SUCTION NOZZLE FEEDING MECHANISMS AND THEIR
EFFECT ON THE PERFORMANCE OF VACUUM
PNEUMATIC CONVEYING SYSTEMS

ANDREW NIGEL PITTMAN

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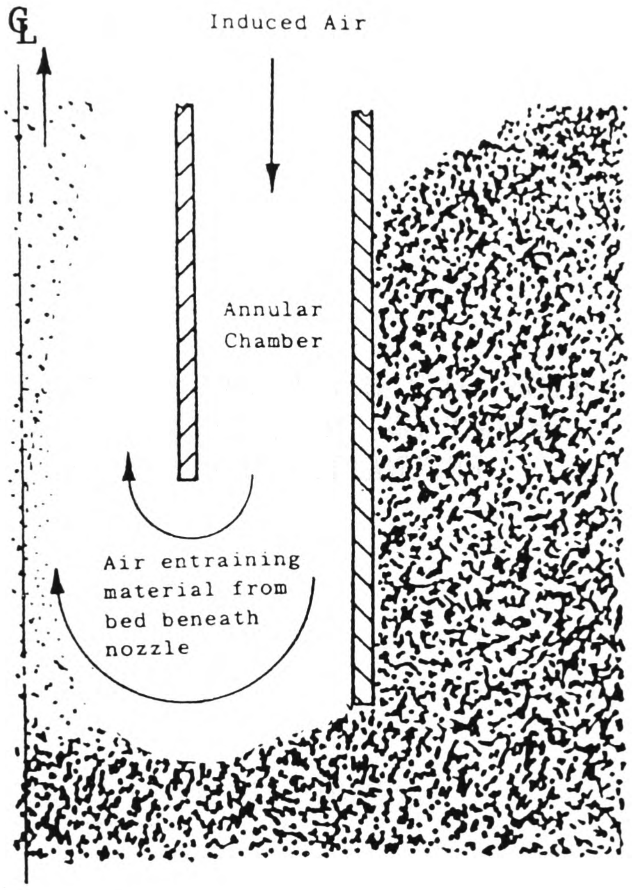

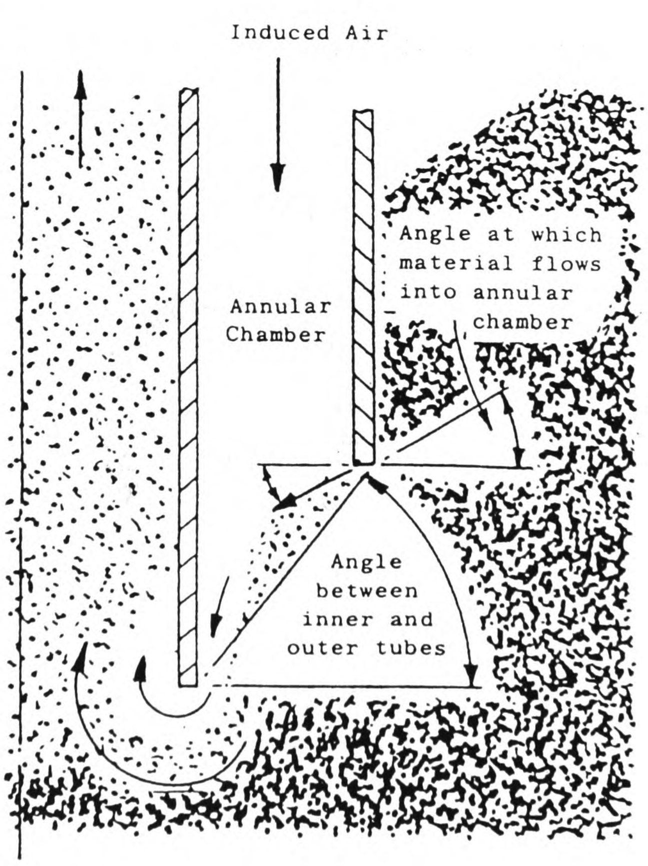

Entrainment Mechanisms for Different Nozzle Geometries

ABSTRACT

SUCTION NOZZLE FEEDING MECHANISMS AND THEIR EFFECT ON THE PERFORMANCE OF VACUUM PNEUMATIC CONVEYING SYSTEMS

by

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The pneumatic conveying of bulk solid materials by the generation of a partial vacuum or negative pressure is the oldest form of conveying such materials through pipelines. It was developed approximately one hundred years ago to assist the rapid unloading of wheat grain from ships, barges and lighters in the London docks. However, the basic design of the device for entraining the bulk solid into the conveying pipeline has changed very little since those times.

This thesis examines the effect of entrainment mechanisms in co-axial tube type suction nozzles on the performance of such conveying systems. The normal design of such nozzles is one where the inner conveying tube is retracted inside the outer tube. The work has demonstrated that with this configuration, high air velocities are necessary to effect entrainment of material beneath the nozzle into the conveying pipeline. By rearranging the nozzle so that the inner tube protrudes from the outer tube, a configuration can be obtained where, for free flowing products, the material flows readily under the action of gravity into the entraining airstream flowing down the annular chamber formed by the two tubes. The work has demonstrated that, for a given air flow rate, the product feed rate, and hence system throughput will be maximised by arranging these tubes such that the angle formed by them in the entrainment region is greater than the angle at which the product flows under the action of gravity into this region. Since a lower air flow rate and hence power consumption is required to achieve a given system throughput, this configuration has been identified as a more efficient entrainment mechanism than that associated with the usual configuration of suction nozzles.

The effect of throttling the air flowing down the annular chamber on product entrainment into such nozzles is also identified and presented.

ACKNOWLEDGEMENTS

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AUTHOR'S NOTE

This thesis is the original work of the Author throughout, except where stated otherwise by reference or by acknowledgement.

NOMENCLATURE

Symbol/ Notation	Meaning	Units
d	Internal Diameter of Inner Conveying Pipe	(m)
D	Outer Sleeve Diameter	(m)
F_o	Cross-Sectional Area of Inner Pipe	(cm^2)
h	Height of Flow Area Beneath Nozzle	(m)
L	Unloading Rate	(t/hr)
$\dot{m}_a$	Mass Flow Rate of Air	(kg/s)
$\dot{m}_p$	Mass Flow Rate of Solids	(kg/s)
Patm	Atmospheric Pressure	(N/m^2)
SF	System Factor	(Ns m^{-3})
SPC	Specific Power Consumption	(W/t/hr)
V_{ao}	Superficial Air Velocity	(m/s)
X	Nozzle Extension	(m)
α	Poured Angle of Repose	(Deg)
β	Drained Angle of Repose	(Deg)
η	Nozzle Constant	(dimensionless)
θ	Nozzle Angle	(Deg)
μ	Permeability Factor	(m^4 Ns^{-1})
ϕ	Solids Loading Ratio	(dimensionless)
ψ	Nozzle Loading	($kg/s/cm^2$)

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6. DISCUSSION and CONCLUSIONS

APPENDICES

CHAPTER 1. OUTLINE OF STUDY

1.1 Introduction

The movement of powders and granular solid materials through pipelines using air or another gas as the carrier medium is a common form of conveying such materials. This facet of bulk handling is normally known as pneumatic conveying. Since this technique is clean, flexible and lends itself readily to automation, it is employed in a wide variety of industries throughout the world. All such systems, whether they operate on a positive pressure or negative pressure basis, require a device by which product can be fed into the conveying pipeline at some controlled rate. Positive pressure systems have the requirement of feeding the product into the conveying pipeline which is operating with an air pressure greater than that of the surrounding atmosphere. As a consequence, there is a tendency for the conveying medium to leak through the feeding device. The most common devices used to feed such systems on a continuous basis are rotary valves, venturi eductors and screw feeders. On the other hand, negative pressure systems feed products into a conveying pipeline where the pressure of the conveying medium is below that of the surrounding atmosphere. Therefore, any air leakages are in the same direction as the product flow. Rotary valves are used in many fixed applications but since a vacuum conveying system is, in effect, like a domestic vacuum cleaner, many systems are mobile and are used for entraining products from vessels, heaps and stockpiles. Such large scale systems

are used extensively for the rapid unloading and trans-shipment of cargoes from the holds of ocean going bulk carriers, using a suction nozzle as the feeding device. These systems range in size from 40 t/hr through conveying pipelines of 150mm (6 inch) diameter up to 500 t/hr in 600mm (24 inch) diameter pipes.
Although basic vacuum conveying systems vary little, the suction nozzle can take many forms. The most widely used type of nozzle is that comprising of two co-axially mounted pipes. As its name implies, this type of suction nozzle consists of two tubes, with the inner tube being nominally the same diameter as the conveying pipeline and the outer tube being of some larger size. When mounted co-axially an annular chamber is formed between them. The British Standard definition of such a nozzle is reproduced in Appendix A. The geometry of the two tubes, that is, the linear position of the tubes relative to one another is usually adjustable, as is the opening through which atmospheric air is induced to flow down the annular chamber to the entrainment region. It is clear that the design of the nozzle can have a significant bearing on the overall performance of the system as a whole, since the conveying system downstream will only convey the product that is presented to it by the feeding device, ie. the nozzle. If the design is such that the product cannot be entrained effectively it is evident that the system will operate inefficiently. Conversely, if too much product is presented to the conveyor, unreliable conveying or pipeline blockages might occur.

1.2 Industrial Interests

As indicated in the previous section, pneumatic conveying is used in a wide variety of industrial applications. With many of these applications conveying rates are dictated by the production process rates. The production or utilisation of powdered or granular solids in many in-plant processes is in the range 2 - 20 t/hr. Therefore, although improving the feeding efficiency of such conveying systems is beneficial, the overall improvement is likely to be small. The real savings to be made with such conveying operations relates to higher throughput applications such as in unloading ships and barges etc. With such applications system throughputs are not dictated by process considerations but rather by the desire to unload bulk cargoes as quickly as possible so as to avoid excessive demurrage charges. Consequently, such systems operate with unloading rates in the range 50 - 500 t/hr. Given that the way in which materials are introduced into the vacuum conveying system will effect the overall performance and efficiency of the system, it is clear that optimising the performance of the suction nozzle is of considerable industrial interest. However, it would appear that prior to the study reported in this thesis, there is little data providing an insight into the operation of such suction nozzles and their effect on system performance.

1.3 Objectives of Research

Based on the foregoing comments, the object of the programme of research reported in this thesis, is to understand the mechanisms by which co-axial tube suction nozzles entrain products into the pneumatic conveying pipelines and understand their effect on system performance, with a view to establishing guidelines for the optimum design for such devices. Although this programme is addressed primarily from the standpoint of pneumatic ship unloading systems, it is clear that the implications of the research will be applicable to smaller scale systems that incorporate such nozzles.

