5 by the number (1). The rated supply volume of each fan is 125,000 c.f.m. (59 m^3/s), which is the same as the Type 8 area S Plant supply fans.

2.6.7 S PLANT EXTRACT - TYPE 14 AREA

There are two S Plant extract fans for the Type 14 area, and as with all the S Plant fans these are located in a corridor along the northern side of the Spinning Tower, at first floor level.

The air is drawn from the Spin Doff area from the area between machines via a duct at ceiling level. There is a common plenum chamber through which air is drawn by these fans. The extraction for each fan is rated at 95,000 c.f.m. (44.8 m^3/s). The system is marked as (2) on Figures 2.3 and 2.5. Dampers operate to regulate the amounts of extract air that is to be mixed with fresh air at the inlet to the supply fan.

2.6.8 BLOWER AIR - TYPE 14 AREA

There are eleven fans which provide air for cooling the extruded filaments from machines in the Type 14 area. These are split into two groups, one of five fans serving machines 25 and 27 to 36, and a group of six serving machines 37-42 and 47-52. Each of these groups is connected through a common header, though individual branches may be isolated. The air is drawn from the Spin Doff area at ceiling level and through electrostatic filters. The distribution and layout of the system is shown in Figure 2.6 and the flows are shown as (3) in Figures 2.3 and 2.5.

As with the Type 8 area Blower Air fans, the volumetric flow is very difficult to measure. A previous study suggests a total for all Type 14 fans of about 20 m^3/s.

2.6.9 EXTRUSION SUPPLY - TYPE 14 AREA

Six connected fans supply cooling air to the Type 14 Extrusion area. The fans are located on the northern side of the Hopper Floor and they provide a small amount of ventilation air for this floor area. Air is drawn into a common chamber through grilles to the outside and some recirculation air can be drawn through louvres into the Hopper Floor. During warm periods in the year water spray evaporative cooling of the incoming air can be carried out.

Each of the fans has a volumetric rating of 60,000 c.f.m. (28.3 m^3/s). (It is unusual for all six fans to operate together). The system and duct outlets are indicated as (4) in Figures 2.3 and 2.5. At Extrusion level, the air is supplied to the areas between the machines by several ducts each leading from the common header duct.

2.6.10 EXTRUSION EXTRACT - TYPE 14 AREA

The fans for this system are positioned on the roof above the Hopper Floor over the Type 14 area. There are twenty-one fans, sixteen of which take air from individual alleyways between machines. The remaining five, some of which are interconnected, draw air mainly from along the southern edge of the Extrusion area. Some air is also extracted from the Hopper Floor through grilles into the duct sides, as they rise up to the roof level.

Each fan has a rating of 30,000 c.f.m. (14.2 m^3/s) though often some fans are not operated. The system is shown as (5) in Figures 2.3 and 2.5.

Table 2.1 summarises the flows of the various systems.

TABLE 2.1 SUMMARY OF AIR CONDITIONING AND VENTILATION AIR FLOWS

	SYSTEM	MAIN AIR FLOWS FROM:	TO:	POTENTIAL MAXIMUM TOTAL RATED FLOW
TYPE 8	S Plant Supply	Outside	Spin Doff	177 m^3/s
	S Plant Extract	Spin Doff	Outside	157 m^3/s
	Blower Air	Spin Doff	Spinning Machines (Extrusion)	(20 m^3/s approx)
	Extrusion Supply (Gallery)	Outside	Extrusion	114 m^3/s
	Extrusion Supply (ABC)	Outside	Extrusion	71 m^3/s
	Extrusion Extract	Extrusion	Outside	184 m^3/s
TYPE 14	S Plant Supply	Outside	Spin Doff	118 m^3/s
	S Plant Extract	Spin Doff	Outside	90 m^3/s
	Blower Air	Spin Doff	Spinning Machines (Extrusion)	(20 m^3/s approx)
	Extrusion Supply	Outside	Extrusion	170 m^3/s
	Extrusion Extract	Extrusion	Outside	298 m^3/s

2.7 INTERFLOOR PRESSURE (I.F.P.)

As mentioned in an earlier section, part of the production process involves the yarn passing down through an open ended conditioning tube. This tube is situated between the Extrusion and Spin Doff levels and contains steam. In order to prevent the steam being drawn through the tube, with the yarn, into the Spin Doff area, a small pressure difference is desirable between these areas. This effect is achieved by using pressure sensors which measure the pressure differential and operate mechanisms to vary the amount of air being extracted from the Extrusion area.

In the Type 8 area this is achieved by altering the speed of the axial flow fans, although a second facility exists to open louvres allowing outside air to be drawn through the fan thus reducing the air extracted from inside the building.

In the Type 14 area dampers are operated which switch to draw a greater amount of air from the Hopper Floor and a reduced amount from the Extrusion area.

The interfloor pressure which the controls seek to produce is 12/1000 th of an inch, water gauge ($3\ N/m^2$).

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The interfloor pressure which the controls seek to produce is 12/1000 th of an inch, water gauge (3 N/m^2).

2.8 THE BUILDING FABRIC

An examination of the building fabric (walls, roofs, floors) in the Spinning Tower was carried out as part of the investigation. Besides personal inspection, building plans were also consulted, which gave information on the construction details dating back to the original factory. This information was to be used to estimate the conduction heat transfers that would take place if a temperature difference existed inside-outside or between different parts of the building.

The Spinning Tower building is a very complex structure and this meant that some areas were difficult to access to check the fabric details. In addition some flows, especially those to adjacent parts of the factory, could not be defined easily, and areas were difficult to measure exactly. This necessitated the making of assumptions and amalgamation of flows. However, since it was expected that these conduction flows would be relatively small by comparison with the heat flows set up by the air conditioning and ventilation systems, it was decided that the assumptions, and possible consequent small errors, could be tolerated. Also since the figures obtained were to be used in a simple steady state model of conduction, extreme accuracy would not be required. The study was still extremely valuable and necessary however, since no previous work had been done to assess this heat transfer, and any information gathered would be useful.

The main structural load of the building is borne by a large number of reinforced concrete pillars and metal girders spread throughout the factory. In fact these pillars form a grid network which enable a location within the factory to be defined. Almost all of the partition walls however, consist in some part of brickwork. The construction of these walls varies considerably; especially obvious are the differences between the older and newer sections of the building.

Five main types of wall can be identified, and these are illustrated, together with a determination of their respective heat transfer coefficients (U-Values) in Figure 2.7 (Wall A), Figure 2.8 (Wall B), Figure 2.9 (Wall C), Figure 2.10 (Wall D) and Figure 2.11 (Wall E).

The intermediate floors are constructed of concrete slabs supported by the pillars and girders. The floor construction is basically identical throughout the factory; the only main variations being due to the positioning of metal hatches between Extrusion and Hopper Floors. Figures 2.12 and 2.13 show the details of the intermediate floor.

(The heat transfer through the Spin Doff level floor into the ground was determined, using the standard method described by the C.I.B.S. [1], as $0.12\ W/m^2\ ^oC$).

There are three main types of roof, indicated in Figure 2.14 (Roof A), Figure 2.15 (Roof B) and Figure 2.16 (Roof C). Again the calculation of their respective U-Values is shown.

In addition to these constructions, the transfer through glass within the walls must also be taken into account. This value is also obtained from the C.I.B.S. data for single glazing, this being $5.62\ W/m^2\ ^oC$.

The potential heat flow paths are summarised in Tables 2.2 (a), (b) and (c) which follow. Details of the location of the possible conduction, the area involved, the construction and the U-Value are shown. The figure given in the final column represents the heat transfer to be expected (under steady state conditions) in watts for each degree centigrade temperature difference.

The building construction and the large areas involved show that there is considerable potential for heat transfer by conduction, both to the external environment, and between areas within the building.

TABLE 2.2 (a) CONDUCTION HEAT TRANSFER COEFFICIENTS
GROUND FLOOR (SPIN DOFF)

FROM	TO	AREA (m^2)	CONSTRUCTION	U-VALUE ($W/m^2\ {}^{o}C$)	HEAT TRANSFER ($W/{}^{o}C$)
Type 8	Drawtwist	213	Wall A	1.92	409
Type 14	Drawtwist	212	Wall B	1.43	303
Type 14	Outside (E)	155	Wall D	0.73	113
Type 14	Outside (S)	200	Wall C	1.51	302
Type 8	Corridor	190	Wall A	1.92	365
Ceiling	Corridor (S)	2984	Int. Floor	2.82	8415
Ceiling	1st Floor	1278	Int. Floor	2.82	3604
Floor	S Plant Area	4262	Gr. Floor	0.12	511

TABLE 2.2 (b) CONDUCTION HEAT TRANSFER COEFFICIENTS
FIRST FLOOR (EXTRUSION)

FROM	TO	AREA (m^2)	CONSTRUCTION	U-VALUE ($W/m^2\ {}^{o}C$)	HEAT TRANSFER ($W/{}^{o}C$)
LOWER LEVEL					
Type 8	S Plant Area	119	Wall A	1.92	228
Type 14	S Plant Area	172	Wall A	1.92	330
Type 14	Outside (E)	78	Wall E	1.23	96
Type 14	Outside (S)	145	Wall E	1.23	178
Type 8	Offices etc	147	Wall A	1.92	282
Floor	Gr. Floor	2984	Int. Floor	2.82	8415
Floor	Corridor	341	Int. Floor	2.82	962
HIGHER LEVEL					
Type 8	S Plant Area	119	Wall A	1.92	228
Type 14	S Plant Area	172	Wall A	1.92	330
Type 14	Outside (E)	78	Wall E	1.23	96
Type 14	Outside (S)	145	Wall E	1.23	178
Type 8	Offices etc	147	Wall A	1.92	282
Ceiling	2nd Floor	2910	Int. Floor	2.82	8206
Ceiling	2nd Floor	416	Hatches	4.72	1964

TABLE 2.3 (c) CONDUCTION HEAT TRANSFER SECOND FLOOR (HOPPER)

FROM	TO	AREA (m^2)	CONSTRUCTION	U-VALUE ($W/m^2\ ^oC$)	HEAT TRANSFER ($W/^oC$)
Type 8	Outside (N)	325	Wall C	1.51	491
Type 8	Outside (N)	65	Glazing	5.62	365
Type 14	Outside (N)	45	Wall E	1.23	55
Type 14	Air Chamber	284	Wall A	1.92	545
Type 14	Outside (E)	92	Wall E	1.23	113
Type 14	Outside (E)	48	Glazing	5.62	270
Type 14	Outside (S)	157	Wall E	1.23	193
Type 14	Outside (S)	79	Glazing	5.62	444
Type 8	Outside (S)	120	Wall C	1.51	181
Type 8	Outside (S)	20	Glazing	5.62	112
Type 8	Steam Clean	270	Wall A	1.92	518
Type 8	Outside (W)	106	Wall C	1.51	160
Type 8	Outside (W)	40	Glazing	5.62	225
Floor	1st Floor	2910	Int. Floor	2.82	8206
Floor	1st Floor	416	Hatches	4.72	1964
Roof	Outside	912	Roof A	2.01	1833
Ceiling	3rd Floor	1718	Int. Floor	2.82	4845
Roof	Outside	1645	Roof B	1.58	2599
Roof	Outside	405	Roof C	0.92	373

The heat loss potential directly to outside is: Spin Doff 0.217 $W/^\circ C/m^2$; Extension 0.15 $W/^\circ C/m^2$; Hopper 2.235 $W/^\circ C/m^2$ (each value is per square metre floor area). Total heat transfer from the building to outside being 8.857 $W/^\circ C$.

2.9 ENERGY CONSUMPTION

The main records of energy use to be found, prior to the instigation of more comprehensive meter readings by the Energy Utilisation Engineer, were those taken for accounts purposes. The main accounting period is the "month" which is made up of whole numbers of weeks, with adjustments in January and December. Most meters around the plant were read at the monthly accounting point, though some major ones were read more frequently. These monthly readings were taken as a basis for further investigation.

2.9.1 GAS

The gas consumption is made up of two main fractions. The first is due to the high pressure hot water system, supplied from a main boilerhouse. As space heating is provided from this system, there is some dependence on external prevailing weather conditions. The second fraction is due to the requirements of the Thermex heating circuits, for the nylon polymer melting process. No major changes, other than those already carried out, were envisaged for these heating systems. A study of the previous few years figures showed that conservation measures had helped reduce energy demand, and that the shutting down of one of the Thermex heating circuits in 1980, had a considerable affect.

2.9.2 ELECTRICITY

The electricity consumption is accounted for by three main load groups. The first is the general machinery power (spinning machines etc) and lighting requirements. The second is the demand made by the air conditioning and mechanical ventilation plants. And the third is that due to service and ancillary equipment (air compressors, power to Thermex system, battery charging, etc).

Since the air conditioning and ventilation systems had been identified as a possible source of energy savings, (with the equipment drawing a major part of the electricity consumption), these figures were investigated in detail.

The meters associated with equipment in the Spinning Tower were found and the figures accumulated. Figure 2.17 illustrates the average weekly electricity use for the months January 1978 to March 1981. (Certain months are excluded because of unusual situations which make them unsuitable for analysis). The top line on the graph indicates the total electrical energy consumption in the Spinning area, whilst the lower line indicates the fraction attributable to air conditioning and ventilation plant. The diagram shows that this fraction forms the major part of the total, and that the variations, month to month, in the total are largely due to variations in the ventilation equipment load.

2.9.3 CORRELATIONS

Correlations were sought between various facets of the electrical load, and production and gas consumption figures, but little useful information could be gleaned. General background data, on energy consumption for the factory, can be found in Appendix A, which gives Energy Usage Analysis tables for the years 1980-1983.

After having made a detailed study of the various figures available, which related to energy use; it was seen that the readings were too infrequent and too non-specific to be of real use in anything more than a general analysis.

2.10 ENVIRONMENTAL SURVEYS

Since much energy and effort has been, and still is, involved in the maintenance of environmental conditions within the Spinning Tower, it was decided to carry out some environmental surveys in the building.

A number of positions were chosen and the conditions measured at four levels (Spin Doff, Extrusion, Extrusion Catwalk and Hopper Floor). Three basic pieces of equipment were used - each standard items employed for environmental assessment. These were a whirling hygrometer, a black globe thermometer and a Kata thermometer.

2.10.1 THE WHIRLING HYGROMETER

This hand-held instrument consists of two mercury-in-glass thermometers, set in a wooden or plastic holder. The bulb of one of the thermometers is covered by a wick moistened by distilled water from a small reservoir. The thermometers in the holder are whirled in a circular pattern to force air flow over the bulbs.

Differences in moisture content of the surrounding air, affect the amount by which the temperature sensed by the "wet-bulb" thermometer is lower than the "dry-bulb". In making a reading, the hygrometer, as the instrument is called, is whirled for approximately ten seconds and the two temperatures are noted. This process is repeated until the readings recorded are static. The two temperatures then define the condition of the air, and from them it is possible to determine various other properties, such as relative humidity.

2.10.2 THE BLACK GLOBE THERMOMETER

This is used in the calculation of mean radiant temperature. The instrument consists of a matt black copper sphere 150 mm (6 inches) in diameter, with a mercury in glass thermometer at its centre. Heat exchange by convection and radiation between the globe and its surroundings, takes place, and approximately fifteen minutes should be allowed for a steady reading to be obtained.

2.10.3 THE KATA THERMOMETER

The Relative Air Velocity can be determined using the Kata thermometer. It has a large silvered bulb filled with a coloured spirit and a glass stem with just two graduation marks on it. These are usually 38 and 35^{o}C, however because of the higher temperatures to be found at the ICI site, the model covering 65.5 to 62.5^{o}C was used.

To use the bulb must first be heated by immersion in a flask of hot water. This causes a column of the coloured spirit to rise up the stem. The flask is removed and the excess water wiped from the bulb. The thermometer is then left unmoving in the measuring position. As the bulb cools due to convection (caused by air movement) the column falls and the time to drop between the two graduation marks is recorded. The first reading is ignored and the process carried out a further five times to obtain an average cooling time.

2.10.4 DERIVED MEASURES OF THE ENVIRONMENT

Though not strictly true, in most instances, the air temperature is assumed equal to the dry bulb temperature. In the surveys carried out limitations of time and equipment meant that this assumption was again made.

The Mean Radiant Temperature (MRT) is calculated in the following way

$$MRT = GT + 2.35\sqrt{v}\ (GT - AT) \qquad (2.1)$$

where

GT = Globe Thermometer reading (^{o}C)

AT = Air Temperature (^{o}C)

v = Air Velocity (m/s)

The Relative Air Velocity is found by using a British Standard Chart [(3)] for which the "cooling factor" and average cooling time of the Kata thermometer, and the air velocity are required data.

2.10.5 RESULTS

The results of the surveys carried out are given in Tables 2.4, 2.5 and 2.6. Survey I was carried out on a different day to the other two surveys.

The results show that conditions varied quite considerably between floors, which was to be expected. However the variations between points on the same floor level were not so expected. Slight differences in the way the surveys were carried out and fluctuating air movements could be responsible; for instance locations a metre or so distant from the measurement points had noticeably different air velocities. In addition, the time and preparation required for each survey, made it impracticable to carry them out frequently at many locations which would have allowed "average" conditions to be determined and some variations to be identified. Clearly some other means of environment measurement was required.

TABLE 2.4

ENVIRONMENTAL SURVEY I - POSITION BY MACHINES 27/28

	SPIN DOFF	EXTRUSION	EXTRUSION CATWALK	HOPPER FLOOR
DRY BULB/AIR TEMPERATURE (Deg.C)	25.0	24.0	30.8	17.9
WET BULB TEMPERATURE (Deg C)	19.0	15.8	19.8	11.9
BLACK GLOBE THERMOMETER (Deg C)	25.5	28.0	33.4	19.7
AVERAGE COOLING TIME FOR KATA THERMOMETER (Seconds)	30	38	34	23
% SATURATION	56	41	34	48
MEAN RADIANT TEMPERATURE (Deg C)	26.3	36.4	38.7	23.4
MEAN AIR VELOCITY (m/s)	0.5	0.8	0.75	0.75

TABLE 2.5

ENVIRONMENTAL SURVEY II - POSITION BETWEEN MACHINES 16 & 17

	SPIN DOFF	EXTRUSION	EXTRUSION CATWALK	HOPPER FLOOR
DRY BULB/AIR TEMPERATURE (Deg C)	23.0	28.9	35.8	20.3
WET BULB TEMPERATURE (Deg C)	17.5	19.1	22.2	14.2
BLACK GLOBE THERMOMETER (Deg C)	24.9	29.4	37.9	23.1
AVERAGE COOLING TIME FOR KATA THERMOMETER (Seconds)	36.5	27	40	25
% SATURATION	59	39	27	50
MEAN RADIANT TEMPERATURE (Deg C)	27.5	34.6	41.7	28.4
MEAN AIR VELOCITY (m/s)	0.25	0.85	0.6	0.75

TABLE 2.6

ENVIRONMENTAL SURVEY III - POSITION BY MACHINES 33/34

	SPIN DOFF	EXTRUSION	EXTRUSION CATWALK	HOPPER FLOOR
DRY BULB/AIR TEMPERATURE (Deg C)	23.5	26.1	38.0	27.2
WET BULB TEMPERATURE (Deg C)	17.2	17.3	24.8	18.6
BLACK GLOBE THERMOMETER (Deg C)	24.8	28.0	39.0	29.3
AVERAGE COOLING TIME FOR KATA THERMOMETER (Seconds)	14	22	36	31
% SATURATION	54	41	34	42
MEAN RADIANT TEMPERATURE (Deg C)	29.1	28.2	40.8	33.1
MEAN AIR VELOCITY (m/s)	2.8	1.25	0.75	0.6

2.11 PRESSURISATION TEST

It had been reported by ICI staff, that large quantities of air could be transferred between areas and floor levels in the factory (due to external influences). Since the air transferred would certainly affect the environmental conditions and generation of ventilation plant, it was decided to investigate the overall air movement in the factory. To do this a pressurisation test was planned and carried out during a Christmas maintenance period. (The use of pressurisation tests as an investigation method, is more fully described in Chapter 3).

In most cases, pressurisation tests are carried out by using an external fan to either pump air into or draw air from a given enclosed area, thus setting up a positive or negative pressure, with respect to the outside, in the area. The flow and pressure differences are recorded and their relationship established. This is usually of the form:

$$Q = C \Delta P^n \qquad (2.2)$$

where

Q = Air flow

C = Leakage coefficient

ΔP = Induced pressure difference

n = Exponent

Because each floor of the Spinning Tower contains such a large volume, it would not have been practicable to have used an external fan for such tests. It was thought possible however, that by the switching on and off fans in the air conditioning and ventilation systems, (to supply or extract excess air to or from particular areas) that pressure variations could be created. To the knowledge of the author, no such tests had been carried out previously in this country, and overseas researchers (in North America) have confined such large scale tests to office type buildings.

The flow rates of the fans, to be used in the tests, were first measured, or where this was not possible, design flow values were taken. Pressure transducers were calibrated and set-up to monitor the following pressure differences:

(i) Outside - Spin Doff

(ii) Spin Doff - Extrusion

(iii) Extrusion - Hopper

(The transducers used were of the B.R.E. wind pressure type, which are described in Chapter 6)

Each of the floor levels was treated as a single large space.

The mechanical ventilation air flows, to and from the Spin Doff and Extrusion areas, were varied, and the resulting pressure differences recorded.

Unfortunately, a number of unforeseen equipment problems presented themselves during the course of the experiments, which severely limited the work which could be undertaken. In addition it was found difficult to control and maintain even small pressure differences in the tests. The pressure measurement points were located within the Type 8 areas at each level, however it was noted that movement of these measurement positions had quite significant effects on the pressure differences found.

Examples of the results of the pressure tests are presented in Figures 2.18 and 2.19, which plot air flow against certain observed pressure differences. These show that even with zero net flow to and from an area, a pressure difference is still to be found.

After close scrutiny of the experiment and results it was decided not to pursue this large scale test further because of the difficulty of gaining a full set of data that would also be reliable and repeatable.

The experiment had been very useful however. Much experience had been gained, of the operation of the ventilation systems, and it had been established that one could not treat each floor level as one, single area, for air flow purposes. It would seem more sensible, considering the degree of partitioning caused by the rows of machines, to divide each floor into a number of similar, sequentially connected, smaller areas. This proposal will be discussed and developed later.

2.12 SMOKE TRACING OF AIR FLOWS

In order to observe more clearly, the air movement in the spinning areas, "smoke" was used to trace the flows. The smoke in fact consisted of an aerosol of paraffin oil created by atomisation of the paraffin in a smoke generator, which was originally designed for use in wind tunnels. The smoke is easily seen as a white opaque cloud. Still photographs of such patterns do not indicate directions of movement and experimental notes made by observers may omit some details. Therefore these techniques were supplemented in this situation by the use of a portable video camera and recorder. The camera is shown set up in Plate 1 (centre bottom) the view being along the Extrusion level catwalk by machines 27 and 28. On the left of this view can be seen a small floodlight required for additional illumination.

Most of the experiments were carried out in the Extrusion area since it was at this level that considerable variations in environmental conditions had been observed (e.g. See Tables 2.4, 2.5, 2.6). Two sets of tests were performed, once during one of the works maintenance periods when the main plant was not operating, and once under normal operating conditions.

Following Page:

PLATE 1

GENERAL ARRANGEMENT FOR
SMOKE TRACER TESTS,
(EXTRUSION LEVEL)
VIDEO CAMERA AND FLOODLIGHT
SHOWN.


415 VOLTS

2.12.1 MAINTENANCE WEEK TEST

This period was chosen for a test because it was only at such times, that it was possible to vary and switch off and on, the fans of the ventilation systems, for experimental purposes. Even so, parts of the heating circuits, still in use to prevent "freezing" of the Thermex system, liberated considerable heat to the environment. As a result, certain background ventilation was required at all times, to prevent overheating of machinery and electronic components.

A variety of system set-ups were investigated, and the most notable features, or differences from the expected results, are noted below.

Amongst the Type 8 area machines, when the ventilation of one "alleyway" was switched off, the ventilation to adjacent alleyways was seen to have an effect. The smoke tracer flowed upwards and then to the southern end of the alleyway.

In the Type 8 area some stagnation was also noticed (which can be seen in Plate 2). That is, the smoke accumulated in some areas forming layers which dispersed only slowly.

The flow to adjacent machine alleyways was found to be unequal, the prevailing smoke movement was to the east side of the Spinning Tower. This may have been due to the influence of external pressures or imbalances in the ventilation system.

Following Page:

PLATE 2

SMOKE TRACER TEST
SHOWING STAGNATION ZONE
AND LAYERING.

In the Type 14 area, at catwalk level, some inconsistencies in the flow were noted. Recirculation zones were seen to occur and the air was not extracted as expected. When smoke was liberated next to the heads of the spinning units, the air was less drawn towards the extract grilles in the duct almost above, than it was to the duct above and behind the supply ducts on the other side of the catwalk. (This is shown in Plate 3). This pattern could have been due, to some extent, to the air flows from the supply ducts, since when the supply flow rate was reduced, the more expected flow pattern occurred.

In general, in the Type 14 area, there was a drift towards the southern end of the machines.

2.12.2 NORMAL PLANT OPERATION TESTS

Many of the tests were repeated during a period of normal plant operation, and in most cases the resulting flow patterns were very similar, if not identical. For process reasons however, it was not possible to vary the fan operation in the same way.

It was noted that smoke dispersed slowly from the Type 8 area Extrusion level. This could have been due to the lack of extract duct located at the lower level (See Figure 2.4).

In the Type 14 area, Extrusion catwalk level, the recirculation zone was once again found, with

Following Page :

PLATE 3

SMOKE TRACER TEST
SHOWING RECIRCULATION.


DANGER
415
VOLTS
DANGER
415
VOLTS

the greater proportion of the flow being drawn to the seemingly inappropriate extract duct.

Flows around the doors and entrances to the areas were also investigated. A definite flow into each area was discovered through doors on the western and northern perimeters. Even if the doors were closed, some infiltration through cracks and gaps occurred. In fact as a number of doors consist of spring-loaded rubber flaps, the differential air pressure can often be sufficient to cause them to open slightly thus permitting air to flow in.

2.13 SUMMARY

This chapter has presented relevant information gathered about the nature and operation of the plant and process at ICI Fibres' Doncaster factory. Meter readings, previous reports, accounting records, site plans and ICI personnel, were consulted to gain a high degree of background knowledge. In addition a number of tests and surveys were carried out to examine certain aspects.

Some criticisms of the plant and its operation can be made, but these are dealt with in a later chapter. It was felt more important to investigate further some points raised, in order that positive recommendations for the improvement of plant efficiency might be made.

The techniques so far used, have been limited in accuracy and scope. It had become clear that specific areas required more detailed work.

Environmental surveys are necessary to determine and record internal conditions. Since the industrial aim of the project was to save on energy costs, (which is very much related to the internal environmental conditions), such records were an essential part of the observation of the plant. The method used for the readings reported earlier in this chapter, would have been far too time consuming, infrequent and non-coincident in time, to give a useful indication of the spacial, and temporal variations of environmental parameters.

The figures for energy use were found to be too non-specific and too infrequently monitored. The main requirement, given the emphasis on the air conditioning and ventilation systems, as both energy consumers and energy flow distributors, was for regular monitoring of these systems.

It was concluded that some form of automated monitoring/data logging system was needed. This would be used to sense and record (i) environmental conditions within the Spinning Tower, and (ii) details of the air conditioning and ventilation systems.

In many studies of the environment masses of data are accumulated. If a clear objective is not in sight the amounts can become so great as to hinder eventual analysis and obscure the significance of the meaning. Some studies in the past have concentrated too greatly on data collection and may not justify the expense in terms of quality of results.

In this investigation the financial constraints for the environmental monitoring was determined by ICI Fibres and the organization of the monitoring was performed bearing in mind the problems and needs of ICI. The measurements made were designed to yield useful information and the system was to have a longer useful life than the timespan of this project.

The results of the pressurization test and the air flow movement traced using smoke indicated some areas for investigation. The ducted air flows supplied and extracted from various areas of the factory did not provide a complete explanation. The large production floors could not be considered as single units for the purposes of air movement. Undoubtedly one of the major factors of influence was the high degree of internal partitioning set up by the production machines and their ancillary service systems. Such a situation combined with its industrial nature meant that the standard guides (e.g. reference (4)) could not be easily applied.

Since the partitioning aspect of the environment seemed very significant, this offered itself as a subject for further investigation. As an important step in the development of the work it was decided to review the factors associated with air movement; the history and evolution of air movement investigations and the techniques which have been employed in such work. In this way a more complete background knowledge would be available on which to base decisions about the course of this work. It was seen that more experimental studies might be required and that these would have to be more restrictive in scope than the whole factory building work so far performed.

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Section A3 Thermal and other properties of building structures

2 F. D. Knapper
Report on Ventilation - Doncaster Works
January 1969

3 British Standards Institute
B.S. 3276 1960

4 Chartered Institution of Building Services
Guide to Current Practice
Books A, B and C

FIGURE 2.
SITE PLAN
I.C.I. FIBRES
DONCASTER.

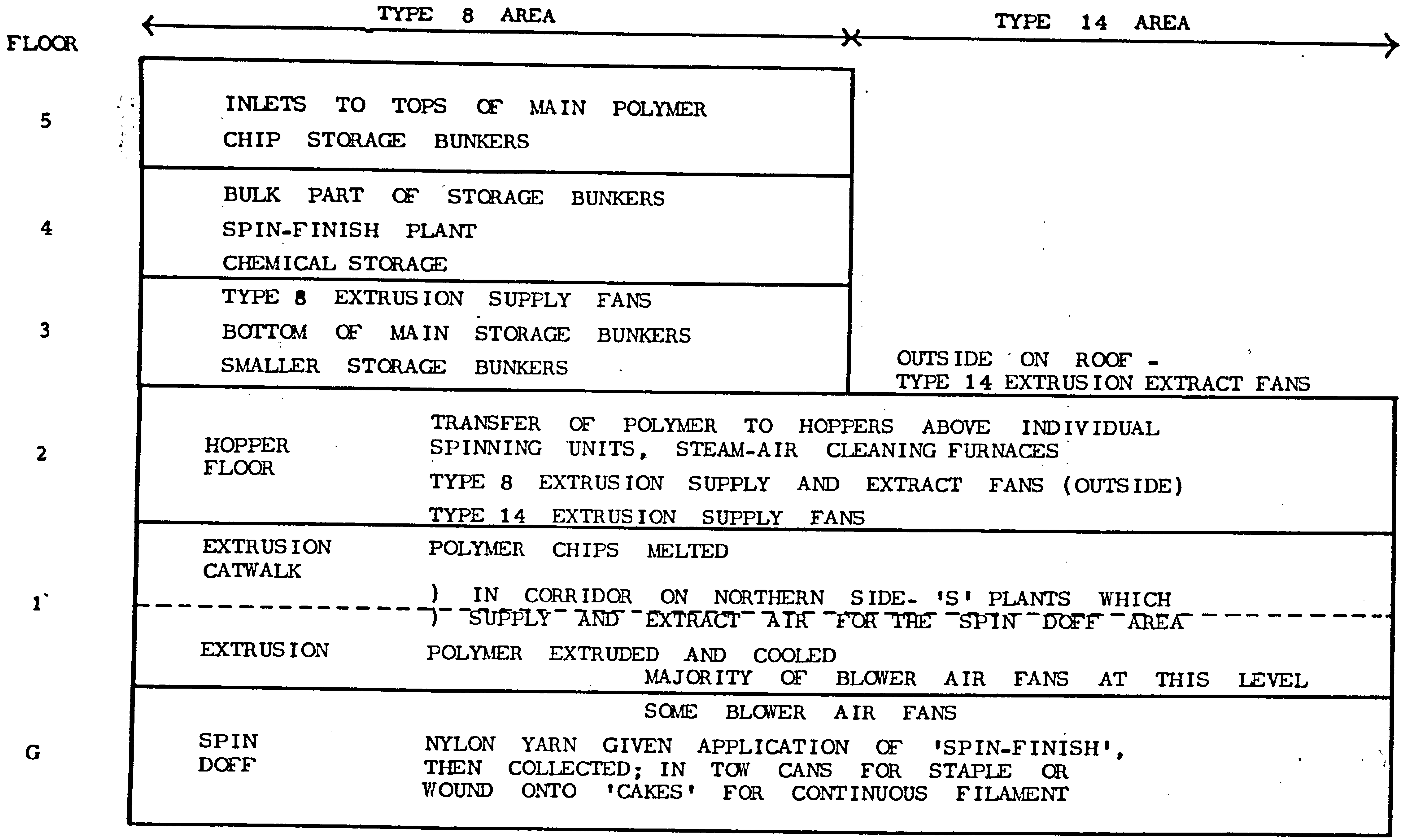
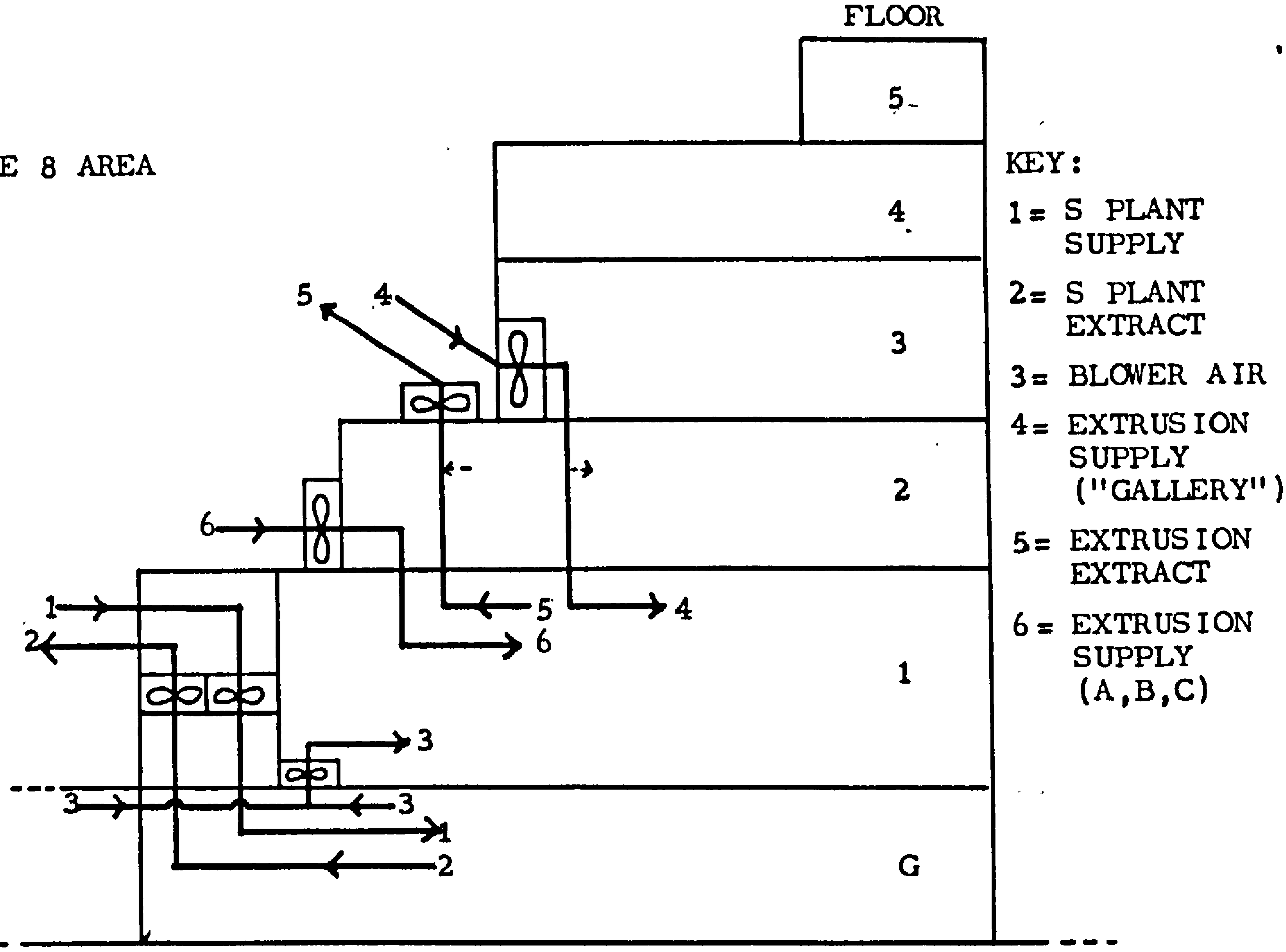


FIGURE 2.2 DIAGRAMATIC SKETCH OF SPINNING TOWER, ICI FIBRES,DONCASTER.

TYPE 8 AREA



TYPE 14 AREA

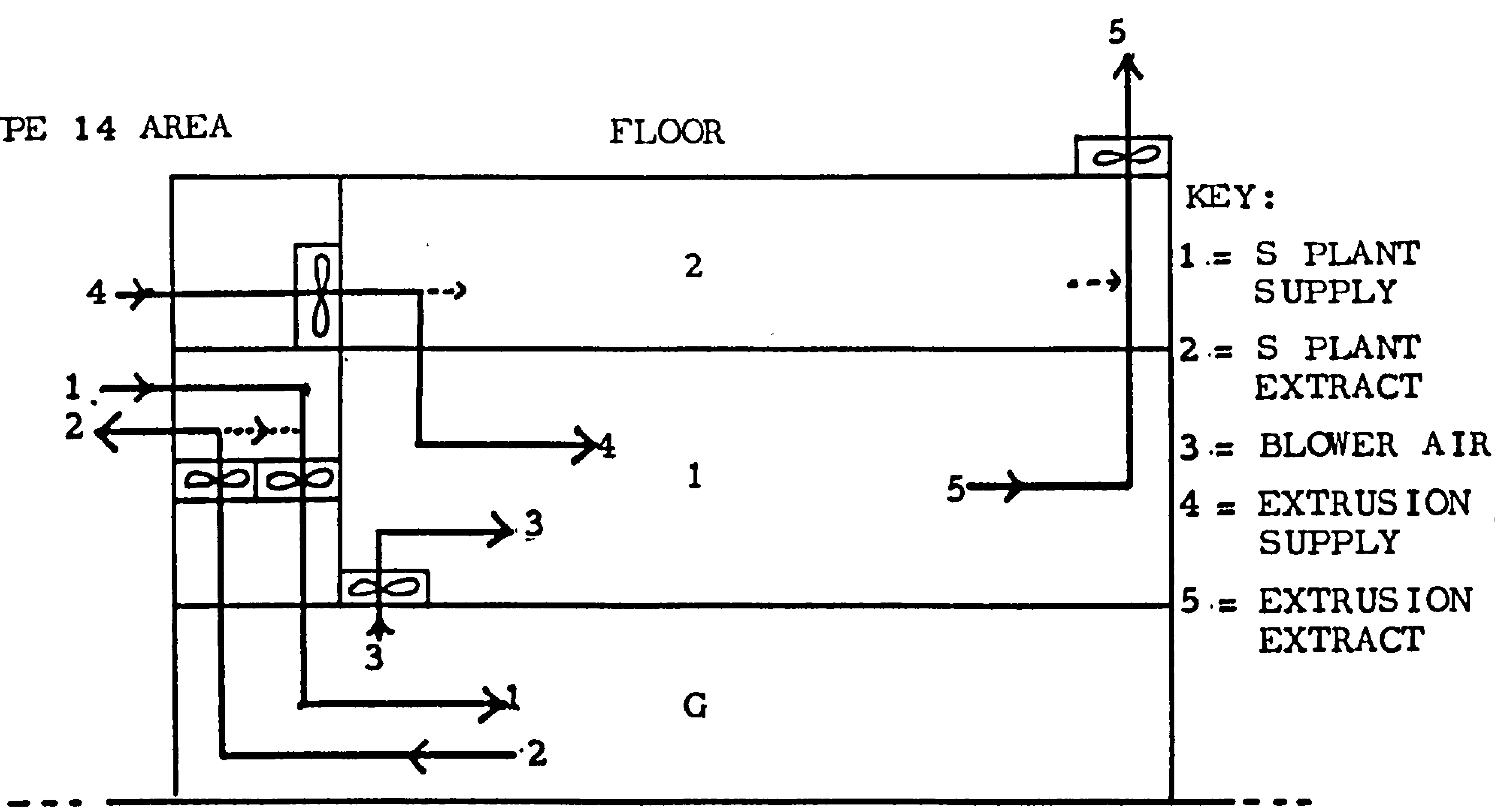


FIGURE 2.3 SCHEMATIC DIAGRAM OF AIR-CONDITIONING AND VENTILATION DUCTED AIR FLOWS

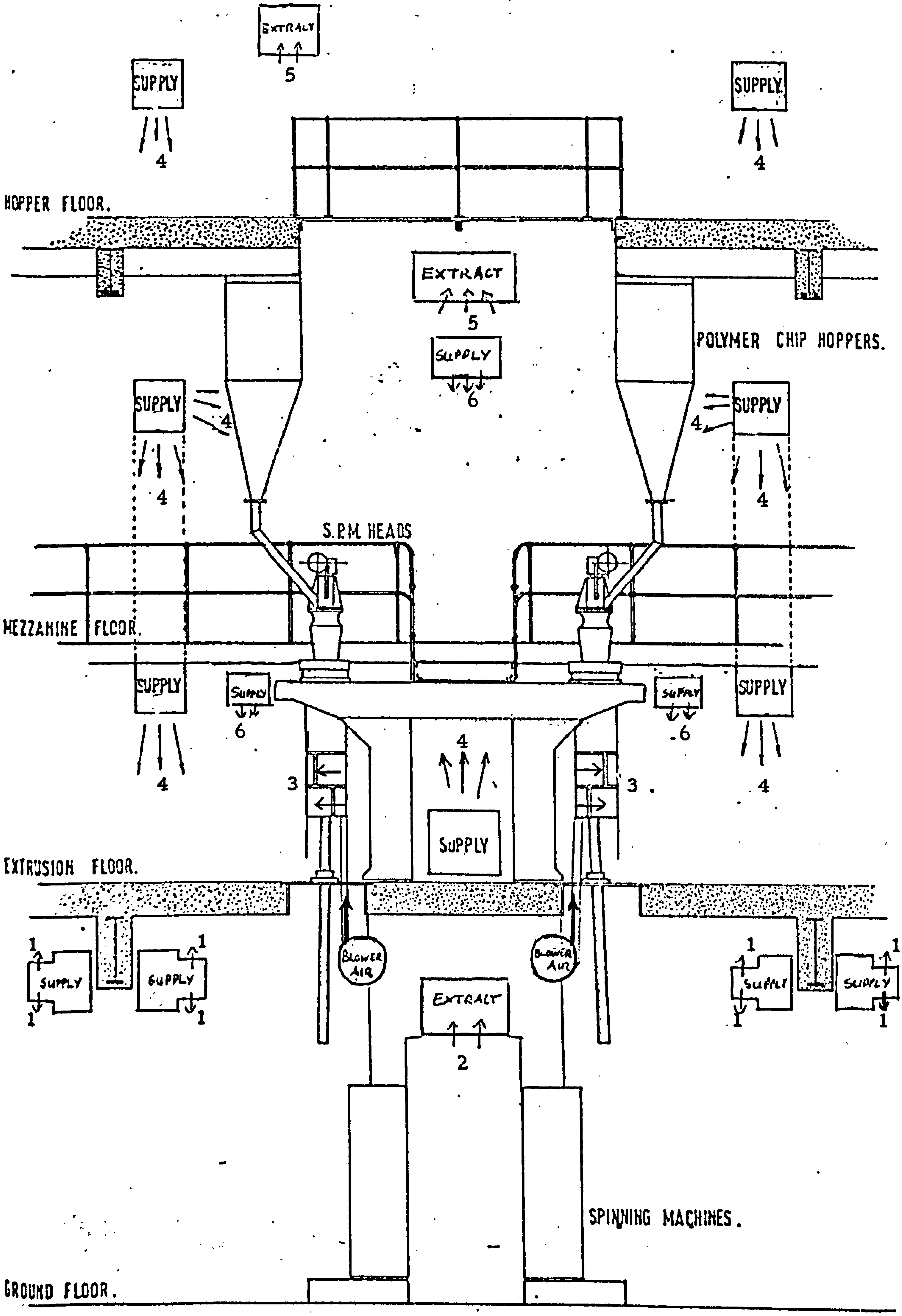


FIGURE 2.4
TYPICAL SECTION THRO' TYPE 8 MACHINES. DONCASTER.[2]

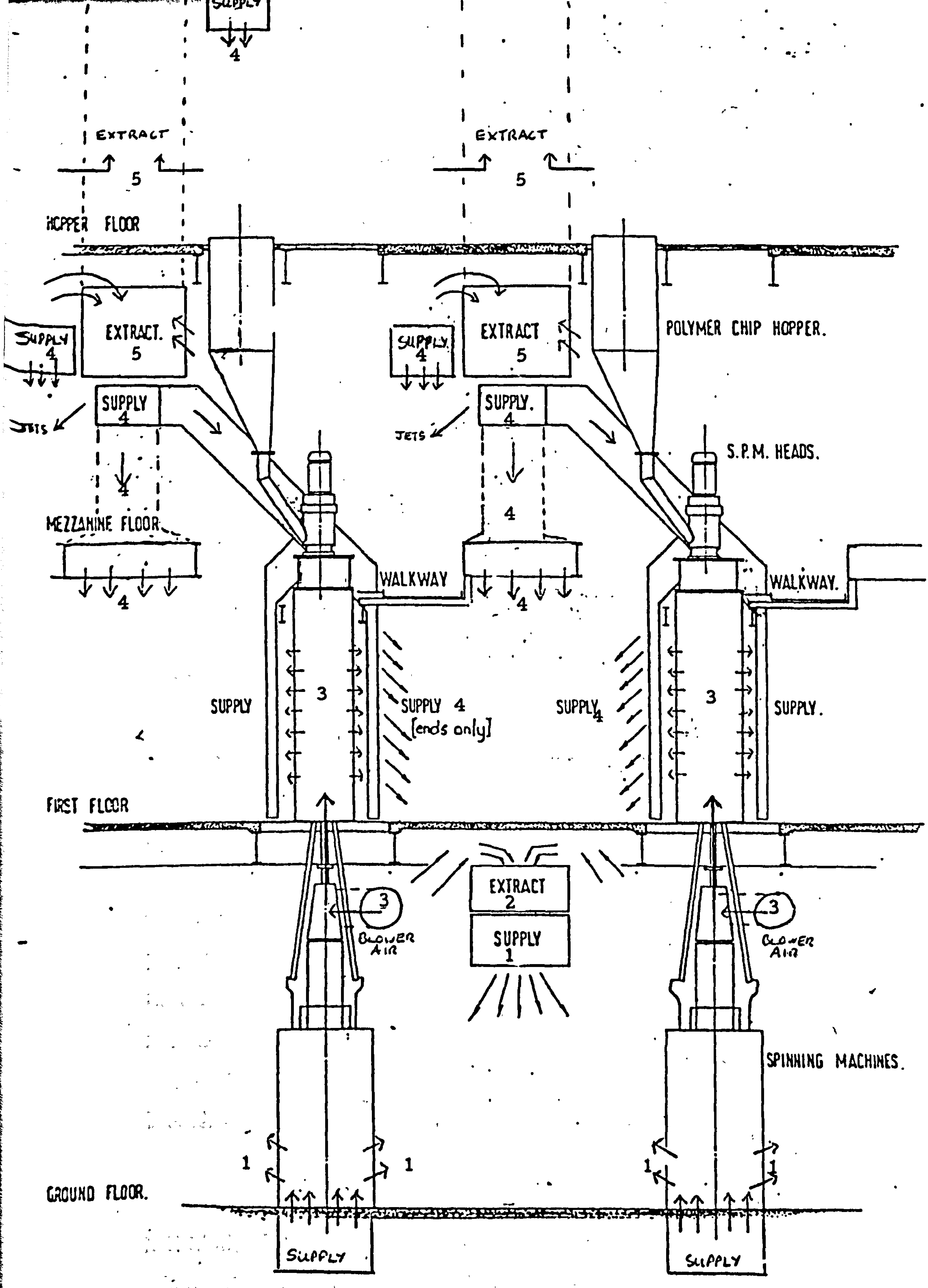


FIGURE 2.5

TYPICAL SECTION THRO' TYPE 14 MACHINES. DONCASTER.[2]

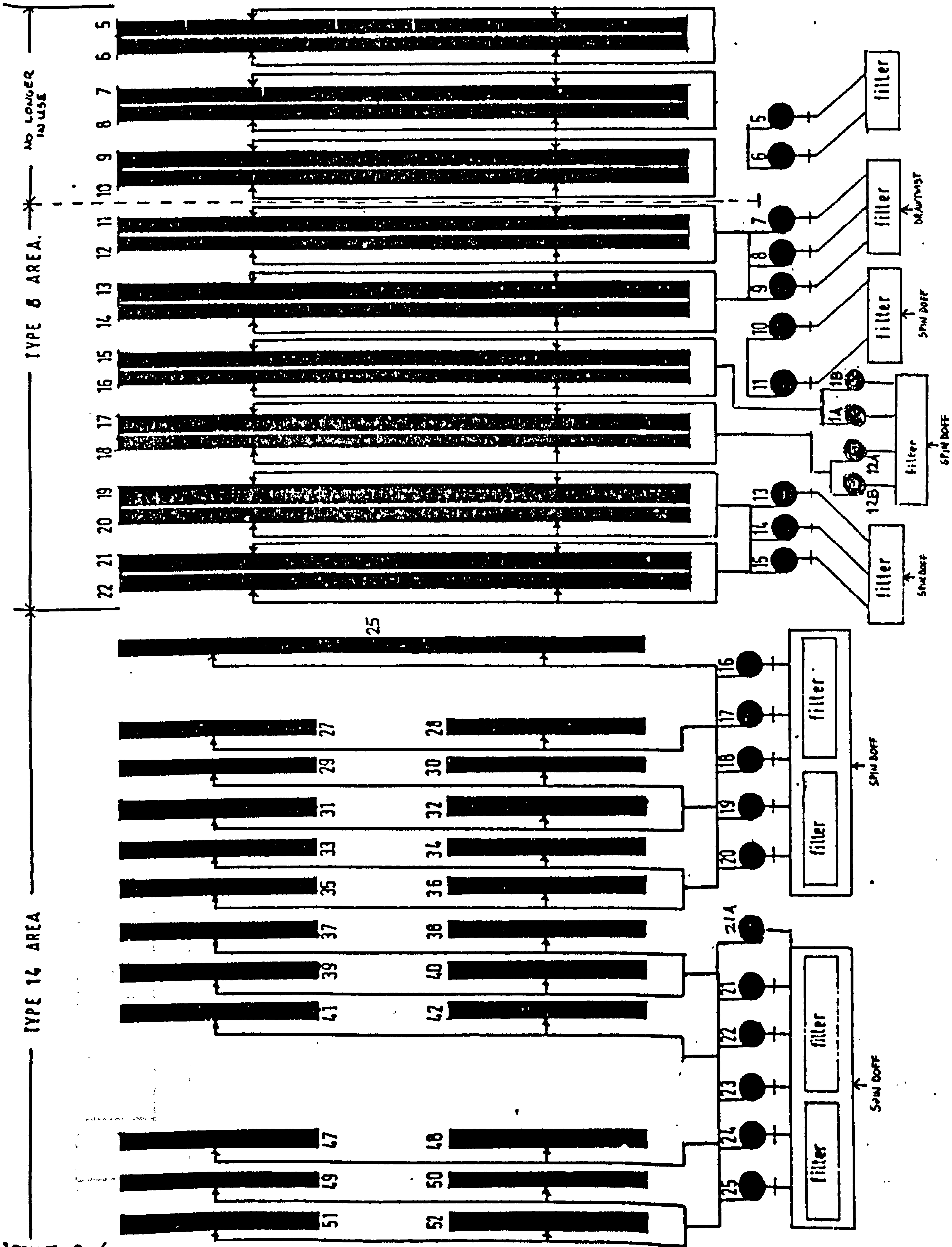
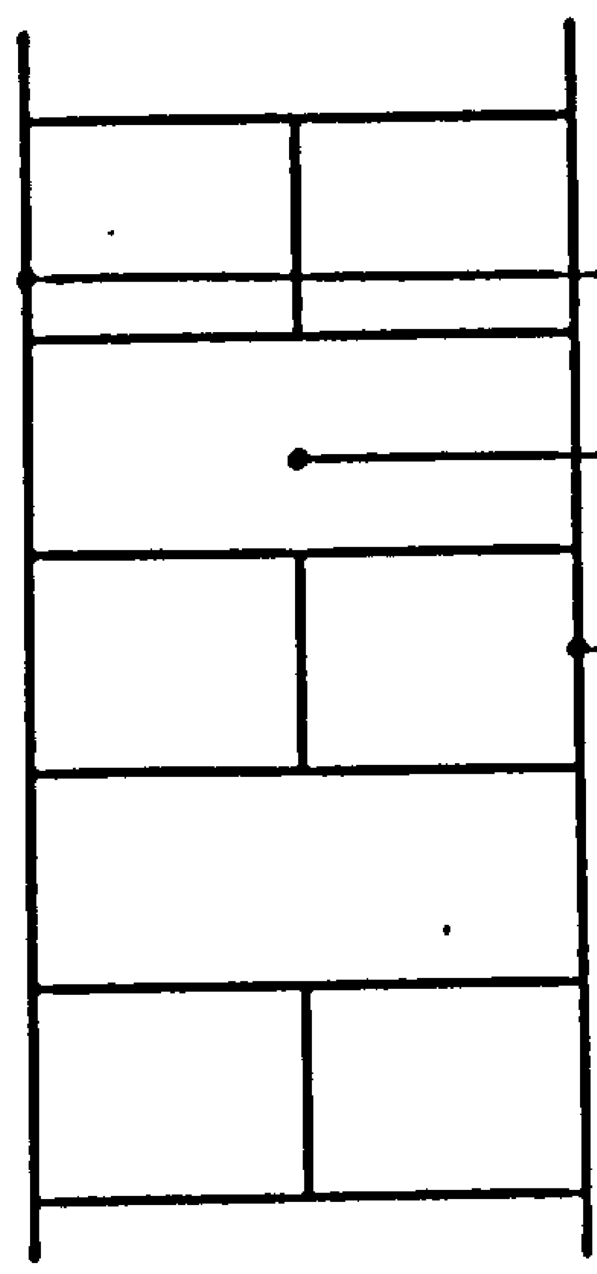


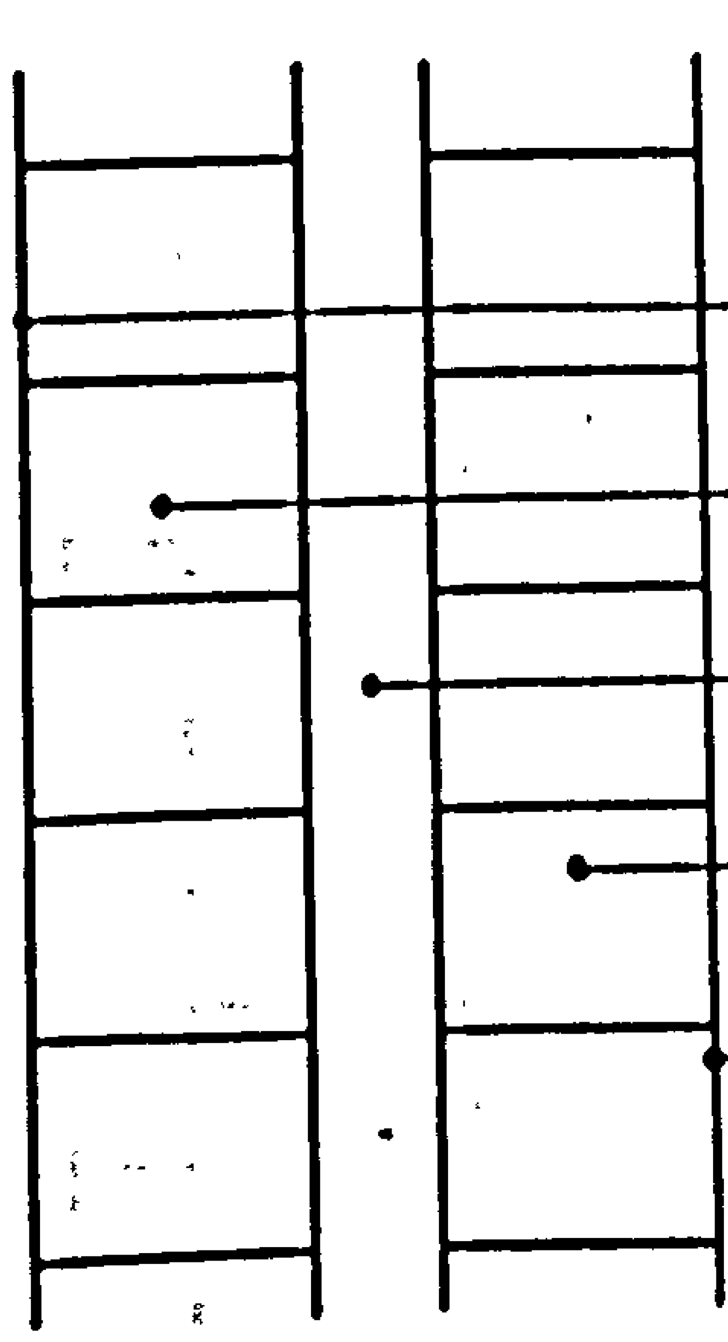
FIGURE 2.6
LAYOUT OF BLOWER AIR & ELECTROSTATIC FILTERS. TYPE 8 & 14 AREAS.[2]



SOURCE OF RESISTANCE	THICKNESS (m)	THERMAL COND. ($W/m^{o}C$)	THERMAL RESISTANCE ($m^{2o}C/W$)
SURFACE RESISTANCE	-	-	0.123
BRICK	0.23	0.84	0.274
SURFACE RESISTANCE	-	-	0.123

U-VALUE = 1.92 $W/m^{2o}C$

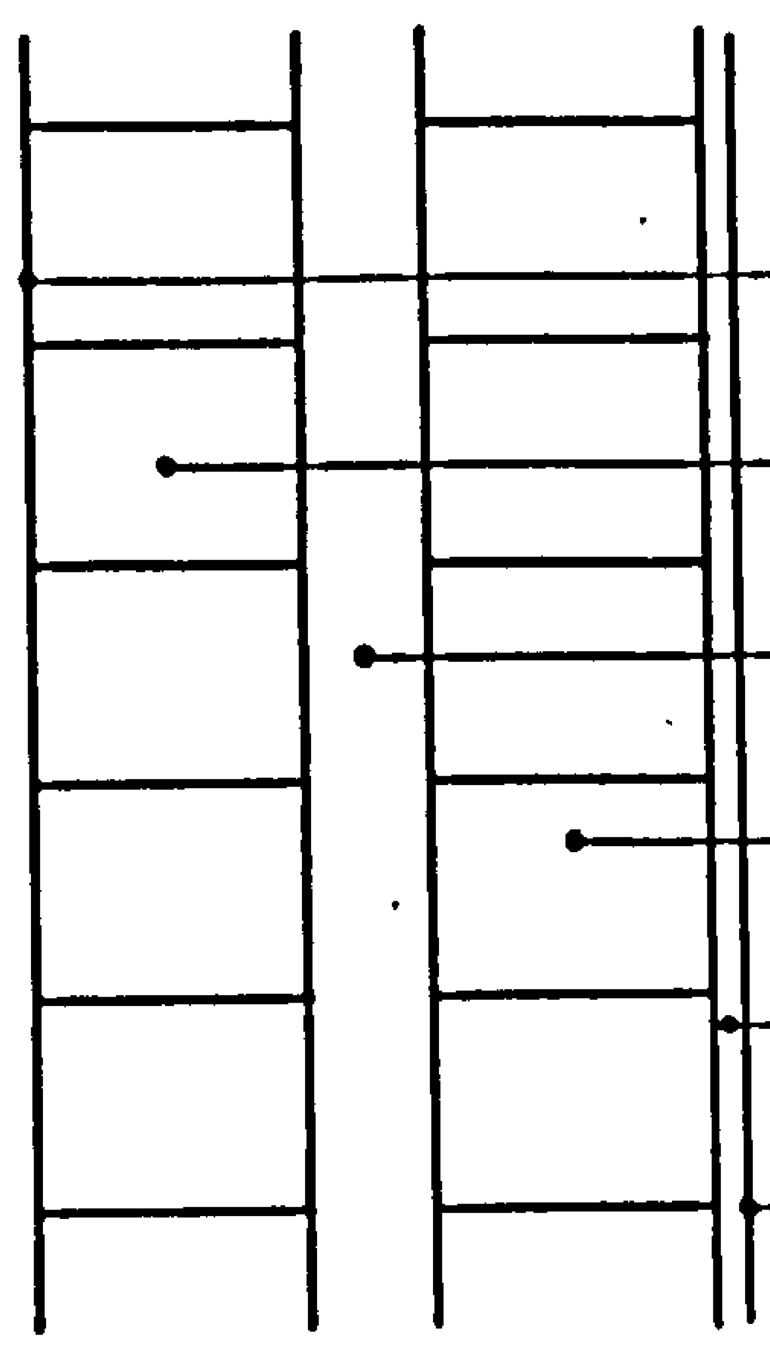
FIGURE 2.7 WALL A (INTERNAL PARTITION WALL)



SOURCE OF RESISTANCE	THICKNESS (m)	THERMAL COND. ($W/m^{o}C$)	THERMAL RESISTANCE ($m^{2o}C/W$)
SURFACE RESISTANCE	-	-	0.123
BRICK	0.115	0.84	0.137
AIR CAVITY	-	-	0.18
BRICK	0.115	0.84	0.137
SURFACE RESISTANCE	-	-	0.123

U-VALUE = 1.43 $W/m^{2o}C$

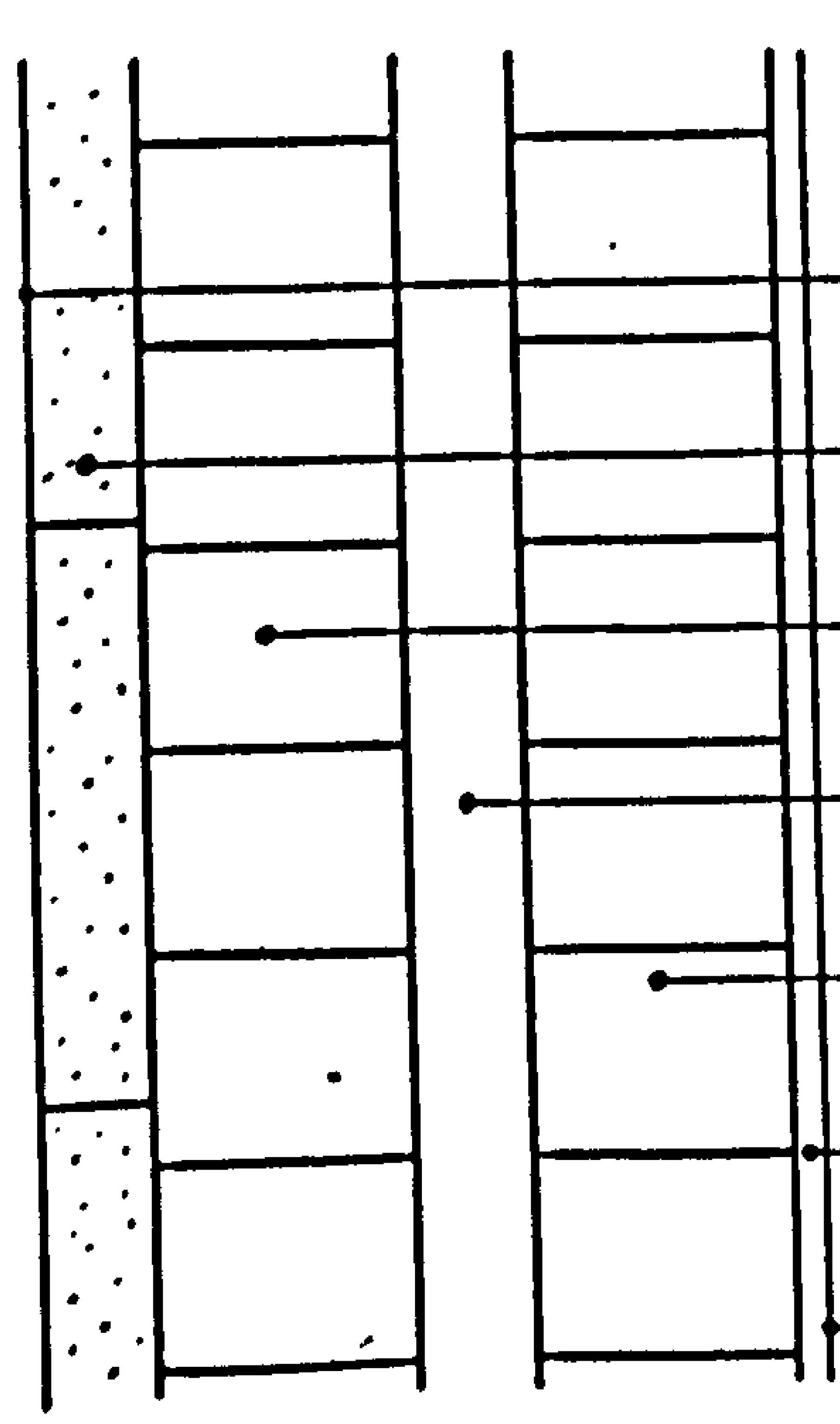
FIGURE 2.8 WALL B (INTERNAL PARTITION WALL)



SOURCE OF RESISTANCE.	THICKNESS (m)	THERMAL COND. ($W/m^{o}C$)	THERMAL RESISTANCE ($m^{2o}C/W$)
SURFACE RESISTANCE (INNER)	-	-	0.123
BRICK	0.115	0.84	0.137
AIR CAVITY	-	-	0.18
BRICK	0.115	0.84	0.137
SURFACE RENDERING	0.015	0.5	0.03
SURFACE RESISTANCE (OUTER)	-	-	0.055

U-VALUE = 1.51 $W/m^{2o}C$

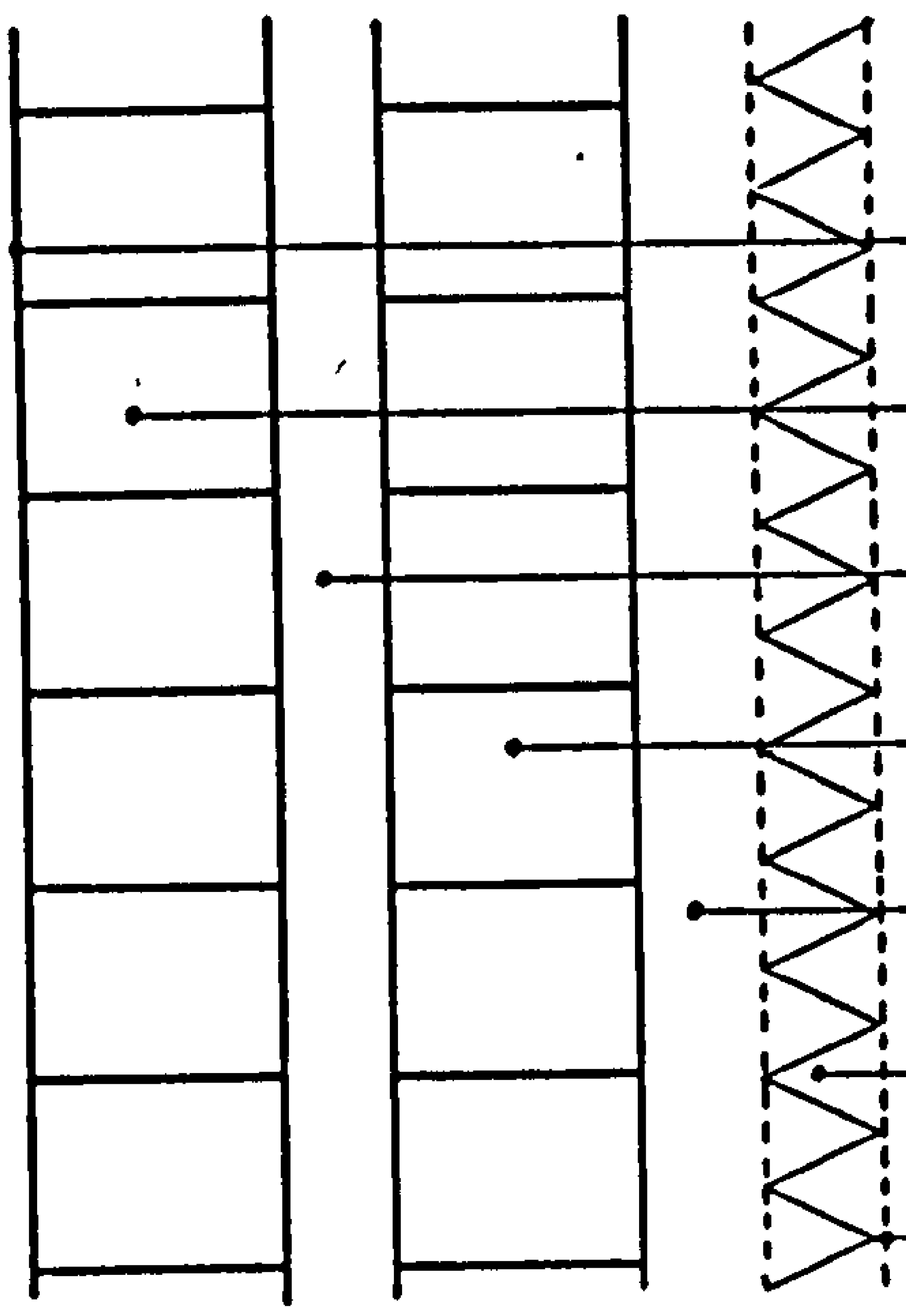
FIGURE 2.9 WALL C



SOURCE OF RESISTANCE	THICKNESS (m)	THERMAL COND. ($W/m^{o}C$)	THERMAL RESISTANCE ($m^{2o}C/W$)
SURFACE RESISTANCE (INNER)	-	-	0.123
INSULATION BOARDS	0.05	0.07	0.714
BRICK	0.115	0.84	0.137
AIR CAVITY	-	-	0.18
BRICK	0.115	0.84	0.137
SURFACE RENDERING	0.015	0.5	0.03
SURFACE RESISTANCE (OUTER)	-	-	0.055

U-VALUE = 0.73 $W/m^{2o}C$

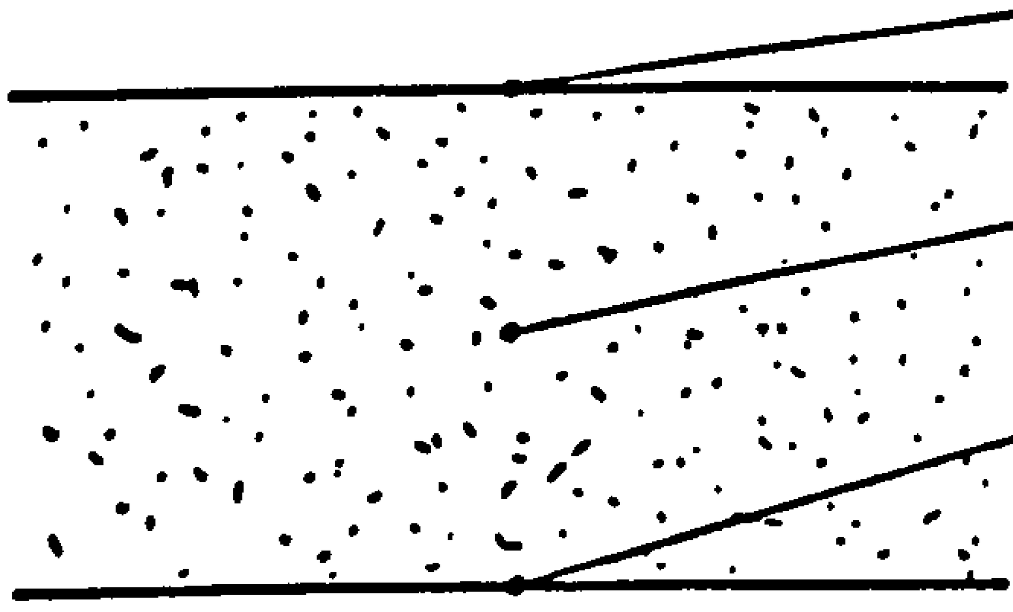
FIGURE 2.10 WALL D



SOURCE OF RESISTANCE	THICKNESS (m)	THERMAL COND. ($W/m^{o}C$)	THERMAL RESISTANCE ($m^{2o}C/W$)
SURFACE RESISTANCE (INNER)	-	-	0.123
BRICK	0.115	0.84	0.137
AIR CAVITY	-	-	0.18
BRICK	0.115	0.84	0.137
AIR CAVITY	-	-	0.18
CORRUGATED METAL DECKING	0.005	160	0.00003
SURFACE RESISTANCE (OUTER)	-	-	0.055

U-VALUE = 1.23 $W/m^{2o}C$

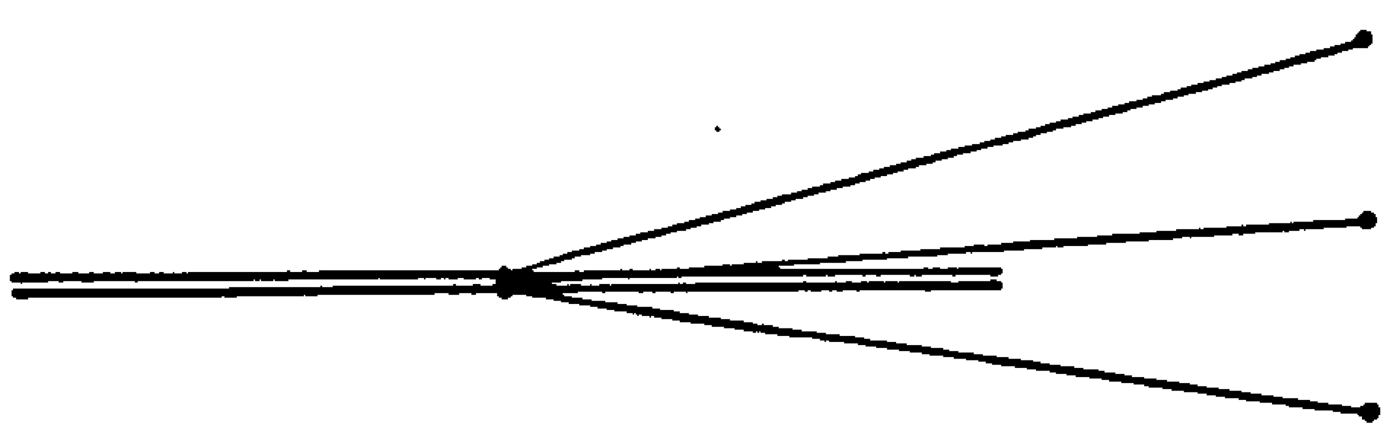
FIGURE 2.11 WALL E



SOURCE OF RESISTANCE	THICKNESS (m)	THERMAL COND. ($W/m^{o}C$)	THERMAL RESISTANCE ($m^{2o}C/W$)
SURFACE RESISTANCE	-	-	0.106
CONCRETE	0.2	1.4	0.143
SURFACE RESISTANCE	-	-	0.106

U-VALUE = 2.82 $W/m^{2o}C$

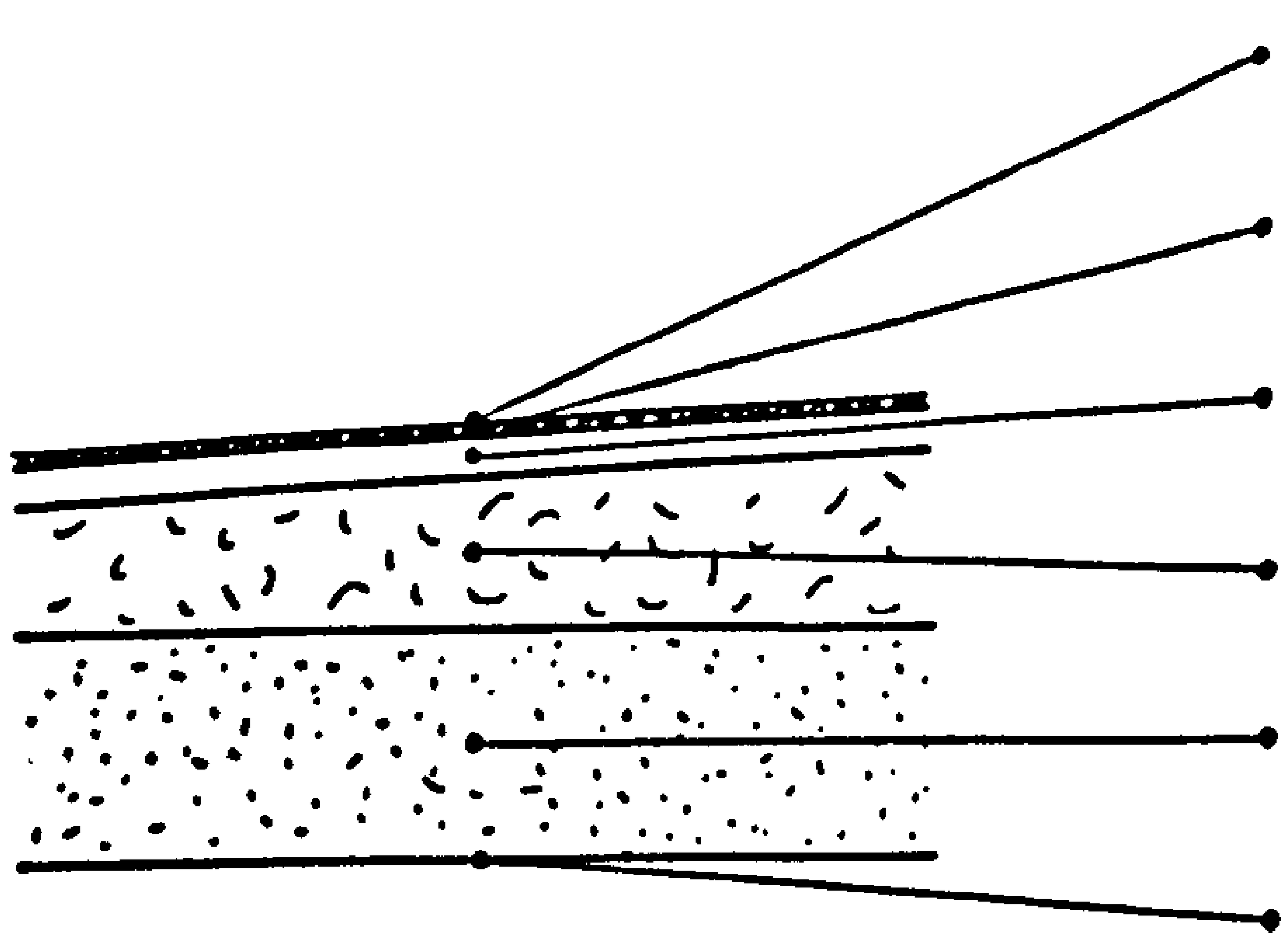
FIGURE 2.12 INTERMEDIATE FLOOR



SOURCE OF RESISTANCE	THICKNESS (m)	THERMAL COND. (W/m°C)	THERMAL RESISTANCE (m²°C/W)
SURFACE RESISTANCE	-	-	0.106
METAL	0.005	200	0.000025
SURFACE RESISTANCE	-	-	0.106

U-VALUE = 4.72 $W/m^2{}^\circ C$

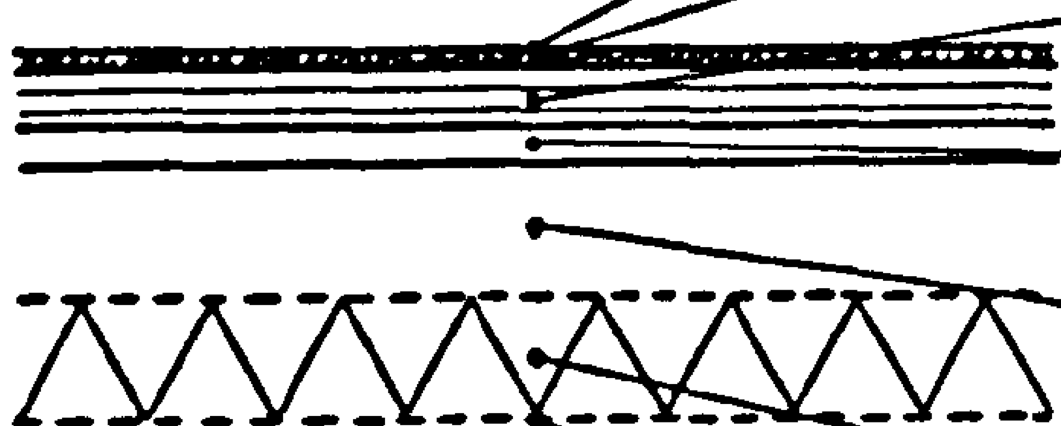
FIGURE 2.13 INTERMEDIATE FLOOR - METAL HATCH COVERS



SOURCE OF RESISTANCE	THICKNESS (m)	THERMAL COND. (W/m°C)	THERMAL RESISTANCE (m²°C/W)
SURFACE RESISTANCE	-	-	0.045
LIMESTONE CHIPS	0.01	0.3	0.033
ASPHALT	0.02	0.5	0.04
FOAM SLAG SCREED	0.065 (average)	0.32	0.203
CONCRETE	0.1	1.4	0.071
SURFACE RESISTANCE	-	-	0.106

U-VALUE = 2.01 $W/m^2{}^\circ C$

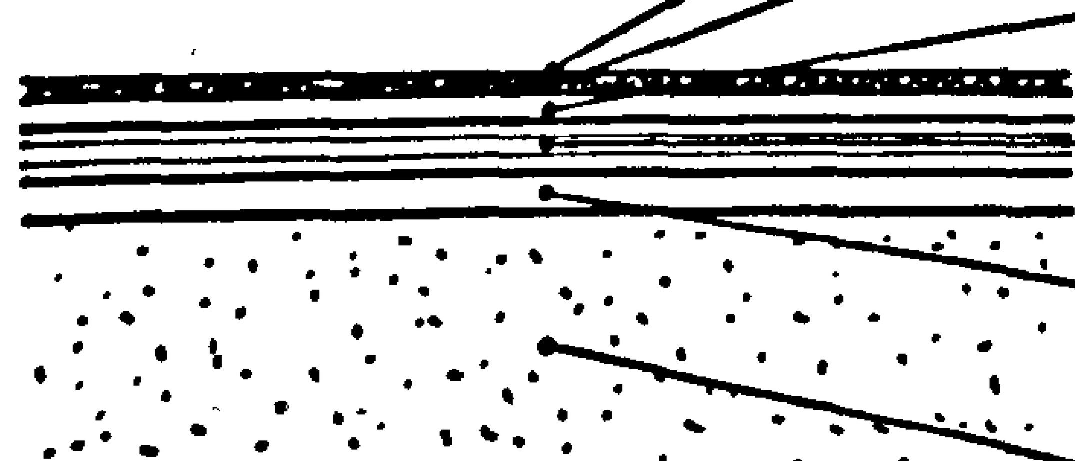
FIGURE 2.14 ROOF A



SOURCE OF RESISTANCE	THICKNESS (m)	THERMAL COND. (W/m°C)	THERMAL RESISTANCE (m²°C/W)
SURFACE RESISTANCE	-	-	0.045
LIMESTONE CHIPS	0.1	0.3	0.033
THREE LAYERS TREATED ROOFING FELT	0.02	0.5	0.4
FIBRE-BOARD INSULATION	0.013	0.057	0.228
AIR CAVITY	-	-	0.18
CORRUGATED METAL DECKING	0.005	160	0.00003
SURFACE RESISTANCE	-	-	0.106

U-VALUE = 1.58 W/m^2°C

FIGURE 2.15 ROOF B



SOURCE OF RESISTANCE	THICKNESS (m)	THERMAL COND. (W/m°C)	THERMAL RESISTANCE (m²°C/W)
SURFACE RESISTANCE	-	-	0.045
LIMESTONE CHIPS	0.1	0.3	0.033
"DUROTILES"	0.1	0.5	0.2
THREE LAYERS TREATED ROOFING FELT	0.02	0.5	0.4
FIBRE BOARD INSULATION	0.013	0.057	0.228
CONCRETE	0.1	1.4	0.071
SURFACE RESISTANCE	-	-	0.106

U-VALUE = 0.92 W/m^2°C

FIGURE 2.16 ROOF C

FIGURE 2.17 ELECTRICAL ENERGY USE IN SPINNING TOWER 1978-1981

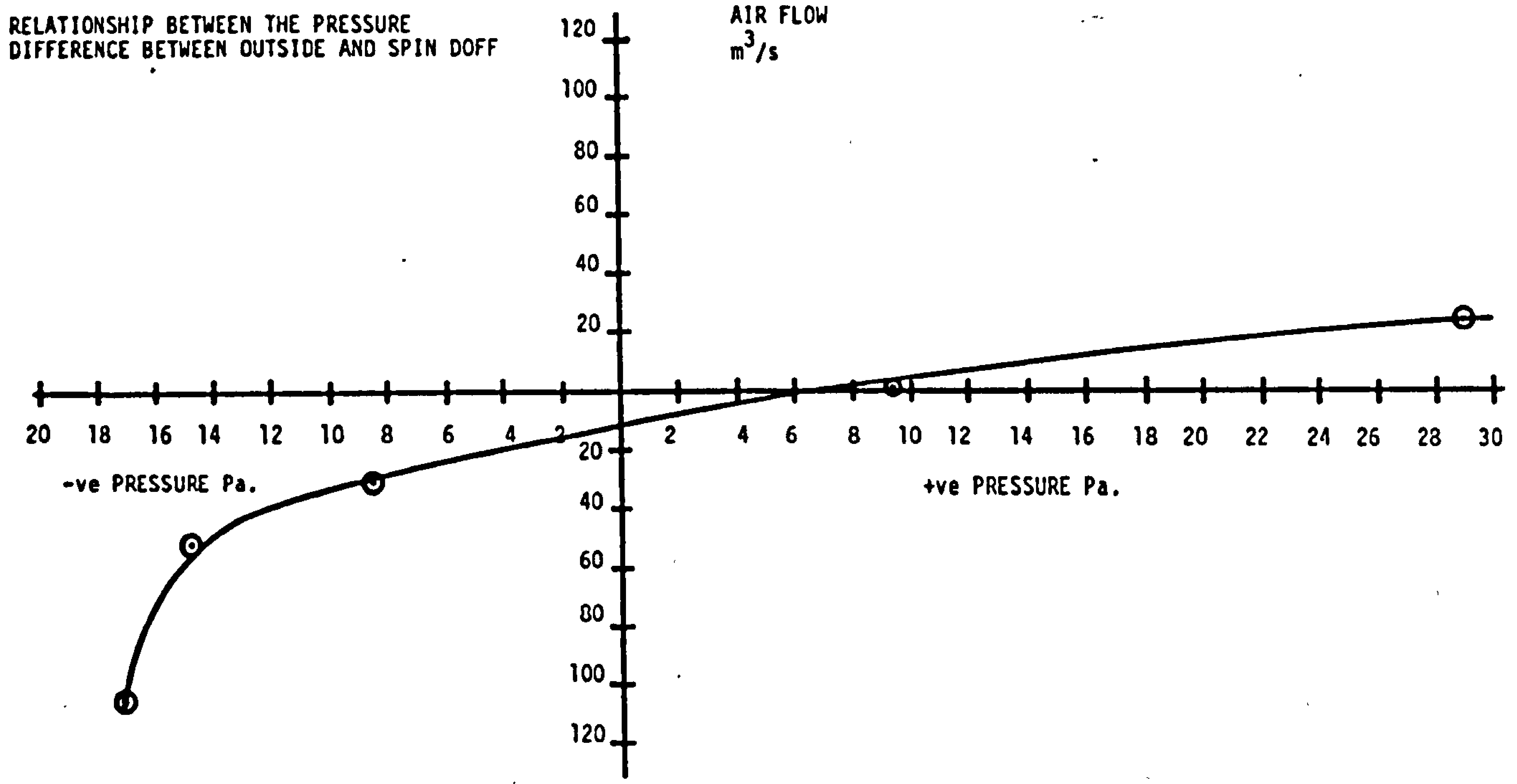


FIGURE 2.18 PRESSURE-FLOW RELATIONSHIP

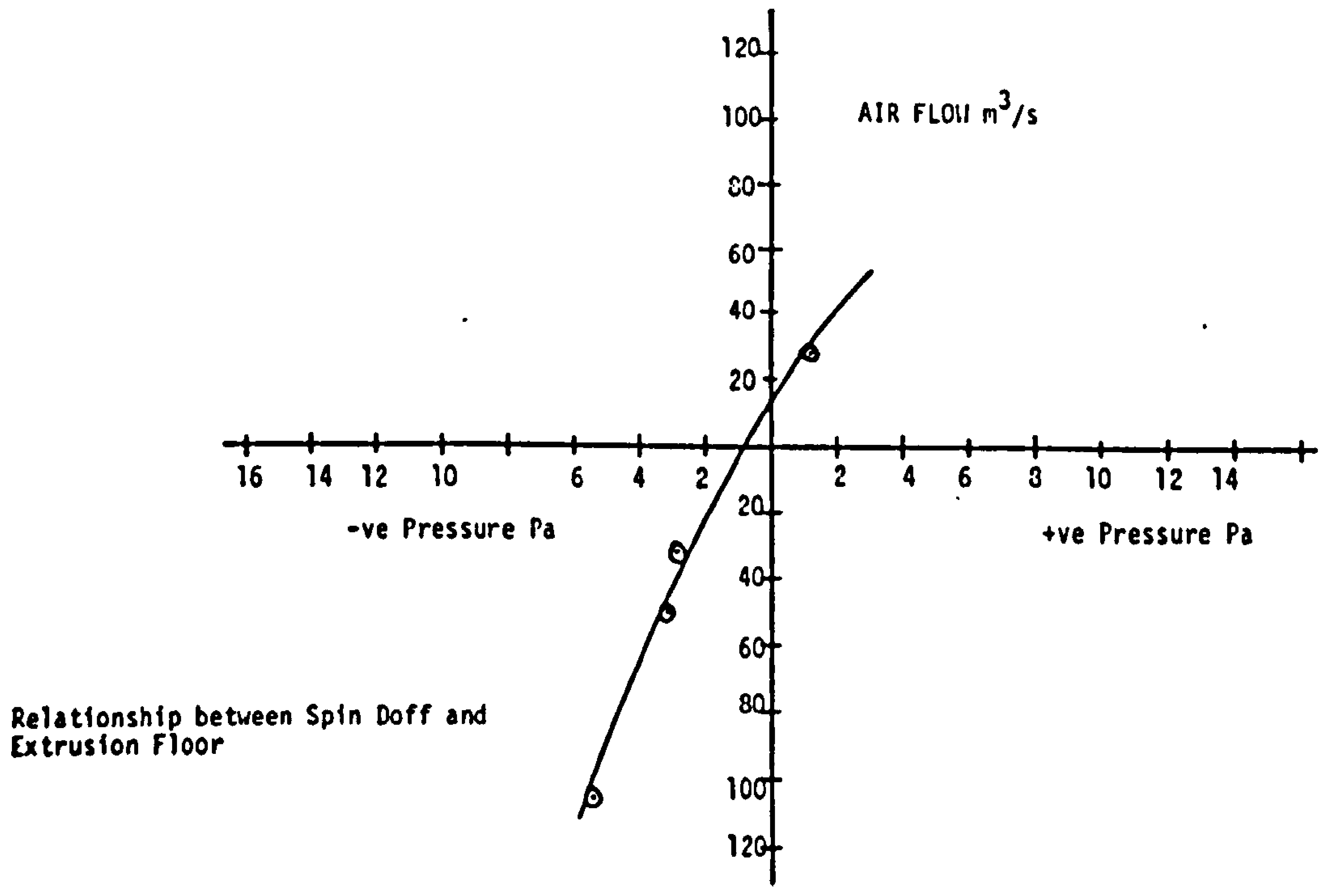


FIGURE 2.19 PRESSURE-FLOW RELATIONSHIP

CHAPTER 3

METHODS FOR THE PREDICTION AND INVESTIGATION OF AIR MOVEMENT IN BUILDINGS

3.1 INTRODUCTION

Various researchers have explored a number of routes for the investigation of air movement in buildings. The aim of most studies has been the prediction of air movement through certain standard structural elements or building types, with a view to assessing and minimising infiltration and ventilation heat losses.

Investigation of air movement was often aided by the use of physical models. These may have been of whole buildings at a reduced scale or of certain sections of the building at a larger, or even full scale.

Some researchers have used analogue methods which have included hydraulic and electrical simulations. Solutions using digital computers have also been widely employed. In these cases a mathematical model of the flow was presupposed in one form or another. The basic model could be elaborate, or could be condensed to give a simpler approach, the results may have been less accurate but the model could be more easily used and adapted.

Full scale tests have been carried out both on specially designed test houses and various types of buildings, both when occupied and when unoccupied. Pressurisation and tracer gas techniques were the main investigative methods employed.

Hitchin and Wilson[1] presented a review of experimental techniques for the investigation of natural ventilation of buildings in 1967. The majority of their paper dealt with the uses of tracer gases and to some extent is now out of date. Air movement measurement was also reviewed concentrating mainly on the various types of anemometer available. Air flow patterns could be observed using a number of visible tracer methods. Modelling techniques were mentioned though very little detail was given.

Since the publication of the Hitchin and Wilson review many changes in technique and method have occurred. It was decided therefore to undertake a further review, to include more recent work.

3.2 PHYSICAL MODELLING

One of the problems in conducting tests to determine air flow is the susceptibility of buildings to variations in weather conditions. Additionally once the building has been built it is difficult to alter location and orientation. Physical models, particularly those on a reduced scale, lend themselves more easily to investigation. They can be moved around with relative ease and in the laboratory various environmental conditions might be simulated with the advantage of repeatability and consistency.

3.2.1 WINDTUNNEL MODELLING

Small scale models have been used in wind tunnels, the main use of this method being to determine relationships between wind speed and the distribution and magnitude of wind pressures over buildings. The techniques for such modelling have been developed over the past 75 years. In most studies, air flow has been produced using fans. In the early days "plain" tunnels were used, often with the model suspended in the middle. Modern designs however, have the model placed on the floor of the tunnel, and the accurate modelling of the air flow distribution is attempted, so as to resemble the real life situation as closely as possible.

Many studies, especially those relating to tall or large structures have been undertaken in order to assess safety limits, so that building designs can withstand the stresses imposed by the wind. In such work the basic speed(on which guidelines are based) was the "3 second gust speed exceeded on average only once in 50 years"(2). This is much higher than typically average wind speeds upon which ventilation rates should be based, however the wind tunnel studies performed in this context did yield useful results relating to pressure distributions.

It has been usual to express the results of wind tunnel studies as a set of pressure coefficients. These related the pressures measured at the model surface to the free stream pressure in the tunnel. This free stream pressure was often referred to as the "dynamic head" or "dynamic wind pressure". The pressure could be calculated as that due to fluid in motion, according to Bernoulli's theorem. The pressure coefficient was defined as

$$C_p = \frac{\text{wind pressure at building surface}}{\text{free stream wind pressure}} \qquad (3.1)$$

Use of pressure coefficients non-dimensionalised test results enabling the data of model tests to be made use of in full scale predictions.

For valid test results it should also be necessary to maintain the same dimensionless flow parameter (known as the Reynolds Number) in the model as at full scale. This was often impracticable; however it has been found that for simple building shapes a lower Reynolds Number can be allowed and this is discussed later.

Air flowing across the surface of the earth is in a state of turbulance due to friction effects between the rough surface and the moving air. Much research has been done to simulate the ensuing air movement (which varies with height and terrain) with varying, but gradually improving, degrees of success. Whilst modern studies exhibit good simulation, early studies were not so accurate.

Bailey and Vincent[3] worked with the 3 ft by 3 ft wind tunnel of the National Physical Laboratory. Seven types of building were studied with varying heights and types of roofs. The model scale employed was 1/240. At this stage the simulation of wind speed distribution with respect to height and terrain was not very far advanced although a wind gradient was recognised as existing. In this case the main concern seems to have been to correctly simulate the wind speed at a reference point 2 inches from the model ground surface (scale height 40 ft). Previous to this study, results of wind tunnel tests had been compared with full scale by Bailey. These showed reasonable agreement, though leeward pressure drops had seemed to be up to about 50 % greater than would have been expected from the model.

A 60 cm square cross-section tunnel was used by Jensen and Franck[4]. They also recognised the existence of a turbulent boundary layer over the earth's surface: variations in ground roughness, causing such a layer, were simulated. A model law based on a "roughness parameter" was developed and experiments to investigate sheltering for houses and smoke dispersal carried out. These studies were of importance since the modelling was based on full scale data and a variety of terrains could be simulated using different types of surface roughness.

A summary of world-wide wind tunnel simulations of the atmospheric boundary layer was presented by

Hunt and Fernholz[5] in their report on the 50th Euromech Colloquium, Berlin (1974).

Modelling scales used by those present at the colloquium varied from 1/20 to 1/2000, with one exception scale of about 1/14000 being used for air movement over mountains. One of the conclusions was that modelling laws could not allow all things to be simulated at the same time, since laws of similarity based on dimensionless groups could only be obeyed within certain limits. Some papers were concerned with energy transfers, others with dispersion of chimney gases. The heat transfer between the ground and the air was difficult to simulate because the model temperature differential needed to be so high to maintain the correct dimensionless number. Water test rigs were reported as being used in some cases to simulate conditions when a higher Reynolds Number was required.

Construction of a suitable wind tunnel to simulate atmospheric boundary layers has been described by a number of authors, Lee[6] for example dealt with a wind tunnel 1.2 m x 1.2 m in cross section, the general approach for construction was reported as being that of Counihan[7]. Cook[8] also described wind tunnel construction and stated that the best simulations of atmospheric boundary layer were given when the flow and turbulence was allowed to build-up over a long section of roughness wall in the wind tunnel, prior to the working section. This required very long tunnels however, (well in excess of 25 m) which was impracticable for most laboratories. Alternatively a boundary layer might be instantly created by artificial means by placing blockages in the air stream immediately before the working section. This method did not simulate the boundary layer very accurately though. Most wind tunnels used an intermediate form where the layer was given an artificial start but was then allowed to develop over a roughness length.

A normal tunnel would consist of honeycombe at the air inlet followed by a "barrier" which created a momentum deficit and gave depth to the boundary layer. This would be followed by a "mixing device" to help develop the layer. The "surface roughness" would then be encountered, it was the most important part - it acted as a "momentum sink" and set up stresses through the layer which in turn controlled the mean velocity profile and the turbulence characteristics. A tunnel of such construction would give a reasonably accurate simulation of the boundary layer without the need for a long roughness section for the flow to develop.

In Lee's tunnel the "barrier" was a castellated fence, the "mixing device" a row of spires, and the "surface roughness" was created by a regular pattern of blocks placed on the floor of the tunnel. This general arrangement is shown in Figure 3.1. The scale factor for the boundary layer produced in this tunnel was 1:350 for urban conditions and 1:500 for rural conditions. Different sizes of the roughness element blocks being used to simulate the different terrain conditions.