1.4 Synopsis of the Thesis

Chapter 2 reviews both past and current industrial practices and discusses the implications of previous research work in relation to the present study. Chapter 3 outlines the experimental rigs constructed for this research and the test work undertaken. Chapter 4. presents the findings of a series of flow visualisation tests using a rig specifically constructed for this purpose. Chapter 5 presents the findings of a series of tests using a pilot scale test rig. Chapter 6 discusses the findings of the test programmes in relation to the performance of vacuum conveying systems as a whole. From this, guidelines for the design and operation of such suction nozzles are proposed. This chapter includes conclusions drawn from this study and recommendations for further work.

CHAPTER 2. A REVIEW OF SHIP UNLOADING PRACTICES AND PREVIOUS WORK ON SUCTION NOZZLES

2.1 Introduction

As inferred in the previous chapter improvements in the design and operation of suction nozzles as a consequence of developing a better understanding of their operation will have a more appreciable effect in pneumatic ship unloading applications due to the sheer scale of such systems. Although this study focuses primarily on ship unloading, the results will be applicable to any size of conveying system employing such nozzles. Since the work focuses on ship/barge unloading applications, this chapter reviews the practices that have been employed over the last hundred years in this area, from both the historical and technical perspectives, as relevant to this study.

2.2 The Origins of Vacuum Pneumatic Conveying

The pneumatic conveying of powders and granular materials in vertical, horizontal and inclined pipes using air as a conveying medium is by no means a recent development. Pneumatic conveying and, in particular, vacuum conveying has been in existence, as far as the author knows, for approximately 100 years. It is not known who first thought of using air as a conveying medium in an industrial context, but a paper presented by Mitchell (Ref 1) in 1914 states that attempts were made to convey grain by such means as far back as 1865. However, these early attempts failed, albeit over very short vertical distances of 1 to 2 metres. Other attempts included passing the grain into the centre of a fan whereupon the grains received an initial impulse from the blades and were then conveyed along the pipeline by the air stream created by the fan. This attempt also failed because, not surprisingly, the grain rapidly degraded into flour. Another attempt included evacuating a large vessel, to which was attached an isolating valve and a length of conveying pipeline. Once the vessel had been evacuated to the desired level of vacuum, the isolating valve would be opened causing a sudden in-rush of air and, with it, the grain. Although the idea worked, it was not a practical solution since the operation was intermittent and only small quantities of grain were conveyed with each cycle. Another problem faced by all of the early attempts at conveying by vacuum, was of emptying the grain from the vacuum receiving vessel. This meant that the vessels had to be emptied after each batch of grain had been conveyed, thereby making the whole process very slow. Mitchell was personally involved with some initial tests in 1894, using high pressure air driven ejectors to generate the operating vacuum, but with very little success due to the large volumes of high pressure air required. It was Duckham (Ref 2) who in 1892 realised the potential and the advantages of using vacuum conveying for the unloading of barges, lighters and steamers. He successfully designed a pneumatic grain elevator which had none of the problems associated with the earlier attempts described above. These elevators were used in the Millwall Docks, London from 1892 to meet the ever increasing demand for wheat to feed the rising population of London. According to the figures outlined in Ref 2, in 1893, approximately 1,450,000 tonnes of grain were imported into the United Kingdom; of which 362,500 tonnes were off-loaded in London, with 181,250 tonnes being off-loaded in the Millwall docks alone. In a paper by Duckham (Ref 3) he refers to the average annual grain imports being 6,157,276 tons for the ten year period ending 1890. Despite the apparent discrepancies in these figures, it can be seen that there was a need to off-load vessels at a much greater rate than had been possible with previous methods of off-loading. In this context the drawing shown in Fig 2.1 shows the type of system in operation around the time of Duckham, which may be compared with a modern day system for unloading Alumina in Fig 2.2.

2.3 Why Use Pneumatic Grain Unloaders ?

Prior to the employment of Duckham's pneumatic unloaders in the Millwall Docks, the Priestman's grab or Bucket grab was used extensively, whilst in the Liverpool Docks the band and bucket (nowadays referred to as a Bucket Elevator) was used. However, even in the 1890's it was already recognised that all of the then current mechanical unloading systems available had problems as mentioned in Refs 2 and 5. These problems being :-

- that the grain had to be "trimmed" or moved to the grabs or bucket elevators,
- grain stored in awkward places could not be reached by the equipment,
- many hatches had to be opened for access, thus inclement weather often prevented unloading operations,
- loss of revenue due to grain being spilt from grabs,
- large quantities of dust being generated within the hold creating an unpleasant working environment, and
- mechanical failure and damage to equipment.

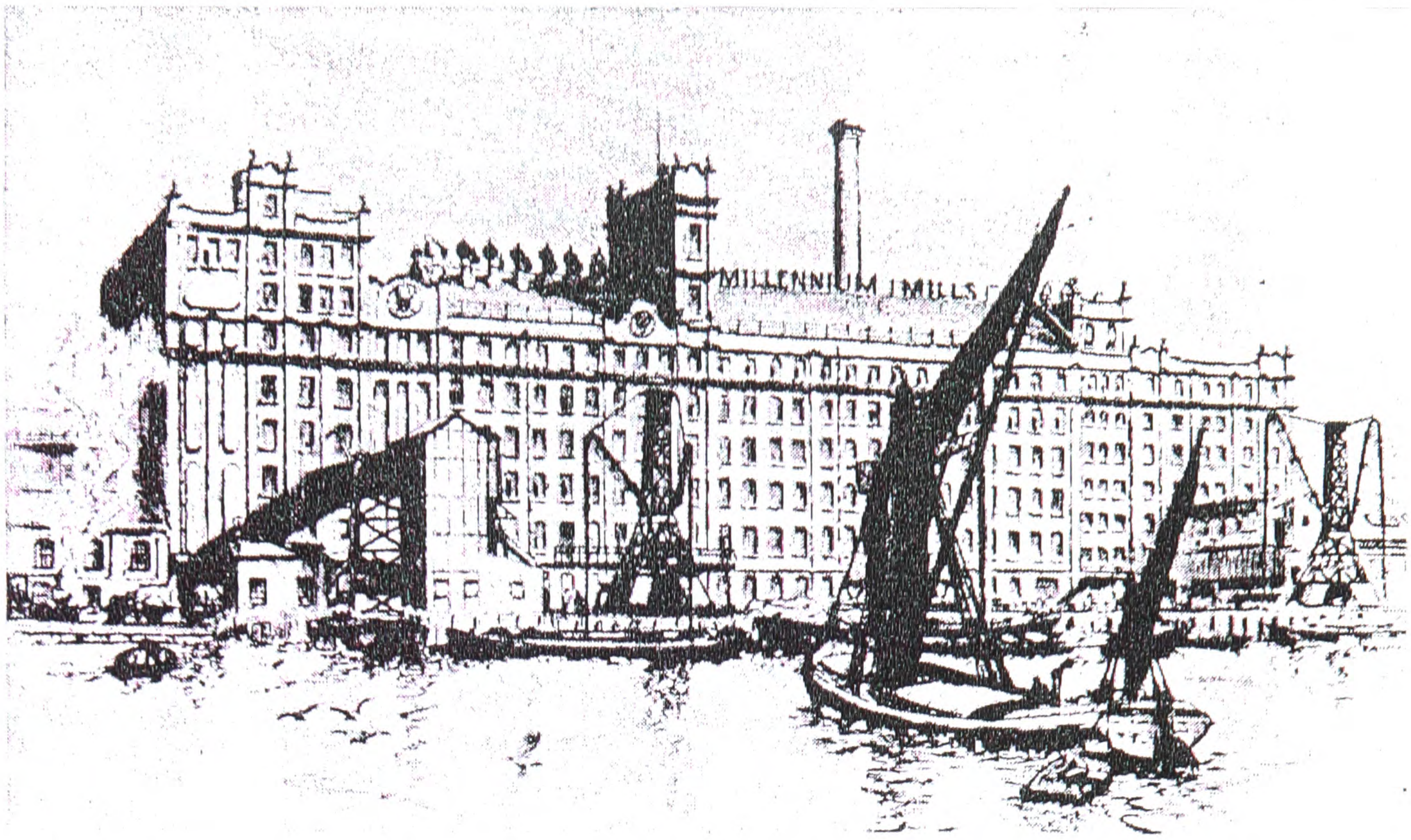


Fig. 2.1 A Typical Scene at The Royal Victoria Dock, London, Clearly Showing Pneumatic Ship Unloading Systems, Circa 1890.

Fig. 2.2 A Present Day Ship Unloader, Holyhead, Anglesey, North-West Wales, Circa 1990.

It is clear from these points that a much simpler and more reliable method for discharging the cargoes from the holds of ships was required and that Duckham's system went a long way to overcoming these problems. The pneumatic grain elevator, designed by Duckham, again came into its own on the River Danube in the mid 1890's (Ref 5), because it was found to be the quickest and most practical method for unloading barges carrying grain for a number of reasons. Due to the navigational problems faced by vessels on the Danube, they were loaded upstream and then immediately lightened prior to travelling downstream to the mouth of the river. Thus, all of the grain removed from the holds to lighten the vessel was placed into barges which followed the vessel downstream. Once the vessel was in deep water the grain would be re-loaded. This meant that some of the grain was handled twice, with the amount varying from day to day according to the prevailing river conditions. Originally all of the loading and lightening operations were undertaken by manual means with men hand filling sacks and then climbing up ladders out of the holds with the sacks on their backs. The sacks were then emptied over the ship's side into barges moored alongside. An attempt was made to mechanise this work by using grabs and buckets but it was found to be too dangerous on the fast flowing river. A considerable amount of manual work was still required to sweep up the grain left in the holds after the grabs had finished their duties. Thus, Duckham's pneumatic grain elevator met every requirement for this particular application and only required men to raise, lower and swing the conveying pipes in the holds.

Ocean going bulk carriers often carry more than one commodity and it is not uncommon to find more than one type in a single hold; with the commodities being separated by simply laying mats on top of one, prior to the next being loaded. During the voyage the mats often "work" thus making it very difficult for a mechanical unloader to operate since there is a risk that the mechanism may become jammed by a mat which had moved to an unexpected position.

2.4 Early Vacuum Pneumatic Grain Elevators

Although there have been many technological advances, for example, in the areas of materials of construction, conveying techniques and air mover designs, the principles upon which the pneumatic elevators work have remained largely unchanged since the 1890's.
Such a pneumatic conveying system comprises of six basic components :-

i) The feeding device to ensure that the required amount of product is supplied to the conveying pipeline

ii) The conveying pipeline through which the product is conveyed to the receiving vessel

iii) A receiving vessel to collect the product

iv) A device for discharging the product from the vacuum receiving vessel whilst maintaining an air seal

v) A filtration system to prevent carry over of dust being drawn into the air mover

vi) An air mover to generate the operating vacuum and hence induce an air flow into the conveying system,

A schematic diagram showing the relative disposition of these components is shown in Figure 2.3. The line diagram of Fig 2.4 shows one of Duckham's early designs. Fig 2.5 shows a present day system, and by comparing the two it may be seen that there is very little difference between them.