Most wind pressure coefficient data is available from wind tunnel studies of tall buildings. Some work on low rise buildings has also been carried out, for instance Lee et al[9]. In that study, pressures were predicted for low rise buildings located amongst a set of similar buildings taking into account geometrical form and spacing parameters. Such data was obviously useful for the majority of buildings, which are only one or two storeys high. The need for such data however is not so great since a large number of full scale studies have been carried out on houses and similar size dwellings (discussed later). Whilst such full scale data cannot be so easily generalised, it gives an inherently more accurate result for a particular case.

Though much wind tunnel work was carried out on scale models of the order 1/400 of full size, wind tunnels were also used for larger scale models. Smith[10] discussed many aspects of the use of models for ventilation determination. Strictly speaking he did not use a wind tunnel, but an "Air Flow Chamber" (see Figure 3.2) which was basically a hexagonally shaped room, and he did not try to model the atmoshpheric boundary layer. Smith was more interested in determining air flow patterns; titanium tetrachloride smoke was used to illustrate these. A scale of 1/16 was used in the tests, which were shown to be reasonably valid after comparison with full scale. The problems of scale effect were mentioned, and though dimensionless groups such as the Reynolds Number could not be simulated properly (given the limitations of the model) the test results could be used with reasonable confidence if certain conditions were satisfied.

Smith developed an expression based on the aerodynamic radius which had to take a value above a certain critical level. (The aero-dynamic radius was defined as "twice the projected area of the building normal to the direction of air flow, divided by the perimeter past which the air flows"). This expression was given by:-

$$E = \frac{2A}{P} v \qquad (3.2)$$

Where A = Projected area of model normal to air flow

P = Perimeter past which air flows

v = Free stream air speed

For the model analogy to be valid, E was to exceed 2000. This allowed model air speeds to take values considerably lower than those required to maintain the same Reynolds Number.

Other conditions for the validity of model test results were that changes in air density and viscosity had to be negligible, and that the effects of thermal convection had also to be negligible.

In a study of the effect of wind air movement inside buildings, one method of investigation used by Malinowski[11] was that of a wind tunnel. The side of a model room or compartment was mounted on one side of the wind tunnel so as to form part of the tunnel wall. At the beginning of each test the compartment was filled with carbon dioxide or nitrogen, the only opening into the box being through the side attached to the tunnel. A controlled air flow was set up through the tunnel and the percentage of oxygen in the compartment recorded at regular intervals. The rate of air exchange was calculated for different external flow rates and different testing conditions. One of the objectives of these model tests was to investigate air exchange due to pulsating and turbulent flow. According to a steady state theory the model wind forces on the compartment should have remained constant, thus for a single opening between the compartment and tunnel, there would have been no air exchange. However pulses in the flow and infiltration of turbulent eddies did cause some air exchange. In tests using two coplanar holes with the compartment wall parralel to the air flow, the ventilation rate increased when the spacing between the holes increased; when the mean air velocity increased; and given a constant mean air velocity, when turbulent intensity increased.

The model chosen by Bilsborrow and Fricke[12] avoided some Reynolds number similarity problems by desiging the holes, which allowed air flow, to have dimensions which would give model flow values of the same order as full scale Reynolds Number. The model was placed in a wind tunnel and both internal and external pressures, and flow rates were measured, the latter by a novel use of an orifice plate mounted in the model (see Figure 3.3). The model results were compared with predicted values from a computerised digital analogue. The model ventilation rates were lower than the computer predicted rates by up to almost 25 %. Unfortunately a comparison between measured full scale readings and the computed values made by Bilsborrow proved to be inconclusive.

A model building in a wind tunnel was also used by Etheridge and Nolan(13). The model measured 400x210x180 mm and used two types of opening (circular holes and model windows) to simulate actual ventilation openings. Measurements were made in a low speed wind tunnel using a tracer gas decay technique. The model was initially filled with 0.8 % helium in air and a katharometer was used to monitor the decay. The main objective of this work was to investigate the relative magnitudes of the ventilation due to steady pressures and to fluctuating pressures. Although the wind tunnel did not give a complete simulation of the full scale wind, the model did generate its own turbulence and in that respect these tests gave a better simulation than those of Malinowski.

The results for the model windows showed a strong scale effect (ie, dependence on Reynolds Number), though the circular holes proved to have better characteristics and the authors suggested modelling leakage areas in buildings by "equivalent circular holes". Turbulent pressure fluctuations increased the rate of ventilation above that caused by steady pressures by about 1/6.

3.2.2 SECTION MODELS

The problems of scale effect experienced in some of the previous studies were avoided by modelling critical flow openings at full scale. This has been done, in particular, for windows since the crack dimensions are usually small and the scaling down operation would have adverse effects on the validity of the model. Thomas and Dick[14] investigated three main types of windows in use in buildings, each specified in British Standards. There were:-

i) the standard wood frame

ii) the "modified" standard wood frame and

iii) the standard metal frame windows

The range of frame clearances chosen was between 0 and 0.1 ins (0 and 2.5 mm) and the range of pressures was 0 to 0.5 ins wg (0 to 124.5 N/m^2). There were two stages in the experiments; first a small uniform length of each of the windows was studied to obtain infiltration characteristics, secondly the flow through the whole gap perimeter of the complete window was measured for non-weather-stripped and weather-stripped cases. Since the construction and dimensions of the short lengths of window frame were more carefully controlled than in the full windows, these first readings gave more consistent results. The testing of windows for air leakage by pressurisation techniques is now carried out by a number of manufacturers, official bodies and research groups. However comparison between laboratory tests and in situ building tests showed some variations in characteristics(15) which may mean not all figures are reliable.

In their study, Hopkins and Hansford(16) used test sections of typical cracks to investigate the flow and its dependence on Reynolds Number. The pressurising test box shown in Figure 3.4 was used to draw air through the three types of crack. These were:

i) a "straight through" crack - as between a door and the floor

ii) an "L-shaped" crack where the air flows round a right angled bend, as between a casement window and frame, and

iii) a "multi-cornered" crack with two right angled bends such as might be found in the groove of a sash window

Crack thickness was varied between 0.5 and 7.5 mm, which were the limits of cracks likely to be found in normal dwellings. A theory was developed to relate discharge coefficient to the crack dimensions for a given Reynold's Number. The experimental results were incorporated into a semi-empirical equation. Deviations from the predicted relationship were explained in terms of change of flow. This work showed that Reynolds Number was an important consideration in relation to cracks, and the authors made a contribution towards quantifying this.

Ideally of course all tests might be carried out at full scale in order to get the best simulation of real conditions. For his comparison with model studies Smith(10) had a full sized house built of common design. It was positioned on a track so that it could be moved to any orientation to take advantage of the prevailing weather conditions. In order to measure full scale wind pressures the Building Research Establishment had a house constructed, in which certain parameters, such as the pitch of the roof, could be changed. Potter(17) described a test room used for fluctuating wind pressure studies which was situated on the first floor of a three storey dwelling in a housing estate. He concluded that an assumption of steady-state pressures gave a prediction of flow rates less than observed rates, or predicted rates using a dynamic method.

3.2.3 THE LIMITATIONS OF PHYSICAL MODELLING

There are a number of advantages to the use of physical models. In relation to the study of infiltration, ventilation and air movement the modelling, in an artificial environment, of a building, building section or wall construction, means that tests are not dependent on the variability of weather. Tests for given conditions can be repeated and this enables different building designs to be compared under the same conditions. There is no need to waste time waiting for a required condition to occur naturally. In general the cost of performing model studies is less than that of full scale tests, so that if the model provides valid results, savings can be made. These factors have led to the fairly widespread use of physical modelling for the investigation of certain parameters. However a number of limitations must be placed on the circumstances in which physical models give valid results, especially if quantitative results are required.

It has already been mentioned that the use of smaller than full size models can lead to a scale effect change. Wannenburg and Van Straaten (18) defined it thus - "a model may be said to be subjected to scale effect if change in Reynolds Number results in a change in the various non-dimensional flow parameters. For example, if pressure measured at a point on the model, expressed non-dimensionally in terms of wind velocity head, is found to be constant over a range of varying Reynolds Number, then the model might be considered free from scale effects over the given range of Reynolds Number".

(The Reynolds Number (R_e) is an indication of the ratio of inertia to viscous forces in fluid flow, and is dependent on the density, viscosity and speed of the fluid, and on the linear size of the model.)

Ideally then the Reynolds Number in model tests should be the same as in the full size equivalent. Unfortunately this is not possible in most models since the much reduced linear dimensions imply an unacceptable increase in flow rate to compensate. However it has been found that objects with sharp edges or corners, and plane faces, usually give the same flow pattern over a wide range of airspeeds. For the experiment to be valid the Reynolds Number should exceed a certain "critical value" (see for instance Smith[10]), but this can be a value much less than that of full scale. This has meant that common building shapes with such edges and faces can be modelled in a wind tunnel. External pressure distributions can be found and other qualitative results gathered, quantitative results especially of internal parameters have a validity much more open to question. For certain critical openings, such as cracks around windows, doors and other components, a change of scale can have quite marked effects as illustrated by Etheridge[19]. Thus whilst the use of model tests to determine external pressure distributions is an established procedure, (and the data is used in theoretical predictions of ventilation), measurement of ventilation rates in models is not.

Aynsley[20] has tried to circumvent some of these problems by use of solid and porous models in wind tunnel studies. The solid models established external pressure distributions, whilst the porous models were used to give more life-like results. His study was concerned with the adventitious use of wind to create ventilation in buildings in the hot humid areas of Australia.

Temperature variations and heat flows can also influence results. In general wind tunnel tests are most suited to simulation of high wind speeds, above 10 m/s, since at such speeds any thermal effects are swamped by wind turbulence. Therefore wind tunnel results are often used for high speed loading and safety simulations.

Because of the aforementioned problems with scale effect, air movement data, especially of a quantitative nature is very difficult to amass for flow within models. As stated, the Reynolds Number can be allowed values less than full scale when sharp edges and plane surfaces are considered, however interior surfaces do not often conform to this rule. The study by Smith[10] does show that in certain accurate models, the full scale air flow pattern can be reasonably simulated. For cases where it is not only the flow pattern that is to be modelled, but also other factors, such as heat transfer, the simulation becomes almost impossible.

Hitchen and Wilson reported the work of Jakob[21] in which he stated that to model free convection completely, twelve dimensionless ratios (involving eleven flow parameters) must be preserved, although some tests could succeed if the ratio of inertia and buoyancy forces (Froude number) was maintained.

Moog[22] investigated reduced scale models in a study concerned with the problems and predictions of room air flow with air-conditioning. He found the micro-structure of the flow to be extremely complex such that a mathematical model was very difficult to form, if not impossible. Model scale tests under both isothermal and anisothermal conditions yielded results which could increase the validity of a mathematical model, which would otherwise lack dimensional certainty. In order to exactly reproduce three dimensional room flow, Moog considered it essential to perform tests at full scale. However in some cases such as in factories it was impossible to perform full scale tests, and in this and other similar cases he allowed the use of models.

3.3 ANALOGUE MODELS OF AIR MOVEMENT

Studies in which analogues have been used to simulate air movement fall into three main categories: digital computer, electrical and water. Digital computer studies will be discussed in a later section.

3.3.1 WATER ANALOGUES

Models in which water is used represent air have been used since the required higher Reynolds Numbers are more easily obtained due to the properties of water. Despite this Kurek[23] in his study using a water analogue, makes no calcualtion of Reynolds Number or comparison with full scale air movement figures.

He used a building model half immersed in water. The flow of water past the model caused variations in depth which Kurek assumed to be proportional to wind pressures. Such a model is more useful for demonstrating general effects and flow patterns, rather than to obtain quantitative results for pressure and flow parameters.

Qualitative information on fluid movement near holes such as cracks in building surfaces, was gathered by Malinowski[11] using water flume tests. His main objective was to study wind effects, including turbulence on air movement inside buildings. Mixing processes and density gradients were studied with the compartment filled with clear water and the external water distinguished by use of aluminium powder. In order to study motion inside the model building, aluminium powder was put into both internal and external water. The patterns caused were recorded on film at an increased speed, and studied later at reduced speeds.

A more complex simulation was attempted by Rydberg[24] as reported by Hitchen and Wilton[1]. Brine was used as the fluid in order to model openable windows across which a temperature gradient existed. The concentration of salt was varied thus allowing different air densities to be modelled and permitting heat transfer due to fluid motion to be illustrated in a restricted sense. Heat transfer, within air, simulated by diffusion of dissolved salts, was found to be too impractical.

From these studies it can be seen that whilst the use of water analogues to simulate air movement offer certain possibilities, particularly with regard to simulation of Reynolds Number, their use has been limited.

3.3.2 ELECTRICAL ANALOGUES

Air movement has also been simulated by electrical analogues or analogue computer. Such techniques have usually been developed from pipe or duct flow analogues since in such cases the flow of air is very similar to the flow of electricity in wires. Scott[25] developed three electrical analogue methods for the flow networks of pipes and ducts. The basic flow equation for air in pipes being given by:-

$$\Delta P = P_1 - P_2 = RQ^z \qquad (3.3)$$

P_1, P_2 are pressures at different points

Q = Flow rate

R (resistance) and z (flow exponent) are constants dependent on Reynolds Number and Pipe Roughness

For an electrical analogue, elements were required which would exhibit similar flow characteristics, that is:-

$$V = KI^z \qquad (3.4)$$

V = Voltage

K = Resistance (constant)

I = Current Flow

z = Current exponent (constant)

Tungsten lamps working with the range 25 to 100 percent of their rated voltage were found to have such a characteristic. This enabled Scott to construct an analogue based on these lamps though the system was expensive to run, had a high initial cost and required many spare resistors. Another of his analogues was computer based: in this case variable linear resistances were made to behave in the required exponential manner by use of servo-mechanisms to alter them. This analogue computer was found to be better than the tungsten lamp set-up but still very costly so Scott developed a third analogue. This was a "manually operated network calculator" which was similar to the analogue computer but with adjustments to reistances made manually. The calculator basically simulated a process of mathematical iteration and its main advantage was a reduction in cost compared to the other analogues.

Jackman[26] compared two types of analogue in his ventilation study of tall office buildings; one being a digital computer analogue, the other an electrical analogue. This electrical analogue was developed by the Institute for Public Health Engineering TNO, in Holland. The instrument, which had been designed specifically for ventilation studies, used one or more electric lamps connected in parallel, in combination with series or shunt resistances, or both, to simulate each door or window crack. The electrical resistances had characteristics given by the equation below:-

$$i = C_1 (E)^{1/n} \qquad (3.5)$$

i = electrical current

C_1 = coefficient of electrical resistance

E = potential difference

$1/n$ = flow exponent

The values of C_1 and $1/n$ could be adjusted to give the correct relationship. Currents corresponded to air flows and potential differences to pressure differentials. Pressures caused by wind and stack effects were simulated by the application of voltages at the corresponding points on the analogue circuit. Currents and voltages at selected points were measured and indicated on a panel, corresponding to flows and pressure differences. This electrical analogue showed favourable results when compared to the digital computer results of Jackman, sufficient for the conclusions of the study to be incorporated into the IHVE/CIBS[27] Guide.

The use of electrical analogues for air flow studies was a suitable form for investigation of flow in ducts and pipes, since the flow equations for air and electricity were quite similar and could be manipulated to give good agreement. The simulation of natural ventilation in a general sense was much more difficult especially if the system of interconnected rooms and corridors had large openings and exhibited low airspeeds.

3.4 MATHEMATICAL MODELS AND DIGITAL COMPUTER ANALOGUES

The application of flow equations and a continuity equation to air movement can be used to calculate flow rates. A number of pieces of information must be incorporated into a mathematical model, describing the air flows, that can then be solved. As the amount and variety of initial information, on which the flow equations are based, increases, so does the complexity of the solution. If a building or structure is considered in which the flows between each room or compartment needs to be calculated, then it becomes necessary to use a computer to solve the resulting equations.

Digital computers have been used by many researchers to deal with the problem of solution of flow equations and prediction of subsequent air movement. The accuracy of these techniques depended upon the accuracy with which the mathematical model was set up and the factors which were taken into account. Some studies have concentrated on the calculation of an overall ventilation rate for a building, others have been concerned with a full description of the air movement within a building.

3.4.1 STUDIES, MODELS AND EQUATIONS

In an air infiltration study by Harrison[28] the information required for the calculations of flows in a simple, uniformly glazed office block consisted of:

i) the pressure across windows

ii) the crack width around windows

iii) the air flow per unit crack length at the operating pressure, and

iv) the length of crack per unit volume of the room.

External pressure due to temperature difference was assumed equal for all sides of the building and was found using the equation of Thomas and Dick[14].

$$P = 2.8 \times 10^{-5} h \ (\theta_i - \theta_o) \qquad (3.6)$$

P = pressure - ins wg

h = building height - ft

θ_i, θ_o = inside and outside temps (^oF)

Four main classes of building were considered by Harrison - these are illustrated in Figure 3.5. The resistances of the flow paths were assumed to occur in simple ratios (the internal and stairwell resistances being multiples of the external wall resistances). Building heights of 50 or 100 ft (15 or 30 m) were used in the calculations, in which only a basic orientation perpendicular to wind direction was considered. The pressure drops across the windward faces were computed for each of the cases indicated in Figure 3.5. In general the results show that stack effect had a large influence except for type I building (each floor sealed from ones above and below). Also the height of a building needed to be taken into account in any generalised flow prediction, because of stack effects.

The general methods of calculation for air movement in multi-storey building were dealt with in a paper by Svetlov[29]. He recognised the difference between pipe flow and flow through building components - he suggested the following equations; for flow through cracks around windows:

$$h = Ax + Bx^2 \tag{3.7}$$

and for flow through doors, extraction ducts and open apertures:

$$h = Sx^2 \tag{3.8}$$

where h = Pressure Differential

x = Air Flow Rate

A, B, S = Specific Resistances

Using a computer and various mathematical techniques Svetlov reported that the solution for a 50 element building was possible in under five minutes. Of course the deisgn and size of computer is one of the main factors here. A comparison between the computed solution and the flows found from a hydraulic analogy gave variations at the flow junctions (or nodes) of less than 3 % on average. This was explained as being due to errors of up to 7 % in the hydraulic analogy.

A paper by two other Russian researchers, Bogslovskii and Titov[30] was mainly concerned with making allowances for infiltration and ventilation losses in heating duty calculations. Account was taken of wind and stack pressures as well as mechanical ventilation. In all of the cases mentioned however, only simple building shapes were considered - rectangular multi-storey office type buildings in which each floor level was identical.

In order to produce accurate figures of infiltration heat losses for designing heating systems, Gabrielsson and Porra[31] developed a digital computation technique. They claimed that their program gave good simulation since in addition to the accurate infiltration loss prediction, it also took into account the heat capacity of the building. Infiltration was calculated using an equation of the exponential form:-

$$\dot{m} = \pm C(p - p_x)^n \qquad (3.9)$$

where $\dot{m}$ = mass flow rate

C = constant dependent on crack characteristics

p = outside pressure

p_x = inside pressure

n = exponent, (taken as 2/3)

Wind and stack pressures were also calculated for the simple building shape considered. The digital computer program relied on the widely used technique of balancing the air flowing to and from each room. The limits of accuracy in the program leave something to be desired, in that the air flow was only balanced to ± 10 % for appartments (each of about five rooms) and ± 20 % for staircases. It also appears that the program (as used on an IBM-1620 computer) was rather slow. Mechanical ventilation was also accounted for by taking ventilation appertures as being cracks. The pressure behind each ventilation opening was calculated allowing for the fan pressure, pressure losses in the duct and any natural draught which could occur in the ducts. An eight floor residential block was used as an example with each floor containing five flats. The main variation from an office block lies in the lack of corridors which affected flow to and from the stairwell. The limitations of the program in terms of allowable inputs were a maximum of; 25 ducts; 100 appartments; four staircases and five fans. The magnitude of wind pressures was found to alter the distribution of infiltration losses from floor to floor, though the total infiltration heat loss varied little with wind speed. The program showed the effect of inside-outside temperature difference (stack effect) - the flow on the lower floors was generally towards the stairwell. The overall accuracy of the program cannot properly be judged since it was not compared with a real building, (in any case it was only intended as a design guide), though it does offer advantages over estimates based on simplified guidelines.

The variation in flow caused by stack effect was also noted by Tamura and Wilson[32]. Their mathematical model was based on the main components shown in Figure 3.6. The main aim of the paper was to analyse the distribution of pressure differences due to stack effect (or "chimney action" as it is sometimes called). Most internal partitions (not floors) were omitted from the model. The leakage areas in each floor - to the outside, between floors and to internal shafts - were lumped together for each component as an equivalent orifice area. The flow through such an orifice was given by:-

$$w = CA\rho^{n}(\Delta P)^{n} \tag{3.10}$$

where m = mass flow

C = proportionality constant

A = orifice area

ρ = air density

ΔP = pressure difference across orifice

n = flow exponent

This equation was of the usual form, the flow exponent took values of between 0.5 and 1.0. A series of simultaneous non-linear equations was produced for the air flow - for each floor and for the vertical shaft the mass flow was required to balance. A digital computer was used to solve these equations, also to yield the absolute pressures inside the building and the pressure differentials across components. Mechanical ventilation was allowed for by defining a pressure difference across a ventilation orifice at each floor. In this study wind pressures were neglected in order that the pressures caused by the temperature differential (outside-inside) could be more fully investigated. A temperature of $0^{o}F$ ($-18^{o}C$) was assumed for outside conditions and $75^{o}F$ ($24^{o}C$) for inside the building. The program was run a number of times for a ten storey block in which the flow openings and building conditions were varied.

A number of conclusions were made by the authors from their work. In relation to pressures caused by chimney or stack effect it was found that in tall buildings the main factor was the resistance to flow, from each floor, into vertical shafts (eg, stairwells). Excessive pressure differentials on some floors could be reduced by providing for excess supply or extraction of air by mechanical means. This study showed that the digital computer program was a useful tool for analysis of air flows and pressures. One of the main obstacles to accurate implementation of these types of program was also illustrated, since the leakage characteristics for flow between floors, outside, and shafts must first be measured, or at least accurate estimates must be available. No indications of the limitations of the program were given by the authors.

In contrast to Tamura and Wilson. Rogelein[33] was more interested in ventilation caused by wind only. The building investigated was located in Munich and was of quite large size being 157 m in length, 22 m in width and 70 m in height. He regarded air movement due to stack effect to be of minor importance since the floors of the building offered considerable resistance to vertical air movement (due to a series of air-tight fire doors). Flow through the uniformly glazed external walls was given by the equation:-

$$V = la(\Delta p)^n \tag{3.11}$$

where V = air flow

l = window joint or crack length

a = specific leakage rate per unit length for unit pressure difference

Δp = pressure difference

n = flow exponent - taken as 2/3

The paper attempted to simplify the calculation procedure by making assumptions about the acting pressures and leakage characteristics. The ventilation rate could then be calculated using the ratio between leakages of the windward and leeward faces.

Augmentation to account for internal partitioning was also possible. The application was limited however to certain building types and the lack of stack effect simulation would prove a hinderance to its use in many tall buildings.

Tall office buildings were also the subject of a study by Jackman[26], in this case both wind and stack pressures were taken into account. The flow equation used was of a form of equation (3.11) and like Rogelein, fairly simple building shapes were investigated. Flat roofed buildings ranging over three heights (15, 30 and 60 metres equivalent to 5, 10 and 20 storeys) of long rectangular and square plan shapes were considered. Windows and doors were assumed to be closed so that the leakage occurred through the cracks around these components. The author reported flow exponents ranging from 1/1.37–1/1.72 for the situations considered; an average of 1/1.6 being taken.

Wind pressures were estimated from model studies in a wind tunnel, and variation with height was also taken into account. Stack effect pressures were also simulated for a variety of building layouts, the inside and outside temperatures being taken as 68^{o}F and 32^{o}F respectively (20^{o}C and 0^{o}C). The leakage characteristics were estimated from figures (given by a number of sources) for common building constructions.

A digital computer program was developed from an earlier version which had been used to determine flows and pressures in pipe networks. The program modelled the building as a series of nodes interconnected by branches representing air flow paths, (this being the normal method employed by most digital programs). A solution was produced by making successive corrections to the pressure value at each node (unless a definite pressure had been previously specified for the node, for example the outside), though the actual iteration process was not described. A comparison was made between the results of the program and an electrical analogue, with quite close agreement between the two. The computer simulation offered greater versatility than the other analogue, though accuracies for flow balancing, and program solution times were not mentioned.

Overall results showed that when wind and stack effects acted together, the ensuing airflow was approximately equal to the flow caused by the greater force. The section inthe current CIBS Guide(27), dealing with air infiltration incorporated a number of the results of this study, including nomograms and correction factors, for the prediction of ventilation. Previous methods of ventilation prediction used by the CIBS/IHVE gave higher rates than were indicated by Jackman, and though his method was based on computer predictions rather than measured rates, it was thought to be more accurate since more of the influencing factors could be taken into account.

A definitive mathematical model was developed by the American Society of Heating, Refrigerating and Air-Conditioning Engineers' Task Group on Energy Requirements for Building[34]. The full procedures for determining heating and cooling loads of buildings by computerised calculations, was also provided by the US National Bureau of Standards in their Load Determination Program (the NBSLD)[35].

The flow equation was a slight variation of (3.11)

$$Q = C.A(\Delta P)^n \tag{3.12}$$

where Q = air flow rate

A = flow opening area

N = pressure exponent

ΔP = pressure difference

C = flow coefficient

Values of the flow coefficient and pressure exponent were given for various types of opening commonly found in buildings (Table 3.1 lists these values).

Thus leakage through most types of building component could be accounted for, though measurement of actual leakage characteristics on a building would yield more accurate answers. The calculation sequence was described in a step by step manner which might be translated into a computer program routine. Wind

TABLE 3.1

VALUES OF EQUIVALENT FLOW COEFFICIENTS AND FLOW EXPONENTS FOR COMMON BUILDING OPENINGS (AFTER NBSLD (35))

	C	N
Windows (double glazed/locked)		
Non-weather-stripped loose fit	6	0.66
Non-weather-stripped average fit	2	0.66
Weather-stripped loose fit	2	0.66
Weather-stripped average fit	1	0.66
Window Frames		
Masonry frame, no caulking	1.2	0.66
Masonry frame, with caulking	0.2	0.66
Wooden frame	1.0	0.66
Swing Doors		
½" (12 mm) crack	160	0.5
¼" (6 mm) crack	80	0.5
1/8" (3 mm) crack	40	0.5
Walls		
8" (203 mm) plain brick	1	0.8
8" (203 mm) brick and plaster	0.01	0.8
13" (330 mm) brick	0.8	0.8
13" (330 mm) brick and plaster	0.004	0.7
13" (330 mm) brick, furring, lath and plaster	0.03	0.9

TABLE 3.1 CONTINUED

	C	N
Frame Wall, lath and plaster	0.01	0.55
24" (610 mm) shingles on 1 x 6 boards (14" centres)	9	0.66
16" (406 mm) shingles on 1 x 4 boards (5" centres)	5	0.66
24" (610 mm) shingles on shiplap	3.6	0.7
16" (406 mm) shingles on shiplap	1.2	0.66

NB Values of C are per foot of linear crack for windows and doors and per square foot area for walls. Units of C are cubic feet per minute

and stack pressures were both incorporated with a fairly complex method being used to give wind pressures for each face according to its neighbourhood. Though cases of the occurrence of tall buildings upstream and downstream, and shorter buildings upstream were accounted for, building shape was not. The pressure coefficients for the faces varied with the surroundings and angle of incident of wind.

To solve the air flow and pressure system the problem was once again reduced to balancing the air flow to and from a room or area, by variation of the pressure in each room. A set of simultaneous non-linear equations had to be solved. The actual method of achieving this was not described in the texts and the algorithm was not incorporated as a written subroutine into the "Load Determination" computer program. The algorithm has however been used in a Fortran IV program by Canadian workers at the Division of Building Research, Ottawa. Two programs with slightly different aims were produced by Sander and Tamura[36] and Sander[37]. The first dealt with simulation of air movement, the second with the more restricted aim of calculation of air infiltration.

The set of simultaneous non-linear equations were solved by a process of successive linear approximations. This process (as described by Sander and Tamura) is given below.

The non-linear flow equation (a variation on (3.11))

$$F = K \Delta P^{x} \tag{3.13}$$

where F = flow rate

K = flow coefficient

ΔP = pressure differential

x = flow exponent ($0.5 \leq x \leq 1.0$)

is shown in Figure 3.7. Around the point $(\Delta P_t, F_t)$ this function was approximated by a straight line that was tangent to the curve at this point. The resulting linear function was:-

$$F = K'(\Delta P - \Delta P_i) \qquad (3.14)$$

where $K' = Kx\Delta P_t^{x-1}$

$\Delta P_i = \Delta P_t - {}^{Ft}/_{K'}$

The flow through each leakage path was replaced by this linear approximation and matrix methods were used to solve the resulting set of linear equations. For the iteration process an initial linear approximation was made for each flow and the resulting equations were solved to give floor and shaft pressures. The flows produced by these pressures were calculated and the calculated flow through each flow path or element was compared with the initial approximated flow. If the difference between these was very low the element was said to satisfy the "convergence criterion" (0.01 pounds Air/minute in this case). If the difference in flows exceeded this criterion then the element was re-linearised. The equations were then solved once again. This iteration continued until all the flows satisfied the criterion.

Up to 100 floors and 10 shafts could be handled by this program which was written for an IBM 360 model

67 computer. For a 20 floor building with two shafts, executuion time was about nine seconds. Whilst for a 60 storey building with seven vertical shafts took about 1½ minutes. Perhaps the main limitation of this model was the restriction on floor plan, the only divisions or partitions allowed being those between the floor and the external air, and between the floor and internal shafts.

The digital analogue developed by Bilsborrow[38] improved on this, by allowing for compartments on each floor level. In his study up to 211 compartments (200 rooms, ten corridors and one stairwell) could be accommodated, the rooms being located on ten floors at most, but each floor separately allowing any number of rooms up to 20. The rooms were each assumed to have two flow paths one through the external wall and one into an internal corridor. The corridors on each floor were linked by a common stairwell, though this nominal stairwell might have incorporated more than one real stairwell. The flow equation used was of the form of equation (3.11) and the usual criterion of balancing air flow to and from compartments and corridors was used.

A paper by De Gids[39] attempted to provide a much simplified mathematical model but one which took into account many factors. His aim was to simplify network models of flow to as few nodes as possible. The more normal flow equation, of a form similar to (3.9), was used, with the flow exponent taking values between 1.0

for laminar flow and 0.5 for turbulent flow. For flow through openings in series or in parallel it was suggested that an equation of the form 3.15 could be used.

$$\Phi = Cr(\Delta p)^{1/n_r} \tag{3.15}$$

where Φ = flow

Cr = air flow coefficient

Δp = pressure difference

$1/n_r$ = flow exponent

The subscript, r, referred to the fact that these values were replacements to simulate the effect of series or parallel flow. A method of calculating these replacement values was indicated and it was reported that inaccuracies within the normal range of use might be up to 4 % but would generally be less than 2 %. A limitation of the model was that it could only be applied to few-junction (or node) situations (one or two node models). The author recognised that a more complex situation would require solution by digital computer methods. A comparison with actual measurements from a simple factory building validated the model to a reasonable degree, though its application to more complex situations was not possible.

British Gas researchers have long been interested in the prediction of ventilation and their work has been reported in a number of papers. Nevrala and Etheridge[40] described the necessary data required by their mathematical model. This included wind speed/direction and external temperature; wind pressure distributions (usually obtained from model scale tests) and the building characteristics, size, geometry, location of leakage areas and general background leakages. Background leakages were difficult to specify and in the model only leakage through the external wall was considered. The model consists of a continuity equation coupled with crack flow equations which were to be solved. The crack flow equations were determined empirically from tests and reported by Hopkins and Hansford[16] and Etheridge[41]. The model was also used for the prediction of heat loss by ventilation, as indicated by Alexander et al[42]. Initially the equations describing the flow were solved using an iterative process. In this wind and stack pressure and mechanical ventilation could be taken into account to give a steady-state solution. Ventilation arising from pulsating flows through cracks is given an approximate value and a quasi-steady state result is produced. The study suggested that simple prediction models may not be very accurate for solution of ventilation rates and recommended further work to investigate the slightly unconventional model proposed.

A more complete description of this model was given by Etheridge and Alexander[43]. They stated that the model had originally been developed to provide an as accurate a model as possible of ventilation for a variety of uses. Other models were unsatisfactory. The main improvements seem to have been a better crack flow description and incorporation of background leakage areas.

For general openings (eg, simple air vent) the flow equation was

$$\bar{Q}_i = C_Z A_i \sqrt{\frac{2\overline{\Delta P_i}}{\rho}} \qquad (3.16)$$

where $\bar{Q}_i$ = mean volume flow rate

C_Z = discharge coefficient

A_i = physical open area

ρ = air density

$\overline{\Delta P_i}$ = mean pressure across opening

However for crack flow, the equations developed by Etheridge[41] were used:

$$CA\bar{Q}^2 + \frac{BzL^2}{4\rho}\mu\bar{Q} - \frac{2A^3\overline{\Delta P}}{\rho} = 0 \qquad (3.17)$$

where C, B = empirical constants (from crack geometry)

z = crack depth

L = crack length

μ = air viscosity

This equation was also used for background leakage areas.

The effect of fluctuating wind pressures was given an allowance based on the assumption of a Gaussian distribution of pressure about a mean value. The flow equation was therefore modified, but it was only applicable to crack flow, not to background leakage flows.

The simultaneous flow equations were combined with a simple conservation of mass continuity equation within a Fortran language computer program. Iterative techniques were used for solution, execution time on a CA1 Alpha micro-computer being about one second for an 11 cell house. A comparison was made with tracer gas ventilation studies of a house, and good agreement was found for the range of typical building pressures. Results of later tests in which individual rooms were monitored were not available for complete comparison, though the authors stated that ventilation of upstairs rooms was generally underestimated whilst that of downstairs rooms was overestimated.

The multi-cell model had a number of advantages over a single-cell version. It had a higher degree of accuracy and had a wider range of application since individual rooms were dealt with. It did require much more input data, some of which was difficult to specify and needed results from pressurisation tests. More influencing factors could be taken into account in the multi-cell model, though it was recognised that specification of background leakages was a problem requiring further work, also needed was a more complete description of pulsating flows. The model's inherent accuracy and availability of solution by computer within a short space of time suggested it as a useful predictive tool for air flow in houses, buts its limitations and applicability to large structures were not mentioned.

3.4.2 SUMMARY

The accuracy of digital computer analogues has been limited by the accuracy with which air movement can be described by the mathematical equations. These equations generally show a non-linear relationship between flow rate and pressure difference, thus with numerous flows to and from a space or spaces, a complex set of non-linear simultaneous equations must be created and solved. This is not a simple task and the usual method for solution has been to use some form of iterative technique, modifications to which could be made to produce a reasonably fast convergence. The speed of convergence also depended upon the size and capability of the computer as well as the program language and the capability of the programmer.

The general technique was to balance the air flow to and from each space, compartment or room in the air flow network. This was achieved by varying the absolute pressure in each space. The permitted errors for the balance condition also affected the execution time of the program.

The mathematical representations for air flow varied somewhat from worker to worker and program to program. The most widely used equation, though giving reasonable simulation and accuracy, has been shown to be deficient and an improved, more complex version has been devised by British Gas researchers (amongst others). In addition it has been recognised that the flow rate cannot be based on the steady state pressure difference only, but that fluctuating pressures (due mainly to wind turbulence) affect flow, and work has already been done to investigate this. However even the most advanced digital analogues make only a token allowance for flow due to pressure fluctuations and turbulence, mainly because of the great difficultly in predicting the effect quantitatively.

3.5 FULL SCALE INVESTIGATIONS

In order to gain the most useful information regarding air flows in buildings, experiments are performed at full scale on real structures. The two areas in which most work has been carried out are those using:-

i) pressurization tests and

ii) tracer gas measurement of infiltration rates

At full scale, experiments produce realistic results but these results may have restricted use and often they are gathered under experimental conditions which can be easily affected by prevailing weather, etc.

3.5.1 MEASUREMENT OF PRESSURE

Most of the full scale wind pressure data that is available is not particularly suited to use in infiltration and ventilation work since it is concerned with wind loading effects at high wind speeds. In addition much data has been collected for tall structures where the wind loading element was of greater interest, but such data is not generally applicable to low rise buildings. Some useful work has been carried out by the Building Research Establishment (or Building Research Station as it used to be) where a need had been recognised to compare predicted wind pressures with real measured values.

One of the first studies by Eaton and Mayne[44] dealt with tall buildings (the numbers of which had increased rapidly in the 1960's), though this was not their only sphere of interest. The two buildings considered, both in London, were a rectangular office block 66 m high (x 43 m x 18 m) and the GPO Tower 177 m high. Pressures at high velocities and gust-speeds rather than lower steady state averaged pressures were the main interest.

These two researchers later concentrated on wind pressures for low rise buildings[45]. A special test house was erected on the West side of a housing estate at Aylesbury. Open country was to be found for 15 km to the South-West (the direction of the prevailing wind) and the specific aim was to investigate wind effects below 10 m height. The test house had a variable roof pitch and had pressure transducers mounted at 72 positions; an additional 44 transducers were located on seven other houses in the estate. Wind velocities were measured at three points (at 3, 5 and 10 m) on a fixed mast and at a mobile mast, on a Land Rover, which was variable up to 20 m in height. Analogue recordings of data were made, on a magnetic tape, which was analysed by the Environmental Sciences Research Unit, Cranfield Institute of Technology. Few results were reported in this first study, though one conclusion was drawn; that the hedges in the fields around the site did not appear to have any significant effect. Pressure coefficient data was given for the house but no attempt was made to theorize.

Another study specifically investigated the effects of varying the pitch of the roof[46]. The range of pitch investigated was from 5 to 45^{o}, this being possible due to the design and construction of the test house. The main concern here was to indicate the peak pressure coefficients. The authors stated an intention to compare the gathered data with wind tunnel tests and with the British Standards Institute Code of Practice, though such a comparison does not seem to have been published.

Tamura and Wilson[47] reported an investigation of pressures in three tall buildings (44, 34 and 17 storeys in height). Pressure tappings were taken externally and internally at a number of floors on each building; the main aim being to assess "chimney" or "stack" effect. A variety of outside-inside temperature differentials were considered, and so as to minimise any wind effects, readings were only taken at wind speed below 10 mph. A comparison was made between the readings and computed theoretical pressures both with and without air conditioning and ventilation systems operating. In most cases the actual values followed those predicted with some variations occurring at machinery/service floor levels. Generally external pressure on the lower floors was higher than internal pressure resulting in net inflow, whilst on the upper floors the situation was reversed. The authors recommended pressurization of the ground floor entrance lobbies in order to prevent a major inflow of air due to stack effect which would increase whole building ventilation rates. The variation, caused by ventilation system operation on the pressure differences acting across the exterior walls of the buildings are shown in Figure 3.8.

In some cases it was possible to discover the rate of air change by direct measurement. In such a situation all air flow paths had to be known for either air supply or air loss from a space. A room in a sheltered location which had only a mechanical air supply or extract system would be such a case. The rate of air supplied or extracted with a resultant significant pressurization or depressurization, would give a fairly accurate measure of the air flow through the space. Hitchen and Wilson[1] described a wide variety of anemometers suitable for the measurement of air flow. In older houses which had chimney flues, the general flow pattern consisted of air being drawn up the chimney and out of the house with a corresponding influx of air through open doors and windows and gaps around structural elements. From these noted facts has been built up a useful, and one of the major, investigative tools for air flow measurements - pressurization or "blower" tests.

3.5.2 PRESSURIZATION TECHNIQUES

The method is based upon the act of supplying or extracting measured air flows to or from rooms or whole buildings. The flows will be higher than naturally caused ones and the effect on resulting pressure differentials can be measured. By using a number of flow rates a relationship between flow and pressure difference can usually be found which is characteristic of the room or building and its air flow paths.

Some research work has concerned only sections of buildings, eg, Thomas and Dick[14], and Hopkins and Hansford[16] who dealt with leakage around windows. British Standard 4315[48] gave a method for the testing of resistance to air penetration of a window. This involved the pressurization of a window at 5 mm water gauge intervals. The form of the expected relationship between pressure and flow is shown in Figure 3.9. This is based on the relation given in the CIBS Guide[27].

$$Q = C\,A(\Delta p)^n \qquad (3.18)$$

where Q = flow

C = constant

A = area (usually constant)

Δp = pressure difference

n = flow exponent - found from the shape of the graph

Many pressurisation tests were carried out by sealing a fan into a window or doorway in order to pressurize the space and, in order to reduce errors, pressures much greater than would be caused naturally, were used.

Alexander et al[49] indicated three contributions to the leakage area, these were:-

i) purpose provided openings such as air vents and open windows

ii) component openings including cracks around windows, and

iii) background leakage areas

These last two could best be investigated using pressurization techniques. Hunt[50] showed how the leakages through various parts of a building could be determined using pressurization and selective sealing of components.

Shaw et al[51] devised a method to determine air leakage areas in the walls of tall buildings. In this case the buildings investigated were pressurized using their own mechanical ventilation plants supplying outside air. A mathematical model was proposed to calculate the leakage rate for a wall area given the overall leakage figures for each building. The model was compared with a computer simulation for validation purposes since no detailed full-scale data was available. Good agreement was found with the mathematical model calculations of leakage areas (for a specified building) agreeing closely with intitial values input to the computer simulation ($\pm$ 10 %). This validation however is only as good as the computer simulation. The model was used to determine leakage areas for four real buildings and the results indicated substantial air leakage - worse than might have been predicted given the construction.

Tamura and Shaw[52] continued this investigation of exterior wall air tightness of tall buildings and extended the study to a further four buildings. To minimize the chance of unpredictable effects, the tests were carried out during unoccupied periods and times when there was little or no wind. The buildings were pressurized using their own supply systems with extracts shut down. By using the same mathematical model as in Ref (51) results from these tests were used to produce a relationship between leakage rate and pressure difference. This pressurization relationship is shown in Figure 3.10. As can be seen the results indicated the leakages for all walls to be higher than the recommended standard.

If the leakage of the exterior wall could be determined at a given pressure difference, as carried out in these tests, then infiltration under normal situations might be calculated. In order to do this the pressures acting upon the exterior walls had to be found by taking into account both wind and stack effects and their distribution over the building facades. In paper by Tamura and Shaw a description of the determination of stack pressures and flows was given along with associated infiltration heat losses.

It is not always possible to perform pressurization tests on whole buildings and for this reason Shaw[53] developed methods for conducting smaller scale tests. These methods involved the use of a portable fan to produce positive or negative pressure for a small test chamber which was sealed around a test area such as an exterior room wall or window. Such tests are desirable in their own right in order to compare, in situ, real building leakage values with those found in artificial laboratory tests. (The ASHRAE Guide Book of Fundamentals[15] gave a wide range of leakage data gathered from laboratory tests.) The small scale pressure tests were most suited to office blocks, often tall buildings in which the exterior wall in each office or room was the same or similar. In order to check the validity of the method, in which the small scale test was used to predict leakage values for whole buildings, a full leakage test was carried out on one building. In order to offset the effect of leakage from the test chamber (shown in Figure 3.11(a)) to adjacent rooms (rather than to the outside) these rooms, at each side and above and below, were also pressurized. (Figure 3.11(b)). The results of tests using this pressure balancing technique showed very good agreement with the whole building leakage tests, whilst without it a significant error was introduced. Leakage to adjacent rooms was likely to occur if heating pipes or other ducts connected rooms through walls. Shaw's test results indicated that three major leakage sources existed, floor-wall joints, windows and window sills; whilst no measureable leakage occurred through ceiling joints.

British Gas have developed a system for measuring whole house leakage characteristics which was described by Alexander et al in their paper on experimental techniques(49). They pointed out some of the drawbacks of whole house experiments in that results of such tests gave no information about the distribution of leakage paths. Prevailing weather conditions could also affect the usefulness of the tests and the majority of workers in this field note temperatures inside and out and perform tests only on calm days (unless the effect of wind itself is to be investigated). Alexander et al also reported variations in leakage with time and found that leakage increased by 70 %, on average, during the first two years of occupancy. This indicated that many leakage measurements which were made on new housing before it was occupied may have little relevance to the real life situation.

Nylund[54] described a method of tightness testing known as "reciprocity". The main advantage of this technique lied in that it eliminated leakage paths into adjacent rooms, to enable the flow through the external wall to be easily calculated. The disadvantage was that all rooms into which leakage may occur from the room under test, had also to be pressurized and flow measured; though only one flow measuring position is required. This disadvantage must limit this type of test to cases where either only one room is being investigated or where many rooms are identical as in an office block.

De Gids[55] also noted that pressurization tests had a number of limitations. In most tests the houses, room or element being investigated would be pressurized or depressurized to a much higher level than would normally be found under natural conditions. Typical pressure tests used levels as high as 50 Pa whilst real life pressures were usually of the order of 5 - 10 Pa. The reason for using such a high pressure was to swamp the naturally occurring pressures (caused by wind and stack action) so that the test would not be strongly dependent on prevailing conditions and thus could be readily repeated. In many cases the tests would still have to be restricted to relatively calm days because of the influence of wind.

The distribution of pressure differences under test conditions is almost homogeneous if internal doors, etc, are left open. However this position would not be found naturally since positive pressures are set up on windward facades with negative pressures on leeward ones. In many buildings stack affect due to inside-outside temperature differences would cause an irregular pressure distribution too.

Pressurization tests gave whole house or room leakage figures, but unless more elaborate tests were performed, there was no information as to the spacial distribution of leakage paths, especially background leakages.

By using higher than normal pressures the flow may have been altered. Flow through small cracks could be laminar at normal pressures but might be changed to turbulent by higher pressures and flow rates. The size and formation of leakage areas may be dynamically altered by such pressures too. The test itself usually involves supplying or extracting air via an existing door or window which can then not be allowed any part in the leakage paths.

Some of the problems mentioned above have been eliminated by a technique developed by Sherman, Grimsrud and Sonderegger[56]. In this case only low pressures were produced but in an alternating pressure and suction technique. Great accuracy has been claimed for this technique since weather induced effects at low pressures were removed by the averaging effect of the method.

Card, Sallman, Graham and Drucker[57] reported another method which used alternating pressures. A schematic diagram of their system is shown in Figure 3.12. This "infrasonic" method was based on the action of pumping a 55 gallon oil drum so as to sequentially compress and rarefy the air in the drum. The volume of the room was effectively being altered by the pumping of the drum. The volume change followed a sinusodial pattern, frequencies between 0.5 and 5 Hertz were used and the resulting pressure fluctuations were measured using a very sensitive transducer employing fibre optics. Under ideal conditions the leakage properties of the structure could be obtained from the frequency response curve, though results indicated some shortcomings in the technique.

3.5.3 TRACER GAS TECHNIQUES

Tracer gases are now widely used to determine infiltration/ventilation rates in buildings. Their main advantage over pressurization techniques lies in the fact that measurements can be made under relatively natural conditions.

The main principle behind the operation of tracer gas techniques is that if a known amount of a gas, with some easily measured property, is liberated in a space then measurement of its concentration can be used to determine the infiltration (or exfiltration) of air to or from that space. A number of reviews of the techniques used, have been produced - notably Hitchen and Wilson[1] and Sherman, Grimsrud, Condon and Smith[58].

All tracer gases must obey a continuity equation, though in some cases (such as with carbon dioxide in occupied rooms) all sources must be carefully identified. Usually if an amount of gas is liberated in a space, its rate of change is determined by the rate of injection and its rate of loss.

$$\frac{dT_S}{dt} = T_I - \frac{dT_E}{dt} \tag{3.19}$$

where t = time (hours)

T_I = injected flow of tracer (m^3/hour)

T_S = volume of tracer in space (m^3)

T_E = volume of tracer lost due to infiltration/exfiltration (m^3)

The average concentration of tracer gas is given by the ratio of the tracer gas volume to the volume of the space

$$\overline{[T_s]} = \frac{T_s}{V} \tag{3.20}$$

where $\overline{[T_s]}$ = average concentration (m^3/m^3)

V = volume of space (m^3)

If the concentration of the tracer gas being used is negligible in outside air it can be shown that

$$\frac{V\, d[T_s]}{dt} + Q\, [T_s] = T_I \tag{3.21}$$

where Q = infiltration (m^3/hour)

The infiltration rate (or air changes per hour) is given by

$$N = \frac{Q}{V} \tag{3.22}$$

Thus if the injection of tracer gas can be measured together with its changing concentration in a given space, then the infiltration rate can be calculated.

The validities of the equations are limited by the accuracy with which the concentration can be measured. In most experimental situations a wide and even dispersion of the tracer gas is attempted and measurements of concentration are taken at a number of points. However the inadequacies cannot be overcome completely and Sherman et al[58] categorised these "mixing problems". Air entering the space may not mix evenly with air already present, causing the concentration of the tracer gas to vary from point to point. This is particularly true of a house with rooms which have varying leakage characteristics (in such a case a multi-chamber analysis may be required). The injected tracer gas may not mix immediately with the air in the space and this will give rise to a mixing time requirement. There is also a delay time to be taken into account since in general injection and monitoring points will not be spacially coincident. The definition of the volume of the space is also questionable since in most cases there will be areas which play no part in infiltration process. These might be cupboards, corners and alcoves in which air does not mix with the rest of the room. Stratification of air into layers may also mean that air near the ceiling does not exchange well with the main body of the space. This leads to the definition of an "effective volume" for the space which may not be the same as its physical volume. In many experiments the assumption that the effective volume is equal to the physical volume is made.

Different measurement techniques have been developed which in some ways eliminate parts of the problems mentioned above.

The tracer gas decay technique is the most widely used of all methods. An initial amount of the gas is injected into the space and mixed with the air, before readings start, often by use of a small fan. The concentration of the gas is then recorded as it decreases due to infiltration of fresh air. The continuity equation takes the form

$$[T]_t = [T]_o e^{-Nt} \tag{3.23}$$

where $[T]_t$ = concentration at time t

$[T]_o$ = concentration at t = 0

N = infiltration rate

(assuming no extra generation of the tracer within the space and that the concentration of the tracer in the outside air is negligible). If the concentration of the tracer gas and its rate of decay is known together with the volume the space, then the infiltration may be found from the relationship.

$$\frac{d\,[T]_t}{dt} = \frac{Q}{V}\,[T]_t = -N\,[T]_t \tag{3.24}$$

The decay technique is relatively simple to perform and can be used repeatedly for short term measurements at a number of positions. The infiltration rate calculated from a decay test will be in error if the effective volume is different from the physical volume. The ratio of these volumes yields the error, which may be as high as 50 % according to Sherman et al. The technique is unsuitable for long term measurements because of the length of the test (upto a few hours) and the interval required between tests, time to set up, etc. When the sampling interval is short, errors in concentration measurement can lead to erroneous calculations. This is shown graphically by Hunt[50] in Figure 3.13 as an infiltration error due to measurement errors causing variations in the quantity $\left[\log_e [T]_t/[T]_{t-1}\right]$ of $\pm$ 2, 5 and 10 %.

From equation (3.21) it can be seen that if there was no change in the concentration of the tracer gas in the space, then the term which involves the room volume would be reduced to zero. Thus if concentration could be kept constant the flow would be given by

$$Q = T_I/[T]_s \tag{3.25}$$

and there would be no error due to incorrected assessment of the effective volume. Such a technique proves to be less practical in reality because of time lags inherent in the system; if the concentration was found to be falling and injection rate of the tracer was increased it would be some time before the change was measured. If the process was automated with a feedback loop to increase or decrease injection rate according to concentration, then there would be a real possibility, due to the response times involved, that an unstable oscillation would be set up. If the "update" time were long enough the instability could be removed but this would inevitably mean that a constant concentration could not be maintained.

A variation of this last theme would be to use a constant flow of tracer gas to the space. Although this method would require a calculation of the effective volume it does extend the time scale of the experiment. The infiltration would be given by

$$Q = \frac{T_I}{[T]_s} - \frac{V}{[T]_s}\frac{d[T]_s}{dt} \qquad (3.26)$$

If the rate of injection of the tracer approximates to its rate of loss then the rate of change of concentration will be small, which in turn places less importance on the need for effective volume to be properly calculated. The constant feed or flow technique is best suited to applications where there is a high rate of internal mixing and relatively homogeneous concentrations exist. For non-homogeneous situations the technique can be used to provide a "transfer-index" of air movement between certain parts of a building. The use of this technique has been pointed out by Hitchen and Wilson[1] and Hunt[50]. It was used not to produce infiltration rates but to assess the air movement between two specific points and it may be used to determine flow patterns and also spread of contaminants in medical applications.

The use of micro or mini-computers to control tracer gas systems now allows more accurate measurements over longer periods. The tracer gas is liberated continuously and its concentration monitored and recorded by the computer. Adjustments to the injection rate are made on the basis of measured concentrations. The effect of this computer control is that a series of constant flow tests, made sequentially, are simulated. The measured flows and concentrations from previous tests are used to modify the set up of the next test.

In the infiltration rates varies from room to room or space to space in an area under investigation and there is insufficient mixing between the rooms, then a multi-chamber analysis is required. Such analses were made by Sherman et al(58) and Sinden(59). Both these studies presented some of the necessary mathematics to deal with a multi-chamber situation. The continuity equation was that for a single room, but inthis case transfers to and from all other connected rooms, as well as the outside, were taken into account. In these analyses matrices were defined to represent the flows between each chamber, and the volumes of the chambers. The continuity equation in matrix notation appeared as

$$\bar{\bar{V}} \frac{d[\bar{T}]_s}{dt} + \bar{\bar{Q}} \; [\bar{T}]_s = \bar{\bar{T}}_I \tag{3.27}$$

where $\bar{\bar{V}}$ = represents a two dimension matrix of the chamber volumes

$[T]_s$ = represents a one dimension matrix of the concentration in each space

$\bar{\bar{Q}}$ = represents a two dimension matrix of the flow rates between chambers

$\bar{\bar{T}}_I$ = represents a one dimensional matrix of the rate of injection of the tracer into each space

The same measurement techniques may be employed with a multi-chamber system as with a single chamber/ outside system. The main problem with a more complex system however was in the number of measurements that must be made, and this is reflected in the lack of field data. If there are N chambers, N^2 independent data points are required to obtain a solution. Either the time of the test can be extended and measurements made in each chamber or a number of tracer gases can be used. For N chambers, N separate tracer gases would be required and each tracer gas would need to be measured in each of the chambers to provide sufficient data. This option would be very expensive, therefore multi-chamber tests are feasible in very few cases at present.

3.5.4 CHOICE OF TRACER GASES AND VAPOURS

A number of tracer gases and vapours have been used in infiltration and ventilation tests. There are a certain desirable criteria for a tracer substance, the choice of tracer will depend upon the relative weight given to each of these criteria since no one substance can satisfy them all. A list of the main considerations is given below.

1) The tracer should be detectable at low concentrations whilst yeilding accurate results over a wide range of concentrations.

2) The tracer should not be absorbed or adsorbed by walls, furnishings or other contents of the space under investigation.

3) The tracer should have a high chemical stability and should not decompose or react with the air or room contents.

4) The tracer should not be inflammable or explosive at the concentrations likely to be produced.

5) The tracer should have no adverse health effects at the levels used (odourless and colourless too, if possible).

6) The density of the tracer and its rate of diffusion should be similar to that of air.

7) The tracer should be a substance not normally found in air or in the area under tests.

8) The measurement techniques should be particular to the tracer being used.

9) The tracer substance and the analytical equipment used should be inexpensive and readily available. If the measurement procedure can be automated this is an advantage.

The use of various gases and vapours for the investigation of infiltration and ventilation has long been considered. Dufton and Marley[60] described an experiment using water vapour as the gas. Unfortunately the method was not very successful due to the number of variables and influences not taken into account. One of the main problems was caused by the absorption of water vapour by items such as furniture.

Another study by Marley[61] gave the method of ventilation measurement recommended by the Building Research Board. It consisted of measuring the increase in carbon dioxide concentration in an unoccupied room, caused by burning three "standard candles" for three hours. A 2.5 litre air sample would be taken using rubber hand bellows and the carbon dioxide estimated by absorption in barium hydroxide and titration with oxalic acid.

The same study however gave a better method of measurement using hydrogen as a tracer gas. This was achieved by taking a measure of the thermal conductivity of the air and since hydrogen has a thermal conductivity seven times that of normal air, its concentration could be gauged. The instrument used to carry out the measurement was a "Katharometer", devised by Shakespear initially to determine carbon dioxide levels in flue gases.

A diagram of the basic instrument is shown in Figure 3.14 R_1 and R_2 are spirals of platinum wire with identical resistances, placed in separate cells A and B. Both A and B are in the same solid copper block to ensure temperatures are equivalent. Two further identical resistances R_3 and R_4 are used to make up a Wheatstone Bridge. A prescribed current is allowed to flow through the circuit heating the resistance spirals R_1 and R_2. These spirals lose heat to the surrounding gas. A standard gas (usually air) is found in cell A whilst room air is allowed to diffuse into cell B. Any differences in thermal conductivity of the air samples will affect the relative temperature of the resistance spirals and thus their resistance. A difference in resistance between R_1 and R_2 will unbalance the bridge circuit and cause current to flow through the galvanometer G. A deflection on the galvanometer scale gives a measure of the hydrogen concentration, a one inch deflection was equivalent to 0.1 % change in concentration.

Since hydrogen is considerably lighter than air a mixing fan was used to prevent stratification and the initial injection of hydrogen was made into the stream of the fan. If people were present in the room under test, they would cause variations in carbon dioxide and water content which would affect the katharometer. reading. To minimize this effect the number of occupants should be kept constant.

This method of hydrogen decay was later used by Dick[62.63], in his studies of ventilation of unoccupied and occupied houses in the late 1940's. He found that decay rates measured in different parts of rectangular rooms did not vary greatly, and a centrally positioned katharometer was considered sufficient. For the hall/ stairs/landing and other irregular areas, a fan was used to promote mixing.

Bedford[64] described a method of determining ventilation by analysis of carbon dioxide concentration. The concentration together with an assumed rate of production by the occupants (0.6 ft^3 per person per hour) can be used to give the ventilation rate. Considerable errors can be found using this method since the assumed rate of CO_2 production may vary from the above figure. Additionally at high ventilation rates and low CO_2 production rates, a small change or error in concentration measurement gives rise to a large change in calculated ventilation rate.

A much later study by Penman[65] also investigated the use of carbon dioxide to measure ventilation. He performed tests on two large stack rooms of a library with the fresh air supplied containing a known concentration of CO_2. The concentration of CO_2 in the extract duct and the number of people in the library were monitored. Good agreement was found between the calculated air change rate and the measured supply rate (four air changes per hour calculated versus 4.2

air changes per hour measured). Problems of estimating rate of production of CO_2 due to metabolic activity still exist and it does not appear that carbon dioxide measurements will be useful for ventilation rate prediction except in specific cases where the assumption can be verified.

The use of ammonia as a tracer gas was proposed by Noronha(66) as suitable for use in factories. The concentration required would be well below threshold levels but doubt is cast on the haphazard technique of dispersal which would not seem to give very uniform levels of concentration. The method of measurement consisted of passing air through a solution of sulphuric acid for later titration. This would seem another source of possible error.

One of the tracer gases widely used is Nitrous oxide (N_2O). Lidwell(67) and Howard(68) performed comparisons of N_2O with other tracer substances. A single measurement by Lidwell gave excellent agreement between infiltration rates measured using N_2O and Acetone (C_3H_6O). Howard was mainly concerned with comparing hydrogen with nitrous oxide, though he also used oxygen as a tracer. The main discovery was that hydrogen diffused through the gypsum walls of the test area thus giving an apparently larger ventilation rate than that observed with nitrous oxide. However non-porous walls (eg, concrete) or painted gypsum yeild the same rates with both H_2 and N_2O. The concentration of hydrogen was measured by katharometer whilst a gas-analyser was used for the nitrous oxide. (The gas-analyser works on the basis of infra-red absorption spectroscopy.) Nitrous oxide is soluble in water but this has not prevented its use in many experiments. The British Gas "Autovent" System used nitrous oxide as its main tracer substance in an automated constant concentration technique.

Hartmann and Muhleback[69] devised an automated system for measuring air change rates using the decay of N_2O tracer gas. Up to six interconnected spaces could be monitored at the same time. Proportional volumes of the gas were injected into each space and mixed for 15 minutes using fans. The scanning took place sequentially through each space in turn, with each sample taking five to ten minutes. Initial tracer gas concentrations were set to be in the range 10 - 20 ppm. The decay was monitored over a long period and the data recorded on an analogue trace and also by a data logger on tape.

The relatively low equipment and running costs combined with its accuracy, make the use of N_2O tracer gas coupled with an infra-red gas analyser one of the most widely used techniques for infiltration research.

Another possible tracer substance is one which emits a measurable amount of radioactivity. The best type of radioactive tracer gas is one in which particles are produced. Collins and Smith[70] used the radioactive isotope of Argon, A^{41}. It could be recommended in buildings where CO_2 or H_2 techniques would be difficult to apply. There were some problems relating to the radio-active emission by Argon and for this reason the isotope Kr^{85} might be used as an alternative. Howland, Kimber and Littlejohn(71) estimated air movement and infiltration in a house using Krypton tracer. They encountered some general mixing problems but found that the response time of the system was rapid. The results were not compared with the results from other methods however. The use of radioactive tracers has its own problems. The general background level of radioactivity should be taken into account and due to the nature of the random decay of the substance, there is a margin of error in all observations which cannot be totally eliminated. Very low concentrations can be used but to offset the problems mentioned either concentrations greater than are actually necessary have to be used, or sampling times increased. Of course the normal precautions applying to radioactive substances must also be taken into consideration, but at concentrations as low as 1×10^{-9} % by volume these are not too great. The cost of equipment and gas is a major prohibitive influence on the use of radioactive tracers in the current studies.

Foord and Lidwell(72) were dissatisfied with the performance of N_2O and radioactive tracers, and thus turned to halocarbons as a source of tracers which would have a large measurable range but be detectable in very low concentrations. The tracer substances used were separated by gas chromatography in a column and then analysed for concentration by an electron capture detector. The technique allows more than one gas to be used at the same time, and three were used in the experiments - Freon 12 (CCl_2F_2), Freon 114 ($CClF_2CClF_2$) and BCF ($C\ Br\ ClF_2$). Concentrations as low as 1 part in 10^{10} parts air could be detected and a range in concentrations of over 1000 to 1 was possible. A number of problems were encountered, the greatest being contamination of the analyser. The tracer gases and method of analysis would be limited in application because of the complexity of operation and problems that might be encountered. They may prove useful, in areas where N_2O or other tracers are not suitable.

Sulphur Hexafluoride (SF_6) has become increasingly popular as a tracer gas, it too is detected by using electron capture, and gas chromatography techniques. It fulfils most of the requirements for a tracer gas and can be detected at very low concentrations due to the electron capture of its six fluorine atoms. Kumar, Ireson and Orr(73) described an automated infiltration measuring system using Sulphur Hexafluoride as the tracer gas. Parts per billion concentration in air could be measured and the system could employ constant concentration and decay methods. Sampling intervals of between one and 15 minutes were used in the decay technique and use of the decay technique meant that absolute concentration calibration of the measuring instrument was not required, only relative concentration. Of course problems mentioned earlier concerning decay methods still apply, particularly in that short term variations in infiltration rate cannot easily be detected and that the method does not lend itself to proper automation. A constant concentration technique was also used by Kumar et al, in a system under the control of a programmable desktop calculator. Careful control must be exercised in such a technique and instruments need to be accurately and frequently calibrated. In this method the gas was discharged into the air by a selenoid valve system in which a measured quantity of gas was held between two valves. The frequency of discharge could be as high as once per 0.9 seconds, (in the system this rate was controlled by the calculator). The test results for this sytem, from laboratory and field work, showed an average error of only 1.5 % and the system could be left unattended for up to six hours.

The automatic system used by Grot, Hunt and Harrje[74] was more ambitious. This was designed for use in large buildings. Sulphur Hexa-fluoride was again used, the main deciding factor being the amount of gas that would be required. The Collins Publishing building, near Glasgow, was to be investigated and since it had a volume of approximately 2.4 million cubic feet ($\approx$ 68000 m^3) it was more sensible to use a gas detectable at lower concentrations in order to reduce tracer gas requirements. (By comparison similar sized cylinders of Sulphur Hexafluoride and Nitrous oxide hold sufficient gas for 10,000 and 10 "seedings" respectively, ie, an advantage of 1000:1 in favour of SF_6). The whole system was controlled by a micro-computer which operated the five gas injection units and the ten port sampling device. Supply and extract fan operation and door/window openings were monitored together with interior and exterior temperatures, wind speed and direction, and pressure differentials across the building. By monitoring the gas concentrations and the various external parameters and fan operations, the computer could calculate the required injection of tracer gas to maintain concentration of fairly constant levels. Incorporation of the various pieces of data, together with infiltration and ventilation rates, given by the tracer gas monitoring, should lead to the determination of the relative importance and influence of prevailing weather conditions, fan operation, etc, on leakage and ventilation rates.

Grimsrud et al [75] made a comparative study of three tracer gases in a test house in California. Sulphur hexafluoride was measured using an electron capture detector, nitrous oxide and methane were measured using infra-red gas analysers. It was found that sulphur hexafluoride gave a slightly higher rate of air change than the two lighter gases. The calculations showed that this difference was + 10 %

($\pm$ 10 %). The authors of this report suggested that the difference could not be attributed to the difference in density of the gases since any stratification or settling out would only be appreciable (in an initially well-mixed atmosphere) after three hours or more. Thus in forced ventilation tests carried out, with approximately six air changes per hour, the difference in measured rates could not be attributed to any buoyancy effects. The different rates may have been caused by inherent differences in the two measurement systems, but in any case the authors pointed out that given the normal variability between infiltration measurements, that this difference would not be particularly significant, even if it is a real, rather than a comparitive or instrument, error.

A study of recent research work reveals that nitrous oxide (N_2O) and sulphur hexafluoride are the two most popular choices for tracer gas studies. For a test in a given volume, less SF_6 is required since it can be detected and measured accurately at lower concentrations. However the gas itself is more expensive and the measuring equipment more complex requiring a finer degree of calibration in use. The choice of gas for a test must inevitably depend on the individual situation and the cost, finance and equipment available.

In all tracer tests care must be taken to eliminate sources of possible error. This particularly applies to gas leaks and instrument malfunctions. Since the tests must be carried out under prevailing weather conditions care must be taken in comparing the results of tests made at different times. This is both an advantage and a disadvantage since the tests can be used to show the influence of certain actions or conditions upon the air change rate but repeatability for comparison is not always possible.

3.5.5 CORRELATIONS OF PRESSURIZATION AND TRACER GAS TESTS

Pressurization tests on buildings and rooms are relatively simple to carry out, repeatable and little affected by prevailing weather conditions. This is in stark constrast to the complexity and difficulties of carrying out tracer gas measures of infiltration. The disadvantage in using pressurization test results for infiltration prediction, is that unreal and artificial pressures have to be used in most cases. There have been some studies, in which some correlation of pressure test data with infiltration rates has been sought in order to simplify a test/investigative procedure for prediction purposes.

Sherman and Grimsrud[76] with Diamond[77] have attempted such correlations. Their methods include the use of full scale pressure test data together with weather data in mathematical model of the building concerned. The surface pressures due to wind and stack action are predicted from the weather data and this can be used with a determined leakage function and flow path distribution to calculate infiltration. A similar method has also been proposed by Warren and Webb[78] but their correlation model is perhaps too simplified to give accurate predictions. So far most correlations seem to show some degree of success, with authors suggesting improvements in their models to take account of more influencing factors. A definitive model for such correlations does not yet exist though current ones may be used to obtain infiltration estimates using weather data and pressure test results.

3.6 SUMMARY

A wide range of techniques have been employed to investigate air movement both inside and around buildings as indicated by the preceeding sections. As in many applied sciences the studies have been concerned with three main areas:

i) Mathematical modelling

ii) Reduced scale physical and analogue modelling

iii) Full scale testing

The main purpose of the review was to consider methods that have been used for investigation of air movement in order that suitable choices might be made in this study.

In deciding on the techniques to be employed it is necessary to take account of the limitations imposed by both equipment availability and the building in question. Certain types of investigation prove impractical in certain situations. In some investigations a large amount of data is required prior to the main work - this can sometimes be very difficult to acquire. Full scale work is limited in complex and large buildings, yet reduced scale model tests also have limitations. In the case of analogue studies, the validity of the model analogy may also have to be proved.

Most studies have been concerned with domestic housing or large office blocks; this being due to a greater interest in buildings with a large number of similar types to which findings might apply. It would appear that Industrial buildings have been ignored in general, or dealt with in a superficial fashion. However such buildings do exhibit some repetitive characteristics which might be analysed.

Since the movement of air is a very complex process, inevitably simplifications are made; for instance it is usually the bulk, overall effect which is investigated rather than the individual flow patterns. In some cases the whole building is treated as one single unit which means that the often significant effects of internal layout and local variations are ignored.

In relation to the investigation to be carried out in this study, little relevant previous work has been performed. Industrial air movement information usually relates to large open spaced structures or to specific air handling units, duct inlets or outlets (see for instance Baturin[79]). Therefore the results of work carried out could provide a useful addition to the knowledge of the air movement in an industrial environment.

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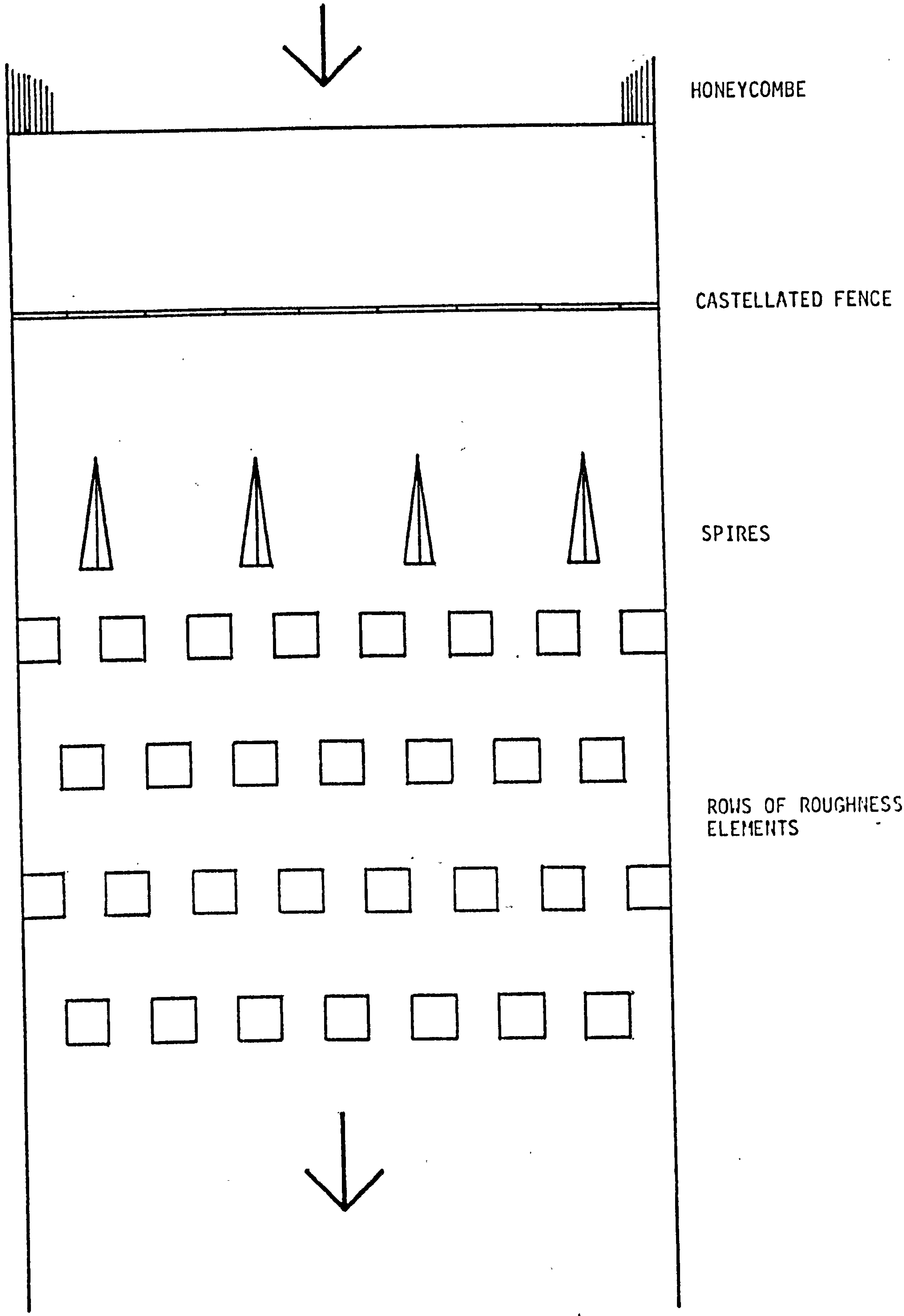


FIGURE 3.1 Typical Arrangement for the Simulation of the Boundary Layer in a Wind Tunnel (After Lee [6])

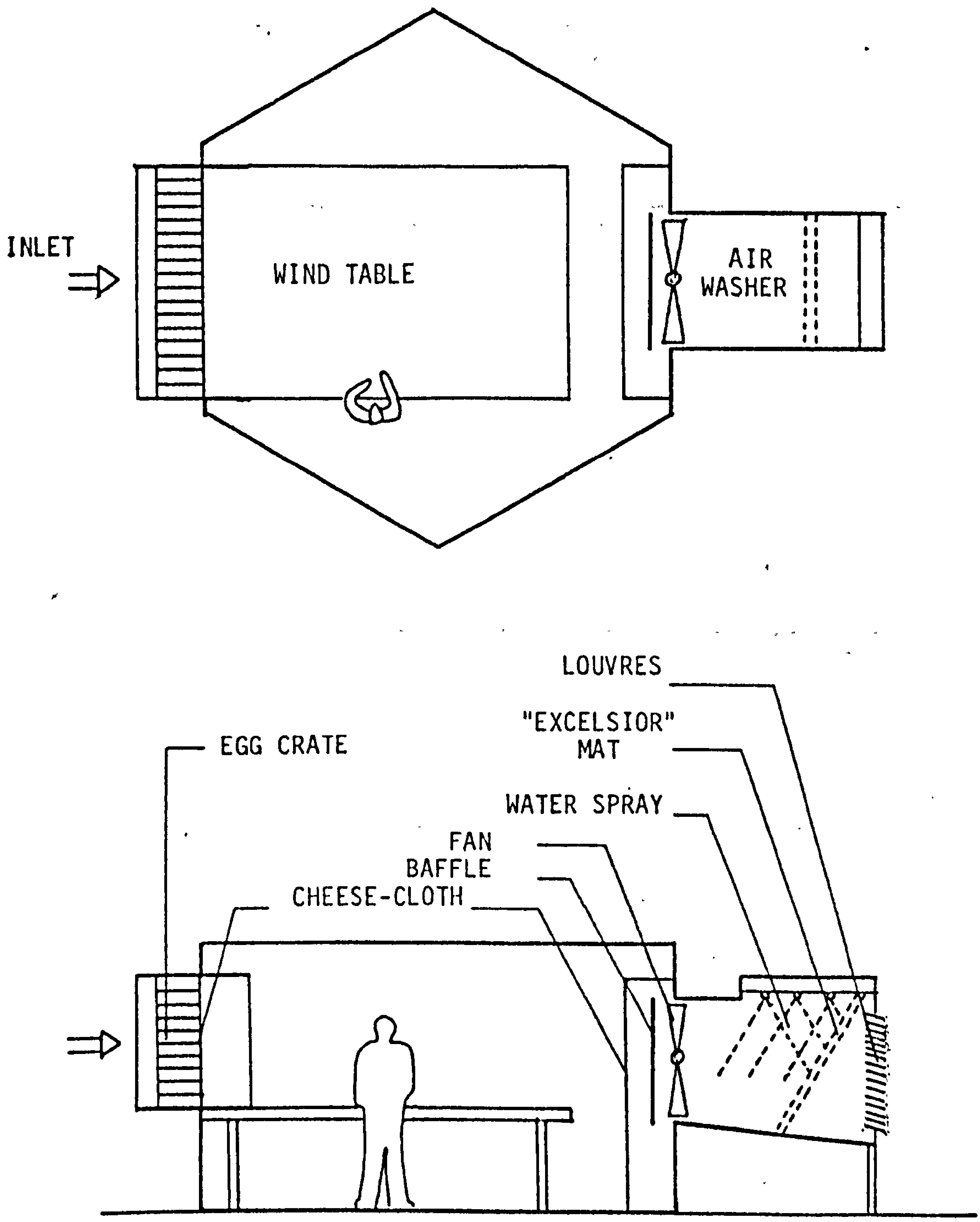
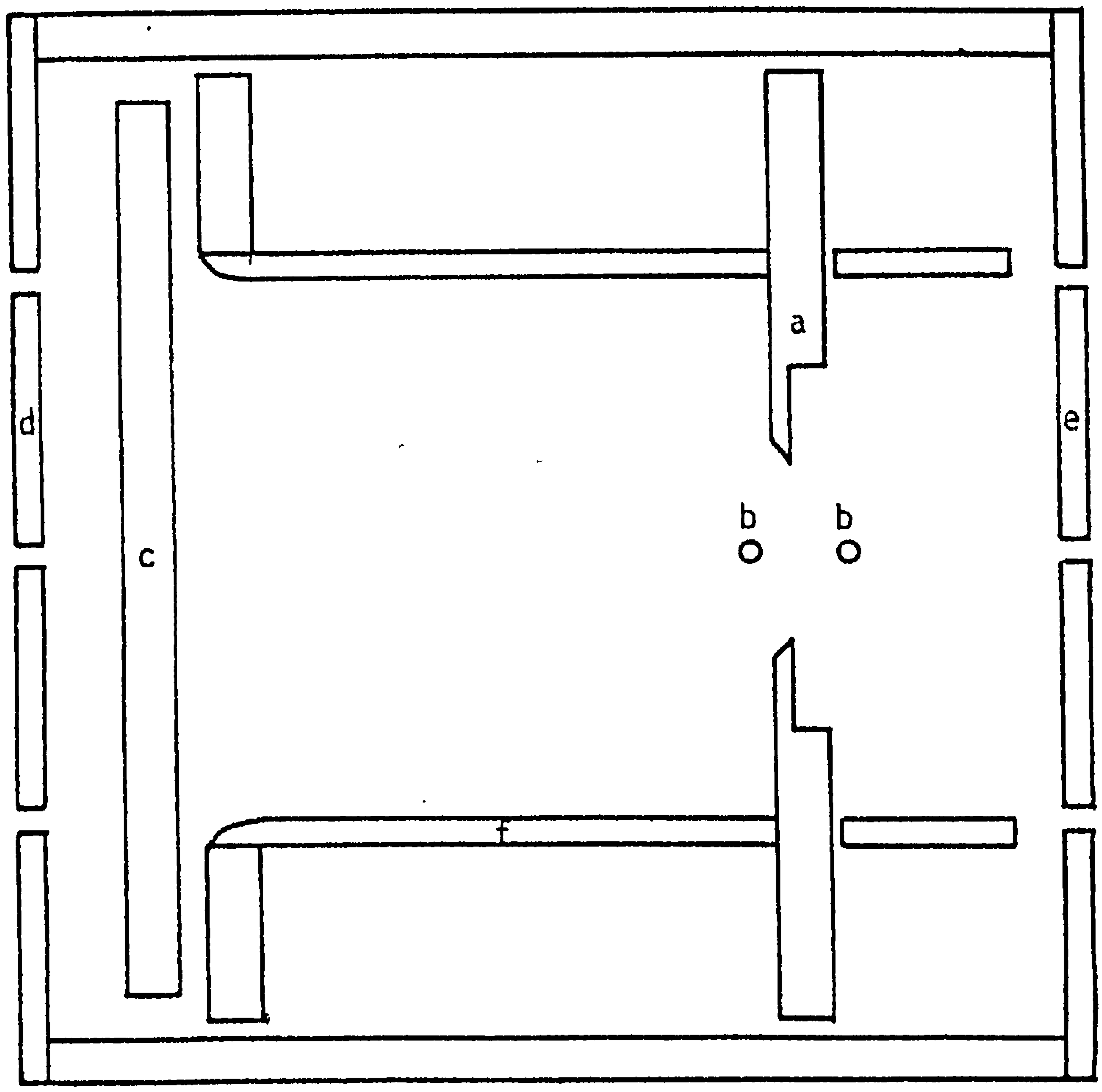


FIGURE 3.2 Air Flow Chamber (After Smith (10))



a. ORIFICE PLATE
b. PRESSURE TAPPING
c. FLOW SPREADER PLATE
d. MODEL FRONT PLATE
e. MODEL BACK PLATE
f. GUIDE PIPELINE

FIGURE 3.3 Section through Orifice Plate Model (After Bilsborrow and Fricke [12])

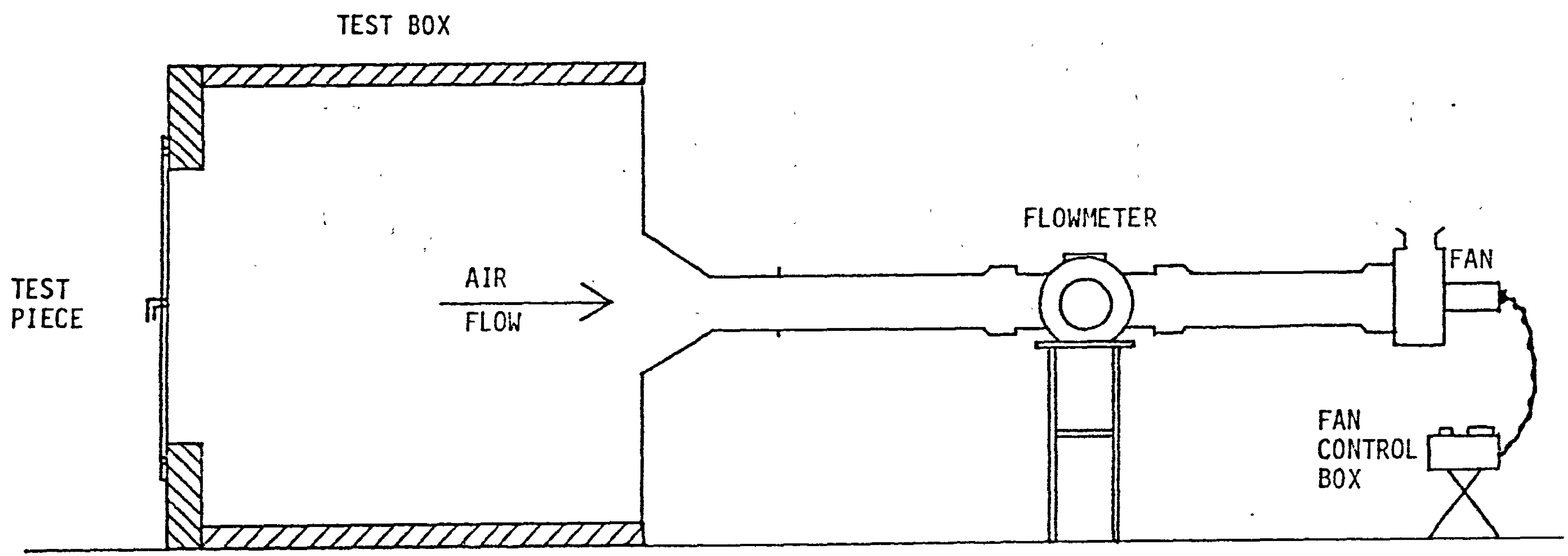
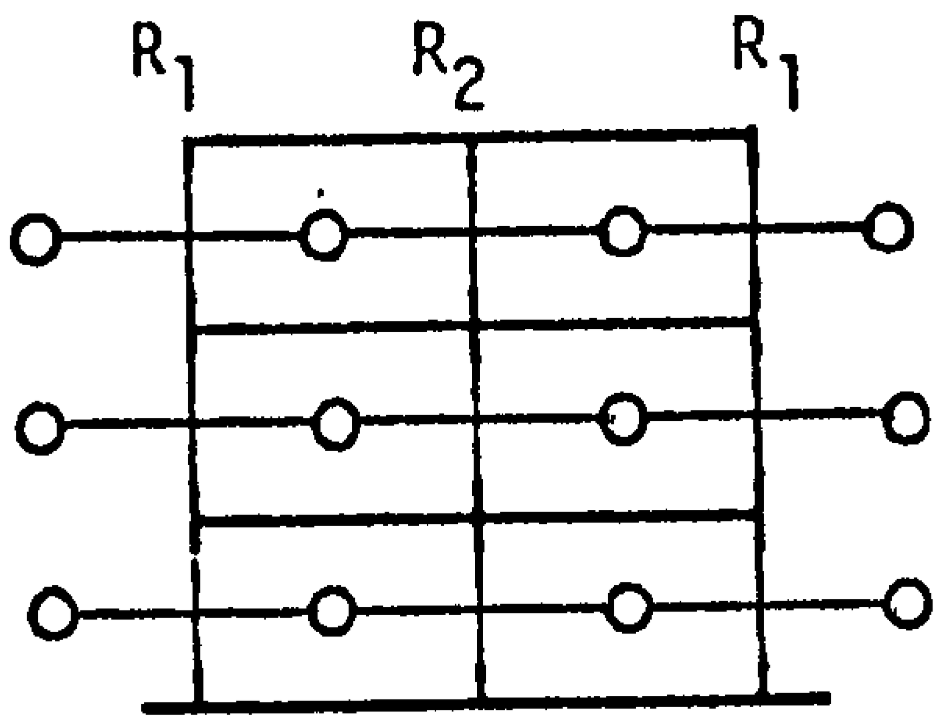


FIGURE 3.4 Arrangement of Test Box (After Hopkins and Hansford [16])

I



CASES CONSIDERED

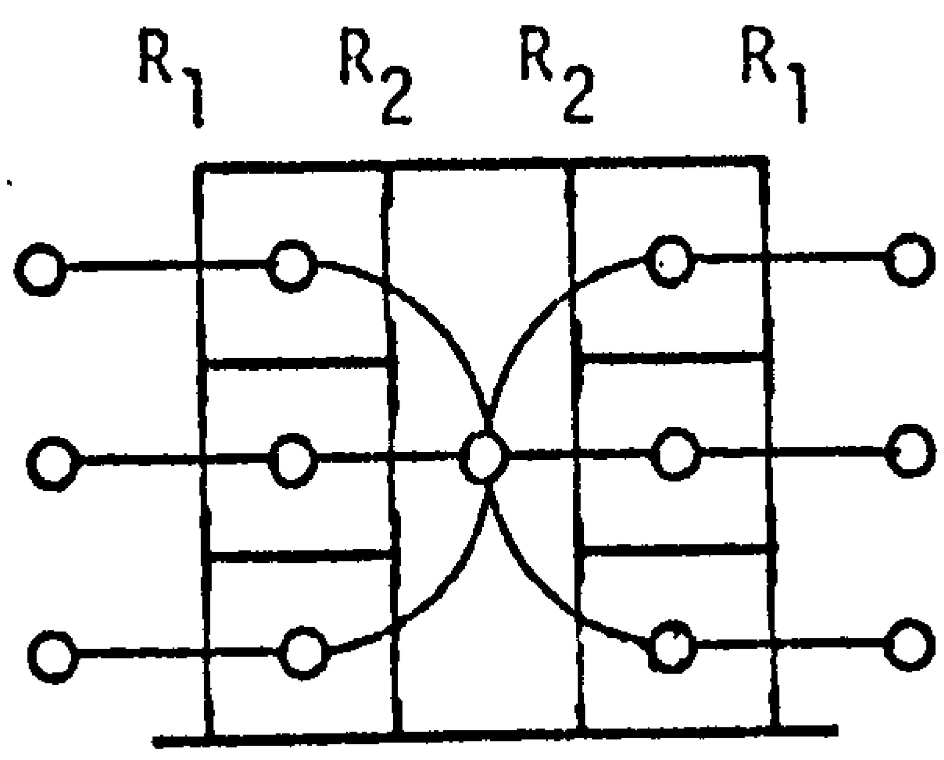
A $R_2 = 0$

B $R_2 = R_1$

C $R_2 = 2R_1$

D $R_2 = 3R_1$

II



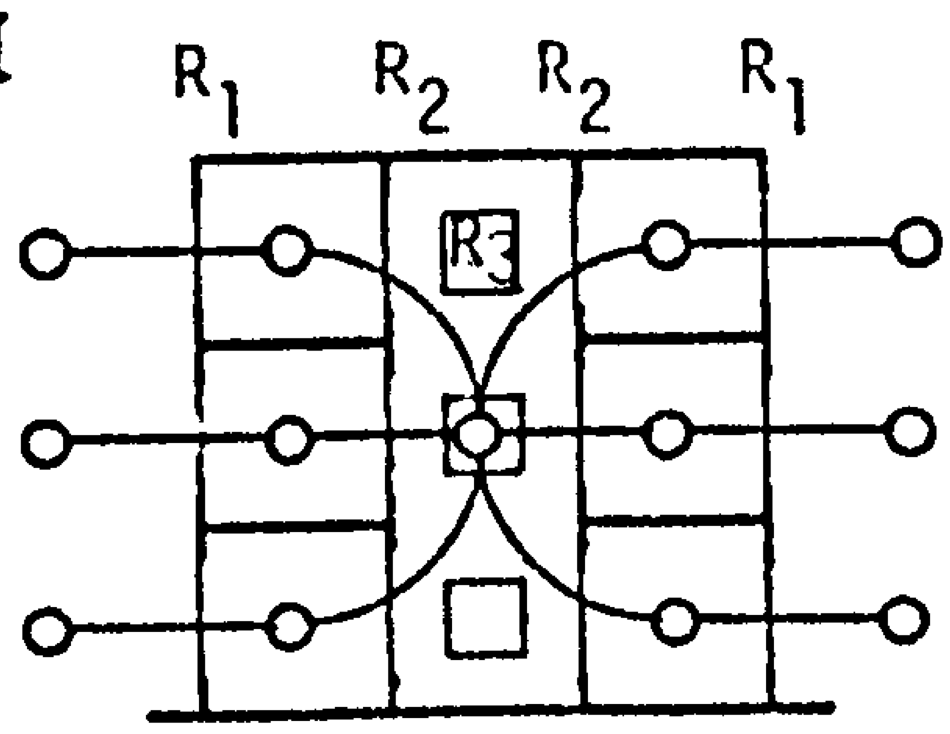
A $R_2 = R_1$

B $R_2 = 2R_1$

C $R_2 = 3R_1$

D $R_2 = 4R_1$

III



A $R_2 = R_1 = R_3$

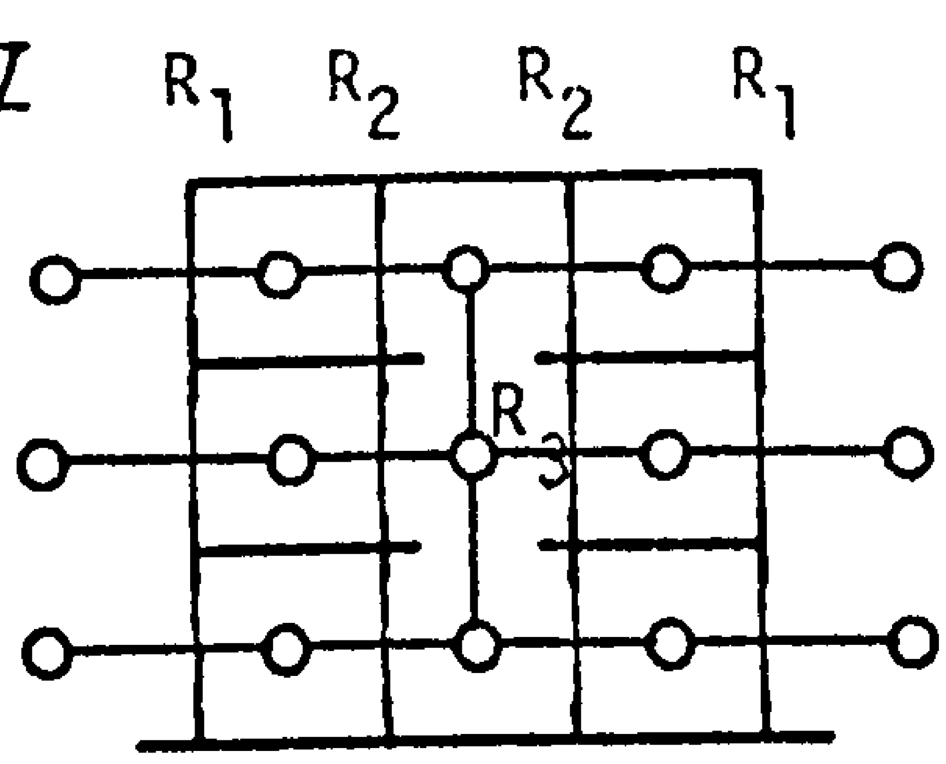
B $R_2 = 2R_1 = 2R_3$

C $R_2 = 3R_1 = 3R_3$

D $R_2 = 4R_1 = 4R_3$

(WITH STARCASE WINDOWS TO WINDWARD AND TO LEEWARD)

IV



A $R_2 = R_1 = 2R_3$

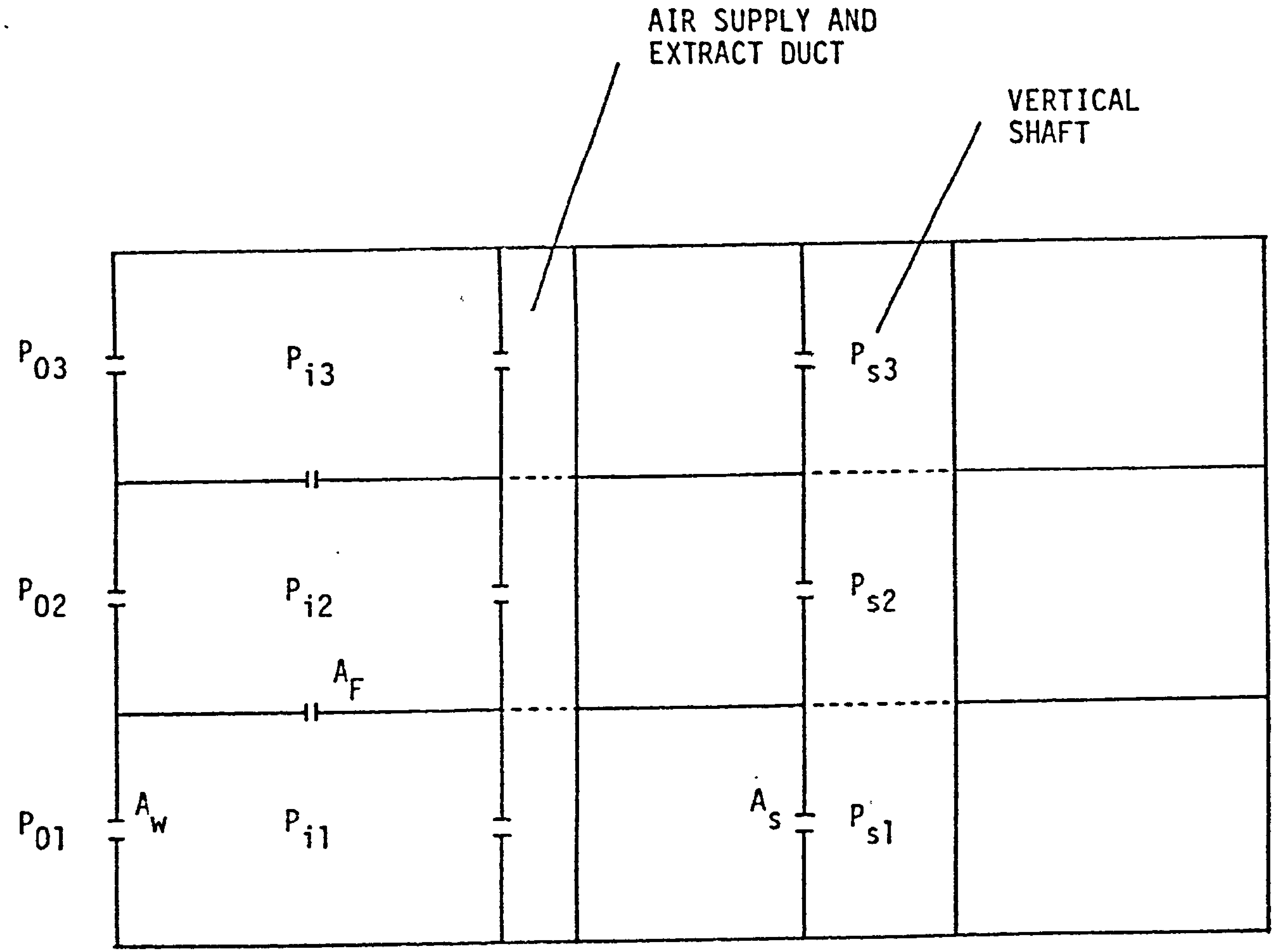
B $R_2 = R_1 = 4R_3$

C $R_2 = 2R_1 = 4R_3$

D $R_2 = 3R_1 = 6R_3$

E $R_2 = 4R_1 = 8R_3$

FIGURE 3.5 Building Types and Flow Paths (After Harrison [28])



A_W = EXTERIOR WALL ORIFICE AREA

A_F = FLOOR ORIFICE AREA

A_S = VERTICAL SHAFT ORIFICE AREA

A_V = VENTILATION DUCT ORIFICE AREA

P = ABSOLUTE PRESSURE
(SUBSCRIPTS REFER TO:
0 - OUTSIDE
i - INSIDE
s - SHAFT
1,2,3 - FLOOR LEVEL)

FIGURE 3.6 Mathematical Model of Building (After Tamura and Wilson [32])