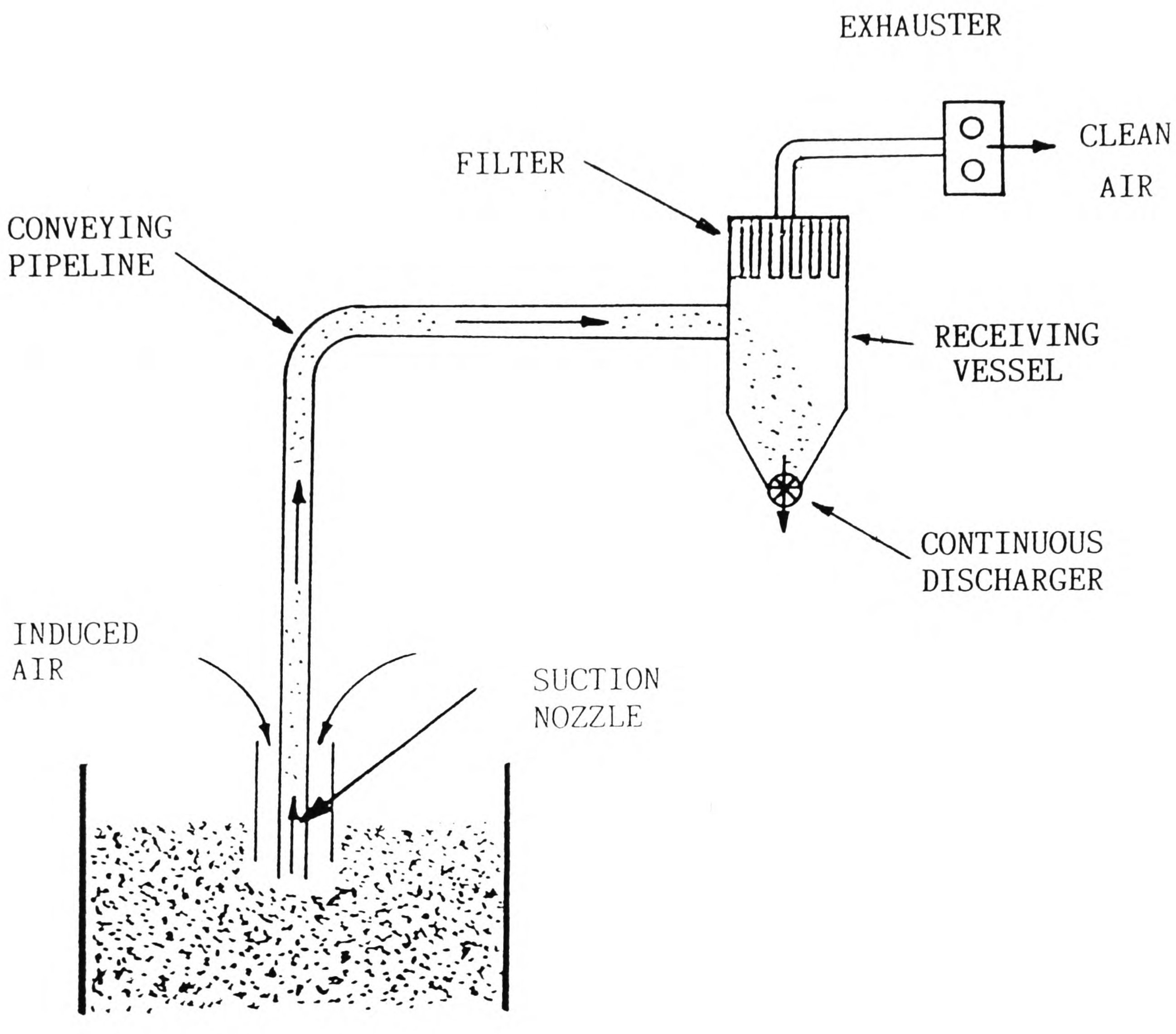


Fig. 2.3 Schematic Diagram of a Vacuum Pneumatic Conveying System as Used for Typical Ship Unloading Applications

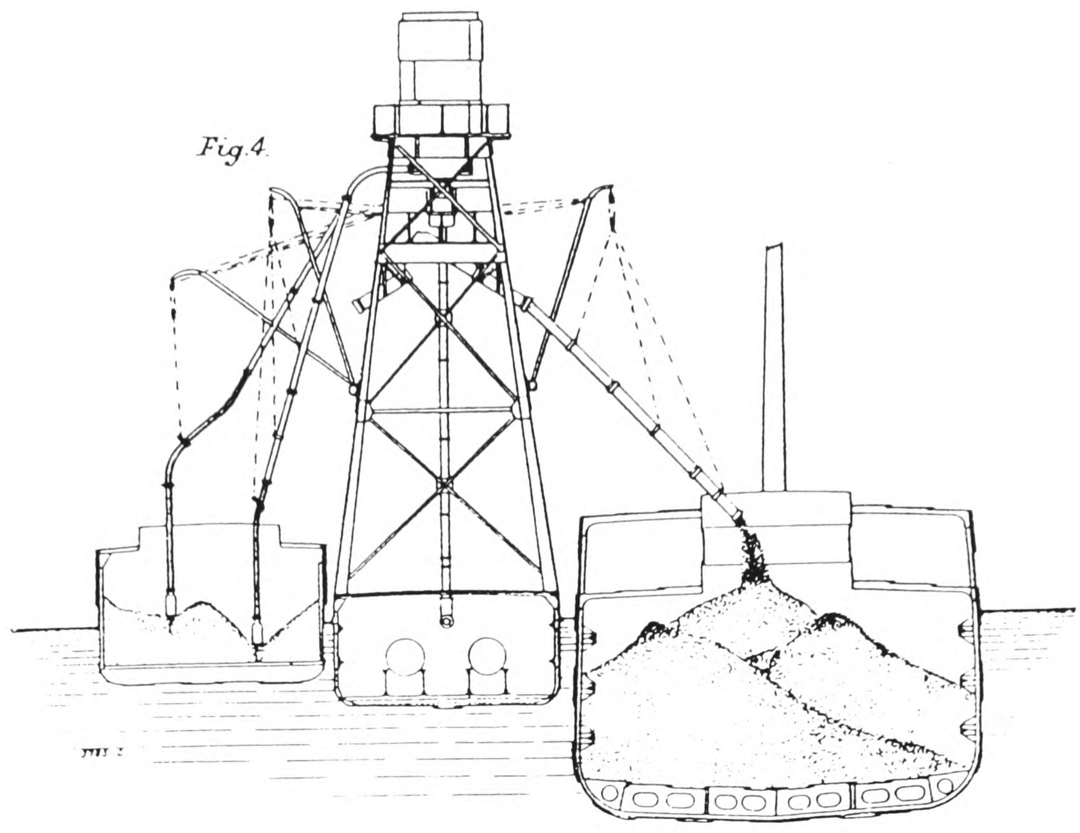


Fig. 2.4 Line Drawing of One of Duckham's Early Ship Unloaders

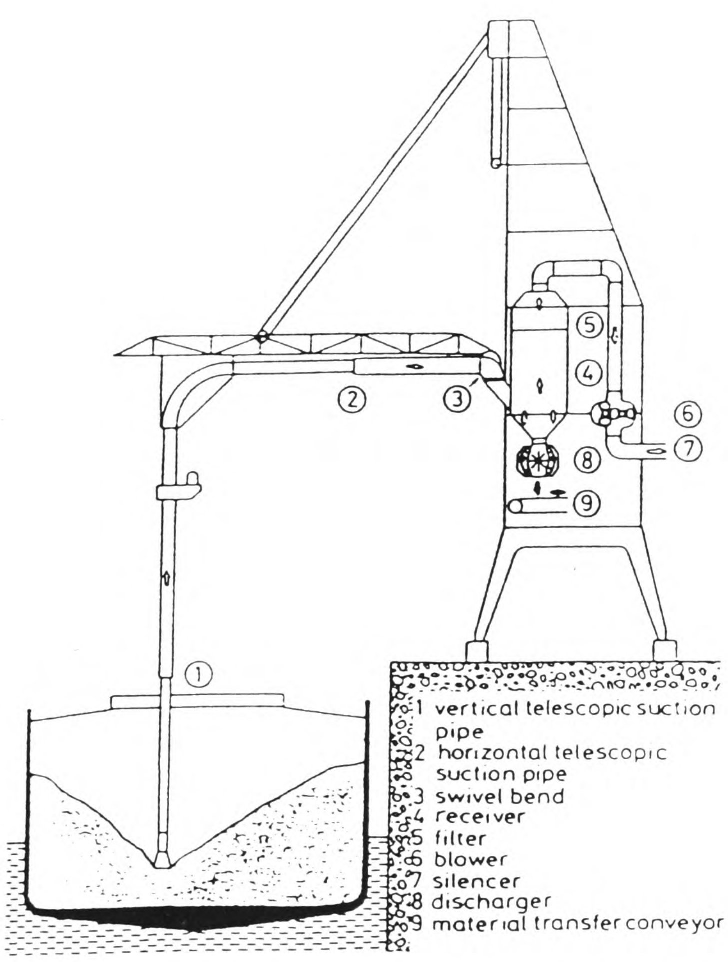


Fig. 2.5 Key Components of a Pneumatic Ship Unloader

2.4.1. Suction Nozzle Feeder

This component is used to control and maintain a feed of product into the pipeline. This was one of the key features that Duckham included in his design. He realised, at an early stage in the development of his design, that a moving airstream was more important than the magnitude of the vacuum present in the system; and that a constant supply of air would have to be admitted into the pipeline if the system was to function reliably in conveying the grain along its entire length without choking. In this context the term "choking" refers to a situation where, should too much product enter the vertical section of the conveying pipeline, the sudden increase in pressure is sufficient to stall the air mover, thereby causing the air velocity in the pipeline to fall to zero. This in turn causes the product to suddenly stop conveying and drop out of suspension, resulting in choking of the pipeline.

Figures 2.6, 2.7 and 2.8 show three basic designs of feeding device. The first is the "standard" co-axial tube design used for the 'main sinking' operations and operates along the lines outlined in the previous chapter. The second type is simply a variation of the standard suction nozzle design allowing the conveying pipeline to be taken away at an incline instead of vertically. The third type is commonly referred to as the "camel back" type and is often used for reaching under comings, side decks or other awkward places where the standard type cannot reach. Although the "camel back" type of nozzle performs the same basic task as the standard type of suction nozzle, described in the previous chapter, it operates in a somewhat different manner. This is because the auxiliary air inlet allows atmospheric air to be drawn directly into the pipeline, effectively by-passing the entrainment region of the nozzle.

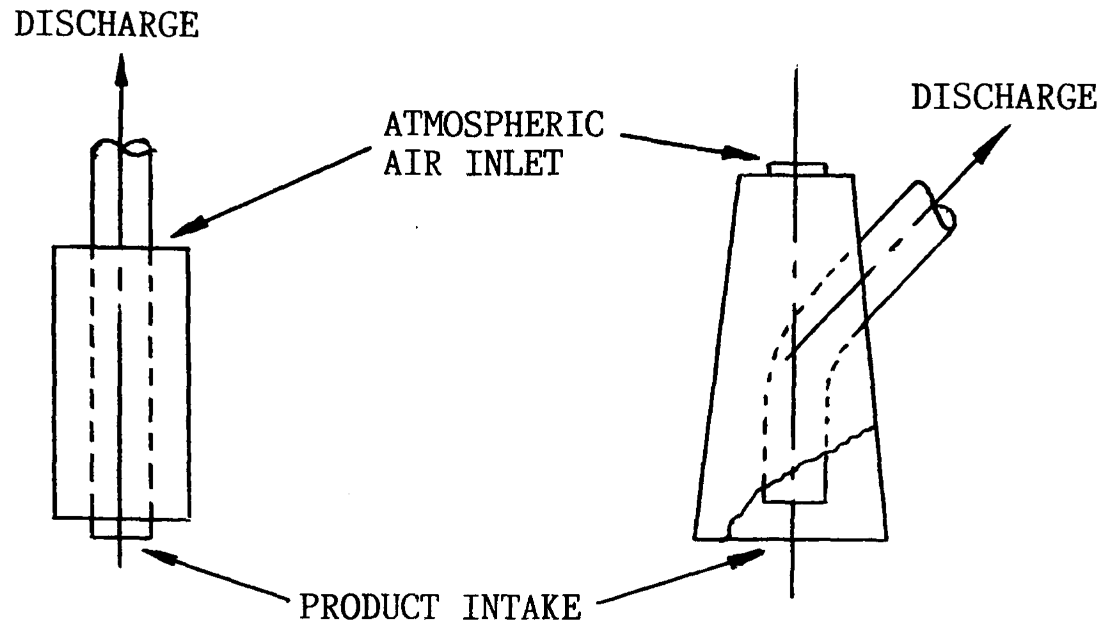


Fig. 2.6
"Standard" Co-axial Tube Type Suction Nozzle

Fig. 2.7
Variant of The Standard Co-axial Tube Type Suction Nozzle (Ref 3)

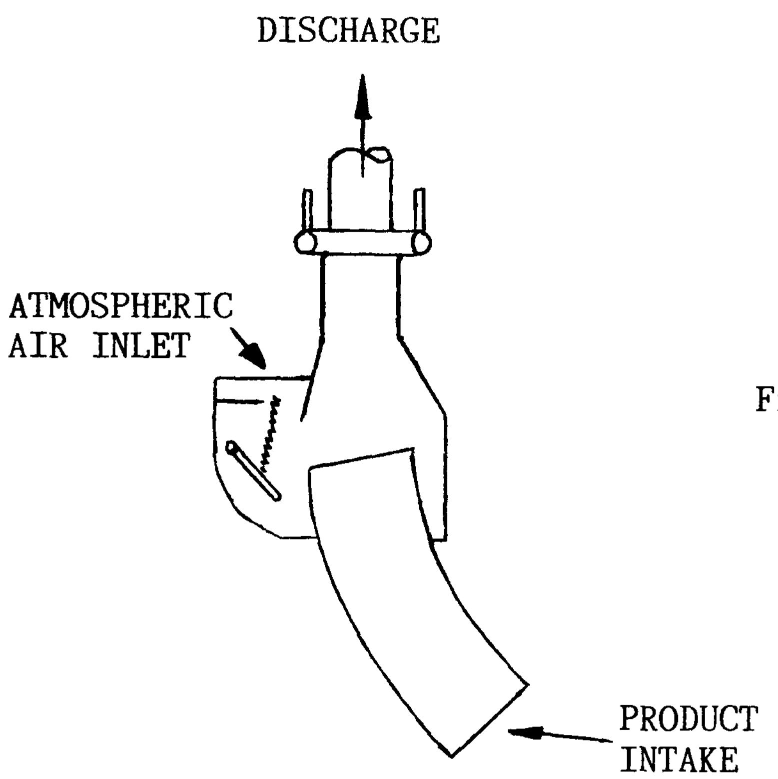


Fig. 2.8 "Camel Back" Type of Suction Nozzle

2.4.2. Conveying Pipeline

To allow the suction nozzle to move within the hold the conveying pipeline consists of rigid and flexible piping. Duckham's early flexible pipes were made of steel cones joined together with the tip of one cone just inserted within the mouth of the next cone and so on along its entire length. The cones were joined together by tapes and then sheathed in fabric to make them air tight. To enable the suction nozzle to be raised and lowered into the hold or be moved about the hold, the rigid sections of steel pipe were telescopic to allow the nozzle to follow the free surface of the material and also to take into account the varying draught of the vessel and tidal effects. The telescopic feature is still an integral part of modern day unloading systems. The joints between the individual sections are either swivel joints, as shown in Figure 2.9 or bends with "flat back" replaceable elements, as shown in Figure 2.10. The only developments associated with this part of the system have been in the materials of construction of the rigid pipes, the replaceable wear backs of bends, the seals within the telescopic joints and the flexible pipes. The flexible pipes described above have now been superseded by various rubber hoses which can withstand the sliding wear associated with the particles moving along the inner surface of the hoses.

2.4.3. Receiving Vessel

This is, simply, a vessel which is designed and constructed to withstand the difference in pressure between the partial vacuum in the receiver and the surrounding atmospheric pressure. As the air and product enter the vessel from the conveying line, the air suddenly expands upon entry causing a reduction in the air velocity. The heavier particles no longer remain in suspension and settle to the bottom of the vessel under the action of gravity, whereupon they are

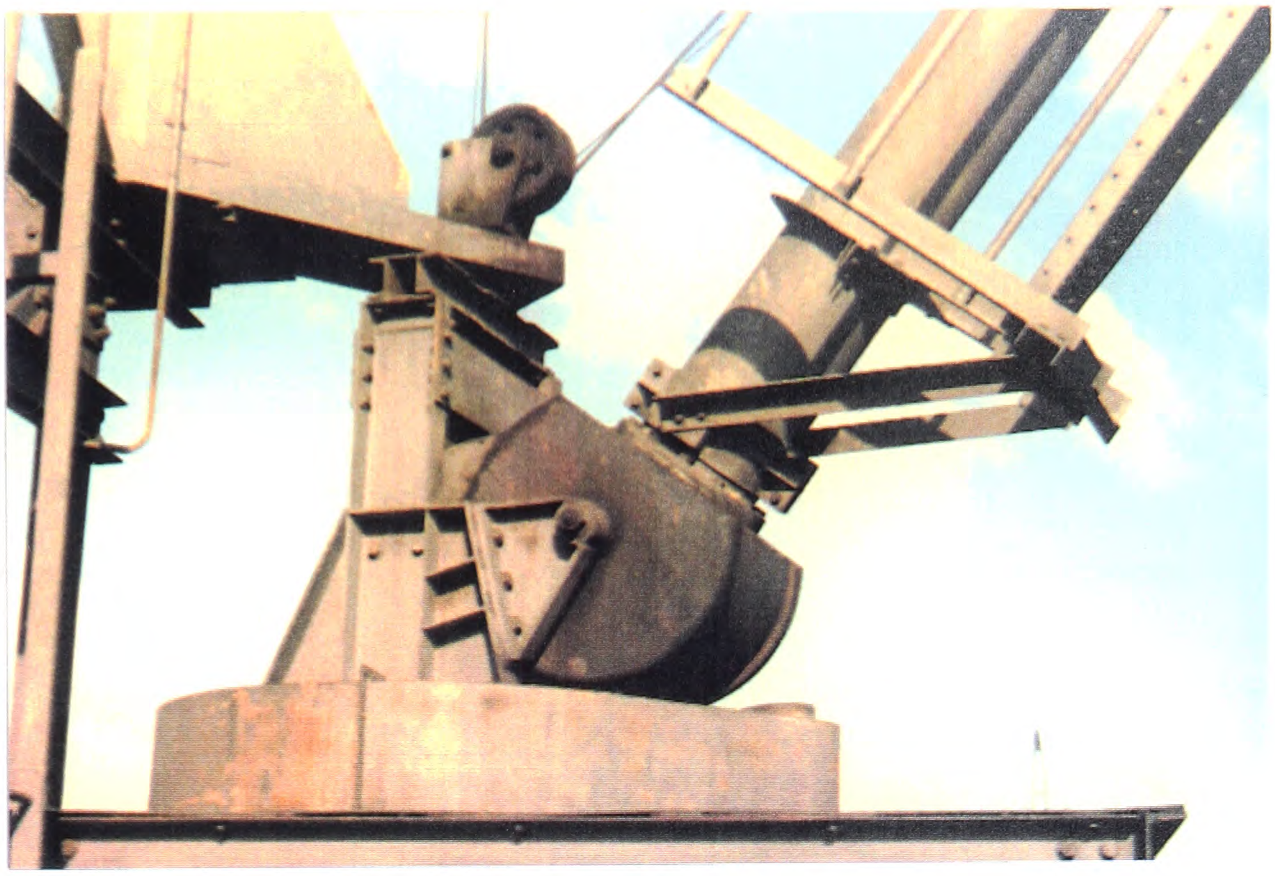


Fig. 2.9 A Typical Swivel Joint

Fig. 2.10 "Flat Back" Bends

discharged. The dust, being lighter, may stay in suspension and can be carried upwards by the airstream towards the filter. The author is not aware of any significant developments associated with this part of the system.

2.4.4. Discharging Device

Duckham devised a way of continuously discharging the grain from the vacuum receiving vessel whilst maintaining an air tight seal. This, together with the constant feeding device, were the two components which helped to overcome the failures associated with earlier attempts at pneumatic conveying by vacuum. As a consequence Duckham's developments made the approach truly continuous in operation. Thus, the pneumatic elevator has remained a practical, flexible and viable solution for the rapid unloading of grain and other bulk commodities, to the present day.