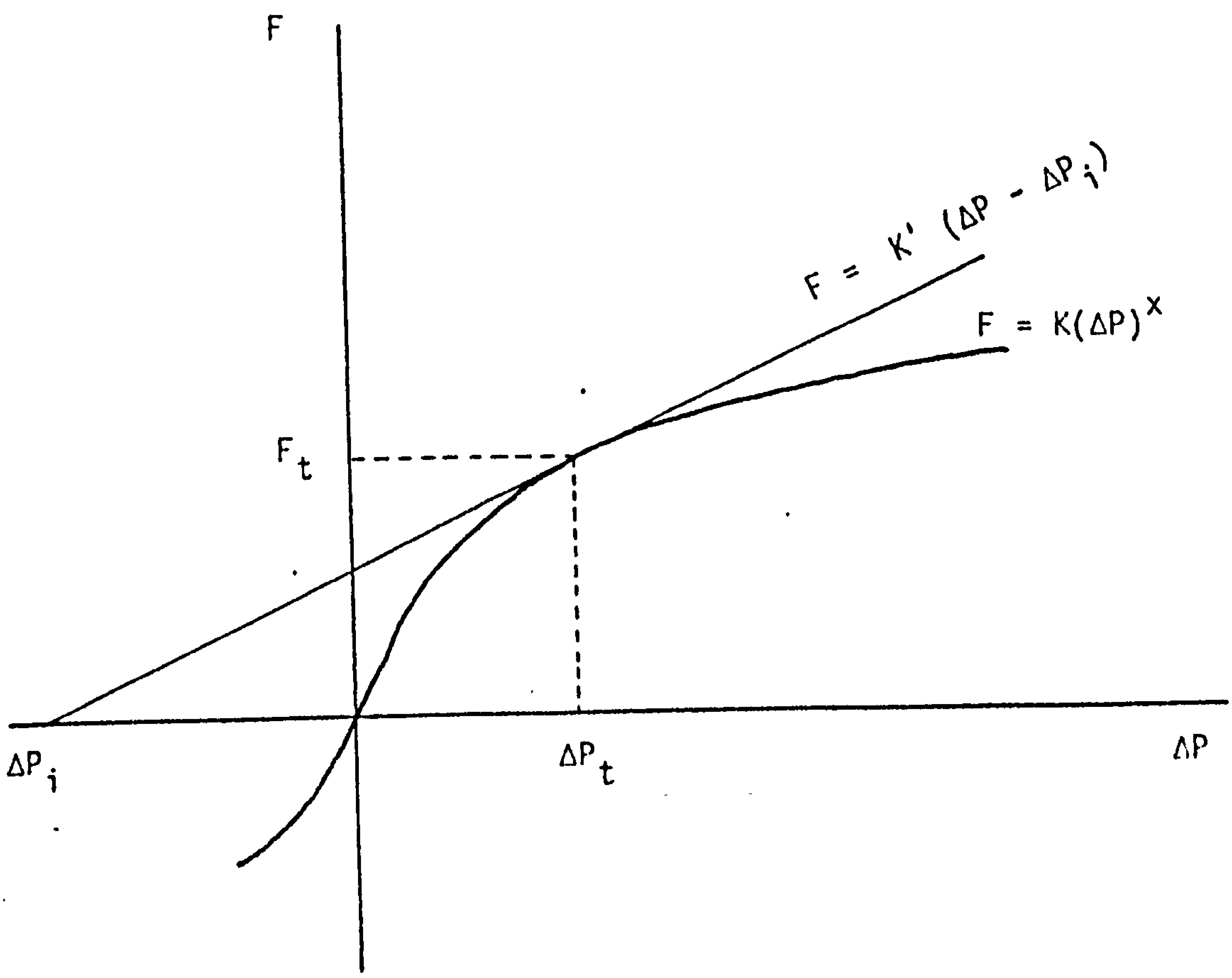


FIGURE 3.7 Linear Approximation of Flow Equation (After Sander and Tamura[36]

FIGURE 3.8 Pressure Differences Cause by "Chimney Effect" (after Tamura and Wilson [47])

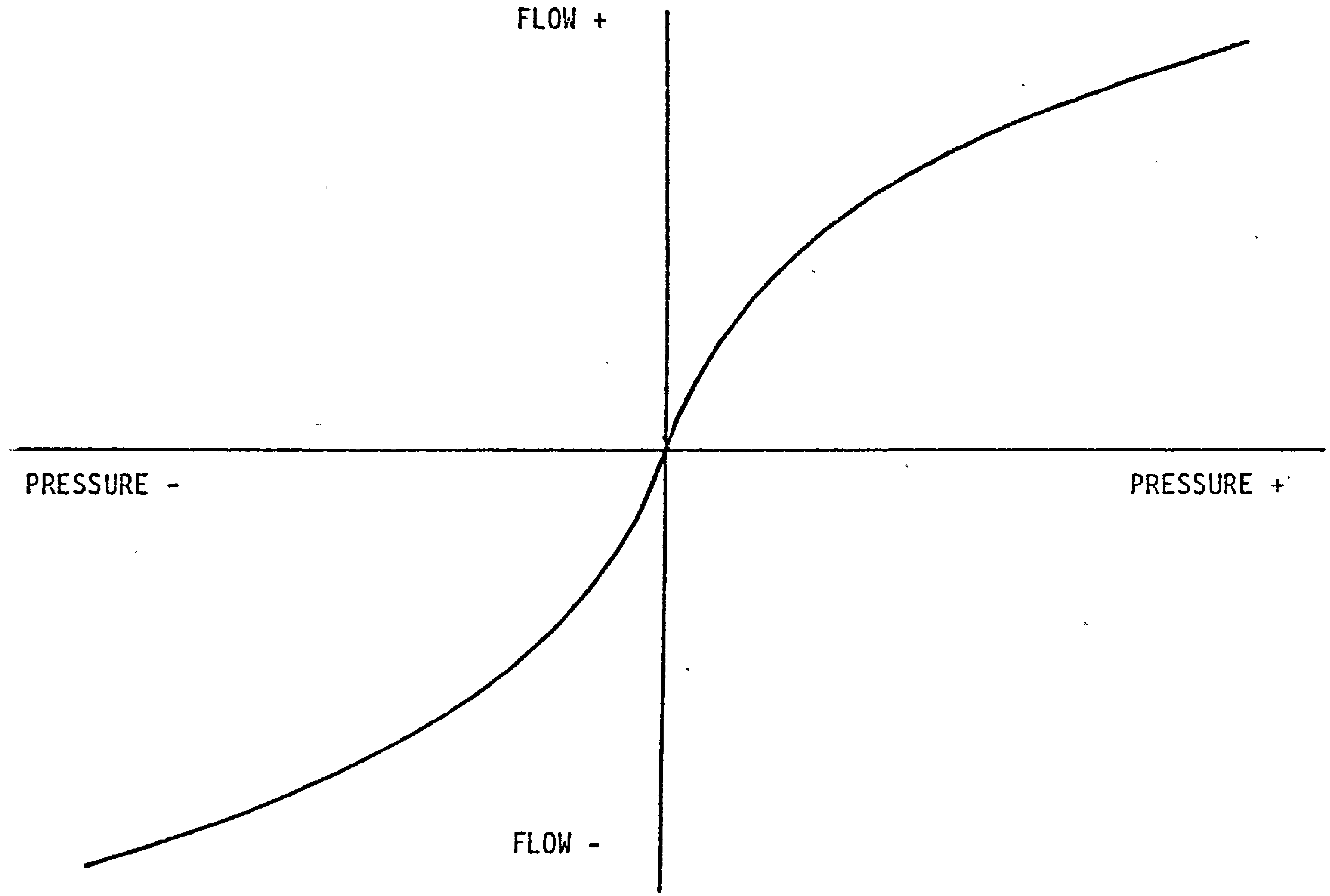
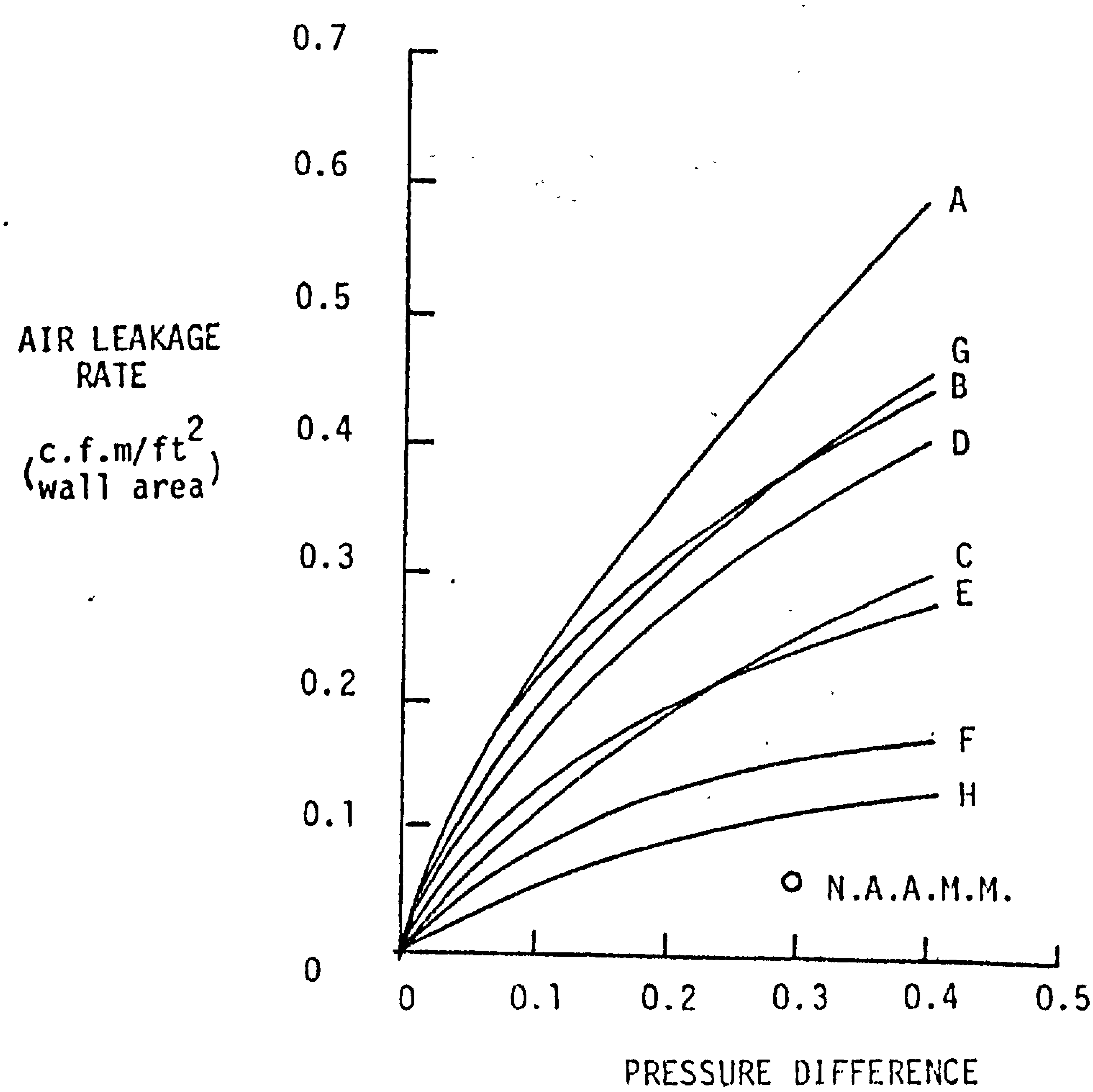


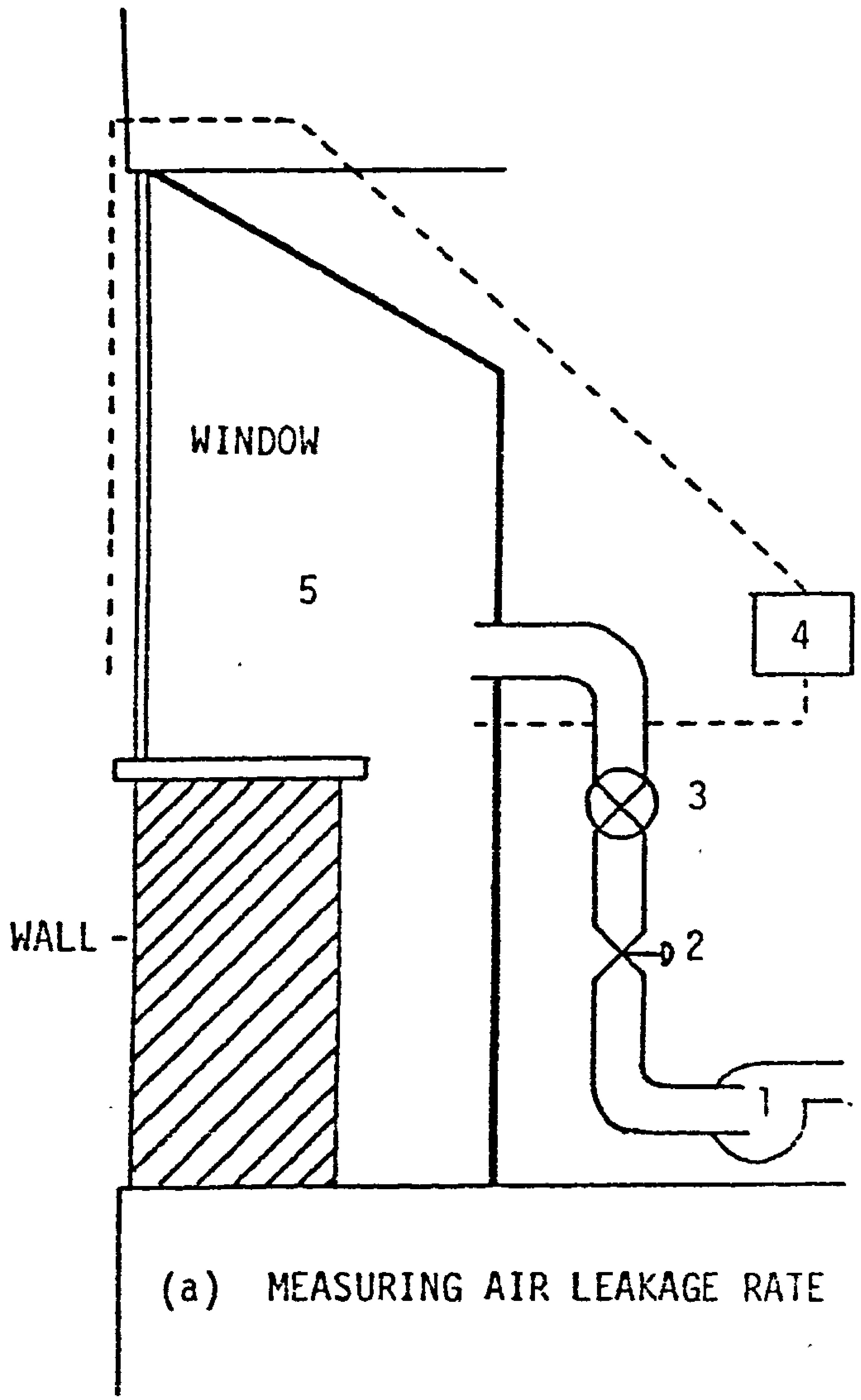
FIGURE 3.9 Idealised Flow-Pressure Relationship for Room or Building



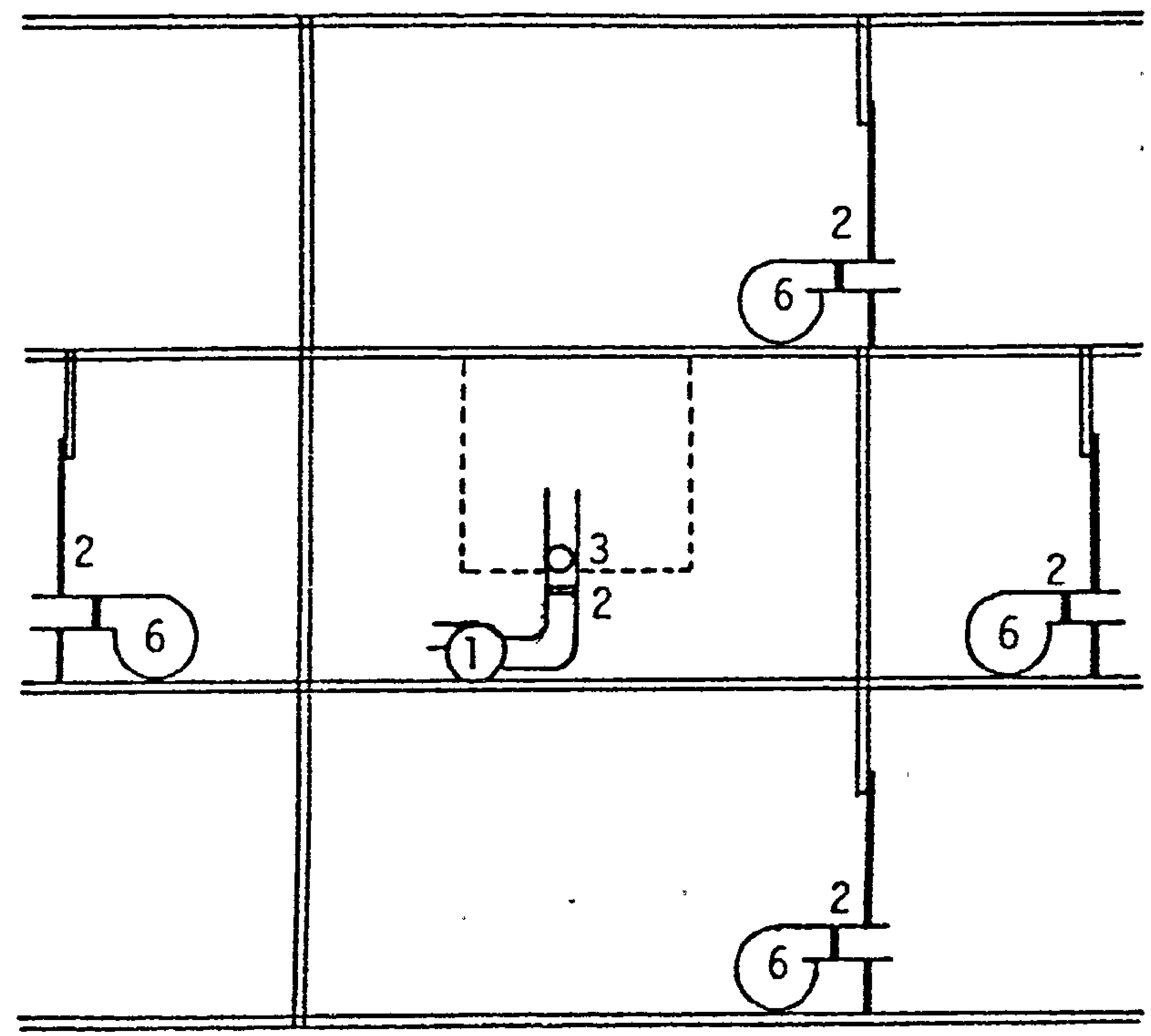
(N.A.A.M.M. = National Association of Architectural Metal Manufacturers Standard
= 0.06 c.f.m/Sq.ft. wall area at 0.3 ins. H_2O pressure)

FIGURE 3.10 Air Leakage Rates of Exterior Walls (After Tamura and Shaw [52])

1. FAN
2. DAMPER FOR FLOW CONTROL
3. FLOWMETER
4. PRESSURE TRANSDUCER
5. RIGID PRESSURE CHAMBER
6. PRESSURE BALANCING FANS



(a) MEASURING AIR LEAKAGE RATE



(b) BALANCING PRESSURES BETWEEN TEST CHAMBER AND ADJACENT ROOMS

FIGURE 3.11 Experimental Set-Up for Measuring Air Lekage Rate through Wall Assembly (After Shaw [53])

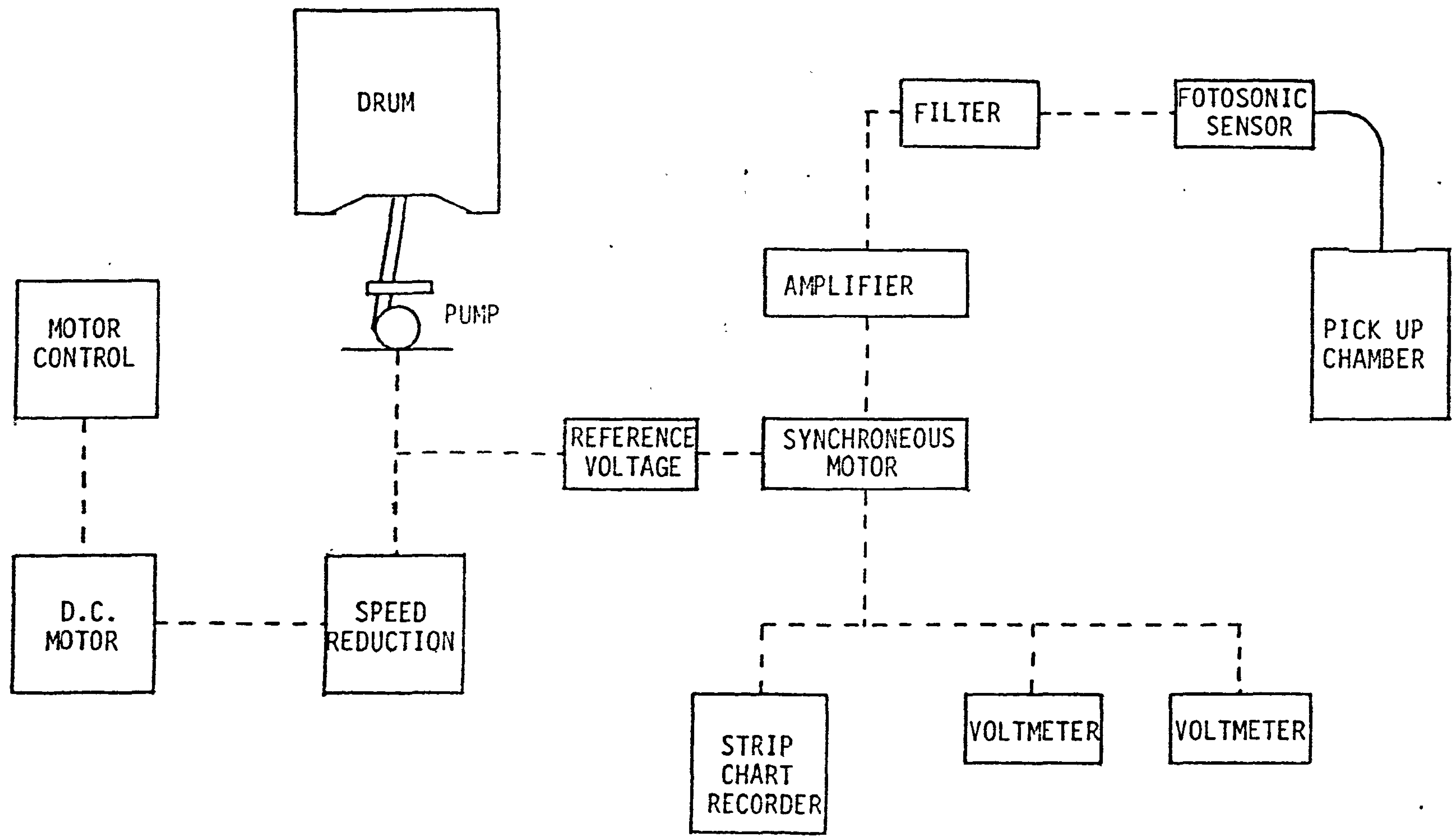


FIGURE 3.12 SCHEMATIC DIAGRAM OF INFRASONIC IMPEDANCE MEASURING SYSTEM (After Card et al [57])

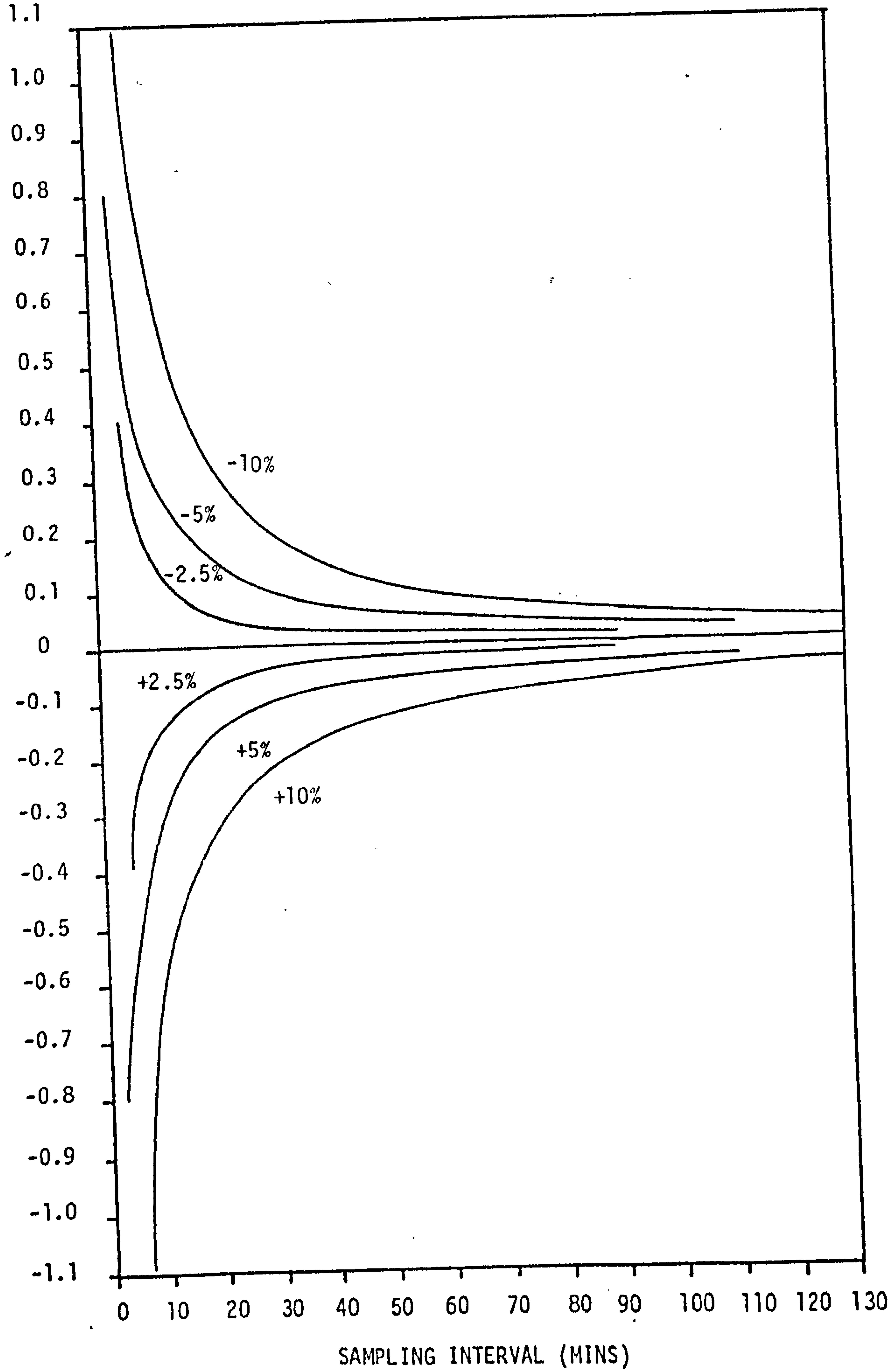


FIGURE 3.13 Infiltration Rate Calculation Errors as Fractional Error in $\mathrm{LOG}\ \left| {}^{[T]_t}/_{[T]_{t-1}} \right|$ (After Hunt [50])

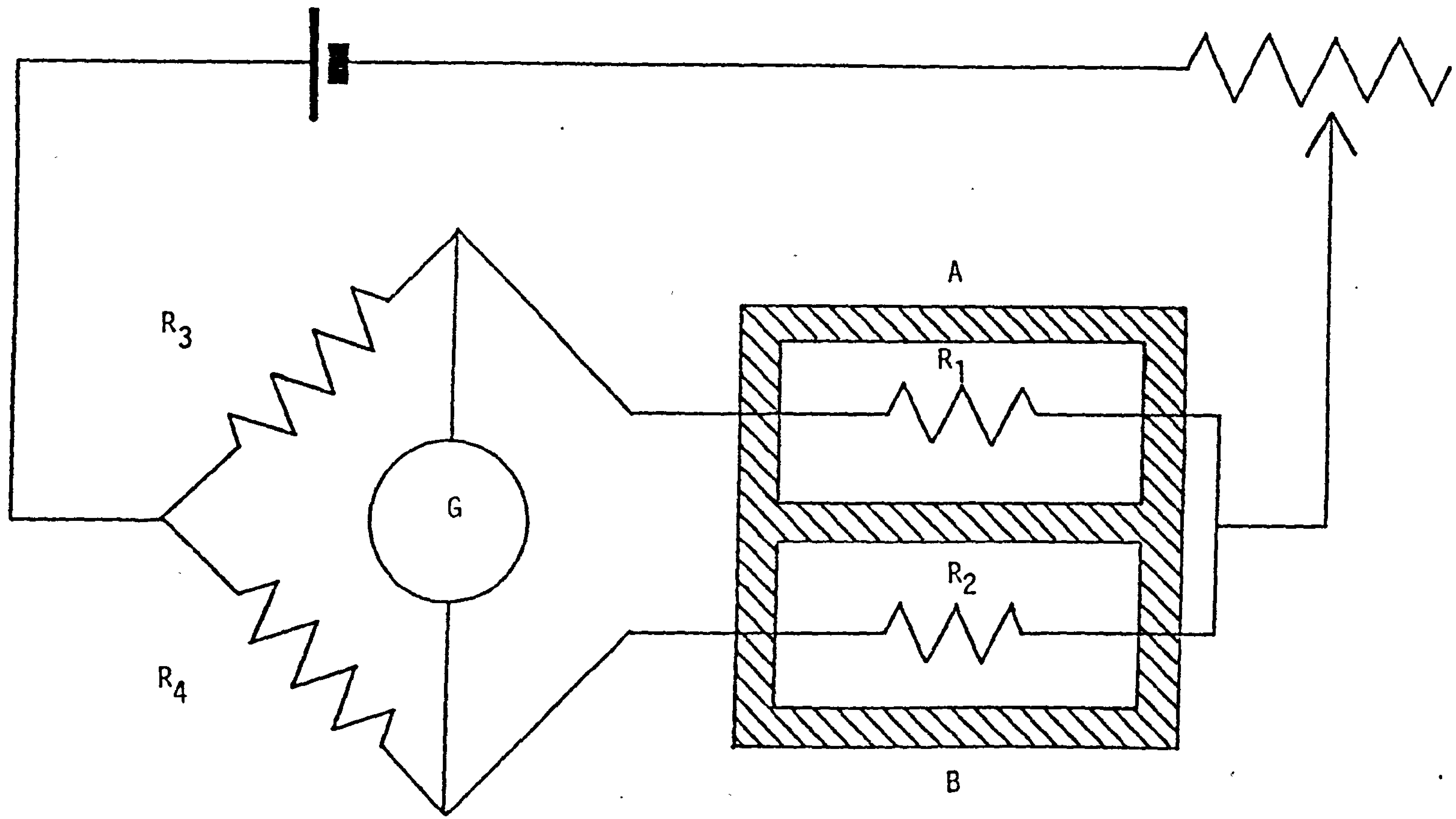


FIGURE 3.14 Basic Katharometer

CHAPTER 4

ENVIRONMENTAL AND VENTILATION PLANT MONITORING

4.1 INTRODUCTION

It had been decided to carry out monitoring at the ICI Fibres factory, Doncaster, both to establish the general environmental conditions in the fibre production areas and to investigate the energy flows associated with the air conditioning and ventilation systems. Because of the large number of ducts, the monitoring of the air movement was confined to one typical pair of machines.

Since the sensors were to be quite numerous and spread out, one requirement was that they produce an output, ie, an electrical signal, that could be recorded at a central location. In order to accumulate the sensor readings a means of recording a number of signals at predetermined and regular intervals was also needed, ie, a data logger.

4.2 DATA LOGGER

The main requirements for the data logger were that it should be able to record signals from a number of sensors at fairly frequent intervals over a period of time. Evidently some form of programming would need to be a feature of the logger. Consultations took place with the factory instrumentation engineer and a number of possible options were considered.

Simple programmable digital voltmeters did not provide sufficient flexibility of operation to be left unattended for the periods envisaged; subsequent data analysis would also have been difficult. Another option would have involved the use of a number of remote data collection points, each taking signals from a number of sensors and then relaying the information to some suitable micro-computer. However this approach was also limited in flexibility and interfacing problems could be foreseen. It was therefore decided to choose an all-in-one option: the sensor signal inputs, measurement, recording and analysis were all to be performed by one set of equipment.

ICI already possessed items of Hewlett Packard equipment and had experience of their operation and reliability; as a result the logger provided for the monitoring was the Hewlett Packard 3054DL. This consisted of two main units - a desk top micro-computer and a data aquisition unit.

4.2.1 THE MICRO-COMPUTER

The micro-computer was a Hewlett Packard model 85F. This had the usual typewriter keyboard with an additional numeric pad and also possessed special function keys. It had 32 K bytes of memory and integral cathode ray tube display, thermal printer and magnetic tape cartridge facility. The cartridges could store over 200 K bytes of additional information. BASIC was the programming language, but the system came complete with software to enable data logging to be carried out with no prior programming experience. The HP 85F was used to control the operation of the data aquisition unit.

4.2.2 DATA AQUISITION UNIT

The HP 3497A Data Aquisition Unit was connected to the HP 85F computer by the Hewlett Packard Interface Bus (HP-IB) which was based on the I.E.E.E. interface. The data aquisition unit was a versatile piece of equipment Provided as standard were a real-time dock, a front panel display and independent control panel, and a digital voltmeter assembly with the following features: 5½ digit resolution, 1 μV sensitivity, autoranging and a current source for resistance measurements. Five plug-in ports on the rear enable the unit to be used in a number of ways. For the data logging to be carried out five, 20 channel, low thermal relay multiplexer assemblies were chosen, thus allowing upto 100 input channels.

4.2.3 SYSTEM SOFTWARE

The data logging system came complete with three levels of software, stored on magnetic tape cartridges. Level 1 was intended as a simple introduction to the logger and allowed upto 30 channels to be monitored. A program would be loaded at start-up which asked questions of the user in order to set up the scanning routine. The types of functions were limited and only simple storage/print-out was allowed. Level 2 enabled all 100 channels to be addressed and had a much wider set of functions that could be employed. Alarm limits, linearization equations and graphical outputs could be set-up. Neither Level 1 nor Level 2 necessitated the use of the BASIC programming language. However Level 3 did require the user to write a program. This was aided by the availability of a number of pre-written subroutines to perform functions. A program would be written containing calls to these subroutines. More facilities would be available if Level 3 was used, combined with a more flexible logging system.

4.3 SENSORS

As has already been stated, some form of recordable output was required from any sensor to be used, indicating the need for transducers producing electrical signals. For environmental measurement the most obvious and useful parameter is temperature, but in order to define the condition of the environment, a second measurement is required. This second measurement is usually related in some way to the moisture contained within the air. From two suitable parameters, all others can be derived; for example, the energy content. In order to determine energy flows associated with the duct air flows a third measurement, that of flow rate, was also needed.

4.3.1 TEMPERATURE MEASUREMENT

For the measurement of temperature, three common transducers were available; thermocouples, resistance temperature detectors (RTD's) and thermistors.

Thermistors were discounted at an early stage: they were too fragile, non-linear and non-standard to be used in the numbers and positions required by the monitoring. The main choice was therefore between thermocouples and RTD's. Though thermocouples themselves were relatively inexpensive, the long cable runs (using thermocouple wire) required to connect to the various parts of the factory, increased the cost quite considerably. RTD's on the other hand were more expensive though normal connecting cables could be used. In addition they had a higher degree of accuracy than thermocouples. After consulting with ICI staff, RTD's were chosen since their cost would be only marginally more overall, and platinum RTD's were commonly used at the factory, which would make installation easier.

There was one drawback in the use of RTD's however. Their principle of operation lies in the small changes in resistance that occur with temperature variations. In the factory environment long cable runs and general electrical "noise" would have a severe detrimental effect upon the clarity of readings of such small changes. There was a way of avoiding these problems with the logging system; this involved taking "4-wire" measurements. The typical measurement circuit is given in Figure 4.1. (Noise elimination was enhanced by the use of a guard connection). The use of such 4 wire circuits is common and was employed in the monitoring to be carried out here. Since two input channels were required to take such a reading, this did reduce the overall number of channels available, but this did not adversely restrict the monitoring.

4.3.2 SECOND ENVIRONMENTAL MEASUREMENT

In a number of environmental measurement systems the second parameter used has been a "wet-bulb" temperature redding. The use of this as part of a whirling hygrometer was described in Chapter 2. Though it is not necessary to take the reading in a whirling mode, (a stationary measurement is allowed but must be used in conjunction with different tables and charts) and neither is it necessary to use a mercury thermometer (therocouples and RTD's can be adapted for use); the principal fault with using a wet-bulb temperature is the need for "wetness". The wetness is normally provided by a wick-like covering for the sensor, which has one end in a reservoir of distilled water. As some sensing positions were to be inside ducts and in other inaccessible locations, the operation of wet-bulb readings would have been very difficult and would have also limited unattended monitoring. Another type of sensor was therefore sought.

The sensor chosen was a Lee-Dickens Humidity Probe. This had the advantage of having been previously used by ICI with satisfactory results, and was considered robust enough for the industrial environment. It operated by sensing a change in capacitance caused by the absorption of water molecules by a polymer dielectric.

A linear output of 0 - 1 mA was produced, representing 0-100% relative humidity. The power supply required was nominally 12 V dc at 10 mA. The probe itself was contained in a tubular metal casing 180 mm in length, 23 mm in diameter, with a 6 pin DIN socket at one end, and the sensors, encased within a sintered bronze filter, at the other. Besides the humidity sensor the probe also contained a platinum RTD, which matched the temperature sensing requirement already discussed.

In order to take the measurements, the current produced by the humidity sensor was fed through a precision, wire-wound 10 kΩ resistor, and it was the voltage across the resistor that was recorded. In this way 0 - 10 V represented 0 - 100 % RH. Any slight errors introduced by using this method would be eliminated by the calibration check carried out (this check is described later).

4.3.3 FLOW MEASUREMENT

The normal methods of flow measurement were not available for use in the monitoring of the air flow in the ducts serving the spinning machines. Pitot-static tubes were not suitable for measuring the low flow rates, did not give a suitable recordable output and were susceptible to blockage in the "dirty" environment. Orifice plates were obviously unsuitable to be placed in the ventilation ducts and again did not give an easily recordable output. The hot-wire anemometer was too delicate an instrument to be used and required frequent calibration. The one piece of apparatus that did appear suitable was the vane anemometer. This consisted of eight thin metallic vanes mounted around a hub. The vanes were balanced and the hub rotated on shielded ball races. The diameter of the whole vane head was approximately 100 mm. Such vane heads were produced by Airflow Development Ltd.

The usual method of flow velocity determination was to connect the vane head to an electronic measurement and readout box, also produced by Airflow Developments. (This having been used by the Department of Building Science in other investigations). However the cost of each measurement and readout unit, and the number of measurement positions, precluded their use, and another means of using the vane anemometers was required.

It was known that the operation of the vane head was dependent on the passing of each vane across a metal plate. When an excitation power supply was provided, a pulse was output at the passing of each vane. Thus if the number of pulses was determined this would allow the derivation of the speed of rotation and so flow velocity. Therefore a frequency to voltage converter was required together with a suitable power supply. The circuit was developed in conjunction with, and produced by the ICI Instrument Workshop. The basic circuit diagram is shown in Figure 4.2. A switch was included to convert to different ranges, though once set up the circuits for all the vane anemometers were used on the same setting. Several identical circuits were produced.

In order to determine the flow velocity from the voltage output of the circuit, a simple calibration rig was constructed. This is depicted in schematic form by Figure 4.3. An air flow was set up by the fan and the corresponding readings of flow velocity (measured using the university's Airflow Developments equipment) and voltage, were noted. This was carried out for a number of flow rates and for each combination of vane anemometer head and frequency to voltage circuit. There was little or no difference between the vane circuits so it was possible to interchange circuit/ vane combinations and use one set of averaged figures from which to derive a relationship. The summarised results are given in Table 4.1

TABLE 4.1 CORRESPONDING MEASUREMENTS OF CIRCUIT VOLTAGE AND FLOW VELOCITY

VOLTAGE (Volts)	VELOCITY (m/s)
0.27	1.0
0.55	2.0
0.85	3.0
1.15	4.0
2.04	7.0
2.33	8.0
2.63	9.0

A simple linear relationship was observed to link these measurements (least squares fit with r = 0.9999)

$$Velocity = (Voltage + 0.034) / 0.296 \qquad (4.1)$$

The vane anemometer was to be mounted in the centre of each duct to obtain an estimate of the air flow, this being the most practical way of obtaining a sensible observation.

4.4 MONITORING POSITIONS LAYOUT

The general locations of the sensors were decided upon and power and signal cables were routed to appropriate positions. The data logger itself, was located in the cloakroom behind the Foreman's Office in the Spin Doff area. This position can be found in Figure 4.4 (which is a plan of the Ground Floor of the Spinning Tower) at grid reference Ta, 6. All signal cables were routed back to that point.

Two main groups of monitoring points were set up: (i) general monitoring of the internal environment spread out over four "floor" levels; and (ii) monitoring of the ventilation systems serving machines 27 and 28. In addition to this, at the request of ICI staff, some addition points were monitored in the Drawtwist area, adjacent to the Spin Doff area.

4.4.1 GENERAL MONITORING

The monitoring took place at Spin Doff, Extrusion, Extrusion Catwalk and Hopper Floor levels within the factory. Also, outside conditions were measured from a point outside on the Eastern wall of the factory at Hopper Floor level. On each of the internal levels temperatures were measured near machines number 16/17; 27/28; 35/36. By machines 27/28, the humidity was also monitored at each level.

Each of the sensors was positioned in such a way to reflect, as accurately as possible, the conditions of its environment and the conditions likely to be experienced by staff working in the area. In general this entailed a position at about head height, attached to a neutral support (ie, neither a heated or a cooled surface) using plastic clips and ties.

During the course of the monitoring some positions were excluded because they gave erroneous or unnecessary readings. The final list of positions, the readings from which were used in the analysis (see Chapter 8) is given in Table 4.2 along with the exact locations. If the grid references given are used with Figures 4,4; 4.5 and 4.6, the positions can be identified.

4.4.2 VENTILATION MONITORING

It was not possible to monitor the ventilation systems serving all spinning machines and it had been decided to concentrate on just two machines (this being a sensible unit for monitoring). Most production was carried out on the more modern Type 14 machines and reference to the production schedule showed that machines 27 and 28 would be in almost continuous use for the duration of the monitoring. Since these machines were also close to the data logger they were chosen.

A number of ventilation ducts served these machines, seven in all (the systems were described, with diagrams in Chapter 2). Temperature and humidity were measured in each duct using a Lee-Dickens probe. The probe was clamped and sealed into the side of each duct with the sensor protruding approximately 150 mm into the duct. The flow rate was measured by a vane anemometer which was positioned in the centre of each duct. The anemometer was supported by a metal bar which was attached to a plate clamped to the side of the duct. All the duct fixings were designed so that the sensors could be removed if required, for cleaning and calibration.

Table 4.2 gives the positions of the duct sensors and the locations can be identified by referring to Figures 4,4; 4.5 and 4.6.

TABLE 4.2

FINAL LOCATIONS OF MONITORING SITES (FLOOR LEVEL AND GRID REFERENCE)

GENERAL ENVIRONMENT MONITORING

Temperature and Humidity	Outside (Hopper Floor - Wn6 Spin Doff - Tb401 Extrusion - Tc301 Extrusion Catwalk - Tc301 Hopper Floor - Tb301
Temperature	Spin Doff - P4, Vb301 Extrusion - P4, Vb301 Extrusion Catwalk - P4, Vb301 Hopper Floor - P4, Vb301

VENTILATION MONITORING (Temperature, Humidity and Flow at each Location)

Spin Doff Supply Duct	Spin Doff - Tb6
Spin Doff Underfloor Supply Duct	Spin Doff - Ta6
Spin Doff Extract Duct	Spin Doff - Tb6
Extrusion Supply (Large) Duct	Extrusion Catwalk - Tc5n
Extrusion Supply (Small) Duct	Extrusion Catwalk - Tc5n
Extrusion Extract Duct	Hopper Floor - Tc2
Blower Air Duct	Extrusion - Tc401

ICI DRAWTWIST AREA MONITORING

Temperature and Humidity	Drawtwist - C Bank - Lag Area, C4 - C5, C7 - C8, C10 - C11

4.4.3 ICI DRAWTWIST MONITORING

The sensors and sensor locations in the Drawtwist area were defined by ICI staff, to whom the results were made available. Apart from the temperature reading of the sensor nearest to the Spin Doff area, the measurements were not otherwise used in this study.

4.4.4 LOGGER CONNECTIONS

For ease of operation of the data logging system the sensors were connected in sequential blocks of the same type of sensor and in the same order (as far as possible) for each type of sensor. This allowed monitoring and analysis programs to make use of repeated loops reducing the complexity and length of the programs.

4.5 SYSTEM PROGRAMMING

As mentioned in section 4.2.3 the data logger was provided with various programs, by the manufacturer. The three levels of software enabled varying degrees of complexity and functions to be encompassed. However even when the highest level was investigated and used in trials, it was still found to be lacking. The method of program "construction" via a data file was very laborious and time consuming. The program produced was not very efficient and was over long since it was designed to cope with almost every eventuality. Since the logger was to be devoted to one prime application, it was unnecessarily complex. The storage and print-out of data was also restricted with fairly low density storage on the magnetic tape. The form of storage made later analysis difficult too. As a result it was decided to produce a special-to-application data logging program (named FANLOG).

The manufacturer's software was not unused however - a number of relevant algorithms were taken and incorporated into programs. The first program produced was for the testing of each sensor at installation and for checking during the duration of the study. It simply took in a reading from a defined input channel and converted it to either a voltage or a resistance depending upon the sensor type. For RTD's the temperature equivalent to the resistance could also be obtained.

A full description of the data logging program that was developed, is given in section 4.7.

4.6 SENSOR CHECKS/CALIBRATION

All sensors were initially checked before installation by the ICI Instrument Workshop and by the author. However, bearing in mind the environment and the duration of the study, further checking of the sensors was required to give confidence in the readings.

4.6.1 RESISTANCE TEMPEATURE DETECTORS

The platinum RTD's were checked at installation by comparison of their resistance (measured by digital multimeter) converted to temperature and accurate mercury-in-glass thermometers. Further, during the course of the study the RTD's temperature, as measured and calculated by the data logger were periodically checked against mercury thermometers. In addition the data logging program rejected and noted obviously out of range readings provided by any sensor. Error producing sensors could then be checked.

4.6.2 HUMIDITY SENSORS

These instruments were provided with a calibration pack containing glass bottles with silicone rubber seals through which the probes could be pushed. Within the glass bottles specific humdidities could be created. A zero humidity was provided by a "molecular sieve dessicant" and various other humidities, by using saturated salt solutions which gave almost constant and fixed humidities in the air above their surfaces. In this study sodium chloride salt was used as it provided a useful nominal reference humidity of 75 %, with less than 2 % change between 15 and 35°C. The small inaccuracy, introduced by the variation, being acceptable in the measurement. The full range of salts and method of probe calibration is described in reference (1). If any deviation in probe reading from that specified, was found, then corrections could be made by adjusting zero and span settings on the probe.

The probes were first checked in this way before installation, in a temperature controlled cabinet. A second check was carried out when the sensor was mounted in position (ie, in situ) and connected to the logger. Further checks and any adjustments necessary, were performed during the course of the study at periodic intervals. The computer program also checked readings and rejected any that were out of range. Damaged probes were replaced by spares if necessary.

4.6.3 VANE ANEMOMETERS

The calibration of vane anemometers with their frequency to voltage circuits has been described in section 4.3.3. In order that the readings obtained were the most accurate available, further checking was required.

The anemometers were each positioned in the centre of their respective ducts, at locations with several metres of straight section before and after the measurement position. The central position was chosen as it would give the most reasonable indication of the bulk flow rate. It was recognised that the actual average flow might vary slightly from that measured, however Legg(2) has shown that such central positions do give fairly accurate results.

Only limited checking of the assumption of central flow rate equal to average flow rate was possible. This being due to awkward access to some ducts and the inability to greatly vary the flow rate of the working plant. However, some comparisons were performed between pitot static tube/manometer observations of duct flow (using a 26 point log-linear technique(3) - described in detail in Chapter 6) and vane anemometer readings, averaged for the period of the test.

TABLE 4.3 COMPARISON OF ACTUAL AVERAGE AND MEASURED DUCT FLOW RATES

DUCT	MEASURED (m/s)	ACTUAL (m/s)
S Plant Supply	9.25	9.1
	7.7	7.8
	7.37	7.2
S Plant Extract	10.2	9.9
	9.51	9.4
Extrusion Extract	7.05	7.1
	8.1	7.9

The comparisons are given in Table 4.3. It can be seen that the central reading gave a good approximation to the actual flow resulting on average, in a slight over-estimation. Given the constraints of the study, this variation was considered acceptable, as no better method was available.

For the purpose of data logging, each time a flow measurement was called for the velocity was scanned ten times, at one second intervals, to obtain an average reading. This removed short term fluctuations and provided a reliable flow rate determination. Readings which were obviously out of range were excluded by the software and an error noted which could be investigated later.

The vane anemometers were regularly checked for physical damage and cleaned (especially those in the extract ducts). As with the other sensors, spares were kept so that damaged and malfunctioning probes could be replaced.

4.7 FANLOG - DATA LOGGER PROGRAM

The basic flow chart for this program is given in Figure 4.7 (Sheets 1 - 7). The program listing is also given at the end of the Chapter.

The purpose of the program was to determine measurements of temperature, humidity and flow at defined periodic intervals; to check the validity of each measurement and then to perform a basic analysis resulting in readings being printed out on thermal paper and/or stored on a magnetic tape cartridge for later, more detailed analysis.

The program was kept on a magnetic tape cartridge and read into the computer memory when required. When the program was run, it asked for a certain amount of initial information in order to set up the scanning, after which it operated unattended. The information required was:-

(i) Time, date and year

(ii) Interval between scans (three minutes to 24 hours)

(iii) Time of first scan

(iv) The number of temperature, humidity and flow rate measurements to be made

(v) Was tape storage required ? - either of daily maximum and minimum readings, or of hourly averages. If so, insert data tape and specify time span of scanning (a blank tape could record in excess of 150 days of daily data or more than 400 hours of hourly averages). Also required was a name for the data file which was to be created.

(vi) Was thermal paper printout required? - the options were:

  (a) daily maximums and minimums
  (b) hourly averages
  (c) selected hourly averages (selected by ICI staff)
  (d) No printout required
  (e) (a) and (b) above
  (f) (a) and (c) above

(vii) Sensor channel specification. If required this could be set individually by the operator, if the channels and groups of sensors were first identified. Alternatively, a standard specification could be input by the program. The standard specification included all sensors operating at the time.

Figure 4.8 shows an example scan start-up (inputs from the operator are underlined). This scan was set up at 11.15 am on 1 September 1983. Scans at ten minute intervals to start at 11.30 am. 27 temperature, 16 humidity and seven flow rate measurements. Hourly averages were recorded on tape for the following 100 hours in a file named "01/09+". Daily maximums and minimums were printed out, as were selected hourly averages. Standard sensor specification was used.

If there were any errors during the input the operator could stop and start again. When all the information was input, the time of next (first) scan was displayed for 30 seconds. (The appropriate next scan time was displayed for 30 seconds after each scanning routine).

The daily maximums and minimums referred to comprised, where applicable for each location, maximum and minimum temperature, humidity, flow velocity, air enthalpy, duct mass flow rate and duct energy flow rate. The hourly averages referred to comprised, where applicable for each location, hourly averaged temperature, humidity, flow velocity and duct energy flow rate.

Several periods of monitoring were undertaken using the FANLOG program, at different times of the year. The analysis of the readings and other computer programs developed are described in Chapter 8.

The main advantages of the FANLOG program were that it performed a degree of initial data analysis; it allowed recording and printout of a variety of measurement data; it stored data in a sufficiently compact form to allow unattended operation for in excess of two weeks; and the form of storage made it accessible for subsequent detailed analysis.

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HUMIDITY MANUAL, TECHNICAL DATA ISSUE 1/82

2 R C LEGG

The measurement of air volume flow rate in rectangular ducts with vane anemometers using a single observation.

Paper presented at Conference "Site Testing of Fans and Equipment" I Mech E 1978

3 E OWER AND R C PANKHURST

The Measurement of Air Flow

Pergamon Press 1977

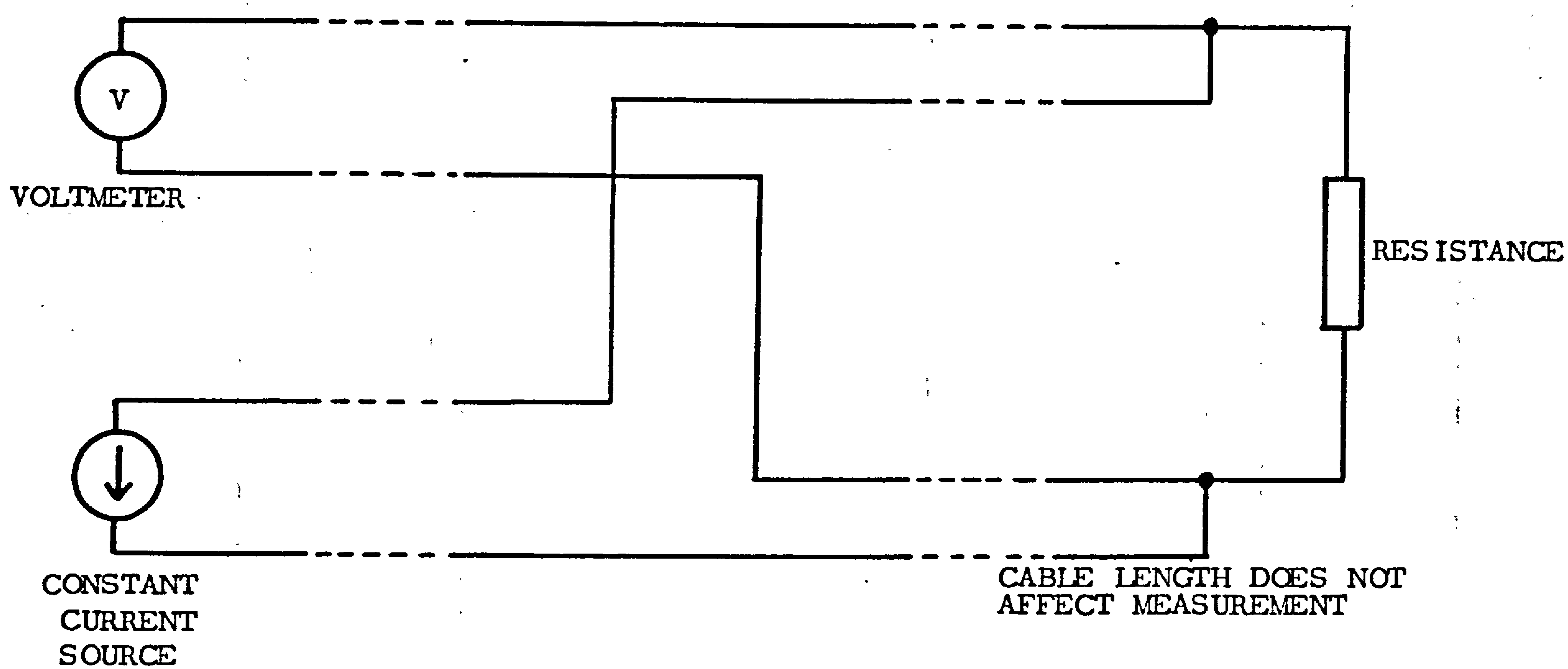

FIGURE 4.1 TYPICAL FOUR-WIRE RESISTANCE MEASUREMENT

FIGURE 4.2 BASIC CICUIT DIAGRAM : VANE ANEMOMETER FREQUENCY TO VOLTAGE CONVERTER

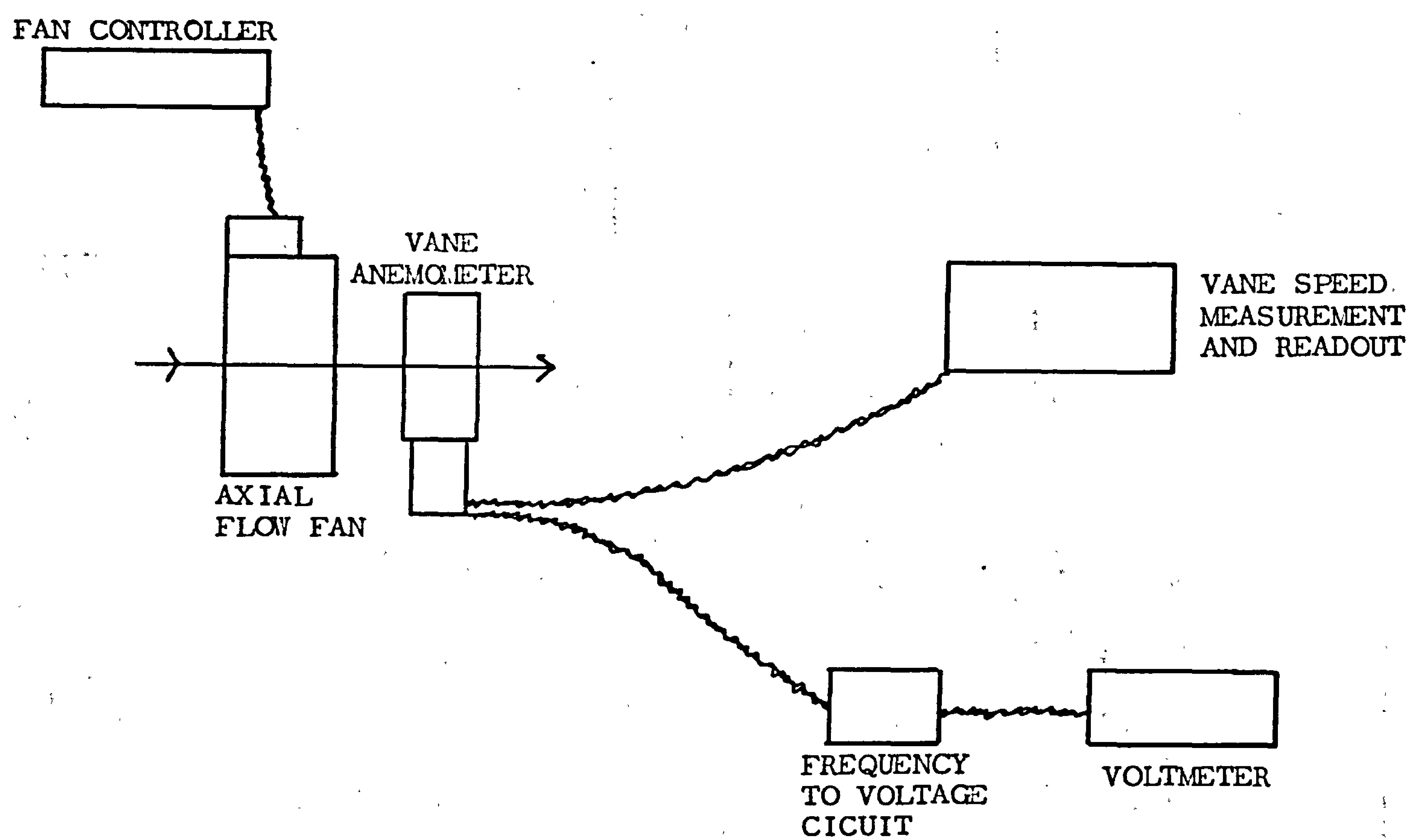

FIGURE 4.3 SCHEMATIC DIAGRAM - CALIBRATION OF VANE ANEMOMETER

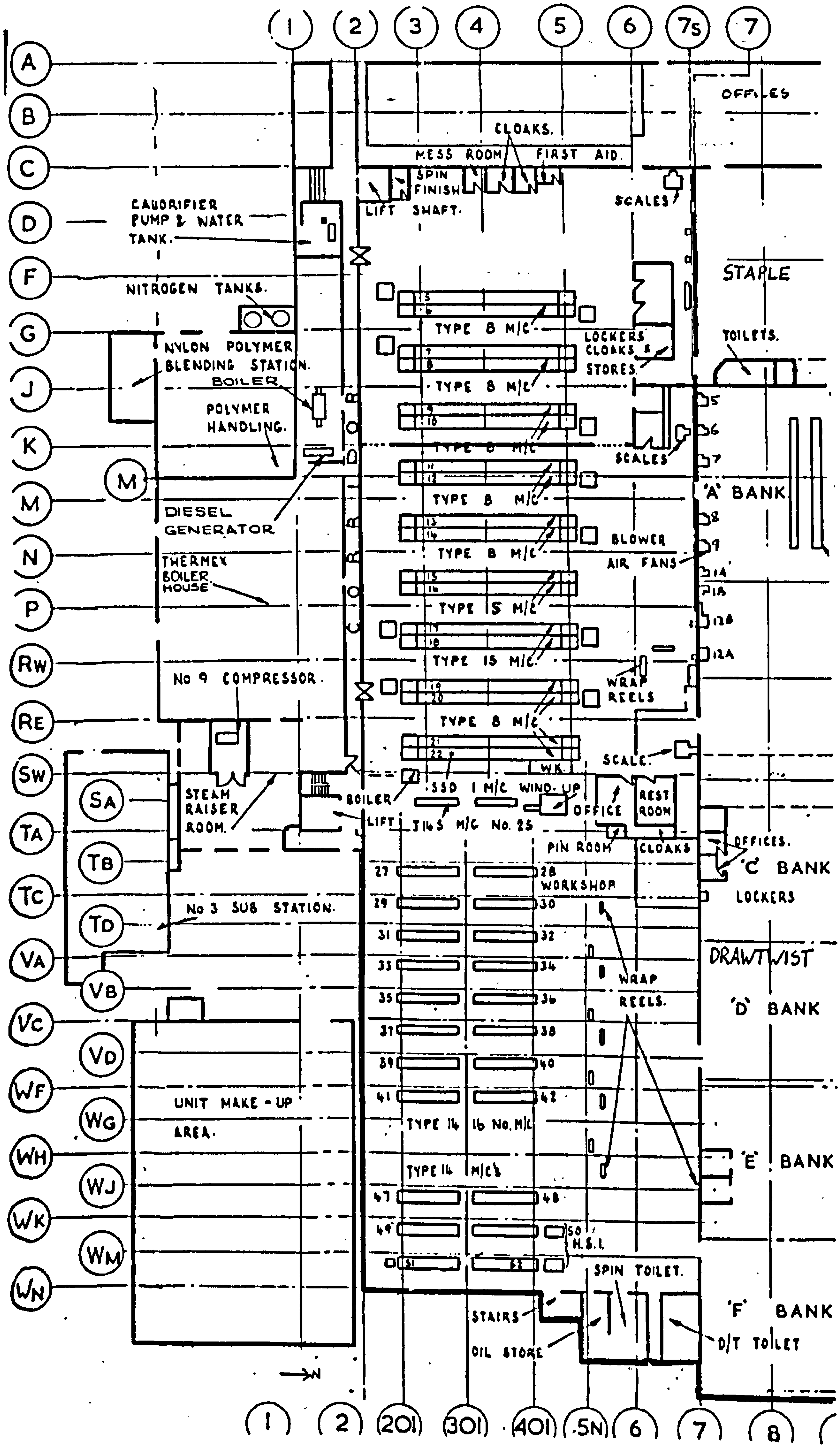

FIGURE 4.4 GROUND FLOOR PLAN (SPIN DOFF)
ICI FIBRES, DONCASTER

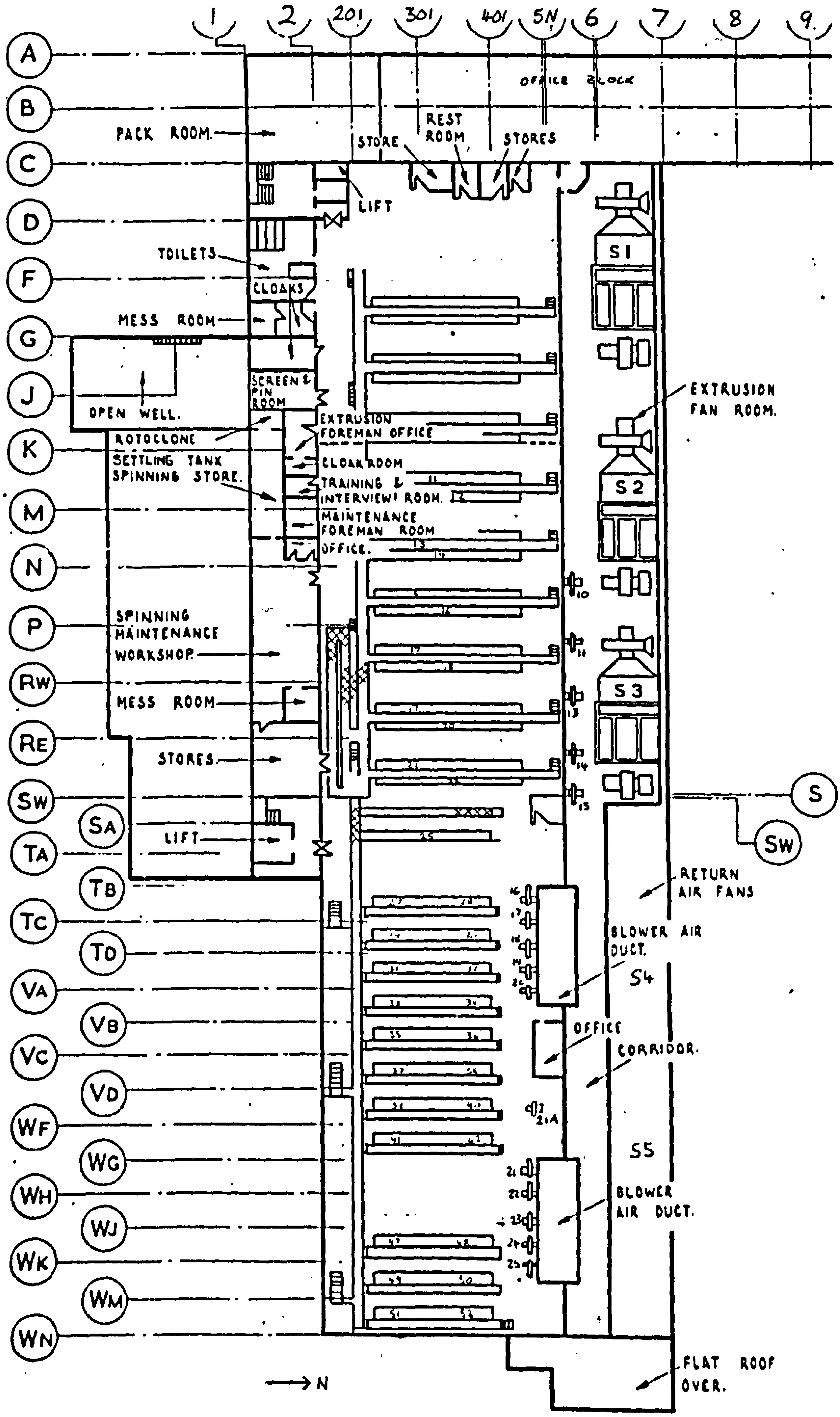

FIGURE 4.5 FIRST FLOOR PLAN (EXTRUSION)
ICI FIBRES, DONCASTER

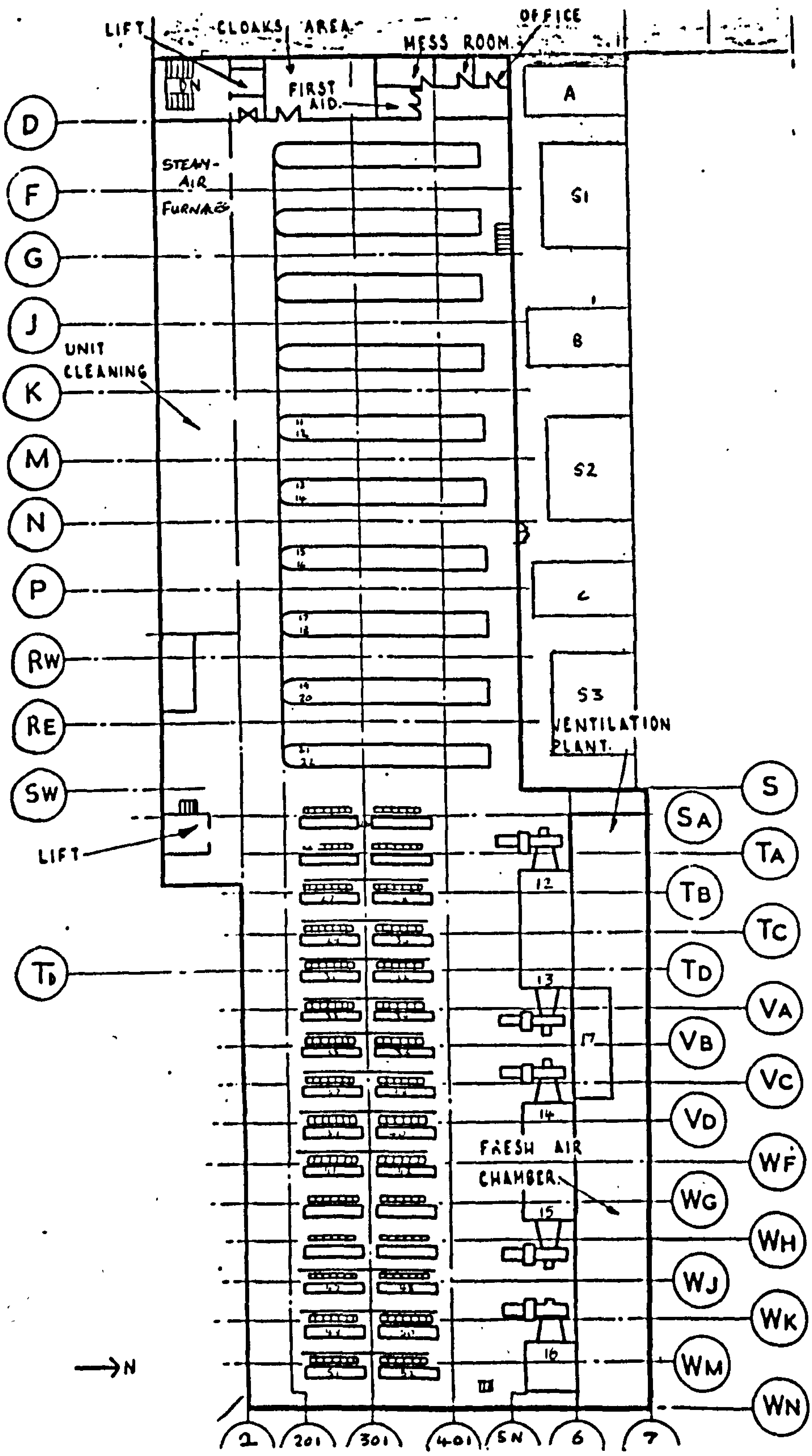

FIGURE 4.6 SECOND FLOOR PLAN (HOPPER FLOOR)
ICI FIBRES, DONCASTER

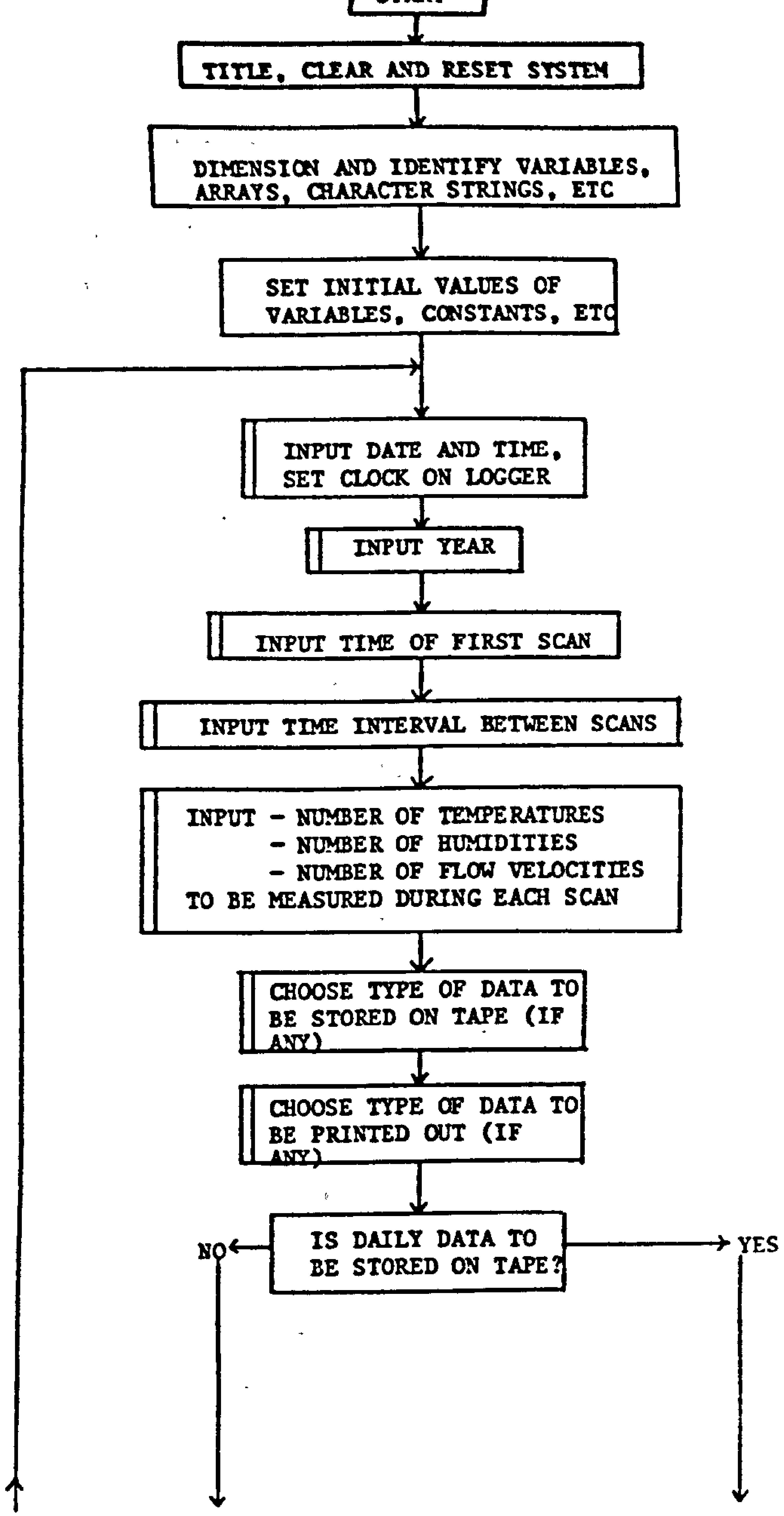

FIGURE 4.7 FLOW CHART FOR "FANLOG" COMPUTER PROGRAM

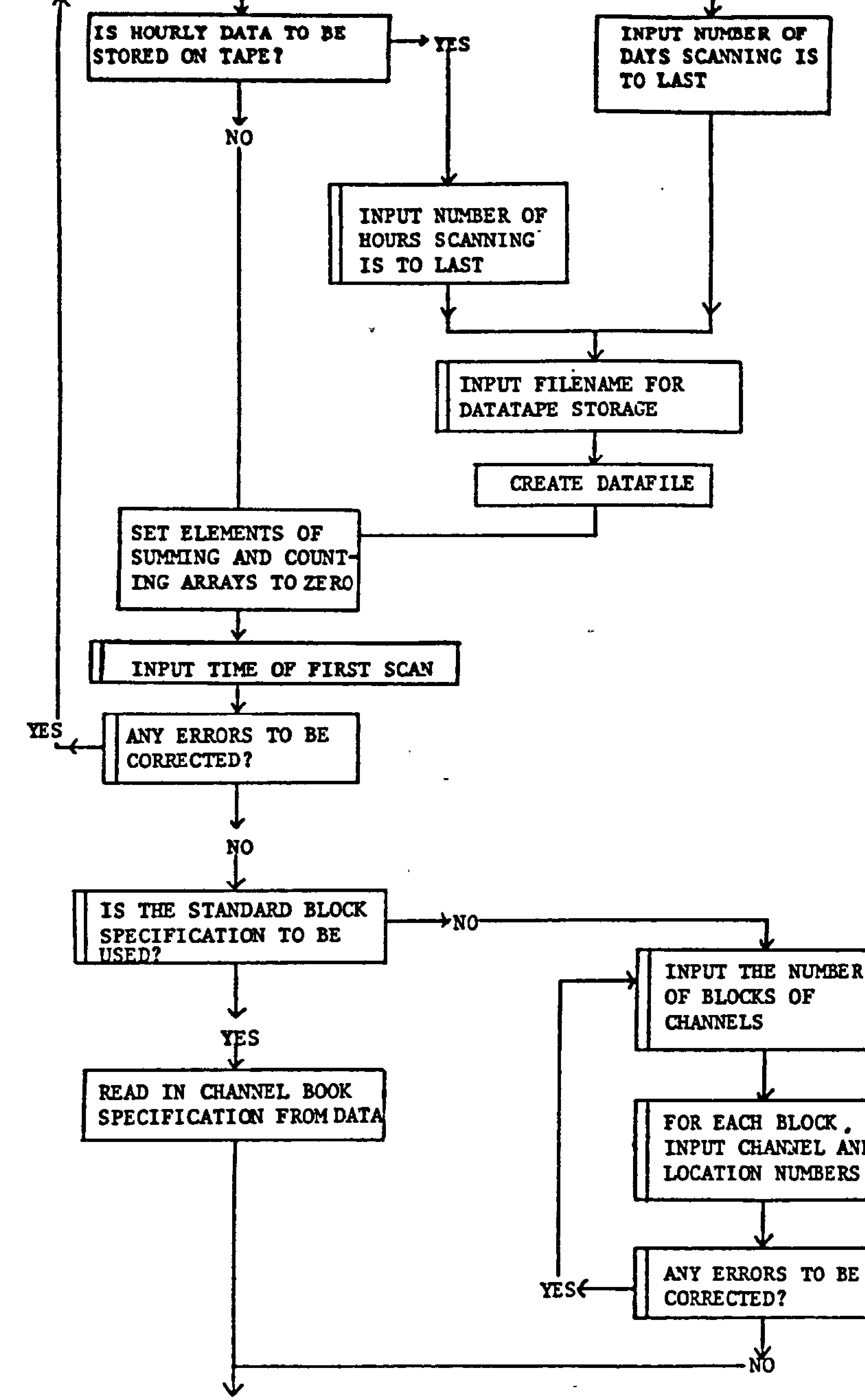

FIGURE 4.7 (continued)

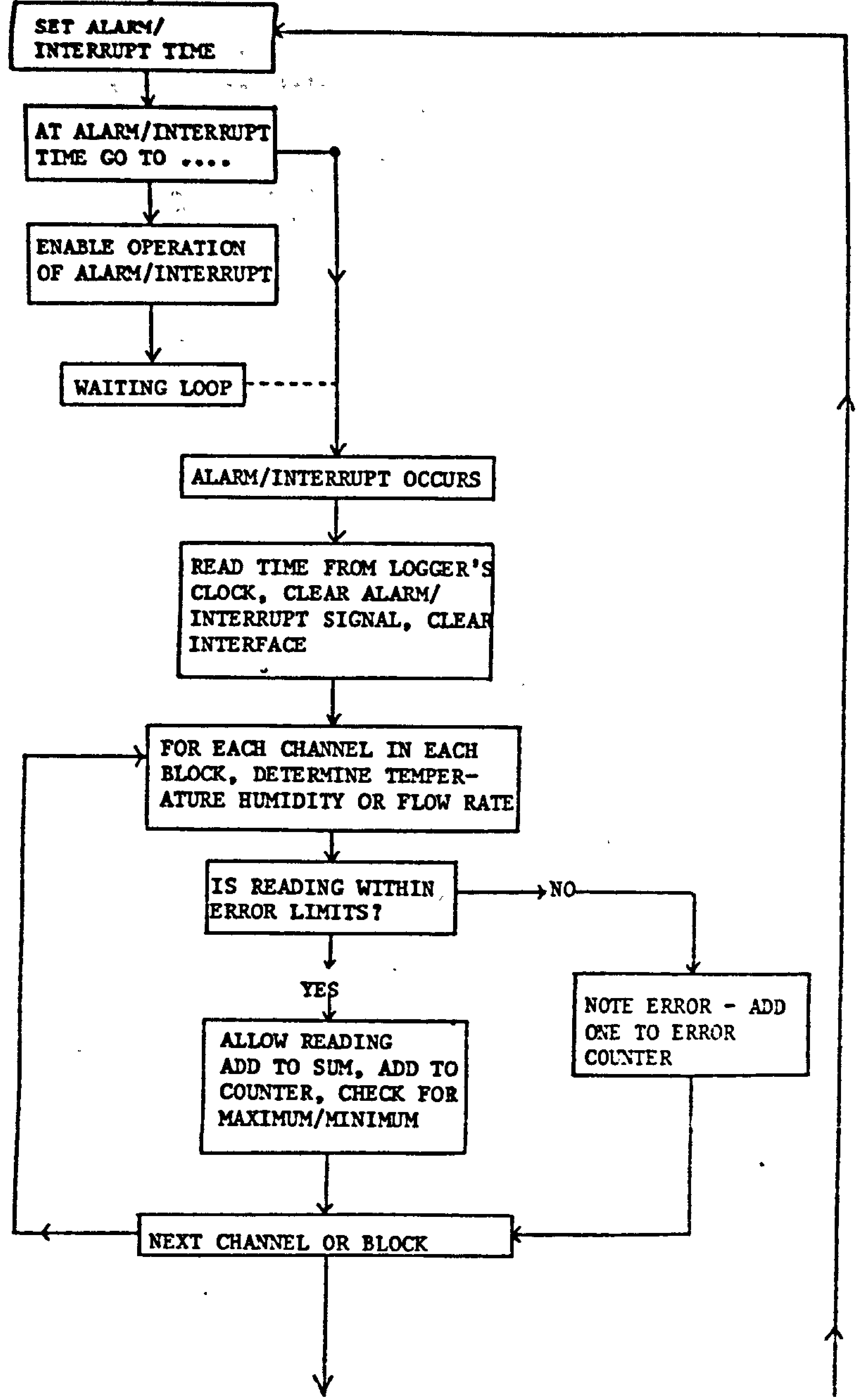

FIGURE 4.7 (continued)

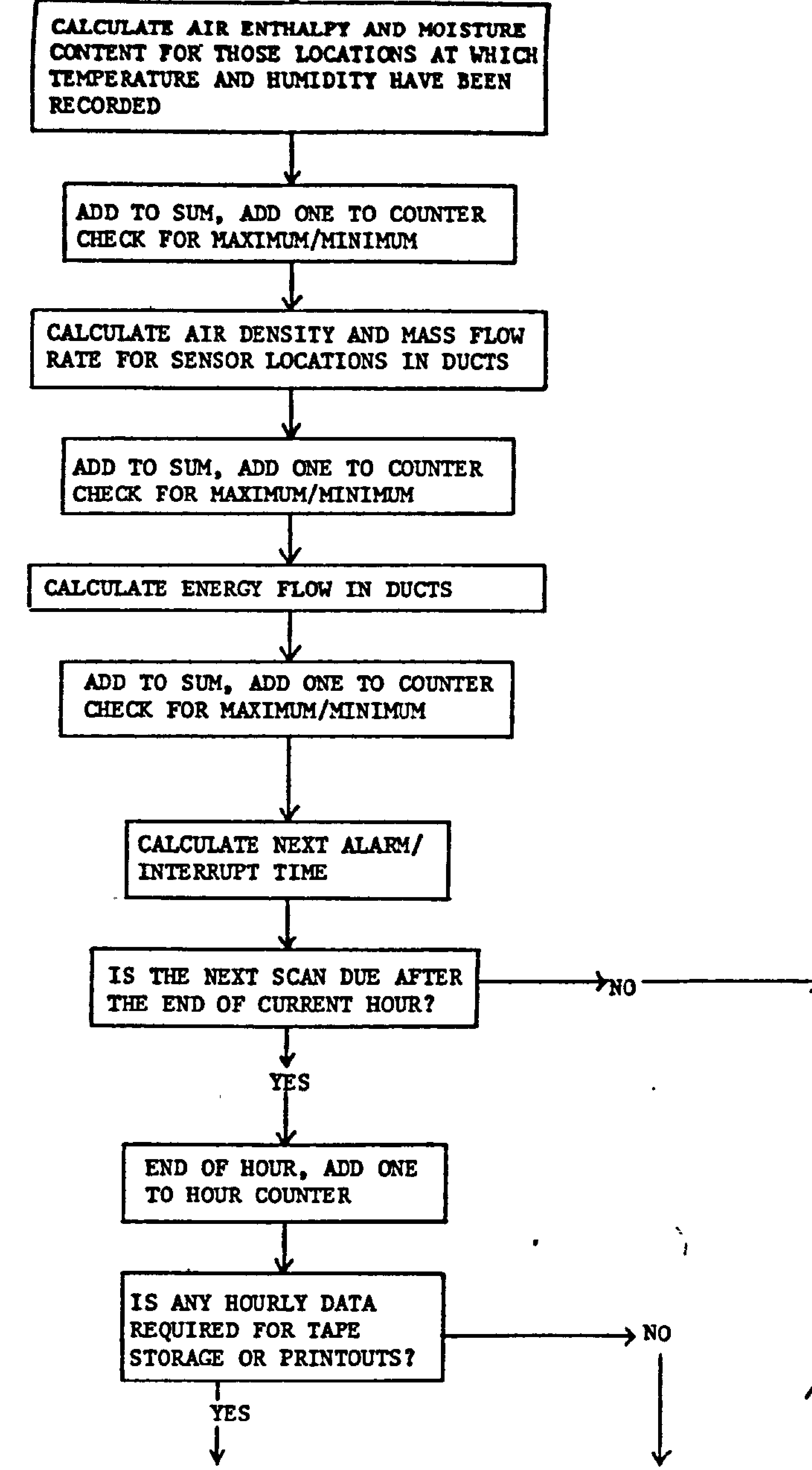

FIGURE 4.7 (continued)

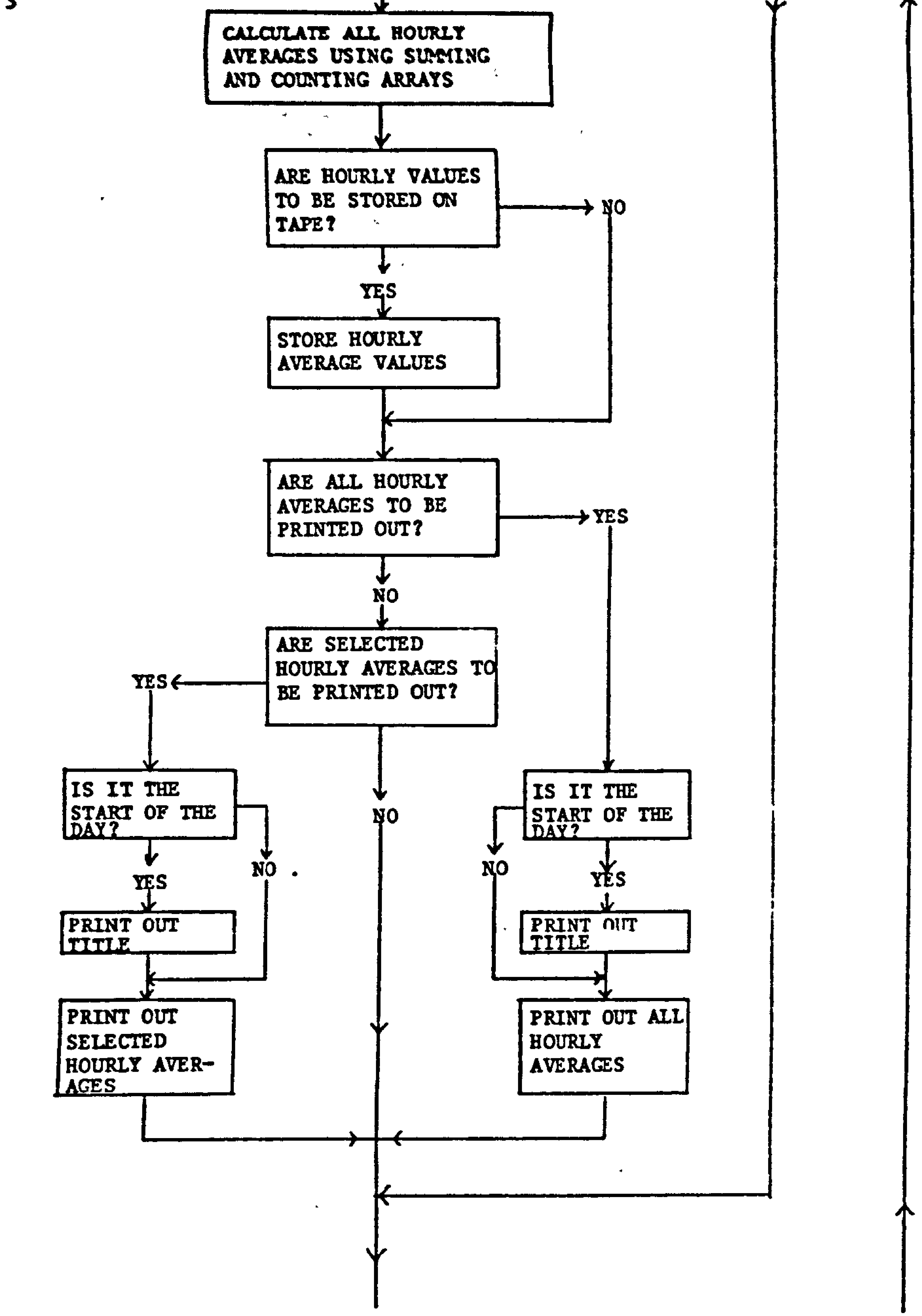

FIGURE 4.7 (continued)

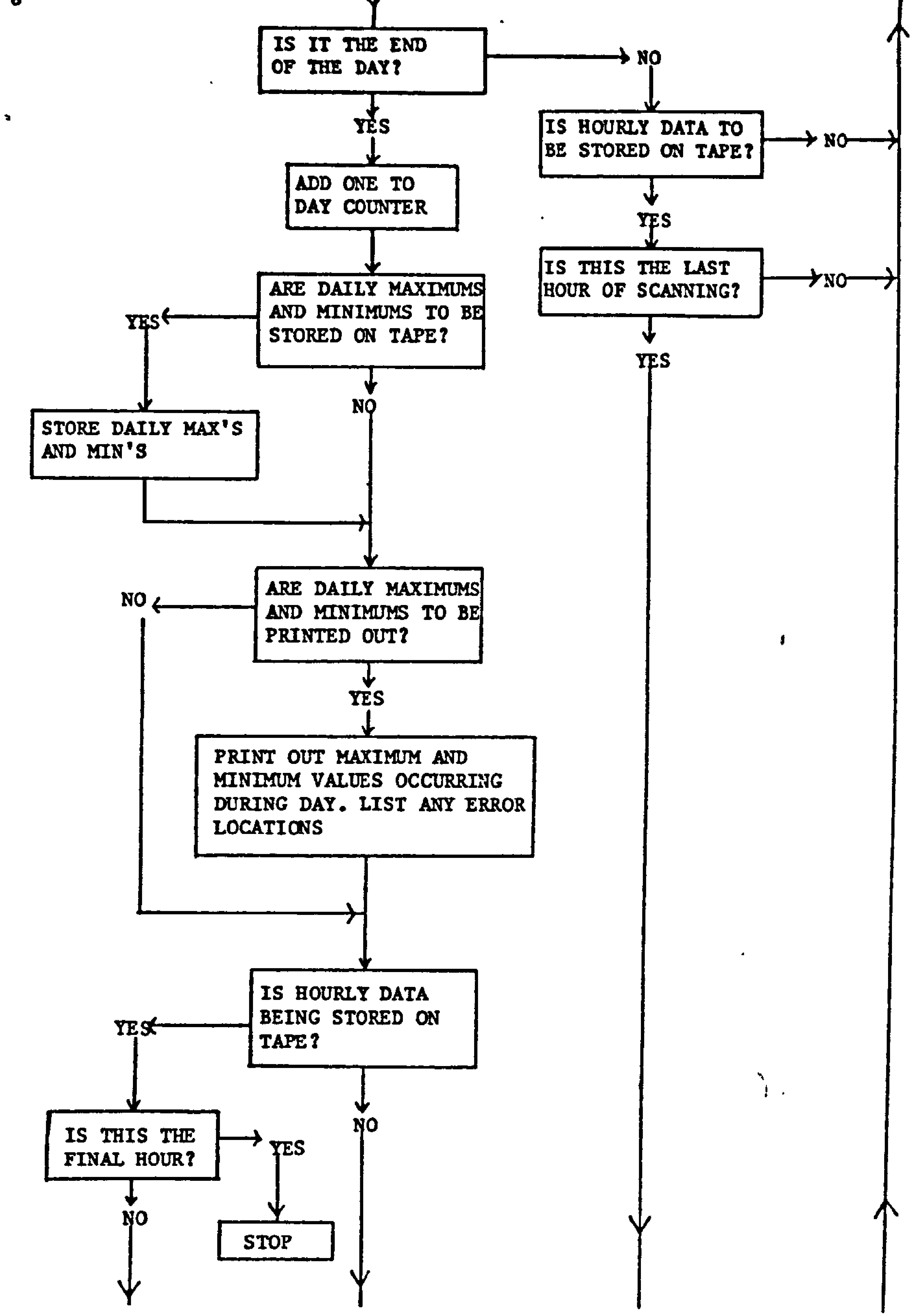

FIGURE 4.7 (continued)

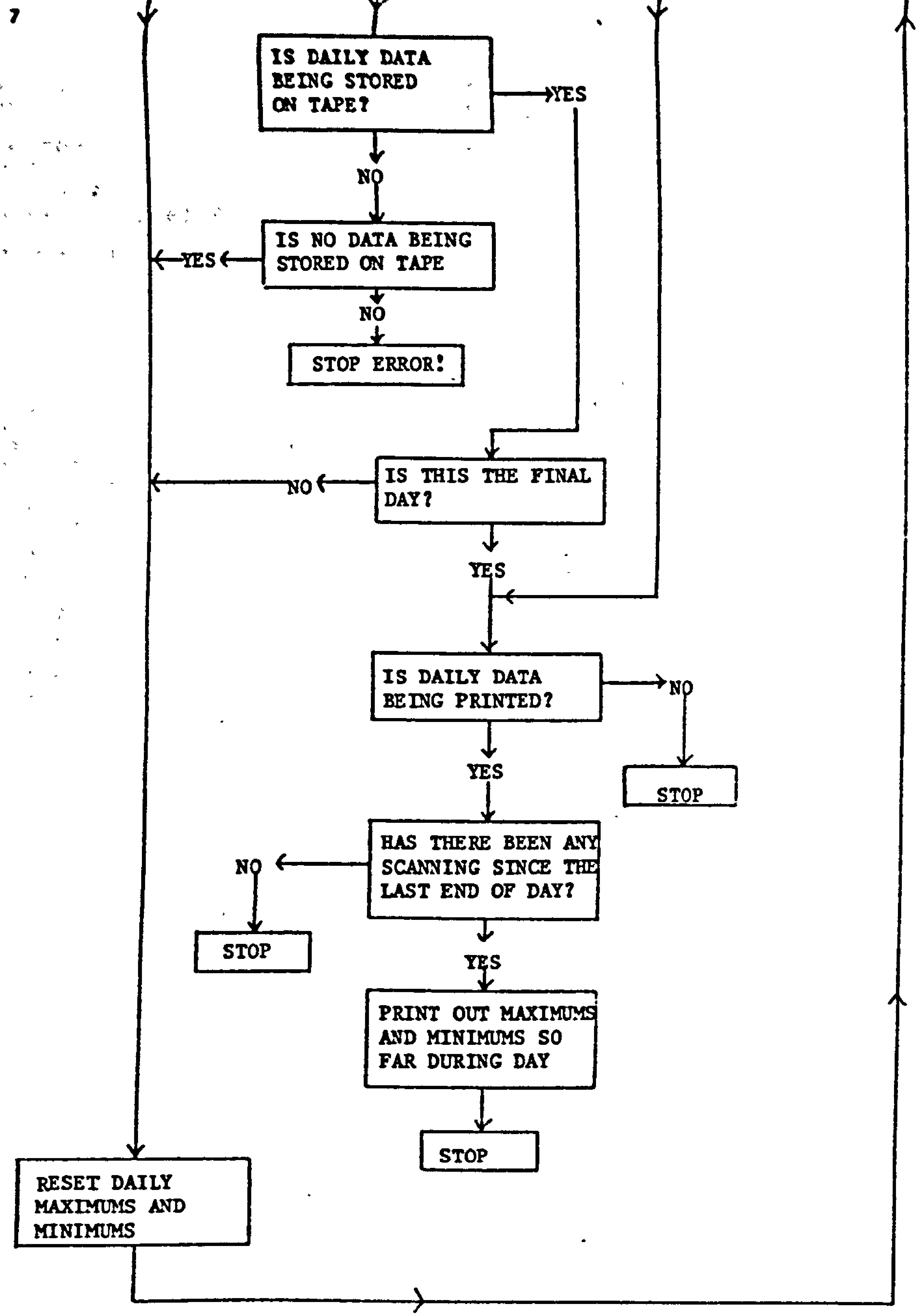

FIGURE 4.7 (final sheet)

```
INPUT DATE AND TIME-
 DD,MM,hh,mm,ss
?
01.09,11.15,00
INPUT YEAR-  yyyy
?
1983
INPUT TIME INTERVAL BETWEEN
SCANS-  hh,mm,ss
?
00.10.00
HOW MANY TEMPERATURE MEASUREMENT
S ?
?
27
HOW MANY HUMIDITY MEASUREMENTS?
?
16
HOW MANY VELOCITY MEASUREMENTS?
?
1
IF ANY DATA IS TO BE STORED ON
TAPE ENSURE THAT THE CORRECT TAP
E IS IN POSITION
WHAT DATA IS TO BE STORED ON
TAPE. CHOOSE BETWEEN...
1=DAILY MAXS AND MINS
2=HOURLY AVERAGES
3=NO TAPE STORAGE
?
2
WHAT DATA IS TO BE PRINTED ?
1=DAILY MAXS AND MINS
2=HOURLY AVERAGES
3=SELECTED HOURLY AVERAGES
4=NO DATA PRINTED
5=1+2 ABOVE
6=1+3 ABOVE
?
6
HOW MANY HOURS IS SCANNING TO
LAST ?
?
100
INPUT FILENAME FOR DATA STORAGE
?
01/09+
INPUT FIRST SCAN TIME-
 hh,mm,ss
?
11.30.00
ANY ERRORS TO CORRECT IN PREVIOU
S INPUT SECTION?(Y/N)
?
N
IS STANDARD SENSOR LAYOUT TO BE
USED?  INPUT Y/N
?
Y
NEXT SCAN AT 11hours 30minutes
00seconds (BST)
```