The continuous discharger of Figure 2.11 shows the development by Duckham in this area. It consists of a pair of symmetrical boxes mounted back to back on a single shaft. The boxes rock back and forth on a shaft being driven by a connecting rod attached to an eccentric mounted on the shaft of an electric motor. The top of the boxes are curved to mate up with the bottom of the receiving vessel, thus providing an air tight seal. This prevents atmospheric air from entering the vessel as they rock back and forth. As they do so, each box passes under the outlet, filling with grain and then discharging the grain on the return stroke. If one were to join several of these boxes together about a central rotating shaft it would result in the well known rotary valve feeder. The rotary valve, Figure 2.12 has been the single major development in the area of continuously discharging products from vacuum receiving vessels. Alternatives have

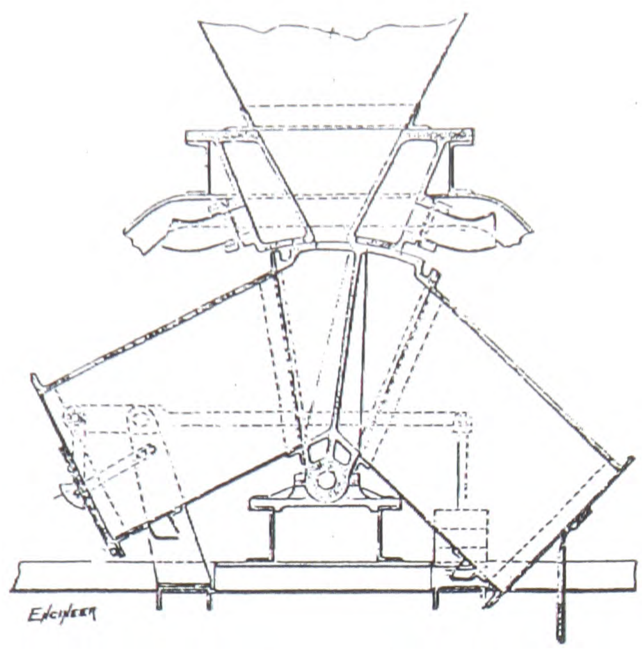


Fig. 2.11 An Early Design of a "Tipper Box"

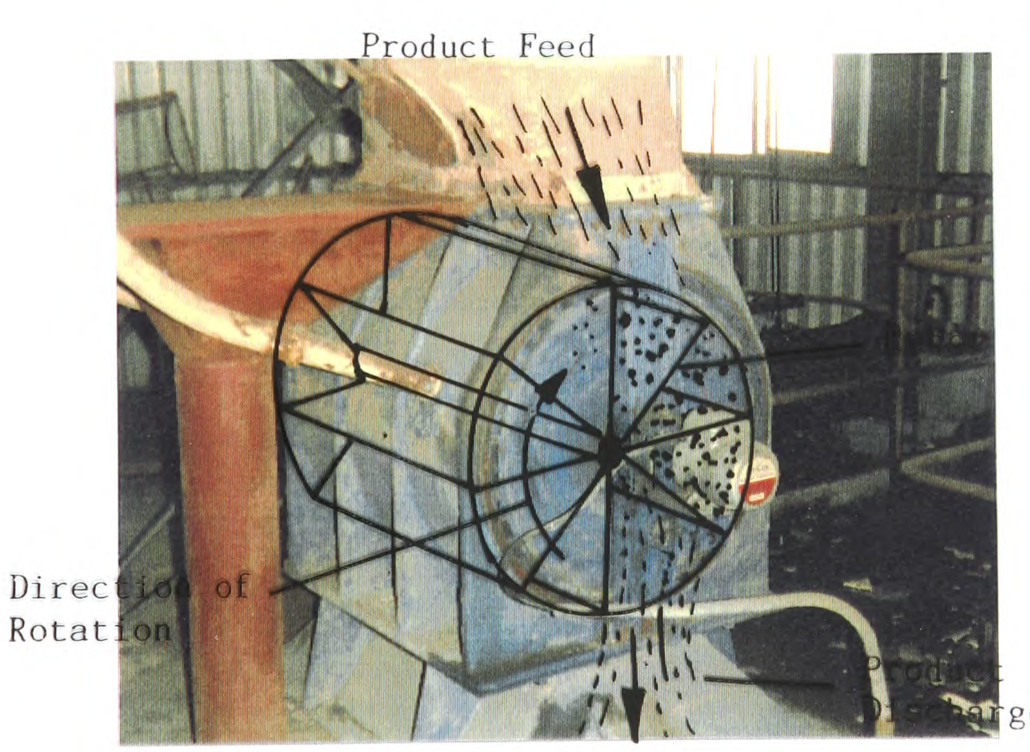
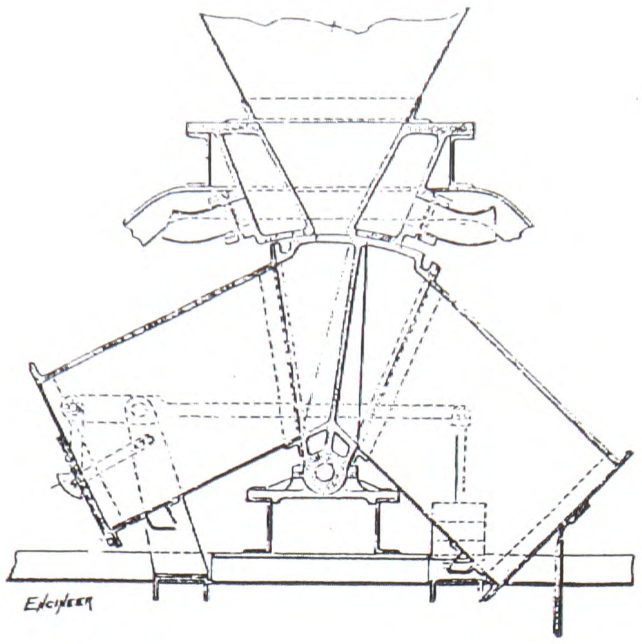


Fig. 2.12 A Typical Rotary Valve Discharger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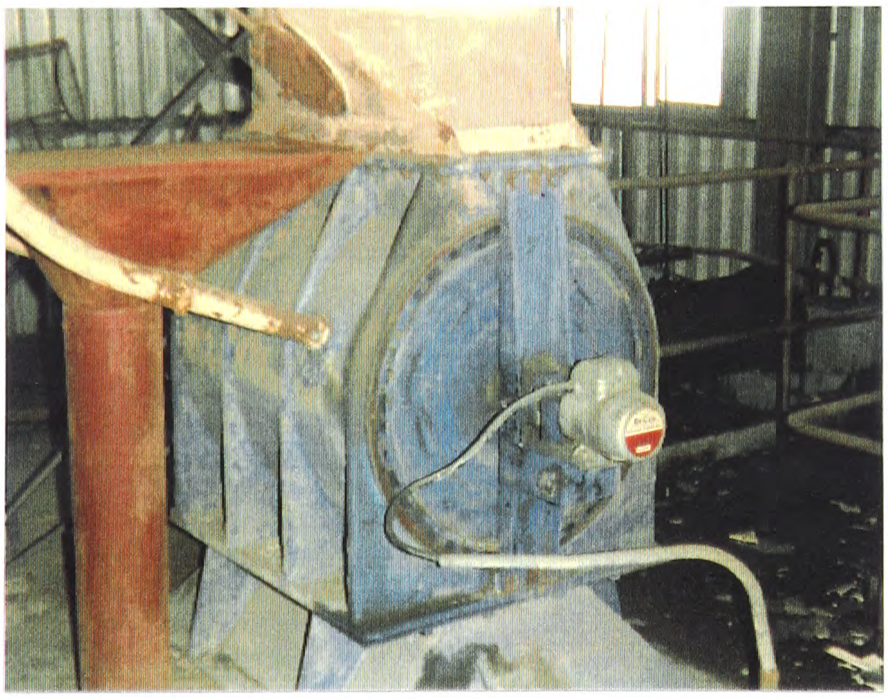


been tried, such as the Hartmann valve and also using columns of material to effect a "material seal", both with varying degrees of success (Ref 6).

2.4.5. Filtration System

Mowat (Ref 7), refers to the early designs of Duckham not having air filters. These particular systems allowed the grain to settle out in a large vacuum chamber (the receiving vessel) and the dust laden air to be drawn through the exhausters. Having passed through the exhausters the dust laden air was allowed to pass into another very large settling chamber where the dust could settle out of the air. The clean air was vented to atmosphere and the dust was collected and re-mixed with the clean grain from the vacuum receiving vessel, thus maintaining the overall weight originally off-loaded from the vessel. This was important to the merchants because they sold the grain by weight and, although nobody doubted the fact that the quality of the grain was improved by the de-dusting action of the vacuum conveying, the weight was reduced and hence their revenue decreased. This problem of loss of revenue was highlighted in Ref 2.

The gravity settling techniques described above, are no longer used in isolation, and have been superseded by the cloth filters and/or cyclone separators. The reasons for this are twofold. Firstly, the Roots type exhausters and axial fans now commonly used for such duties cannot tolerate fines continually flowing through them and secondly, environmental regulations in the UK such as the 1956 Clean Air Act etc, are becoming even more stringent, virtually restricting all dust emissions into the atmosphere.

2.4.6. Air Mover

The first vacuum pneumatic elevators, as designed by Duckham, used double acting reciprocating vacuum pumps of the type shown in Figure 2.13, to induce an airflow through the system. These vacuum pumps were either connected to steam powered cylinders via a connecting rod; or alternatively, they were driven by a belt drive system via a flywheel and connecting rod. An alternative to the reciprocating piston pump, was the turbo exhauster. The advantages of this machine were its lightness and compactness, but its disadvantage was its efficiency, which was lower than for the reciprocating pump. In the 1920's the Roots type positive displacement exhauster, Figure 2.14 was perfected and incorporated into many pneumatic conveying systems. However, it was not until the 1940's that the Roots type exhauster was accepted as the principal design of air mover for pneumatic ship unloading applications, due to it being both relatively light and compact for a given duty. Reciprocating pumps are not used nowadays, but many small fixed and mobile vacuum conveying systems use multi stage fans. However, the author is aware of an operational system (Ref 8) that still employs a reciprocating pump as the air mover, but using an electric motor to provide the necessary motive power.