FIGURE 4.8 EXAMPLE "START-UP" SEQUENCE FOR FANLOG LOGGING PROGRAM (OPERATOR INPUTS UNDERLINED)

```
10 ! A.C.PITTS  LOGGING/ANALYSIS
  PROGRAM
20 ! FANLOG
30 CLEAR
40 RESET 7
50 CLEAR 709
60 RESUME 7
70 OPTION BASE 1
80 ! CHARACTER STRINGS
90 ! D$'s(1-5) = DATE & TIME
100 DIM D1$[2]
110 DIM D2$[2]
120 DIM D3$[2]
130 DIM D4$[2]
140 DIM D5$[2]
150 ! C$'s(1-3) = NEXT SCAN TIME
160 DIM C1$[2]
170 DIM C2$[2]
180 DIM C3$[2]
190 ! B$'s(1-3) = INTERVAL
200 DIM B1$[2]
210 DIM B2$[2]
220 DIM B3$[2]
230 DIM D6$[6] ! FILENAME FOR DA
TA STORAGE
240 DIM D7$[14] ! TIME AT START
OF SCAN
250 DIM D8$[1]
260 DIM Q$[1]
270 DIM Y$[4] ! YEAR
280 ! NUMBER ARRAYS
290 DIM A(7) ! DUCT c.s.a.'s
300 ! S's(1-6+9) =SUM ARRAYS FOR
 HOURLY AVERAGES
310 DIM S1(27)
320 DIM S2(16)
330 DIM S3(7)
340 DIM S4(16)
350 DIM S9(16)
360 DIM S5(7)
370 DIM S6(7)
380 ! N's(1-6) =NUMBER OF VALID
READINGS ADDED TO SUM DURING HOU
R
390 DIM N1(27)
400 DIM N2(16)
410 DIM N3(7)
420 DIM N4(16)
430 DIM N5(7)
440 DIM N6(7)
450 ! C1-C4 RELATE TO BLOCKS OF
CHANNELS TO BE SCANNED-SEE LATER

460 DIM C1(30)
470 DIM C2(30)
480 DIM C3(30)
490 DIM C4(30)
500 DIM C8(27) ! =1 IF ANY ERROR
S AT LOCATION DURING SCAN/=0 IF
NO ERRORS
510 DIM P(27) ! TEMPERATURES
520 DIM H(16) ! HUMIDITIES
530 DIM E(16) ! AIR ENTHALPIES
540 DIM W(16) ! MOISTURE CONTENT
S
550 DIM V(7) ! FLOW VELOCITIES
560 DIM D(7) ! AIR DENSITIES
570 DIM M(7) ! MASS FLOW IN DUCT
S
580 DIM Q(7) ! ENERGY FLOW IN DU
CTS
590 DIM C9(27) ! COUNTS NO. OF E
RRORS AT LOCATION IN DAY
600 ! Y's(1,2,3,4,5,6,9) =MAXIMU
M VALUES DURING DAY
610 DIM Y1(27)
620 DIM Y2(16)
630 DIM Y3(7)
640 DIM Y4(16)
650 DIM Y9(16)
660 DIM Y5(7)
670 DIM Y6(7)
680 ! Z's(1,2,3,4,5,6,9) =MINIMU
M VALUES DURING DAY
690 DIM Z1(27)
700 DIM Z2(16)
710 DIM Z3(7)
720 DIM Z4(16)
730 DIM Z9(16)
740 DIM Z5(7)
750 DIM Z6(7)
760 ! D(1,2,3,4,5)=1st SCAN INDI
CATORS
770 DIM D1(27)
780 DIM D2(16)
790 DIM D3(7)
800 DIM D4(16)
810 DIM D5(7)
820 ! LOOP COUNTERS USED ARE K1,
L1,L2,L3,L4   ALSO USED L8,L
9
830 ! SET ERROR COUNTER TO ZERO
840 FOR L1=1 TO 27
850 C9(L1)=0
860 NEXT L1
870 ! SET MAX'S & MIN's TO ZERO-
REQUIRED ONLY TO AVOID NULL
DATA ERRORS
880 GOSUB 5400
890 GOSUB 7400
900 ! VALUES OF DUCT c s a. s
910 A(1)=.3726
920 A(2)=.3726
930 A(3)=.81
940 A(4)=.6916
950 A(5)=.2244
960 A(6)=.72
970 A(7)=.272
980 T9=0 ! NUMBER OF SCANS SINCE
  START/END OF DAY
990 I7=0
1000 T8=0
1010 B7=0 ! TIME DEFAULT VALUE
1020 B1=0
1030 B2=0
1040 B3=0
1050 DISP "INPUT DATE AND TIME-
                DD,MM,hh,mm,ss"
 @ INPUT D1$,D2$,D3$,D4$,D5
$
1060 OUTPUT 709 ;"TD";D1$,D2$,D3
$,D4$,D5$;"TD"
1070 DISP "INPUT YEAR-  yyyy" @
INPUT Y$
1090 DISP "INPUT TIME INTERVAL B
ETWEEN      SCANS-  hh,mm,ss
" @ INPUT B1$,B2$,B3$
1100 DISP "HOW MANY TEMPERATURE
MEASUREMENTS ?" @ INPUT I2
1110 DISP "HOW MANY HUMIDITY MEA
SUREMENTS?" @ INPUT I3
1120 DISP "HOW MANY VELOCITY MEA
SUREMENTS?" @ INPUT I4
1130 DISP "IF ANY DATA IS TO BE
STORED ON  TAPE ENSURE THAT
 THE CORRECT TAPE IS IN POS
ITION"
1140 DISP "WHAT DATA IS TO BE ST
ORED ON    TAPE  CHOOSE BET
WEEN..."
1150 DISP "1=DAILY MAXS AND MINS
               2=HOURLY AVERAGE
S                    3=NO TAPE S
TORAGE"
1160 INPUT B4
1170 DISP "WHAT DATA IS TO BE PR
INTED ?    1=DAILY MAXS AND
 MINS              2=HOURLY AV
ERAGES"
1180 DISP "3=SELECTED HOURLY AVE
RAGES         4=NO DATA PRINTE
D"
1190 DISP "5=1+2 ABOVE
               6=1+3 ABOVE"
1200 INPUT B5
1210 IF B4=1 THEN 1240
1220 IF B4=2 THEN 1280
1230 GOTO 1340
1240 DISP "HOW MANY DAYS IS SCAN
NING TO    LAST ?"
1250 INPUT B6
1260 I5=1302
1270 GOTO 1310
1280 DISP "HOW MANY HOURS IS SCA
NNING TO   LAST ?"
1290 INPUT B6
1300 I5=496
1310 DISP "INPUT FILENAME FOR DA
TA STORAGE"
1320 INPUT D6$
1330 CREATE D6$,B6,I5
1340 ! CONTINUE
1350 GOSUB 5610 ! ZERO SOME ARRA
YS
1355 DISP "INPUT FIRST SCAN TIME
 -                hh,mm,ss" @ INP
UT C1$,C2$,C3$
1360 CLEAR
1370 DISP "ANY ERRORS TO CORRECT
  IN PREVIOUS INPUT SECTION?
 (Y/N)" @ INPUT Q$
1380 IF Q$="Y" THEN 1050
1390 CLEAR
1400 DISP "IS STANDARD SENSOR LA
YOUT TO BE USED^  INPUT Y/N
" @ INPUT Q$
1410 IF Q$="Y" THEN 7320
1420 IF Q$="N" THEN 1440
1430 GOTO 1400
1440 DISP "HOW MANY BLOCKS OF SE
NSORS IS   THE SCAN TO BE S
PLIT INTO?"
1450 DISP "[ EACH BLOCK MUST CON
TAIN ONLY  ONE TYPE OF SENS
OR AND THE "
1460 DISP "CHANNELS MUST BE CONS
ECUTIVELY  NUMBERED. IN ADD
ITION THE BLOCKS"
1470 DISP "MUST BE ARRANGED FOR
CORESPONDING LOCATION NUMBE
RS FOR THE DIFFERENT SENSOR
S ]"
1480 INPUT I6
1490 ! INPUT DATA FOR EACH BLOCK
 OF SENSORS TO BE SCANNED
1500 FOR L1=1 TO I6
1510 CLEAR
1520 DISP "BLOCK ";L1
1530 DISP "WHAT TYPE OF SENSOR "
1540 DISP "  1. P.R.T
                 2. HUMIDITY SE
NSOR                  3. VELOCI
TY SENSOR"
1550 INPUT C1(L1)
1560 DISP "FIRST CHANNEL NUMBER=
 " @ INPUT C2(L1)
1570 DISP "LAST CHANNEL NUMBER=
 " @ INPUT C3(L1)
1580 DISP "LOCATION NUMBER CORRE
SPONDING TO FIRST CHANNEL N
UMBER= " @ INPUT C4(L1)
1590 NEXT L1
1600 CLEAR
1610 DISP "ANY ERRORS TO CORRECT
  IN PREVIOUS INPUT SECTION?
 (Y/N)" @ INPUT Q$
1620 IF Q$="Y" THEN 1440
1630 CLEAR
1640 ! SET UP INTERRUPT AND ENAB
LE IT
```

```
1650 OUTPUT 709 ;"TD"
1660 ON INTR 7 GOTO 1800
1670 OUTPUT 709 ;"SE4TA";C1$;C2$
     ;C3$;"TD"
1680 ENABLE INTR 7,8
1690 ! 'WAITING FOR INTERRUPT' L
     OOP
1700 DISP "NEXT SCAN AT ";C1$;"h
     ours ";C2$;"minutes ";C3$;"
     seconds (BST)"
1710 FOR L1=1 TO 60
1720 WAIT 500
1730 NEXT L1
1740 CLEAR
1750 B7=B1*7200+B2*120+B3*2
1760 IF B7=0 THEN 1710
1770 FOR L1=1 TO B7
1780 WAIT 500
1790 NEXT L1
1800 ! INTERRUPT
1810 OFF INTR 7
1820 STATUS 7,1 ; A
1830 OUTPUT 709 ;"TD"
1840 ENTER 709 ; D7$
1850 CLEAR 709
1860 T3=VAL(D7$[7]) ! =HOUR OF D
     AY
1870 FOR L1=1 TO I6
1880 IF C1(L1)=1 THEN 1940
1890 IF C1(L1)=2 THEN 2470
1900 IF C1(L1)=3 THEN 2740
1910 STOP
1920 ! CHECK FOR TYPE OF SENSOR
     BLOCK BY BLOCK
1930 ! PRT TEMPERATURE READING
1940 C6=C4(L1)-1 ! LOCATION NO.
1950 FOR L2=C2(L1) TO C3(L1)
1960 C6=C6+1
1970 K2=L2
1980 ! OHM FIND RESISTANCE OF PR
     T
1990 OUTPUT 709 USING "AA,DDD,A,
     DDD" ; "AC",K2;",";K2+10
2000 OUTPUT 709 ;"VCOVR5VN1VA1VF
     1VD5VS0VW0VT3"
2010 ENTER 709 ; O5
2020 IF ABS(O5)<=.5 THEN 2070
2030 K3=0
2040 C8(C6)=1
2050 C9(C6)=C9(C6)+1
2060 GOTO 2440
2070 O6=1
2080 O7=O6
2090 OUTPUT 709 ;"VC";O6
2100 OUTPUT 709 ;"VT3"
2110 ENTER 709 ; K3
2120 OUTPUT 709 ;"VC0"
2130 O6=MAX(1,MIN(3,O6+(ABS(K3-O
     5)<.01)+(ABS(K3-O5)<.001)-(
     ABS(K3))=11)))
2140 IF O6<>O7 THEN 2080
2150 K3=(K3-O5)*10^(6-O6)
2160 O7=K3/100
2170 IF O7<1 THEN 2240
2180 IF O7>3 THEN 2310
2190 O3=3367.82144098
2200 O4=13065764.8637
2210 O5=-1723543.60565
2220 P(C6)=IP((O3-SQR(O4+O5*O7))
     *100+.5)/100
2230 GOTO 2290
2240 O2=-241.996759172
2250 O3=222.560617915
2260 O4=25.2488238315
2270 O5=-5.81268262456
2280 P(C6)=IP((O2+O7*(O3+O7*(O4+
     O7*O5)))*100+.5)/100
2290 IF P(C6)>-20 AND P(C6)<60 T
     HEN 2340 ! WITHIN RANGE OK
2300 ! ERROR
2310 C8(C6)=1
2320 C9(C6)=C9(C6)+1
2330 GOTO 2440
2340 S1(C6)=S1(C6)+P(C6)
2350 N1(C6)=N1(C6)+1
2360 ! FIND IF MAX OR MIN (TEMP)
2370 IF D1(C6)=0 THEN 2410 ! 1st
     VALID SCAN OF DAY
2380 Y1(C6)=MAX(P(C6),Y1(C6))
2390 Z1(C6)=MIN(P(C6),Z1(C6))
2400 GOTO 2440
2410 Y1(C6)=P(C6)
2420 Z1(C6)=P(C6)
2430 D1(C6)=1
2440 NEXT L2
2450 GOTO 3030
2460 ! HUMIDITY READING
2470 C6=C4(L1)-1
2480 FOR L2=C2(L1) TO C3(L1)
2490 C6=C6+1
2500 K2=L2
2510 ! VOLTAGE READING
2520 OUTPUT 709 ;"AC";K2
2530 OUTPUT 709 ;"VR5VN1VA1VF1VD
     5VC0VS0VW0VT3"
2540 ENTER 709 ; K3
2550 H(C6)=IP(K3*1000+.5)/100
2560 IF H(C6)>0 AND H(C6)<100 TH
     EN 2610 ! WITHIN RANGE OK
2570 ! ERROR
2580 C8(C6)=1
2590 C9(C6)=C9(C6)+1
2600 GOTO 2710
2610 S2(C6)=S2(C6)+H(C6)
2620 N2(C6)=N2(C6)+1
2630 ! FIND IF MAX OR MIN (HUM)
2640 IF D2(C6)=0 THEN 2680 ! 1st
     VALID SCAN OF DAY
2650 Y2(C6)=MAX(H(C6),Y2(C6))
2660 Z2(C6)=MIN(H(C6),Z2(C6))
2670 GOTO 2710
2680 Y2(C6)=H(C6)
2690 Z2(C6)=H(C6)
2700 D2(C6)=1
2710 NEXT L2
2720 GOTO 3030
2730 ! VANE ANEMOMETER -FLOW VEL
     OCITY READING
2740 C6=C4(L1)-1
2750 FOR L2=C2(L1) TO C3(L1)
2760 C6=C6+1
2770 V(C6)=0
2780 FOR L3=1 TO 10
2790 K2=L2
2800 ! VOLTAGE READING
2810 OUTPUT 709 ;"AC",K2
2820 OUTPUT 709 ;"VR5VN1VA1VF1VD
     5VC0VS0VW0VT3"
2830 ENTER 709 ; K3
2840 V(C6)=V(C6)+K3
2850 WAIT 650
2860 NEXT L3
2870 V(C6)=IP((V(C6)/10+.034)/.2
     96*100+.5)/100
2880 IF V(C6)>0 AND V(C6)<30 THE
     N 2920
2890 C8(C6)=1
2900 C9(C6)=C9(C6)+1
2910 GOTO 3020
2920 S3(C6)=S3(C6)+V(C6)
2930 N3(C6)=N3(C6)+1
2940 ! FIND IF MAX OR MIN (VEL)
2950 IF D3(C6)=0 THEN 2990 ! 1st
     VALID SCAN OF DAY
2960 Y3(C6)=MAX(V(C6),Y3(C6))
2970 Z3(C6)=MIN(V(C6),Z3(C6))
2980 GOTO 3020
2990 Y3(C6)=V(C6)
3000 Z3(C6)=V(C6)
3010 D3(C6)=1
3020 NEXT L2
3030 NEXT L1
3040 ! ENTHALPY CALCULATION
3050 FOR L3=1 TO I3
3060 IF C8(L3)=1 THEN 3270
3070 F=P(L3)
3080 H=H(L3)
3090 GOSUB 6390
3100 E(L3)=IP(F4*100+.5)/100
3110 W(L3)=X7
3120 S4(L3)=S4(L3)+E(L3)
3130 S9(L3)=S9(L3)+W(L3)
3140 N4(L3)=N4(L3)+1
3150 ! FIND IF MAX OR MIN (ENTH)
3160 IF D4(L3)=0 THEN 3220 ! 1st
     VALID SCAN OF DAY
3170 Y4(L3)=MAX(E(L3),Y4(L3))
3180 Y9(L3)=MAX(W(L3),Y9(L3))
3190 Z4(L3)=MIN(E(L3),Z4(L3))
3200 Z9(L3)=MIN(W(L3),Z9(L3))
3210 GOTO 3270
3220 Y4(L3)=E(L3)
3230 Y9(L3)=W(L3)
3240 Z4(L3)=E(L3)
3250 Z9(L3)=W(L3)
3260 D4(L3)=1
3270 NEXT L3
3280 ! AIR DENSITY AND MASS FLOW
     CALCULATION
3290 FOR L3=1 TO I4
3300 IF C8(L3)=1 THEN 3500
3310 D(L3)=1013.25/((P(L3)+273.1
     6)*(2.87+4.61*L3))
3320 M(L3)=IP(V(L3)*A(L3)*D(L3)*
     1000+.5)/1000
3330 S5(L3)=S5(L3)+M(L3)
3340 N5(L3)=N5(L3)+1
3350 Q(L3)=IP((E(L3)-E(8))*M(L3)
     *100+.5)/100
3360 S6(L3)=S6(L3)+Q(L3)
3370 N6(L3)=N6(L3)+1
3380 ! FIND IF MAX OR MIN VALUE
3390 IF D5(L3)=0 THEN 3450 ! 1st
     VALID SCAN OF DAY
3400 Y5(L3)=MAX(M(L3),Y5(L3))
3410 Z5(L3)=MIN(M(L3),Z5(L3))
3420 Y6(L3)=MAX(Q(L3),Y6(L3))
3430 Z6(L3)=MIN(Q(L3),Z6(L3))
3440 GOTO 3500
3450 Y5(L3)=M(L3)
3460 Z5(L3)=M(L3)
3470 Y6(L3)=Q(L3)
3480 Z6(L3)=Q(L3)
3490 D5(L3)=1
3500 NEXT L3
3510 FOR L1=1 TO 27
3520 C8(L1)=0
3530 NEXT L1
3540 T9=T9+1
3550 ! CALCULATE NEXT SCAN TIME
3560 K4=VAL(C1$)
3570 K5=VAL(C2$)
3580 K6=VAL(C3$)
3590 B1=VAL(B1$)
3600 B2=VAL(B2$)
3610 B3=VAL(B3$)
3620 K7=K4+B1
3630 K8=K5+B2
3640 K9=K6+B3
3650 IF K9>=60 THEN 3840
3660 IF K8>=60 THEN 3870
3670 IF K7>=24 THEN 3900
3680 IF K7>=10 THEN 3720
3690 C1$="0"
3700 C1$[2,2]=VAL$(K7)
3710 GOTO 3730
3720 C1$=VAL$(K7)
3730 IF K8>=10 THEN 3770
3740 C2$="0"
3750 C2$[2,2]=VAL$(K8)
```

```
3760 GOTO 3780
3770 C2$=VAL$(K8)
3780 IF K9>=10 THEN 3820
3790 C3$="0"
3800 C3$[2,2]=VAL$(K9)
3810 GOTO 3830
3820 C3$=VAL$(K9)
3830 GOTO 3930
3840 K9=K9-60
3850 K8=K8+1
3860 GOTO 3650
3870 K8=K8-60
3880 K7=K7+1
3890 GOTO 3660
3900 K7=K7-24
3910 GOTO 3670
3920 ! TEST FOR END OF HOUR/DAY/
     SCANNING
3930 IF K7=T3 THEN 1630
3940 ! END OF HOUR
3950 I7=I7+1
3960 IF B4=2 THEN 4040
3970 IF B5=2 THEN 4040
3980 IF B5=3 THEN 4040
3990 IF B5=5 THEN 4040
4000 IF B5=6 THEN 4040
4010 ! NO HOURLY DATA REQUIRED
4020 GOTO 4280
4030 ! CALCULATE AVERAGES
4040 GOSUB 4720
4050 IF B4=2 THEN 4080
4060 GOTO 4110
4070 ! STORE HOURLY AVERAGES ON
     TAPE
4080 ASSIGN# 1 TO D6$
4090 PRINT# 1,I7 ; D7$,S1(),S2()
     ,S3(),S6()
4100 ASSIGN# 1 TO *
4110 IF B5=2 THEN 4170
4120 IF B5=3 THEN 4230
4130 IF B5=5 THEN 4170
4140 IF B5=6 THEN 4230
4150 ! NO PRINTOUT REQUIRED
4160 GOTO 4280
4170 IF T3>0 THEN 4210
4180 ! PRINT HOURLY AVERAGES TIT
     LE
4190 GOSUB 5020
4200 ! PRINT HOURLY AVERAGES
4210 GOSUB 5180
4220 GOTO 4280
4230 IF T3>0 THEN 4270
4240 ! PRINT SELECTED HOURLY AVE
     RAGES TITLE
4250 GOSUB 6890
4260 ! PRINT SELECTED HOURLY AVE
     RAGES
4270 GOSUB 7080
4280 IF K7=0 AND T3=23 THEN 4330
4290 IF B4=2 THEN 4310
4300 GOTO 1630
4310 IF I7>=B6 THEN 4530
4320 GOTO 1630
4330 T8=T8+1
4340 IF B4=1 THEN 4430
4350 IF B5=1 THEN 4480
4360 IF B5=5 THEN 4480
4370 IF B5=6 THEN 4480
4380 ! NO DAILY DATA REQUIRED
4390 T9=0
4400 GOSUB 7400
4405 GOSUB 5400
4410 GOTO 4500
4420 ! STORE DAILY MAXS AND MINS
     ON TAPE
4430 ASSIGN# 1 TO D6$
4440 PRINT# 1,T8 ; D7$,Y1(),Z1()
     ,Y2(),Z2(),Y3(),Z3(),Y4(),Z
     4(),Y5(),Z5(),Y6(),Z6()
4450 ASSIGN# 1 TO *
4460 GOTO 4350
4470 ! PRINT DAILY MAXS AND MINS
4480 PRINT
4490 GOSUB 5950
4500 IF B4=1 THEN 4540
4510 IF B4=2 THEN 4560
4520 IF B4=3 THEN 4620
4530 STOP
4540 IF T8>=B6 THEN 4670
4550 GOTO 4680
4560 IF I7>=B6 THEN 4530
4570 GOTO 4680
4580 ! END OF SCANNING
4590 IF B5=1 THEN 4670
4600 IF B5=5 THEN 4670
4610 IF B5=6 THEN 4630
4620 STOP
4630 IF T9>0 THEN 4660
4640 STOP
4650 ! PRINT MAXS AND MINS
4660 GOSUB 5950
4670 STOP
4680 GOSUB 7400
4685 GOSUB 5400
4690 GOTO 1630
4700 ! END OF HOUR
4710 ! CALCULATE HOURLY AVERAGES
     ROUNDING
4720 FOR L3=1 TO I4
4730 IF N1(L3)=0 THEN 4750
4740 S1(L3)=IP(S1(L3)/N1(L3)*10+
     .5)/10
4750 IF N2(L3)=0 THEN 4770
4760 S2(L3)=IP(S2(L3)/N2(L3)*10+
     .5)/10
4770 IF N3(L3)=0 THEN 4790
4780 S3(L3)=IP(S3(L3)/N3(L3)*100
     +.5)/100
4790 IF N4(L3)=0 THEN 4820
4800 S4(L3)=IP(S4(L3)/N4(L3)*100
     +.5)/100
4810 S9(L3)=IP(S9(L3)/N4(L3)*100
     0+.5)
4820 IF N5(L3)=0 THEN 4840
4830 S5(L3)=IP(S5(L3)/N5(L3)*100
     0+.5)/1000
4840 IF N6(L3)=0 THEN 4860
4850 S6(L3)=IP(S6(L3)/N6(L3)*100
     +.5)/100
4860 NEXT L3
4870 FOR L3=I4+1 TO I3
4880 IF N1(L3)=0 THEN 4900
4890 S1(L3)=IP(S1(L3)/N1(L3)*10+
     .5)/10
4900 IF N2(L3)=0 THEN 4920
4910 S2(L3)=IP(S2(L3)/N2(L3)*10+
     .5)/10
4920 IF N4(L3)=0 THEN 4950
4930 S4(L3)=IP(S4(L3)/N4(L3)*100
     +.5)/100
4940 S9(L3)=IP(S9(L3)/N4(L3)*100
     0+.5)
4950 NEXT L3
4960 FOR L3=I3+1 TO I2
4970 IF N1(L3)=0 THEN 4990
4980 S1(L3)=IP(S1(L3)/N1(L3)*10+
     .5)/10
4990 NEXT L3
5000 RETURN
5010 ! AVERAGE HOURLY READINGS T
     ITLE
5020 PRINT
5030 PRINT "LOCATIONS OF SENSOR
     POSITIONS     (no's 1-7 are
     for ducts around 27/28 mach
     ine)"
5040 PRINT
5050 PRINT "1=S Plant Supply Duc
     t              2=S Plant Extra
     ct Duct"
5060 PRINT "3=S Plant Underfl. S
     upply Duct   4=Extrusion Sup
     ply-Large Duct "
5070 PRINT "5=Extrusion Supply-S
     mall Duct    6=Extrusion Ext
     ract Duct          7=Blower A
     ir Duct"
5080 PRINT "8=Outside (East end
     Hopper (1 ) 9=Spin Doff Tc4
     01"
5090 PRINT "10=Extrusion Tc301"
5100 PRINT "11=Extrusion Catwalk
      Tc301        12=Hopper Floor
      Tb301"
5110 PRINT "13=Drawtwist C Bank
     Lag Area        14=Drawtwist C4
     -C5                 15=Drawtwi
     st C7-C8"
5120 PRINT "16=Drawtwist C10-C11
                   17=Spin Doff P4
     "
5130 PRINT "18=Spin Doff Asp. Tb
     301            19=Spin Doff Vb
     301                   20=Extrusi
     on P4"
5140 PRINT "21=Extrusion Asp. Tb
     301           22=Extrusion Vb
     301"
5150 PRINT "23=Extrusion Catwalk
      P4(m/c17)  24=Extrusion Ca
     twalk Asp. Tb301"
5160 PRINT "25=Extrusion Catwalk
      Vb301          26=Hopper Fl P4
                         27=Hopper
     Fl.Vb301"
5170 RETURN
5180 PRINT "HOURLY AVERAGES "
5190 T5=VAL(D7$[1])
5200 T6=VAL(D7$[4])
5210 PRINT
5220 PRINT "HOUR";T3;" ON";T5;"/
     ";T6;"/";Y$
5230 PRINT
5240 PRINT "POS     TEMP     P.H.
     FLOW  ENERGY"
5250 FOR L1=1 TO I4
5260 PRINT USING 5340 ; L1,S1(L1
     ),S2(L1),S3(L1),S6(L1)
5270 NEXT L1
5280 FOR L1=I4+1 TO I3
5290 PRINT USING 5350 ; L1,S1(L1
     ),S2(L1)
5300 NEXT L1
5310 FOR L1=I3+1 TO I2
5320 PRINT USING 5360 ; L1,S1(L1
     )
5330 NEXT L1
5340 IMAGE DDD,3X,DD.D,3X,DD.D,3
     X,DD.D,X,DDDD.D
5350 IMAGE DDD,3X,DD.D,3X,DD.D,1
     5X
5360 IMAGE DDD,3X,DD.D,22X
5370 GOSUB 5620 ! ZERO SOME ARRA
     YS
5380 RETURN
5390 ! SET MAX AND MIN ZERO
5400 FOR L1=1 TO 27
5410 Y1(L1)=0
5420 Z1(L1)=0
5430 NEXT L1
5440 FOR L1=1 TO 16
5450 Y2(L1)=0
5460 Z2(L1)=0
5470 Y4(L1)=0
5480 Z4(L1)=0
5490 Y9(L1)=0
5500 Z9(L1)=0
5510 NEXT L1
5520 FOR L1=1 TO 7
5530 Y3(L1)=0
5540 Z3(L1)=0
```

```
5550 Y5(L1)=0
5560 Z5(L1)=0
5570 Y6(L1)=0
5580 Z6(L1)=0
5590 NEXT L1
5600 RETURN
5610 ! ZERO ARRAYS
5620 FOR L1=1 TO I4
5630 S1(L1)=0
5640 N1(L1)=0
5650 S2(L1)=0
5660 N2(L1)=0
5670 S3(L1)=0
5680 N3(L1)=0
5690 S4(L1)=0
5700 S9(L1)=0
5710 N4(L1)=0
5720 S5(L1)=0
5730 N5(L1)=0
5740 S6(L1)=0
5750 N6(L1)=0
5760 C8(L1)=0
5770 NEXT L1
5780 FOR L1=I4+1 TO I3
5790 S1(L1)=0
5800 N1(L1)=0
5810 S2(L1)=0
5820 N2(L1)=0
5830 S4(L1)=0
5840 S9(L1)=0
5850 N4(L1)=0
5860 C8(L1)=0
5870 NEXT L1
5880 FOR L1=I3+1 TO I2
5890 S1(L1)=0
5900 N1(L1)=0
5910 C8(L1)=0
5920 NEXT L1
5930 RETURN
5940 ! DAILY MAXS & MINS PRINTOU
     T
5950 PRINT "===================
     ============"
5960 T5=VAL(D7$[1])
5970 T6=VAL(D7$[4])
5980 PRINT T5;"/";T6."/";Y$
5990 PRINT " POSITION     MAXIMUM
       MINIMUM"
6000 PRINT "   NO.        READING
       READING"
6010 PRINT "-------------------
     -------------"
6020 PRINT "TEMPERATURE    DegC
       DegC "
6630 FOR L1=1 TO I2
6040 PRINT TAB(4);L1;TAB(14);Y1(
     L1);TAB(25);Z1(L1)
6050 NEXT L1
6060 PRINT "HUMIDITY        %rh
       %rh"
6070 FOR L1=1 TO I3
6080 PRINT TAB(4);L1,TAB(14);Y2(
     L1);TAB(25);Z2(L1)
6090 NEXT L1
6100 PRINT "FLOW VELOCITY   m/s
       m/s"
6110 FOR L1=1 TO I4
6120 PRINT TAB(4),L1.TAB(14);Y3(
     L1);TAB(25),Z3(L1)
6130 NEXT L1
6140 PRINT "ENTHALPY        kJ/kg
       kJ/kg"
6150 FOR L1=1 TO I3
6160 PRINT TAB(4);L1:TAB(14);Y4(
     L1);TAB(25),Z4(L1)
6170 NEXT L1
6180 PRINT "MASS FLOW       kg/s
       kg/s"
6190 FOR L1=1 TO I4
6200 PRINT TAB(4),L1 TAB(14);Y5(
     L1);TAB(25),Z5(L1)
6210 NEXT L1
6220 PRINT "ENERGY FLOW   kJ/s=kW
       kJ/s=kW"
6230 FOR L1=1 TO I4
6240 PRINT TAB(4),L1.TAB(14);Y6(
     L1);TAB(25),Z6(L1)
6250 NEXT L1
6260 T9=0
6270 GOSUB 7400
6280 PRINT
6290 FOR L1=1 TO 27
6300 IF C9(L1)=0 THEN 6330
6310 PRINT "ERRORS AT "
6320 PRINT "POSITION ",L1," (";C
     9(L1);"TIMES)"
6330 NEXT L1
6340 FOR L1=1 TO I2
6350 C9(L1)=0
6360 NEXT L1
6370 RETURN
6380 ! ENTHALPY CALCULATION
6390 X1=1013.25
6400 X2=P+273.16
6410 IF P<0 THEN 6450
6420 X3=LOG(X2)/LOG(10)
6430 X4=31.5901-8.2*X3+.0024804*
     X2-3142.31/X2
6440 GOTO 6460
6450 X4=10.5381-2663.91/X2
6460 X6=10^X4
6470 X5=H*X6/100
6480 IF P<11 THEN 6520
6490 IF P<26 THEN 6540
6500 IF P<=60 THEN 6560
6510 GOTO 6560
6520 F1=-(.0000073*P)+1.00444
6530 GOTO 6570
6540 F1=.0000132*P+1.00425
6550 GOTO 6570
6560 F1=.0000405*P+1.003497
6570 X8=.62197*F1*X6/(X1-F1*X6)
6580 X9=H*(X1-X6)/(X1-X5)
6590 X7=X9*X8/100
6600 IF P<0 THEN 6670
6610 IF P<=10 THEN 6700
6620 IF P<=20 THEN 6730
6630 IF P<=30 THEN 6760
6640 IF P<=40 THEN 6790
6650 IF P<=50 THEN 6820
6660 GOTO 6850
6670 F2=1.00545*P
6680 F3=2500.822+1.8375*P
6690 GOTO 6870
6700 F2=1.00597*P
6710 F3=2500.92+1.84*P
6720 GOTO 6870
6730 F2=1.00597*P
6740 F3=2519.32+1.83*(P-10)
6750 GOTO 6870
6760 F2=1.0062*P-.0045
6770 F3=2537.62+1.82*(P-20)
6780 GOTO 6870
6790 F2=1.00676*P-.0213
6800 F3=2555.82+1.8*(P-30)
6810 GOTO 6870
6820 F2=1.00707*P-.0337
6830 F3=2573.82+1.77*(P-40)
6840 GOTO 6870
6850 F2=1.00778*P-.0692
6860 F3=2591.52+1.76*(P-50)
6870 F4=F2+X7*F3
6880 RETURN
6890 PRINT
6900 PRINT
6910 T5=VAL(D7$[1])
6920 T6=VAL(D7$[4])
6930 PRINT "AVERAGED HOURLY READ
     INGS"
6940 PRINT "   1=OUTSIDE"
6950 PRINT "   2=SPIN DOFF (Tc401
     )"
6960 PRINT "   3=DRAWTWIST C BANK
      LAG AREA"
6970 PRINT "   4=DRAWTWIST C4-C5"
6980 PRINT "   5=DRAWTWIST C7-C8"
6990 PRINT "   6=DRAWTWIST C10-C1
     1"
7000 PRINT "   7=UNDERFLOOR SUPPL
     Y DUCT"
7010 PRINT "[ MOIST=Moisture Con
     tent g/kg ]"
7020 PRINT
7030 PRINT "DATE   ";T5;"/";T6;"/
     ";Y$
7040 PRINT "                 POSIT
     ION NO."
7050 PRINT "HOUR        1   2   3
      4   5   6   7 "
7060 RETURN
7070 ! TECHNICAL PRINTOUT
7080 S1(8)=IP(S1(8)+.5)
7090 S2(8)=IP(S2(8)+.5)
7100 S9(8)=IP(S9(8)+.5)
7110 S1(9)=IP(S1(9)+.5)
7120 S2(9)=IP(S2(9)+.5)
7130 S9(9)=IP(S9(9)+.5)
7140 FOR L1=13 TO 16
7150 S1(L1)=IP(S1(L1)+.5)
7160 S2(L1)=IP(S2(L1)+.5)
7170 S9(L1)=IP(S9(L1)+.5)
7180 NEXT L1
7190 S1(3)=IP(S1(3)+.5)
7200 S2(3)=IP(S2(3)+.5)
7210 S9(3)=IP(S9(3)+.5)
7220 PRINT USING 7230 ; T3,"T(C)
",S1(8),S1(9),S1(13),S1(14),S1(1
5),S1(16),S1(3)
7230 IMAGE X,2Z,X,4A,2X,3D,3D,3D
,3D,3D,3D,3D
7240 PRINT USING 7250 ; "R.H.",S
2(8),S2(9),S2(13),S2(14),S2(15),
S2(16),S2(3)
7250 IMAGE 4X,4A,2X,3D,3D,3D,3D,
3D,3D,3D
7260 PRINT USING 7270 ; "MOIST",
S9(8),S9(9),S9(13),S9(14),S9(15)
,S9(16),S9(3)
7270 IMAGE 4X,5A,X,3D,3D,3D,3D,3
D,3D,3D
7280 GOSUB 5610 ! ZERO SOME ARRA
YS
7290 RETURN
7300 STOP
7310 ! BLOCK SPECIFICATION FOR S
TANDARD INPUT
7320 I6=13
7330 FOR L1=1 TO I6
7340 READ C1(L1),C2(L1),C3(L1),C
4(L1)
7350 NEXT L1
7360 DATA 2,0,0,8,2,4,7,9,2,8,14
,1,2,15,18,13,3,20,26,1,1,40,40,
S,1,41,47,1
7370 DATA 1,48,48,11,1,49,49,10,
1,63,63,9,1,64,69,13,1,80,88,19,
1,89,89,12
7380 CLEAR
7390 GOTO 1650
7400 FOR L1=1 TO 27
7410 D1(L1)=0
7420 NEXT L1
7430 FOR L1=1 TO 16
7440 D2(L1)=0
7450 D4(L1)=0
7460 NEXT L1
7470 FOR L1=1 TO 7
7480 D3(L1)=0
7490 D5(L1)=0
7500 NEXT L1
7510 RETURN
 5585
```

CHAPTER 5

AIR FLOW EQUATIONS FOR BUILDINGS

5.1 INTRODUCTION

Air may be transferred between two locations if a suitable flow path and driving force (pressure difference) exist. In buildings this transfer may be from one internal area to another, or to and from the external environment. Such transfers are important; they can provide fresh air, they also provide the means by which air borne odours and contaminants may be transported to, from and within buildings. Since there are often differences in the temperature and water content of air at points within a building and between internal and external air; the movement of air also implies the transfer of heat and moisture. Thus for many reasons it it important to understand, and to some degree to be able to predict, the air flow pattern. In order to make such predictions, it is essential to be able to relate the main factors governing flow by a mathematical expression.

The first step in such a treatment would be to define the boundaries of the area, or areas, under consideration. These boundaries will usually coincide with major structural partitions such as walls, floors and ceilings. The second step involves the identification of the flow paths by which air might be transferred.

These flow paths may be obvious and large (open doorways), small (cracks around windows) or sometimes almost insignificant (diffusion through walls). The last of these categories (often referred to as "background leakage") is difficult to define, and is sufficiently small to be ignored in most cases. If the pressure difference acting across the defined flow paths, or flow openings, is given, then the flow rate might be predicted by use of a suitable flow equation.

In the following sections a brief description of the development of such flow equations is given. The equations used for flow prediction by current guides are shown and equations for particular circumstances discussed. It will be seen that normal flow equations are not useful in certain situations, and new equations are developed for such cases.

5.2 THE DEVELOPMENT OF FLOW EQUATIONS

Bedford[1] credited Sir Napier Shaw[2] with the definition of four fundamental laws related to air flow. Shaw's definitions were based on the idea of air flowing in circuits; these circuits beginning and ending in the external air. The first law was a simple proposition of continuity of flow and conservation of mass, it stated that the total flow into a ventilated space would be balanced by the total flow out.

Shaw's second law concerned the relationship between the "head" or "aeromotive force", (ie, pressure difference) and the flow rate.

This followed a square law:-

$$H = RV^2 \tag{5.1}$$

where H = pressure difference (inches water gauge)

R = resistance

V = flow rate (ft^3/h)

The third and fourth laws dealt with the interaction of multiple openings between compartments. The third law concerned two or more air flow ducts "in parallel" (ie, the ducts connected the same two compartments and had the same pressure difference acting across them). Such ducts in parallel would produce the same effect as that of a single duct whose "equivalent orifice" was equal to the sum of the "equivalent orifices" of the separate ducts. The "equivalent orifice" was defined as a simple opening in a flat plate which would allow the same quantity of air to pass through, at a given pressure difference, as the flow path under consideration. The resistance of an orifice was given as being inversely proportional to the square of the orifice area:-

$$R = K\ ({}^{1}/a^{2}) \tag{5.2}$$

where K = Constant

a = Orifice area

The fourth law concerned air flow paths "in series" (ie, the air must flow through sequentially through a number of ducts or openings in series). In this situation the total resistance would be calculated by summing the resistances of the separate ducts or openings:-

$$H = (R_1 + R_2 + R_3 + \ldots)\ V^2 \qquad (5.3)$$

where H = total pressure difference

R_1, R_2, R_3, etc = resistances of individual ducts.

These basic laws or definitions still apply, for the most part, in the present day, however much work has been carried out to assess the range af applicability of these laws and to extend and modify them where necessary.

James Dick performed tests in the Laboratory and carried out field studies concerned with housing ventilation. Some of his Laboratory tests(3) comfirmed Shaw's second law by suggesting a square law relationship between the pressure difference across, and air flow through a building element. Dick also noted however, that this relationship was only an approximation, though a fairly good approximation. He considered that changes in the flow, due to variations in flow path size and geometry in the building element, (evident as differences in Reynolds Number) would have an effect.

He was able to quantify his approximate relationship as:-

$$V = 1070.a.H^{\frac{1}{2}} \tag{5.4}$$

where a = Equivalent orifice area (ins^2)

H, V as before

Typical equivalent orifice areas were found by Dick and these are listed in Table 5.1.

The equation (5.4) showed that openings in building components behaved in a similar fashion to simple sharp edged orifices in thin plates which had a coefficient of discharge of 0.65. (The coefficient of discharge relates the actual and theoretical flows through an opening. It is discussed in more detail later).

Dick also rearranged the third and fourth of Shaw's definitions. For flow paths in parallel:-

$$V = 1070\ (a_1 + a_2 + a_3 + a\ \ldots)H^{\frac{1}{2}} \tag{5.5}$$

V, H as before

a_1, a_2, a_3, etc = the equivalent orifice areas of the individual flow paths

For two flow paths or components acting in series:-

$$V = 1070\ \frac{a_1 a_2}{(a_1^2 + a_2^2)}\ H^{\frac{1}{2}} \tag{5.6}$$

where H = total pressure difference = $H_1 + H_2$

TABLE 5.1

TYPICAL EQUIVALENT ORIFICE AREAS (AFTER DICK[3])

COMPONENT	EQUIVALENT AREA ins^2	EQUIVALENT AREA cm^2
Windows (25 ft crack length)		
- non-weatherstripped	13	84
- weatherstripped	3	19
- frame	1	6
Doors (18 ft crack length)		
- non-weatherstripped	13	84
- weatherstripped	7	45
Walls (100 ft^2 of 9 ins wall)		
- unplastered	3	19
- plastered	0	0
Air Bricks	10-50	65-323
Floors (12 ft x 12 ft)		
- solid	0	0
- tongued and grooved boards	35	226
- square boards	200	1290
- ventilators (60 ft wall run)	40-90	258-581
Flues		
- open fire	50	323
- gas fired	20-50	129-323
Heating Appliances	Large Range	Large Range
Ventilators		
- fixed louvre	24	155
- constant flow wind 2 mph	13.5	87
wind 20 mph	3.5	23

H_1 and H_2 being the pressure difference acting across orifices of areas a_1 and a_2 respectively.

The total equivalent orifice area (a) was found for the "in series" case by

$$^1/a^2 = {}^1/a_1{}^2 + {}^1/a_2{}^2 \qquad (5.7)$$

Thus when considering the openings in series, it can be seen that the total equivalent area is usually reduced, lying within the range 0.7 to 1.0 times the equivalent area of the smaller opening. This is shown in Figure 5.1. For many flow paths in series, the effective area is further reduced, with the smallest opening being the main determining factor.

The Chartered Institute of Building Services (CIBS) Guide to Current Practice[4] incorporated the results of Dick's work. In the present edition, the following form of equation (5.4) is used:-

$$Q = 0.827.A.(\Delta P)^{0.5} \qquad (5.8)$$

where Q = Rate of Air Flow (m^3/s)

A = Area of Orifice (m^2)

ΔP = Pressure Drop across orifice (N/m^2)

As Dick supposed, this equation gives only an approximate relationship and it is often replaced by a more appropriate version, of a generalised form -

$$Q = K.A.(\Delta P)^n \tag{5.9}$$

where Q = constant

n = exponent

This equation presents a relationship which may be applied to different types of flow path, often it represents the average effect of a number of openings. The constant, K, takes a value dependent upon the air density and the discharge coefficient of the flow path opening. The exponent, n, has been assigned various values in different studies. For laminar flow conditions - an exponent of unity would be expected; for turbulent flow - an exponent of 0.5. (Thin cracks and small holes sometimes produce laminar flow, whilst most other openings usually give rise to turbulent flow). Since in most building environments, both types of opening might be found, many studies have taken an "averaged" exponent value of between 0.6 and 0.7.

The Building Research Establishment[5] considered dimensional analysis, and produced the generalized flow equation.

$$Q = A.F.\sqrt{\frac{2.\Delta P}{\rho}} \tag{5.10}$$

where ρ = density of air (kg/m^3)

Q, A, ΔP as before

F is a function of the Reynolds Number and the geometry of the opening.

For large openings (ie, openings with a dimension typically greater than 10 mm) such as airbricks, open windows and doors, the function, F, was considered to be constant and equal to the discharge coefficient. Thus equation (5.10) become;

$$Q = A.C_d . \sqrt{\frac{2.\Delta P}{\rho}} \qquad (5.11)$$

where C_d = Discharge coefficient

A = Equivalent orifice area

The B.R.E. took a value for C_d of 0.61, this being representative of a sharp edged opening and high Reynolds Number. The equivalent orifice area was considered to be close to the geometric open area for windows and doors, but might be significantly different for some openings such as airbricks.

For small crack type openings, the function, F, took on a more complex form. For high flow rates (which implied a high Reynolds Number) the function approached the form of equation (5.11). However at very low flow rates, the function was assumed proportional to the Reynolds Number, and the flow rate proportional to the acting pressure difference. Under normal circumstances a situation somewhere between these extremes might be found and the equation proposed was,

$$Q \doteq L.k.(\Delta P)^n \qquad (5.12)$$

where L = Length of crack (m)

k = Leakage coefficient (litres/s/m/Pa)

The value of n, the exponent was expected to be between 0.6 and 0.7, some typical values of the constant k being shown in Table 5.2.

TABLE 5.2

VALUES FOR WINDOW LEAKAGE COEFFICIENT (B.R.E.[5])

WINDOW TYPE	LEAKAGE COEFFICIENT (k) (LITRES/SECOND PER METRE AT 1 Pa)	
	AVERAGE	RANGE
Sliding	0.08	0.02 - 0.03
Pivoted	0.21	0.06 - 0.80
Pivoted (weatherstripped)	0.08	0.005 - 0.20

5.3 FLOW EQUATIONS FOR DESIGN AND PREDICTION

A fairly straightforward technique is required for designers to be able to predict air flows in buildings. In Britain this need is principally satisfied by the CIBS Guide to Current Practice[4]. The methods proposed therein, were derived in the main from the work of Jackman[6]. His study compared the predicted infiltration rates for tall buildings determined from a digital computer technique and from an electrical analogue. The main flow paths considered were cracks and small gaps around windows and doors, hence the flow equation used was of the form of equation (5.12).

Values of the leakage coefficent were taken for typical building elements, whilst the flow exponent was assumed to lie between 0.59 and 0.73; 0.625 being considered representative of average conditions. For simplification Jackman condensed information on window crack lengths and the amount of glazing found in tall office type buildings into a single parameter of crack length per unit area of building facade. The value of this parameter was found to be between 0.09 and 0.9 ft/ft^2 (0.3 and 3.0 m/m^2 respectively). For office doors a crack length of 0.152 ft/ft^2 (0.5 m/m^2) was assumed. For entrance doors and corridor/stairwell doors, the overall crack length was taken as 33 ft (10 m).

Using the results of this study, Jackman was able to produce an "Infiltration Chart" which allowed the prediction of flow rates. This chart is presented in the CIBS Guide and is shown in Figure 5.2.

The chart is used as follows: The Building height and its environment are used to predict a mean pressure difference to be expected (left hand side of the chart - units not shown). This value is then projected to the right hand side of the chart. The window infiltration coefficient, for the type of window being considered, is found from Table 5.3. This information is then used to find the Infiltration Rate in litres per second per metre run of window opening joint.

Figures derived by this method rely on the window cracks offering the most significant resistance to air

TABLE 5.3

AIR INFILTRATION THROUGH WINDOWS (CIBS[4])

WINDOW TYPE	WINDOW INFILTRATION COEFFICIENT (LITRES/m/s/Pa)
Horizontally or Vertically Pivoted (non-weatherstripped)	0.25
Horizontally or Vertically Pivoted (weatherstripped)	0.05
Horizontally or Vertically Sliding (non-weatherstripped)	0.25
Horizontally or Vertically Sliding (weatherstripped)	0.125

flow through the building. If there is a substantial amount of internal partitioning, this can affect the infiltration rate and a correction factor should be applied. Table 5.4 indicates such correction factors.

The total infiltration for a room can be found by,

$$Qr = Q.Lr.f \qquad (5.13)$$

where Qr = Room infiltration (litres/s)

Q = Infiltration rate determined from chart (litres/s/m)

Lr = Crack length (m)

f = Correction factor

Some adjustment to the rate so calculated, is made to account for the influence of stack effect, though the total infiltration rate is not greatly altered, rather its distribution within the building (stack effect is caused by buoyancy forces due to inside - outside temperature differences). The size and overall plan dimensions of the building and the distribution of glazed facades also have an influence, details of which are given in the CIBS Guide[4].

As a predictive tool for design purposes, so that heating loads etc might be calculated, this method is very useful. However the simplifications and assumptions it makes mean that it is necessarily limited in application.

TABLE 5.4

CORRECTION FACTORS FOR INFILTRATION RATE CIRCULATION (CIBS[4])

WINDOW TYPE	INTERNAL BUILDING LAYOUT	CORRECTION FACTOR (f)
All Types	Open Plan (no full partitioning)	1.0
Short length of well fitting window opening joint (20% of facade openable).	Single Corridor (with many side doors). Liberal Internal Partitioning (with few interconnecting doors).	1.0 1.0
Long length of well fitting window or short length of poor fitting window joint (20-40% of facade openable)	Single Corridor Liberal Partitioning	1.0 0.8
Long length of poor fitting window joint (40-50% of facade opening)	Single Corridor Liberal Partitioning	0.8 0.65
Very long length of poor fitting window joint (greater than 50% of facade openable)	Single Corridor Liberal Partitioning	0.65 0.4

Cockroft[7] in a study of air flows in residential housing employed a flow equation of the form of equation (5.9). This was used both for cracks and for larger openings. He recommended a series of values to be adopted for the flow constant, K, and for the flow exponent, n, which were dependent on the size of the flow opening. For small cracks the flow was assumed laminar and a flow exponent of unity was used. As crack width increased, the flow changed, resulting in a decreasing flow exponent. This took a value of 0.5 for openings of 10 mm or more, through which turbulent flow was assumed. The figures used are listed in Table 5.5.

TABLE 5.5

VALUES OF PARAMETERS FOR BASIC FLOW EQUATION (COCKROFT[7])

CRACK WIDTH	K	n	A
0.1 mm	0.001	1.0	
0.5 mm	0.01	0.95	
2.5 mm	1.0	0.72	Crack Length (m)
5 mm	2.0	0.61	Flow Units (litres/s)
10 mm	8.4	0.5	
Greater than 10 mm	0.84	0.5	Opening Area (m^2) Flow Units (m^3/s)

In the United States a more comprehensive guide to building design is provided by the American Society of Heating, Refrigerating and Air-Conditioning Engineers

(ASHRAE)[8]. They proposed a form of equation (5.9) for flow through cracks.

$$Q = C(\Delta P)^n \qquad (5.14)$$

where Q = Volume flow rate per unit length of crack (litres/s/m) or per unit area (litres/s/m^2)

C = Flow coefficient, volumetric flow per unit length of crack or per unit area, at unit pressure difference

Much information was also provided in the guide, on the leakage of various building components under a variety of prevailing conditions. It assumed that infiltration due to stack effect was the major portion of the total for multi-storey buildings in cold weather. This is a different assumption to that made by Jackman and could leak to inaccurate prediction, especially in buildings with significant interfloor partitioning. Data on infiltration through and around windows is shown in Figure 5.3 which was based on information contained within the ASHRAE Handbook[8].

In general, the equations offered by the major design guides seem to fulfil a necessity for design calculations, but are unable to deal with all situations. Both are most concerned with flow to and from the outside and lack the scope to provide information relating to internal partitions and air movements, and buildings much different from office block type accommodation.

5.4 EXPERIMENTALLY DERIVED FLOW EQUATIONS

5.4.1 FLOW THROUGH CRACKS

The difficulties in defining a flow equation with universal application, has led to studies in which empirically derived relationships have been proposed. Thomas and Dick[9] investigated the leakage of air through window cracks. Three common types of window, each specified by a British Standard, were used in the tests. They found, as might have been expected, that the flow depended on the width and length of the cracks; on the acting pressure differentials; and on other resistances within the flow circuit. The following relationship was derived from an analysis of the flow-pressure curves for each window.

$$\Delta P = AV + BV^2 \qquad (5.15)$$

where ΔP = Pressure difference (ins. water gauge)

V = Flow rate (ft^3/hour)

A, B = Constants, determined for each curve

Equation (5.15) can be rearranged to:

$$\Delta P = A(V + B/A.V^2) \qquad (5.16)$$

where B/A defines the shape of the curve and A its level on the graph.

These equations, by using two terms, one proportional to the flow rate, the other proportional to the square of the flow rate, attempted to take into account the possible mixed nature of the flow (laminar and turbulent), through window cracks.

In a number of studies which have investigated the flow-pressure relationship, the data has been fitted to an equation of the form of equation (5.9). Values of the constant, K, and the exponent, n, are then derived. In general, such equations are dimensionally incorrect and this can restrict their use once K and n have been defined. Hopkins and Hansford[10], in an investigation of different types of window cracks, incorporated flow dependence on Reynolds Number, and attempted to produce an accurate, dimensionally correct, flow equation. They began with an initial assumption that the flow rate was proportional to the square root of the pressure difference. However, a comparison of this theory and experimental results (see Figure 5.4) showed certain deviations. The authors attributed these to three possible reasons:-

i) The open area could have been increased or decreased as the pressure differential was varied, due to distortion of the openings.

ii) The discharge coefficient of the openings might not have been constant

iii) The square law relationship was not strictly true for cracks and depended upon geometry and flow rates.

An equation was developed which related the discharge coefficient to crack dimensions, taking Reynolds Number into account.

$$\frac{1}{Cd^2} = \frac{C_A Z}{Re_h D_h} + K_1 \qquad (5.17)$$

where
C_d = Crack discharge coefficient

C_A = "Apparent" coefficient based upon the aspect ratio of the crack and Re

Z = Centre line distance through the crack (m)

Re_h = Reynolds Number based upon D_h

D_h = Hydraulic Diameter (m)

K_1 = Empirical Constant

Laboratory tests were carried out to test the validity of the equation and to determine the values of C_A and K_1. The three basic crack types investigated are shown in Figure 5.5(a). The empirically determined values of C_A and K_1 can be found from Figures 5.5(b) and 5.5(c) respectively. The equation (5.17) was recommended as a semi-empirical method to evaluate, more accurately, the discharge coefficient, which is an important parameter in the description of flow.

5.4.2 FLOW THROUGH LARGER OPENINGS

In most building studies, it has been the flow paths to the external environment which have been the major concern. Since these flow paths are usually crack-like, this has led to a lack of experimental work on larger flow openings.

Some work has been carried out by the Building Services Research Unit(11) (12), but this dealt in the main with flows due to natural convection effects. Whyte and Shaw accounted for flow, when pressure and temperature differentials were zero across openings, as due to turbulence. They were able to use a flow equation in which the coefficient of discharge increased towards infinity as such zero differential conditions were approached.

In general, flow through large openings has been assumed to follow theory with less variation than crack openings.

5.4.3 EQUATIONS FOR FLOW THROUGH LARGER OPENINGS

The ASHRAE Handbook(8), proposed equations for flow through such openings. When wind impinges on a facade with large flow paths open, the following equation would be used.

$$Q = C_v A V \qquad (5.18)$$

where Q = Air flow rate (litres/s)

A = Free area of opening (m^2)

V = Wind velocity (m/s)

C_v = Effectiveness of the opening

The "effectiveness" of the opening depended on the angle of incidence of the wind. For perpendicular winds, values for C_V were between 0.5 and 0.6; for diagonally approaching winds C_V and values of between 0.25 and 0.35.

Equation (5.18) was to be principally used to determine adventitious ventilation due to the wind acting on opened windows. For flow caused by stack effect only, equation (5.19) was recommended.

$$Q = C_t\, A \sqrt{h.(t_i - t_o)/t_i} \qquad (5.19)$$

where h = Height difference between inlets and outlets (m)

t_i = Average indoor air temperature (^oC)

t_o = Average outdoor air temperature (^oC)

Q = Air flow (litres/s)

A = Free area of inlets or outlets (m^2)

C_t = Constant of proportionality

The constant of proportionality took into account the "effectiveness" of the openings. Normally the effectiveness was taken as 65% and C_t = 119, but under unfavourable conditions, a 50% effectiveness was assumed and C_t = 89. If under certain circumstances, outdoor air temperature was greater than the indoor air temperature, then t_o would replace t_i as the denominator in the expression. Equation (5.19) would normally be used for sets of ventilation openings, spaced vertically within the building facade.

For the cases of horizontal openings and small vertical openings, the pressure difference acting is usually assumed to be constant across the whole cross-section of the opening (in the case of turbulent pressure fluctuations, this may not be so). However where a temperature difference exists across the opening, the flow can be altered, or a flow induced, by the effect of small variations in density over the height of the opening. This can result in air flow in both directions at the same time - Figure 5.6 illustrates this case. As with stack effect induced ventilation in buildings, a height (referred to as the "neutral level" or "neutral height") can be defined at which internal and external pressures balance, and at which there is no flow. Above this, flow would be in one direction only, and below it, in the opposite direction.

Cockroft[7] described a procedure for the calculation of the flow through such an opening, the flow being assumed proportional to the square root of the pressure difference.

$$dQ = C_d w dx \sqrt{\frac{2 \Delta P_p}{\rho}} \qquad (5.20)$$

where Q = Flow

C_d = Coefficient of discharge

w = Opening width

x = Height of horizontal plane under consideration

$P_p = P_1 - P_2$ = pressure difference at plane considered

ρ = Mean air density

Equation (5.20) was integrated and substitutions made, resulting in:-

$$Q = 2/3.B.A. \frac{(Ca^{3/2} - Cb^{3/2})}{Ct} \tag{5.21}$$

where $B = Cd\sqrt{2/\rho}$

A = Opening area

$Ca = (1-r_p)\ Ct + (P_1 + P_2)$

$Cb = (P_1 - P_2) - r_p Ct$

$Ct = \frac{gPh}{R}\ (1/T_2 - 1/T_1)$

P = Average pressure

R = Gas constant

$r_p = h_p/h$

h_p = Height of plane

h = Height of opening

T_1 and T_2 = Temperatures (K) on each side of opening

The evaluation of equation (5.21) would yield real and imaginary parts corresponding to the flows from space 1 to 2, and from space 2 to 1 respectively. For the cases in which the temperature difference between the spaces was small, Ct would be approximately zero. To avoid the consequent problems of division by zero in the equation, Cockroft gave another equation;

$$Q = B.A.\ (P_1 - P_2)^{\frac{1}{2}} \left[1 + \frac{Ct(1-2r_p)}{(P_1 - P_2).4} \right] \qquad (5.22)$$

In this case, as in equation (5.21), real and imaginary solutions would be produced representing the two flow directions.

5.5 THE APPLICATIONS OF FLOW EQUATIONS

The flow equations so far discussed, relate, in the main, to flow through the internal - external boundary of a building. This is evidenced by the number of studies which have been concerned with cracks around windows and other small openings in the building envelope. This bias is to be expected, since it is these flows, combined with temperature differences, which account for substantial energy flows and heating loads. It is generally assumed that flows between compartments wholly contained within the building are of less importance.

In some cases, however, internal flow patterns are important; for instance in the spread of contaminants in a hospital environment, or the movement of smoke and other pollutants. It may also be advisable in the design context, to be able to predict the heating requirements of individual rooms or areas. Substantial environmental variations between internal areas can occur in industrial and other buildings, increasing the need for flow prediction. Some knowledge of internal flow is also required, in order that the potential for flow through the external building boundary is correctly interpreted by taking into account internal resistances.

For building, consisting of a number of interconnected rooms, the interaction of the various flows is complex. Often a digital computer method is required to solve the resulting sets of simulataneous flow equations. The assessment of flow is aided if flow paths can be amalgamated. The basic means by which such simplifications may be made, by the use of series and parallel flows has already been outlined in Section 5.2. As has been stated however most flow equations have been derived with a view to use in crack/small opening situations. The flow openings found within buildings are usually larger. Internal cracks are not likely to be weatherstripped, and the cracks themselves are usually quite large. In considering such large openings it is necessary to re-examine the flow equations. The following sections discuss the behaviour of an opening as an orifice and begin with a basic derivation of the flow equation (which follows the method of

Owen and Pankhurst[13]).

5.6 FLOW THROUGH A CONSTRICTION

A pipe is considered, shaped as in Figure 5.7, in which the cross-sectional area at AA is greater than that at BB. The conditions in plane AA are p_1, v_1, ρ_1, a_1 and in plane BB - p_2, v_2, ρ_2, a_2 (where p is the absolute static pressure, v is the mean velocity, ρ is the air density and a is the cross-sectional area). The pipe walls at AA are parallel to the direction of flow; the flow also being assumed frictionless. The ratio of the static pressures p_2/p_1 is often little different from unity and given this, the densities of the air in the two planes are considered equal.

Taking the general Bernoulli equation

$$\frac{v^2}{2} + \int \frac{dp}{\rho} = \text{constant} \tag{5.23}$$

and considering the sections AA and BB then

$$\frac{v_2^2 - v_1^2}{2} = \int_{p_2}^{p_1} \frac{dp}{\rho} \tag{5.24}$$

Since it is assumed $\rho_1 = \rho_2$ (ie, the air is incompressible) equation (5.24) becomes

$$v_2^2 - v_1^2 = \frac{p_1 - p_2}{\rho} \qquad (5.25)$$

The mass of air passing AA must equal the mass of air passing BB, therefore

$$\rho v_1 a_1 = \rho v_2 a_2 \qquad (5.26)$$

or $$v_1 = \frac{v_2 a_2}{a_1} \qquad (5.27)$$

Substituting into equation 95.25)

$$\frac{v_2^2 - \dfrac{v_2^2 a_2^2}{a_1^2}}{2} = \frac{p_1 - p_2}{\rho}$$

$$v_2^2 \left(1 - \frac{a2^2}{a_1^2}\right) = \frac{2(p_1 - p_2)}{\rho}$$

$$v_2 = \sqrt{\frac{2(p_1 - p_2)}{\rho(1 - a_2^2/a_1^2)}} \qquad (5.28)$$

The theoretical volume of air flowing is given by

$$Q_t = v_2 a_2 = a_2 \sqrt{\frac{2.\Delta P}{\rho(1-a_2^2/a_1^2)}} \qquad (5.29)$$

and the theoretical mass flow by

$$M_t = \rho v_2 a_2 = a_2 \sqrt{\frac{\rho.2.\Delta P}{(1 - a_2^2/a_1^2)}} \qquad (5.30)$$

5.7 FLOW THROUGH A THIN PLATE ORIFICE

The equations used to describe flow through openings in building elements, likens the relationship to that of an orifice in a thin plate. If such an orifice, being a sharped-edge, circular hole cut in a thin plate, is inserted transversely into a pipe, such that the hole is co-axial with the pipe, the position shown in Figure 5.8 is obtained. Thus, the constriction of Figure 5.7 is replaced by a thin plate orifice. In this situation the flow from the orifice continues to contract for a short distance downstream, the minimum flow cross-section, which is usually between 0.6 and 0.7 of the orifice area, is reached approximately one pipe diameter from the opening. This point is often referred to as the "vena contracta". The flow expands to the full cross-section at some later stage downstream.

If the term xa_2 is used to represent the area of flow at the vena contracta, then the theoretical volume of air flowing is given by;

$$Q_t = xa_2\sqrt{\frac{2\Delta P}{\rho(1-x^2a_2^{\,2}/a_1^{\,2})}} \qquad (5.31)$$

And if y is the ratio between the actual and theoretical flows, ie, $Q = yQ_t$ (Q = actual flow) then,

$$Q = yxa_2\sqrt{\frac{2\Delta P}{\rho(1-x^2a_2^{\,2}/a_1^{\,2})}} \qquad (5.32)$$

The factors x and y cannot be computed from theory or be easily determined experimentally, we therefore use an overall "coefficient of discharge", C_d to represent these. The equation for the actual volumetric flow becomes;

$$Q = C_d\, a_2\sqrt{\frac{2\Delta P}{\rho(1-a_2^{\,2}/a_1^{\,2})}} \qquad (5.33)$$

Similarly, actual mass flow is given by

$$M = C_d\, a_2\sqrt{\frac{\rho.2.\Delta P}{(1-a_2^{\,2}/a_1^{\,2})}} \qquad (5.34)$$

If an area of the orifice is small compared with the cross-section of the pipe, then the quantity $a_2^{\,2}/a_1^{\,2}$ tends to zero and equation (5.33) becomes,

$$Q = C_d\, a_2 \sqrt{\frac{2\Delta P}{\rho}} \qquad (5.35)$$

Equation (5.34) becomes

$$M = C_d\, a_2 \sqrt{\rho . 2 . \Delta P} \qquad (5.36)$$

Therefore it is only in the situations in which a_2 is very small by comparison with a_1, that equations (5.35) and (5.36) may be used. Figure 5.9 shows the error caused in the calculation of flow rate by assuming the ratio a_2/a_1 to be insignificant. A 5% error is caused when a_2/a_1 exceeds 0.31. As the ratio a_2/a_1 increases, the flow rate also increases for a given pressure difference. The overall effect so using equations of the form of (5.33) or (5.34) rather than (5.35) or (5.36), is to increase the predicted flow rate.

5.8 BUILDING FLOW EQUATIONS

For air flows in the built environment it would seem sensible to adopt an equation of the form of (5.33). This would be used for flows through openings which are significant in size by comparison with the cross-sectional area of the spaces they join. There are difficulties to be encountered in the definition of the space flow area, a, however. In some cases such as the placing of a doorway across a corridor, this area can be measured, but there are many occasions when this would not be possible.

5.8.1 PARALLEL FLOWS

Consider the simple case of parallel flows shown in Figure 5.10. Conventionally, the area a_2 for equation (5.33) would be defined as the sum of areas, $A_b + A_c + A_d + A_e$ and this is assumed to hold true. However, it may not be correct to define $a_1 = A_a$, therefore the factor "k" is introduced into equation (5.33) to represent the variation. The ratio a_2/a_1 is replaced by "m"

$$Q = C_d \, a_2 \sqrt{\frac{2\Delta P}{\rho(1 - k^2 m^2)}} \tag{5.37}$$

This factor k might also be used to account for cases in which the partition does not lie perpendicular to the overall flow regime. In order to simplify the procedure, k can be incorporated into the Discharge Coefficient term thus

$$Q = C_{dk} \cdot a_2 \sqrt{\frac{2\Delta P}{\rho(1-m^2)}} \tag{5.38}$$

5.8.2 SERIES FLOW

In order to combine the effect of a number of partitions with openings in series, it is usual to sum the resistances. In Figure 5.11 the basic layout of series flow is shown. The areas A_1, A_2 and A_3 represent the sum of the open areas in their respective partitions.

Resistance is defined as the acting pressure difference divided by the square of the flow rate, ie, $R = \Delta P/Q^2$

By re-arranging equation (5.38)

$$\frac{\Delta P}{Q^2} = \frac{\rho(1-m^2)}{2C_{dk}^2 \, a_2^2} = R \qquad (5.39)$$

For Figure 5.10, the total resistance, R_T is given by

$$R_T = R_1 + R_2 + R_3 \qquad (5.40)$$

where R_1, R_2 and R_3 are the resistances of the individual partitions.

and

$$R_1 = \frac{(P_1 - P_2)}{Q^2} = \frac{\rho(1-A_1^2/A_T^2)}{2\, C_{dk1}^2 \, A_1^2}$$

$$R2 = \frac{(P_2 - P_3)}{Q^2} = \frac{\rho(1 - A_2^2/A_T^2)}{2\, C_{dk}^2 \, A_2^2}$$

$$R_3 = \frac{(P_3 - P_4)}{Q^2} = \frac{\rho(1 - A_3^2/A_T^2)}{2\, C_{dk3}^2 \, A_3^2}$$

Therefore;

$$R_T = \frac{(P_1-P_2)}{Q^2} + \frac{(P_2-P_3)}{Q^2} + \frac{(P_3-P_4)}{Q^2} = \frac{(P_1-P_4)}{Q^2} \quad (5.41)$$

$$R_T = \frac{(1-A_1^2/A_T^2)}{C_{dk1}^2\ A_1^2} + \frac{(1-A_2^2/A_T^2)}{C_{dk2}^2\ A_2^2} + \frac{(1-A_3^2/A_T^2)}{C_{dk3}^2\ A_3^2} \quad (5.42)$$

In this form the equation is too clumsy and involved to be of great practical benefit, indeed it would require the individual determination of each discharge coefficient. However there is a situation in which it can be of use.

5.9 FLOW THROUGH A SERIES OF SIMILAR PARTITIONS

If in the case shown by Figure 5.11, each of the open areas in the respective partitions are identical, then equation (5.42) can be simplified.

$$R_T = \frac{\rho}{2}\left[\frac{(1-A^2/A_T^2)}{C_{dk1}^2\ A_1^2} + \frac{(1-A^2/A_T^2)}{C_{dk2}^2\ A^2} + \frac{(1-A^2/A_T^2)}{C_{dk3}^2\ A^2}\right] \quad (5.43)$$

where $A = A_1 = A_2 = A_3$

Substituting $M = A/A_T$, this reduces to,

$$R_T = \frac{\rho}{2}\left[\frac{(1-M^2)}{C_{dk1}^{\ 2}\ A^2} + \frac{(1-M^2)}{C_{dk2}^{\ 2}\ A^2} + \frac{(1-M^2)}{C_{dk3}^{\ 2}\ A^2}\right]$$

and

$$R_T = \frac{\rho(1-M^2)}{2A}\left[\frac{1}{C_{dk_1}^{\ 2}} + \frac{1}{C_{dk_1}^{\ 2}} + \frac{1}{C_{dk_3}^{\ 2}}\right] \qquad (5.44)$$

In this case the resulting equation applies to combination of three partitions in series, however a more general form, applicable to any number of partitions in series is shown by the equation below,

$$R_T = \frac{\rho(1-M^2)}{2a_2^{\ 2}} \sum_{i=1}^{n} 1/(C_{di})^2 \qquad (5.45)$$

where n = number of partitions and C_d is a combined coefficient of discharge taking into account all factors.

Equation (5.45) provides a means by which the resistance to flow, set up by a series of similar partitions, can be calculated. If the resistance is found, then the flow rate for a given pressure difference can be determined.

The main factor governing this calculation process, is the value of the discharge coefficient for each partition. It may be that the discharge coefficient for each partition takes the same value; in this case the total resistance would be given by the product of the number of partitions and the resistance of one partition. This is shown in Figure 5.12. However if the partitions are laid out in close proximity, then the position of the openings "in line" may cause the partitions encountered after the first to exhibit a reduced resistance to flow. If each of the remaining partitions were to take the same reduced value of resistance, then the variation in total resistance would be shown as in Figure 5.13. Alternatively the reduction in resistance might follow a logarithmic decay, this being depicted by Figures 5.14. These three alternatives form the basis of the hypotheses which could indicate how the resistance from a series of similar partitions combine to produce an overall resistance to air flow through a confined building space.

It was decided to investigate these possibilities in the context of two different types of partition at model scale. The description of this model and its results are discussed in the following chapters.

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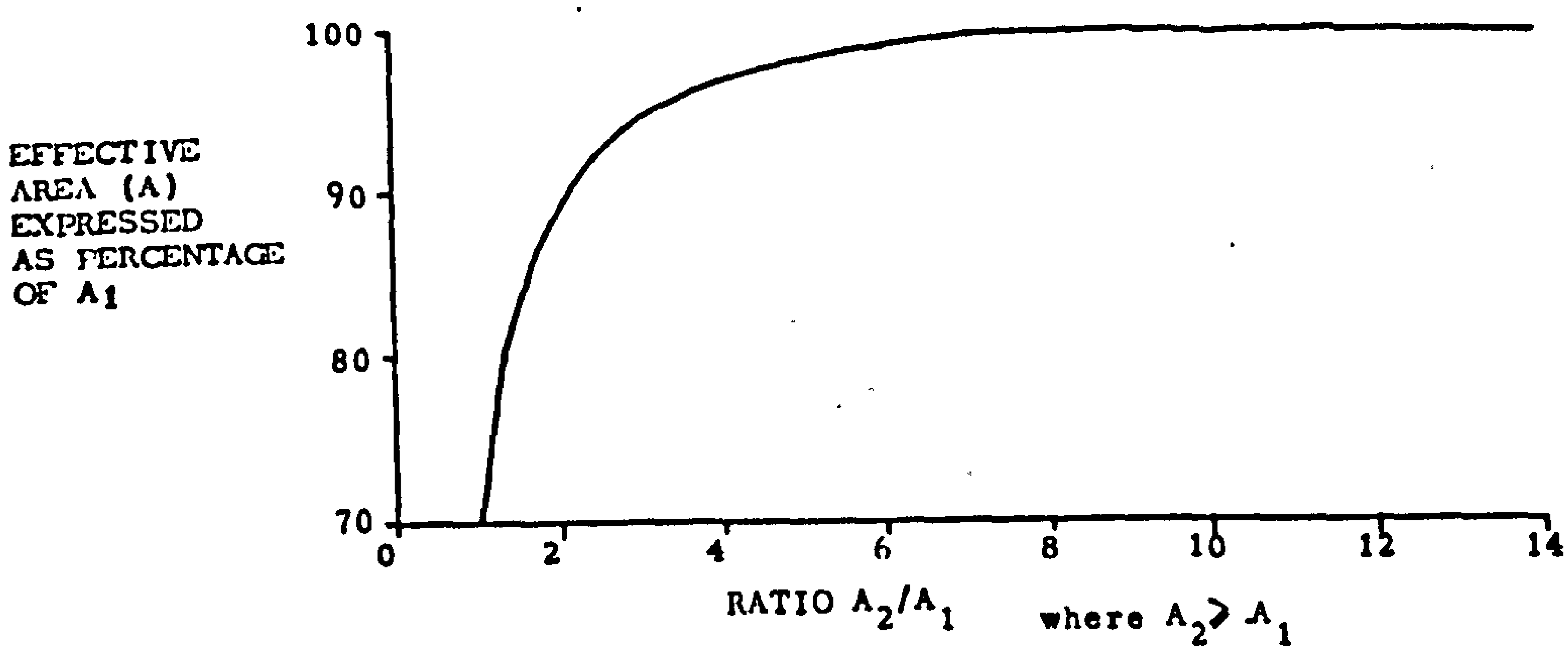


FIGURE 5.1 THE COMBINED EFFECTIVE AREA OF TWO OPENINGS IN SERIES

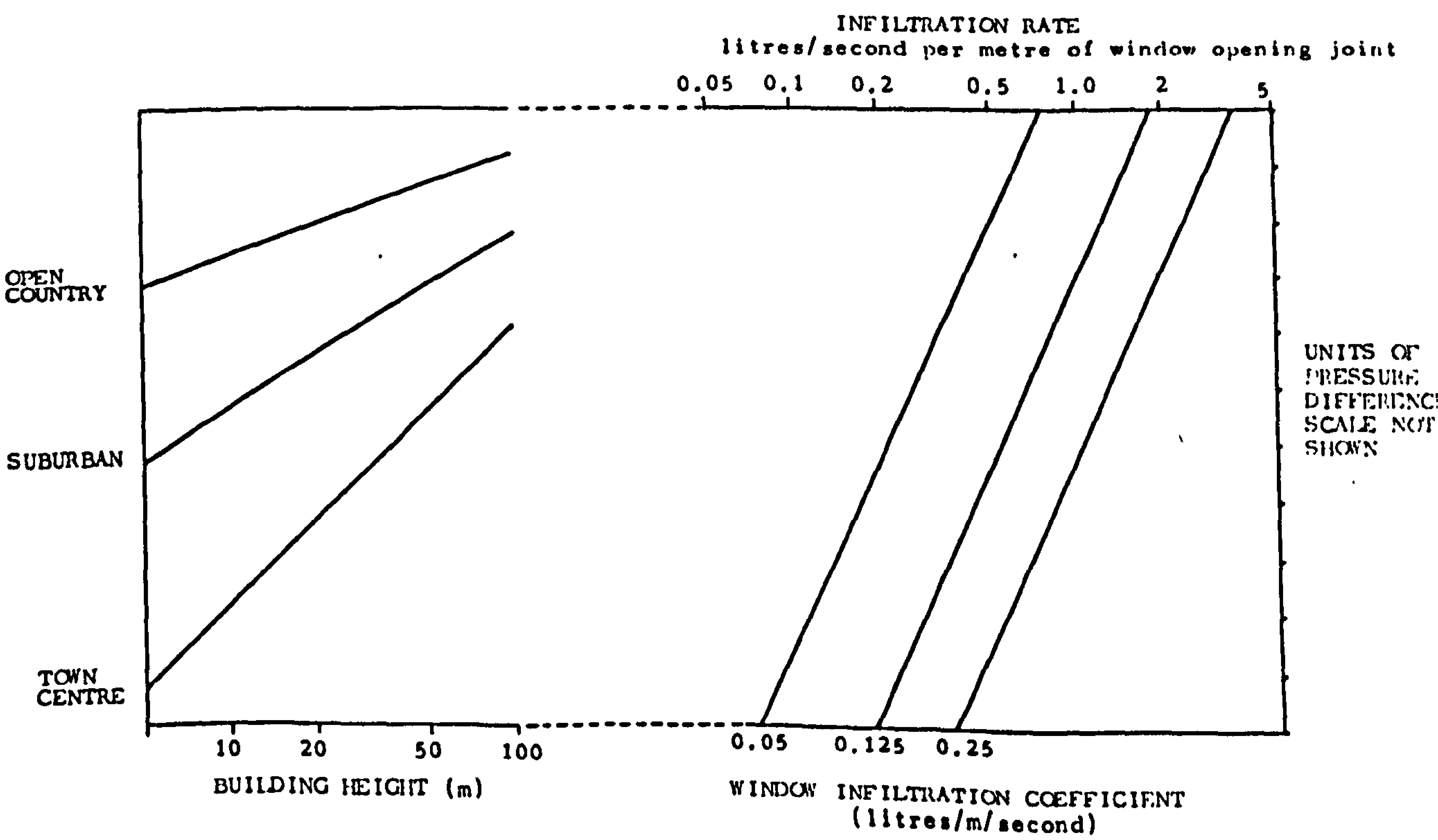
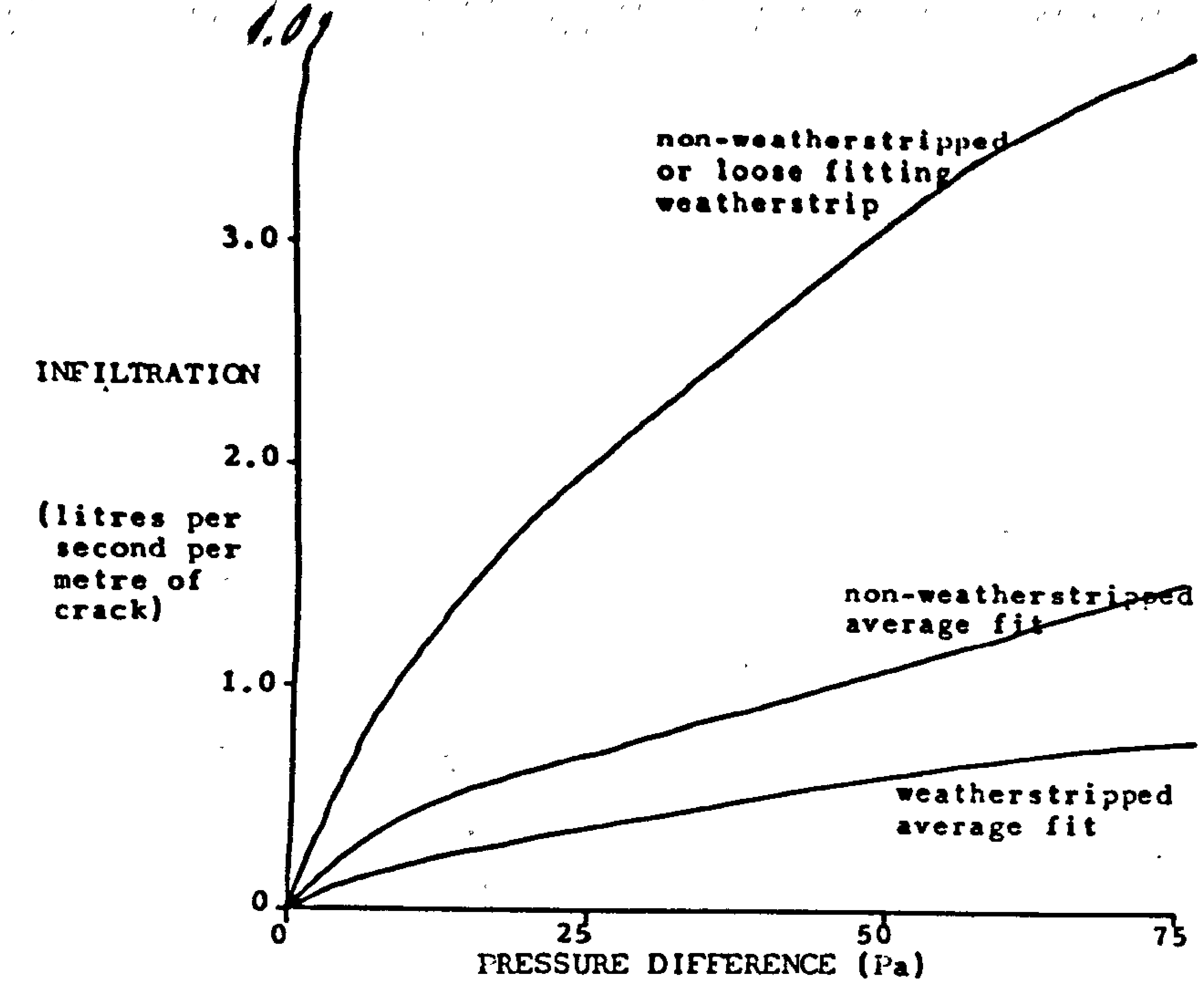
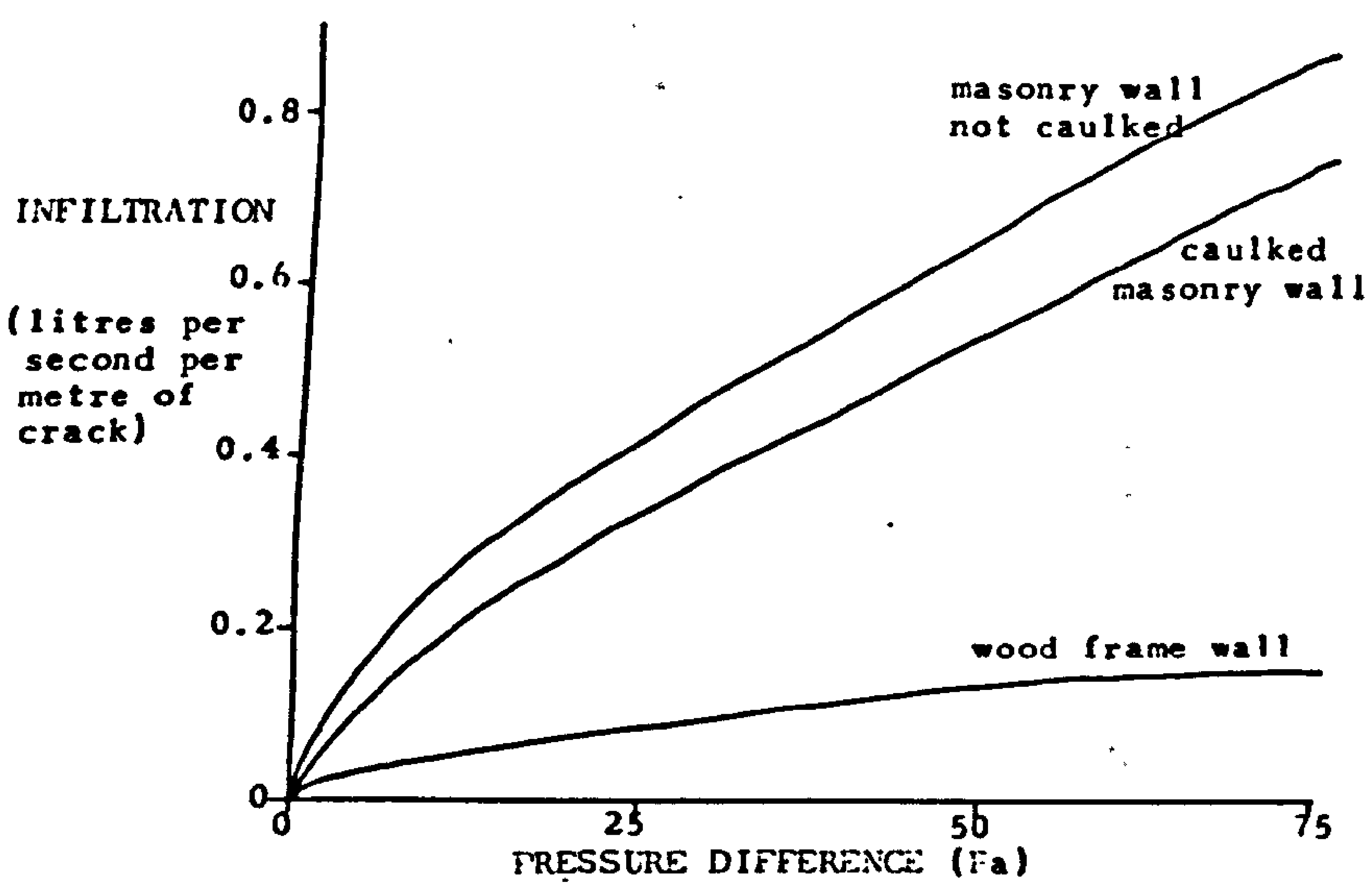


FIGURE 5.2 INFILTRATION CHART (AFTER C.I.B.S.[4])



(a) FLOW THROUGH LOCKED WINDOW



(b) FLOW THROUGH FRAME-WALL GAP

FIGURE 5.3 INFILTRATION THROUGH DOUBLE GLAZED WINDOWS (A.S.H.R.A.E.[8])

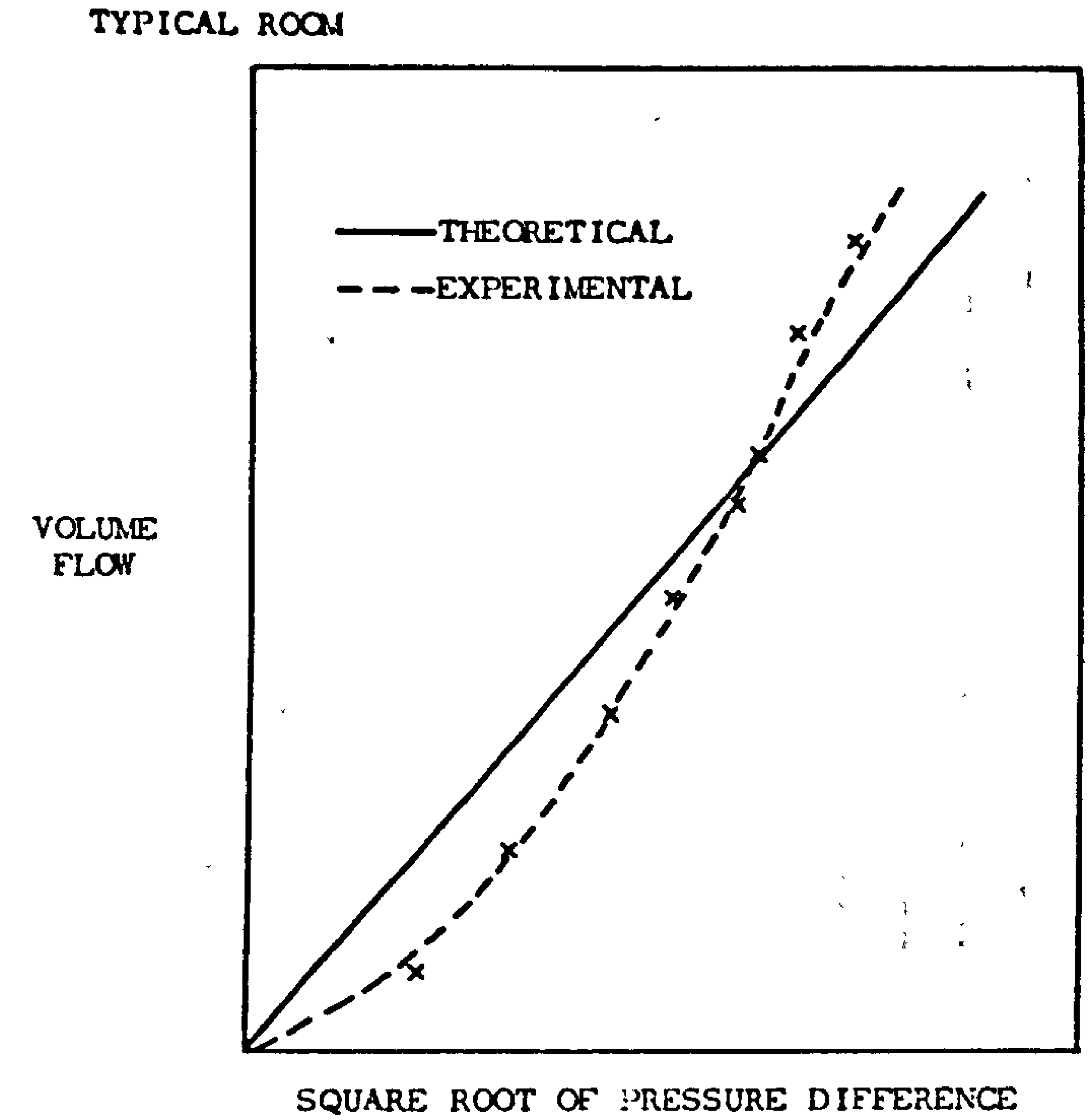


FIGURE 5.4 GRAPH SHOWING THEORETICAL AND EXPERIMENTAL VOLUME FLOW RATES AND CORRESPONDING SQUARE ROOTS OF PRESSURE DIFFERENCE (HOPKINS AND HANSFORD[10])

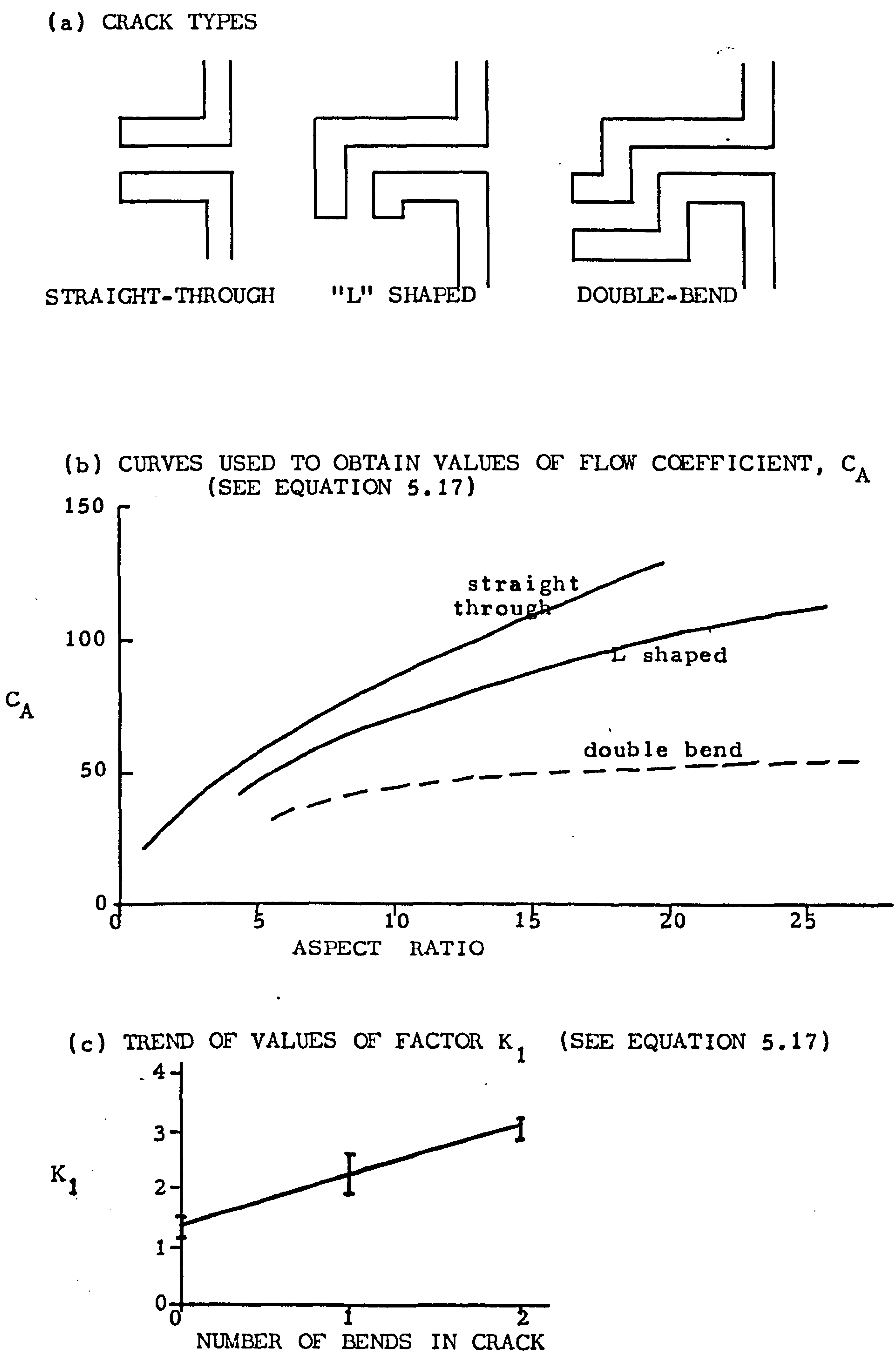


FIGURE 5.5 CRACK TYPES AND SUMMARY OF RESULTS (HOPKINS AND HANSFORD[10])

KEY:

h = height of opening
h = height of point being considered, at which the acting pressures are p_1 and p_2
h_n = "neutral height" (zero flow)
x = distance (vertical) between neutral height and point being considered