2.5 Development Of Vacuum Pneumatic Ship Unloaders

By far the greatest effort in the development of vacuum conveying systems for ship unloading applications has been concerned with improving the discharge rate and, more importantly, the overall efficiency with respect to power consumption per unit rate of product conveyed. It is widely accepted that pneumatic conveying systems, and in particular pneumatic ship unloading systems, are inefficient compared with their mechanical counterparts. However, the inherent flexibility of pneumatic systems over mechanical systems often far

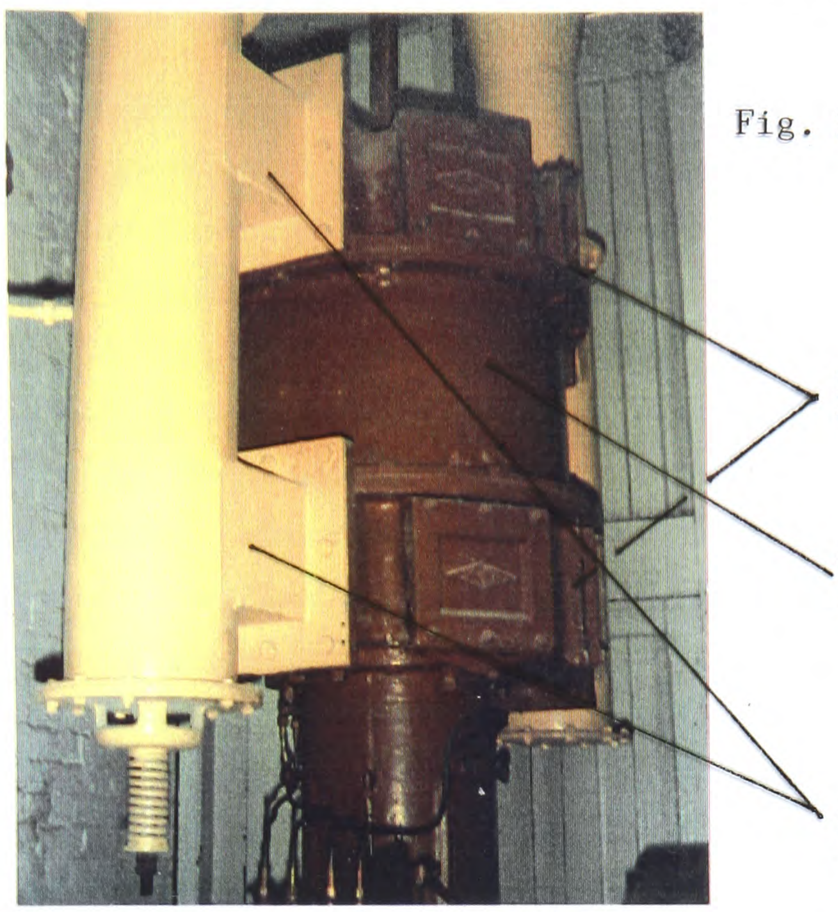
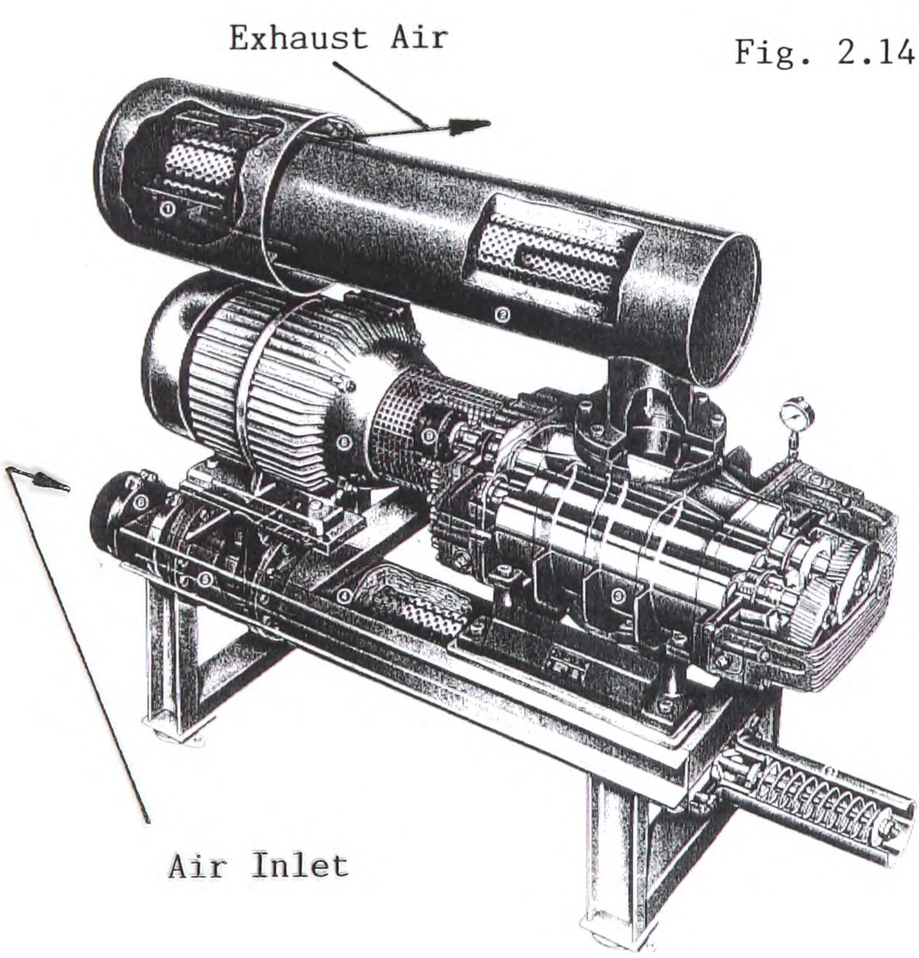


Fig. 2.13 A Typical Double Acting Reciprocating Vacuum Pump



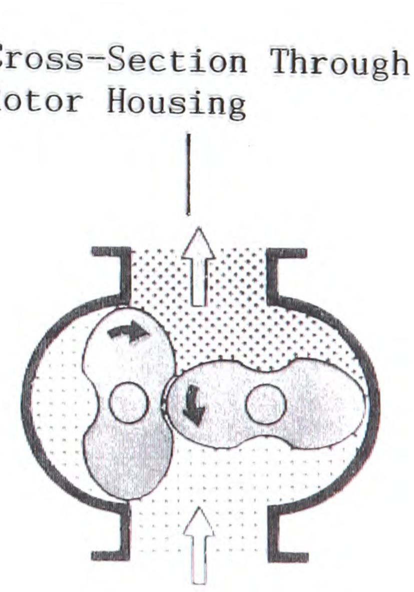
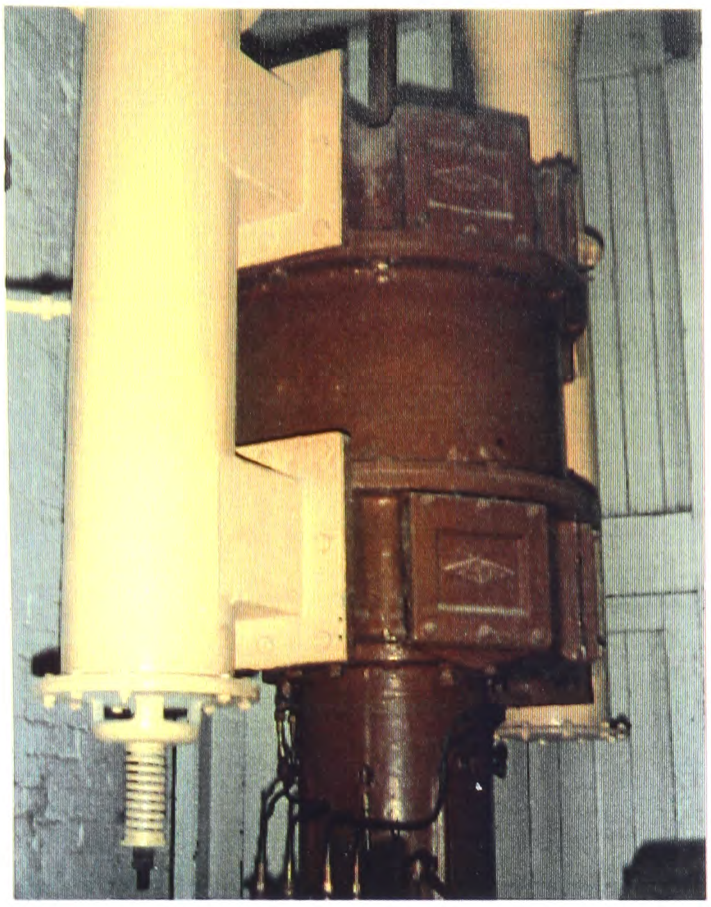
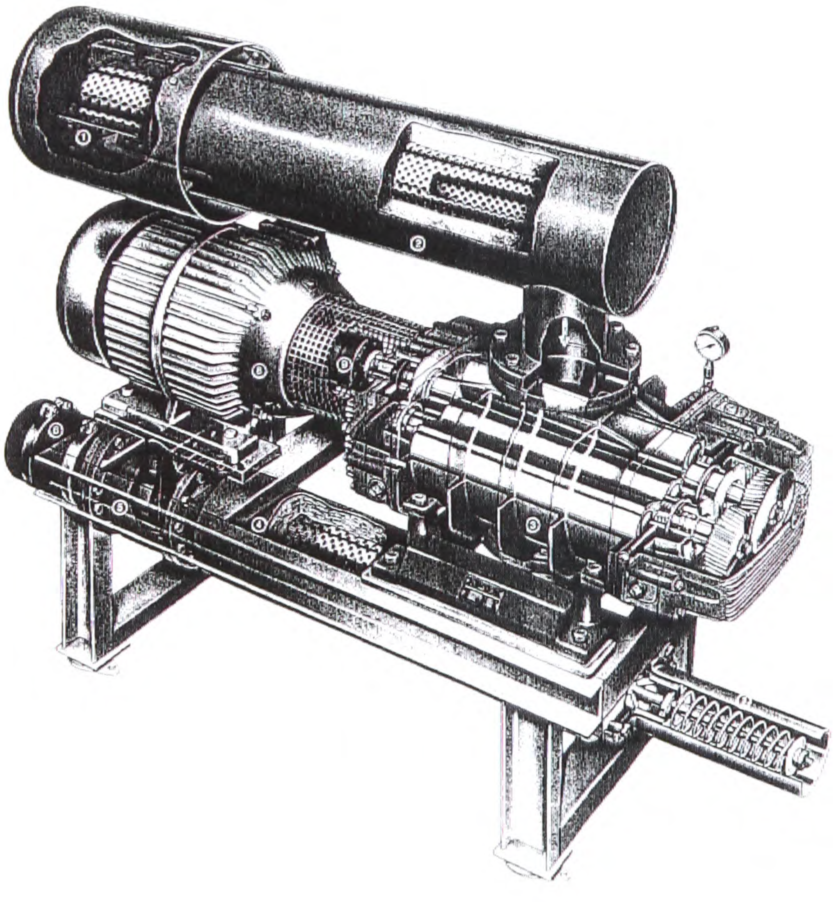
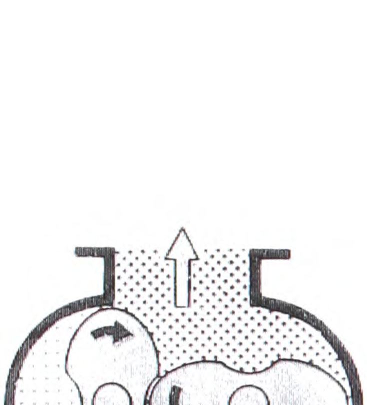


Fig. 2.14 A Roots Type Positive Displacement Vacuum Pump







outweighs their higher power consumption. Grimshaw (Ref 9) states that the power consumption of Duckham's early elevators was of the order of 5.0 hp/ton/hour (3.68 kW/t/hr) of grain elevated from ship to quay. He also states that from around 1906 Mitchell, who was working in this field, reduced this figure to approximately 1.5 hp/ton/hour (1.1 kW/t/hr) and that the power consumption of the then modern day systems (circa 1965) were reduced even further to 1.2 hp/ton/hour (0.88 kW/t/hr). The author has not found any specific documentary evidence to confirm that the figures relating to the early designs are correct, but if one studies the technical details outlined in several articles relating to Duckham's machines (Refs 4, 7 and 10) the power per tonne of grain conveyed figures can be calculated, giving 3.35 hp/ton/hour (2.46 kW/t/hr), 2.75 hp/ton/hour (2 kW/t/hr) and 3.0 hp/ton/hour (2.2 kW/t/hr) respectively. During the period 1892 to 1906 very little development work was undertaken in the UK to improve the performance and effectiveness of pneumatic elevators since there were several patents, assigned to Duckham himself, protecting the basic design of his system. However, it is believed that improved designs based on Duckham's machines were produced in Europe during this period.