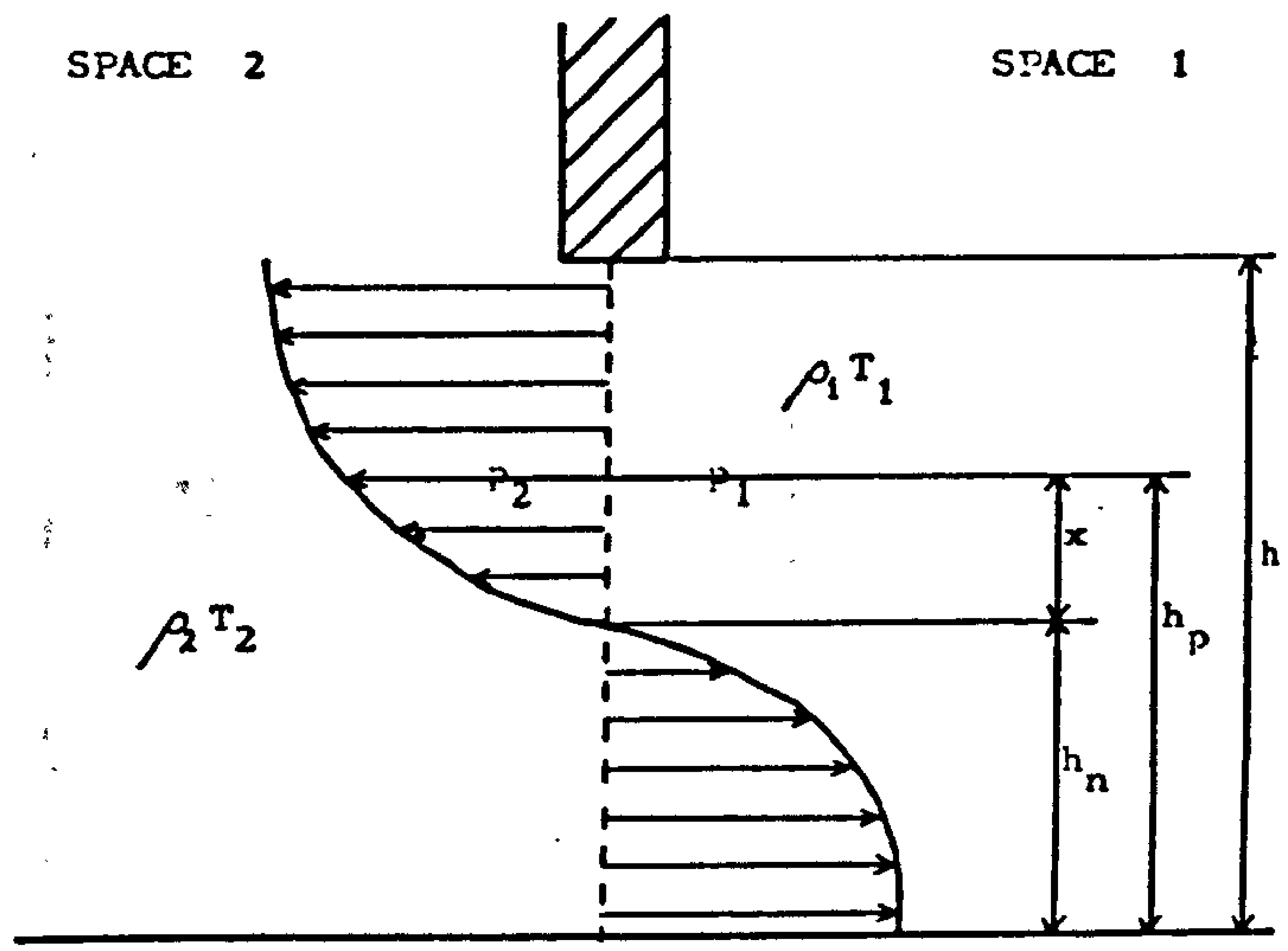


FIGURE 5.6 BI-DIRECTIONAL AIR FLOW THROUGH A DOORWAY

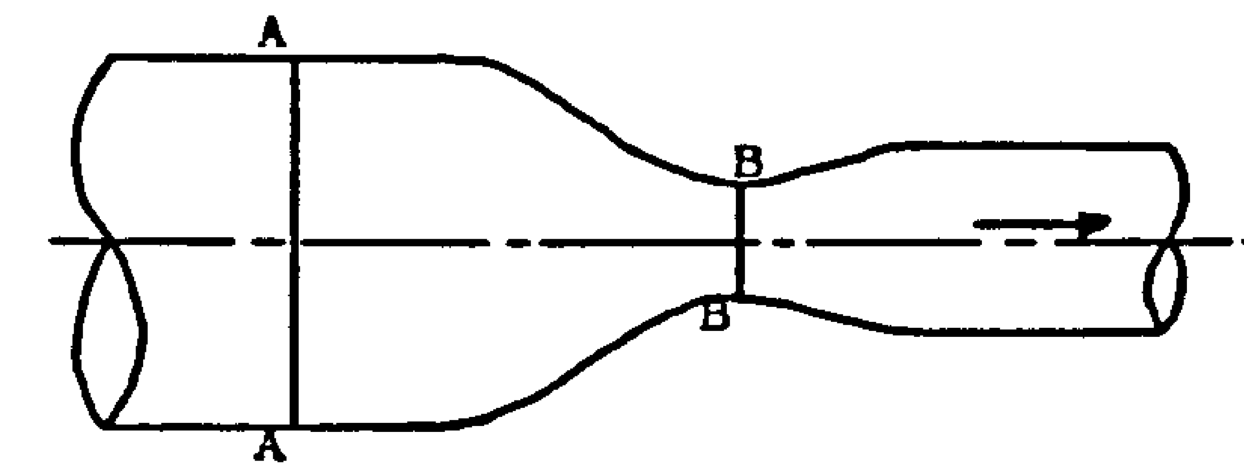


FIGURE 5.7 FLOW THROUGH A CONSTRICTION

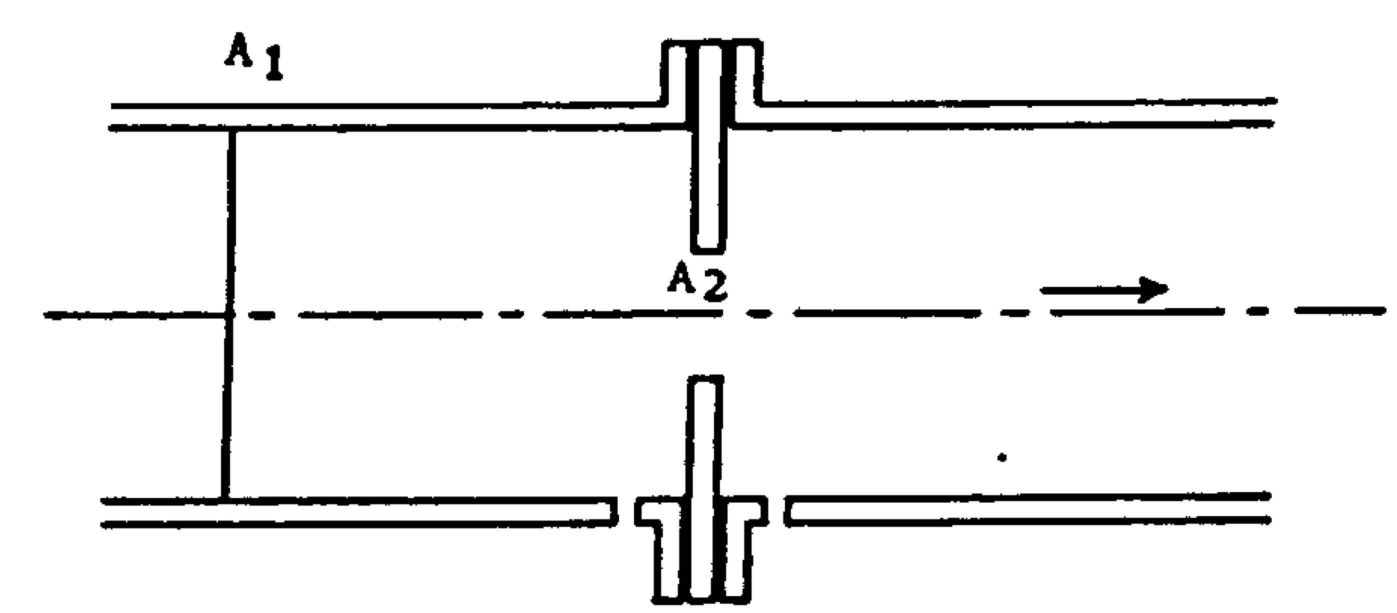


FIGURE 5.8 FLOW THROUGH A PLATE ORIFICE

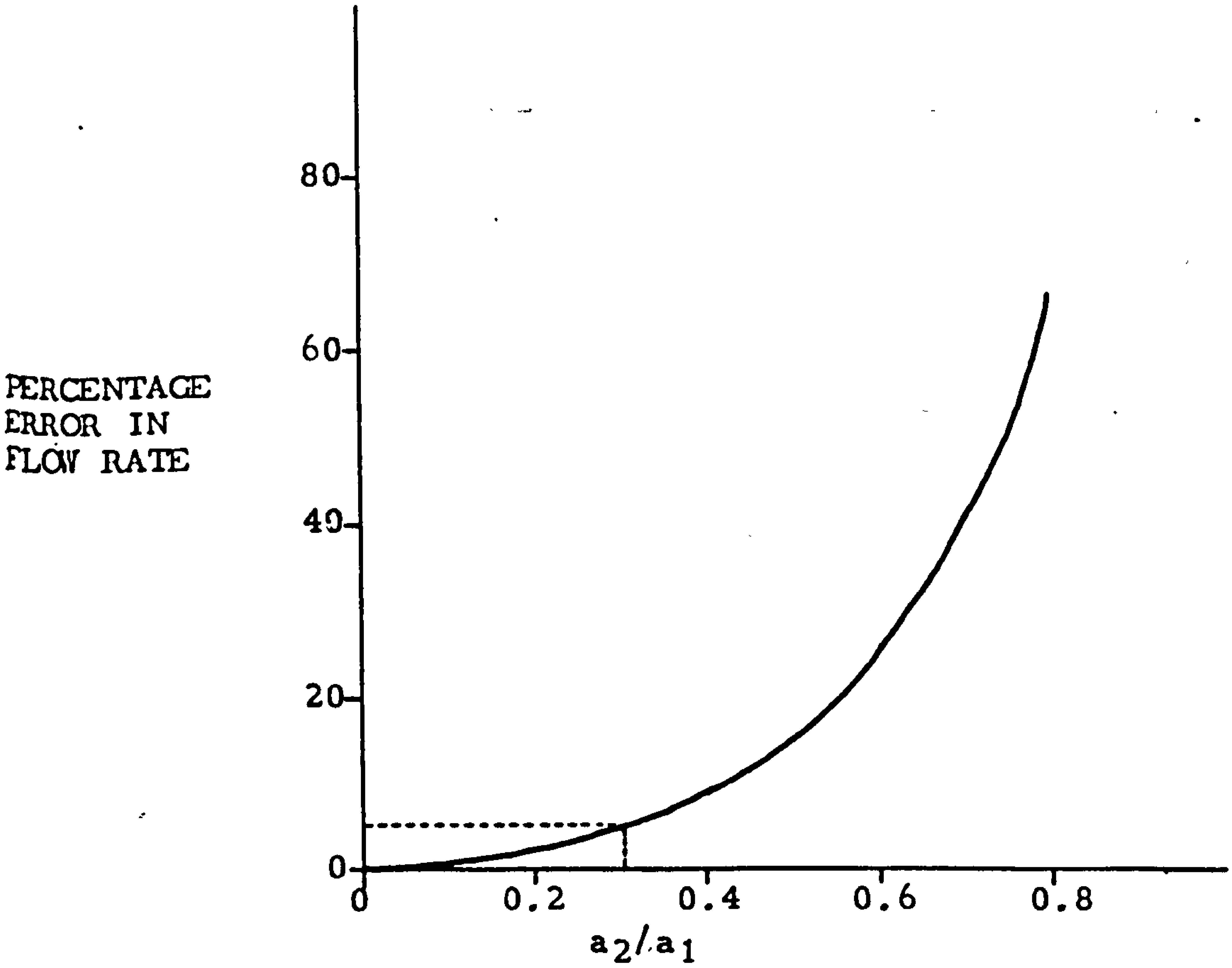


FIGURE 5.9 ERRORS IN FLOW RATE PREDICTION WHEN OPEN AREA BECOMES SIGNIFICANTLY LARGE

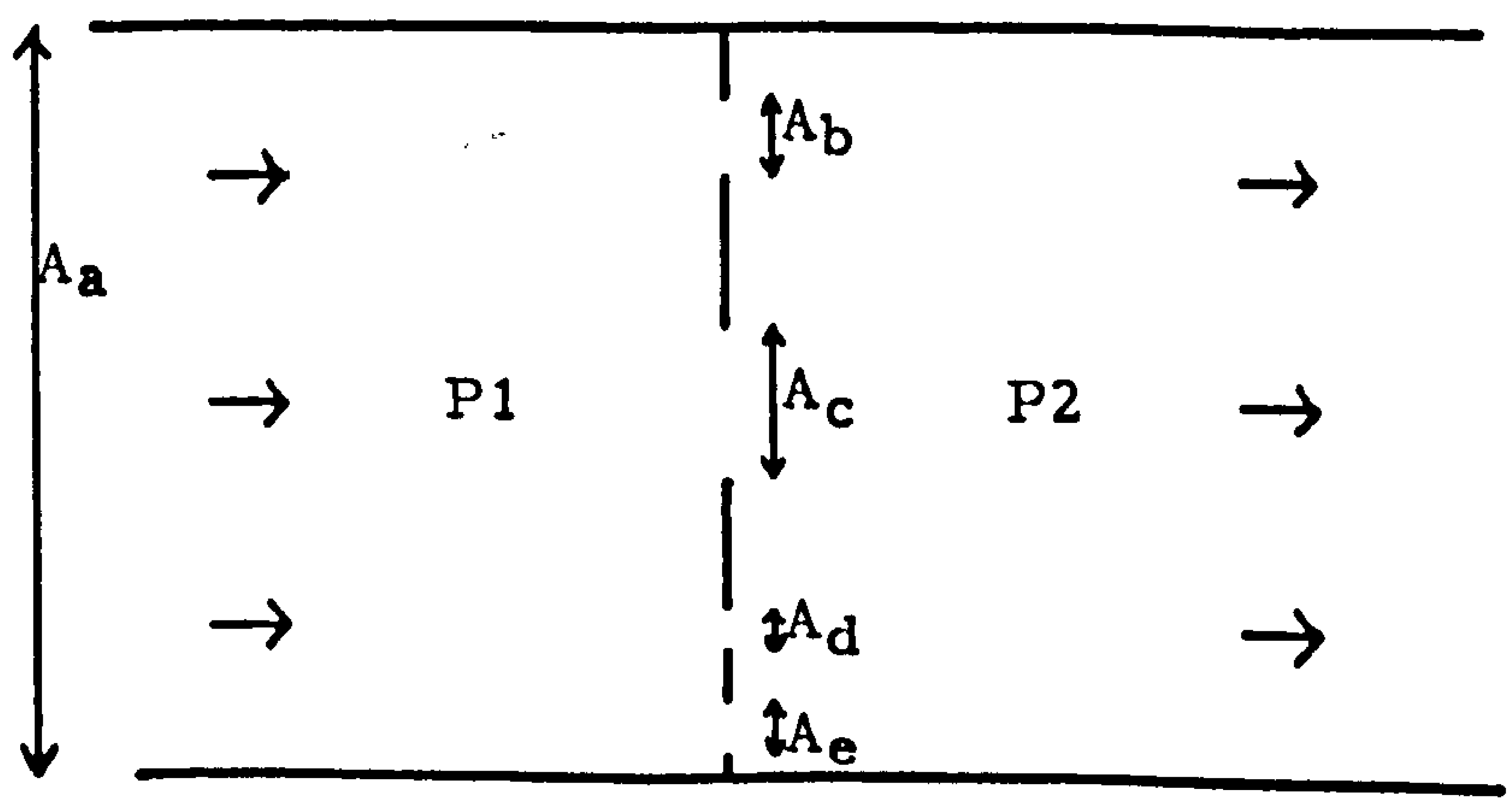


FIGURE 5.10 PARALLEL AIR FLOWS

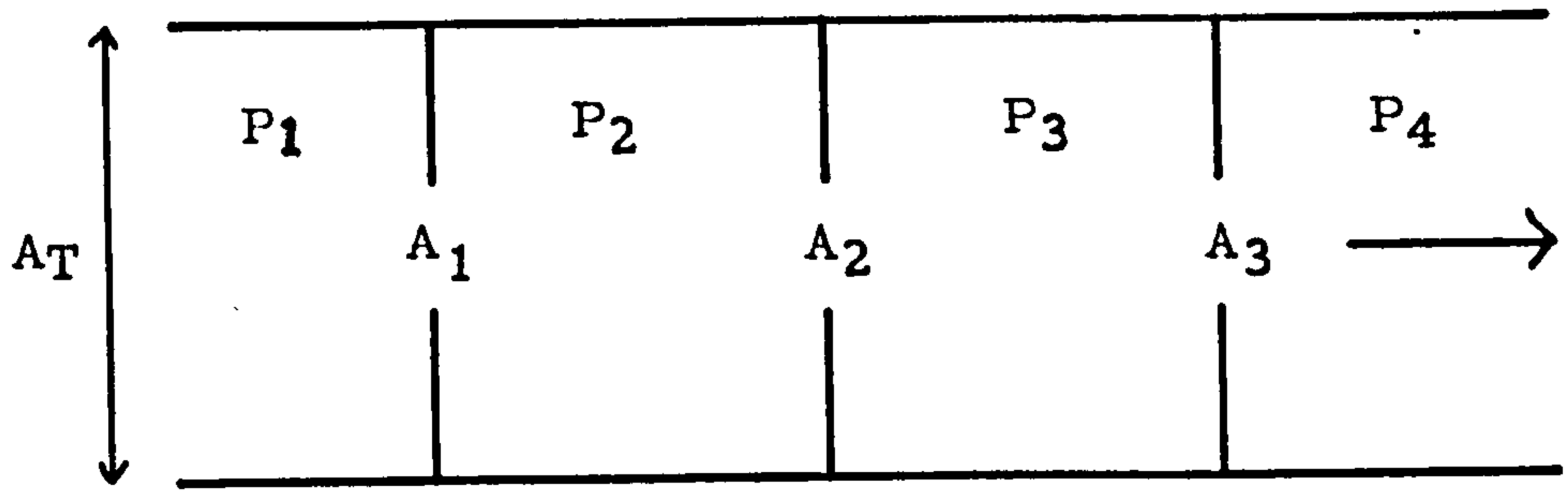


FIGURE 5.11 SERIES AIR FLOWS

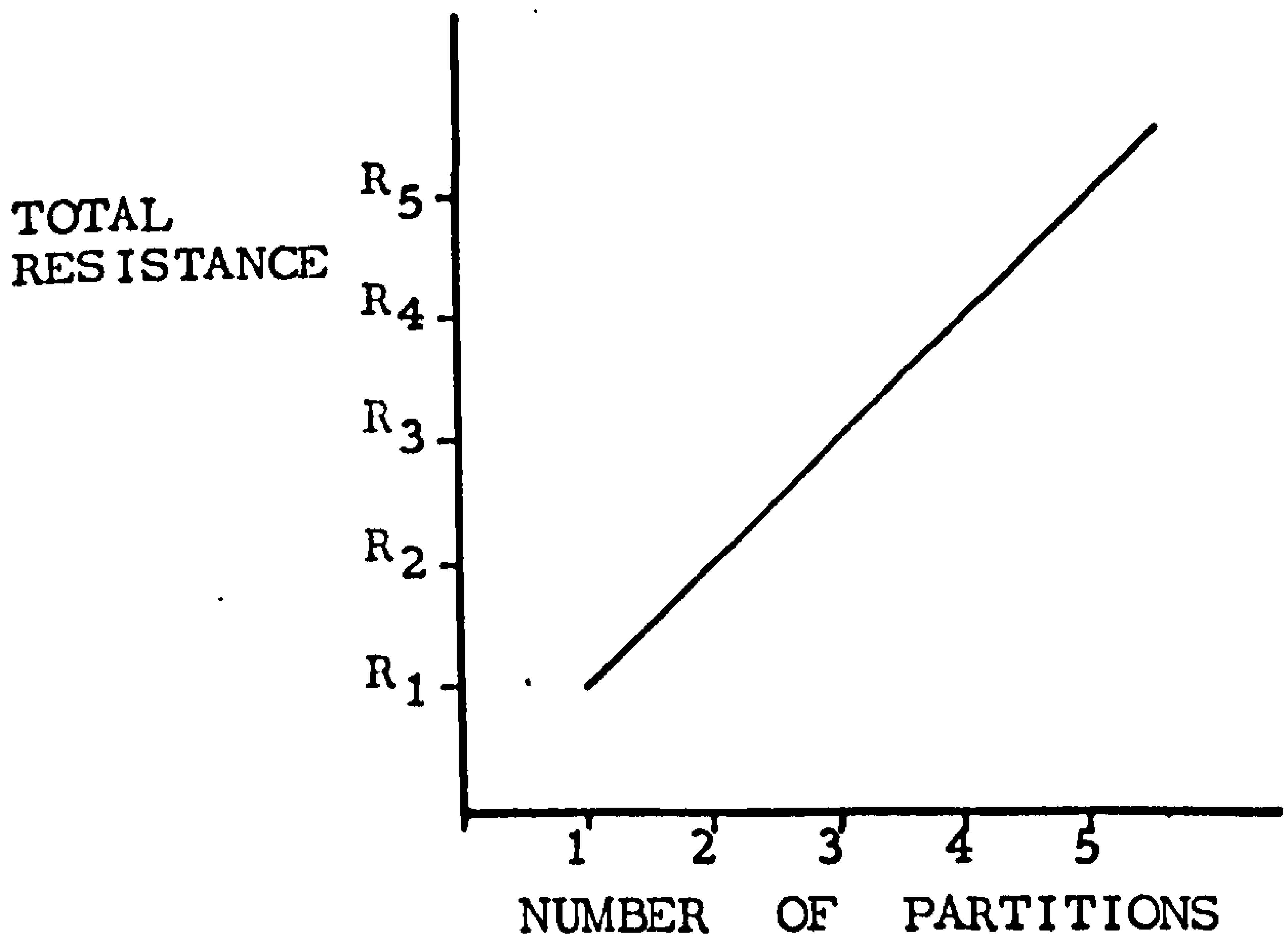


FIGURE 5.12 RESISTANCE TO FLOW
(ALL PARTITIONS EQUAL RESISTANCE)

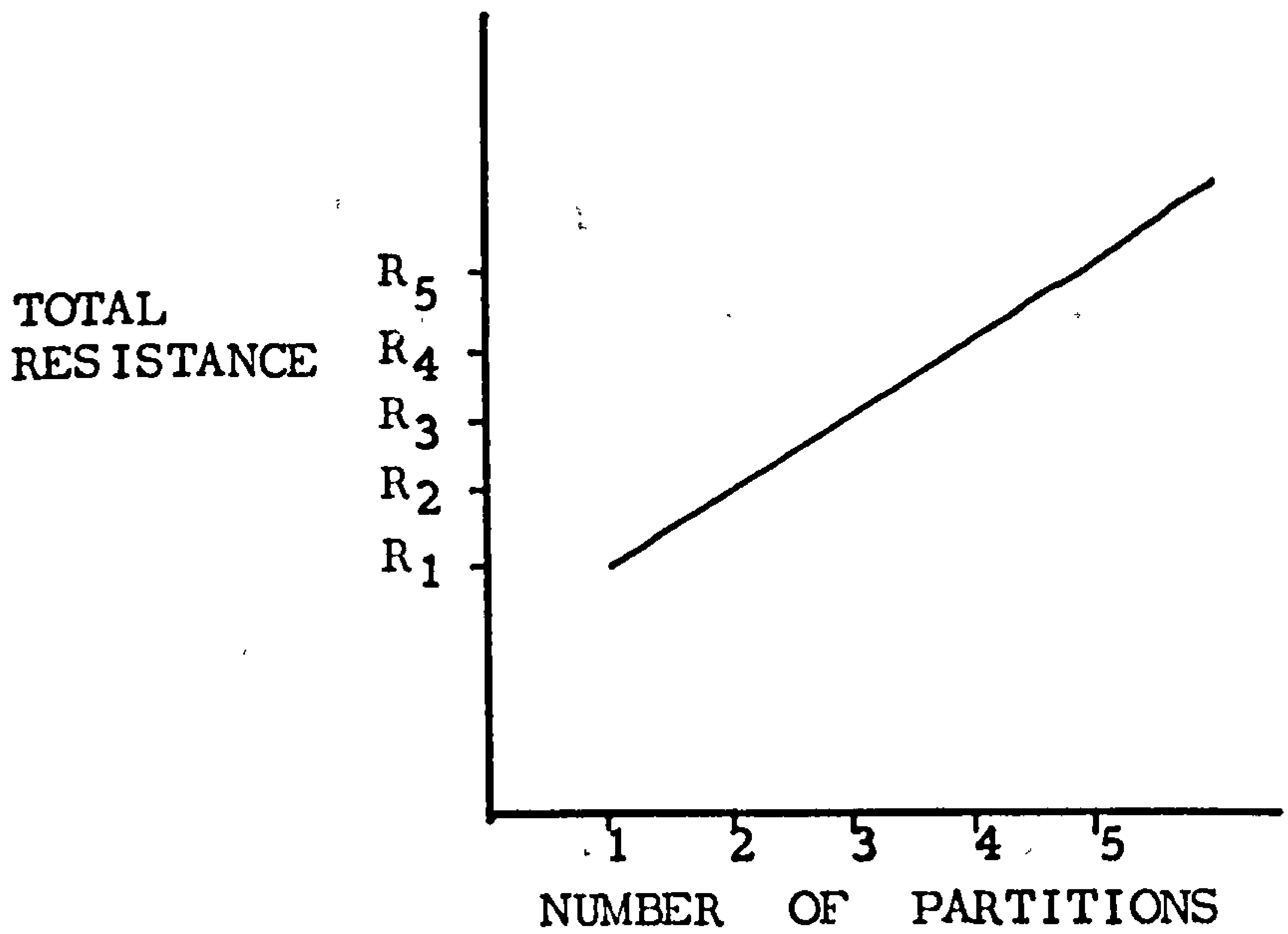


FIGURE 5.13 RESISTANCE TO FLOW
(FIRST PARTITION IS MAIN RESISTANCE,
SUBSEQUENT PARTITIONS WITH SMALLER
BUT EQUAL RESISTANCE)

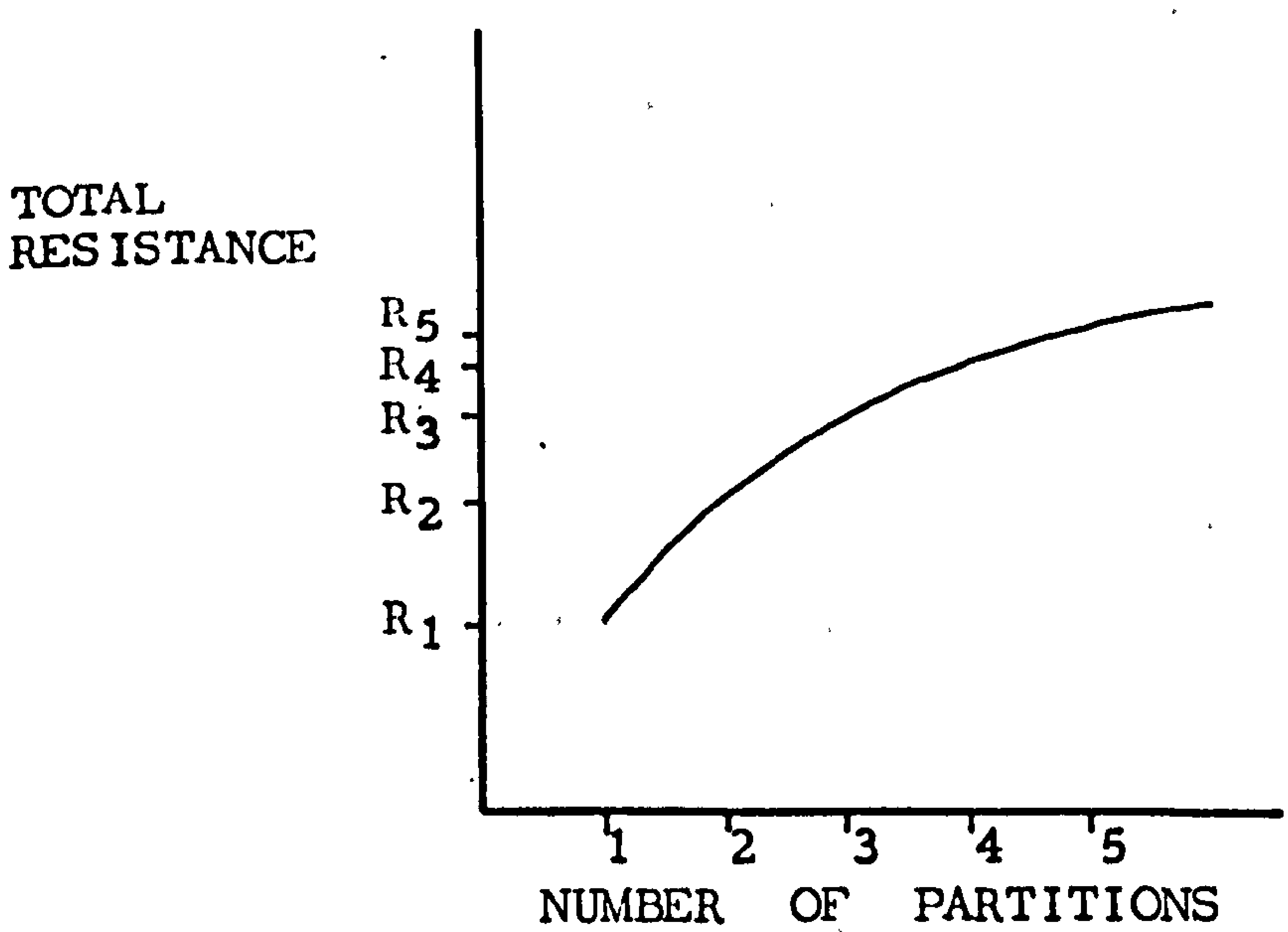


FIGURE 5.14 RESISTANCE TO FLOW
(EACH PARTITION IN THE SERIES HAVING
A RESISTANCE EQUAL TO A FRACTION OF
THE PRECEEDING PARTITION)

CHAPTER 6

MODEL SCALE TESTS: DESIGN AND DEVELOPMENT

6.1 INTRODUCTION

Tests performed in an industrial environment indicated that it was not appropriate to consider a large space containing substantial partitioning as a single "room" for air movement purposes. A review of standard mathematical methods used for air flow calculation, showed these to be lacking and unsuited to such a situation. In order to examine aspects of internal partitioning in a large space and to try to determine flow relationships, model scale tests were carried out.

Previous studies of internal air flow have shown that in most situations, air movement has a turbulent nature. The exceptions occur when very thin cracks are considered, in which the flow is assumed to be laminar. In the cases to be examined at model scale, the retention of turbulent flow was considered very important as the openings considered were not of the thin type.

The principal situation of interest was the flow through a space of rectangular cross-section due to the existence of a pressure difference. Partitions were to be located in the space, perpendicular to the flow direction. The partitions would be identical and positioned at regular intervals within the space under consideration.

The model scale tests were not carried out in order that, for example, values of air flow rate at full scale could be directly determined, but rather that they could be used to compare a number of situations and thereby establish some sort of relationship. In this way the effect of the partitioning layout upon the air flow and the importance of this effect could be used to improve the understanding of air movement in large partioned spaces.

6.2 SCALE MODELS

In order that information gathered from model scale tests is representative of the situation found at larger or full scale (ie, that the experiments are "similar" to full scale), it is attempted to keep the values of certain dimensionless parameters the same. A number of such dimensionless parameters have been defined, however in this case, since no temperature differences are to be found, the Reynolds number was chosen to be used as the basic representative parameter. It can be defined thus:

$$Re = \frac{Dv\rho}{\mu} \qquad (6.1)$$

Where D = dimension of length, m

v = flow velocity, m/s

ρ = density, kg/m^3

μ = dynamic viscosity, kg/sm

The values of the density and dynamic viscosity of the air do not depend upon linear scale and it can be seen that variations in the value of the Reynolds number are caused by variations of length or flow velocity. In this study, the length, D, is representative of the size of the flow opening in the space considered. It can be seen that in order to maintain a value of Reynolds number at a reduced scale, the flow velocity must be increased proportionally.

A range of flow rates were to be used in the model tests but these had to be sufficient to meet a criterion mentioned earlier; that is, that the type of flow be turbulent in the model - as at full scale. Turbulent flow is often regarded as occurring in flows with Reynolds numbers in excess of 3,000. However Ower and Pankhurst[1] suggested a minimum value of 100,000 was required for the greatest accuracy. By comparison Croome and Roberts[2] allowed values as low as 1,500. Generally, in the model tests conducted, the minimum value was of the order of 10,000.

6.3 MODEL BOX TESTS

The first series of experiments to be performed were carried out in a model chamber. The internal dimensions of this box were: width 1.83 m (6 ft), height 1.22 m (4 ft) and length 3.66 m (12 ft). (One of the main limitations on size was the laboratory space available). In order to facilitate viewing of the flow regimes using smoke, the chamber walls were constructed partly from clear plastic and the other parts from chipboard. To allow both ease of handling and the ability to change the dimensions of the chamber, each of the sides, base and top, consisted of a number of separate sections. These sections were constructed from a sheet of clear plastic or chipboard fixed to a supporting wooden frame. The sections were bolted together for use and all cracks and gaps sealed using heavy duty tape. The box was open at each end to allow flow.

In order to create a pressure difference and cause the air flow through the chamber, two fans were to be used which drew air through a 25 cm (10 inch) diameter duct. An inlet section to this duct was constructed to connect with one of the open ends of the chamber. This connecting section was formed by affixing polythene sheeting to a variable timber frame. Since space did not permit the construction of an 'ideal' connecting section, an empirical method was used for its design. A smoke generator was used to produce the means to visualise the flow. (This machine operates by heating a paraffin oil which is then atomized,using pressurized carbon dioxide, to form an aerosol with the appearance of white smoke. Though the 'smoke' is initially at a higher temperature than the surrounding air, it soon reaches an equilibrium). Air was drawn through the chamber, both with and without partitions of the types to be used, present. The resulting flow patterns were observed and the shape of the connecting section was modified until a satisfactory, fairly uniform flow was produced under a variety of conditions.

Two types of partition were selected for use in the model studies, each would have the same open area (equal to 50 % of the total cross section). The first type was a simple rectangular wall which extended across the full width of the model. It was half of the full height and the flow of air would occur across the top of the wall. The second type was composed of circular holes cut into a partition of the same dimensions as the full cross section. Each type of partition was constructed from sheets of chipboard. Each type of partition was to be used either singularly or in a regular combination with other similar partitions. The plain rectangular partition was to represent the type of distinct wall or barrier partitioning to be found in a variety of environments, but in this case with particular relevance to an industrial situation. The partition with circular holes would provide a basis for comparison since many flow theories have suggested that flow openings can be considered as circular holes in plates. The plain wall partitions measured 1.83 m (6 ft) by 0.61 m (3 ft). Eight circular holes were cut in the other partition at regular intervals, each of diameter 0.42 m (16.6 ins). These holes gave an open area equivalent to that of the area above the wall partition.

In order to even out flow fluctuations and external influences, a double sheet of fabric wadding was fixed across the inlet to the model chamber.

At the other end of the chamber the extract connecting section led into a 25 cm diameter duct. This turned through a 90 degree bend before reaching the straight flow measurement section. First the air flowed through a honeycombe flow straightner, then at about ten pipe diameters downstream, the measurement locations. After a further three diameters the duct opened out through a conical section to a width to accommodate the two fans. After the fans the air was discharged back into the Laboratory in which the experiments took place. Care was taken to shield the inlet to the test chamber from effects due to the discharges. Figure 6.1 shows a schematic diagram of the model chamber test layout.

6.4 EQUIPMENT

6.4.1 FANS

It was necessary to use artificial means to create flows and pressure differences within the model. This duty was performed by a two stage pair of 19 inch (48.3 cm) contra-rotating fans. These were supplied by Woods of Colchester, and were from their Aerofoil Axial Fan range using type "J" impellers. The two fans had pitch angles of 16^{o} and 14^{o}.

Solid State thyristor control circuits were fitted to each fan to allow speed, and hence flow control. The speed was continuously variable from 100 % down to 20 % - 50 % of full speed, using two dial controls.

6.4.2 FLOW MEASUREMENT

For data collection purposes it was necessary to choose a form of flow measurement device which could be easily observed, and one which would provide an electrical output signal if required. This need was satisfied by use of a vane anemometer, the version chosen being the Airflow Developments Ltd EDRA 5 unit. An easily readible analogue display of flow velocity was provided along with an electrical signal proportional to this quantity.

A problem with a vane anemometer is that whilst fairly low velocities can be measured, the reading will only represent the flow at one point in the duct. In general, due to flow reduction at points adjacent to the duct walls, a measurement in the centre of a duct, if applied to the total cross sectional area, would tend to overestimate the flow. Legg[(3) (4)] has explored ways of using a single observation to accurately estimate flow rates, but in fairly well defined environments, and usually in ducts of rectangular cross section. For the most reliable results it was decided, in this case, to calibrate the vane anemometer "in situ", against sets of flow measurements made using pilot-static tube traverses and an inclined tube manometer.

Since the duct arrangement was not ideal, the most accurate form of flow measure was to be used, to eliminate errors. The method was described by Ower and Pankhurst[1] and consisted of two 10-point log-linear traverses, on perpendicular axes. This method was originally devised by Winternizt and Fischl and was recommended by the British Standards Institute as yielding Class A accuracy.

The mean velocity of air flow, Vm, is given by

$$Vm = k\sqrt{hm} \qquad (6.2)$$

where h_m = mean velocity pressure (cm H_2O) from manometer

$$k = 20.56 \; \frac{273 + \text{Temp} (^{o}C)}{\text{Barometric Pressure (mmHg)}}$$

$$h_m = \frac{1}{n} (h_1 + h_2 + h_3 + \ldots + h_n) \qquad (6.3)$$

where n = number of reading points

h = velocity head

Twenty readings were taken using the pitot-static tube at each flow rate. Figure 6.2 shows the measurement positions on each axis. Eight different flow rates were set using the fan control circuits. These ranged from 15 % to 100 % of the maximum, though the lowest flow rate was only achieved by the switching off completely of one of the stages. Whilst the traverses were being carried out, the measured flow velocity from the vane anemometer was being output and recorded on a chart recorder. When the measurements had been completed the mean air velocity was calculated from the pitot-static tube readings using the above equations. The average flow velocity measured by the vane anemometer was also calculated for the period of each flow rate. The results are summarised in Table 6.1.

TABLE 6.1

FLOW VELOCITIES: VANE ANEMOMETER AND DUCT AVERAGE

FAN CONTROL CIRCUIT SETTINGS		MEASURED FLOW VELOCITY, m/s (VANE ANEMOMETER)	AVERAGE DUCT FLOW VELOCITY, m/s (PITOT-STATIC TRAVERSE)
1ST STAGE	2ND STAGE		
10	10	20.7	16.92
7	7	17.3	13.96
6	6	13.4	11.02
5	5	10.1	8.16
4	4	7.8	6.25
2.5	2.5	6.2	4.91
0	0	5.2	4.08
0	Off	3.25	2.52

The results exhibit a good linear relationship as can be seen from Figure 6.3 (Correlation Coefficient = 0.99988).

$$\underset{\text{VELOCITY (m/s)}}{\text{ACTUAL MEAN FLOW}} = \underset{\text{VELOCITY (m/s)}}{\text{MEASURED FLOW}} \times 0.812 \qquad (6.4)$$

6.4.3 PRESSURE MEASUREMENT

As with the flow measuring device, the means of pressure measurement was required to give an output which could easily be recorded. Some electrical device was therefore in order. A transducer, originally designed by Mayne[6] for the measurement of wind pressures on the external surfaces of buildings, was available.

The basic transducer consists of a pressure plate supported by three small cantilevers. Four foil resistance strain gauges are attached to each of the cantilevers. As the plate moves, due to pressure fluctuations, the cantilevers bend causing compression and tension in the strain gauges. These strain gauges are wired to form a Wheatstone bridge, each arm is made up by three gauges in series from corresponding positions on each cantilever. When the bridge is energized, variations in the strain gauge resistances, due to movement in the pressure plate, will cause measurable variations in the output signal from the bridge. A fuller description of the basic transducer is given in reference (6).

This transducer allowed measurement of the ambient pressure on the front of the pressure plate, referenced to a back pressure, applied through the venting nozzle on the rear of the casing. However for the work to be undertaken, a pressure differential measurement was required. This need was met by a modified form of the transducer developed in the Department of Building Science at the University of Sheffield. In this the membrane clamp ring is removed, and replaced by a front cover plate containing a venting nozzle, similar to that in the back plate. In this way an airspace is created on each side of the pressure plate and membrane; each airspace being accessed via a venting nozzle.

The pressure transducer will produce a signal if an energizing voltage is applied across the input terminals. Since the measurement of this signal was so important it was investigated in some depth. Known pressures could be applied by laying the basic transducer in a horizontal plane, and positioning weights on the pressure plate. The weight divided by the area of the plate gives the average equivalent pressure.

In the first case the raw signal from the energized Wheatstone bridge circuit was monitored. It was found to take between ½ an hour and an hour for the reading to stabilize and even after this period some zero drift was noted. The sensitivity was also low, being about 0.01 mV per Pascal. Since low pressure differentials were to be measured, it was clear that some amplification of this signal was required.

When the pressure transducers had been originally purchased by the department, power supply and amplification units had also been obtained. The amplifier was an Electro Mechanisms Ltd type DSL 4.

An investigation using this set up showed a much stronger output signal, being approximately 0.8 mV per Pascal. However this improvement was negated by other problems with the amplifier, in particular the zero drift problem was exacerbated and other unexplained errors occurred. Because of these problems, it was decided to choose a new amplifier which could be used with greater reliability.

The option chosen was to build a circuit based upon the "Radio Spares" Strain Gauge Amplifier. A suitable circuit was to be found in one of the Radio Spares Data Sheets[7]. This circuit is given in Figure 6.4, it was constructed in the departmental workshop using a circuit supply voltage of $\pm$ 12 V. After some initial problems, the circuit was found to operate very well. The bridge supply excitation voltage was set at 10V, and at this level the circuit produced an output signal of about 4 mV per Pascal. It was still clear however that there were likely to be small movements in the zero, so it was decided that before and after each measure of pressure, a reading of the zero ought to be taken. The first case in which this system was used, was the accurate calibration of the transducer.

The transducer was laid on a purpose built tripod, the level of which was adjusted until the pressure plate was horizontal (checked using a spirit level). Precision weights were then placed on the plate and the voltage output from the amplifier noted, before, during and after the weight was in place. Five weight levels were used in sequence and the whole procedure was repeated three times. The change in output signal due to the placement of each of the weights was averaged for the three test sequences. This figure was then compared with the equivalent pressure levels. The data is summarised in Table 6.2 and shown in Figure 6.5.

TABLE 6.2

PRESSURE TRANSDUCER CALIBRATION

WT	AMPLIFIER OUTPUT (mV)							EQUIVALENT PRESSURE (Pa)
	TEST A		TEST B		TEST C		AVERAGE CHANGE	
	SIGNAL	CHANGE	SIGNAL	CHANGE	SIGNAL	CHANGE		
0g	0.7		1		2			
5g	25	24	25	24	26	24	24	6.24
0g	0.6		1		2			
10g	49.4	49	50	49	50	48	49	12.49
0g	0.7		1		2			
20g	98.4	98	99	98	99	97	98	24.97
0g	0.7		1		2			
50g	247	246	247	246	247	245	246	62.43
0g	1.2		2		2			
100g	494	492	496	493	497	494	493	124.87
0g	3		4		4			

The diameter of the pressure plate is 10 cm, thus its area is 0.007854 m^2. The equivalent pressure is found by dividing the force due to the weight applied by this area.

That is:-

$$\text{Pressure (Pascals)} = \frac{\text{(weight).g}}{\text{area}} \qquad (6.5)$$

where g is the acceleration due to gravity = 9.81 m/s^2

Thus:-

$$\text{Pressure (Pascals)} = \text{weight in grammes} \times 1.249 \qquad (6.6)$$

The output signal varies linearly with the pressure as can be seen from Figure 6.5. The relationship is:-

$$\text{Pressure (Pascals)} = 0.255 \times \text{signal (mV)} \qquad (6.7)$$

(correlation coefficient 0.999)

This calibration was also checked using small weights during the tests and was fully rechecked at the completion of the trials. The variation was found to be less than 1% which suggested that the signal and calibration could be relied upon.

6.5 METHOD OF DATA COLLECTION

6.5.1 CHART RECORDER

The signals to be measured were recorded as pen traces on a Linseis, type 7060, chart recorder. Upto six measurements could be made at any one time. A range of voltage scales and chart speeds could be chosen from. The recorded traces were checked for accuracy by using test signals which were also measured in parallel by a digital multi-meter.

6.5.2 PRESSURE MEASUREMENT POINTS

In order to determine the most suitable positions between which to measure the pressure differentials a number of trial runs were carried out. For orifice plates used in flow measurement, Sprenkle[7] and Ower and Pankhurst[1] provided details of six types of measurement positions:

a) Vena Contracta Taps; these are located one pipe diameter preceeding the orifice, and in the plane of greatest jet contraction (the "vena contracta") following the orifice.

b) 1D and ½D Taps; these are sometimes called "radius taps" and are located in a similar position to the vena contracta taps, but are always one diameter preceeding and a half a diameter following the orifice.

c) Flange Taps; the taps are placed in the holding flange, one inch on either side of the orifice.

d) Pipe Taps; the connections are two and a half pipe diameters before, and eight pipe diameters after the orifice.

e) Corner Taps; these are more often used in Europe and open into the pipe in the corner where the pipe meets the plate in which the orifice is located.

f) Annular Taps; similar to the corner taps but formed from two annular chambers which are located in the corner between pipe and orifice plate, thus averaging slight variations.

None of these however, appeared to provide an ideal solution. The annular taps might have given the best results, considering the far from normal tests being carried out. The construction of a fitting to enable this in the variable partition layout planned would not have been possible. First of all single tapping positions were investigated using various partition options. These tappings were located at a variety of positions up-and downstream and sets of readings obtained. The trials for certain layouts were repeated and their consistency checked. Overall these trials did not yield particularly good results so another option was tried.

An averaging mechanism was sought, and this was achieved by taking four tappings at both the upstream and downstream position. These tappings were mounted flush with the model inner surfaces and placed at the midpoint of each - (ie middle of floor, roof and both side walls). After trials it was decided that these tappings could be left in one set of positions for all the tests, these being just after the inlet and at the outlet of the model chamber. A further aspect which favoured such measurement positions was that they represented the positions at full scale between which the pressure difference would operate i.e. between the two furthest extents of the space.

6.5.3 FLOW RATE RECORDING

It was found during the twenty trial test runs carried out to check the system, that the chart record of vane anemometer flow velocity was unnecessary. Once the fan speed had been set, and the flow allowed to stabilize, a virtually constant flow velicity was observed, which could easily be noted on a record sheet.

6.5.4 TEST ROUTINE

A set routine was followed for the performance of each experiment. Before any experiments were carried out the pressure transducer amplifier and chart recorder were switched on and allowed to "warm-up" for an hour. After this period the drift of the zero signal from the amplifier was checked; when this found satisfactory, the tests began. The following start sequence was used:

a) Measure ambient temperature and pressure.

b) Set chart recorder moving - usually at a speed of 1 cm/min.

c) Switch on vane anemometer.

d) Set chart pen to mid-position on the paper giving a scale of $\pm$ 10 mV.

e) View pen recording and other equipment - check all in order.

At this time the partition(s) would have been placed in position in the test chamber and the test run could commence. This took the following sequence:

i) Select control settings for fan speed.

ii) Switch on fans.

iii) View chart recording and vane anemometer reading - when steady, proceed.

iv) Mark chart - start of measurement period.

v) View vane anemometer for approximately one minute checking that the reading is steady.

vi) After minute note vane speed on chart and mark chart for end of measurement period (also note vane speed in log book).

vii) Switch off fans.

viii) View vane speed and fans - allow flow to return to zero and chart reading to settle (chart shows new zero if this has altered at all).

ix) Repeat (i) to (viii) for approximately sixteen different flow settings.

Test runs were carried out for a variety of different partition numbers and spacings. The cases tested are given in Chapter 7.

6.6 "WIND TUNNEL" TESTS

After the results of the model air flow chamber tests had been collected and analysed, it was decided to extend the testing of the plain wall partitions. One of the aerodynamic wind tunnels in the Department of Building Science offered the most suitable facility, and since it had been designed specifically for aerodynamic tests, it was envisaged that a reliable set of data would be produced.

6.7 EQUIPMENT

6.7.1 WIND TUNNEL

The wind tunnel used had a working section with a 0.61 m (2 ft.) square cross section. It was mainly of wood construction with two perspex panels fitted along one side (mounted to be flush with the inner surface) to allow viewing. The working section was approximately 3.6 m in length and this was preceeded by further square cross section area which provided a narrowing channel from the shaped inlet. The inlet contained a honeycombe flow straightener and grilles to prevent unwanted items from being drawn into the tunnel. After the working section the tunnel led to a variable speed fan which was used to draw air through the tunnel. Between the working section and the fan were further grilles to catch material which could damage the fan if drawn-in. The control for the fan speed setting had quite a wide range, but for the tests to be carried out it was established that the lower end of the scale would provide quite sufficient scope for the experiments.

6.7.2 PARTITIONS

The plain rectangular partitions to be placed in the tunnel were made from thin steel plate, and were each 0.305 m x 0.61 m (1 ft x 2 ft) in size. In order that these could be positioned both variably and securely within the tunnel a method of attachment was devised. This consisted of four thin steel runners bolted to the sides and bottom of the tunnel, and which covered the full length of the working section. These runners were constructed so that the partitions could be fixed at any of the required positions by screwing into the four runners at that position. Though this method proved time consuming when positions were changed, it was the only one available which kept the partitions secure in place and did not interfere with the air flow.

6.7.3 PRESSURE MEASUREMENT

A new low pressure transducer had become available for use in these experiments. It was a type FC040 produced by Furness Controls Limited. The version used covered the range of differential pressures between 0 and 10 mm water gauge (0-98 Pa). The transducer required a type M0177 Power Supply, which provided $\pm$ 15 V dc input voltage; the output from the transducer being 0 - 1 V dc calibrated to represent 0 - 10 mm water gauge pressure. The unit was mounted vertically with the two pneumatic connections pointing downwards, as recommended by the manufacturers.

The zero point of the unit was checked as was the zero drift. There was practically no zero drift to be found indicating the equipment to be much more useful than the previous transducer and amplifier used. The factory calibration of the range was also viewed against an inclined tube manometer (though the transducer gave a much higher resolution than could be adequately checked by these means).

The positions chosen for the measurement of the pressure differential were similar to those used in the previous set of tests. That is at either end of the working section. Initial trials indicated fairly steady readings and as a result only one tapping was used at each of the upstream and downstream sides. This was mounted to be flush with the side walls of the tunnel at a height equal to that of the partition walls.

6.7.4 FLOW MEASUREMENT

For the same reasons as outlined in section 6.4.2 a vane anemometer was to be used for flow measurement in the wind tunnel. It was mounted in the centre of the duct at the beginning of the working section. In order to take account of variations in flow across the duct, the velocity recorded at the centre of the duct must be compared with the total average flow (this was also discussed in section 6.4.2). In this case the duct is of square cross-section and the method of calibration was the 26 point log-linear version described by Ower and Pankhurst[1], though this was originally devised by Myles, Whitaker and Jones. The flow was set up in the duct with the vane anemometer in position, then a pitot-static tube was used with an inclined tube manometer to determine the flow at each of the 26 points on the measuring grid. (These points are shown in Figure 6.7). The velocity of the air at each measurement position is given by

$$v = k \sqrt{h} \qquad (6.8)$$

where k is as in equation 6.2

h is the velocity head

The average duct velocity is calculated thus

$$Vav = \frac{1}{96} (2v_1 + 2v_2 + 5v_3 + 6v_4 + 5v_5 + 2v_6 + 2v_7 + 3v_8 + 3v_9 + 6v_{10} + 6v_{11} + 3v_{12} + 3v_{13}$$

$$+ .3v_{14} + .3v_{15} + .6v_{16} + .6v_{17} + .3v_{18} + .3v_{19}$$
$$+ .2v_{20} + .2v_{21} + .5v_{22} + .6v_{23} + .5v_{24} + .2v_{25} + .2v_{26}) \quad (6.9)$$

Where the subscripts 1 - 26 refer to positions in Figure 6.7.

The averaged flow was then compared with the flow indicated by the vane anemometer. Figure 6.8 shows the possible partition locations. In the extreme case the upstream partition would be quite close to the flow measurement position. Because of this the leading partition could affect the preceeding flow pattern and flow measurement, therefore it was decided to calibrate the vane anemometer using several different positions for the leading partition. In this way any variations could be incorporated into the vane anemometer flow correction multiplier. Tables 6.3 a,b,c and d, indicate the results obtained, and these are represented graphically in Figures 6.9 - 6.12. (correlation coefficients of at least 0.999)

TABLE 6.3

FLOW VELOCITIES: VANE ANEMOMETER AND DUCT AVERAGE

a) NO PARTITION

Measured centre point flow m/s (Vane Anemometer)	Average duct flow, m/s (Pitot-Static Tube Traverses)
13.33	11.63
11.12	9.69
8.71	7.47
6.25	5.25
3.81	3.26

b) PARTITION AT POSITION O (See Figure 6.8)

Measured centre point flow, m/s (Vane Anemometer)	Average duct flow, m/c (Pitot-Static Tube Traverses)
6.65	5.58
5.60	4.68
3.75	3.33
2.25	2.04

c) PARTITION AT POSITION 6 (See Figure 6.8)

Measured centre point flow, m/s (Vane Anemometer)	Average duct flow, m/s (Pitot-Static Tube Traverses)
6.75	5.65
5.35	4.41
3.40	2.95
2.20	1.85

d) PARTITION AT POSITION 12 (See Figure 6.8)

Measured centre point flow, m/s (Vane Anemometer)	Average duct flow, m/s (Pitot-Static Tube Traverses)
7.05	5.89
5.65	4.66
3.65	3.10
2.30	1.91

Using "least-squares" technique with the ducts, a linear relationship with a very good correlation coefficient was found for each set of results. With the line constrained to pass through the origin the following corrections to the vane anemometer, centre point measured, flow velocities were produced:

a) No partition -

AVERAGE FLOW = 0.866 (VANE ANEMOMETER FLOW) (6.10)

b) Partition position 0

AVERAGE FLOW = 0.849 (VANE ANEMOMETER FLOW) (6.11)

c) Partition position 6

AVERAGE FLOW = 0.837 (VANE ANEMOMETER FLOW) (6.12)

d) Partition position 12

AVERAGE FLOW = 0.834 (VANE ANEMOMETER FLOW) (6.13)

These figures show that as the distance of the first partition from the vane anemometer decreased there was an increased tendency for the vane anemometer to over estimate flow and so the correction multiplier decreased. In the trials carried out the effect of the nearest partition to the vane amemometer was compensated for, by using a variable correction factor based upon the results above.

6.7.5 CHART RECORDER

A Vitatron, two pen, chart recorder was used to take a permanent record of the data. As in the earlier tests, it was found unnecessary to continuously record the vane anemometer flow rate as this remained almost constant throughout each test flow rate. The main measurement taken therefore, was the output from the Furness pressure transducer. The chart speed used was 1 cm per minute and the usual scale range was 0 - 200 mV. The recorded output was checked using a digital multimeter.

6.8 TESTING AND OPERATION

6.8.1 TESTING

All the equipment was tested individually and when all the items had been assembled, a number of trial test runs were carried out to check performance which was found to be quite satisfactory.

6.8.2 TEST ROUTINE

At the start of each experiment the pressure transducer circuit and chart recorder were switched on and allowed a warming-up period. At this time the partitions would be located and fixed into position in the wind tunnel. The following steps were then followed

(a) Measure ambient temperature and pressure.

(b) Switch on vane anemometer circuit.

(c) Set chart recorder moving.

(d) Check pressure transducer zero and make any adjustment to chart zero.

When these actions had been carried out the test run commenced

(i) Set fan speed control to zero.

(ii) Check equipment.

(iii) Switch on fan and allow to stabilize (some flow found even at zero setting).

(iv) When flow steady mark chart (start of reading)

(v) View vane anemometer for approximately one minute and note flow reading in log book. Also note setting on chart.

(vi) Mark chart again (end of reading).

(vii) Increase fan speed control by set fraction. (N.B. No requirement to return to zero as in previous tests because of improved pressure transducer stability.)

(viii) Repeat (iv) - (vii) approximately 14 times.

(ix) Reset fan speed control to low value.

(x) Carry out (iv) - (vii) approximately 5 times but with larger speed increments.

(xi) Fan speed control to zero and switch off.

The second set of five readings was made to increase the accuracy of each test run as a whole and also to act as a comparison to the first set of readings to check for time dependent variabilities.

The range of partition layouts tested and the results of these experiments are given in the following chapter.

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Paper Bl - Symposium Flow Measurement in Closed Conduits

H.M.S.O. Edinburgh 1962.

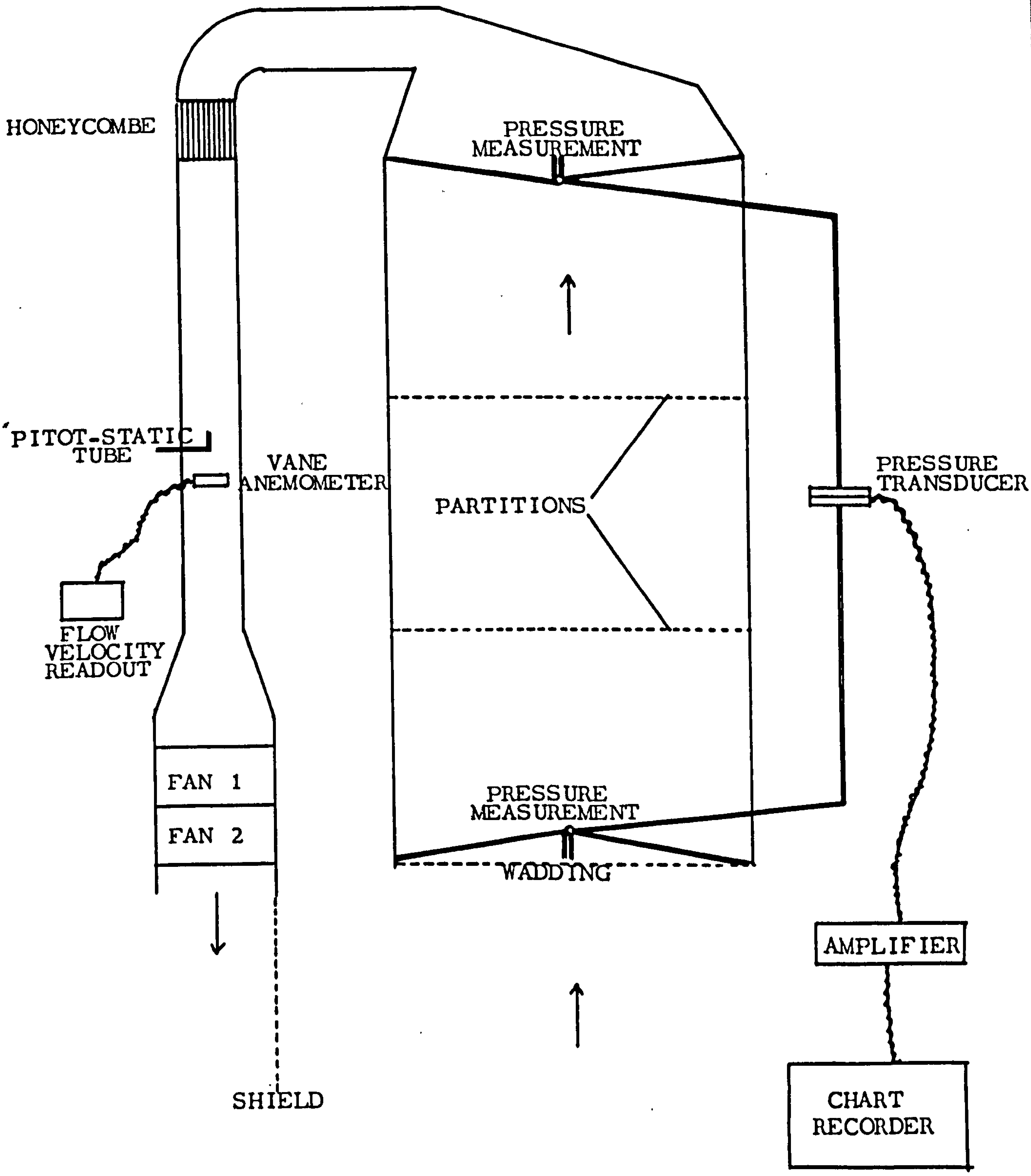


FIGURE 6.1 SCHEMATIC PLAN OF MODEL AIR FLOW CHAMBER

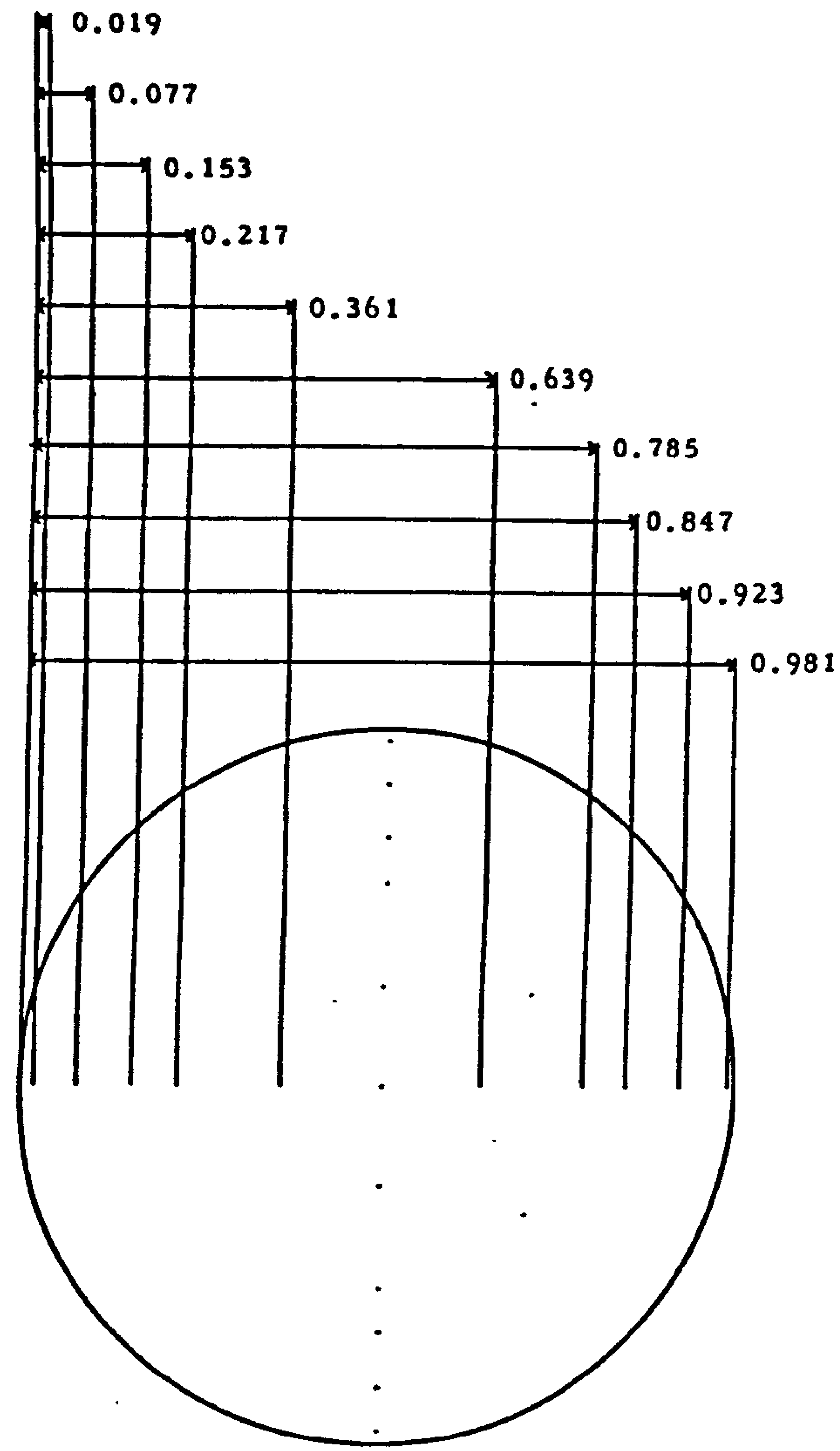


FIGURE 6.2 MEASUREMENT POINTS FOR PITOT-STATIC TUBE TRAVERSES

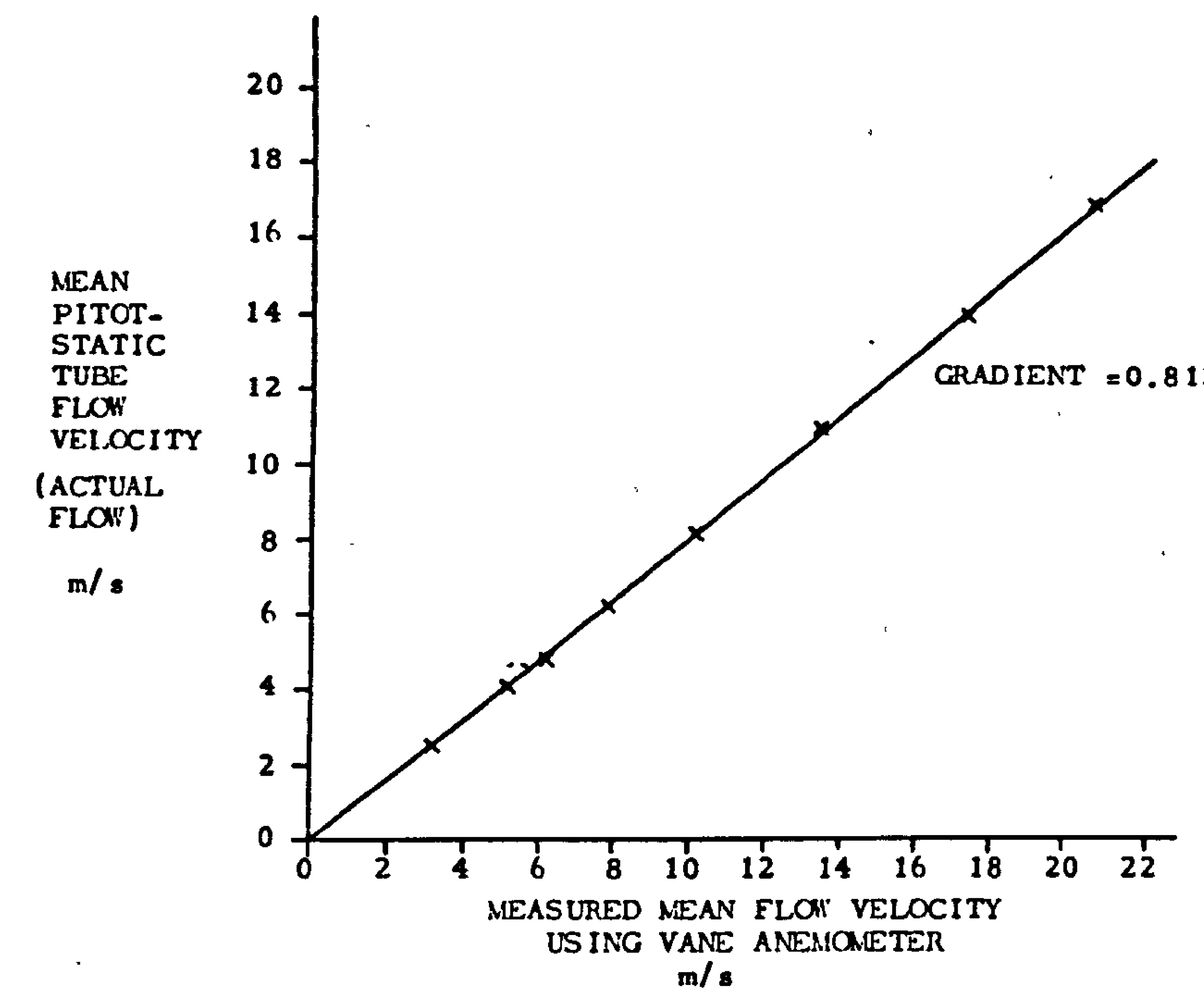
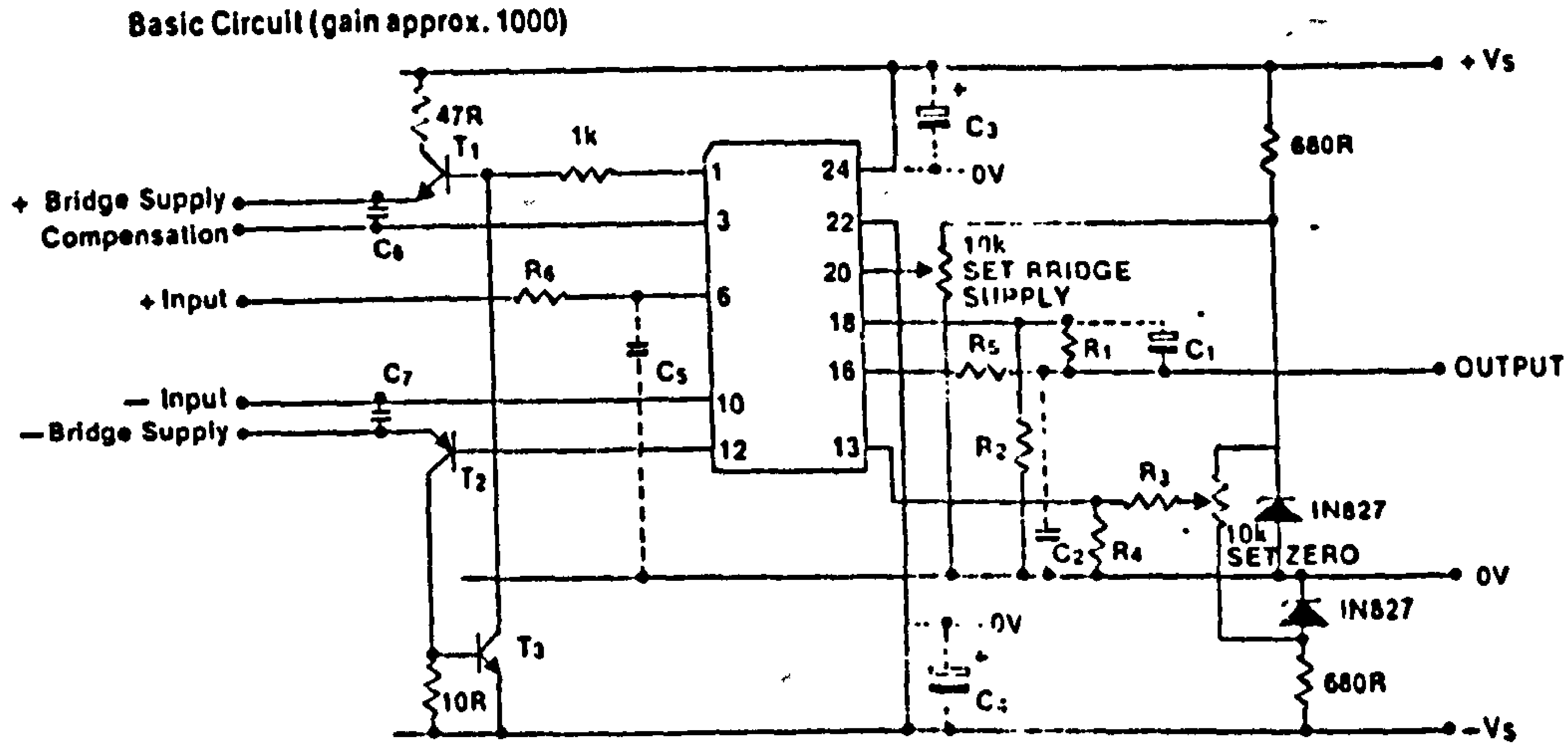


FIGURE 6.3 CALIBRATION LINE : VANE ANEMOMETER VELOCITY AND ACTUAL AVERAGE FLOW VELOCITY (FROM PITOT-STATIC TUBE TRAVERSE OBSERVATIONS)



Component Values:

R_1 100k	C_1, C_6, C_7 100n (typ.)
R_2 100R	C_2, C_5 10n (typ.)
R_3 100k*	C_3, C_4 10μ (tant.)
R_4 68R*	T_1 BD135
R_5 10R	T_2 BD136
R_6 100R (typ.)	T_3 BC108

Only typical values are given for certain components, as adjustment of these values may be necessary in specific applications to obtain optimum noise reduction (see Minimisation of Noise, page 5).
*R_3 and R_4 values may be adjusted to alter the zero adjustment range when compensating for bridge imbalance.

FIGURE 6.4 STRAIN GAUGE BRIDGE AMPLIFICATION CIRCUIT

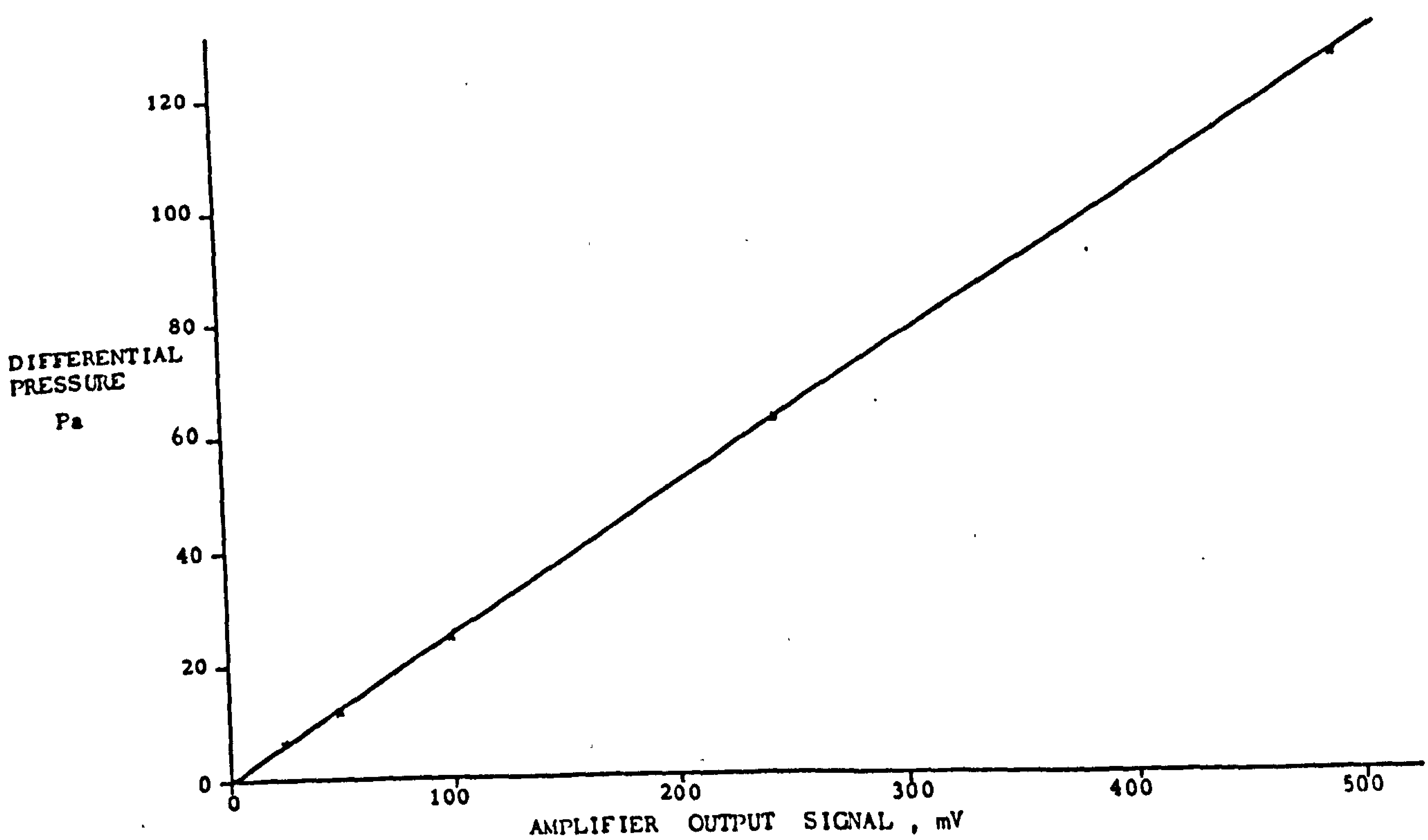


FIGURE 6.5 CALIBRATION FO PRESSURE TRANSDUCER

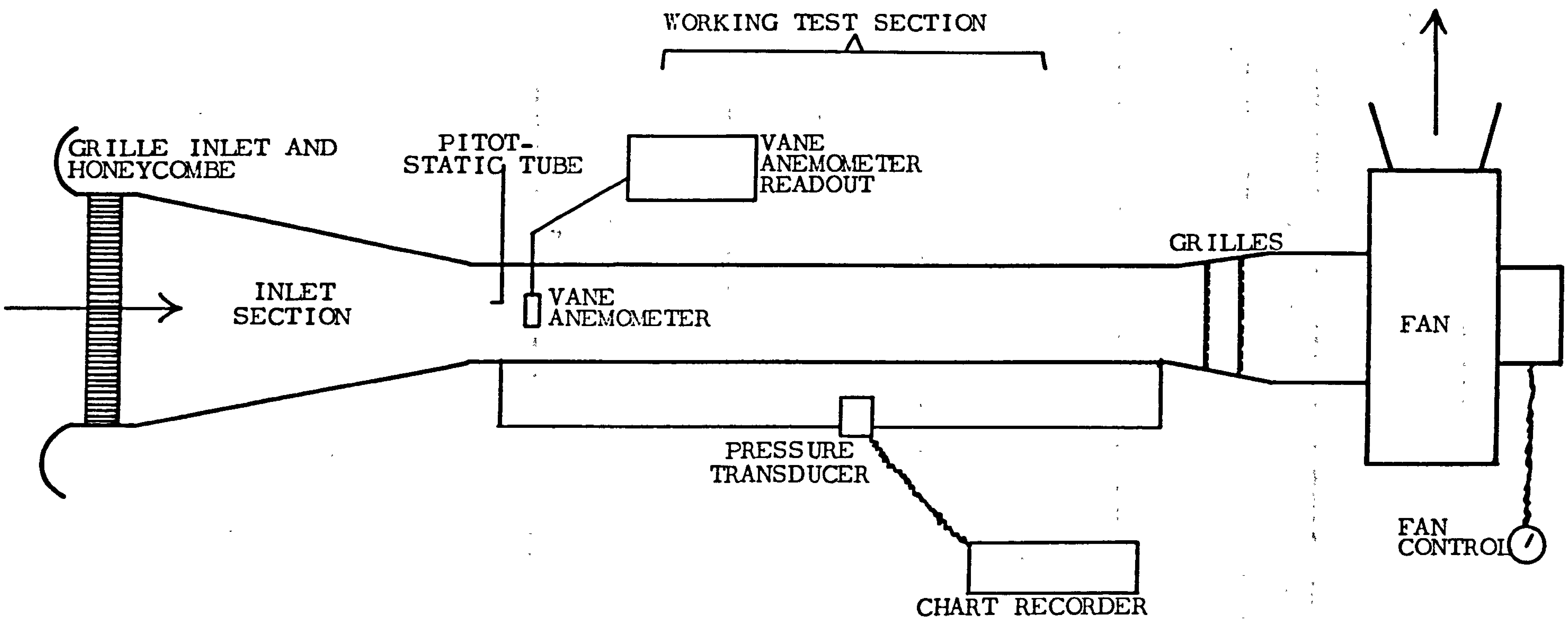


FIGURE 6.6 SCHEMATIC DIAGRAM OF WIND TUNNEL AND TEST EQUIPMENT LAYOUT

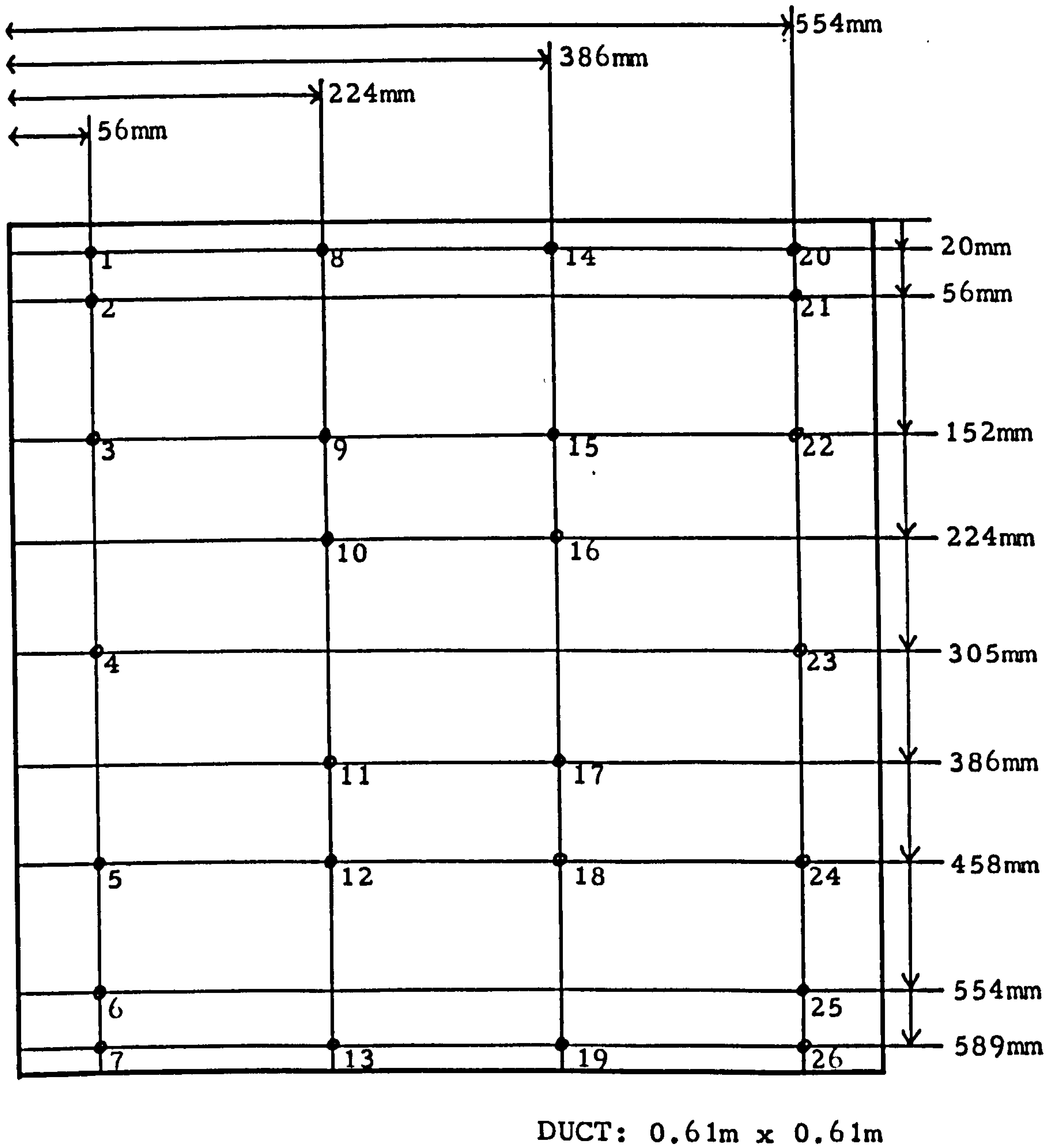


FIGURE 6.7 POSITIONS FOR PITOT-STATIC TUBE MEASUREMENTS IN SQUARE CROSS SECTION OF WIND TUNNEL

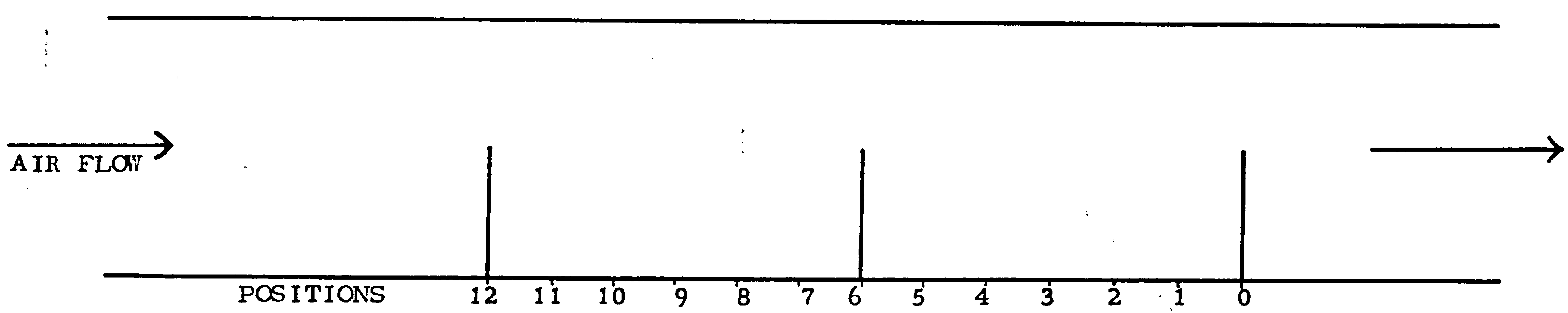


FIGURE 6.8 PARTITION POSITIONING IN WORKING SECTION

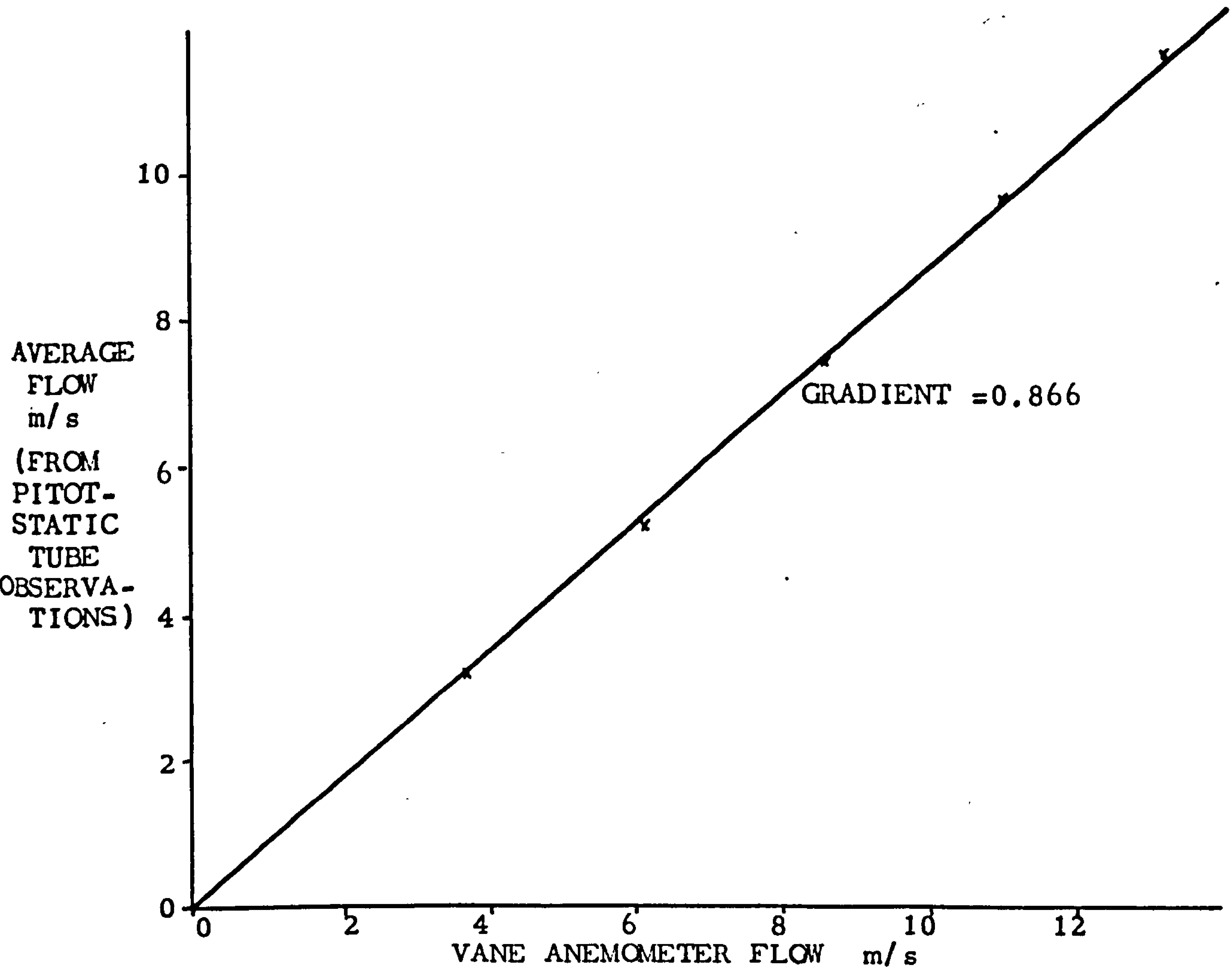


FIGURE 6.9 VANE ANEMOMETER FLOW AND AVERAGE FLOW (NO PARTITION CASE)

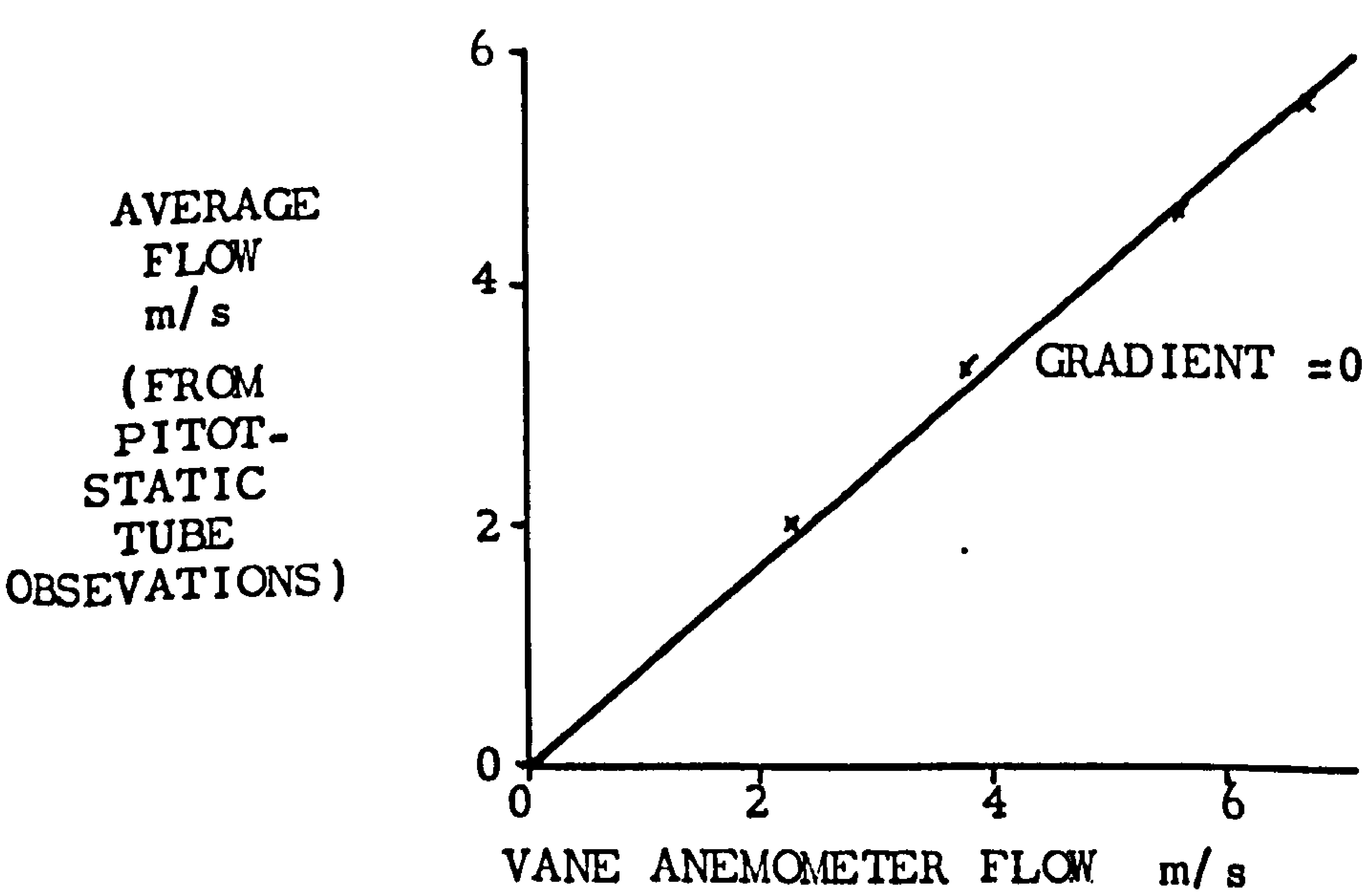


FIGURE 6.10 VANE ANEMOMETER FLOW AND AVERAGE FLOW (PARTITION POSITION 0)

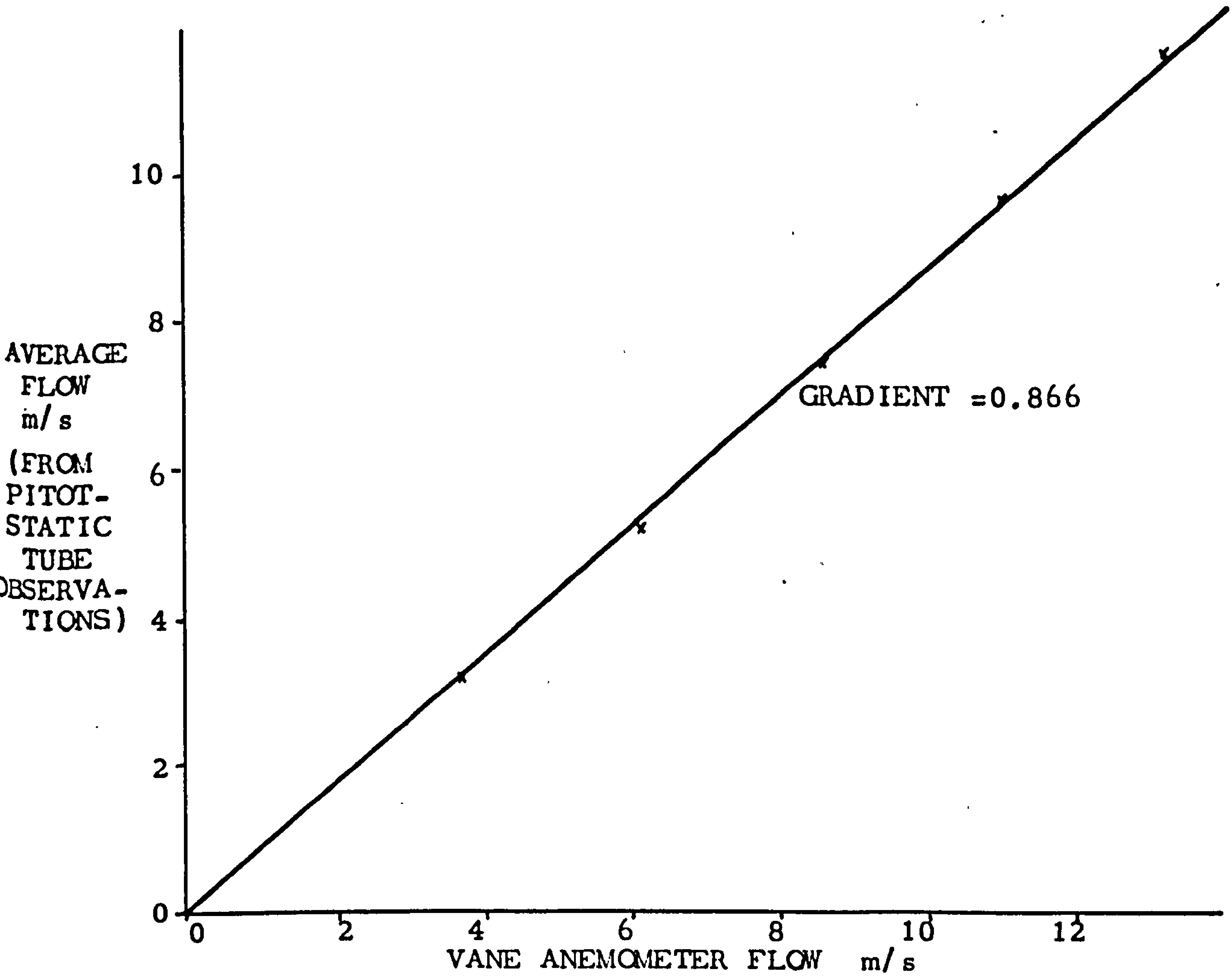


FIGURE 6.9 VANE ANEMOMETER FLOW AND AVERAGE FLOW (NO PARTITION CASE)

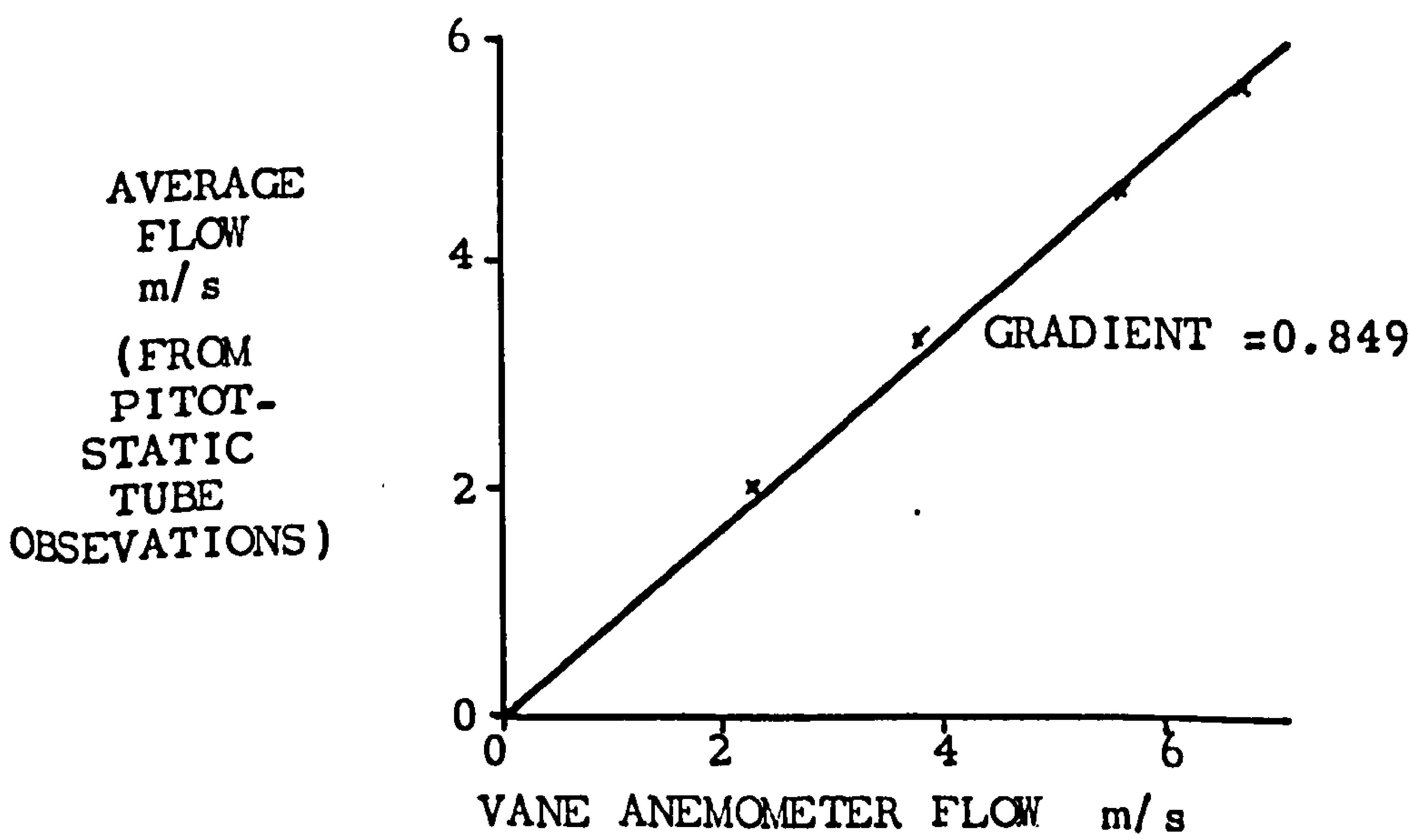


FIGURE 6.10 VANE ANEMOMETER FLOW AND AVERAGE FLOW (PARTITION POSITION 0)

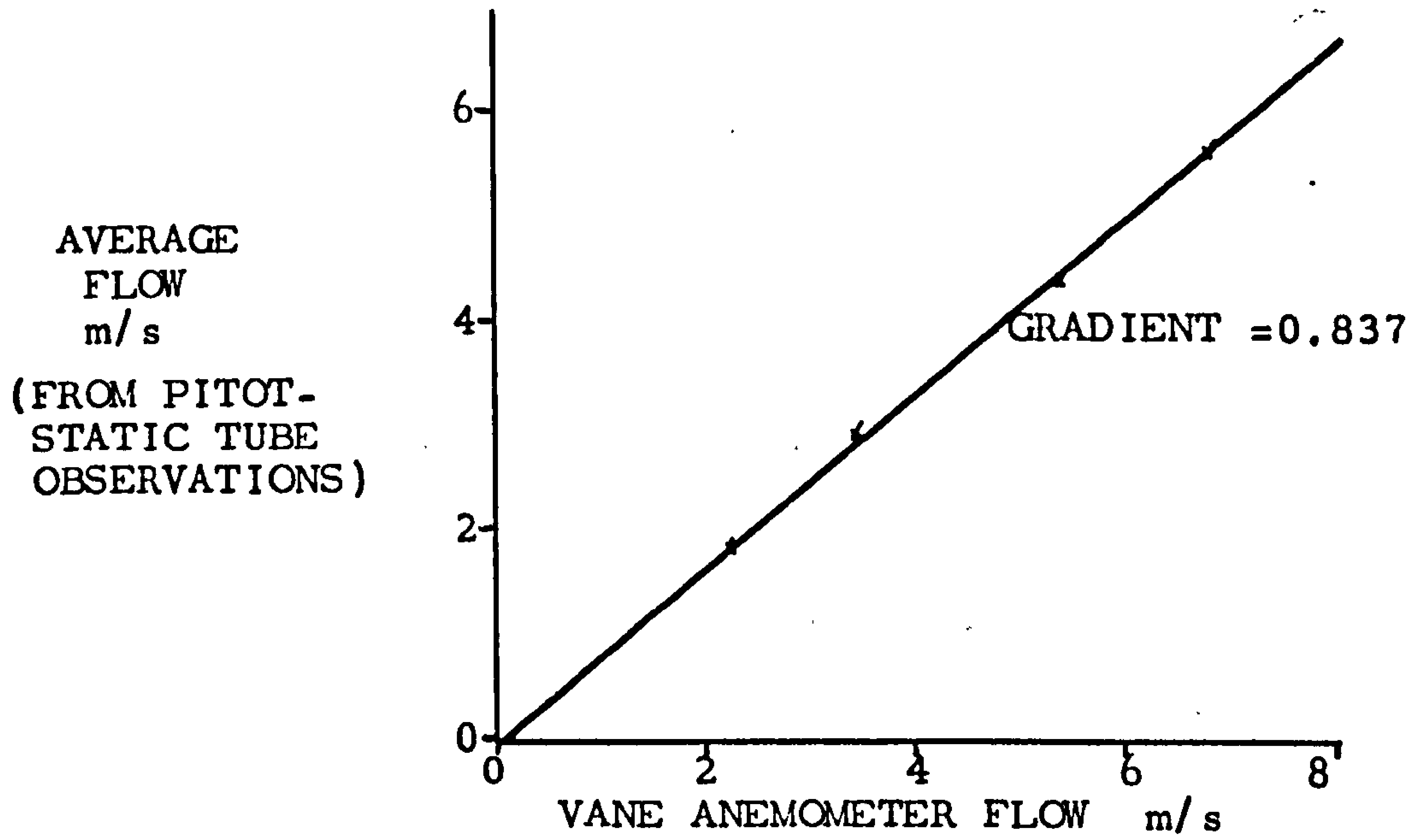


FIGURE 6.11 VANE ANEMOMETER FLOW AND AVERAGE FLOW (PARTITION POSITION 6)

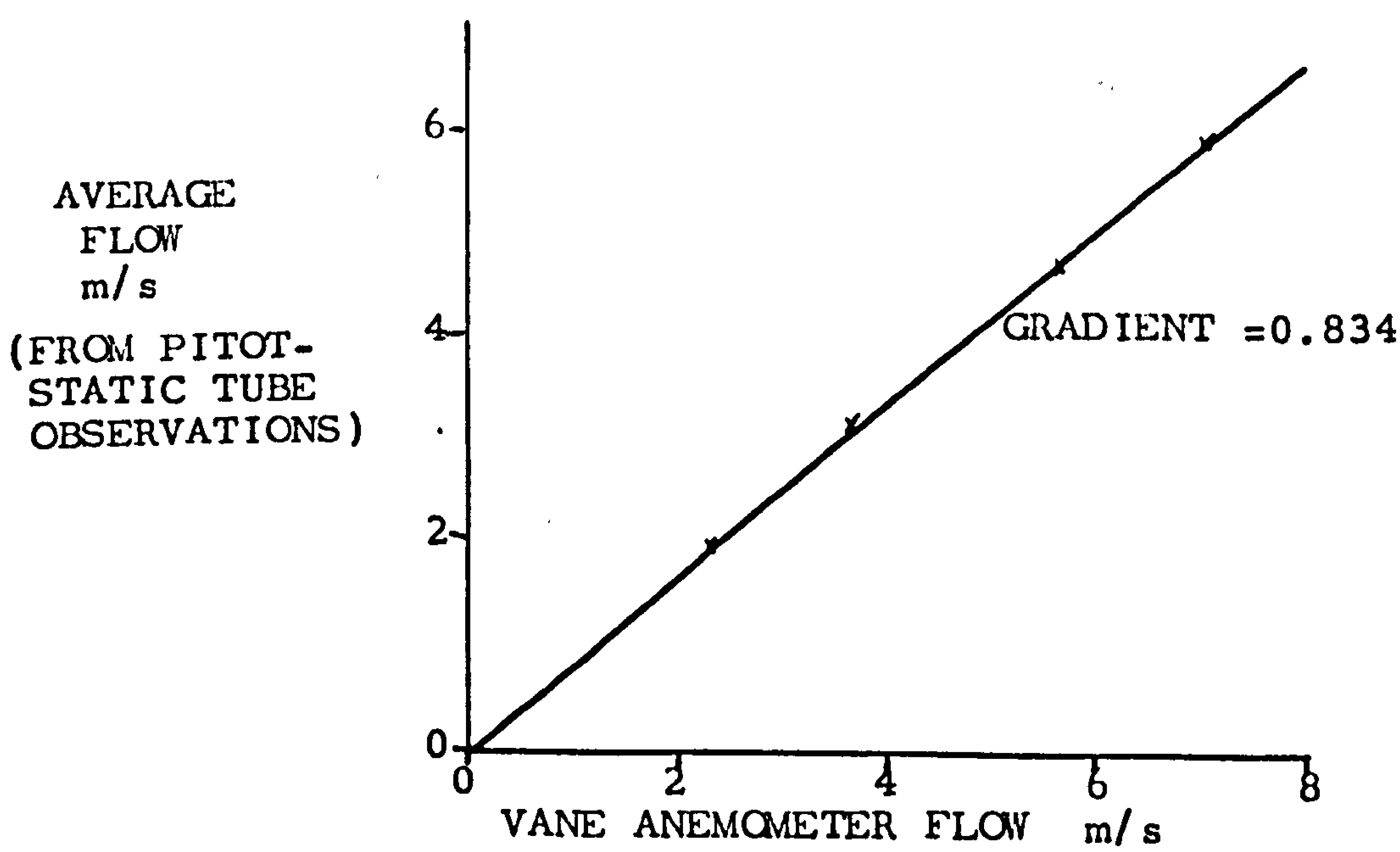


FIGURE 6.12 VANE ANEMOMETER FLOW AND AVERAGE FLOW (PARTITION POSITION 12)

CHAPTER 7

MODEL SCALE TESTS : RESULTS AND DISCUSSION

7.1 INTRODUCTION

Experiments were carried out to investigate air flow through regularly partitioned spaces at model scale. These experiments were divided into three main groups; the first two groups were performed consecutively and used the same equipment; the third group of tests utilised different apparatus and was carried out at a later date. The grouping of the tests was as follows:

(i) Model chamber testing of up to five partitions aligned in series, each partition having identical circular holes cut in it.