It was not until the early 1920's that Cramp (Refs 11, 12 and 13) studied the performance of pneumatic grain elevators in a more scientific manner. In Cramp's paper of 1921 (Ref 11) he introduces the basic features of a vacuum pneumatic conveyor, and proceeds to discuss the performance and efficiency of vacuum conveying in vertical pipes approximately 9.0 metres long having a 64.5mm bore. Having shown that various equations can be used to describe the performance of conveying grain in these vertical pipes he referred to the "drawing power of the nozzle". He stated that, regardless of the design and efficiency of the conveyor, there is a definite limit to the amount of grain that can be introduced into the conveyor through a particular design of nozzle; and that the limit is dependent upon the air velocity flowing through the nozzle. His research work showed that, for any configuration of nozzle tested, a nozzle constant or ratio could be ascertained which took the form :-

$$\eta = L \,/\, (F_o \times V_{ao})$$

where η = Nozzle Constant
L = Unloading Rate (t/hr)
F_o = Cross-Sectional Area of Inner Pipe (cm^2)
V_{ao} = Superficial Air Velocity (cm^2)

Thus, knowing the nozzle constant for a particular design of nozzle, it is possible to predict the product entrainment rate for any given air inlet velocity or, to determine the cross-sectional area of the nozzle for a given product entrainment rate and air inlet velocity. Cramp summed up his discussion by stating that the "auxiliary air inlet" or outer sleeve is of little value except for limiting the drawing power of the nozzle. This effectively limits the solids loading and hence overall vacuum within the system to below, or equal to, the maximum vacuum that can be generated by the air mover.

Cramp's paper of 1925 (Ref 12) is simply a variation of the paper of Ref 11 and will not be referred to any further.

During the same period, Jennings (Ref 14) studied the operation of a small scale (15 to 20 t/hr) vacuum pneumatic conveyor for either elevating grain from lighters, or for removing grain from a lorry tipping pit at a flour mill in Bristol. This paper was presented with a view to highlighting the advances that had been made between 1890 and 1920 by comparing the then current pneumatic elevators against the early designs of Duckham. Many readings were taken relating the mass of grain unloaded per hour to the horse power developed at the pump. Analysis of the data suggests that the specific power consumption (hp/ton/hour) did not rise above 1.0 hp/ton/hour, which is low when compared with earlier comments in this section quoting specific power consumption figures of up to 5.0 hp/ton/hour. Unfortunately this article does not reveal any more useful information since the long horizontal run from the lorry tipping pit had just been installed at the time of publication.

2.5.1 Research On Suction Nozzles

Cramp and Priestley's paper of 1924 (Ref 13) is similar to Refs 11 and 12 in that it discusses the relationships between the conveying line pressure drops, air velocities and particle velocities, and also refers to the drawing power of the nozzle with similar comments. They undertook a series of tests to ascertain whether or not the nozzle constant was affected by the cross-sectional shape of the nozzle (in plan view) and its attitude within the bed of product. None of the nozzles tested had an outer sleeve (auxiliary air inlet), and simply consisted of the conveying pipe being placed directly into the hopper filled with grain. Three of the nozzles tested were circular in cross-section and had areas of 18 cm^2, 31 cm^2 and 80 cm^2 respectively. The first two had the same value of nozzle constant in spite of the fact that the second nozzle, because of its increased cross-sectional area, had a lower pick-up velocity. The third nozzle had a lower nozzle constant and an even lower pick-up velocity but still managed to convey the product. The most likely reason why conveying still occurred, according to Cramp and Priestley, was that only the product immediately next to the wall of the pipe was drawn in and the rest of the free surface beneath the inlet plane of the nozzle was static. Thus, it is possible for the air to flow preferentially through the live bed of product. However, this does not explain how conveying

velocities are maintained higher up the nozzle.
The author assumes that there must have been other mechanisms influencing the flow behaviour. Other nozzle shapes such as eliptical and rectangular were also tested by Cramp and Priestley. The eliptical nozzle was of similar area to the second standard circular nozzle and tests revealed that the nozzle constant was virtually identical. The smallest rectangular nozzle tested had a lower constant than its circular counterpart, with the likely reason being that the two flows from opposite faces interfered with each other. A circular nozzle placed horizontally into the hopper had a nozzle constant approximately twice that of the value relevant to the vertical attitude. The most likely reason for this is that the product can flow into the nozzle without having to turn 180° before it enters the conveying pipeline. There is also no static region by the nozzle face, thus the whole inlet plane of the nozzle is "seeing" a live product. These particular tests showed that the nozzle design can have a bearing on system performance, but no single shape is significantly superior and as a consequence it is doubtful whether the expense associated with employing nozzles of non circular designs can be justified.
Between the work of Cramp and Priestley of the 1920's and the early 1980's there appears to be little reference to work on suction nozzles in relation to vacuum pneumatic conveying systems. However, Chen (Ref 15) in his review of the development of pneumatic ship unloaders in The Peoples Republic of China between the 1950's and 1980's discusses some development work during this period. The tests by Chen showed that the performance of the co-axial type suction nozzle (which is similar to that shown in Figure 2.6) is related to the area ratio of the outer secondary air pipe, A, to the cross sectional area of the inner pipe, A_1. Ratios of between 0.8 - 1.0 were shown to be better for materials such as coal and sand, and ratios of between 0.4 - 0.6 were suggested as better for products such as grain. The area A, should equate to the cross-sectional area of the annulus formed between the inner and outer tubes and not the total cross sectional area of the outer pipe.

Unfortunately, the data is given without reference to the design or geometry of the suction nozzle itself, and therefore it is not possible to offer an opinion on his conclusions. However, if one assumes that all the air flowing through the system enters via the outer sleeve, the air velocity along the annulus will increase as the area ratio decreases. Also, there will be a corresponding increase in pressure drop along the annular chamber in the direction of flow. It is suggested that a small annular area is not so important with grain because the air can flow through the bulk of the product just as easily as flowing down the annulus. Whereas, with sand it is probable that the majority of the air flows down the annulus since it cannot flow through the bulk as easily. Hence the outer sleeve is more important and thus the pressure drop should be minimised by increasing the annular area.

More recently a programme of work undertaken by Foster (Ref 16) concerned with conveying coal by vacuum, discusses the need to admit air as near to the start of the conveying pipeline as possible, thereby reducing the solids loading ratio and hence the overall vacuum generated in the system. In his work a co-axial tube suction nozzle was used as the feeding device. It was found that altering the position of the two tubes relative to each other could significantly increase the mass flow rate of coal. During these tests it was also noticed that small horizontal movements of the whole suction nozzle assembly within the bulk increased the mass flow rate above that obtainable for a static fixed nozzle. As a consequence the programme of tests was repeated, but this time with a vibrator mounted on the suction nozzle assembly. Figure 2.15 depicts the variation in mass flow rate of coal with nozzle extension for both static nozzle and vibrated nozzle conditions. This figure suggests that as the inner tube of the nozzle is extended with respect to the outer tube, the mass flow rate of product increases until an optimum point is reached where, any further increases has a detrimental effect on the mass flow rate of coal.

The most likely reason for the vibrated nozzle yielding a higher mass flow rate relates to the grade of coal under test. This was not a particularly free flowing grade, which probably limited the rate at which the coal can flow under the action of gravity to the nozzle, thereby limiting the amount of coal that can be entrained by the suction nozzle.

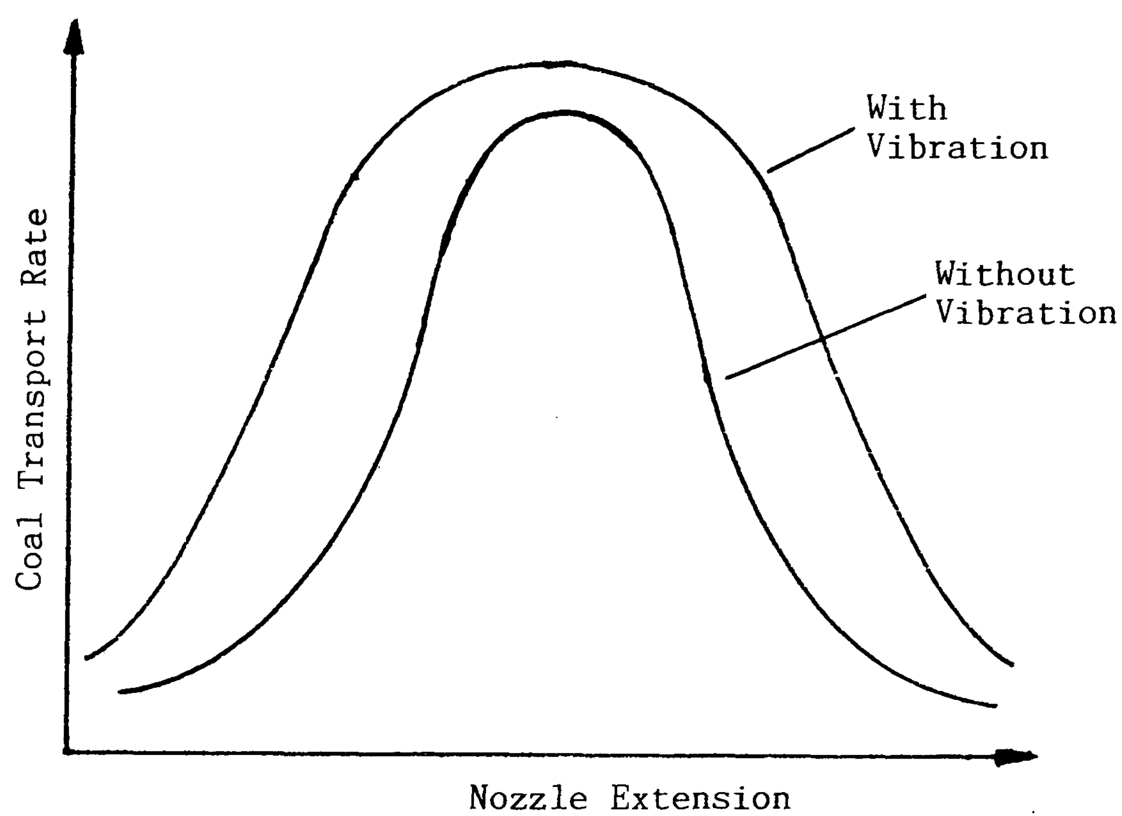


Fig. 2.15 Variation In Product Flow Rate With Nozzle Extension - Taken From The Paper by Foster (Ref 16)

However, by vibrating the nozzle these vibrations will be imparted into the bulk causing the coal to flow more readily to the nozzle resulting in a higher overall flow rate. With respect to the general trend up to the "optimum" position, it is reasonable to assume this to be correct since if atmospheric air can flow down the annular chamber and into the inner conveying pipeline without coming into contact with the product, as would be the case if the inner conveying pipe were retracted inside the outer sleeve, there is no reason to assume that any product would be entrained. However, if the air flowing down this chamber has to pass through the product to reach the inlet of the inner pipe it is logical to assume that product would be entrained into this air flow.