(ii) Model chamber testing of up to five partitions aligned in series, each partition being an identical, plain rectangular wall.

(iii) "Wind Tunnel" testing of up to four partitions aligned in series, each partition being an identical, plain rectangular wall.

The experiments were designed to give a turbulent flow regime within the models, the overall aim being to investigate the resistance of various partition layouts, to air flow. The basic relationship by which resistance was determined was:

$$R = \Delta P/Q^2 \qquad (7.1)$$

where ΔP = pressure difference

Q = flow rate

and R = total resistance to flow of all partitions.

7.2 MODEL CHAMBER EXPERIMENTS (Test groups (i) and (ii))

For the two types of partition to be as a number of layout arrangements were chosen are listed below:

1. Partitions - central position
2. Partitions - at spacings of 0.305m, 0.455m, 0.61m, 0.915m, 1.22m, 1.83m and 2.44m.
3. Partitions - at spacings of 0.305m, 0.61m, 0.915, and 1.22m.
4. Partitions - at spacings of 0.305m, 0.405m, 0.61m and 0.81m.
5. Partitions - at spacings of 0.305m, 0.455m and 0.61m.

These distances between partitions equated to spacings from one quarter to twice the height of the chamber, (half to four times the height of the rectangular wall partition). The spacings were to cover the general range to be found at full scale.

In addition a test run was carried out with no partition in the model. In this case the resistance was so low as to be negligible by comparison with partitioned cases, and was subsequently ignored.

The results of the tests are given in Appendix B1 for the circular hole partitions, and B2 for rectangular wall partitions. The results tables of appendices quote the measured pressure differences of flow rates. However for the purposes of resistance determination, it was the relationship between pressure and the square of the flow rate that was required. This was calculated and the results are shown for each arrangement in the diagrams - Figures 7.1 (a) - (s) and 7.2 (a) - (s).

A computer program "POLF" available on the University of Sheffield's PRIME 750 computer, was used to fit a line (using least squares regression) to each set of data. The line was constrained to pass through the origin since at zero flow rate there was zero pressure difference. A fairly strong linear relationship was observed to exist between the pressure difference and the square of the flow rate. This supported the supposition of turbulent flow and also indicated that the resistance (given by the gradient of the line - see equation 7.1) was relatively constant over the range of flow rates used. Tables 7.1 and 7.2 indicate the resistances found in units of Pa s^2m^{-6}.

[N.B. For more details of statistical analyses refer to Appendix D]

TABLE 7.1 RESISTANCES DETERMINED FROM MODEL TESTS - CIRCULAR HOLE PARTITIONS

TEST NO.	NO. OF PARTITIONS (m)	SPACING	RESISTANCE
21	1	-	0.613
22	2	0.61	0.673
23	3	0.61	0.834
24	4	0.61	1.066
25	5	0.61	1.348
26	5	0.455	1.157
27	5	0.305	0.927
28	4	0.305	0.801
29	4	0.405	0.935
30	4	0.81	1.387
31	3	1.22	1.292
32	3	0.915	1.135
33	3	0.305	0.647
34	2	0.305	0.589
35	2	1.22	0.891
36	2	2.44	1.094
37	2	1.83	0.986
38	2	0.915	0.831
39	2	0.455	0.695

TABLE 7.2 RESISTANCES DETERMINED FROM MODEL TESTS - RECTANGULAR WALL PARTITIONS

TEST NO.	NO. OF PARTITIONS	SPACING (m)	RESISTANCE
40	5	0.61	0.734
41	5	0.455	0.672
42	5	0.305	0.662
43	4	0.305	0.779
44	4	0.405	0.672
45	4	0.61	0.636
46	4	0.81	0.827
47	3	1.22	0.791
48	3	0.915	0.741
49	3	0.61	0.747
50	3	0.305	0.844
51	2	0.305	0.835
52	2	0.455	0.862
53	2	0.61	0.952
54	2	0.915	0.782
55	2	1.22	0.779
56	2	1.83	0.698
57	2	2.44	0.867
58	1	-	0.888

A plot of the change of resistance with respect to partition spacing is shown in Figure 7.3 for the circular hole partitions; and in Figure 7.4 for the rectangular wall partition. The resistance produced by one single partition of either type should be identical to that produced by any number of thin partitions at zero spacing. Therefore this result has also been included in the diagrams, to represent the zero spacing case.

7.3 DISCUSSION OF MODEL CHAMBER RESULTS

The results for the circular hole partitions show that the resistance increased as the separation between the partitions was increased. It can also be seen that the resistance of two partitions (at any of the spacings used here) was less than twice that produced by a single partition. This indicates that the resistances of a series of such partitions cannot be summed in a simple fashion. By extrapolating the data for the two partition case, it is indicated that for consecutive partitions to behave independently (in terms of resistance) they must be at least two duct diameter apart (based on the hydraulic diameter of the model chamber) or six "hole diameters" apart.

The increase in resistance with respect to spacing appears to be almost linear for each specific number of partitions. Using "least square" regression following relationships were estimated.

For two partitions

Resistance = 0.218 (spacing) + 0.584

For three partitions

Resistance = 0.605 (spacing) + 0.535

For four partitions

Resistance = 0.93 (spacing) + 0.564

For five partitions

Resistance = 1.21 (spacing) + 0.596

(Resistance units : Pa s^2m^{-6}, spacing in metres)

Although relationships were found for the circular hole partition, this was not so for the plain rectangular wall partitions (see Figure 7.4). There seemed to be a degree of variability within the tests and no simple relationship could be discerned. However, considering the results for each number of partitions separately; the minimum resistance occurred approximately in the middle of the range of spacings employed. Further studies were undertaken to investigate the behaviour of the rectangular wall partitions.

7.4 WIND TUNNEL TRIALS (Test group (iii))

In order to try to provide the most stable environment to extend the testing of plain rectangular wall partitions, one of the aerodynamic wind tunnels within the Department of Building Science was utilized. This had an invariable square cross section and its layout and design have been described in the previous chapter.

The partition arrangements used are listed below:

One Partition

Two Partitions - at spacing of 0.152m, 0.305m, 0.46m, 0.61m, 0.76m, 0.915m, 1.07m, 1.22m, 1.37m, 1.52m, 1.68m and 1.83m.

Three Partitions - at spacings of 0.152m, 0.305m, 0.46m, 0.61, 0.76m and 0.915m.

Four Partitions - at spacings of 0.152m, 0.305m, 0.46, and 0.61m.

No Partition.

The spacing chosen represented distances between partitions of a half to six times the height of the wall partition. This range being a slightly extended version of that chosen for the model chamber experiments.

The results of pressure differences and corresponding flow rates for each of the trials are given in Appendix B3. As with the model chamber experiments, it is the resistance that is of interest hence pressure differences versus the squares of the flow rate plots are shown in the graphs of Figure 7.5. Again a good linear relationship was quantified using the POLF computer program (used as previously mentioned) and the resistance in each case was determined. [N.B. For more details of statistical analyses refer to Appendix D.]

For the case with no partition, the resistance was sufficiently small by comparison with that of the partitions, that the flow resistance of the tunnel

TABLE 7.3 RESISTANCES DETERMINED FROM WIND TUNNEL TESTS - RECTANGULAR WALL PARTITIONS

TRIAL NO.	NO. OF PARTITIONS (m)	SPACING	RESISTANCE
6 + 7 + 34 + 34a	1	- (pos 0)	37.5
8	0	-	0.6
9 + 10	2	0.152	36.0
11 + 11a	2	1.83	31.9
12 + 12a	3	0.915	32.4
13 + 13a	4	0.61	28.8
14 + 14a	3	0.61	24.9
15 + 15a	2	1.22	26.7
16 + 16a	1	- (pos 6)	35.0
17 + 17a	1	- (pos 12)	25.0
18 + 18a	2	1.68	30.7
19 + 19a	2	1.52	28.7
20 + 20a	3	0.76	27.5
21 + 21a	2	1.37	27.6
22 + 22a	4	0.46	26.1
23 + 23a	3	0.46	22.9
24 + 24a	2	0.915	24.7
25 + 25a	4	0.305	23.5
26 + 26a	3	0.305	22.1
27 + 27a	2	0.305	34.4
28 + 28a	3	0.152	34.5
29 + 29a	4	0.152	29.5
30 + 30a	2	0.46	31.2
31 + 31a	2	1.07	26.2
32 + 32a	2	0.76	25.2
33 + 33a	2	0.615	26.4

surfaces could be neglected. The two parts of the set of results obtained for each layout (eg 11 + 11a) were found to be very similar and were incorporated into a single results group. The resistance values obtained are given in Table 7.3 (resistance taking units of Pa s^2m^{-6}). A graphical comparison of the results is shown in Figure 7.6.

7.5 DISCUSSION OF WIND TUNNEL RESULTS

The results shown by the graphs of Figure 7.6 and the relationships calculated using least squares indicated that the gradients (representing the resistance) are more stable than those of the model chamber tests, thus justifying the extension of the tests to the wind tunnel.

The comparison of the results (Figure 7.6) exhibits some similar tendencies to those found in the plain rectangular wall testing in the model chamber. Though the results were not as one might, at first, expect.

Taking the results for each number of partitions separately, the resistance produced first decreased to a minimum then increased as the spacing between partitions was widened. A plot of resistance against total "spread" distance of the partitions, (Distance between first and last partition in space), showed that the apparent minimum for each number of partitions occurred at the same distance (Figure 7.7). In order to try to discover a reason for this, smoke tracer was introduced into the wind tunnel with partitions in position, so that the air flow patterns could be seen. The flow patterns observed indicated that the partition spread at which the minimum occurred, was slightly greater than the length of the main eddy set up by the first partition. When the second partition was introduced at the minimum resistance distance it just contained the eddy circulation flow and slightly reduced some other turbulence. As the third and fourth partitions were added, between the first two, in each case, the general level of turbulence in the wake of the partitions seemed reduced and a fairly steady flow was set up across the top of the partitions. Figure 7.8 illustrates the observations made.

These observations go some way towards explaining the results shown in Figure 7.6; that is, why the addition of partitions apparently reduced the resistance. The rectangular wall partitions used in the experiments would only begin to behave, (at least partly) independently when the spacing exceeded the eddy length. Referring to Figures 7.6 and 7.7, the distance is approximately 0.9m, that is, three times the height of the partition. Resistance and partition spread distance were also plotted for the wall partition experiments performed in the model chamber (Figure 7.9). A minimum is less in evidence in this case, the lowest point occurs at a spread of about 1.8m . This distance is also three times the height of the partitions used.

7.6 CONCLUSIONS

The range of partition spacings used in the experiments, were chosen to be representative of real life/full scale layout. At none of these spacings could the partitions be said to be independent, in terms of resistance to air flow (excepting, of course, the single partition)

The partitions with uniformly set out circular holes had been chosen to represent the situation usually assumed by theoretical studies (opening equivalent to circular holes in thin partitions). The alignment of the holes in each partition led to a lower degree of independence than if the holes had been randomly distributed and the flow had been returned to an homogeneous state. However, within the constraints of the experiments, it was not possible to vary the arrangement to allow that option. In any case, such an "independent" arrangement might be expected to yield the usual results of separate spaces connected sequentially, in which the resistances would be summated, and an investigation of such was not the objective of this work.

The results from the circular hole partitions indicated that the first partition provided the main resistance to flow. Adding further partitions increased the resistance but by less than that predicted by simply multiplying the figure for the single partition by the number of partitions. Taking a spacing equal to the height of a partition and using the relationships given in equations 7.2 to 7.5 the following resistances were derived:

One Partition : $= 0.613$ Pa s^2m^{-6}

Two Partitions :

$(0.218)(1.22) + 0.584 = 0.85$ Pa s^2m^{-6}

Three Partitions :

$(0.605)(1.22) + 0.535 = 1.27$ Pa s^2m^{-6}

Four Partitions :

$(0.93)(1.22) + 0.564 = 1.7$ Pa s^2m^{-6}

Five Partitions :

$(1.21)(1.22) + 0.596 = 2.07$ Pa s^2m^{-6}

These results are shown graphically in Figure 7.10.

The evidence suggests that partitions subsequent to the first, add to the resistance but as a reduced proportion of the single partition value. That is a relationship of the form:

$$R = R1 + (n-1)R2 \qquad (7.6)$$

where R = Total Resistance

R1 = Resistance of first partition

R2 = Resistance of subsequent partitions (depends upon spacing)

n = Number of partitions

The rectangular wall partitions had been designed to simulate wall type partitions found in industrial environments. Two sets of experiments were performed using the wall partitions; one in the model chamber (also for the circular hole partitions), and the second in aerodynamic wind tunnel.

The results obtained showed that the main resistance to flow was caused by the first partition. Indeed the resistance of a single partition in the model chamber was significantly greater than the circular hole partition with the same open area under the same conditions. However, when further partitions were added, the total resistance fell due to decreased turbulence effects and improved flow patterns. The resistance began to rise again once the spread of partitions exceeded the length of the main eddy caused by the first partition. The minimum resistance occurred when the spread distance was about three times the height of the partition. This result could prove significant for the design layout partitions and will be considered in greater depth in Chapter 9.

Comparing the results with the possibilities outlined at the end of Chapter 5; the circular hole partition seemed to indicate that a relationship of the form shown in Figure 5.13 is probable. This may also be the case for the wall partitions when spread over a significant distance.

In conclusion, the results of the model scale tests, using plain types of partitions, indicate that the most significant aspect of a series of regularly spaced partitions across which a pressure difference exists, is the effect of the first partition in the series.

For partitions with circular openings, the first partition has the greatest resistance to flow. Subsequent partitions appear to have resistances equal to one another but less than that of the first.

For wall type partitions, the first partition sets up a prime flow regime into which subsequent partitions are placed. Because of this prime flow the distance over which the partitions are spread, rather than their number, seems to be the main factor in determining resistance at many partition spacings utilized.

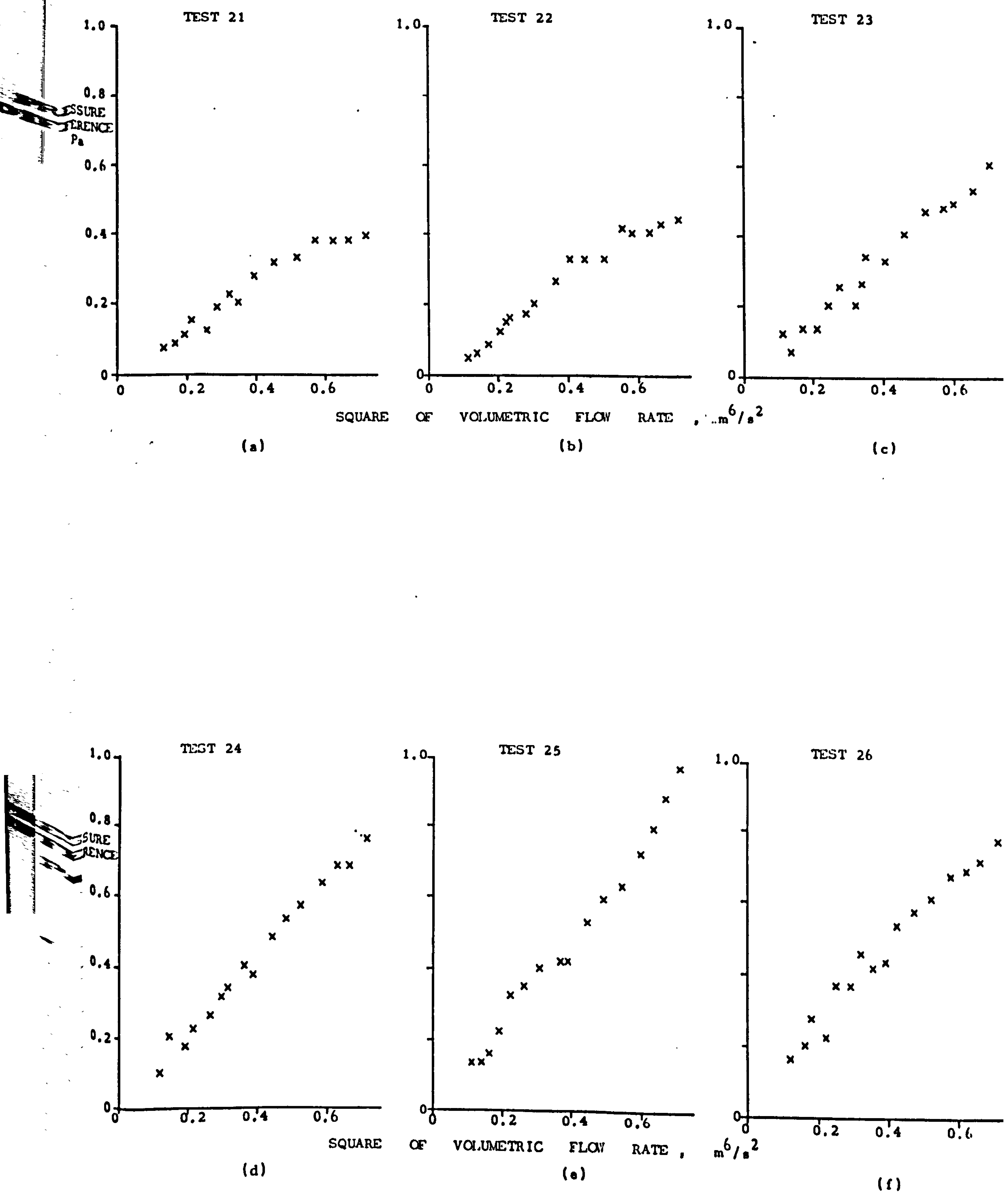

TEST 21
TEST 22
TEST 23
TEST 24
TEST 25
TEST 26
PRESSURE DIFFERENCE Pa
SQUARE OF VOLUMETRIC FLOW RATE , m^6/s^2
(a)
(b)
(c)
(d)
(e)
(f)

FIGURE 7.1

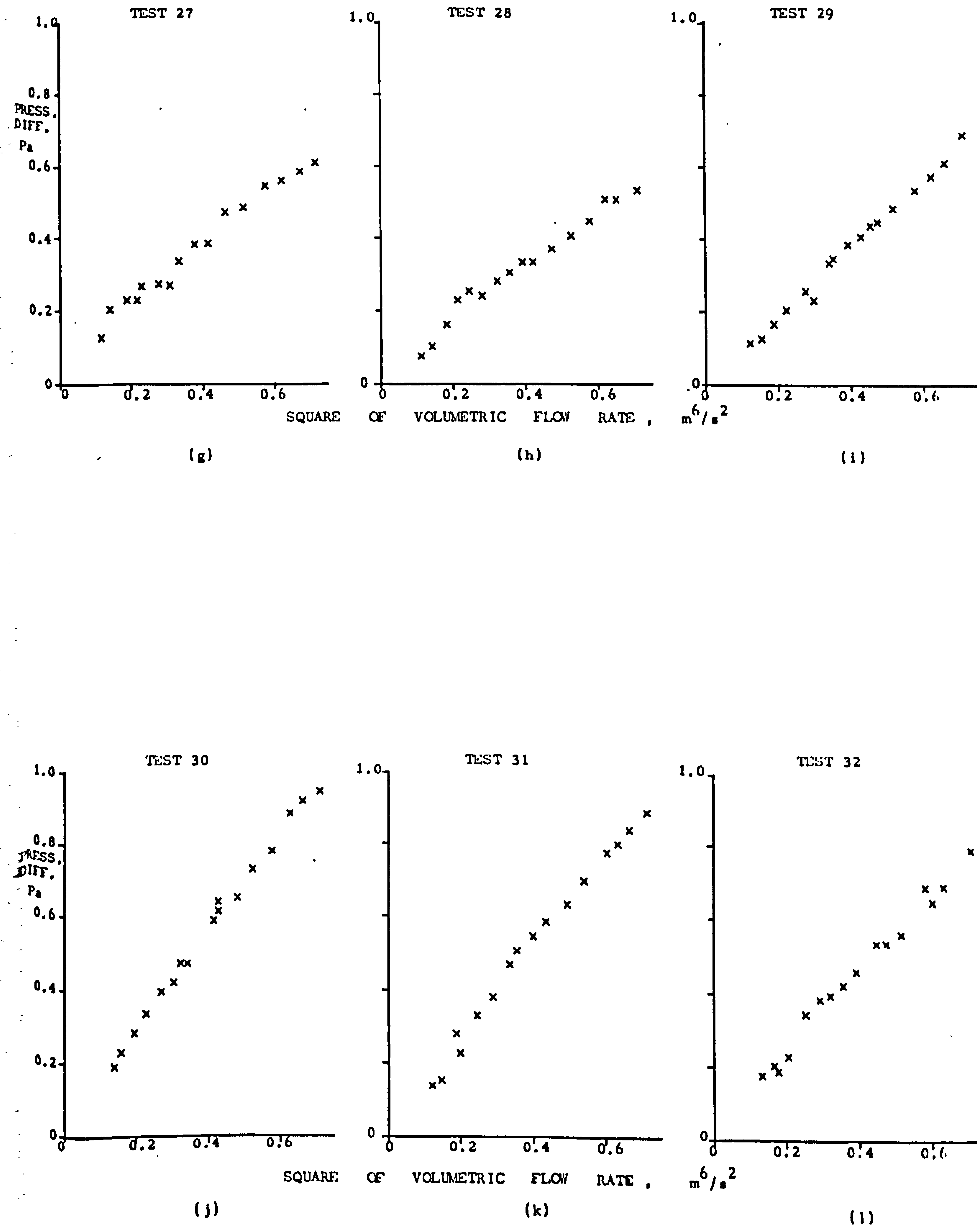


FIGURE 7.1 (cont.)

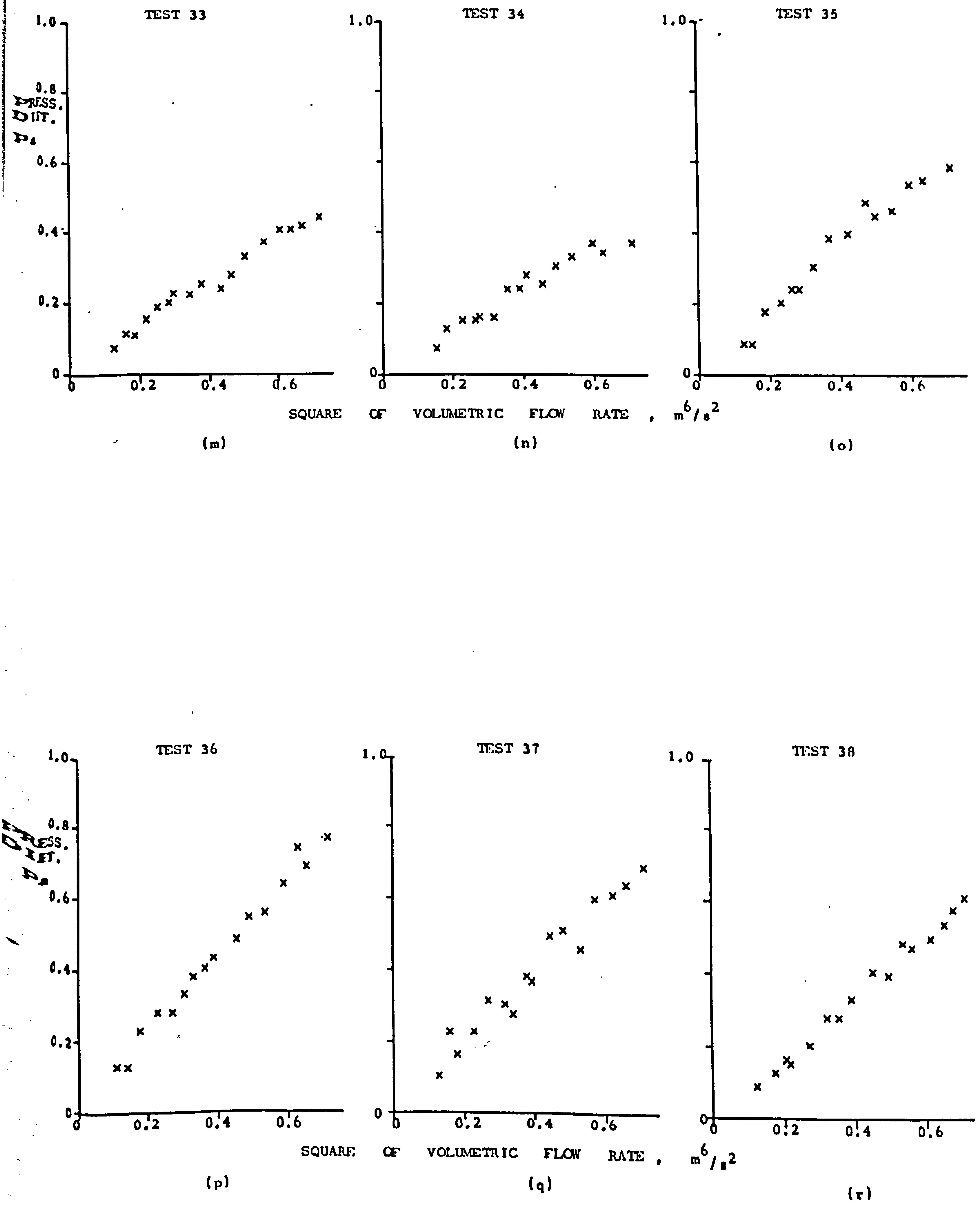


FIGURE 7.1 (cont.)

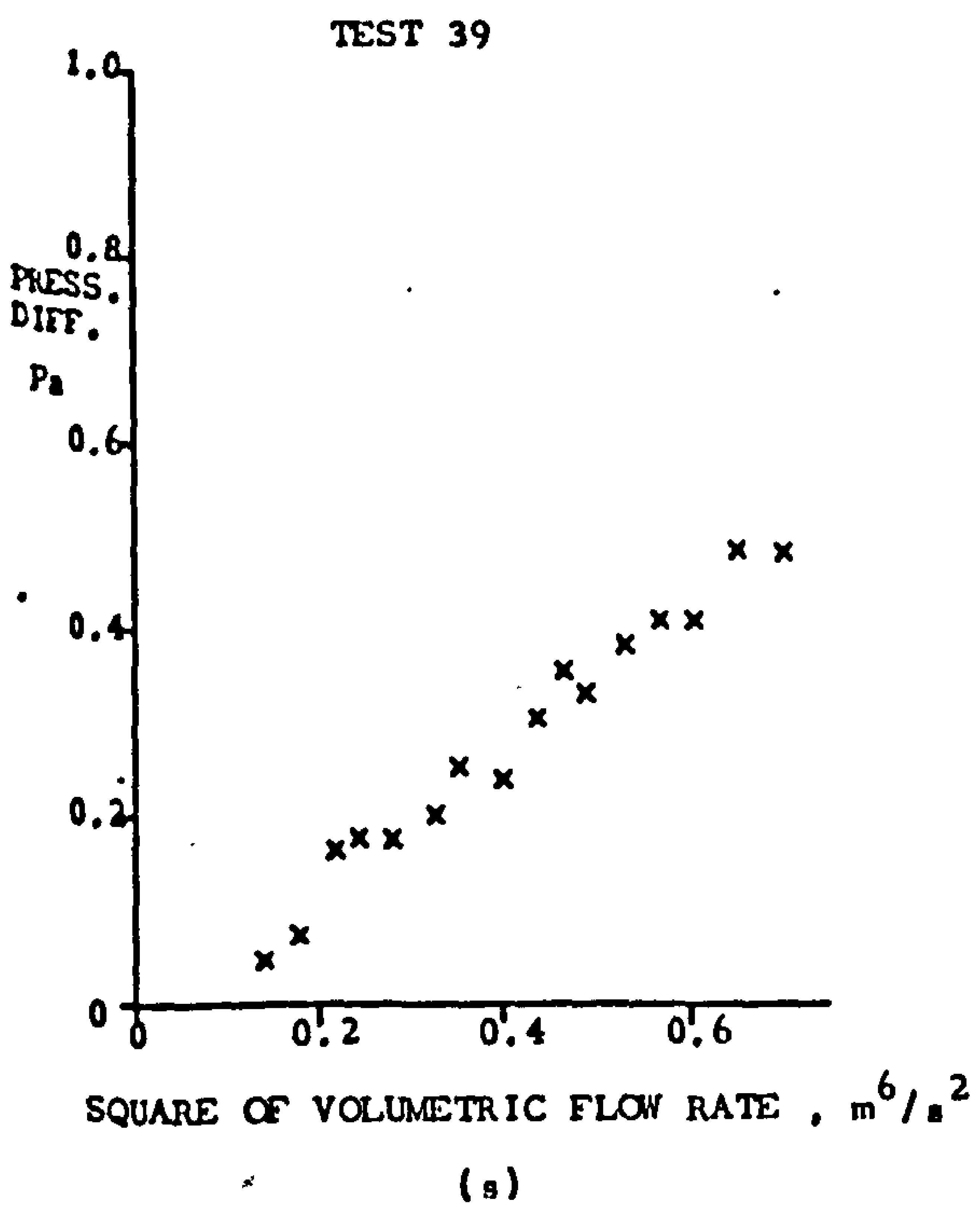


FIGURE 7.1 (cont.)

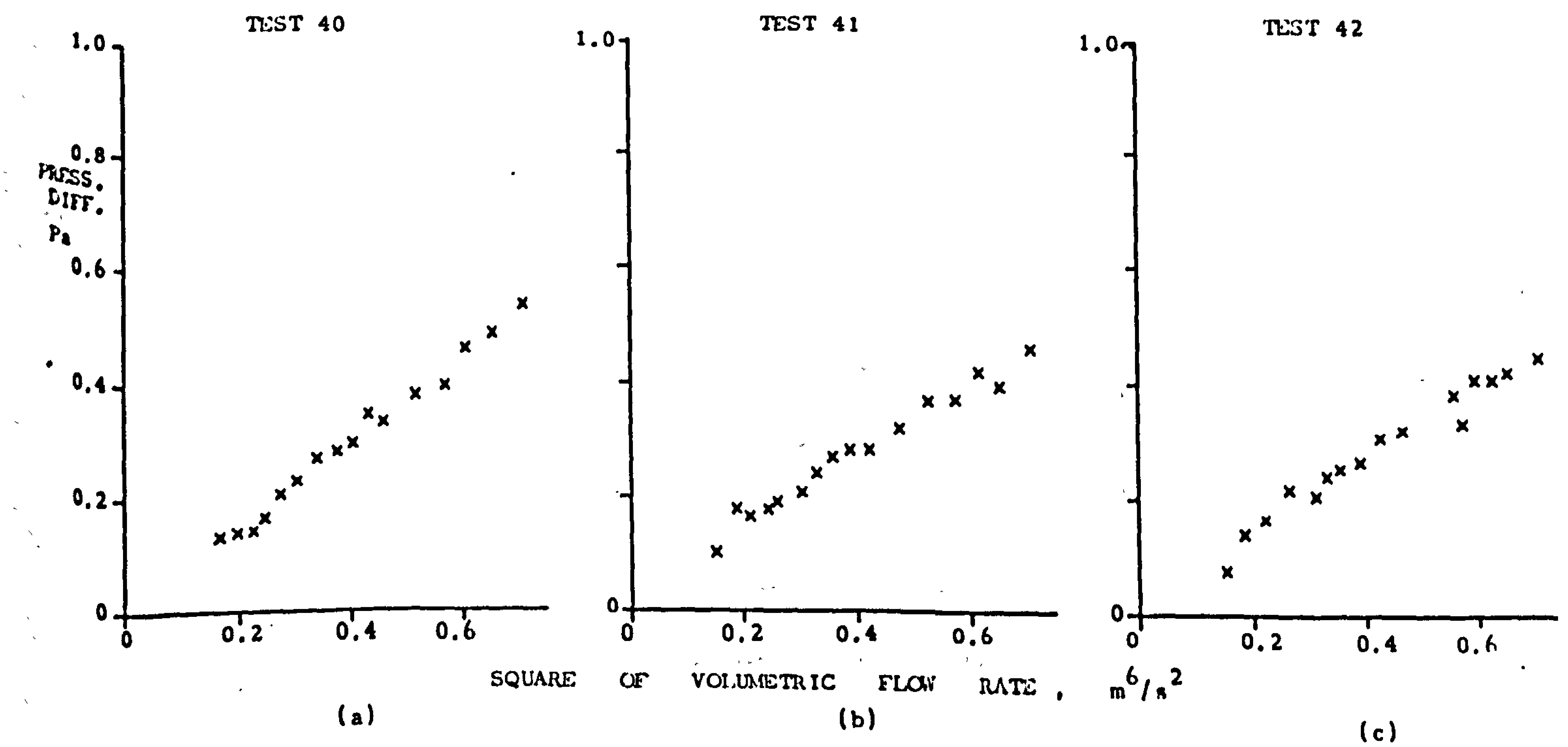


FIGURE 7.2

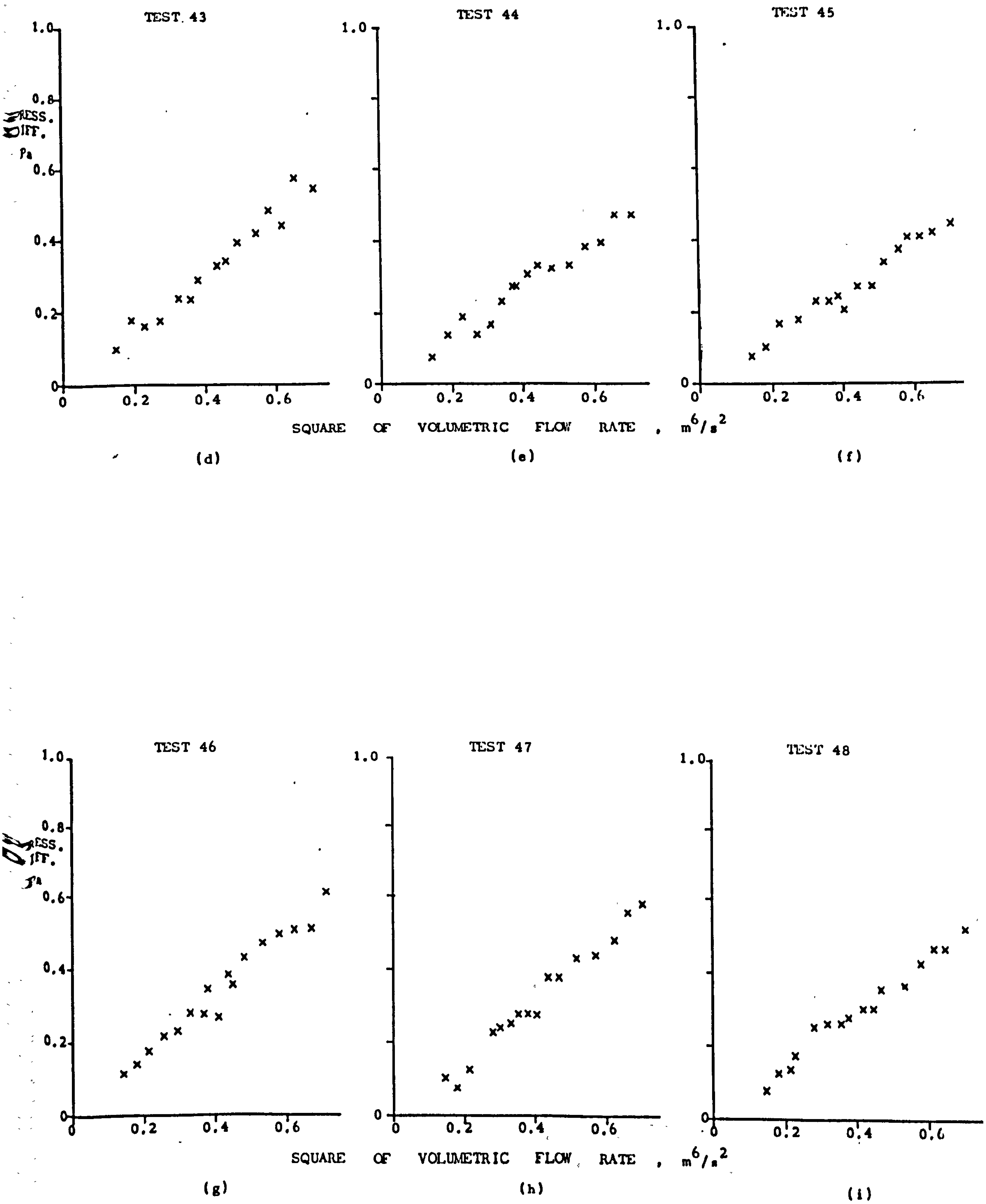
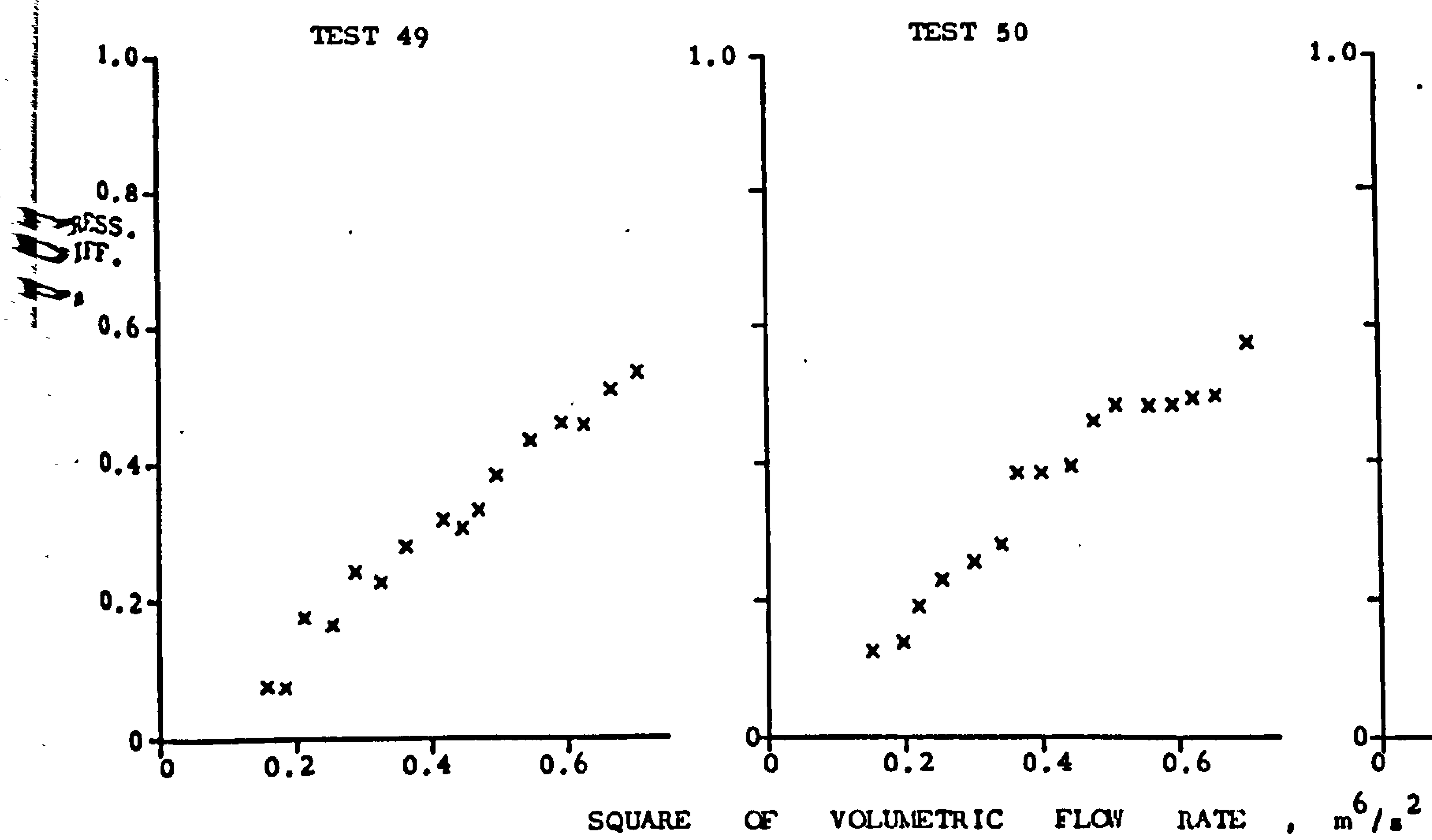
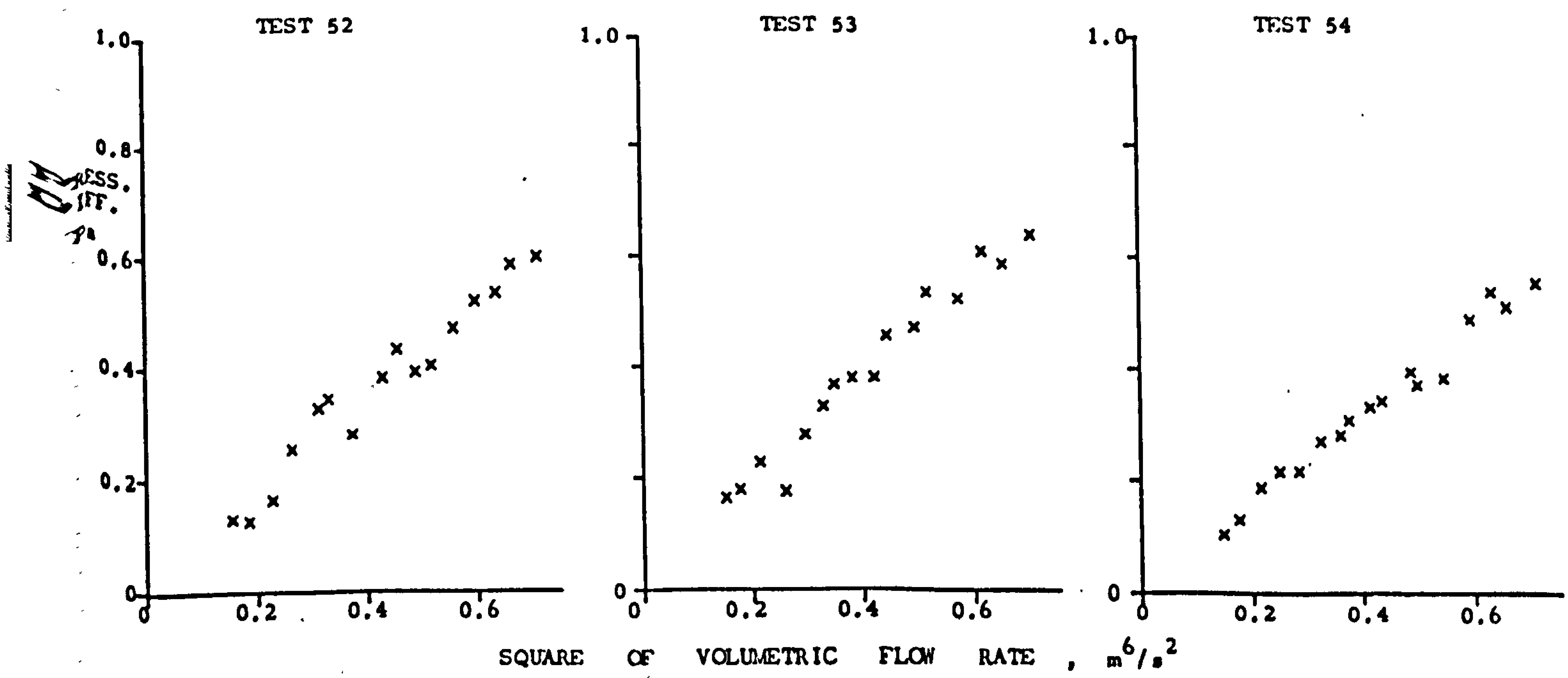


FIGURE 7.2 (cont.)

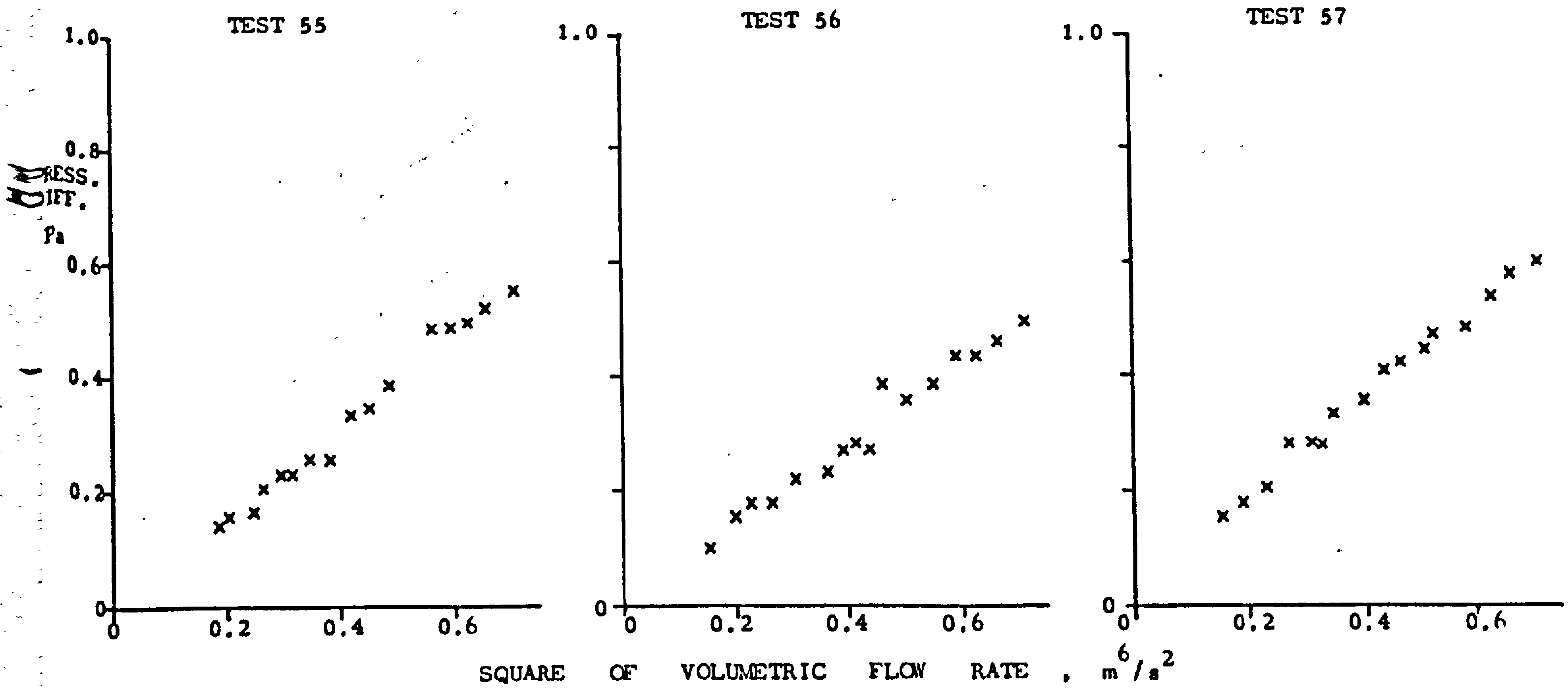


(j) (k) (l)

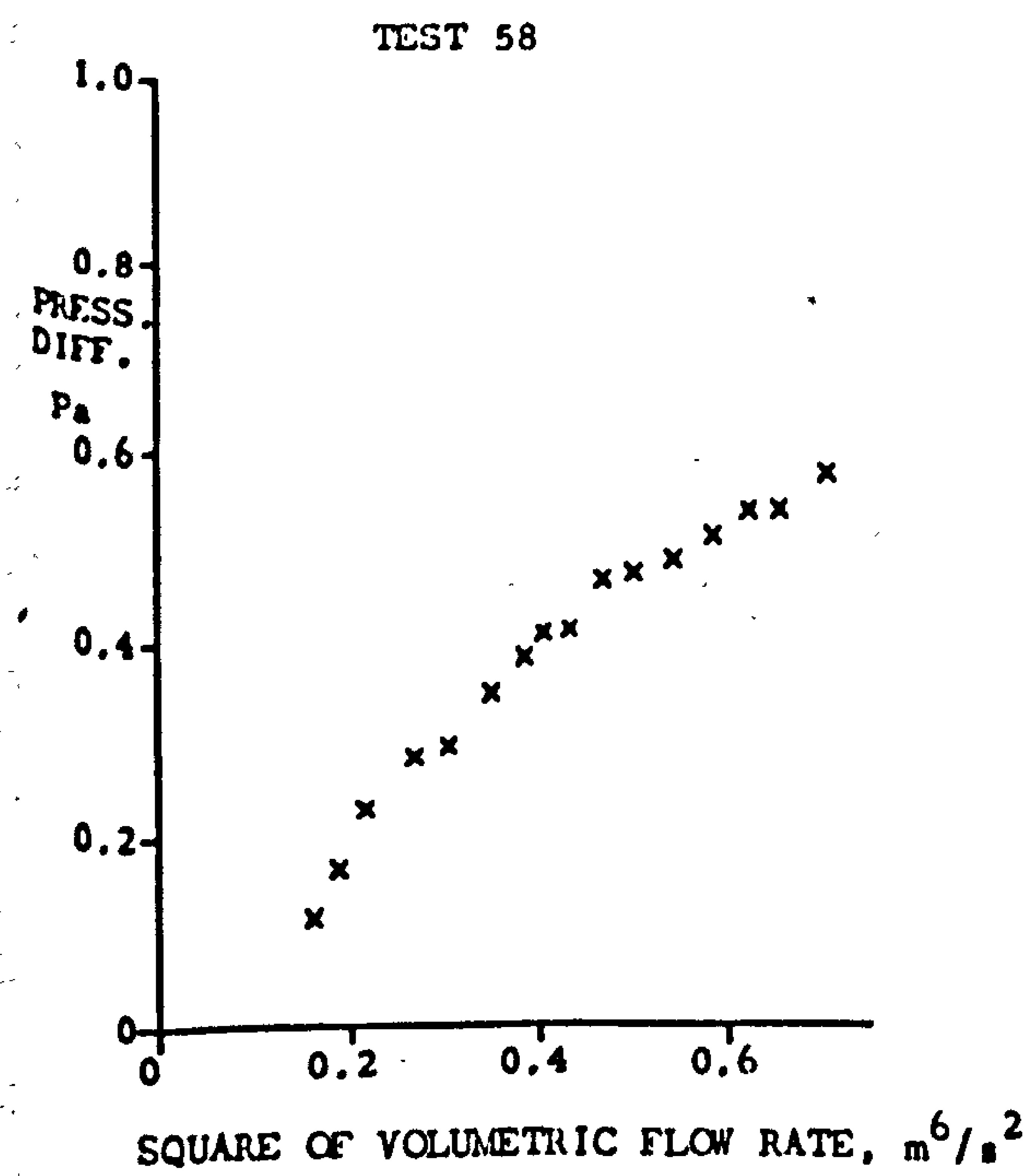


(m) (n) (o)

FIGURE 7.2 (cont.)



(p) (q) (r)



(s)

FIGURE 7.2 (cont.)

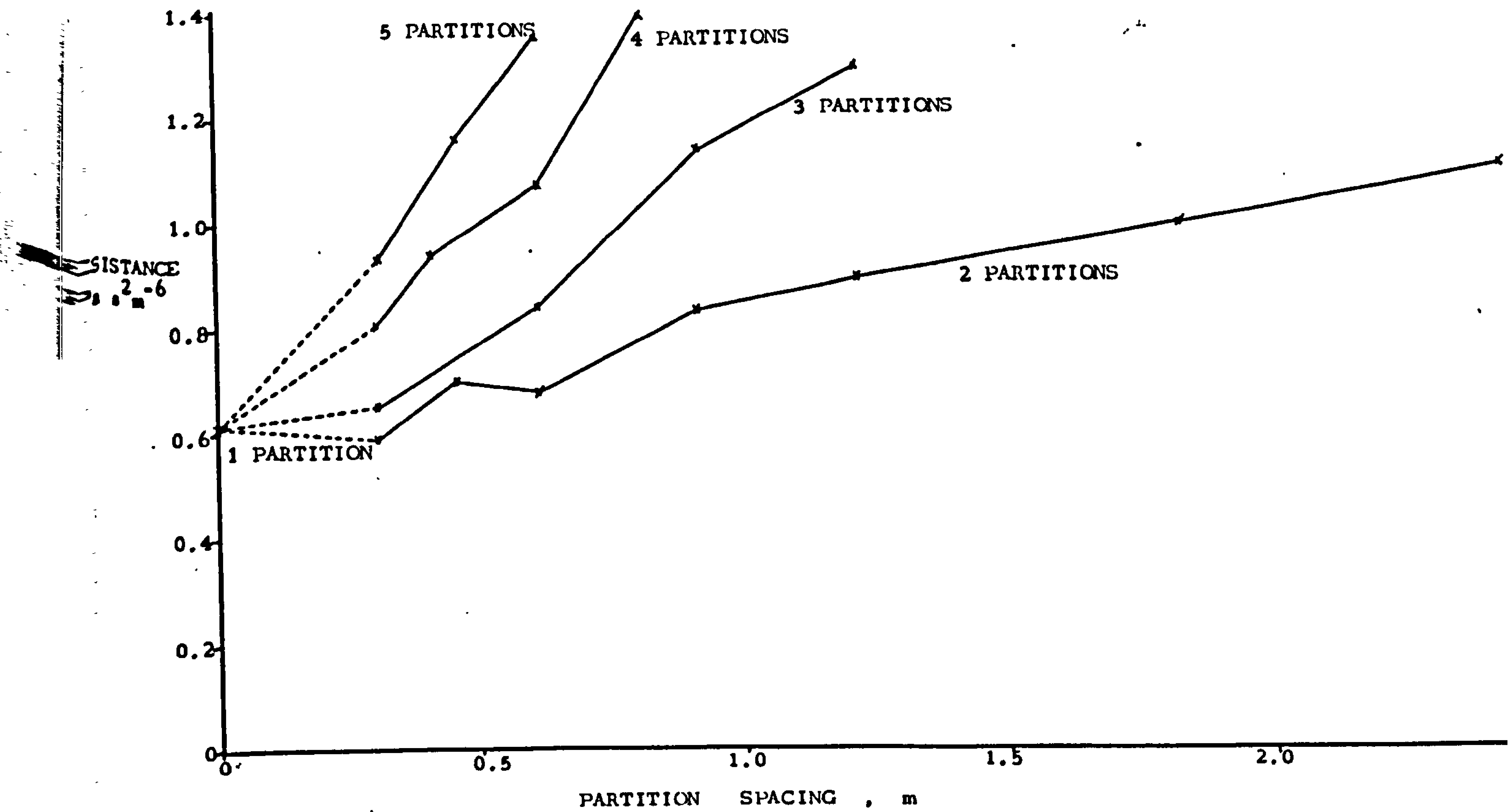


FIGURE 7.3 RESULTS FOR CIRCULAR HOLE PARTITIONS IN MODEL CHAMBER TESTS

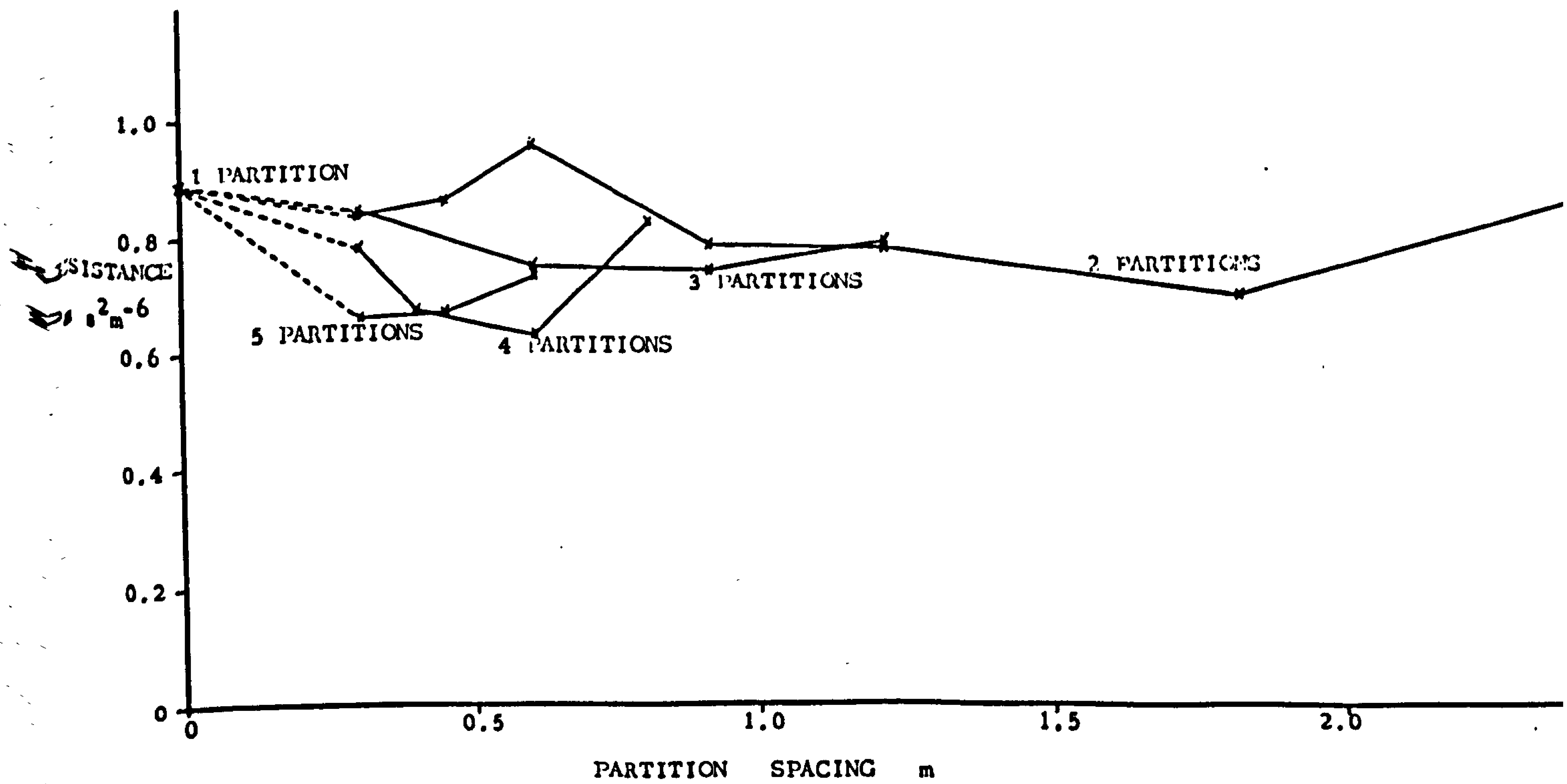


FIGURE 7.4 RESULTS FOR WALL PARTITIONS IN MODEL CHAMBER TESTS

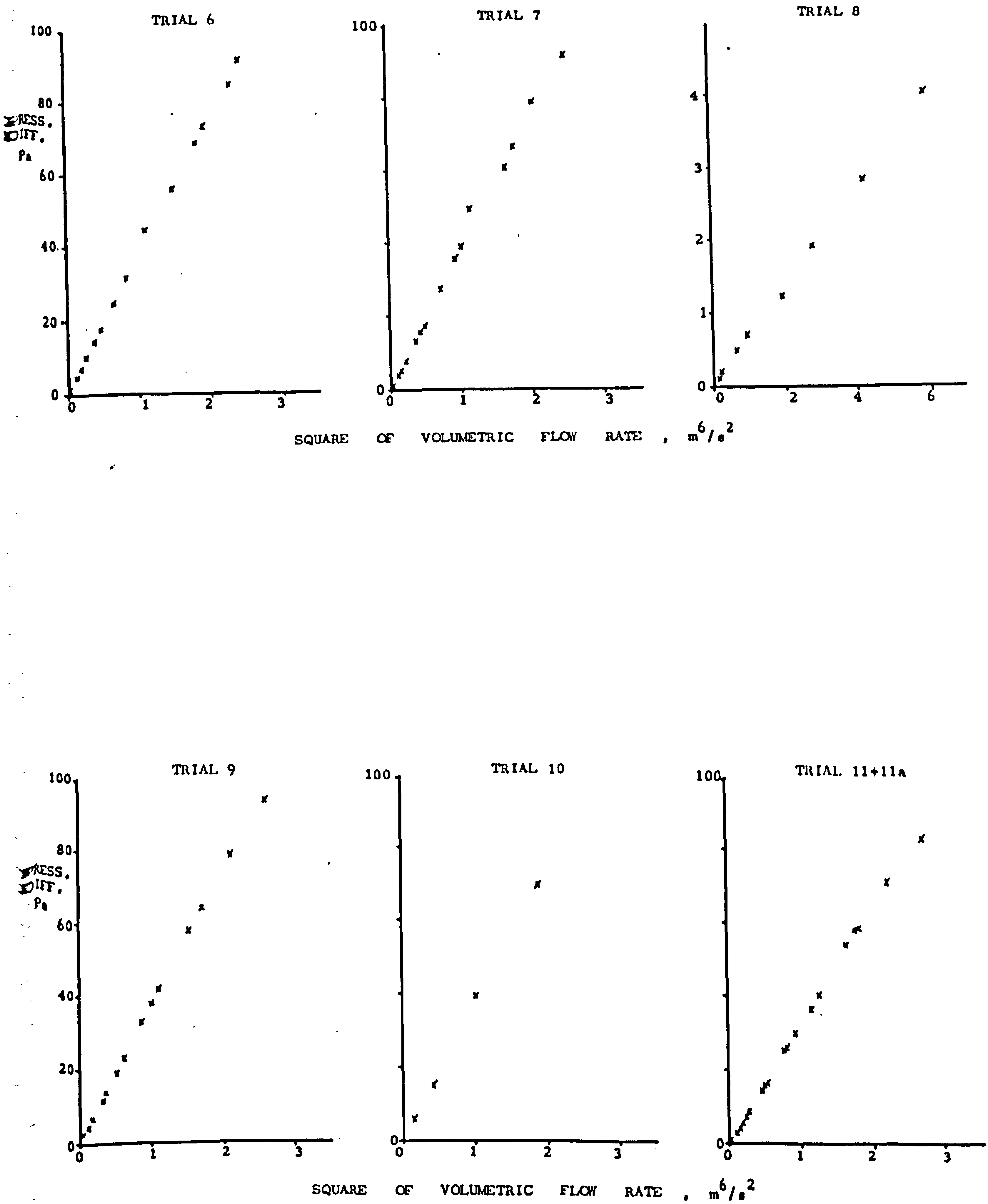

TRIAL 6
TRIAL 7
TRIAL 8
TRIAL 9
TRIAL 10
TRIAL 11+11a
PRESS. DIFF. Pa
SQUARE OF VOLUMETRIC FLOW RATE , m6/s2

FIGURE 7.5

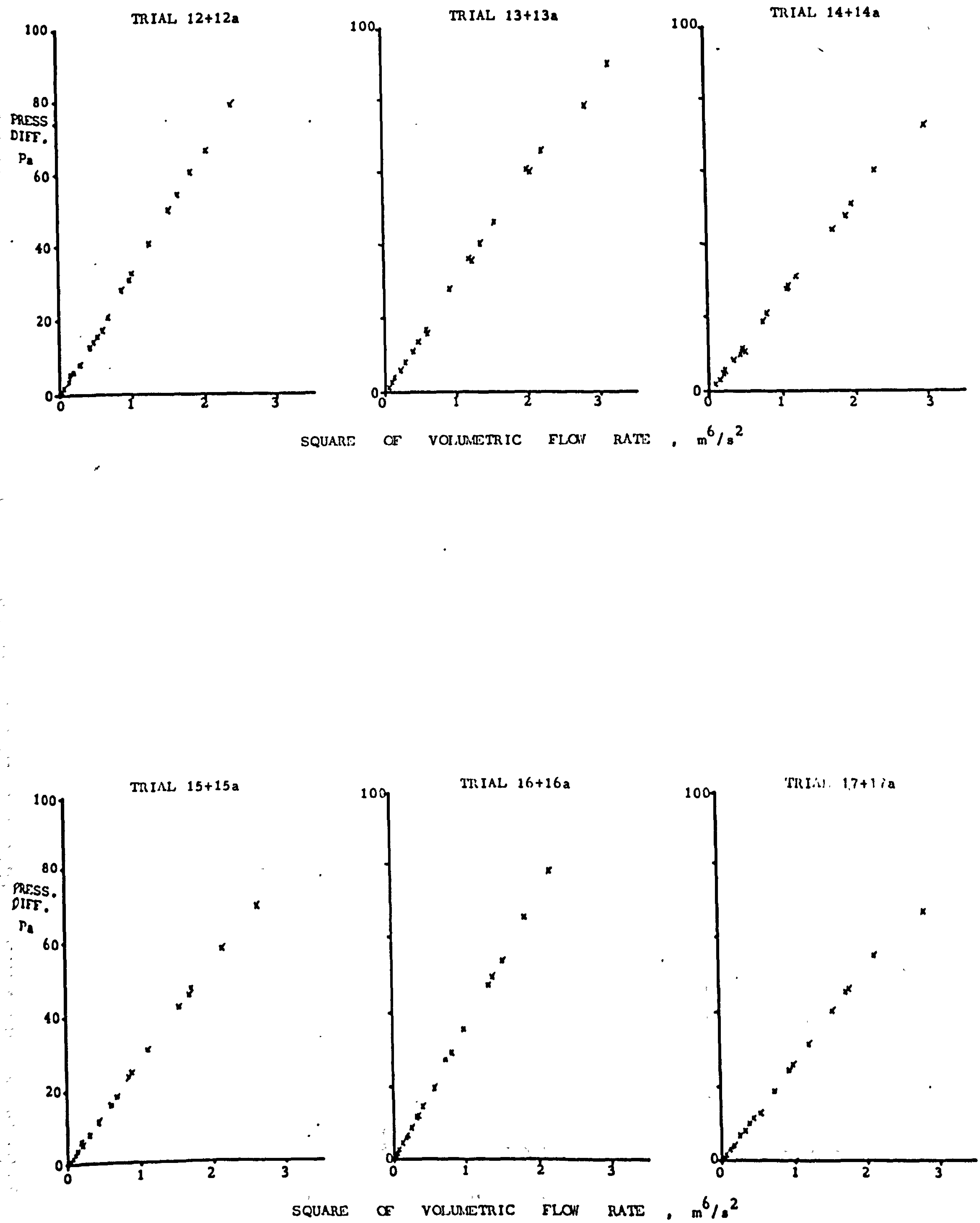
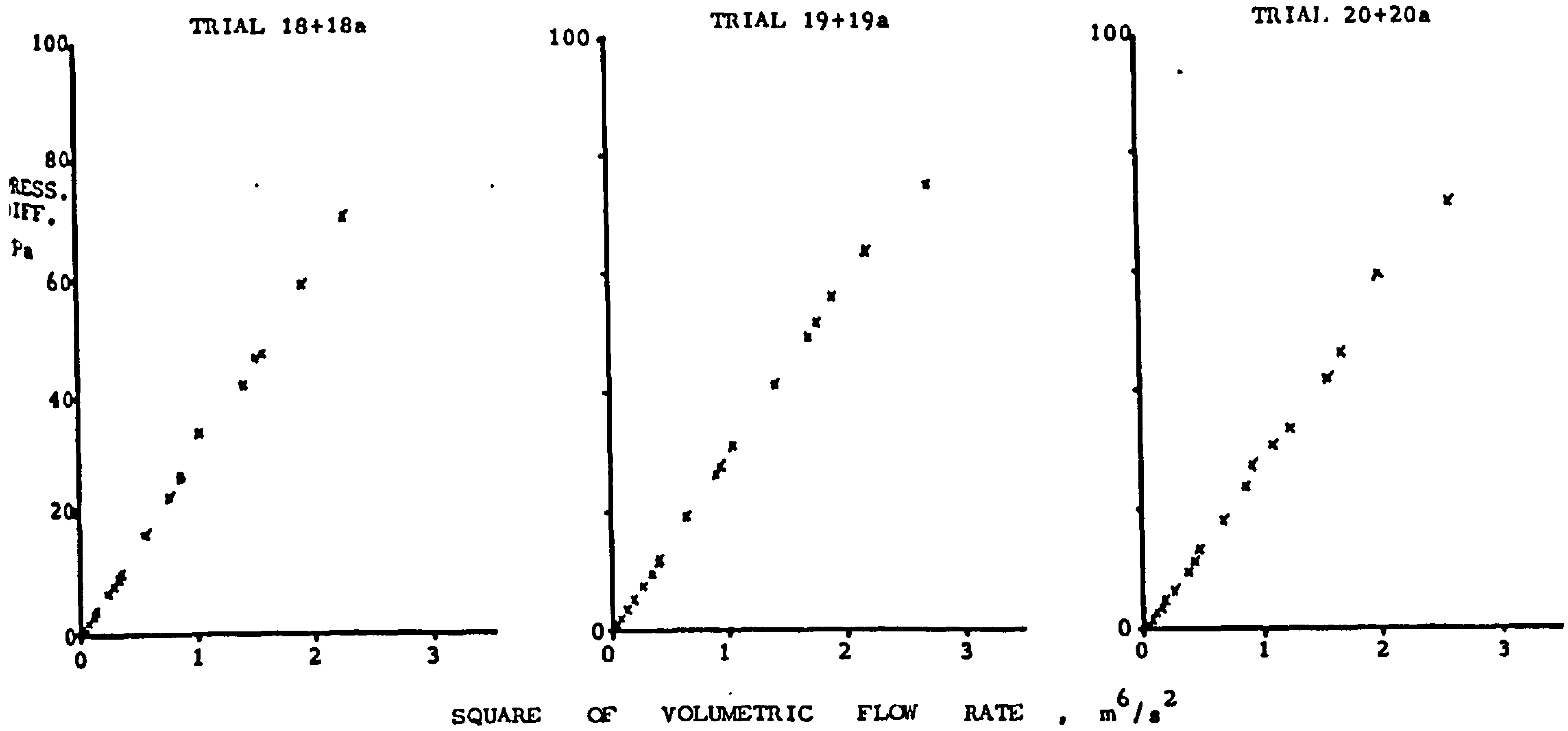


FIGURE 7.5 (cont.)



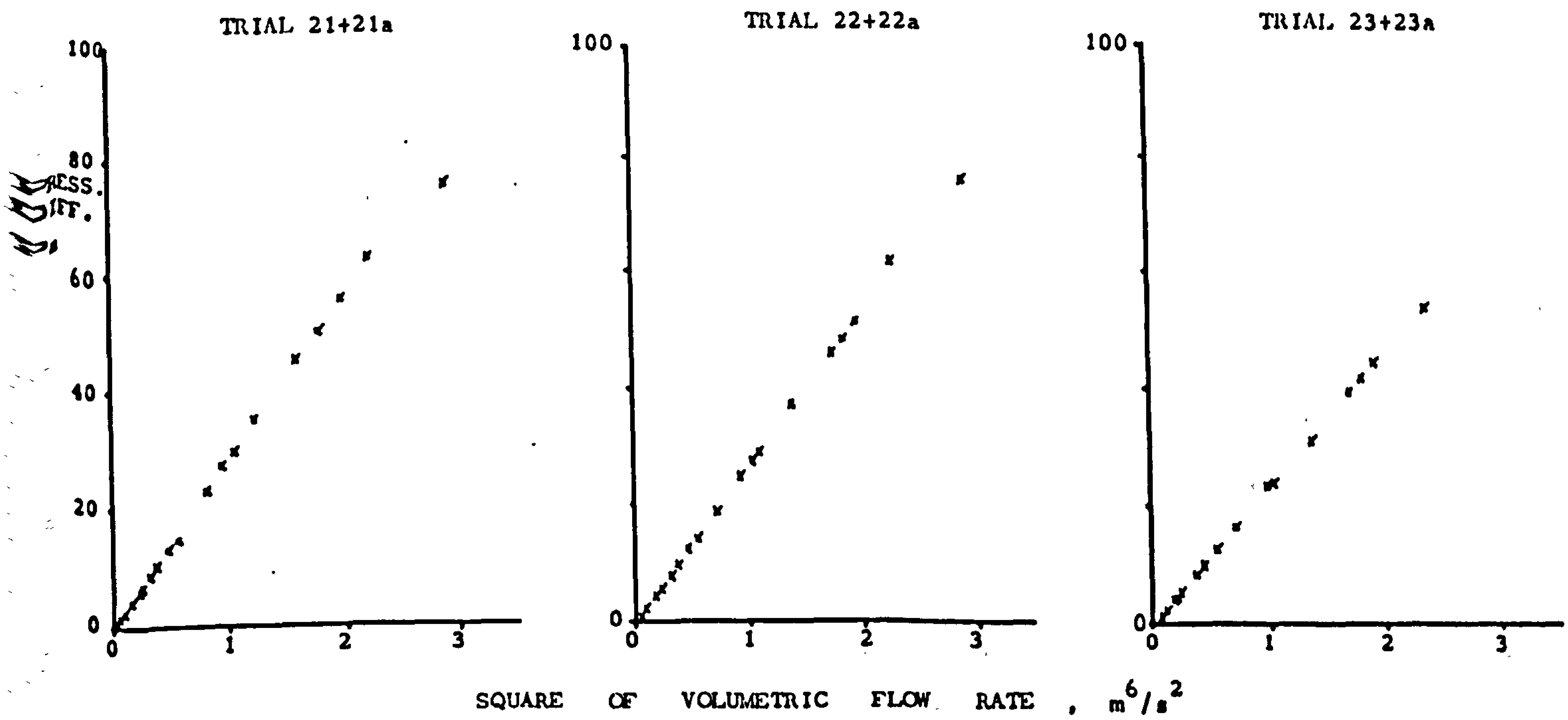


FIGURE 7.5 (cont.)

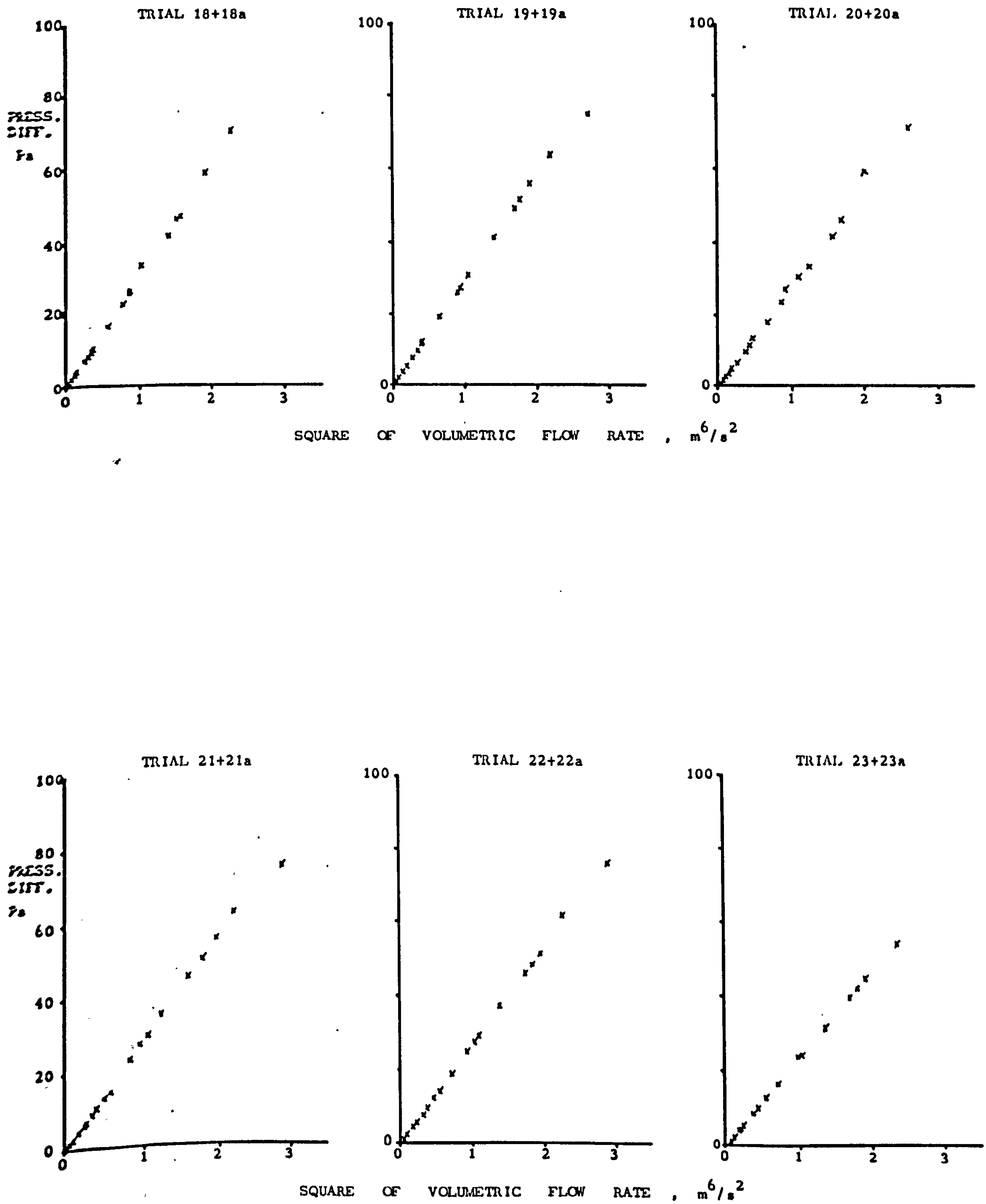


FIGURE 7.5 (cont.)

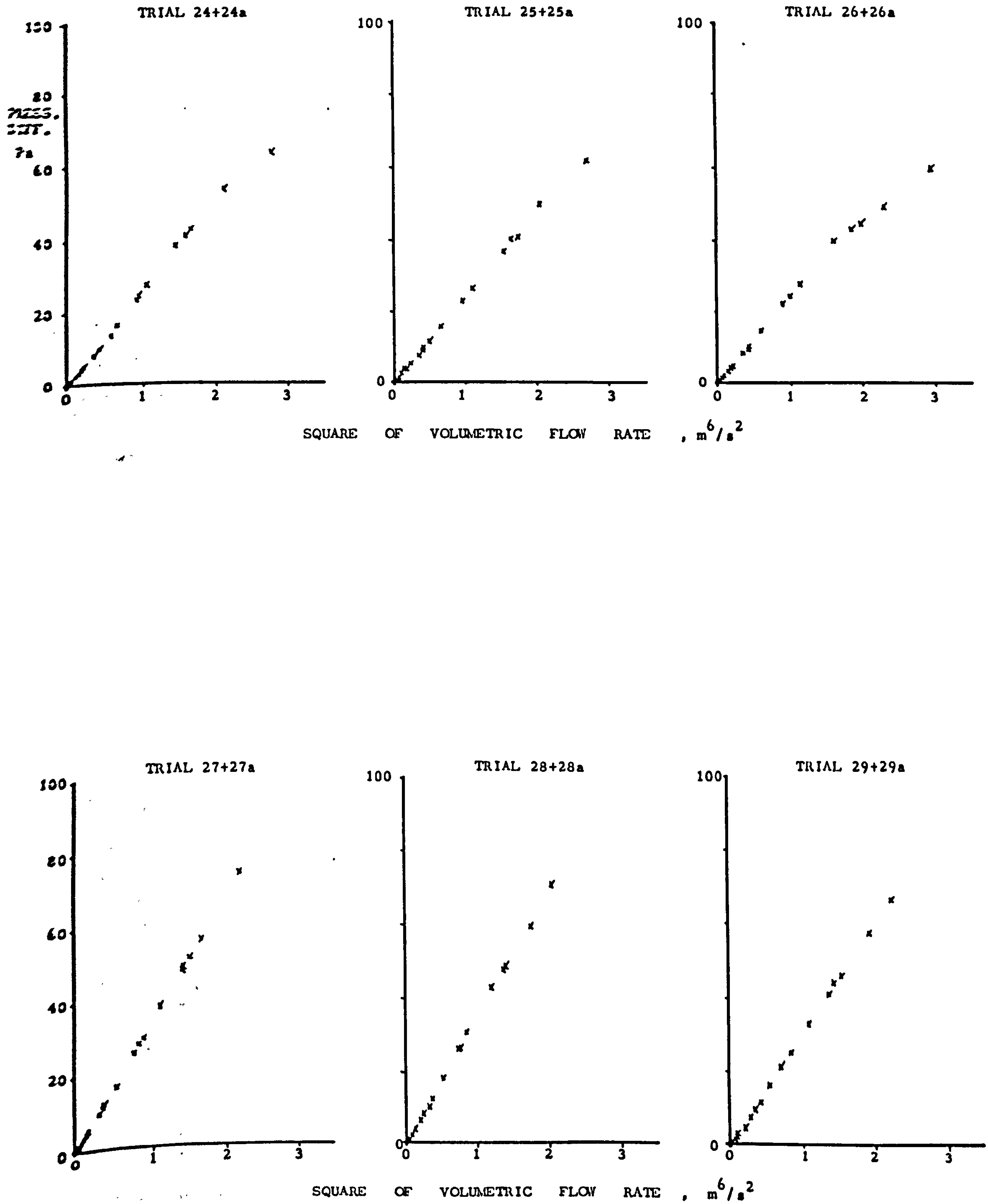
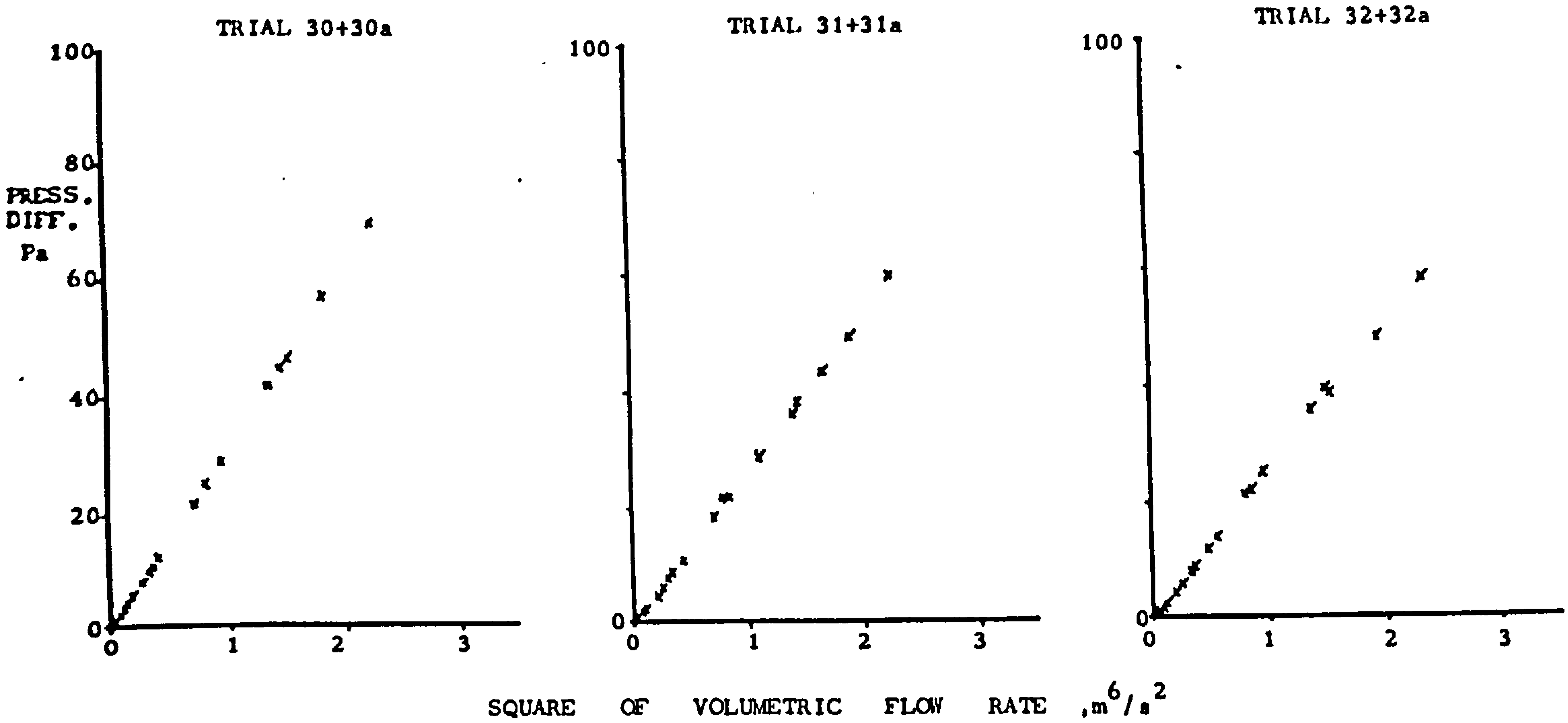


FIGURE 7.5 (cont.)



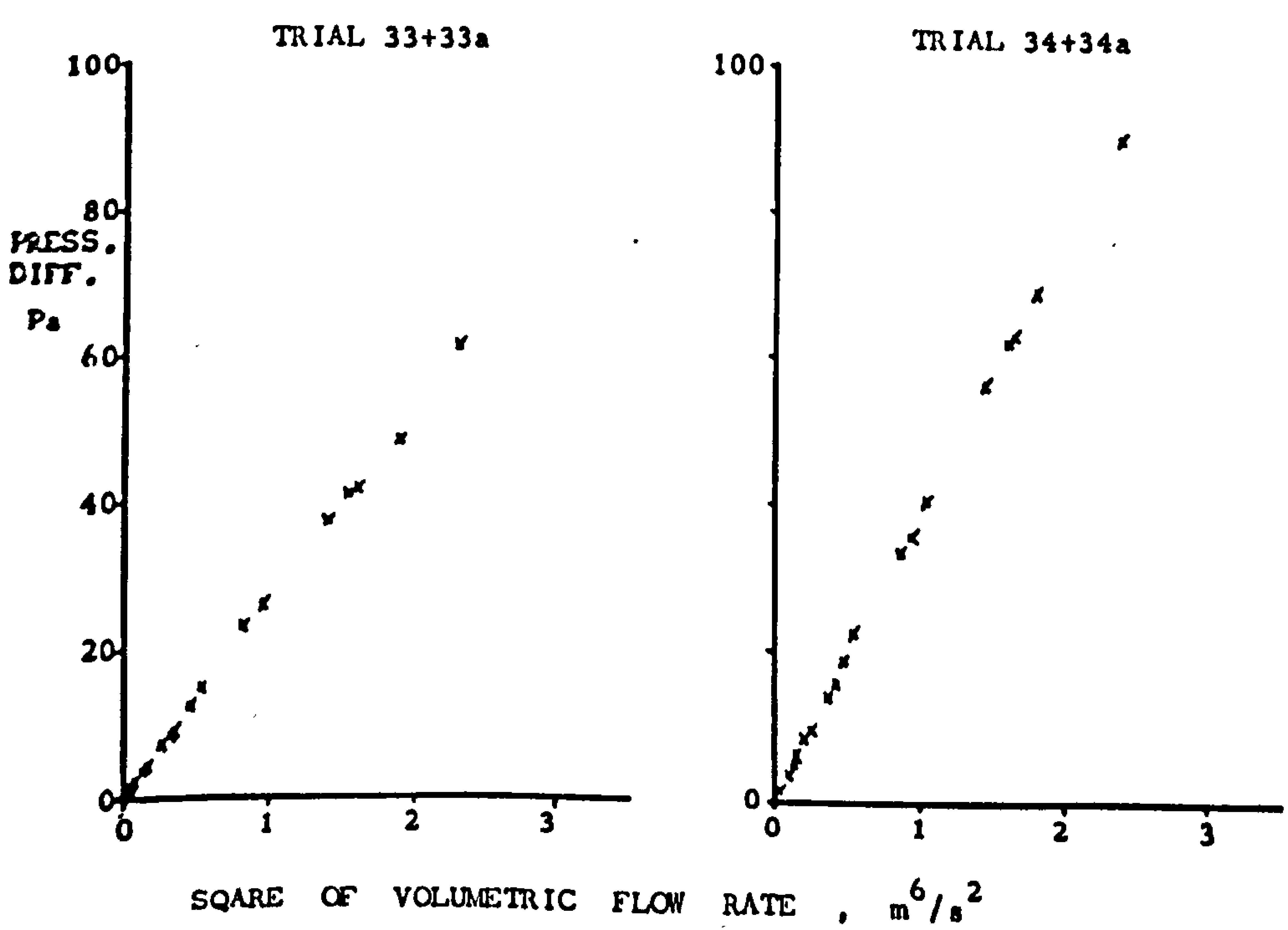


FIGURE 7.5 (cont.)

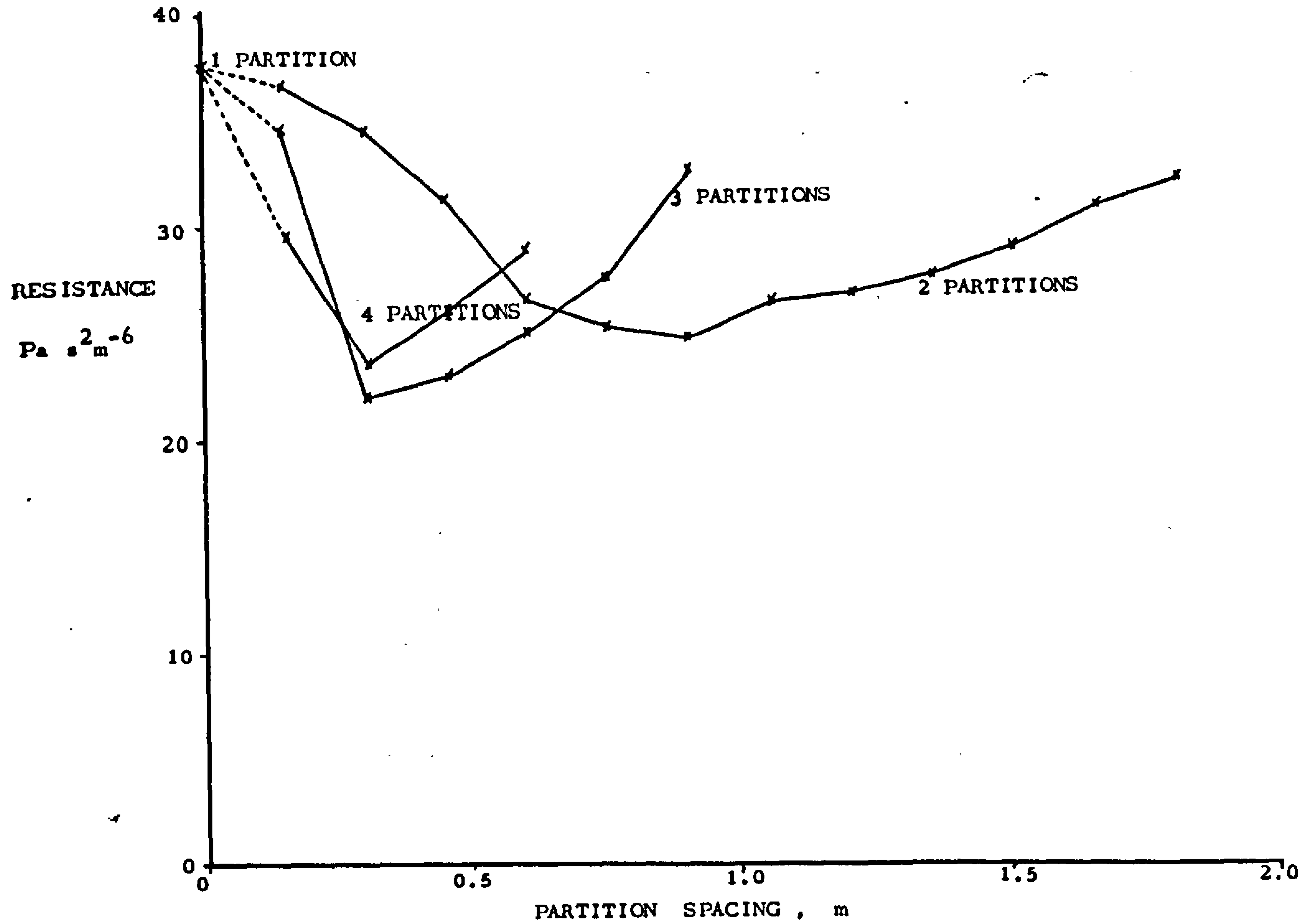


FIGURE 7.6 RESULTS FOR WALL PARTITIONS IN WIND TUNNEL TRIALS

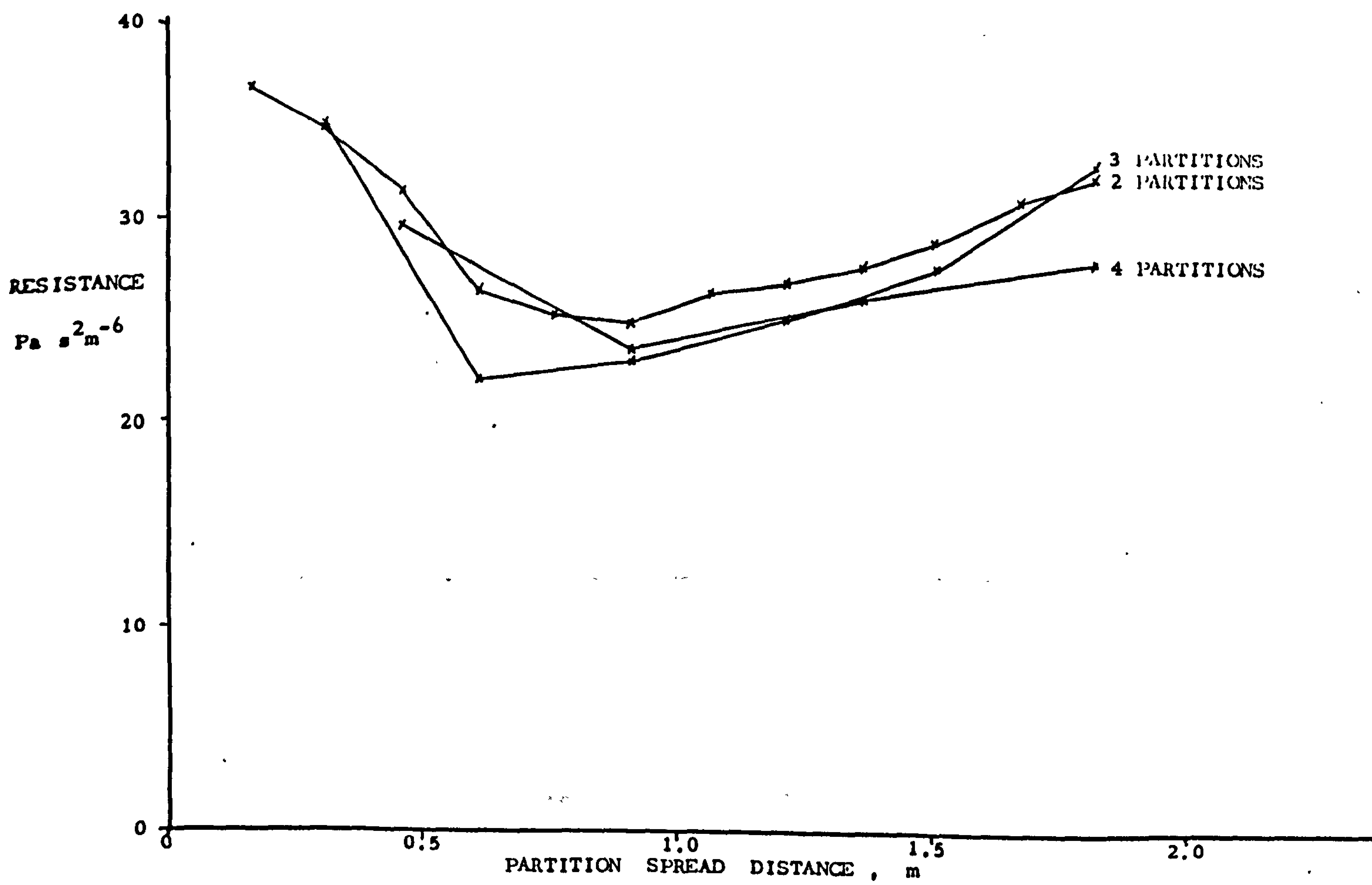
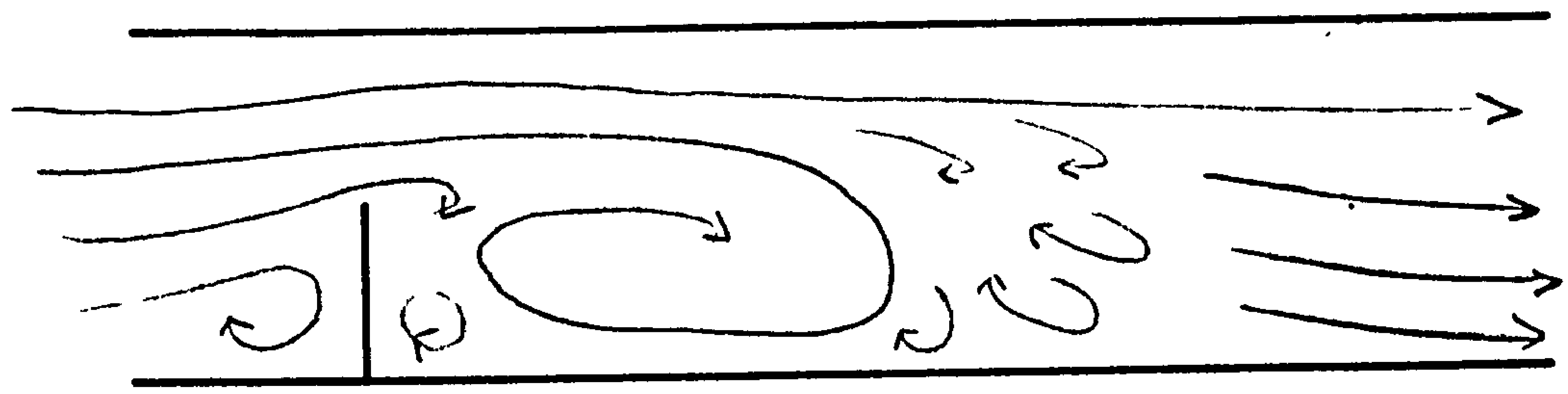
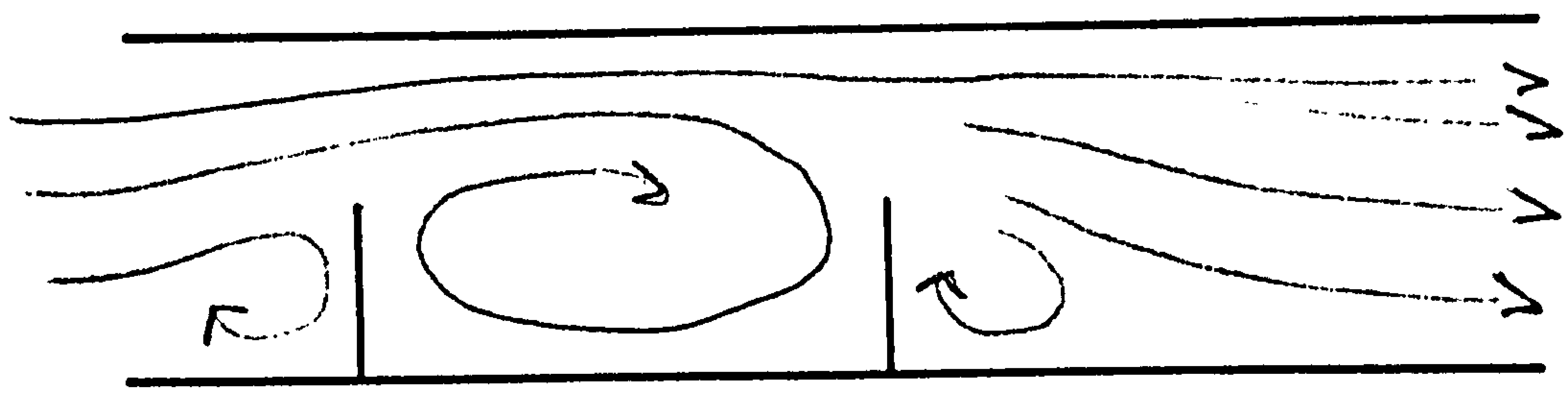


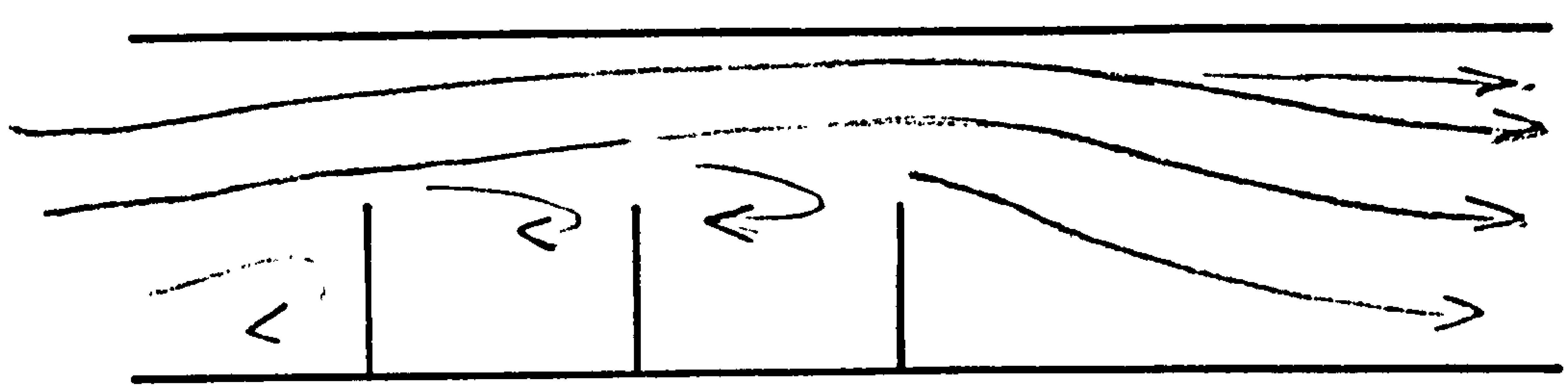
FIGURE 7.7 COMPARISON OF RESISTANCE WITH SPREAD DISTANCE FOR WIND TUNNEL TRIALS USING WALL PARTITIONS



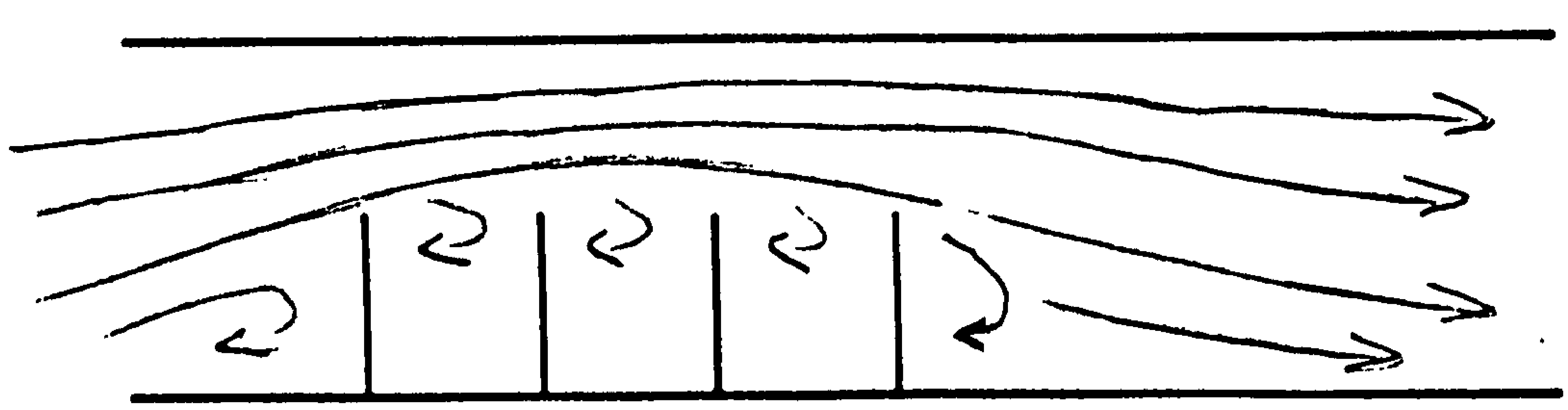
1 PARTITION



2 PARTITIONS



3 PARTITIONS



4 PARTITIONS

FIGURE 7.8 AIR FLOW PATTERNS PAST WALL PARTITIONS IN WIND TUNNEL

FIGURE 7.9 COMPARISON OF RESISTANCE WITH SPREAD DISTANCE FOR MODEL CHAMBER TESTS USING WALL PARTITIONS

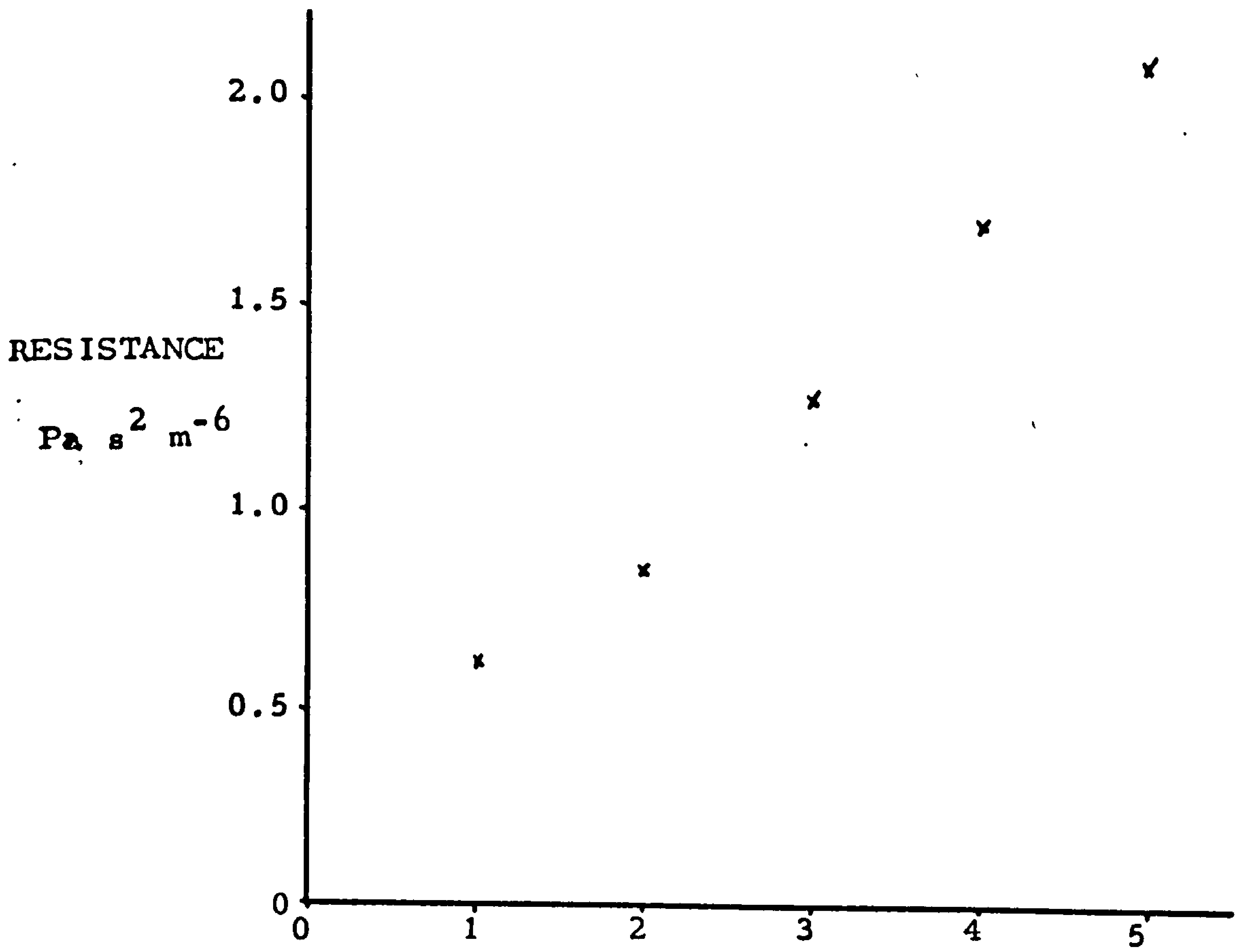


FIGURE 7.10 EXAMPLE RELATIONSHIP SHOWING RESISTANCES PREDICTED AT MODEL SCALE FOR PARTITIONS SPACED AT 1.22m.

CHAPTER 8

RESULTS AND DISCUSSION OF PLANT MONITORING

8.1 INTRODUCTION

This chapter deals in part with the results gathered during the environmental and ventilation plant monitoring period, using the Hewlett Packard data logging system. Also described is a series of experiments using a tracer gas to investigate ventilation and air transfer at the ICI factory. Upon completion of the data recording and analysis required for the work mentioned above, the monitoring system was reorganised and a program developed to enable ICI staff to continue to use the measurement of environmental parameters to aid with the production process. This further development is also described.

8.2 "CONTINUOUS" ENVIRONMENTAL AND VENTILATION MONITORING

The system was capable of monitoring temperature and other environmental parameters in the production areas and ventilation ducts, on a continuous basis with minimal attention. However, changes of monitoring points, fault detection and correction, and improvements in the monitoring scheme, meant that the resulting data was acquired in blocks of about 10 to 20 days at a time. This was recorded on magnetic tape cassettes during the period autumn 1982 to summer 1983.

At the same time as data was being recorded on tape, a printout of selected hourly averaged data, was provided in order for ICI staff to check certain readings relating to the production process. Also printed out were daily maximum and minimum readings recorded by all sensors during that day. Example copies of these printouts are shown in Figure 8.1 and 8.2 which relate to dates during the logging period, (included are some measurement locations which were omitted from the final analysis). The daily maximum and minimum printout provided an additional check for faulty sensors.

8.3 DATA TAPE RECORDS

The data recorded on tape consisted of hourly averages (averages of 6 readings at 10 minute intervals) of temperature, humidity, flow rate and energy flow (where applicable) relating to the various measurement locations.

A number of computer programs were written (mainly for use with the Hewlett Packard HP 85 computer) to inspect and check the data. Eventually four sets of records were selected for further analysis, other sets of data having been rejected due to either incompleteness or faulty readings at an important location. A faulty interface connection lead between the data logger and controlling micro-computer also caused problems which meant a number of data tapes had to be discarded. The records used were also chosen to cover periods during which production on the spinning machine was at normal full load.

The data which was used for further analysis was recorded during the following periods:

(a)	9.12.82	-	15.12.82	(141 hours)
(b)	18. 1.83	-	28. 1.83	(235 hours)
(c)	18. 2.83	-	3. 3.83	(310 hours)
(d)	4. 7.83	-	18. 7.83	(325 hours)

8.4 COMPUTER PROGRAM "ANALIS"

A computer program was developed which read in the hourly average values from a data tape for analysis. A basic functional flow diagram of this program is given in Figure 8.3 (Sheets 1 and 2)

In addition to the recorded data other information was required for analysis and calculation of energy flows. Such information (duct cross-sectional areas, buildings heat conduction data, etc.) was input as constants to the program. Standard corrections for known faults and other assumptions were included as befitted the data. Where temperature and humidity readings were available for a location, the specific enthalpy of the air was calculated (kJ/kg) as was the moisture content (g/kg air). For duct measurement locations the air density and mass flow (kg/s) were also determined. Since air was taken from the outside by the ventilation system and exhausted back to it, the condition of the outside air was taken as a basis for energy flow calculations. The energy flowing in a duct being equal to the difference between the specific enthalpy of the duct air and the outside air, multiplied by the mass of air moving along the duct.

Where the amounts of air flowing to and from an area by mechanical systems did not balance, an air flow was assumed to take place to adjacent areas to which openings existed (i.e. Drawtwist to Spin Doff and Extrusion to Hopper Floor) Energy flows associated with these air movement were estimated based on the air conditions in the areas.

Steady-state conduction heat transfers between floors and to the outside were also determined.

The energy flows to and from each area were found by adding/subtracting the various determined flows. A printout of the relevant data was then made by interfacing to a FACIT 4510 Serial Matrix Printer. A sample of the output is given in Figure 8.4, and a key to the various items is given in Figure 8.5.

8.6 RESULTS

[For details of experimental variations see Appendix D.]

A wide range of external conditions was covered by the period of the test ranging from sub-zero temperatures during the winter months to almost 30°C during the summer. The sets of results produced (e.g. see Figure 8.4) were used with a view to examining the variations due to seasonal changes and diurnal differences; also to attempt to identify relationships amongst some of the measurements and calculated values.

Conduction heat losses through the fabric of the building varied, as one might expect, with the outside temperature. However, the losses during warm spells were still very high: typically 230 kW with an outside temperature of 20°C, compared with 350 kW at 0°C. Extrapolation of the results and approximation of the relationship to a linear one, indicated that to produce nil conduction heat loss, an outside temperature of over 50°C would be required. The reason for the loss at higher outside temperatures was that at such times, internal temperatures also rose thus maintaining a significant temperature differential. This was particularly true for the Hopper Floor area which also had a large proportion of its envelope surfaces in contact with the external air. Figure 8.6(a) shows the typical variation of conduction losses (covering all four sets of data) with respect to outside temperature. Figure 8.6(b) shows Hopper Floor temperature variations with outside temperature, again using samples of all sets of results.

The temperature within the Spin Doff area was maintained within the range 25-27°C for most of the time. During particularly warm spells, this was apt to rise up to about 30°C; whilst even at the coldest time it rarely dropped below 24°C. This indicated that the heating apsect of the conditioning system was able to cope with prevailing conditions but casts some doubt on the ability of the evaporative spray cooling system to deal with warm, humid days.

Temperatures within the Extrusion floor also varied but in a less definite manner. Temperatures, especially at Extrusion Catwalk level were very high particularly during the summer months. However, the evaporative spray coolers in the air conditioning plant for this area of the factory, were operated during the summer period thus reducing the temperatures below what might otherwise have been expected. Good correlation between outside temperature variation and extrusion temperature variations could be found in the short term, but comparisons over longer periods was less conclusive - probably due to variations in the operation of the plant causing modification of the heating load.

Significant flow imbalances were detected for both the Spin Doff and Extrusion areas when only mechanical ventilation was considered. In the Spin Doff area, although the S plant supply fans were designed to provide a greater volume of air than the associated extracts were designed to exhaust, the removal of air by the Blower air system had the effect of producing a net defficit. This could only be met by an influx of air from the Drawtwist area (and to a lesser extent from other adjacent areas). Though the air in the Drawtwist area was also conditoned, as a general rule it is usual to attempt to avoid such influxes into conditioned areas in order to maintain control.

In the Extrusion area there appeared to be an excess of supply over extract in the measured duct flows. The balance in this case being made up by a loss of air to the Hopper floor area above.

To illustrate the variations and significance of the various energy flows, Figures 8.7 to 8.10 have been constructed. These show the flows in kW for both Spin Doff and Extrusion areas referring to one machine pair. In general the energy flows are those associated with the flow of air; and the energy flows were calculated in these cases by multiplying flow rate by specific enthalpy.

The figures are based on average conditions and flows for the areas at two times in the year - winter and summer. Outside air temperatures for these periods being typically 2-7°C and 20-25°C respectively.

In all cases the fabric conduction heat losses are comparatively small. It can also be seen that the heat liberated by the process is not as overriding as might have been expected, though it is certainly significant, especially in consideration of the Extrusion area.

These results showed, at an overall plant level, that the heat load of the systems was not so great as to be uncontrollable. However, individual measurements and readings indicated that the temperatures sometimes became excessive and that conditions required in the Spin Doff area were not always met.

In order to further assess the ventilation systems, the recorded figures were analysed with a view to considering their efficiency.

8.6 VENTILATION EFFICIENCY

In recent years Sandberg [1][2] has developed the concept of "ventilation efficiency" as a means of judging ventilation systems, in particular systems incorporating mechnaical supply or exhaust, or both. The judgement is based on being able to measure a property of the supply and extract air streams and also that property at specified points within the space or enclosure under consideration. Generally the measured property is the concentration of a contaminant in the air and the ventilation efficiency describes the ability to remove the contaminant. Alternatively a tracer substance may be artificially introduced into the environment for test purposes.

Where long-term/continuous discharges of contaminent or tracer are concerned, the ventilation efficiency is simply described:

$$\text{Efficiency, } E_v = \frac{C_e - C_s}{C_j - C_s} \times 100 \qquad (8.1)$$

where C_e is the concentration in the exhaust duct

C_s is the concentration in the supply duct

C_j is the concentration at the point under consideration

A high efficiency is produced when the point considered has a low concentration due to good supply of fresh air and when the extracted air contains as high a concentration of contaminant as possible. It should be noted that efficiencies in excess of 100% are possible, indeed they are desirable to provide expulsion of the maximum amount of contaminant, and provision of maximum "fresh" (supply) air.

Sandberg extended the idea of efficiency to warm air heating systems to provide a measure of thermal or temperature efficiency. The relationship was a simple modification of equation 8.1.

$$\text{Efficiency, } E_t = \frac{T_e - T_s}{T_j - T_s} \times 100 \qquad (8.2)$$

where T = Temperature, subscripts as before.

As in the previous case it is desirable that the temperature at the point in the space be close to that at supply : that is, only a small drop; whilst the temperature at the extract should be the coldest air possible in the room.

Sandberg used these definitions of ventilation and heating systems in housing with particular interest in the positioning of inlet and outlet grilles. Together with Svensson (3) he also used thermal efficiency to values the ability of a ventilation system to remove heat. This uses the same equation (8.2) however in this case the temperatures in the extract should be as high as possible.

Such measurements, however, consider only the differences between the sensible heat contents of the air, and not latent heat gains/losses. These may be of significant proportions in industrial rather than domestic or commercial situations. In fact there are occasions when the changes in moisture content or humidity level of the air are the prime considerations. In such cases a moisture removal/dehumidifying, or moisture supply/humidification efficiency could be defined with moisture content replacing temperature in equation 8.2.

In the context of this project which is concerned with the industrial process of nylon fibre production, air humidity is an important factor and changes in moisture content of the air are in evidence. Also with regard to the cooling ventilation provided in parts of the production areas, the latent heat transfers ought to be considered. Whilst moisture content is not as important as temperature in determining comfort levels it is still significant, and it should be incorporated when ascertaining energy flows. In such cases it would be more useful to use values of specific enthalpy for the air, and to determine a total thermal efficiency:

Total Thermal Efficiency,

$$E_{tt} = \frac{E_e - E_s}{E_j - E_s} \times 100 \qquad (8.3)$$

where E = Specific Enthalpy, subscripts as before.

Using this Total Thermal Efficiency to evaluate ventilation would allow comparisons of duct arrangements for maximum heat removal, though not necessarily maximum comfort.

If total mixing of fresh, supply air were to occur to produce a homogeneous mixture in the area being considered, then the conditions measured at the point under consideration and the conditions in the extract air would be the same. This would give an "efficiency" of 100% using any of the equations 8.1, 8.2, 8.3. Such a situation can be considered as the base by which to judge other systems, efficiencies in excess of 100% being the desired deviation.

The recorded data of temperature and humidity, levels within the factory and of the supply and extract ducts enabled the specific enthalpy values to be calculated. Thus determination of both temperature and total thermal efficiency of the ventilation system was possible. The principal factory area of interest was the Extrusion floor since it was in this area that heat was required to be removed. Where more than one duct was involved the measurement used was a mass flow weighted average. Efficiencies were calculated for each complete day of the four test periods which have been considered in this study.

[N.B. For details of experimental variations see Appendix D]

The average temperature efficiency for the the Extrusion area for periods between December 1982 and March1983 was 99.2%, whilst for the same periods the total thermal efficiency was less at 87.6%. During the summer test period (July 1983) the average temperature efficiency was 87.8% but the total thermal efficiency averaged only 65.4%. These figures show that though the removal of high temperature air is not performed very well, the removal of (total) heat is even worse. Figures were calculated for the Spin Doff area for comparison purposes, though of course the function of the air conditioning system is different to that of the Extrusion area. For the December to March periods the average temperature efficiency was 140.6% and the average total thermal efficiency 198.4%; for the July period these figures were 175.5% and 66.3%. In each category the Spin Doff duct arrangements out perform the Extrusion duct systems. It is noteworthy however, that whilst the temperature efficiency improved for the July period the total thermal efficiency fell markedly. The reasons for this drop were that during the summer months the system was operated with little or no recycled air and it was this element which had the stabilizing effect during the winter. Also on hot humid days the evaporative spray coolers were unable to reduce temperature sufficiently for the supply and in addition (as a consequence) the moisture content of the supply air would be higher than normal. This resulted in an absormally high specific enthalpy for the supply air which affected air conditions in the area with the eventual reduction in total thermal efficiency.

Since the duct layouts in the Extrusion area were limited by process and machine access requirements, this can explain some of the poor levels of efficiency in this area. Also a significant volume of air escaped from the Extrusion area to the Hopper floor area above either through the designed openings (for stairs, etc.) or through the cracks around the hatch openings above each spinning unit. This air was assessed to take considerable quantity of heat with it thus lowering the potential performance of the extract ducts.

Odd flow patterns, especially at the Extrusion Catwalk level have been noted in Chapter 2. These showed that the flow to the extract duct was not, as one would have expected, from the adjacent machine, but rather from above the machine behind the duct. This fact, plus the results produced in this section suggest scope for improvement. This is discussed in Chapter 9.

8.7 INVESTIGATION USING TRACER GAS

8.7.1 INTRODUCTION

In order to investigate the ventilation rate and air transfers in the spaces between the spinning machines at the factory, it was decided to use nitrous oxidee (N_2O) tracer gas. The investigation had to be restricted to one main space/alleyway and its neighbours because of the difficulties in monitoring a larger area and the large quantities of tracer gas that would have been required to provide measurable concentrations in a larger space.

The experiments could not be carried out during normal production periods, but during the Christmas shutdown week, the plant conditions were very similar to those normally found. Additionally, during the shutdown period plant items could be adjusted to suit the requirements of the experiments.

For the investigation measurements of tracer gas concentrations were made concurrently with measurements of mechanical ventilation flow rates and pressure differentials existing across the area under investigation.

Nitrous Oxide was chosen as the tracer gas as this had been previously used by the Department at the University and suitable apparatus was available. In order to check for potential problems in its use at the factory ICI personnel were consulted to advise on any effects or interactions which might occur - none could be envisaged. Since the tests were to be performed during non-production periods, no adverse reactions could be foreseen for the nylon fibre itself.

[N.B. For details of experimental variations see Appendix D]

8.7.2 APPARATUS

For the liberation of the tracer gas, cylinders of nitrous oxide with a suitable pressure reduction valve, were used. The flow of gas was monitored using a vertical tube gap flowmeter. To ensure adequate mixing of the gas with the air, a mixing device, using a small fan, was utilized and positioned centrally in the alleyway/area under study.

In order to draw air for sampling, flexible plastic tubing was connected between metal inlets and a multi-way mixing valve box. The inlets were mounted on vertical poles and their height was adjustable between ground level and about 2.5m. All the tubing was of identical bore and length. The multi-way mixer allowed the selection of different inlets for sampling purposes. The outlet from the mixer valve was attached to a small suction pump which delivered the air to an Infra-Red Gas Analyser. The gas analyser determined the concentration of nitrous oxide in the sample of air, from the degree of Infra Red light absorption.

The pressure differential existing across the machine alleyway under test, and its immediate neighbour on each side, was monitored using the BRE type pressure transducer and amplifier (as described in Chapter 6). The output from this amplifier and the measurement of gas concentration from the analyser, were recorded on a Linseis multiple pen, chart recorder.

Some variations in pressure could be created using the factory's ventilation systems, but were created in these tests using extra movable fans with long flexible ducting attachments.

The flow rates of the mechanical ventilation systems was monitored using vane anemometers set in each duct (as described in Chapter 4).

8.7.3 EQUIPMENT CALIBRATION

The Infra-Red Gas Analyser was calibrated using a test gas sample of known nitrous oxide concentration to ensure operation in the correct range. Initial tests in the factory environment indicated no reason to suspect that factors in that environment would affect the readings. The time lag for air entering the system at the monitoring point to reach the analyser was determined as 26 seconds (Since identical tubing had been used for each point the time lag would be the same).

The pressure transducer was calibrated in the usual way, using weights placed upon the pressure plate (method as described in earlier chapters) Figure 8.11 shows the result (Gradient 0.253 Pa/mV).

8.7.4 METHOD

The experiments were performed in the Spin Doff area of the factory; the layout of the apparatus is shown in sketch form in Figure 8.12. Plate 4 illustrates the test environment. Two types of

Following Page:

PLATE 4

GENERAL ARRANGEMENT FOR
TRACER GAS TESTS,
SHOWING GAS INPUT AND
SAMPLING POINTS, AND
DATA RECORDING EQUIPMENT.

investigation were undertaken; one using tracer gas concentraion decay rates, the other measuring equilibrium gas concentrations achieved with a known gas input. Each set of tests was carried out under three types of prevailing plant conditions these being:-

(i) Normal ventilation plant operation

(ii) Background ventilation (i.e. ventilation switched off in area of test)

(iii) Normal ventilation off, additional fans switched on in order to create a pressure differential

Between each trial using the tracer gas a period of time was allowed to elapse. This allowed any residual nitrous oxide to be removed or diluted to negligibly small concentrations.

The dimensions of the space under investigation were length : 15m, width : 3m, height : 3.5m giving an overall volume of 157.5m^3.

Twelve sampling inlet points were used, four in the main "alleyway" under test and four in each of the adjacent alleyways. The inlet points were positioned using vertical poles, two inlets per pole, one at a height of 1m from the ground, the other at a height of 2m. This meant that there were two poles per alleyway and they were positioned centrally with a separation of 3m. For the decay experiments, only the main central alleyway was investigated.

8.7.5 TRACER GAS DECAY

The mathematics of the use of tracer gas concentration decay to predict ventilation rates has already been discussed in Chapter 3.

To reiterate:

$$[T]_t = [T]_o e^{-Nt} \tag{8.4}$$

where $[T]_t$ = Concentration of Tracer at time t

$[T]_o$ = Concentration of Tracer at start

N = air exchange rate (per second)

$$\frac{d[T]_t}{dt} = -N[T]_t = \frac{-Q}{V}[T]_t \tag{8.5}$$

i.e.

$$N = \frac{Q}{V} \tag{8.6}$$

where Q = air flow m^3/s

V = volume of space, m^3

The measurements of the gas concentration decay are given in Tables 8.1, 8.2 and 8.3 (representing Trial A - Normal plant ventilation, Trial B - Background ventilation and Trial C - extra fans to create pressure difference, respectively).

In order to determine the exponential decay a computer program ("CURVE") available on one of the University's computers, was used to fit a curve to the data using ordinary least squares regression. The curves produced had the following equations (N.B. for convenience time periods of 100 seconds chosen).

TABLE 8.1 DECAY OF NITROUS OXIDE CONCENTRATION - TRIAL A

TIME (SECONDS)	CONCENTRATION (Parts per million)
0	103
10	96
20	77
30	64
40	52
50	34
60	24
70	22
80	15
90	10
110	7
120	6
140	4

TABLE 8.2 DECAY OF NITROUS OXIDE CONCENTRATION - TRIAL B

TIME (SECONDS)	CONCENTRATION (Parts per million)
0	157
10	152
20	149
30	149
40	142
50	135
60	134
70	122
80	117
90	114
100	114
110	108
120	96
130	84
140	84
150	81
160	80
170	81
180	72
190	60
200	68
210	76
220	67
230	67
240	52
250	47
260	43
270	42
280	42
290	40
300	39
310	37
320	36
330	35
340	33

TABLE 8.3 DECAY OF NITROUS OXIDE CONCENTRATION - TRIAL C

TIME (SECONDS)	CONCENTRATION (Parts per million)
0	67
10	53
20	43
30	36
40	26
50	22
60	17
70	16

Trial A :

$$\text{Concentration} = 124.e^{-2.64t} \quad (8.7)$$

$$(R^2 = 0.984)$$

Trial B :

$$\text{Concentration} = 174.e^{-0.51t} \quad (8.8)$$

$$(R^2 = 0.977)$$

Trial C :

$$\text{Concentration} = 66.e^{-2.15t} \quad (8.9)$$

$$(R^2 = 0.99)$$

To calculate the air flow to and from the space, the exponential power was multiplied by the volume of the space and divided by 100 to obtain flow per second.

Trial A - normal ventilation plant

Air flow = (2.65 x 157.5)/100 = 4.16 m^3/s

Trial B - background ventilation

Air flow = (0.51 x 157.5)/100 = 0.8 m^3/s

Trial C - additional fans only operating

Air flow = (2.15 x 157.5)/100 = 3.39 m^3/s

8.7.6 EQUILIBRIUM CONCENTRATION

If a tracer gas is liberated, at a known rate, into a space, the equilibrium concentration achieved enables the rate of dilution to be found and hence the rate of influx of new air into that space. This can be evaluated from the following equation in which the reciprocal can be equated to the "Transfer Index" as described by Lidwell[4].

$$\frac{1}{\text{Air Supply or Extract Rate}} = \frac{\text{Equilib conc tracer at point}}{\text{Tracer liberation rate}}$$

$$= \text{Transfer Index}$$

For the first test, the ventilation plant was set up to operature as normal and the tracer gas was liberated at a rate of 0.6 litres/second. The concentration found in the main central alleyway (marked B in Figure 8.12) was 82 parts per million. This concentration was unevenly distributed however as further readings showed the equilibrium for the measurement points 2m from the ground to be about 40 p.p.m. whilst at 1m from the ground it was approximately 125 p.p.m. The equilibrium concentrations found in the adjacent alleyways (due to transfer from the central one) was 3 p.p.m. and less than 1 p.p.m. in areas A and C respectively (as marked in Figure 8.12).

In the second test run, normal ventilation systems were switched off to leave only the background ventilation. This resulted in concentrations of 150 p.p.m. in the central (B) alleyway; 30 p.p.m. in alleyway A and 8 p.p.m. in alleyway C. In this case the rate of tracer liberation was 0.3 litres second.

The third test run was also carried out with a liberation rate of 0.3 litres second and produced the following equilibrium concentrations : 65 p.p.m. in alleyway B ; 4 p.p.m. in A; 8.5 p.p.m. in C.

For the first test the transfer index for the test area B is calculated to be 0.137 s/m^3 and thus the total air flow in/out was 7.3 m^3/s. In the second test the alleyway had a transfer index of 0.5 s/m^3 equivalent to an air flow of 2 m^3/s. The third test gave a transfer index of 0.217 s/m^3, that being an air transfer for the alleyway of 4.62 m^3/s.

8.7.7 MEASURED DUCT FLOWS

During the course of the tests, vane anemometers were used to record the air supply and extract rates due to the mechanical ventilation systems. The average supply rate was 4.5 m^3/s and the average extract was 3.7 m^3/s. In the general area there is another extract to be considered, this being the air taken by the Blower Air System. Since the inlet to this system was spacially separated from the areas under test, it has not been included here, though the effect would be felt as part of the background ventilation.

8.7.8 PRESSURE DIFFERENCES

The average pressure differences recorded across the alleyways, between points 1 and 2 (Figure 8.12) were:

Normal Plant Operation	:	+2mV	=	+0.51 Pa
Background Ventilation	:	+7mV	=	+1.77 Pa
Extra Fan Operation	:	-3mV	=	-0.76 Pa

These figures represent the pressure difference for four partitions, therefore the pressure differences per partition in each case were 0.13 Pa, 0.44 Pa and -0.19 Pa respectively.

8.7.9 DISCUSSION OF RESULTS OF TRACER GAS TESTS

When the normal mechanical ventilation systems were operated, the tracer gas decay technique gave a ventilation rate of 4.16 m^3/s whilst the equilibrium concentration method estimated the rate at 7.3 m^3/s. The air actually supplied by the ventilation systems was averaged at 4.5 m^3/s. These results show quite a degree of variability.

One would expect the true "ventilation rate" to be higher than that measured as due solely to the mechanical systems, because of flows to adjacent areas. The reason for the low ventilation rate calculated using the tracer gas decay method is not certain. However, with the fast decay rates experienced a degree of variability was introduced particularly with regard to the non-homogeneous state of the air. The variability was illustrated by the equilibrium conditions measured at heights of 1 and 2 metres.

It was decided to place more emphasis on the results of the equilibrium concentration tests since this method ought to eliminate some of the short temporal variations which might have affected the decay technique. Also it was assumed that the air transfer due to the mechanical ventilation systems was 4.5 m^3/s (i.e. the measured rate) and that air supply in the two adjacent areas due to this cause was the same, (since they were all served from common header ducts). Because of the proximity of the adjacent alleyways it would also seem likely that other external effects would act equally on each of the partitioned areas.

It might therefore be assumed that the air transferred to and from a machine alleyway under normal plant operating conditions at Spin Doff level was 7.3 m^3/s. Of this 4.5 m^3/s was caused by the mechanical ventilation systems serving an alleyway. The remainder was accounted for by flow to adjacent alleyways and flows out of the alleyways to other areas and the blower air system inlet.

The actual flow to the adjacent areas can be estimated:

Air Flow (Area 1 to 2)

$$= \frac{\text{Tracer conc. Area 2}}{\text{Tracer conc. Area 1}} \times \text{Air Suppy Area 2} \qquad (8.11)$$

Therefore:

Air transferred from central alleyway (B) to alleyway (A)

$$= (3\text{ppm}/82\text{ppm}) \times 7.3\ m^3/s = 0.27\ m^3/s$$

Air transferred from central alleyway (B) to alleyway (C)

$$= (1\text{ppm}/82\text{ppm}) \times 7.3\ m^3/s = 0.09\ m^3/s$$

Similarly the transfers under the different test conditions can be estimated

Background Ventilation:

Air transferred from central alleyway (B) to alleyway (A)

$= (30ppm/150ppm) \times 2\ m^3/s = 0.4\ m^3/s$

Air transferred from central alleyway (B) to alleyway (C)

$= (8ppm/150ppm) \times 2\ m^3/s = 0.11\ m^3/s$

Extra Fans:

Air transferred from central alleyway (B) to alleyway (A)

$= (4ppm/65ppm) \times 4.62\ m^3/s = 0.28\ m^3/s$

Air transferred from central alleyway (B) to alleyway (C)

$= (8.5ppm/65ppm) \times 4.62\ m^3/s = 0.6\ m^3/s$

These figures indicate that when the ventilation plant was operated "normally", there was little air transfer to adjacent alleyways. When only background ventilation operated the transfers increased by about one third. Using extra fans to produce a pressure difference altered the distribution and increased the flow to the adjacent area. If we use as a criteria of significance of flow, 10% of the total alleyway ventilation (i.e. 10% of 7.3 m^3/s), then only this last case gave rise to such a level of flow. In the other cases, each alleyway was relatively isolated from its neighbours.

The difference between the ventilation rates predicted by the equilibrium tracer gas concentration method and the measured duct flows was in the main due to flows out of the ends of each alleyway, primarily due to the Blower Air intakes. For the flows between alleyways to become important a fairly substantial pressure difference would be required. This relative isolation of each alleyway could have considerable significance as regards plant operation and this is to be discussed in Appendix C.

8.8 FURTHER ENVIRONMENTAL MONITORING AT ICI FIBRES

8.8.1 USE OF DATA LOGGER

After the completion of the data recording carried out for the curent work, the data logging/monitoring system (Hewlett Packard 3054DL) was to continue being made use of. Besides the program "FANLOG" described in Chapter 4, which produced information printed out on the computers thermal printer, further programs were developed. The main requirement was for a relatively easily usable program which would provide regular printouts of information which would aid members of ICI staff with plant and process operation. This program was named "DATLOG".

8.8.2 "DATLOG" COMPUTER PROGRAM

This program was designed to produce a printed record at regular intervals, of environmental and associated measurements made in certain parts of the factory. The basic functional flow diagram is given in Figure 8.13 (Sheets 1 and 2).

Up to 45 measurement locations could be identified and within the overall limit of 100 input channels, upto four different measurements could be made at each location (assuming of course, that suitable sensors and transducers were supplied and wired at the locations). The measurements were temperature (degrees centigrade) humidity (% r.h.), duct flow rate (m/s) and voltage (volts). The derived measure of moisture content (g/kg air) was also calculated where applicable.

As an alternative to allowing the operator to type in relevant input channel numbers and location names, a standard set of data could be chosen and entered directly from the program. This standard set covered all normally connected channels.

If an error was detected then a flag would be set in the program and this would be noted in the printout.

Scanning of channels and printout was set to take place at intervals determined by the operator at the start up of the program. The information was printed out by a FACIT 4510 Serial Matric Printer and a typical output is shown in Figure 8.14. The printer was positioned in one of the process areas so that the information was quickly available to staff. The most important parameters printed, as far as the basic nylon production process was required, were humidity and moisture content. The printer was positioned in one of the process areas so that the information was quickly available to staff. The most important parameters printed, as far as the basic nylon production process was required, were humidity and moisture content. The printout has allowed almost immediate detection of air conditioning changes and malfunctions by reference to these values.

REFERENCES

1. SANDBERG, M., "What is Ventilation Efficiency?" Building and Environment, Vol. 16, No. 2, 1981.

2. SANDBERG, M., "Definition of Ventilation Efficiency and the Efficiency of Mechanical Ventilation Systems." Paper 13 - Energy Efficient Domestic Ventilation Systems for Achieving Acceptable Indoor Air Quality, 3rd AIC Conference, Septemer 1982, London, UK.

3. SANDBERG, M. and SVENSSON, A., "The Ventilation and Temperature Efficiency of Mechanical Ventilation Systems" Proceedings of CIB W67 Third International Symposium Energy Conservation in the Built Environment, Vol. 6, Dublin 1982.

4. LIDWELL, O.M., "the Evaluation of Ventilation ", Journal of Hygiene, Vol. 58, p.297, Cambridge 1960.

DATE 4 / 4 /1983

HOUR		POSITION NO. 1	2	3	4	5	6	7
00	T(C)	5	23	25	26	26	24	21
	R.H.	58	58	48	42	45	51	77
	MOIST	3	10	10	9	9	10	12
01	T(C)	4	23	26	26	26	24	21
	R.H.	57	58	47	42	45	52	77
	MOIST	3	10	10	9	9	10	12
02	T(C)	3	23	25	26	26	24	21
	R.H	64	58	48	43	45	50	76
	MOIST	3	10	10	9	9	9	12
03	T(C)	3	23	25	26	26	24	20
	R.H.	67	58	49	42	45	51	76
	MOIST	3	10	10	9	9	10	11
04	T(C)	3	23	25	26	26	24	20
	R.H.	67	58	49	42	45	52	76
	MOIST	3	10	10	9	9	10	11
05	T(C)	4	23	25	26	26	24	20
	R.H.	62	59	48	42	44	51	76
	MOIST	3	10	10	9	9	9	11
06	T(C)	4	23	25	25	26	23	20
	R.H.	62	59	49	43	45	52	77
	MOIST	3	10	10	9	9	9	11
07	T(C)	4	23	25	26	26	23	20
	R.H.	63	59	50	44	45	53	77
	MOIST	3	10	10	9	9	9	11
08	T(C)	6	23	25	26	26	23	20
	R.H.	55	59	49	43	45	54	76
	MOIST	3	10	10	9	9	9	11
09	T(C)	7	23	26	26	26	23	20
	R.H.	50	59	49	43	45	56	77
	MOIST	3	10	10	9	9	10	12
10	T(C)	8	23	25	26	26	23	21
	R.H	44	59	49	42	46	56	77
	MOIST	3	10	10	9	9	10	12
11	T(C)	9	23	26	25	25	23	21
	R.H.	33	59	49	42	45	55	75
	MOIST	2	10	10	9	9	10	12
12	T(C)	10	23	26	26	26	23	21
	R.H.	28	59	48	41	45	54	76
	MOIST	2	10	10	9	9	10	12
13	T(C)	10	23	26	26	26	23	21
	R.H.	26	58	47	41	44	54	76
	MOIST	2	10	10	9	9	10	12
14	T(C)	9	23	25	26	26	23	21
	R.H	30	58	48	42	44	54	76
	MOIST	2	10	10	9	9	10	12
15	T(C)	11	23	26	26	26	23	21
	R.H	22	58	46	41	44	55	75
	MOIST	2	10	9	9	9	10	12
16	T(C)	11	24	26	26	27	24	21
	R.H.	18	56	45	40	43	54	75
	MOIST	1	10	9	8	9	10	12
17	T(C)	10	24	26	26	27	24	21
	R.H.	22	57	45	40	43	54	75
	MOIST	2	10	9	9	9	10	12
18	T(C)	9	24	26	26	26	24	21
	R.H.	27	58	46	42	44	54	75
	MOIST	2	10	10	9	9	10	12
19	T(C)	8	24	26	26	26	24	21
	R.H.	34	58	47	42	44	53	76
	MOIST	2	11	10	9	9	10	12
20	T(C)	7	24	26	26	26	24	21
	R.H.	38	58	48	43	45	53	75
	MOIST	2	10	10	9	9	10	12
21	T(C)	6	24	25	25	26	24	21
	R.H.	41	58	47	43	45	53	75
	MOIST	2	11	9	9	9	10	12
22	T(C)	6	24	25	25	26	24	21
	R.H.	45	58	49	44	45	52	76
	MOIST	3	11	10	9	9	10	12
23	T(C)	5	23	25	25	26	24	21
	R.H.	47	58	48	43	45	52	75
	MOIST	3	10	10	9	9	10	12

AVERAGED HOURLY READINGS
1=OUTSIDE
2=SPIN DOFF (Tc401)
3=DRAWTWIST C BANK LAG AREA
4=DRAWTWIST C4-C5
5=DRAWTWIST C7-C8
6=DRAWTWIST C10-C11
7=UNDERFLOOR SUPPLY DUCT
[MOIST=Moisture Content g/kg]

FIGURE 8.1 EXAMPLE OF PRINTED OUTPUT FROM DATA LOGGER

4 / 4 /1983

POSITION NO.	MAXIMUM READING	MINIMUM READING
TEMPERATURE	DegC	DegC
1	20.82	19.66
2	28.65	27.75
3	21 25	20.28
4	17.83	16.24
5	17.29	15.57
6	27 36	25.74
7	30.46	29.56
8	10.92	2.69
9	24.6	23.83
10	23.8	23.11
11	22.65	21.56
12	36.3	29.89
13	22 1	20.43
14	24.06	22.41
15	26.25	24.29
16	26.31	24.94
17	26.59	25.38
18	24.47	22.88
19	20.58	19.68
20	24.17	23.8
21	29.43	27.9
22	25.61	22.41
23	26.41	24.81
24	31.34	29.32
25	35 76	32.11
26	39.3	37.15
27	36.46	33.87
28	23.11	19.45
29	26.25	24.96
HUMIDITY	%rh	%rh
1	76.37	73.79
2	45.35	43.89
3	77.15	74.47
4	42.25	32.11
5	40.15	29.8
6	29 33	24.43
7	29.57	28.08
8	67.75	17.03
9	50.22	47.39
10	59.32	56.41
11	69.21	66.45
12	38.35	31.73
13	36.74	27.69
14	36.32	28.09
15	51.27	43.97
16	44.04	39.64
17	46.69	42.23
18	56.94	49.51
FLOW VELOCITY	m/s	m/s
1	11.81	11.45
2	8.19	7.94
3	2.12	1.53
4	5.42	4.99
5	8.55	8.05
6	7.77	7.37
7	12.4	11.98

FIGURE 8.2 EXAMPLE OF PRINTED OUTPUT FROM DATA LOGGER

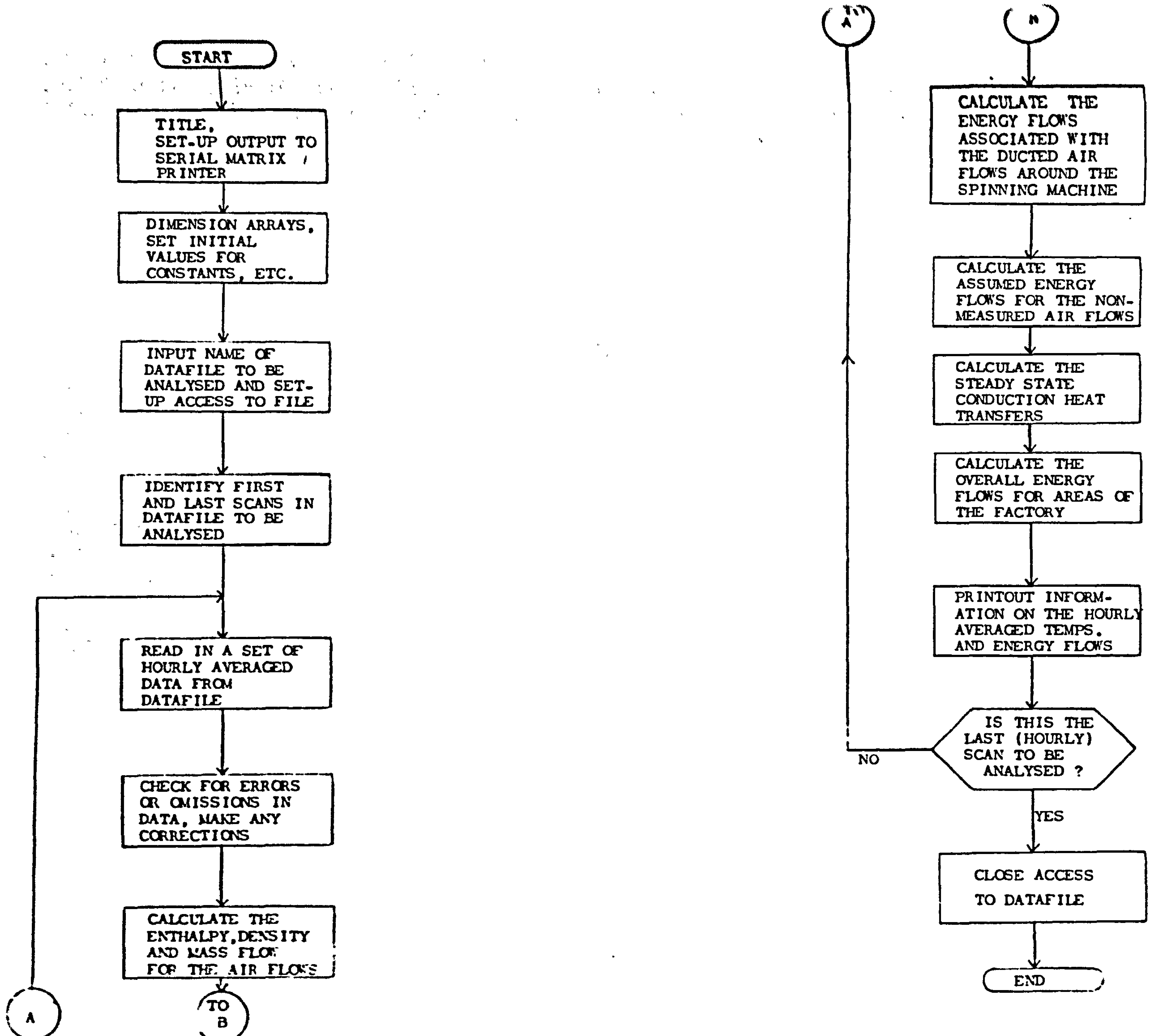


FIGURE 8.3 FLOW CHART: COMPUTER PROGRAM "ANALIS"

```
209             27:01:07:55:00
TEMPS:OUTSIDE=15.6  SD=25.1  EXTR=31.5  EXT CAT=35.4   HOPP=28.8  TOT FABRIC LOSS=287.22
DUCT ENGY FLOWS 56.18  119.34 38.27  35.8   15.98  140.02 77.86  F17 64.26  F8-164.21 F9 66.77
TOT.BUILDING E-S=113.13 ,TOT.S(added energy)=159.83 ,SD E-S=22.38  EXT E-S=174.59

210             27:01:08:55:00
TEMPS:OUTSIDE=15.8  SD=25.1  EXTR=32.3  EXT CAT=37     HOPP=29    TOT FABRIC LOSS=273.2
DUCT ENGY FLOWS 58.05  121.9  36.98  34.84  15.4   138.16 79.09  F17 64.81  F8-183.64 F9 70.5
TOT.BUILDING E-S=114.79 ,TOT.S(added energy)=159.55 ,SD E-S=21.18  EXT E-S=192.47

211             27:01:09:55:00
TEMPS:OUTSIDE=15.9  SD=25.1  EXTR=33.8  EXT CAT=37.4   HOPP=28.5  TOT FABRIC LOSS=269.55
DUCT ENGY FLOWS 60.94  125.74 38.7   30.75  12.72  136.02 83.09  F17 68.8   F8-193.31 F9 73.02
TOT.BUILDING E-S=118.65 ,TOT.S(added energy)=157.4  ,SD E-S=21.88  EXT E-S=202.77

212             27:01:10:55:00
TEMPS:OUTSIDE=15.9  SD=25.1  EXTR=33    EXT CAT=36.3   HOPP=28.5  TOT FABRIC LOSS=269.07
DUCT ENGY FLOWS 65.65  131.51 42.1   33.8   14.94  144.78 89.86  F17 73.46  F8-182.41 F9 73.97
TOT.BUILDING E-S=119.8  ,TOT.S(added energy)=172.89 ,SD E-S=23.25  EXT E-S=188.59

213             27:01:11:55:00
TEMPS:OUTSIDE=16.1  SD=25.2  EXTR=33.2  EXT CAT=36.4   HOPP=28.3  TOT FABRIC LOSS=264.65
DUCT ENGY FLOWS 62.92  127.69 40.93  31.94  13.75  138.53 86.49  F17 69.89  F8-179.15 F9 70.94
TOT.BUILDING E-S=116.68 ,TOT.S(added energy)=166.14 ,SD E-S=22.79  EXT E-S=185.5

214             27:01:12:55:00
TEMPS:OUTSIDE=16.2  SD=24.8  EXTR=33.1  EXT CAT=35.9   HOPP=29.1  TOT FABRIC LOSS=268.83
DUCT ENGY FLOWS 64.37  128.34 42.09  35.41  15.42  153.05 89.06  F17 70.02  F8-166.62 F9 74.24
TOT.BUILDING E-S=124.1  ,TOT.S(added energy)=176.33 ,SD E-S=17.66  EXT E-S=179.78

215             27:01:13:55:00
TEMPS:OUTSIDE=15.2  SD=24.6  EXTR=32.9  EXT CAT=35.2   HOPP=29.4  TOT FABRIC LOSS=280.11
DUCT ENGY FLOWS 65.73  128.34 41.42  39.1   17.84  155.92 88.02  F17 69.95  F8-164.46 F9 74.04
TOT.BUILDING E-S=120.17 ,TOT.S(added energy)=182.16 ,SD E-S=17.1   EXT E-S=175.42

216             27:01:14:55:00
TEMPS:OUTSIDE=13.9  SD=24.8  EXTR=32    EXT CAT=34.5   HOPP=29    TOT FABRIC LOSS=294.92
DUCT ENGY FLOWS 72.77  139.99 47.28  43.87  20.44  158.56 96.09  F17 80.34  F8-179.69 F9 79.13
TOT.BUILDING E-S=114.19 ,TOT.S(added energy)=200.11 ,SD E-S=21.15  EXT E-S=177.85
```

FIGURE 8.4 EXAMPLE PRINTOUT FROM DATA ANALYSIS PROGRAM
(SEE NEXT FIGURE FOR EXPLANATION)

EXAMPLE:

EXPLANATION OF DATA GIVEN IN FIGURE 8.4

SCAN NUMBER : 211

TIME AND DATE AT END OF HOUR OF SCAN : 27th.JAN. 9.55 a.m.

TEMPERATURES (Degrees Celcius)

OUTSIDE	15.9
SPIN DOFF	25.1
EXTRUSION	33.8
EXTRUSION CATWALK	37.4
HOPPER FLOOR	28.5

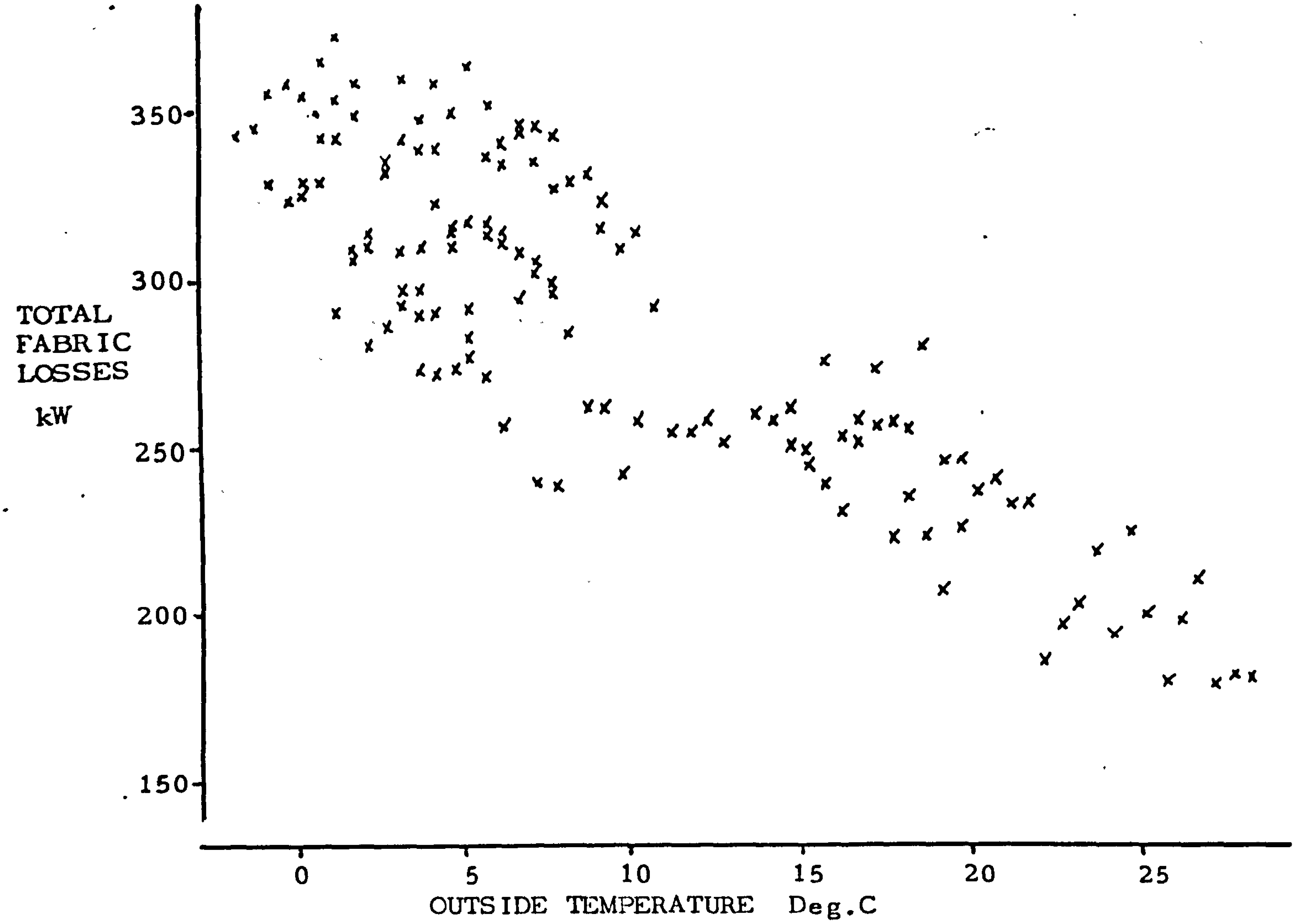
ESIMATED TOTAL CONDUCTION HEAT LOSS : 269.55 kW

DUCT ENERGY FLOWS
(N.B. DUCT FLOWS REFER TO ONE MACHINE PAIR)

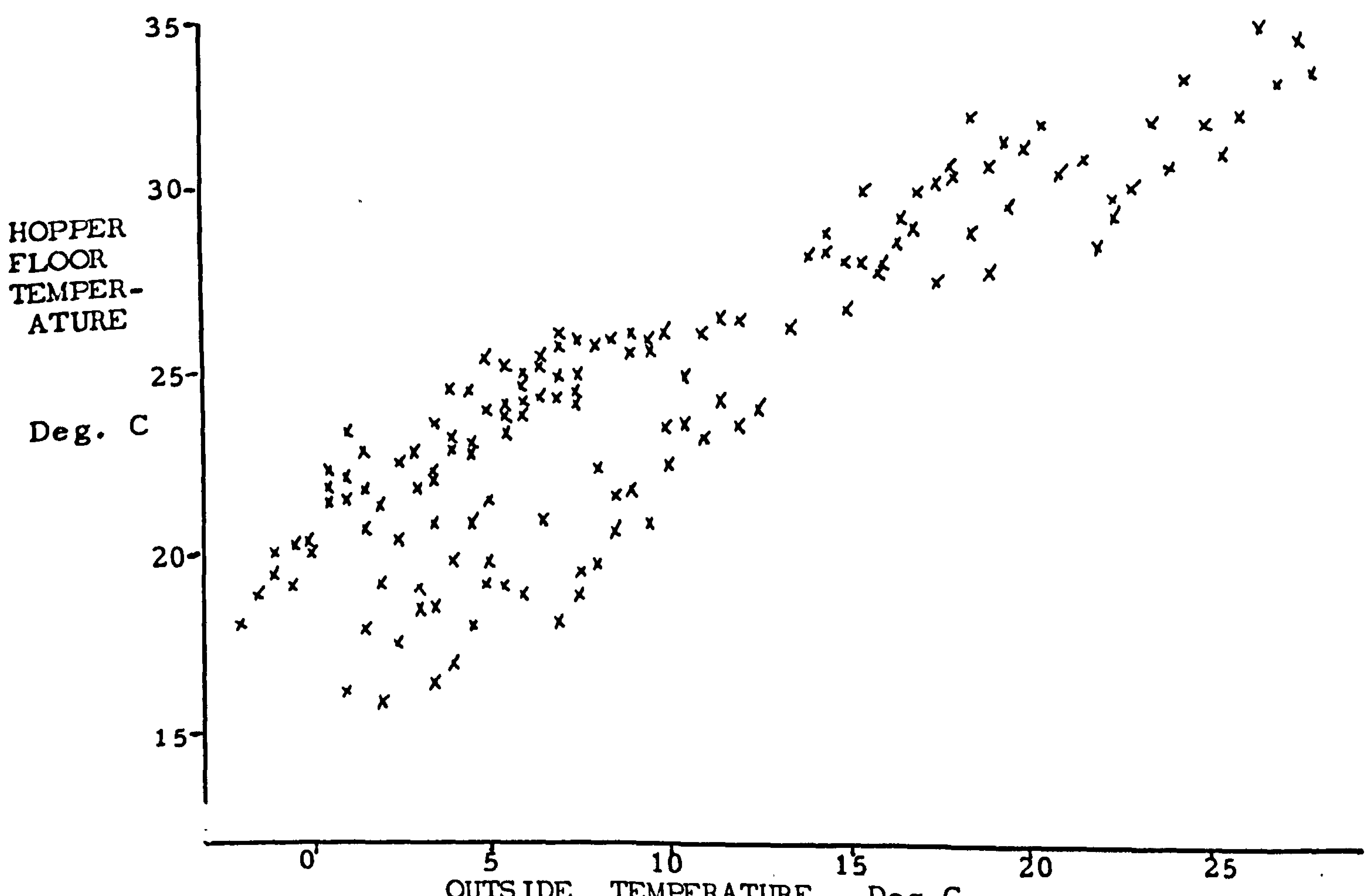
SPIN DOFF SUPPLY	60.94 kW
SPIN DOFF EXTRACT	125.74 kW
SPIN DOFF UNDERFLOOR SUP.	38.7 kW
EXTRUSION SUPPLY (LARGE)	30.75 kW
EXTRUSION SUPPLY (SMALL)	12.72 kW
EXTRUSION EXTRACT	136.02 kW
BLOWER AIR SUPPLY	83.09 kW

F17 (ENERGY IN AIR TAKEN FROM SPIN DOFF BY BLOWER AIR)	68.8 kW
F8 (ASSUMED ENERGY IN EXTRUSION TO HOPPER AIR FLOW)	193.31 kW
F9 (ASSUMED ENERGY IN DRAWTWIST TO SPIN DOFF AIR FLOW)	73.02 kW
TOT. BUILD E-S (DIFFERENCE BETWEEN ENERGY EXTRACTED BY ALL DUCTS AND ENERGY SUPPLIED BY ALL DUCTS)	118.65 kW
TOT. S (ENERGY ADDED TO OUTSIDE AIR TO ACHIEVE CONDITIONS IN AIR SUPPLY DUCTS)	157.4 kW
SD (E-S) (DIFFERENCE BETWEEN ENERGY FLOWS OF EXTRACTS AND SUPPLIES FOR SPIN DOFF AREA)	21.88 kW
EXT (E-S) (DIFFERENCE BETWEEN ENERGY FLOWS OF EXTRACTS AND SUPPLIES FOR EXTRUSION AREA)	202.77 kW

FIGURE 8.5 EXPLANATION AND KEY FOR FIGURE 8.4
TAKING SCAN 211 AS AN EXAMPLE



(a) RELATIONSHIP BETWEEN FABRIC HEAT LOSSES AND OUTSIDE TEMPERATURE



(b) RELATIONSHIP BETWEEN HOPPER FLOOR AND OUTSIDE TEMPERATURES

FIGURE 8.6

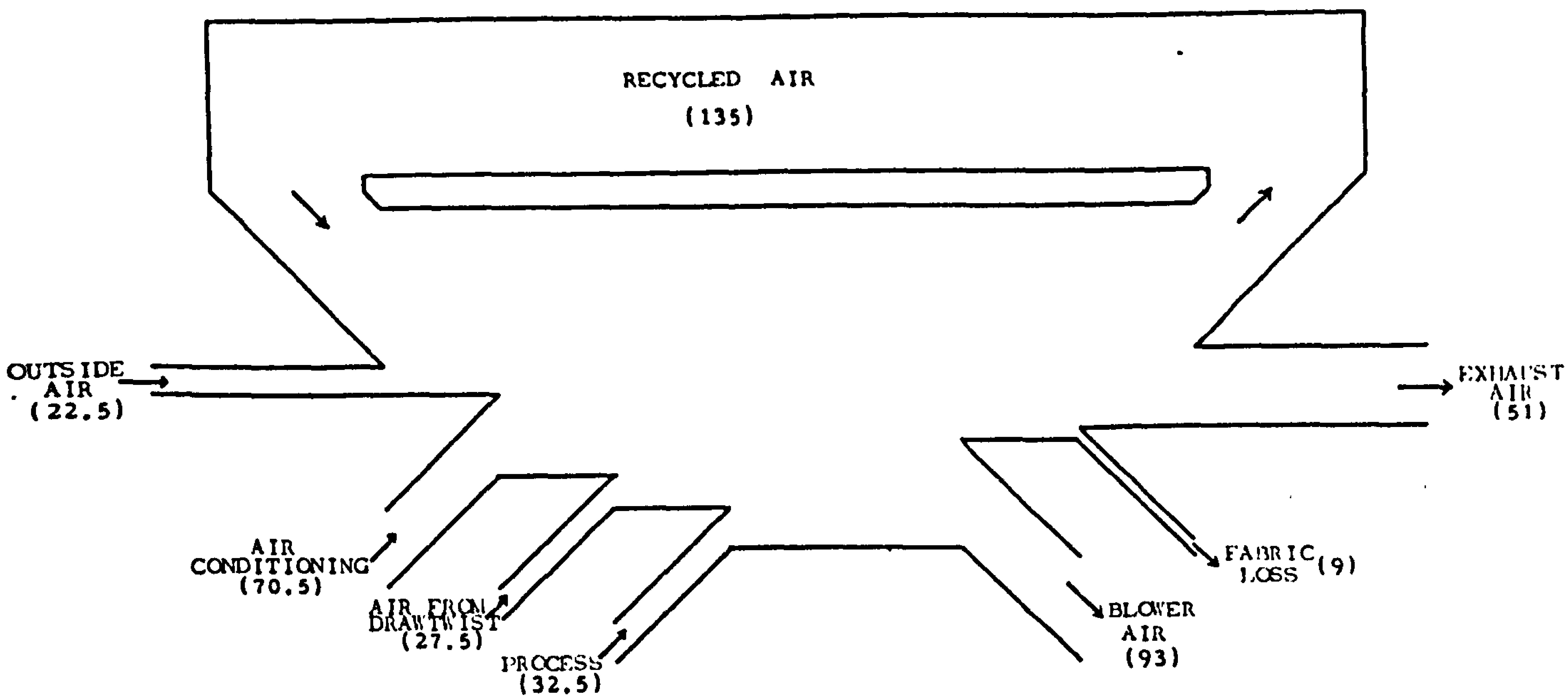


FIGURE 8.7 ENERGY FLOWS - SPIN DOFF AREA : WINTER
(ALL FIGURES IN kW)

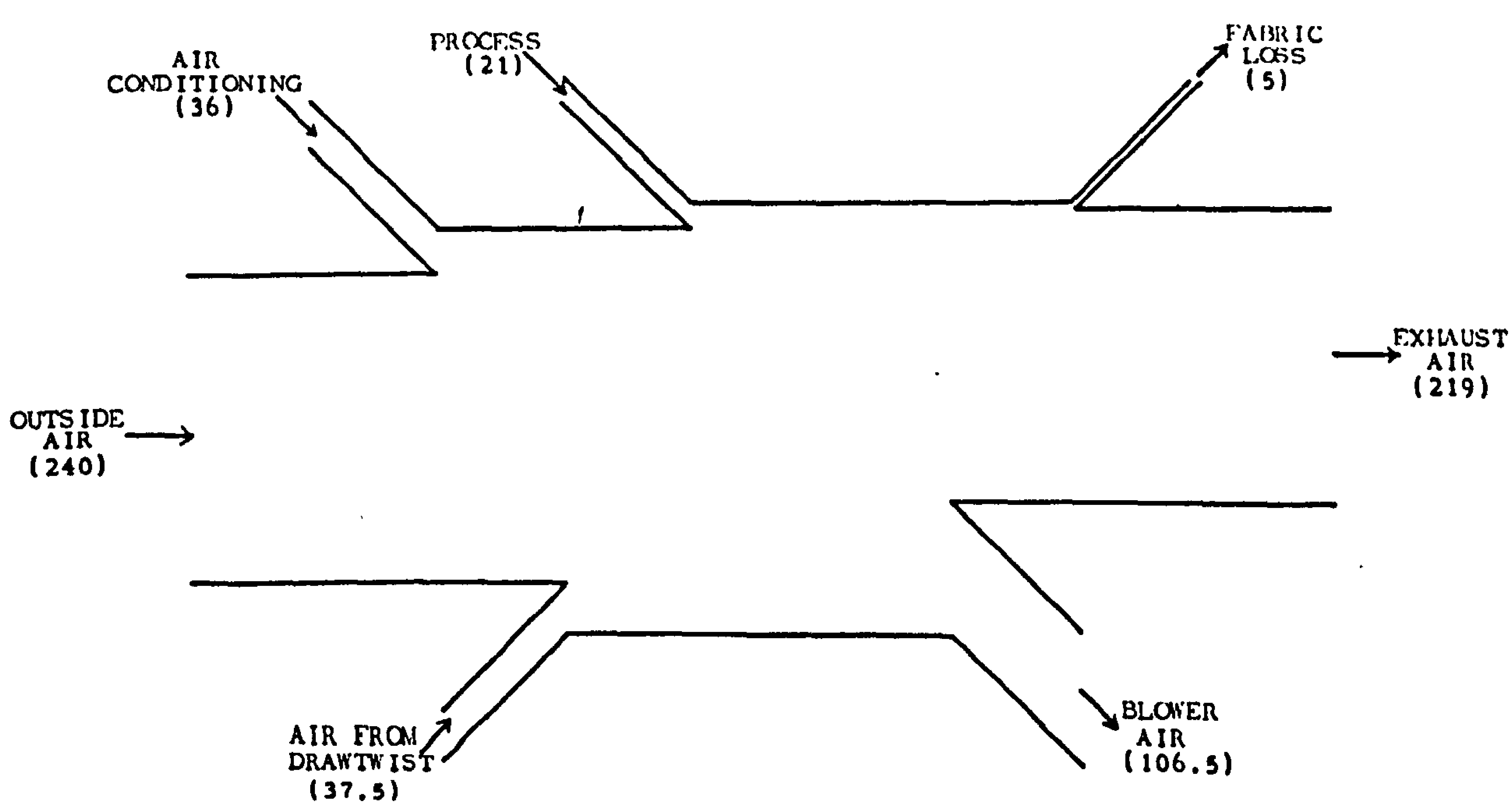


FIGURE 8.8 ENERGY FLOWS - SPIN DOFF AREA : SUMMER
(ALL FIGURES IN kW)

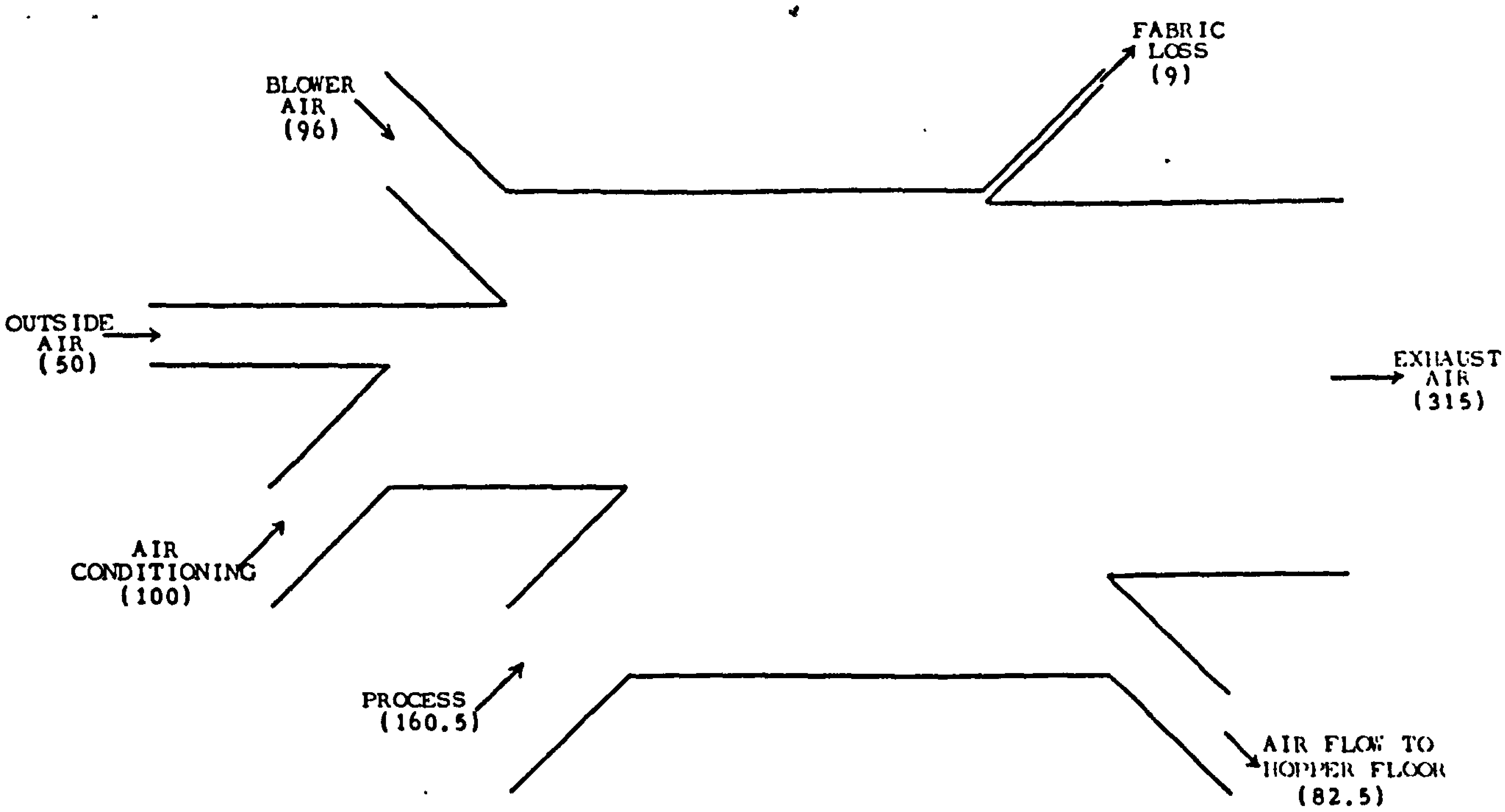


FIGURE 8.9 ENERGY FLOWS - EXTRUSION AREA : WINTER
(ALL FIGURES IN kW)

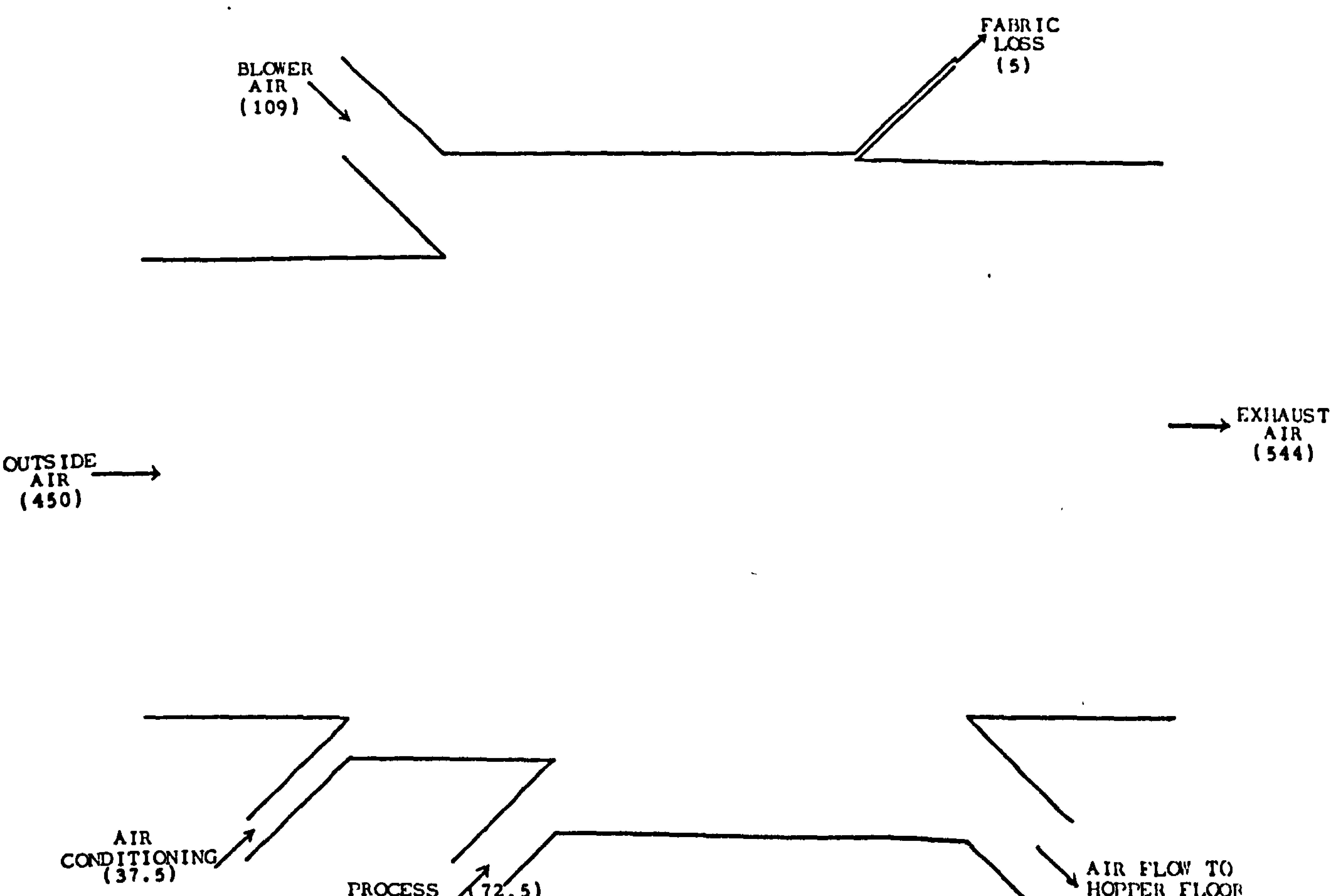


FIGURE 8.10 ENERGY FLOWS - EXTRUSION AREA : SUMMER (ALL FIGURES IN kW)

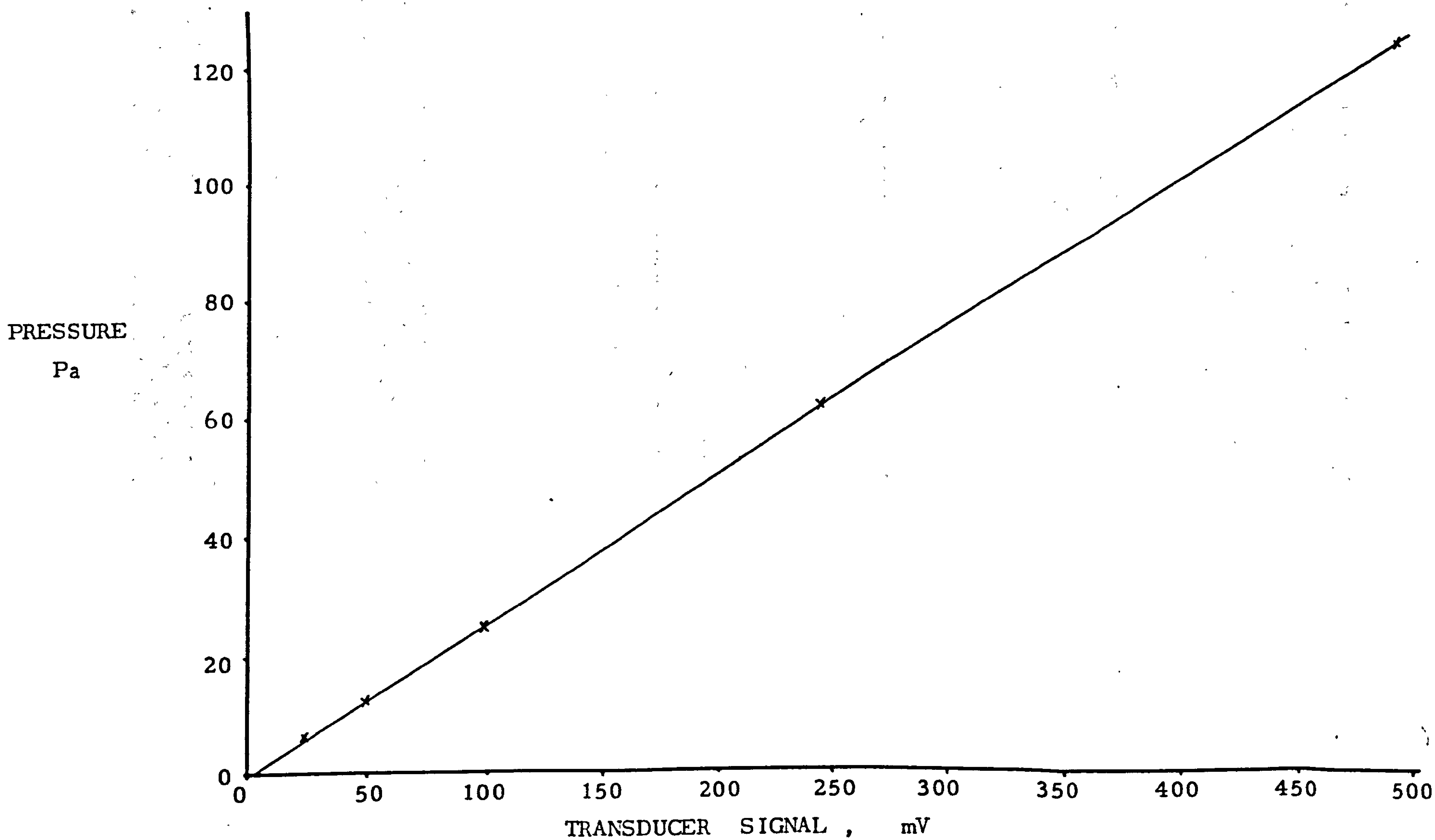


FIGURE 8.11 CALIBRATION OF PRESSURE TRANSDUCER

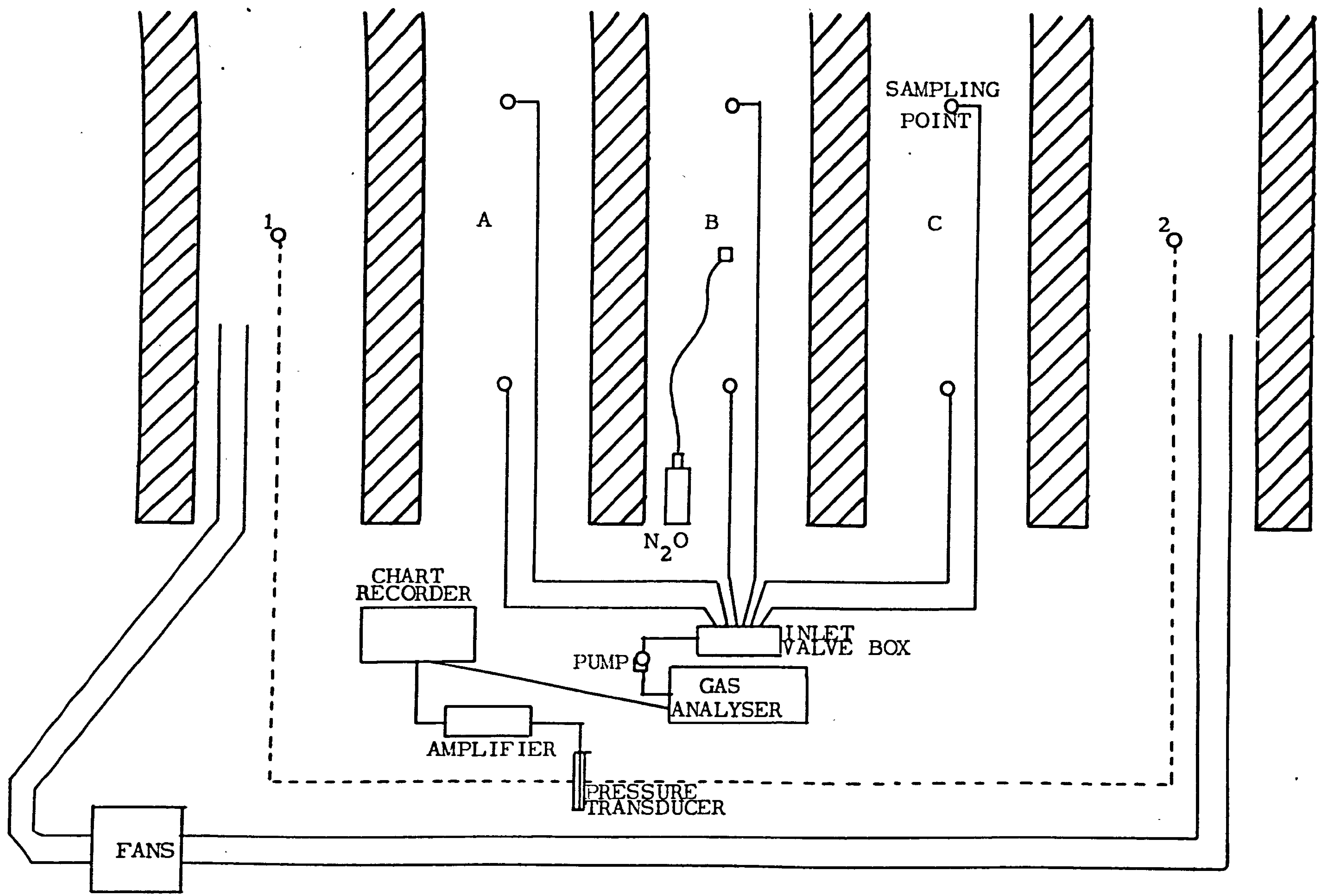


FIGURE 8.12 SKETCH PLAN OF LAYOUT FOR TRACER GAS INVESTIGATION

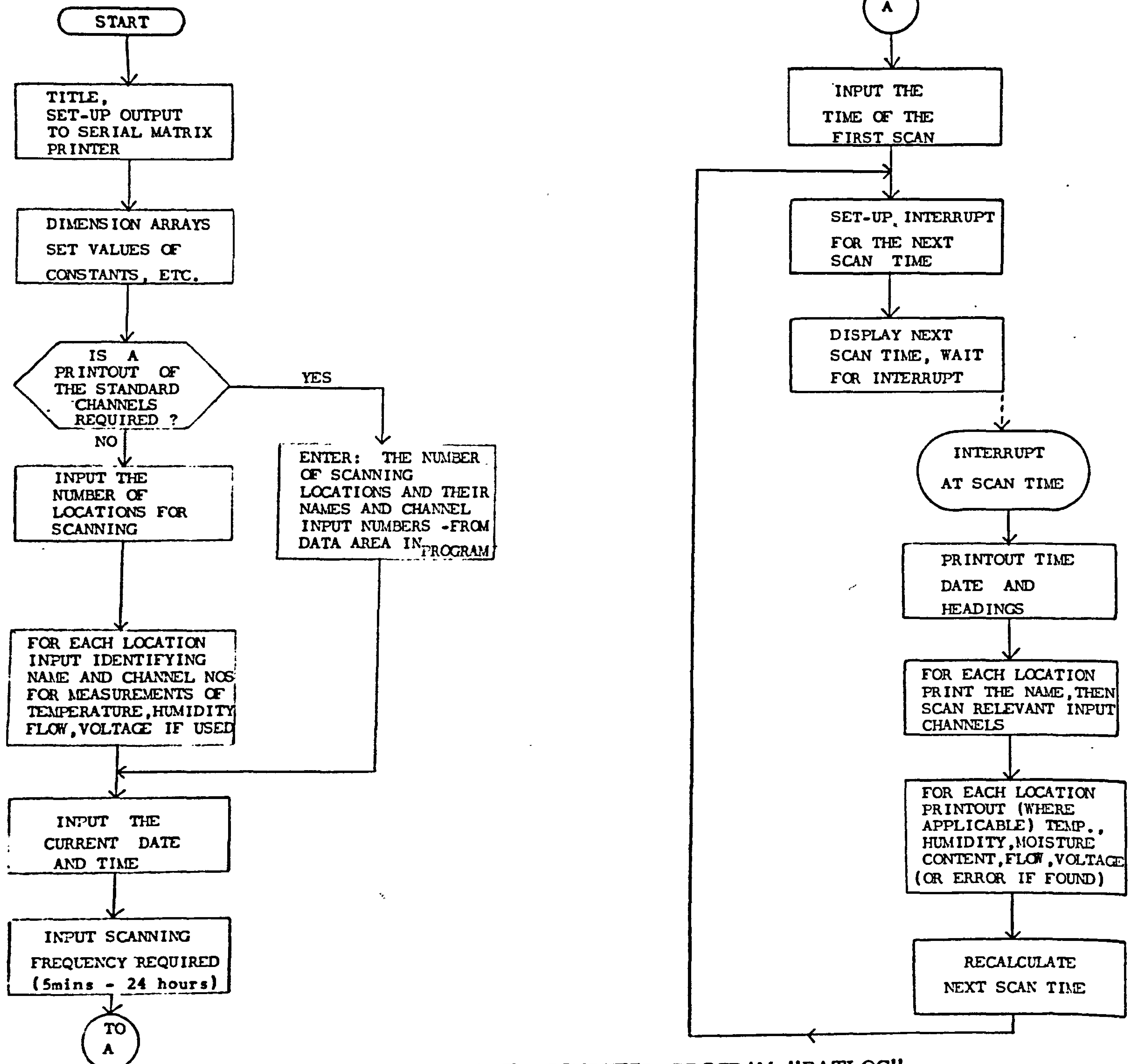


FIGURE 8.13 FLOW CHART: COMPUTER PROGRAM "DATLOG"

SCAN TIME: 05:11:12:00:00 1983

	LOCATION	TEMP DegC	HUM %rh	MOIST g/kg	FLOW m/s	VOLTAGE
1	OUTSIDE (EAST END)	14	62.7	6.2		
2	SPIN DOFF (P4)	27.7				
3	SPIN DOFF (Tb5)	24.8	ERROR	ERROR		
4	SPIN DOFF (Vb4)	20.8				
5	EXTRUSION (P4)	32.9				
6	EXTRUSION (Tc301)	26.8	47.4	10.5		
7	EXTRUSION (Vb301)	23.3				
8	EXTRUSION CAT(P4)	43				
9	EXTRUSION CAT(Tc301)	31.1	34.9	9.9		
10	EXTRUSION CAT(Vb301)	29.4				
11	HOPPER (P4)	22.6				
12	HOPPER (Tb301)	22	48.3	8		
13	HOPPER (Vb301)	24.2				
14	DRAWTWIST C BANK LAG	24.9	52.7	10.4		
15	DRAWTWIST C4-C5	26.7	43.6	9.6		
16	DRAWTWIST C7-C8	25	52.2	10.4		
17	DRAWTWIST C10-C11	27.4	44.6	10.2		
18	DRAWTWIST 1	24.4	50.5	9.7		
19	DRAWTWIST 2	25.4	46.4	9.4		
20	DRAWTWIST 3	25.2	39.8	8		
21	S PLANT SUPPLY	21.1	72.3	11.3	16.1	
22	S PLANT EXTRACT	27.1	47.6	10.7	11.7	
23	S PLANT UNDERFLOOR S	21.5	70.1	11.3	2.8	
24	EXTRUSION LARGE SUPP	18	61.3	7.9	8.4	
25	EXTRUSION SMALL SUPP	17.3	57.5	7.1	.1	
26	EXTRUSION EXTRACT	25.4	33.4	6.7	10	
27	BLOWER AIR DUCT	23.5	54.6	9.9	.1	

FIGURE 8.14 EXAMPLE OUTPUT FROM DATLOG COMPUTER PROGRAM

CHAPTER 9

ASSESSMENT OF STUDIES AND CONCLUSIONS

9.1 RECAPITULATION

Synthetic fibre manufacture is very energy intensive both in terms of process requirements and ancillary services. Since energy costs have been increasing disproportionately faster than other costs in recent years, and this trend is likely to continue, there has been a general resolve to investigate the use of energy and the means of effecting savings.

At the ICI Fibres factory being studied, major energy flows were known to be involved with the Spinning Process. "Good Housekeeping" measures at the factory had produced some reductions in energy consumption but a particular factor seen as requiring further investigation was that of air movement, and air conditioning and ventiation systems operation.

This situation produced the need to satisfy dual requirements in this study; the first to advance the knowledge and understanding of air movement in such an industrial location; the second to provide tangible results which could be used to modify and improve plant operation and produce reduced energy use and costs.

The salient feature of the production areas of the factory was the high level of internal partitioning - produced as a result of the machinery layout. The ground floor, known as the Spin Doff area was provided with conditioned air to meet the requirements of yarn. On the first floor, known as the Extrusion area, a great deal of heat was liberated from the nylon melting and extrusion process. This resulted in a need for substantial ventilation to provide cool fresh air and extract warm air, to prevent overheating.

The structure and fabric of the building was examined, as were construction drawings, to enable prediction of the building's thermal properties and conduction heat transfer.

The basic design guides and most texts, lack information relating to internal partitioning, except in fairly general terms, and industrial buildings are usually assumed to be large open structures with high ceilings. Thus the information gathered in this study should provide an extension of knowledge in such areas. The initial tests carried out at the factory (a pressurization test and smoke tracing of air movement together with basic environmental measurements) helped identify aspects worthy of further investigation.

The problem to be tackled in this study was seen to have two main facets. First, for air movement purposes how would a series of similar, regularly spaced partitions react to a pressure difference across them and what were the implications for air flow. (It was known that wind pressures and features of the mechanical air movement systems often cause a differential pressure across the production areas perpendicular to the partitions.) Such information was required for the second facet, that of improving the site personnel's knowledge of the plant operation and so allow better decision making and better functioning.

In Chapter 5, air flow equations were considered and three possible options were proposed for the interaction of a series of resistances to air flow. The first of these assumed each partition to behave independently with the total resistance being the sum of the individual resistances. The second took the resistance of the first partition at full value, but subsequent ones each at the same but reduced resistance. The third hypothesized that after the first partition, the resistance of subsequent partitions would be reduced, each being a constant fraction of that previously. Thus in this third case the total resistance would have a logarithmic form.

9.2 SUMMARY OF EXPERIMENTAL WORK

Physical limitations of size meant that investigation of resistances to air flow could not be performed at full scale. A reduced scale model test offered the additional benefits of allowing a variety of partition spacings and partition types to be considered. Because of scale effects the model experiments were only able to operate with plain types of partition. Though this meant the results could not be identified exactly with the full scale factory environment, the range of tests performed did permit some basic guidelines to be determined.

At none of the spacings or partition numbers used was the total resistance found to be the product of the number of partitions multiplied by the resistance of a single partition. (This being an extension of the simple theory given in most texts.) This result could only be expected if each partition were independent and unaffected by others in the series. Based on the work carried out, each partition would need to be separated by a distance in excess of ten times the partition height.

Considering first, the partition with evenly distributed circular holes, the resistance was found to depend upon the partition spacing and the number of such partitions. At each spacing used, the total resistance produced was of the form

$$R = R1 + (n - 1)R2 \tag{9.1}$$

where R = Total Resistance

R1 = Resistance of First/Single Partition

R2 = Resistance of Subsequent Partitions (depends on spacing)

n = Number of Partitions

This relationship represents the second hypothesis option considered earlier for the determination of overall resistance. The result is illustrated in Figure 9.1.

Results from the plain wall type partition were not so clear. In this case it seemed that the air flow pattern (eddy lengths etc.) and partition spread distance were of greater importance than the number of partitions. The minimum resistance to flow occurred when the partitions were spread over a distance approximately three times the partition height. Above and below this spread greater interference to the flow pattern was found. As the wall partitions were moved further and further apart they began to display increased independence such that at large separations the flow over each would be independent and the resistances additive, generalised results are shown in Figure 9.2.

Since in building design one is trying to maximize or minimize resistance to air flow, it would be useful to use partition heights and spacings to produce, or avoid, the three times the height spread distance.

In order to collect information on environmental conditions; temperatures and humidities, and to determine duct flow rates a computer based data logging system was designed and installed around a machine pair at the factory. The various sensors and transducers were calibrated and checked, and a computer program was devised and implemented to operate the system and record the required values. The measurements made allowed the specific enthalpy (which is a measure of heat content) of the air to be determined. Further computer programs were devised to analyse the recorded data. The results of this study showed that the conduction heat transfers, from and within the building, were small by comparison with the energy flows associated with air movement.

Though the recorded data was that of readings averaged over an hour, fairly short term (e.g. day to day) variations in factors such as plant operation, made comparison difficult. Thus energy flows were averaged for longer periods the results being given for typical winter and summer periods in Chapter 8 (Figures 8.7 to 8.10) These show the relative energy flows but do not give an indication of the effectiveness of the systems. One suitable way of assessing heat removal by ventilation is by use of an efficiency developed from the work of Sandberg,[(1)(2)] on ventilation efficiency. This is described in section 9.4.

In order to look at air movement and transfers more closely, experiments using nitrous oxide tracer gas were designed and carried out. These indicated that within the partitioned environment investigated, air transfers between adjacent machine alleyways were quite low even with an exaggerated pressure difference across the partitioned layout. The main flow in the situations examined, was the flow out of the ends of the alleyways, with each alleyway relatively isolated from its neighbours.

9.3 COMPARISON AND EVALUATION

It was understood when the experiments at model and full scales were defined, that they would not be directly compatible. This was as a result of the need to use plain shapes in the model scale tests as compared to the fairly complex design of the machine-formed partitions at full scale. Therefore the opportunity has been taken at model scale to investigate a wider variety of partition layouts than that in existence at the factory.

The results of the experiments performed show that the simple additive theory of resistance for partitions in series underestimates the flow rate for a given pressure difference. Additionally for the case of simple wall like partitions, the flow regime set up exaggerates the underestimation. The extent of these underestimations cannot be quantified directly for all cases, from the experiments performed and further work is recommended for such information.

If such work were to be carried out a different approach and additional equipment to that used in this study would be suggested. A test chamber would be required which might be of about half full scale. The main influence and requirement on size is that of a suitable space being available in which to perform the tests, since it must allow access to the chamber to permit modifications and layout variations, as well as for monitoring purposes. It is particularly important to allow for pressure measurements and a series of accurate low pressure devices would be required operating in the ranges 0-1, 0-5, 0-10 Pascals. The air inlets and outlets to and from the test chamber should be of better design taking account of flow distribution in the chamber and flow measurement requirements.

A means of measuring local flow velocities in the chamber would be desirable as well as measurement of bulk air flow through the model. Lower flow rates, but with greater pressure drops resulting from partition and other effects should be considered and so the fans to produce such a pressure drop would have to be chosen carefully. The facilities to view air flows using smoke or particle tracers should be incorporated in further work as this was found to be very useful in the experiments already described.

In the current study, though allowances might be made due to factors inherent in, and affected by the different scales, there was a significant factor evident in the full scale test results. Even with an induced pressure differential across the machine partition there was little air transfer, and the effect of reduced resistance to flow could not be investigated. This factor was exhibited as the marked isolation of the machine alleyways at the Spin Doff level which was apparently due to the disturbance of cross flows caused by the mechanical ventilation systems. Thus whilst a naturally ventilated building containing partitioning might show the reduced resistance to flow previously mentioned; the case of mechanically ventilated and air-conditioned buildings must be considered separately.

9.4 VENTILATION

As a means for evaluating system performance, the concepts of ventilation efficiency and temperature efficiency are very useful. By comparing values of a measurable and relevant property of the air in fresh (supply) air streams; exhaust air streams; and within the building space, a guide to the effectiveness of the ventilation of air-conditioning can be obtained. In order to take account of latent heat gains and changes in moisture content of air, the idea of a total thermal efficiency has been presented in this thesis. This development was necessary so that the variations encountered within the factory environment could be assessed. Total thermal efficiency is given by a ratio of specific enthalpies thus:

$$E_{tt} = \frac{H_e - H_s}{H_i - H_s} \times 100\% \qquad (9.2)$$

Where H = Specific Enthalpy

Subscript e = extract air

s = supply air

i = building space air

This efficiency was of particular use for examining the heat removal in the Extrusion area. An acceptable value for the "efficiency" might be 100% with the desired result being in excess of this value. However, the Extrusion area systems did not appear to perform favourably when temperature efficiency was considered, and they exhibited even worse results for Total Thermal Efficiency. The figures are illustrated and compared in Figure 9.3. The difference which exists between the two efficiencies provides support for the use of Total Thermal Efficiency in suitable environments. The methods for determining its value require fairly simple environmental measurements and its adoption for valuation of heat removing ventilation systems can be recommended.

9.5 CONCLUSIONS

The work described and embodied in this thesis was pursued in order to extend the knowledge of air movement, in particular, within the context of an industrial environment. This involved the examination of a number of air-conditioning and ventilation systems in which the main criterion for operation was not personal comfort. Previous work concerning industrial environments have dealt primarily with large open structures with high roofs or ceilings. In this study a more novel environment was under consideration; a factory with several production floor levels, typically 3-5m in height, and a high degree of internal partitioning. Such an environment is not uncommon being found in a number of fibre manufacturing and processing plants.

Within the industrial context of the work was a very practical aim; that of improving knowledge of the air movement so that better operation and efficiencies might be attained. In such a way an energy and cost saving might be made which could be of substantial amounts given the overall energy input at the factory.

Internal partitioning was investigated at some length. Design guides, text books and reports of research studies were consulted and a series of model scale tests were planned and executed. Particular emphasis was placed on the way in which a series of resistances to air flow would behave. On examination, a simple model of additive resistances was thought to be too simple for the cases under study and three options were considered. The results from the model scale tests indicated that the simple model was likely to underestimate flow, and by quite a large margin in the case of plain wall partitions.

Work carried out at full scale, however, showed that the air flows set-up by the mechanical systems interfered to a very significant degree with the air flows caused by pressure differences due to external effects. The isolation of areas between partitions proved very important and there may be a need to reconsider ideas on internal air flows and their interaction.

A tool for ventilation system evaluation, in its ability to remove heat has been developed in the concept of a Total Thermal Efficiency. Such an efficiency can be recommended for investigation of a wide variety of systems in which moisture and heat exchanges are important.

Future work can be recommended to investigate the interaction between air flows within buildings caused by external pressure differences (e.g. due to wind) and those flows caused by mechanical ventilation systems. Because of the difficulties in obtaining an ideal environment at full scale which would incorporate and allow variation in both flows, a model scale environment would prove more suitable. Knowledge of the interactions should provide a means for better flow prediction and more effective design.

One particular area of interest uncovered by the model scale work involved the interaction of partition spacing, partition height and flow regime. The relationship between these parameters may be independent of scale given certain conditions, and might be expressed by specific values of dimensionless ratios. Such techniques are used in many engineering disciplines which involve fluid mechanics and dynamics.

Previous studies which were concerned with the effect of fences, hedges and windbreaks (which are similar in nature to the partitions of this study) have yielded information relating to the values of dimensionless ratios. However in these studies, the object has been placed in models using simulation of air movements (wind) in the earths atmosphere at ground level. As such the boundary conditions are very different to those existing with internal partitions.

The limited model scale work described in this study suggested that the ratio of 3:1 in spacing and height for the wall partitions, appears to give rise to certain effects. A more detailed investigation of the influence of this ratio on resistance to air flow would be justified.

It would be desirable in such an investigation to consider the distance between the top of the partition and the "ceiling" of the model and also to consider variations in the height, width and depth geometry of the partition.

The results of such work may allow the development of a more universal relationship for air flow behaviour in partitioned environments.

Since the project described in this thesis was also to be of practical benefit in the industrial context, a series of recommendations for improved operation at ICI Fibres, Doncaster are given in Appendix C which can be acted upon if funds are available.

REFERENCES

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What is Ventilation Efficiency?
Building and Environment, Vol. 16, No. 2, 1981.

2. M. SANDBERG
Definition of Ventilation Efficiency and the Efficiency of Mechanical Ventilation Systems.
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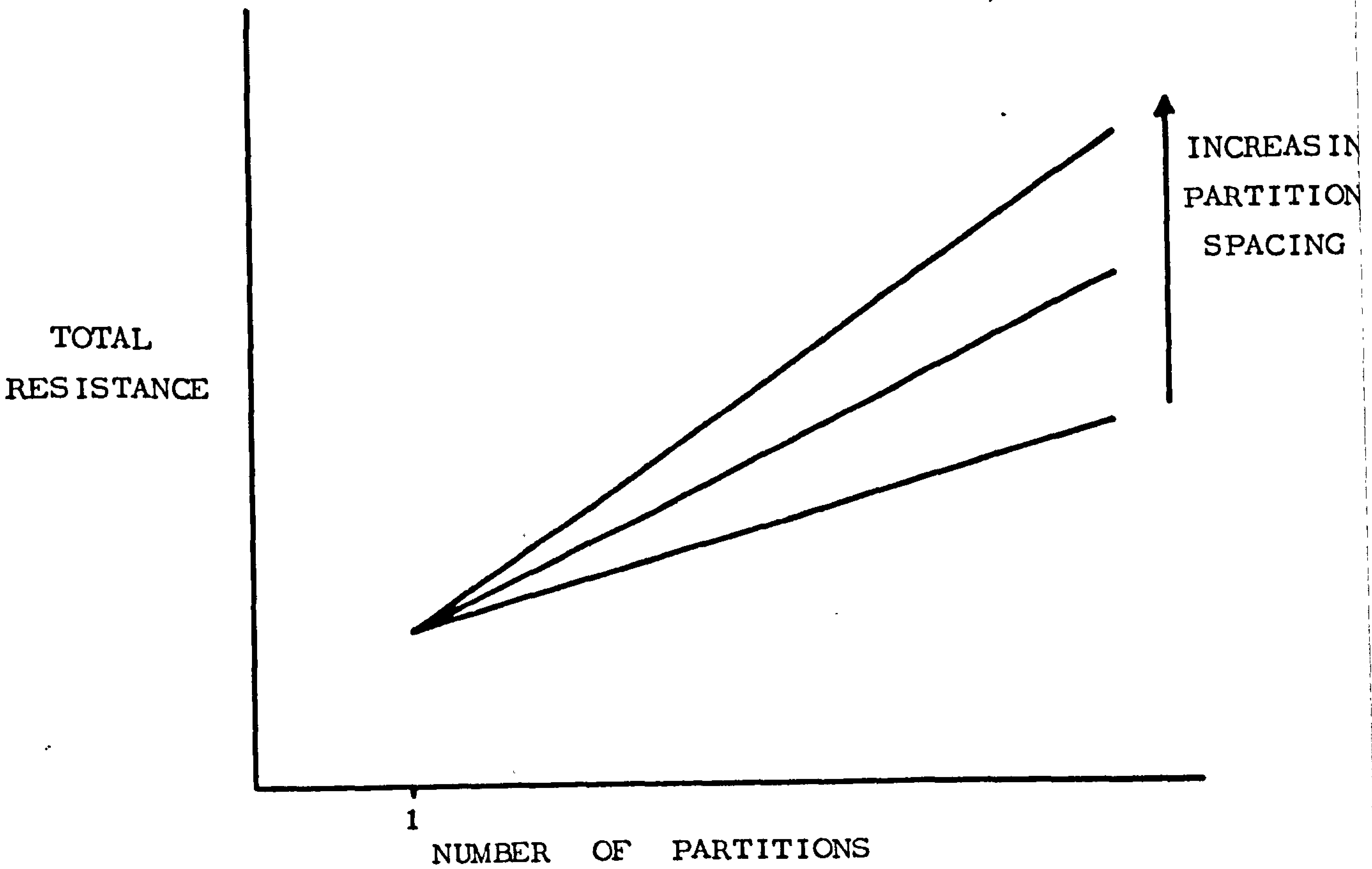

FIGURE 9.1 VARIATION IN TOTAL RESISTANCE FOR CIRCULAR HOLE PARTITIONS

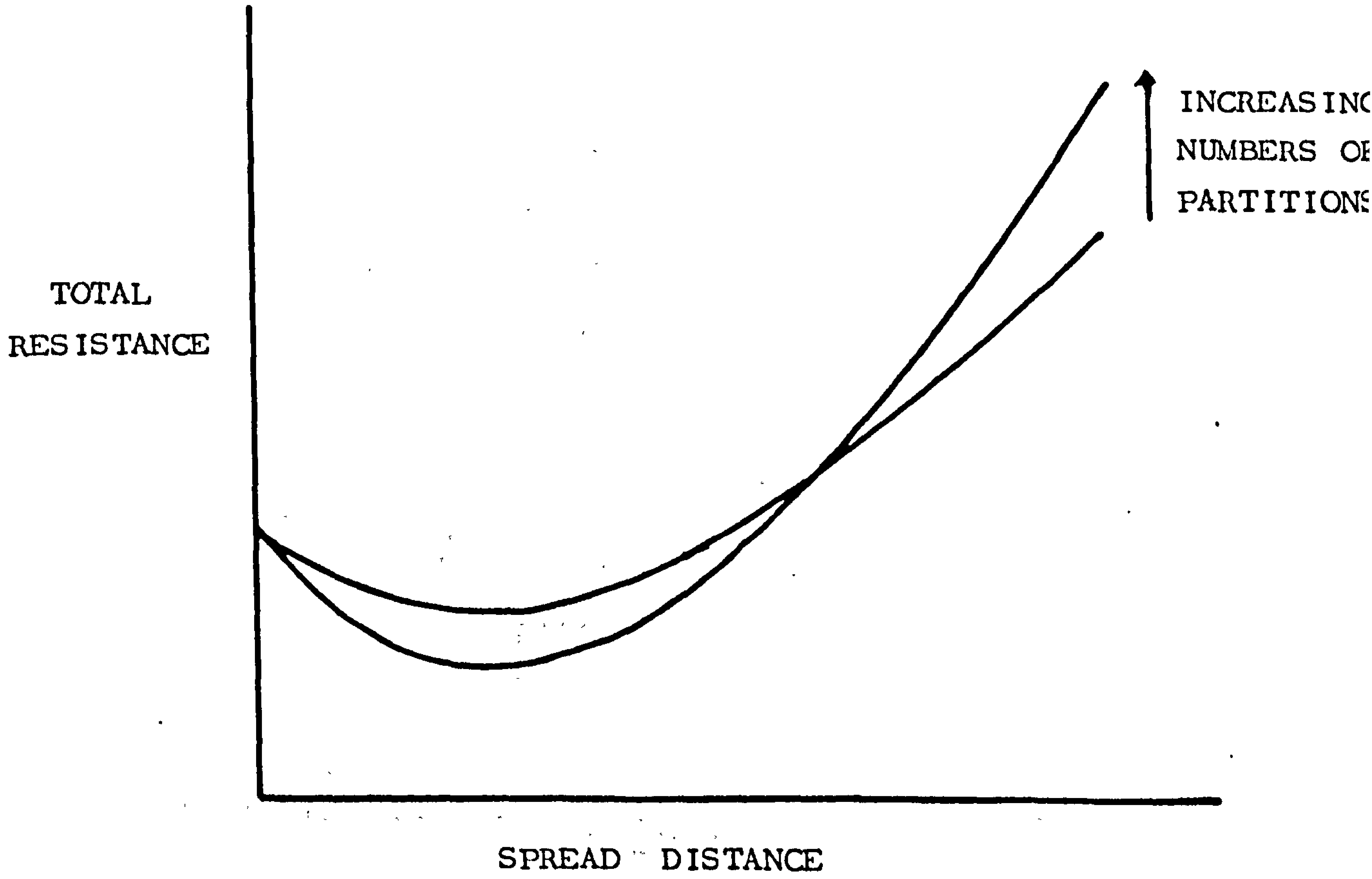

FIGURE 9.2 VARIATION IN TOTAL RESISTANCE FOR WALL PARTITIONS

APPENDIX A ENERGY USAGE

ICI FIBRES, DONCASTER

The following tables give summarised details of energy use (gas and electricity), and production month by month 1980-83.

The data is based upon the duration of the firm's accounting "month" which does not exactly coincide with the calendar month. The figures are taken from internal documents prepared by Mr D Watson of ICI Fibres.

It should be noted that the months of July and December usually contain a week during which the main plant is shut down for maintenance purposes, and that no production takes place at these times.

The figure for Degree Days is also based upon the accounting month. The number of Degree Days in a month is a measure of the difference between a base temperature (which is related to internal conditions) and the 24 hour mean outside temperature. It is included in the following tables to give an indication of general climate. Further information concerning Degree Days is to be found in the Department of Energy Fuel Efficiency Booklet No. 7.

All energy use figures are given in the common unit of Mega-Watt hours for comparison purposes.

APPENDIX A: TABLE 1

ENERGY USAGE ANALYSIS 1980

	DAYS (ACCOUNTING MONTH)	DEGREE DAYS (ACCOUNTING MONTH)	PRODUCTION (TONNES)	GAS USED (MWh)	ELECTRICITY USED (MWh)	TOTAL ENERGY USED (MWh)	% SPLIT GAS : ELEC
JAN	31	403.6	3116	15649	5528	21177	73.9 : 26.1
FEB	28	287.0	3586	13242	5147	18389	72.0 : 28.0
MAR	28	299.4	3879	13502	5365	18867	71.6 : 28.4
APR	35	243.6	4627	14793	6419	21212	69.7 : 30.3
MAY	28	138.2	2992	9156	4783	13939	65.7 : 34.3
JUN	28	78.7	2196	6039	3708	9747	62.0 : 38.0
JUL	35	94.1	1374	4377	3006	7383	59.3 : 40.7
AUG	28	37.4	1856	5236	3452	8688	60.3 : 39.7
SEPT	28	57.5	2514	5806	3823	9629	60.3 : 39.7
OCT	35	232.0	3238	9874	5235	15109	65.4 : 34.6
NOV	28	234.1	3025	9963	4354	14317	69.6 : 30.4
DEC	34	361.8	2762	11339	4282	15621	72.6 : 27.4
TOTAL	366	2467.4	35165	160003	74941	234974	68.1 : 31.9

APPENDIX A: TABLE 2

ENERGY USAGE ANALYSIS 1981

	DAYS (ACCOUNTING MONTH)	DEGREE DAYS (ACCOUNTING MONTH)	PRODUCTION (TONNES)	GAS USED (MWh)	ELECTRICITY USED (MWh)	TOTAL ENERGY USED (MWh)	% SPLIT GAS : ELEC
JAN	29	317.7	3069	11591	4677	16268	71.3 : 28.7
FEB	28	343.8	3212	11300	4722	16022	70.5 : 29.5
MAR	28	237.0	3328	10731	4552	15283	70.2 : 29.8
APR	35	280.5	3737	11717	5694	17411	67.3 : 32.7
MAY	28	139.3	2978	9051	4593	13644	66.3 : 33.7
JUN	28	55.5	2855	7548	4458	12006	62.9 : 37.1
JUL	35	64.2	2565	6417	4488	10905	58.8 : 41.2
AUG	28	43.1	2997	6634	4377	11011	60.2 : 39.8
SEP	28	52.7	3348	7035	4518	11553	60.9 : 39.1
OCT	35	223.2	4279	11547	5688	17235	67.0 : 33.0
NOV	28	212.9	3702	10349	4767	15116	68.5 : 31.5
DEC	35	529.0	3646	13490	4920	18410	73.3 : 26.7
TOTAL	365	2498.9	39716	117410	57454	174864	67.1 : 32.9

APPENDEX A: TABLE 3

ENERGY USAGE ANALYSIS 1982

	DAYS (ACCOUNTING MONTH)	DEGREE DAYS (ACCOUNTING MONTH)	PRODUCTION (TONNES)	GAS USED (MWh)	ELECTRICITY USED (MWh)	TOTAL ENERGY USED (MWh)	% SPLIT GAS : ELEC
JAN	28	390.5	3237	13077	4603	17680	74.0 : 26.0
FEB	28	304.4	3638	12240	4798	17038	71.8 : 28.2
MAR	28	264.2	3431	11094	4754	15848	70.0 : 30.0
APR	35	257.9	3687	11425	5819	17244	66.3 : 33.7
MAY	28	142.6	3013	7941	4428	12369	64.2 : 35.8
JUN	28	59.7	2918	6596	4342	10938	60.3 : 39.7
JUL	35	41.9	2623	5698	4176	9874	57.7 : 42.3
AUG	28	34.2	1928	5859	3537	9396	62.4 : 37.6
SEPT	28	62.6	2965	5926	3933	9859	60.1 : 39.9
OCT	35	191.9	4063	10290	5255	15545	66.2 : 33.8
NOV	28	195.1	3520	9229	4361	13590	67.9 : 32.1
DEC	36	427.0	3366	11716	4513	16229	72.2 : 27.8
TOTAL	365	2372.0	38389	111091	54519	165610	67.1 : 32.9

APPENDIX A: TABLE 4

ENERGY USAGE ANALYSIS 1983

	DAYS (ACCOUNTING MONTH)	DEGREE DAYS (ACCOUNTING MONTH)	PRODUCTION (TONNES)	GAS USED (MWh)	ELECTRICITY USED (MWh)	TOTAL ENERGY USED (MWh)	% SPLIT GAS : ELEC
JAN	27	242.5	3072	10829	4323	15152	71.5 : 28.5
FEB	28	392.9	3648	12431	4538	16969	73.3 : 26.7
MAR	35	322.6	4555	13199	5784	18983	69.5 : 30.5
APR	28	261.4	3506	10633	4499	15132	70.3 : 29.7
MAY	28	169.9	3839	9618	4546	14164	67.9 : 32.1
JUN	35	113.5	4689	9458	5673	15131	62.5 : 37.5
JUL	28	22.2	2519	4756	3505	8261	57.6 : 42.4
AUG	28	26.2	2498	6490	4317	10807	60.1 : 39.9
SEPT	35	91.5	4510	9501	5815	15316	62.0 : 38.0
OCT	28	136.6	3841	8749	4805	13554	64.5 : 35.5
NOV	28	237.7	3697	9724	4717	14441	67.3 : 32.7
DEC	37	368.8	3026	11593	4633	16226	71.4 : 28.6
TOTAL	365	2385.8	43400	116981	57155	174136	67.2 : 32.8

APPENDICES B1 AND B2

RESULTS OF MODEL CHAMBER TESTS USING CIRCULAR HOLE AND RECTANGULAR WALL PARTITIONS.

The results of these tests are presented on the following pages in tabular form.

Appendix B1: Model chamber tests
Circular hole partitions
Test Run numbers 21-39.

Appendix B2: Model chamber tests
Wall type partitions
Test Run numbers 40-58.

TEST RUN NO. 21
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 1
PARTITION SPACING = n/a

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	1.55	0.393
19.9	0.818	1.5	0.380
19.2	0.79	1.5	0.380
18.4	0.757	1.5	0.380
17.5	0.72	1.3	0.329
16.4	0.675	1.25	0.317
15.3	0.629	1.1	0.279
13.8	0.568	0.9	0.228
14.4	0.592	0.8	0.203
13.1	0.539	0.75	0.190
12.4	0.51	0.5	0.127
11.3	0.465	0.6	0.152
10.75	0.442	0.45	0.114
9.9	0.407	0.35	0.089
8.9	0.366	0.3	0.076

Barometric Pressure = 749 mm Hg   Temperature = 19.5°C

TEST RUN NO. 22
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 0.61 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	1.75	0.444
19.9	0.818	1.7	0.431
19.4	0.798	1.6	0.406
18.6	0.765	1.6	0.406
18.2	0.749	1.65	0.418
17.3	0.712	1.3	0.329
16.3	0.67	1.3	0.329
14.7	0.605	1.05	0.266
15.5	0.638	1.3	0.329
13.4	0.551	0.8	0.203
12.9	0.531	0.7	0.177
11.8	0.485	0.65	0.165
11.5	0.473	0.6	0.152
11	0.452	0.5	0.127
10.2	0.42	0.35	0.089
9.1	0.374	0.25	0.063
8.3	0.341	0.2	0.051

Barometric Pressure = 749 mm Hg   Temperature = 19°C

TEST RUN NO. 23
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 3
PARTITION SPACING = 0.61 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.4	0.839	2.4	0.608
19.75	0.812	2.1	0.532
18.9	0.777	1.95	0.494
18.4	0.757	1.9	0.482
17.6	0.724	1.85	0.469
16.6	0.683	1.6	0.406
15.6	0.642	1.3	0.329
14.5	0.596	1.35	0.342
14.3	0.588	1.05	0.266
13.8	0.568	0.8	0.203
12.9	0.531	1.0	0.253
12.1	0.498	0.8	0.203
11.2	0.461	0.55	0.139
10.1	0.415	0.55	0.139
9.1	0.374	0.3	0.076
8.3	0.341	0.5	0.127

Barometric Pressure = 748 mm Hg   Temperature = 19°C

TEST RUN NO. 24
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 4
PARTITION SPACING = 0.61 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	3.0	0.760
19.8	0.814	2.7	0.684
19.3	0.794	2.7	0.684
18.6	0.765	2.5	0.634
17.6	0.724	2.25	0.570
16.9	0.695	2.1	0.532
16.2	0.666	1.9	0.482
15.2	0.625	1.5	0.380
14.7	0.605	1.6	0.406
13.3	0.547	1.25	0.317
13.7	0.563	1.35	0.342
12.5	0.514	1.05	0.266
11.4	0.469	0.9	0.228
10.7	0.44	0.7	0.177
9.4	0.387	0.8	0.203
8.5	0.35	0.4	0.101

Barometric Pressure = 748 mm Hg   Temperature = 19°C

TEST RUN NO. 25
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 5
PARTITION SPACING = 0.61 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	3.85	0.976
19.9	0.818	3.5	0.887
19.4	0.798	3.15	0.798
18.8	0.773	3.25	0.824
18.0	0.74	2.9	0.735
17.1	0.703	2.75	0.697
16.3	0.67	2.5	0.634
15.2	0.625	2.1	0.532
14.8	0.609	2.1	0.532
13.6	0.559	1.6	0.406
12.5	0.514	1.4	0.355
11.6	0.477	1.3	0.329
10.7	0.44	0.9	0.228
9.9	0.407	0.65	0.165
9.2	0.378	0.55	0.139
8.3	0.341	0.55	0.139

Barometric Pressure = 748 mm Hg Temperature = 19°C

TEST RUN NO. 26
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 5
PARTITION SPACING = 0.455 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	3.05	0.778
19.8	0.814	2.8	0.714
19.2	0.79	2.7	0.689
18.5	0.761	2.65	0.676
17.6	0.724	2.4	0.612
16.8	0.691	2.25	0.574
15.9	0.654	2.1	0.536
15.3	0.629	1.75	0.446
13.8	0.568	1.8	0.459
14.6	0.6	1.65	0.421
13.2	0.543	1.45	0.37
12.2	0.502	1.45	0.37
11.5	0.473	1.0	0.225
9.9	0.407	0.8	0.204
10.4	0.428	1.05	0.268
8.6	0.354	0.65	0.166

Barometric Pressure = 753 mm Hg Temperature = 18.5°C

TEST RUN NO. 27
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 5
PARTITION SPACING = 0.305 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	2.4	0.612
20.0	0.823	2.3	0.587
19.2	0.79	2.2	0.561
18.5	0.761	2.15	0.548
17.5	0.72	1.9	0.485
16.6	0.683	1.85	0.472
15.7	0.646	1.5	0.383
15.0	0.617	1.5	0.383
14.0	0.576	1.3	0.332
13.5	0.555	1.05	0.268
12.8	0.526	1.1	0.281
11.7	0.481	1.05	0.268
11.3	0.465	0.9	0.23
10.6	0.436	0.9	0.23
9.1	0.374	0.8	0.204
8.2	0.337	0.5	0.128

Barometric Pressure = 753 mm Hg Temperature = 19°C

TEST RUN NO. 28
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 4
PARTITION SPACING = 0.305 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.1	0.536
19.7	0.81	2.0	0.51
19.2	0.79	2.0	0.51
18.5	0.761	1.75	0.446
17.7	0.728	1.6	0.408
16.8	0.691	1.45	0.37
15.8	0.65	1.3	0.332
15.3	0.629	1.3	0.332
14.6	0.6	1.2	0.306
13.8	0.568	1.1	0.281
12.9	0.531	0.95	0.242
12.1	0.498	1.0	0.255
11.25	0.463	0.9	0.23
10.4	0.428	0.65	0.166
9.2	0.378	0.4	0.102
8.2	0.337	0.3	0.077

Barometric Pressure = 754 mm Hg Temperature = 19°C

TEST RUN NO. 29
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 4
PARTITION SPACING = 0.405 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.7	0.689
19.8	0.814	2.4	0.612
19.2	0.79	2.25	0.574
18.5	0.761	2.1	0.536
17.5	0.72	1.9	0.485
16.4	0.675	1.7	0.434
16.8	0.691	1.75	0.446
15.9	0.654	1.6	0.408
15.3	0.629	1.5	0.385
14.7	0.605	1.35	0.344
14.3	0.588	1.3	0.332
12.8	0.526	1.0	0.255
13.3	0.547	0.9	0.23
11.5	0.473	0.8	0.204
10.6	0.436	0.65	0.166
9.6	0.395	0.5	0.128
8.5	0.35	0.45	0.115

Barometric Pressure = 755 mm Hg   Temperature = 20°C

TEST RUN NO. 30
TYPE OF PARTITION : HOLES

NUMBER OF PARTITIONS = 4
PARTITION SPACING = 0.81 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	3.7	0.944
19.8	0.814	3.6	0.918
19.3	0.794	3.45	0.88
18.5	0.761	3.05	0.778
17.6	0.724	2.85	0.727
16.9	0.695	2.55	0.65
15.9	0.654	2.4	0.612
15.9	0.654	2.5	0.638
15.7	0.646	2.3	0.587
13.8	0.568	1.85	0.472
14.2	0.584	1.85	0.472
13.4	0.551	1.65	0.421
12.6	0.518	1.55	0.395
11.6	0.477	1.3	0.332
10.7	0.44	1.1	0.281
9.6	0.395	0.9	0.23
9.1	0.374	0.75	0.191

Barometric Pressure = 755 mm Hg   Temperature = 20°C

TEST RUN NO. 31
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 3
PARTITION SPACING = 1.22 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	3.5	0.893
19.9	0.818	3.3	0.842
19.4	0.798	3.15	0.803
18.9	0.777	3.05	0.778
17.9	0.736	2.75	0.701
17.1	0.703	2.5	0.638
16.1	0.662	2.3	0.587
15.4	0.633	2.15	0.548
14.7	0.605	2.0	0.51
14.1	0.58	1.85	0.472
13.1	0.539	1.5	0.383
12.1	0.498	1.3	0.332
10.6	0.436	1.1	0.281
10.9	0.448	0.9	0.23
9.3	0.383	0.6	0.153
8.5	0.35	0.55	0.14

Barometric Pressure = 755 mm Hg   Temperature = 20°C

TEST RUN NO. 32
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 3
PARTITION SPACING = 0.915 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	3.1	0.791
19.4	0.798	2.7	0.689
18.9	0.777	2.55	0.65
18.6	0.765	2.7	0.689
17.5	0.72	2.2	0.561
16.8	0.691	2.1	0.536
16.3	0.67	2.1	0.536
15.3	0.629	1.8	0.459
14.6	0.6	1.65	0.421
13.8	0.568	1.55	0.395
13.2	0.543	1.5	0.383
12.25	0.504	1.35	0.344
11.0	0.452	0.9	0.23
10.3	0.424	0.75	0.191
10.0	0.411	0.8	0.204
8.9	0.366	0.7	0.179

Barometric Pressure = 756 mm Hg   Temperature = 20°C

TEST RUN NO. 33
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 3
PARTITION SPACING = 0.305 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	1.75	0.446
19.9	0.818	1.65	0.421
19.4	0.798	1.6	0.408
18.9	0.777	1.6	0.408
18.2	0.749	1.45	0.37
17.25	0.709	1.3	0.332
16.6	0.683	1.1	0.281
16.0	0.658	0.95	0.242
15.0	0.617	1.0	0.255
14.3	0.588	0.9	0.23
13.3	0.547	0.9	0.23
13	0.535	0.8	0.204
12.2	0.502	0.75	0.191
11.4	0.469	0.6	0.153
10.5	0.432	0.45	0.115
9.8	0.403	0.45	0.115
8.7	0.358	0.3	0.077

Barometric Pressure = 756 mm Hg   Temperature = 19.5°C

TEST RUN NO. 34
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 0.305

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	1.45	0.37
19.25	0.792	1.35	0.344
18.8	0.773	1.45	0.37
17.9	0.736	1.3	0.332
16.4	0.675	1.0	0.255
17.1	0.703	1.2	0.306
15.2	0.625	0.95	0.242
15.6	0.642	1.1	0.281
14.5	0.596	0.95	0.242
13.7	0.563	0.65	0.166
12.6	0.518	0.6	0.153
11.7	0.481	0.6	0.153
10.5	0.432	0.5	0.128
9.6	0.395	0.3	0.077
13.0	0.535	0.65	0.166

Barometric Pressure = 757 mm Hg   Temperature = 19°C

TEST RUN NO. 35
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 1.22 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
[illegible]	0.843	2.3	0.587
[illegible]	0.798	2.15	0.548
[illegible]	0.773	2.1	0.536
[illegible]	0.74	1.8	0.459
[illegible]	0.689	1.9	0.485
[illegible]	0.707	1.75	0.446
[illegible]	0.65	1.55	0.395
[illegible]	0.611	1.5	0.383
[illegible]	0.572	1.2	0.306
[illegible]	0.535	0.95	0.242
[illegible]	0.485	0.8	0.204
[illegible]	0.514	0.95	0.242
[illegible]	0.432	0.7	0.179
[illegible]	0.387	0.35	0.089
[illegible]	0.358	0.35	0.089

Barometric Pressure = 757 mm Hg   Temperature = 18.5°C

TEST RUN NO. 36
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 2.44 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	3.0	0.765
19.3	0.794	2.9	0.74
18.7	0.769	2.5	0.638
17.8	0.732	2.2	0.561
19.7	0.81	2.7	0.689
16.4	0.675	1.9	0.485
17.0	0.699	2.15	0.548
14.7	0.605	1.6	0.408
15.2	0.625	1.7	0.434
13.4	0.551	1.3	0.332
14.0	0.576	1.5	0.383
12.7	0.522	1.1	0.281
11.6	0.477	1.1	0.281
10.3	0.424	0.9	0.23
9.2	0.378	0.5	0.128
8.25	0.339	0.5	0.128

Barometric Pressure = 757 mm Hg   Temperature = 18.5°C

TEST RUN NO. 33
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 3
PARTITION SPACING = 0.305 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	1.75	0.446
19.9	0.818	1.65	0.421
19.4	0.798	1.6	0.408
18.9	0.777	1.6	0.408
18.2	0.749	1.45	0.37
17.25	0.709	1.3	0.332
16.6	0.683	1.1	0.281
16.0	0.658	0.95	0.242
15.0	0.617	1.0	0.255
14.3	0.588	0.9	0.23
13.3	0.547	0.9	0.23
13	0.535	0.8	0.204
12.2	0.502	0.75	0.191
11.4	0.469	0.6	0.153
10.5	0.432	0.45	0.115
9.8	0.403	0.45	0.115
8.7	0.358	0.3	0.077

Barometric Pressure = 756 mm Hg   Temperature = 19.5°C

TEST RUN NO. 34
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 0.305 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	1.45	0.37
19.25	0.792	1.35	0.344
18.8	0.773	1.45	0.37
17.9	0.736	1.3	0.332
16.4	0.675	1.0	0.255
17.1	0.703	1.2	0.306
15.2	0.625	0.95	0.242
15.6	0.642	1.1	0.281
14.5	0.596	0.95	0.242
13.7	0.563	0.65	0.166
12.6	0.518	0.6	0.153
11.7	0.481	0.6	0.153
10.5	0.432	0.5	0.128
9.6	0.395	0.3	0.077
13.0	0.535	0.65	0.166

Barometric Pressure = 757 mm Hg   Temperature = 19°C

TEST RUN NO. 35
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 1.22 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.3	0.587
19.4	0.798	2.15	0.548
18.8	0.773	2.1	0.536
18.0	0.74	1.8	0.459
16.75	0.689	1.9	0.485
17.2	0.707	1.75	0.446
15.8	0.65	1.55	0.395
14.85	0.611	1.5	0.383
13.9	0.572	1.2	0.306
13.0	0.535	0.95	0.242
11.8	0.485	0.8	0.204
12.5	0.514	0.95	0.242
10.5	0.432	0.7	0.179
9.4	0.387	0.35	0.089
8.7	0.358	0.35	0.089

Barometric Pressure = 757 mm Hg   Temperature = 18.5°C

TEST RUN NO. 36
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 2.44 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	3.0	0.765
19.3	0.794	2.9	0.74
18.7	0.769	2.5	0.638
17.8	0.732	2.2	0.561
19.7	0.81	2.7	0.689
16.4	0.675	1.9	0.485
17.0	0.699	2.15	0.548
14.7	0.605	1.6	0.408
15.2	0.625	1.7	0.434
13.4	0.551	1.3	0.332
14.0	0.576	1.5	0.383
12.7	0.522	1.1	0.281
11.6	0.477	1.1	0.281
10.3	0.424	0.9	0.23
9.2	0.378	0.5	0.128
8.25	0.339	0.5	0.128

Barometric Pressure = 757 mm Hg   Temperature = 18.5°C

TEST RUN NO. 37
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 1.83 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.7	0.689
19.8	0.814	2.5	0.638
19.2	0.79	2.4	0.612
17.8	0.732	1.8	0.459
18.4	0.757	2.35	0.599
16.9	0.695	2.0	0.51
16.3	0.67	1.95	0.497
15.25	0.627	1.45	0.37
14.2	0.584	1.1	0.281
13.7	0.563	1.2	0.306
12.7	0.522	1.25	0.319
11.7	0.481	0.9	0.23
15.0	0.617	1.5	0.383
9.8	0.403	0.85	0.217
10.3	0.424	0.65	0.166
8.8	0.362	0.4	0.102

Barometric Pressure = 756 mm HG   Temperature = 19.5°C

TEST RUN NO. 38
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 0.915 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	2.4	0.612
20.1	0.827	2.25	0.574
19.75	0.812	2.1	0.536
19.1	0.786	1.95	0.497
17.9	0.736	1.9	0.485
18.3	0.753	1.85	0.472
17.2	0.707	1.55	0.395
16.4	0.675	1.6	0.408
15.3	0.629	1.3	0.332
14.6	0.6	1.1	0.281
13.8	0.568	1.1	0.281
12.7	0.522	0.8	0.204
11.1	0.457	0.65	0.166
11.4	0.469	0.6	0.153
10.2	0.42	0.5	0.128
8.6	0.354	0.35	0.089

Barometric Pressure = 739 mm Hg   Temperature = 16.5°C

TEST RUN NO. 39
TYPE OF PARTITION : HOLE

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 0.455 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.4	0.839	1.9	0.485
19.7	0.81	1.9	0.485
19.0	0.781	1.6	0.408
18.4	0.757	1.6	0.408
17.8	0.732	1.5	0.383
17.1	0.703	1.3	0.332
16.1	0.662	1.2	0.306
16.7	0.687	1.4	0.357
15.4	0.633	0.95	0.242
13.9	0.572	0.8	0.204
12.9	0.531	0.7	0.179
14.5	0.596	1.0	0.255
11.4	0.469	0.65	0.166
12.0	0.494	0.7	0.179
10.3	0.424	0.3	0.077
9.1	0.374	0.2	0.051

Barometric Pressure = 739 mm Hg   Temperature = 16.5°C

TEST RUN NO. 40
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 5
PARTITION SPACING = 0.61 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.05	0.523
19.7	0.81	1.9	0.485
19.0	0.781	1.8	0.459
18.4	0.757	1.55	0.395
17.5	0.72	1.45	0.37
16.5	0.679	1.3	0.332
16.0	0.658	1.35	0.344
15.4	0.633	1.15	0.293
14.9	0.613	1.1	0.281
14.2	0.584	1.0	0.268
13.4	0.551	0.9	0.23
12.7	0.522	0.8	0.204
12.1	0.498	0.65	0.166
11.5	0.473	0.55	0.14
10.8	0.444	0.55	0.14
9.9	0.407	0.5	0.128

Barometric Pressure = 739 mm Hg   Temperature = 17°C

TEST RUN NO. 41
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 5
PARTITION SPACING = 0.455 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	1.8	0.459
19.7	0.81	1.55	0.395
19.1	0.786	1.65	0.421
18.5	0.761	1.45	0.37
17.7	0.728	1.45	0.37
16.8	0.691	1.25	0.319
15.8	0.65	1.1	0.281
15.2	0.625	1.1	0.281
14.6	0.6	1.05	0.268
14.0	0.576	0.95	0.242
13.4	0.551	0.8	0.204
12.4	0.51	0.75	0.191
12.0	0.494	0.7	0.179
11.2	0.461	0.65	0.166
10.6	0.436	0.7	0.179
9.4	0.387	0.4	0.102

Barometric Pressure = 739 mm Hg   Temperature = 17°C

TEST RUN NO. 42
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 5
PARTITION SPACING = 0.305 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	1.75	0.446
19.7	0.81	1.65	0.421
19.25	0.792	1.6	0.408
18.8	0.773	1.6	0.408
18.2	0.749	1.5	0.383
17.6	0.724	1.3	0.332
16.7	0.687	1.25	0.319
15.9	0.654	1.2	0.306
15.3	0.629	1.05	0.268
14.6	0.6	1.0	0.255
13.7	0.563	0.8	0.204
14.1	0.58	0.95	0.242
12.5	0.514	0.85	0.217
11.5	0.473	0.65	0.166
10.5	0.432	0.55	0.14
9.6	0.395	0.3	0.077

Barometric Pressure = 740 mm Hg   Temperature = 16.5°C

TEST RUN NO. 43
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 4
PARTITION SPACING = 0.305 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.15	0.548
19.7	0.81	2.25	0.574
19.1	0.786	1.75	0.446
18.6	0.765	1.9	0.485
18.0	0.74	1.65	0.421
17.1	0.703	1.55	0.395
16.5	0.679	1.35	0.344
16.0	0.658	1.3	0.332
15.0	0.617	1.15	0.293
14.6	0.6	0.95	0.242
13.8	0.568	0.95	0.242
12.7	0.522	0.7	0.179
11.6	0.477	0.65	0.166
10.7	0.44	0.7	0.179
9.4	0.387	0.4	0.102

Barometric Pressure = 741 mm Hg   Temperature = 16.5°C

TEST RUN NO. 44
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 4
PARTITION SPACING = 0.405 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	1.85	0.472
19.8	0.814	1.85	0.472
19.2	0.79	1.55	0.395
18.5	0.761	1.5	0.383
17.8	0.732	1.3	0.332
16.9	0.695	1.25	0.319
16.2	0.666	1.3	0.332
15.7	0.646	1.2	0.306
14.9	0.613	1.05	0.268
15.0	0.617	1.05	0.268
14.25	0.586	0.9	0.23
13.5	0.555	0.65	0.166
12.6	0.518	0.55	0.14
11.6	0.477	0.75	0.191
10.6	0.436	0.55	0.14
9.2	0.378	0.3	0.077

Barometric Pressure = 748 mm Hg   Temperature = 16°C

TEST RUN NO. 45
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 4
PARTITION SPACING = 0.61 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.4	0.839	1.75	0.446
19.7	0.81	1.65	0.421
19.1	0.786	1.6	0.408
18.6	0.765	1.6	0.408
18.2	0.749	1.45	0.37
16.9	0.695	1.05	0.268
17.5	0.72	1.3	0.332
16.2	0.666	1.05	0.268
15.2	0.625	0.95	0.242
15.5	0.638	0.8	0.204
14.7	0.605	0.9	0.23
14.0	0.576	0.9	0.23
12.8	0.526	0.7	0.179
11.5	0.473	0.65	0.166
10.4	0.428	0.4	0.102
9.3	0.383	0.3	0.077

Barometric Pressure = 748 mm Hg   Temperature = 16.5°C

TEST RUN NO. 46
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 4
PARTITION SPACING = 0.81 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.4	0.612
19.9	0.818	2.0	0.51
19.2	0.79	2.0	0.51
18.5	0.761	1.95	0.495
17.8	0.732	1.85	0.472
16.9	0.695	1.7	0.434
16.1	0.662	1.5	0.383
16.3	0.67	1.4	0.357
15.0	0.617	1.35	0.344
15.6	0.642	1.05	0.268
14.8	0.609	1.1	0.281
14.0	0.576	1.1	0.281
13.2	0.543	0.9	0.23
12.3	0.506	0.85	0.217
11.2	0.461	0.7	0.179
10.3	0.424	0.55	0.14
9.25	0.38	0.45	0.115

Barometric Pressure = 748 mm Hg   Temperature = 16.5°C

TEST RUN NO. 47
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 3
PARTITION SPACING = 1.22 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.4	0.839	2.3	0.587
19.8	0.814	2.2	0.561
19.2	0.79	1.9	0.485
18.4	0.757	1.75	0.446
17.5	0.72	1.7	0.434
16.7	0.687	1.5	0.383
16.1	0.662	1.3	0.332
15.5	0.638	1.1	0.281
15.0	0.617	1.1	0.281
14.5	0.596	1.1	0.281
14.1	0.58	1.0	0.255
12.9	0.531	0.9	0.23
13.4	0.551	0.95	0.242
11.2	0.461	0.5	0.128
10.3	0.424	0.3	0.077
9.3	0.383	0.4	0.102

Barometric Pressure = 748 mm Hg   Temperature = 17°C

TEST RUN NO. 48
TYPE OF PARTITION : WALL

NUMBLR OF PARTITIONS = 3
PARTITION SPACING = 0.915 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.4	0.839	2.05	0.523
19.6	0.806	1.85	0.472
19.1	0.786	1.85	0.472
18.5	0.761	1.7	0.434
17.75	0.73	1.45	0.37
16.7	0.687	1.4	0.357
16.3	0.67	1.2	0.306
15.8	0.65	1.2	0.306
15.0	0.617	1.1	0.281
14.6	0.6	1.05	0.268
13.8	0.568	1.05	0.268
13	0.535	1.0	0.255
11.3	0.465	0.55	0.14
11.6	0.477	0.7	0.179
10.4	0.428	0.5	0.128
9.3	0.383	0.3	0.077

Barometric Pressure = 748 mm Hg   Temperature = 17°C

TEST RUN NO. 49
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 3
PARTITION SPACING = 0.61 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.1	0.536
19.9	0.818	2.0	0.51
19.3	0.794	1.8	0.459
18.8	0.773	1.8	0.459
18.0	0.74	1.7	0.434
17.2	0.707	1.5	0.383
16.7	0.687	1.3	0.332
16.2	0.666	1.2	0.306
15.75	0.648	1.25	0.319
14.7	0.605	1.1	0.281
13.9	0.572	0.9	0.23
13.1	0.539	0.95	0.242
12.25	0.504	0.65	0.166
11.3	0.465	0.7	0.179
10.5	0.432	0.3	0.077
9.7	0.399	0.3	0.077

Barometric Pressure = 748 mm Hg   Temperature = 18°C

TEST RUN NO. 50
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 3
PARTITION SPACING = 0.305 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.25	0.574
19.8	0.814	1.95	0.497
19.3	0.794	1.95	0.497
18.8	0.773	1.9	0.485
18.3	0.753	1.9	0.485
17.5	0.72	1.9	0.485
16.9	0.695	1.8	0.459
16.3	0.67	1.55	0.395
15.5	0.638	1.5	0.383
14.8	0.609	1.5	0.383
14.3	0.588	1.1	0.281
13.4	0.551	1.0	0.255
12.4	0.51	0.9	0.23
11.5	0.473	0.75	0.191
10.9	0.448	0.55	0.14
9.6	0.395	0.5	0.128

Barometric Pressure = 749 mm Hg   Temperature = 18°C

TEST RUN NO. 51
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 0.305 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.5	0.638
19.7	0.81	2.25	0.574
19.3	0.794	1.9	0.485
18.8	0.773	1.9	0.485
18.2	0.749	1.75	0.446
17.2	0.707	1.4	0.357
16.8	0.691	1.45	0.37
16.25	0.668	1.5	0.383
15.5	0.638	1.5	0.383
15.0	0.617	1.5	0.383
14.8	0.609	1.25	0.319
13.7	0.563	1.2	0.306
12.7	0.522	0.9	0.23
11.7	0.481	0.6	0.153
10.7	0.44	0.5	0.128
9.8	0.403	0.4	0.102

Barometric Pressure = 749 mm Hg   Temperature = 17.5°C

TEST RUN NO. 52
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 0.455

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.35	0.599
19.8	0.814	2.3	0.587
19.4	0.798	2.1	0.536
18.8	0.773	2.05	0.523
18.2	0.749	1.85	0.472
17.5	0.72	1.6	0.408
17.0	0.699	1.55	0.395
16.4	0.675	1.7	0.434
15.9	0.654	1.5	0.383
14.9	0.613	1.1	0.281
13.6	0.559	1.3	0.382
13.9	0.572	1.35	0.344
12.5	0.514	1.0	0.255
11.7	0.481	0.65	0.166
10.5	0.432	0.5	0.128
9.5	0.391	0.5	0.128

Barometric Pressure = 749 mm Hg   Temperature = 17.5°C

TEST RUN NO. 53
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 0.61 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.4	0.839	2.5	0.638
19.7	0.81	2.3	0.587
19.1	0.786	2.4	0.612
18.4	0.757	2.05	0.523
17.5	0.72	2.1	0.536
17.1	0.703	1.85	0.472
16.2	0.666	1.8	0.459
15.8	0.65	1.5	0.383
15.0	0.617	1.5	0.383
14.4	0.592	1.45	0.37
14.0	0.576	1.3	0.332
13.3	0.547	1.1	0.281
12.4	0.51	0.7	0.179
11.2	0.461	0.9	0.23
10.3	0.424	0.7	0.179
9.5	0.391	0.65	0.166

Barometric Pressure = 750 mm Hg   Temperature = 17.5°C

TEST RUN NO. 54
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 0.455 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	2.15	0.548
19.8	0.814	2.0	0.51
19.4	0.798	2.1	0.536
18.8	0.773	1.9	0.485
18.0	0.74	1.5	0.383
17.0	0.699	1.55	0.395
17.2	0.707	1.45	0.37
16.1	0.662	1.35	0.344
15.7	0.646	1.3	0.332
15.0	0.617	1.2	0.306
14.7	0.605	1.1	0.281
14.0	0.576	1.05	0.268
13.1	0.539	0.85	0.217
12.3	0.506	0.85	0.217
11.4	0.469	0.75	0.191
10.3	0.424	0.5	0.128
9.5	0.391	0.4	0.102

Barometric Pressure = 748 mm Hg   Temperature = 18°C

TEST RUN NO. 55
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 1.22 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.15	0.548
19.75	0.812	2.05	0.523
19.3	0.794	1.95	0.497
18.8	0.773	1.9	0.485
18.3	0.753	1.9	0.485
17.0	0.699	1.5	0.383
16.4	0.675	1.35	0.344
15.8	0.65	1.3	0.332
15.1	0.621	1.0	0.255
14.4	0.592	1.0	0.255
13.7	0.563	0.9	0.23
13.3	0.547	0.9	0.23
12.1	0.498	0.65	0.166
11.0	0.452	0.6	0.153
12.5	0.514	0.8	0.204
10.5	0.432	0.55	0.14

Barometric Pressure = 748 mm Hg   Temperature = 18°C

TEST RUN NO. 56
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 1.83

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	1.95	0.497
19.9	0.818	1.8	0.459
19.3	0.794	1.7	0.434
18.75	0.771	1.7	0.434
18.1	0.744	1.5	0.383
17.3	0.712	1.4	0.357
16.1	0.662	1.05	0.268
16.6	0.683	1.5	0.383
15.7	0.646	1.1	0.281
15.3	0.629	1.05	0.268
14.7	0.605	0.9	0.23
13.5	0.555	0.85	0.217
12.5	0.514	0.7	0.179
11.7	0.481	0.7	0.179
10.9	0.448	0.6	0.153
9.6	0.395	0.4	0.102

Barometric Pressure = 748 mm Hg   Temperature = 18°C

TEST RUN NO. 57
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 2
PARTITION SPACING = 2.44 m

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.6	0.847	2.35	0.599
19.9	0.818	2.25	0.574
19.4	0.798	2.1	0.536
18.7	0.769	1.9	0.485
17.4	0.716	1.75	0.446
17.7	0.728	1.85	0.472
16.7	0.687	1.65	0.421
16.2	0.666	1.6	0.408
15.6	0.642	1.4	0.357
14.5	0.596	1.3	0.332
14.1	0.58	1.1	0.281
13.7	0.563	1.1	0.281
12.8	0.526	1.1	0.281
11.8	0.485	0.8	0.204
10.8	0.444	0.7	0.179
9.75	0.401	0.6	0.153

Barometric Pressure = 748 mm Hg   Temperature = 18.5°C

TEST RUN NO. 58
TYPE OF PARTITION : WALL

NUMBER OF PARTITIONS = 1
PARTITION SPACING = n/a

FLOW RATE		PRESSURE DIFFERENTIAL	
Vane Anemometer (m/s)	Equivalent Volume (m^3/s)	Transducer Output (mV)	Equivalent Pressure (Pa)
20.5	0.843	2.25	0.574
19.75	0.812	2.1	0.536
19.3	0.794	2.1	0.536
18.7	0.769	2.0	0.51
18.0	0.74	1.9	0.485
17.3	0.712	1.85	0.472
16.8	0.691	1.8	0.459
16.1	0.662	1.6	0.408
15.6	0.642	1.6	0.408
15.2	0.625	1.5	0.383
14.5	0.596	1.35	0.344
13.5	0.555	1.15	0.293
12.7	0.522	1.1	0.281
11.4	0.469	0.9	0.23
10.6	0.436	0.65	0.166
9.8	0.403	0.45	0.115

Barometric Pressure = 748 mm Hg   Temperature = 18°C

APPENDIX B3

RESULTS OF WIND TUNNEL TRIALS USING
RECTANGULAR WALL PARTITIONS.

The results of these trials are presented on the following pages in tabular form.

Appendix B3: Wind Tunnel trials
Wall type partitions
Trial numbers 6-34.

TRIAL NUMBER 6

Number of Partitions 1
Spacing -
Temperature 23.8 DegC
Barometric Pressure 754 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.520	.16	.009	.9
1.130	.36	.047	4.6
1.400	.44	.070	6.9
1.690	.53	.103	10.1
1.970	.62	.142	13.9
2.190	.69	.179	17.6
2.550	.81	.250	24.5
2.890	.91	.320	31.4
3.480	1.10	.455	44.6
3.860	1.22	.570	55.9
4.270	1.35	.700	68.6
4.410	1.39	.745	73.1
4.780	1.51	.865	84.8
4.950	1.56	.935	91.7

TRIAL NUMBER 7

Number of Partitions 1
Spacing -
Temperature 23.9 DegC
Barometric Pressure 753.5 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.100	.03	.006	.6
1.050	.33	.040	3.9
1.200	.38	.051	5.0
1.450	.46	.079	7.7
1.900	.60	.135	13.2
2.100	.66	.162	15.9
2.200	.70	.182	17.8
2.700	.85	.275	27.0
3.050	.96	.360	35.3
3.200	1.01	.395	38.7
3.600	1.14	.500	49.0
4.050	1.28	.620	60.8
4.200	1.33	.675	66.2
4.600	1.45	.800	78.5
5.000	1.58	.935	91.7

TRIAL NUMBER 8

Number of Partitions 0
Spacing -
Temperature 24.3 DegC
Barometric Pressure 753 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
1.175	.38	.001	.1
1.450	.47	.002	.2
2.500	.81	.005	.5
3.000	.97	.007	.7
4.325	1.39	.014	1.4
5.200	1.68	.019	1.9
6.400	2.06	.029	2.8
7.600	2.45	.041	4.0

TRIAL NUMBER 9

Number of Partitions 2
Spacing 0.152m
Temperature 24.4 DegC
Barometric Pressure 753 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.800	.25	.023	2.3
1.075	.34	.041	4.0
1.325	.42	.064	6.3
1.800	.57	.112	11.0
1.950	.61	.138	13.5
2.275	.72	.190	18.6
2.500	.79	.235	23.0
2.975	.94	.330	32.4
3.200	1.01	.380	37.3
3.375	1.06	.425	41.7
3.950	1.24	.585	57.4
4.175	1.31	.650	63.7
4.625	1.46	.795	78.0
5.150	1.62	.950	93.2

TRIAL NUMBER 10

Number of Partitions 2
Spacing 0.152m
Temperature 24.5 DegC
Barometric Pressure 753 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
1.275	.40	.058	5.7
2.050	.65	.152	14.9
3.250	1.02	.400	39.2
4.375	1.38	.710	69.6

TRIAL NUMBER 11+11a

Number of Partitions 2
Spacing 1.83m
Temperature 24.6 DegC
Barometric Pressure 753 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
1.050	.33	.030	2.9
1.225	.38	.040	3.9
1.400	.43	.055	5.4
1.700	.53	.085	8.3
2.200	.68	.144	14.1
2.350	.73	.168	16.5
2.850	.88	.255	25.0
2.900	.90	.265	26.0
3.450	1.07	.370	36.3
3.600	1.12	.410	40.2
4.125	1.28	.550	53.9
4.350	1.35	.595	58.3
4.800	1.49	.725	71.1
5.300	1.64	.850	83.4
.600	.19	.009	.9
1.600	.50	.071	7.0
2.250	.70	.155	15.2
3.100	.96	.305	29.9
4.300	1.33	.590	57.9

TRIAL NUMBER 12+12a

Number of Partitions 3
Spacing 0.91m
Temperature 24.9 DegC
Barometric Pressure 753 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.675	.21	.010	1.0
.850	.26	.019	1.9
1.125	.35	.033	3.2
1.425	.44	.054	5.3
1.700	.53	.082	8.0
2.225	.69	.142	13.9
2.325	.72	.155	15.2
2.450	.76	.175	17.2
2.650	.82	.212	20.8
3.000	.93	.285	27.9
3.225	1.00	.330	32.4
3.625	1.12	.415	40.7
4.000	1.24	.510	50.0
4.350	1.35	.615	60.3
4.600	1.43	.675	66.2
5.000	1.55	.805	78.9
.500	.16	.005	.5
1.375	.43	.054	5.3
2.125	.66	.128	12.6
3.175	.99	.310	30.4
4.150	1.29	.550	53.9

TRIAL NUMBER 13+13a

Number of Partitions 4
Spacing 0.61m
Temperature 23.0 DegC
Barometric Pressure 757 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
1.020	.32	.029	2.8
1.200	.37	.038	3.7
1.475	.46	.058	5.7
2.025	.63	.112	11.0
2.225	.69	.138	13.5
2.475	.77	.173	17.0
3.100	.96	.280	27.5
3.575	1.11	.360	35.3
3.725	1.16	.410	40.2
4.025	1.25	.470	46.1
4.625	1.44	.610	59.8
4.800	1.49	.670	65.7
5.400	1.68	.795	78.0
5.750	1.78	.915	89.7
.700	.22	.012	1.2
1.750	.54	.082	8.0
2.450	.76	.170	16.7
3.550	1.10	.366	35.8
4.575	1.42	.615	60.3

TRIAL NUMBER 14+14a

Number of Partitions 3
Spacing 0.61m
Temperature 23.4 DegC
Barometric Pressure 757 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.875	.27	.018	1.8
1.175	.37	.032	3.1
1.425	.44	.049	4.8
1.875	.58	.085	8.3
2.075	.65	.103	10.1
2.250	.70	.122	12.0
2.775	.86	.190	18.6
2.900	.90	.210	20.6
3.375	1.05	.285	27.9
3.550	1.10	.315	30.9
4.225	1.31	.445	43.6
4.525	1.41	.515	50.5
4.900	1.52	.610	59.8
5.600	1.74	.735	72.1
1.475	.46	.053	5.2
2.200	.68	.117	11.5
3.375	1.05	.280	27.5
4.425	1.38	.485	47.6

TRIAL NUMBER 15+15a

Number of Partitions 2
Spacing 1.22m
Temperature 23.6 DegC
Barometric Pressure 757 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.675	.21	.010	1.0
.800	.25	.016	1.6
1.000	.31	.025	2.5
1.175	.37	.034	3.3
1.475	.46	.055	5.4
1.750	.54	.080	7.8
2.100	.65	.115	11.3
2.475	.77	.162	15.9
2.625	.82	.185	18.1
3.025	.94	.250	24.5
3.375	1.05	.315	30.9
4.025	1.25	.430	42.2
4.225	1.31	.475	46.6
4.725	1.47	.585	57.4
5.250	1.63	.705	69.1
.525	.16	.008	.8
1.400	.44	.060	5.9
2.125	.66	.115	11.3
3.000	.93	.240	23.5
4.175	1.30	.465	45.6

TRIAL NUMBER 16+16a

Number of Partitions 1
Spacing -
Temperature 23.6 DegC
Barometric Pressure 757 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.675	.21	.016	1.6
.925	.29	.029	2.8
1.200	.37	.045	4.4
1.350	.42	.058	5.7
1.625	.51	.085	8.3
1.925	.60	.122	12.0
2.100	.65	.146	14.3
2.455	.76	.198	19.4
2.925	.91	.295	28.9
3.200	1.00	.360	35.3
3.725	1.16	.480	47.1
3.975	1.24	.545	53.4
4.375	1.36	.665	65.2
4.800	1.49	.795	78.0
.450	.14	.006	.6
1.350	.42	.060	5.9
1.875	.58	.118	11.6
2.825	.88	.275	27.0
3.800	1.18	.505	49.5

TRIAL NUMBER 17+17a

Number of Partitions 1
Spacing -
Temperature 23.6 DegC
Barometric Pressure 757 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.625	.19	.008	.8
.850	.26	.016	1.6
1.150	.36	.029	2.8
1.650	.51	.064	6.3
1.850	.57	.080	7.8
2.025	.63	.100	9.8
2.300	.71	.130	12.7
2.775	.86	.190	18.6
3.125	.97	.250	24.5
3.575	1.11	.320	31.4
4.025	1.25	.410	40.2
4.250	1.32	.460	45.1
4.725	1.47	.560	54.9
5.400	1.68	.680	66.7
.525	.16	.007	.7
1.325	.41	.040	3.9
2.175	.67	.112	11.0
3.250	1.01	.265	26.0
4.325	1.34	.470	46.1

TRIAL NUMBER 18+18a

Number of Partitions 2
Spacing 1.68m
Temperature 24.1 DegC
Barometric Pressure 757 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.675	.21	.012	1.2
.925	.29	.023	2.3
1.175	.36	.038	3.7
1.625	.50	.076	7.5
1.775	.55	.087	8.5
1.925	.60	.105	10.3
2.425	.75	.172	16.9
2.775	.86	.235	23.0
2.975	.92	.270	26.5
3.325	1.03	.345	33.8
3.800	1.18	.430	42.2
4.025	1.25	.480	47.1
4.450	1.38	.600	58.8
4.850	1.51	.715	70.1
.450	.14	.006	.6
1.225	.38	.042	4.1
1.925	.60	.103	10.1
2.975	.92	.265	26.0
3.975	1.23	.475	46.6

TRIAL NUMBER 19+19a

Number of Partitions 2
Spacing 1.52m
Temperature 24.3 DegC
Barometric Pressure 757 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.700	.22	.012	1.2
.900	.28	.023	2.3
1.200	.37	.040	3.9
1.675	.52	.079	7.7
1.925	.60	.100	9.8
2.075	.64	.123	12.1
2.600	.81	.197	19.3
3.050	.95	.265	26.0
3.300	1.03	.315	30.9
3.800	1.18	.420	41.2
4.275	1.33	.525	51.5
4.425	1.37	.570	55.9
4.750	1.48	.650	63.7
5.300	1.65	.760	74.5
.500	.16	.008	.8
1.400	.43	.052	5.1
2.075	.64	.120	11.8
3.125	.97	.280	27.5
4.175	1.30	.500	49.0

TRIAL NUMBER 20+20a

Number of Partitions 3
Spacing .76m
Temperature 24.4 DegC
Barometric Pressure 757 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.825	.26	.018	1.8
.950	.30	.024	2.4
1.175	.37	.036	3.5
1.600	.50	.065	6.4
1.950	.61	.097	9.5
2.100	.65	.115	11.3
2.625	.82	.183	17.9
3.000	.93	.240	23.5
3.400	1.06	.310	30.4
3.600	1.12	.340	33.3
4.025	1.25	.425	41.7
4.200	1.30	.470	46.1
4.575	1.42	.600	58.8
5.200	1.62	.725	71.1
.475	.15	.006	.6
1.325	.41	.048	4.7
2.225	.69	.138	13.5
3.100	.96	.275	27.0
4.200	1.30	.470	46.1

TRIAL NUMBER 21+21a

Number of Partitions 2
Spacing 1.37m
Temperature 22.8 DegC
Barometric Pressure 758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.150	.05	.005	.5
.775	.24	.018	1.8
.975	.30	.027	2.6
1.250	.39	.044	4.3
1.500	.47	.061	6.0
1.825	.57	.092	9.0
2.000	.62	.109	10.7
2.375	.74	.155	15.2
2.925	.91	.240	23.5
3.175	.99	.280	27.5
3.575	1.11	.360	35.3
4.050	1.26	.465	45.6
4.525	1.41	.570	55.9
4.800	1.49	.640	62.8
5.500	1.71	.770	75.5
.575	.18	.010	1.0
1.550	.48	.064	6.3
2.225	.69	.135	13.2
3.325	1.03	.305	29.9
4.325	1.34	.525	51.5

TRIAL NUMBER 22+22a

Number of Partitions 4
Spacing 0.46m
Temperature 23 DegC
Barometric Pressure 757 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.150	.05	.002	.2
.800	.25	.014	1.4
1.075	.33	.027	2.6
1.375	.43	.046	4.5
1.825	.57	.082	8.0
2.000	.62	.102	10.0
2.225	.69	.125	12.3
2.750	.85	.195	19.1
3.125	.97	.255	25.0
3.375	1.05	.295	28.9
3.800	1.18	.380	37.3
4.275	1.33	.470	46.1
4.500	1.40	.525	51.5
4.875	1.51	.630	61.8
5.500	1.71	.740	72.6
.550	.17	.009	.9
1.500	.47	.056	5.5
2.400	.74	.146	14.3
3.325	1.03	.285	27.9
4.375	1.36	.495	48.5

TRIAL NUMBER 23+23a

Number of Partitions 3
Spacing 0.46m
Temperature 23.4 DegC
Barometric Pressure 758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.125	.04	.003	.3
.825	.26	.014	1.4
1.075	.33	.025	2.5
1.375	.43	.043	4.2
1.550	.48	.054	5.3
2.000	.62	.090	8.8
2.175	.68	.105	10.3
2.400	.75	.132	12.9
2.725	.85	.170	16.7
3.300	1.03	.247	24.2
3.750	1.17	.320	31.4
4.200	1.30	.405	39.7
4.450	1.38	.455	44.6
4.950	1.54	.550	53.9
.500	.16	.006	.6
1.400	.43	.044	4.3
2.150	.67	.102	10.0
3.225	1.00	.240	23.5
4.325	1.34	.425	41.7

TRIAL NUMBER 24+24a

Number of Partitions 2
Spacing 0.91m
Temperature 23.5 DegC
Barometric Pressure 758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.725	.23	.013	1.3
.975	.30	.023	2.3
1.150	.36	.031	3.0
1.400	.44	.049	4.8
1.850	.58	.084	8.2
2.025	.63	.102	10.0
2.400	.75	.140	13.7
2.575	.80	.170	16.7
3.100	.97	.250	24.5
3.300	1.03	.280	27.5
3.900	1.21	.390	38.2
4.150	1.29	.440	43.1
4.700	1.46	.550	53.9
5.400	1.68	.655	64.2
.475	.15	.005	.5
1.350	.42	.043	4.2
2.025	.63	.100	9.8
3.075	.96	.240	23.5
4.050	1.26	.420	41.2

TRIAL NUMBER 25+25a

Number of Partitions 4
Spacing 0.305m
Temperature 23.8 DegC
Barometric Pressure 758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.725	.23	.012	1.2
.950	.30	.020	2.0
1.200	.37	.034	3.3
1.500	.47	.051	5.0
1.850	.58	.078	7.6
2.025	.63	.094	9.2
2.250	.70	.118	11.6
2.600	.81	.160	15.7
3.150	.98	.235	23.0
3.375	1.05	.270	26.5
3.975	1.24	.375	36.8
4.125	1.28	.410	40.2
4.625	1.44	.505	49.5
5.300	1.65	.625	61.3
.450	.14	.004	.4
1.225	.38	.034	3.3
2.050	.64	.098	9.6
3.125	.97	.235	23.0
4.200	1.31	.415	40.7

TRIAL NUMBER 26+26a

Number of Partitions	3
Spacing	0.305m
Temperature	23.8 DegC
Barometric Pressure	758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.775	.24	.015	1.5
.925	.29	.020	2.0
1.200	.38	.035	3.4
1.400	.44	.048	4.7
1.875	.59	.048	4.7
2.125	.66	.105	10.3
2.450	.77	.150	14.7
3.050	.95	.225	22.1
3.200	1.00	.250	24.5
3.425	1.07	.285	27.9
4.075	1.27	.405	39.7
4.350	1.36	.440	43.1
4.875	1.52	.500	49.0
5.500	1.72	.600	58.8
.500	.16	.007	.7
1.325	.41	.044	4.3
2.075	.65	.100	9.8
3.200	1.00	.250	24.5
4.525	1.41	.455	44.6

TRIAL NUMBER 27+27a

Number of Partitions	2
Spacing	0.305m
Temperature	24.2 DegC
Barometric Pressure	758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.575	.18	.012	1.2
.700	.22	.017	1.7
.900	.28	.028	2.7
1.025	.32	.035	3.4
1.300	.41	.055	5.4
1.750	.55	.101	9.9
1.925	.60	.125	12.3
2.250	.71	.175	17.2
2.750	.86	.265	26.0
2.950	.93	.305	29.9
3.325	1.04	.390	38.2
3.750	1.18	.490	48.1
3.900	1.22	.525	51.5
4.075	1.28	.575	56.4
4.700	1.48	.760	74.5
.450	.14	.006	.6
1.225	.38	.050	4.9
1.925	.60	.123	12.1
2.850	.90	.285	27.9
3.800	1.19	.500	49.0

TRIAL NUMBER 28+28a

Number of Partitions	5
Spacing	0.152m
Temperature	24.4 DegC
Barometric Pressure	758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.675	.21	.014	1.4
.850	.27	.023	2.3
1.125	.35	.041	4.0
1.425	.45	.066	6.5
1.575	.49	.085	8.3
1.800	.57	.105	10.3
1.975	.62	.130	12.7
2.300	.72	.185	18.1
2.775	.87	.270	26.5
2.975	.93	.310	30.4
3.500	1.10	.435	42.7
3.775	1.19	.495	48.5
4.200	1.32	.605	59.3
4.550	1.43	.720	70.6
.425	.13	.005	.5
1.125	.35	.042	4.1
1.825	.57	.110	10.8
2.800	.88	.270	26.5
3.725	1.17	.485	47.6

TRIAL NUMBER 29+29a

Number of Partitions	4
Spacing	0.152m
Temperature	24.3 DegC
Barometric Pressure	758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.650	.20	.011	1.1
.875	.27	.020	2.0
1.000	.31	.027	2.6
1.400	.44	.051	5.0
1.675	.52	.079	7.7
1.875	.59	.098	9.6
2.075	.65	.120	11.8
2.375	.74	.165	16.2
2.925	.92	.255	25.0
3.325	1.04	.335	32.9
3.725	1.17	.415	40.7
4.000	1.25	.470	46.1
4.425	1.39	.580	56.9
4.800	1.50	.675	66.2
.475	.15	.005	.5
1.225	.38	.039	3.8
1.875	.59	.098	9.6
2.700	.85	.215	21.1
3.875	1.21	.450	44.[illegible]

TRIAL NUMBER 30+30a

Number of Partitions	2
Spacing	0.46m
Temperature	24.6 DegC
Barometric Pressure	758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.650	.20	.010	1.0
.850	.27	.020	2.0
1.100	.34	.033	3.2
1.375	.43	.055	5.4
1.650	.52	.081	7.9
1.825	.57	.098	9.6
2.025	.63	.125	12.3
2.650	.83	.220	21.6
2.825	.89	.255	25.0
3.050	.96	.295	28.9
3.700	1.16	.430	42.2
3.900	1.22	.475	46.6
4.300	1.35	.585	57.4
4.775	1.50	.710	69.6
.450	.14	.004	.4
1.200	.38	.040	3.9
1.875	.59	.105	10.3
2.850	.89	.255	25.0
3.825	1.20	.460	45.1

TRIAL NUMBER 31+31a

Number of Partitions	2
Spacing	1.07m
Temperature	24.6 DegC
Barometric Pressure	758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.600	.19	.008	.8
.825	.26	.015	1.5
.950	.30	.020	2.0
1.425	.44	.048	4.7
1.600	.50	.060	5.9
1.750	.54	.075	7.4
2.125	.66	.110	10.8
2.700	.84	.190	18.6
2.950	.92	.225	22.1
3.400	1.06	.295	28.9
3.775	1.17	.370	36.3
4.150	1.29	.445	43.6
4.400	1.37	.505	49.5
4.825	1.50	.610	59.8
.450	.14	.003	.3
1.075	.33	.026	2.5
1.825	.57	.084	8.2
2.875	.89	.215	21.1
3.875	1.21	.390	38.2

TRIAL NUMBER 32+32a

Number of Partitions	2
Spacing	0.76m
Temperature	24.7 DegC
Barometric Pressure	758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.650	.20	.009	.9
.850	.27	.016	1.6
.950	.30	.022	2.2
1.375	.43	.046	4.5
1.525	.48	.059	5.8
1.825	.57	.083	8.1
2.200	.69	.120	11.8
2.375	.74	.145	14.2
2.900	.90	.220	21.6
3.150	.98	.255	25.0
3.750	1.17	.360	35.3
3.950	1.23	.400	39.2
4.475	1.40	.490	48.1
4.900	1.53	.590	57.9
.400	.12	.003	.3
1.100	.34	.028	2.7
1.925	.60	.090	8.8
2.950	.92	.225	22.1
3.975	1.24	.395	38.7

TRIAL NUMBER 33+33a

Number of Partitions	2
Spacing	0.61m
Temperature	24.5 DegC
Barometric Pressure	758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.675	.21	.007	.7
.800	.25	.014	1.4
.900	.28	.021	2.1
1.300	.41	.042	4.1
1.625	.51	.070	6.9
1.825	.57	.085	8.3
2.150	.67	.125	12.3
2.350	.73	.150	14.7
2.900	.91	.235	23.0
3.125	.98	.265	26.0
3.775	1.18	.385	37.8
4.050	1.27	.425	41.7
4.400	1.38	.490	48.1
4.875	1.52	.625	61.3
.400	.13	.003	.3
1.200	.38	.034	3.3
1.900	.59	.090	8.8
2.900	.91	.235	23.0
3.975	1.24	.420	[illegible]

TRIAL NUMBER 34+34a

Number of Partitions 1
Spacing -
Temperature 22.6 DegC
Barometric Pressure 758 mmHg

FLOW RATE		PRESSURE DIFFERENCE	
Vane Anemometer m/s	Volumetric cu.m/s	Voltage V	Equivalent Pa
.675	.21	.018	1.8
1.000	.32	.038	3.7
1.150	.36	.051	5.0
1.500	.47	.084	8.2
1.575	.50	.094	9.2
1.925	.61	.140	13.7
2.225	.70	.190	18.6
2.350	.74	.225	22.1
2.975	.94	.340	33.3
3.250	1.03	.410	40.2
3.800	1.20	.565	55.4
4.000	1.26	.630	61.8
4.250	1.34	.700	68.6
4.900	1.55	.915	89.7
.525	.17	.011	1.1
1.250	.39	.058	5.7
2.050	.65	.158	15.5
3.075	.97	.360	35.3
4.050	1.28	.635	62.3

APPENDIX C

RECOMMENDATIONS FOR IMPROVED PERFORMANCE AT ICI FIBRES, DONCASTER.

C1 The site and building layout, the process operation and the ventilation and air conditioning systems relating to the spinning area of the factory have already been described.

Though production takes place on several distinct floor levels, the floor levels themselves are further subdivided by machinery layout. This effectively sets up a large number of regularly spaced partitions across each floor. This partitioning feature of the building is important when considering the air flows and air movement systems.

A building pressurization test and air flow visualization using smoke tracer, provided information relating to air movement in the production areas. In addition, the environmental monitoring system gave data for ducted air flows and for temperatures and humidities in the factory.

C2 Within the ground floor Spin Doff area there seems to be an imbalance between the amounts of supply and extract air which causes the area to be at a negative pressure with respect to adjacent areas and the outside air. This results in the influx of air. For air conditioned areas such an influx is undesirable since it can lead to variations which produce air temperatures and moisture contents much different to those required. In fact it is usually the case that air conditioned areas are designed to operate at slight positive pressures. The reason for the imbalance is the use of the blower air system which extracts considerable volumes of air from the Spin Doff area and supplies these to machines at the Extrusion level. Thus although the basic Spin Doff systems do operate with an excess of supply over extract the situation is reversed by inclusion of blower air at the volumes encountered.

In the short term this situation might be improved by operation of blower air fans being restricted to the amounts required for the machines in use. However, since the blower air cooling of the nylon yarn is a critical process, such reductions would have to be managed very carefully. There are dampers which could be closed in ducts within the Type 14 area blower air system, which coupled with reduced fan operation could achieve the desired results. In the long term if more complete electronic control could be exercised from a central location, then the running of the system would be much improved. The use of such a system is dealt with later in this appendix.

C3 The use of heaters in the return air ducts of the Spin Doff system should be stopped if this has not already been done. These heaters were designed to produce return air at a certain temperature, thus if no production was being carried out on a specific area, with consequent reduced heat liberation, then the heaters have to compensate. This is very wasteful in terms of energy use. The need for conditioned air in such areas needs to be examined and this is discussed later. Since a machine in production will produce a relatively constant load on the system, then it should be possible to predict the required supply air conditions and design the systems to provide such a supply.

C4 Turning to the nylon melting and extrusion process which takes place on the Extrusion floor level. The thermal efficiency of the process has been determined by ICI to be very low. This occurs because the melting takes place within metal spinning units, which have little or no insulation as this could otherwise hinder operation and maintenance. Consequently there is a high heat emission (particularly radiative) into the area. In order to allow personnel to work in such areas, cool air must be supplied and hot air extracted. Much of the heat is exhausted at relatively low grade into the external air.

Smoke visualization of the air flows showed them to be, on occasion, somewhat different to what might have been expected, with recirculation and stagnation zones set up. This means that the hot air extracted may be below the maximum temperature and warm air may be blown back over staff working on the machines.

A significant improvement could be achieved by use of heat reflecting panels at the extrusion catwalk level, between the melting units and the walkway. These would have three main effects. Firstly, a large amount of the radiant heat emission would be reflected and prevented from impinging on working personnel and thus improving working conditions. Secondly, by containing the heat near the melting units, a reduced temperature gradient would be found with consequent less heat loss. This would improve the thermal efficiency of the melting process. The third effect would be that the direction of the air flows could be more easily controlled. Hot air could be more efficiently extracted, perhaps at a high enough temperature so that some form of heat recovery could be contemplated; and cool air could be supplied to the catwalk side of the panels when required by maintenance personnel. Some provision for cooling electrical motors or systems might have to be made.

For the best effect the panels should be made from Aluminium, or perhaps Zinc. A cheaper alternative would be to coat some other material with a reflective Aluminium or Zinc based paint. Panels made from other substances could be considered as almost any material could offer improvements on the present situation.

The main problem with the use of such panels is the hindrance to maintenance access which the panels might cause. Good organization and management would be required to ensure full benefit was derived from the panels since broken or removed panels would significantly reduce their effectiveness.

C5 At the Extrusion floor level there is also an imbalance between the amounts of ducted supply and extract ventilation. In this case supply exceeds extract mainly because of the addition of blower air. The majority of the excess air appears to escape up into the Hopper floor area via the large number of cracks and openings between the two floors. Better control of the amounts and directions of the air flows might reduce the total requirement for supply air and so reduce the imbalance. Such an improved control system is dealt with later.

C6 The main problem for the Hopper floor level appears to be the warm temperatures found during the summer period. This is caused by warm air and heat rising from the Extrusion floor below and by solar heat gains, through the large amount of glazing found particularly around the Type 14 area and through the rather lightweight structure of the upper wall and roof. Some improvements might be made by use of solar control glass or improved wall structure, though such remedies could be costly and awkward to install. If better control of the extrusion ventilation systems were achieved then this could offer improvements. Greater provision for use of natural ventilation at the Hopper floor level could be considered, though this would have to be managed appropriately so as to have the desired effects without causing other problems or discomfort.

C7 The environmental and duct flow monitoring carried out using the Hewlett-Packard Data Logging system showed that there were considerable energy flows associated with the air flows. Excluding the Thermex and hot water heat transfer systems, the major energy flows were the ducted air flows, and the air flows produced as a result of ducted flow imbalances. The temperature of the air exhausted into the external environment is insufficiently high to offer much prospect of heat recovery which would be useful to the factory. Most of the exhausts are physically spread out which might further reduce performance. Some heat pumps could provide heat reclamation but an air conditioning system making use of recirculated air may be the best option. The currently installed systems do use recirculated air particularly for Spin Doff areas winter operation.

The measurements made in the ducted air flows allowed the evaluation of the efficiency of the systems in removing heat. This evaluation was performed on a temperature and on a total heat basis, with particular reference to the Extrusion area.

Though the ventilation systems would have been designed according to recognized principles, the operation in the particular environment of the ICI Fibres factory means that warm air is not extracted at as high a temperature or heat content as might be possible. This must in part be due to the number of systems operating within each area which can adversely interact and the overall block approach to control (i.e. off or on).

Though en masse the systems can produce environments that are acceptable, the systems' operation, especially after shut-downs is not an exact affair. Even at the "acceptable" state, quite wide variations across the production areas are evident. This is not to suggest that the methods of plant operation found during the course of this study were incompetent, far from it, but rather that there has not existed the means to implement a reasoned control strategy. The initial step to such a strategy is proposed under section C9 of this appendix.

C8 The use of Nitrous Oxide tracer gas to examine air transfers and ventilation was described in the body of the thesis. It was shown that the spaces, or alleyways, between each bank of spinning machines, could be considered as relatively isolated from their neighbours, at the Spin Doff level.

The two parts (Type 8 and Type 14 areas) of the Spin Doff area were each air-conditioned as a whole; nonproducing machine as well as producing machine areas both being served alike. Since the spaces between the machines are relatively isolated, the need to air condition non-producing areas is in some doubt. The variations from place to place found with "normal" plant operation suggest that slight variations introduced by the halting of air conditioning to non-producing areas would be acceptable. The methods available for stopping the flow of conditioned air to selected areas of the Spin Doff floor are not easily utilized. In some cases this involves barring the duct run offs from inside the header chambers. It is proposed that a much more effective and efficient means of air conditioning the Spin Doff area would be to use a central, electronically operating control system.

C9 The first stage in the development of a centralized air conditioning and ventilation operation and control system, is that of demand prediction. This would entail the prediction of the required amounts of conditioned air in the Spin Doff area, and cooling air/hot air extract in the Extrusion area. The main influences on the prediction would be external climatic conditions and the proposed production schedule. Such a prediction could provide the information to calculate the number of fans required to meet demand within a given time period. The period used for any particular system would be determined by the ease with which the fans in the system could be switched on and off. This time period would generally be shorter for the smaller fans. Since the fans of each system are interconnected by header ducts or chambers in most cases, this would allow less than the full complement of fans to be operated to serve spread out areas of production.

Dampers would be required to be installed in ducts where not present at the moment and servo-mechanisms would be attached to all dampers.

A certain amount of fan capacity would have to be kept running as, what one might term, "spinning reserve". This would allow minute by minute or hour by hour variations in requirement to be met. Alternatively a number of controllable variable speed axial flow fans could be incorporated into critical systems. The use and operation of this extra fan capacity would allow fine tuning of the air conditioning and ventilation dependent on environmental parameters (temperature and humidity) measured within and across the Spin Doff and Extrusion areas. It would also be possible to incorporate a variable volume aspect to the air flows to and from each machine area by variations in the damper mechanism.

The operation of fans would have to be organized so that at any time a balance between supply and extract rates was maintained. The Spin Doff area ought to be operated at a slight positive pressure and the Extrusion area at a slight negative pressure. Such operation would allow scope for the maintenance of the small inter-floor pressure required for the steam conditioner tubes (described earlier in the thesis).

The obvious means for operation and control of the ventilation and air conditioning systems is that of a computerised system. This would need to be on a larger scale than the Hewlett Packard Data Logger since a larger number of sensors would be required to monitor conditions and the ability to power servo-mechanisms for the dampers and aspects of the fans and conditioning plant would be required.

C10 An option for the proposed layout is shown in Figure C1. At Extrusion level (First Floor) the supply ducts A and B should normally be operated with the extract duct. Additional cooling could be provided by the use of supply duct C which would produce jets of air to cool personnel on the walkway as required.

At the Spin Doff level (Ground Floor), better circulation and mixing could be achieved by increased use of supply duct B which serves the wind-up area of the machine. As previously suggested however, the amount of air conditioning for this area might be reduced.

Increased thermal insulation is also proposed around and between some of the ducts to prevent adverse heat transfers.

C11 The overall effect of these proposals should be to allow greater control and management of the environmental energy flows within the production areas. In this way more efficient plant operation with reduced costs should be achieved, whilst still maintaining suitable conditions for fibre production and relative human comfort.

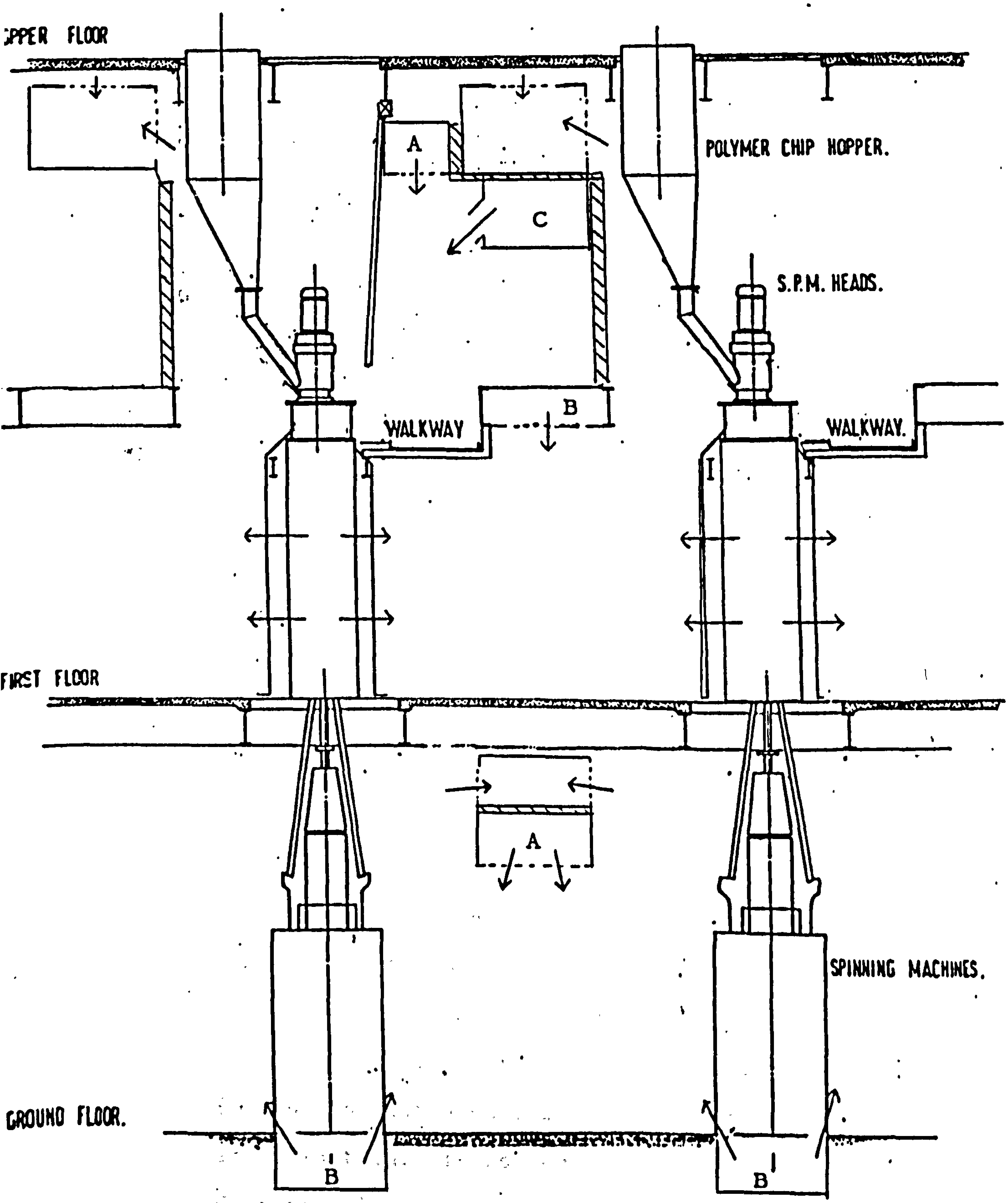

FIGURE C1 ILLUSTRATION OF PROPOSALS FOR I.C.I. FIBRES

▨ = THERMAL INSULATION

APPENDIX D

EXPERIMENTAL VARIABLES AND STATISTICAL EVALUATION

D1 In all experimental work, observations and measurements made are subject to experimental error. In order to make the effects of experimental error small, high quality materials and apparatus should be used with carefully controlled experimental conditions and the experiment should be repeated a number of times. Of course the facilities in which and with which to perform such ideal experiments rarely exist due to limitations of time, money, personnel, etc.

Industrial experimentation is further hampered as it may be impossible to control all the variables and the wide variation/range of inputs required for optimum results may not be feasible. In addition, if a continuous production process is involved, the quality of the product may be impaired due to the effects of the experiment and therefore the experiment is restricted.

D2 For experiments in which perhaps a relatively small number of measurements are made, or where repetition on a number of occasions is not possible, then we must consider the accuracy and significance of results obtained.

A variety of statistical techniques are available for use in such a context, and have been used in this investigation. It is necessary to pre-suppose that any deviation in a measured value of a variable from its true value is due to the effect of a large number of statistically independent influences or variables. The sum influence of the variables should have a Normal probability distribution. The greater the number of values measured then the smaller the range in which we can say the true value lies, with a certain degree of confidence. Thus, there is great virtue in being able to take substantial numbers of measurements.

In cases where only a small number of observations are possible (typically less than 30) it is usually more appropriate to make use of the t-distribution in statistical analyses. This has generally been the case throughout this study.

When it has been required to fit a line to sets of related data in graphical formats, correlation and regression techniques have been employed, using least sum of squares criteria.

D3 Considering first the model scale experiments performed in the laboratory, great care was taken in the setting up of measurement equipment and in such a controlled environment regular checking and calibration of performance was possible. During such checks no discernible bias was detected to the positive or negative indicating that deviations from true value were of a general nature, (i.e. supportive of a normal distribution).

The main items of equipment measured air flow rates and pressure differences and the calibration lines for these are given in the main text.

Instrument accuracy specified for the flow measurement was $\pm$ 0.002 ms^3/s; for pressure measurements in the model chamber tests $\pm$ 7%; and for pressure measurement in the wind tunnel $\pm$ 2%.

The results of the model scale tests were used to derive a relationship between pressure and square of flow rate using a least squares linear regression technique. The gradient of the line so derived, represented the resistance to air flow of partitions. The graphical representation of the results has been given in Figures 7.1, 7.2 and 7.5. In Tables D1 and D2 the correlation coefficients for the regression lines are given, this indicates the strength of the linear relationship. For the second set of results relating to the wind tunnel trials, the correlation coefficients were all very good and further statistical tests were performed to evaluate the 95% confidence intervals for the gradient of the line (i.e. 95% confidence intervals for the value of resistance). These results are presented in Table D3.

In Chapter 7 the results of the model scale tests were used to produce a predictive relationship relating the numbers and spacings of partitions to the resistance. Due to the small number of sets of results available no great confidence can be placed in the absolute values of these relationships (Section 7.3) but there is a statistically significant difference between each relationship.

D4 The experimental measurements made at full-scale in the factory environment had inherently greater inaccuracies, however careful checking and calibration was still practied as has been described in Chapter 8.

TABLE D1

MODEL SCALE TESTS

CORRELATION COEFFICIENTS FOR LEAST SQUARES LINEAR REGRESSION

(PRESSURE DIFFERENCE - SQUARE OF FLOW RATE)

TEST	CORRELATION COEFFICIENT	TEST	CORRELATION COEFFICIENT
21	0.971	40	0.994
22	0.977	41	0.989
23	0.983	42	0.985
24	0.996	43	0.981
25	0.993	44	0.972
26	0.982	45	0.976
27	0.989	46	0.978
28	0.991	47	0.991
29	0.995	48	0.988
30	0.997	49	0.987
31	0.995	50	0.964
32	0.991	51	0.970
33	0.986	52	0.969
34	0.968	53	0.978
35	0.981	54	0.988
36	0.992	55	0.993
37	0.976	56	0.981
38	0.994	57	0.995
39	0.986	58	0.979

TABLE D2

"WIND TUNNEL" TRIALS

CORRELATION COEFFICIENTS FOR LEAST SQUARES LINEAR REGRESSION

(PRESSURE DIFFERENCE - SQUARE OF FLOW RATE)

TEST	CORRELATION COEFFICIENT	TEST	CORRELATION COEFFICIENT
6	0.9975	25 + 25a	0.999
7	0.9998	26 + 26a	0.9937
8	0.9996	27 + 27a	0.9999
9	0.9996	28 + 28a	0.9998
10	0.9997	29 + 29a	0.9998
11 + 11a	0.9994	30 + 30a	0.9999
12 + 12a	0.9997	31 + 31a	0.9999
13 + 13a	0.999	32 + 32a	0.9993
14 + 14a	0.9987	33 + 33a	0.9991
15 + 15a	0.9995	34 + 34a	0.9997
16 + 16a	0.9999		
17 + 17a	0.9985		
18 + 18a	0.9997		
19 + 19a	0.999		
20 + 20a	0.9988		
21 + 21a	0.9982		
22 + 22a	0.9989		
23 + 23a	0.9999		
24 + 24a	0.9966		

TABLE D3

95% CONFIDENCE INTERVALS FOR VALUES OF RESISTANCE DETERMINTED IN WIND TUNNEL TESTS

TRIAL	RESISTANCE
6	37.4 ± 0.18
7	32.2 ± 0.47
8	0.6 ± 0.028
9 + 10	36.0 ± 0.07
11 + 11a	31.9 ± 0.6
12 + 12a	32.4 ± 0.67
13 + 13a	28.8 ± 0.66
14 + 14a	24.9 ± 0.68
15 + 15a	26.7 ± 0.44
16 + 16a	35.0 ± 0.35
17 + 17a	25.0 ± 1.0
18 + 18a	30.7 ± 0.42
19 + 19a	28.7 ± 0.21
20 + 20a	27.5 ± 0.7
21 + 21a	27.6 ± 0.86
22 + 22a	26.1 ± 0.62
23 + 23a	22.9 ± 0.16
24 + 24a	24.7 ± 1.11
25 + 25a	23.5 ± 0.55
26 + 26a	22.61 ± 1.37
27 + 27a	34.4 ± 0.29

TRIAL	RESISTANCE
28 + 28a	34.5 $\pm$ 0.398
29 + 29a	29.5 $\pm$ 0.33
30 + 30a	31.2 $\pm$ 0.3
31 + 31a	26.2 $\pm$ 0.28
32 + 32a	25.2 $\pm$ 0.48
33 + 33a	26.4 $\pm$ 0.57
34 + 34a	38.0 $\pm$ 0.46

Measurement of temperature by platinum resistance thermometer had a basic accuracy of $\pm$ 0.1°C with a specified drift of less than 0.05°C at temperatures less than 500°C. The humidity sensor had a nominal accuracy of $\pm$ 2% which compares very favourably with all other normally used measurement systems. The vane anemometer has a specified accuracy of $\pm$ 5% and in the calibration of vane and circuits the correlation coefficient was very high (see Chapter 4). Of course the use of the van anemometers in the ducts led to some variability (already described) which was unavoidable.

The evaluation of the energy flows which was eventually presented in diagramatic form (Figures 8.7 - 8.10) encompassed a number of variable factors giving an indication of a range in values as high as $\pm$ 20% and therefore the figures quoted should be taken as typical (derived by averaging) rather than definitive. However, since these figures represent the first real step to measure such energy flows their usefulness is not to be underestimated.

D5 The values determined for the temperature and total thermal efficiencies of Chapter 8 were averaged for summer and winter periods, and exhibited a number of variations. However, at the 95% confidence level, real differences between the seasons and between the types of efficiency were established. The 95% confidence intervals are given below for the extrusion areas.

Winter

Temperature Efficiency	99.23	± 8.36%
Total Thermal Efficiency	87.58	± 4.26%

Summer

Temperature Efficiency	87.83	± 8.98%
Total Thermal Efficiency	65.42	± 14.68%

D6 In the experiments performed using a tracer gas, several methods were used to evaluate the ventilation rate. Substantial variations were found using the tracer gas decay technique which led to its results being disregarded. The measurement of the air change due to duct flow gave a ventilation rate of 4.5 m^3/s for the space under consideration. The 95% confidence interval being 4.2 - 4.8m^3/s.

The ventilation rate derived by the equilibrium concentration of tracer gas method gave a 95% confidence interval of

7.3 ± 0.85 m^3/s for normal plant operation

2.0 ± 0.25 m^3/s for background ventilation

and 4.62 ± 0.05 m^2/s for the ventilation rate using the additional fans.