However, it is not so easy to understand the trend for nozzle extensions greater than those corresponding to that required for maximum throughput. This is based on the observation that should the free end of a vacuum cleaner hose is accidentally allowed to plunge slightly too far into a bed of material whilst clearing a spillage the system chokes; ie. the hose near the inlet fills completely with product causing a sudden increase in pressure, thereby effectively stalling the fan. It is suggested that this, in effect, would be the same as extending the nozzle geometry to such an extent that the outer sleeve becomes redundant. It would appear from this statement that there is a fine dividing line between success and failure and not a gradual reduction, as depicted by curves A and B of the figure of Foster. However, caution must be advised here since the above statement is based purely on supposition and analogous experience and not quantitative data.

The author has found only a limited number of technical papers regarding the development of the feeding devices for vacuum pneumatic conveying systems, which tends to suggest that most of the development work has been conducted within the companies supplying pneumatic ship unloading equipment. It is understandable that there is a lack of such information in the public domain since manufacturers will, for obvious commercial reasons, not want to share their "expertise" with a competitor. Although this statement may be a true, it is clear that there is a general lack of understanding due to the multiplicity of types and forms of suction nozzles employed with such applications.

Another reason for this general lack of understanding is that any suction nozzle and hence the conveying system, will work to some degree unless the design is so poor that no material can be entrained into the conveying pipeline at all. As a result, many companies operating such equipment tend to undertake their own development work "in-house" to improve the discharge rate.

Kano et al, (Ref 17) investigated the effects of nozzle design on the performance of a vacuum conveying system conveying polystyrene pellets for four different configurations. It is not possible to make comments on the conclusions drawn by Kano as a result of this particular study because an English transcript of the paper was unavailable. However, it would appear from the different designs of suction nozzles examined, Fig 2.16, he was considering the possibility of increasing the performance of the suction nozzle, by providing a smoother flow pattern within the entrainment region, for example see Figure 2.16.c. Depending upon the type and nozzle configuration used, it is possible to reduce the cross sectional area of the annular chamber as depicted in Figures 2.16.b, c and d. It is probable that Kano was assuming that the restriction would give rise to an increase in the air velocity leading to an increase in the rate of product entrainment.

A paper by Davies et al. (Ref 18) reports the findings of a series of tests undertaken on a full size system unloading alumina at a rate of around 500 t/hr, with a view to initially understanding the behaviour of the system and then improving the unloading rates. Although the system was used to unload both petroleum coke and alumina, the test work concentrated on the latter, since it represents the greatest proportion of the total tonnage unloaded at the facility under consideration.

In this paper the author then discusses the operational problems caused by using a co-axial tube nozzle having a "standard" protruding nozzle configuration with no throttling (restriction) of the atmospheric air entering the annular chamber, compared to a "modified" retracted nozzle configuration with throttling. The author states that if the

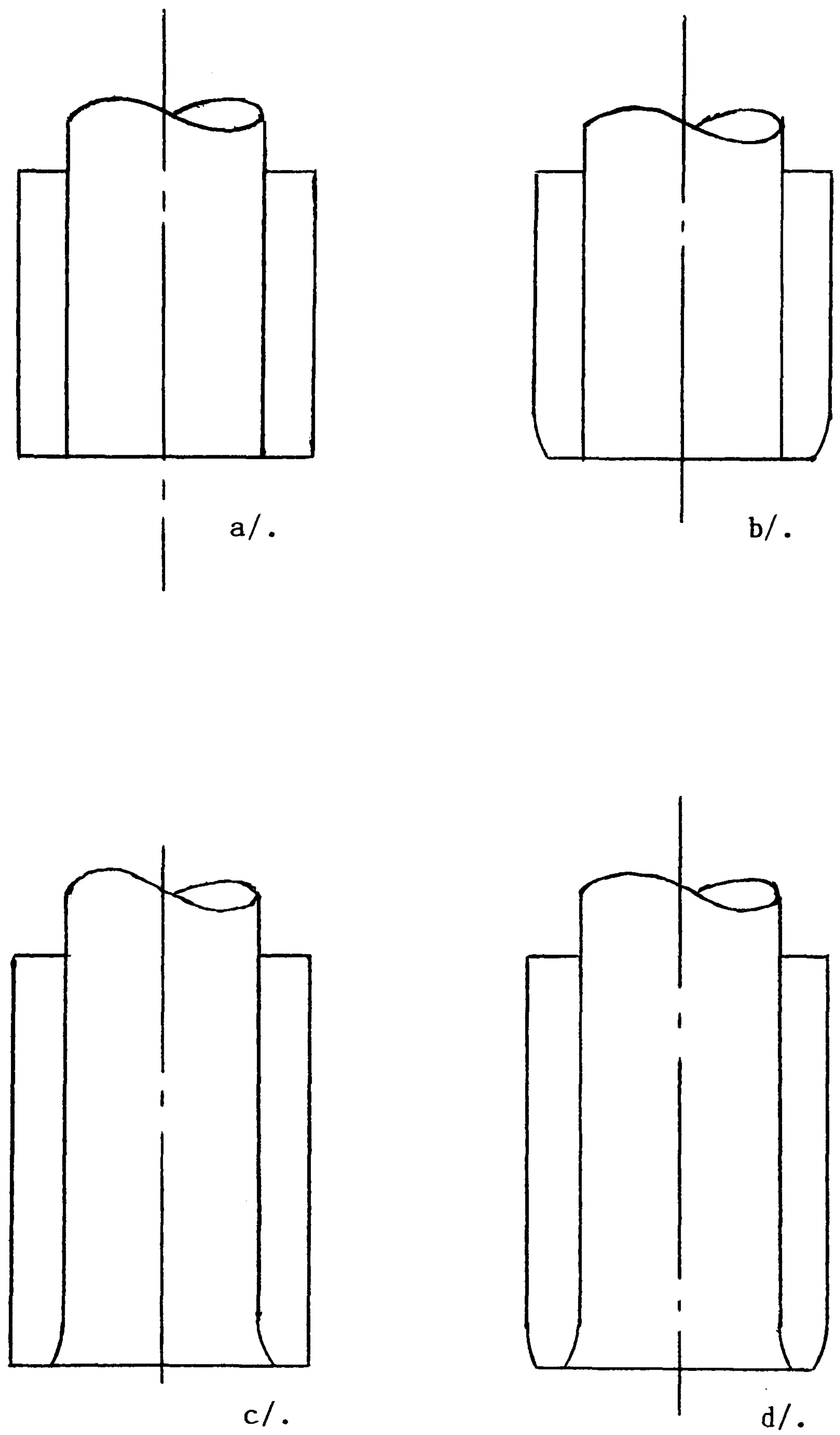


Fig. 2.16 Nozzle Configurations Tested by Kano

"standard" nozzle is lowered into the alumina, the nozzle will initially draw too much powder because the superficial air velocity is in the order of 60m/s. As the conveying line pressure drop rises, the air velocity will decrease until a condition is reached where choking occurs in the vertical riser. When considering the "modified" configuration the same problem could arise. This is because the superficial air velocity will also be of the order of 60m/s prior to the nozzle being dipped into the alumina, and thus the nozzle will also draw too much powder. The author proposes that lowering the "modified" nozzle very slowly into the hold it should be possible for the system to operate satisfactorily. However, one must consider the fact that the operators will, most likely, plunge the nozzle into the alumina regardless. Unfortunately, it is not clear whether the statements made by the author are based on supposition or practical evidence.

Davies then goes on to suggest that it might be possible to construct a suction nozzle having a variable geometry which can be adjusted after the nozzle has been lowered into the powder. However, it is considered that a simpler arrangement would be to fit an air bleed to the vertical riser just downstream of the suction nozzle. This air bleed could be fully open when the suction nozzle is initially lowered into the alumina, and then it could be closed slowly allowing the superficial air velocity in the nozzle to increase. This would, in turn, entrain more product causing the system vacuum to increase gradually to its operating condition. Davies concludes his paper with a somewhat obvious statement that the harder the exhauster is driven (ie. the greater the system pressure drop) the greater the mass flow rate of alumina. Perhaps more importantly he also states that the mass flow rate of product is insensitive to nozzle position for suction nozzles where throttling is employed.

2.6 Techniques for Unloading Non Free-Flowing Products

In an attempt to reduce transportation costs, many non free-flowing products are shipped in bulk and subsequently unloaded by pneumatic means. To enable the pneumatic ship unloaders to unload these non free-flowing products many novel devices have been developed to allow such systems to compete against their mechanical counterparts. Some of these designs, taken from a paper by Reed (Ref 19), are shown in Figures 2.17 and 2.18. They all rely on some form of mechanical agitator to either, loosen the product in the immediate vicinity of the nozzle, thereby allowing it to be sucked into the conveying pipeline or, they scrape the product to the nozzle. It is clear that the actual rate of discharge from the hold of the ship is dictated by the rate at which these devices can present product to the suction nozzle, assuming that the conveying system has been designed to handle the maximum discharge rates likely to be encountered.
In a paper by Chen (Ref 15), he discusses the use of a suction nozzle fitted with external guide vanes. These guide vanes are motor driven such that they rotate around the outer sleeve agitating and moving the product to the entrainment region of the nozzle. Although Chen's sketch depicting the device has no scale, it would appear that the upper set of guide vanes have a relatively small swept diameter. This means that only material quite close to the nozzle will be dislodged. Thus, it is not unreasonable to assume that a relatively small "bore hole" around the nozzle will result.
Teichmann (Ref 20) cites a case where a standard suction nozzle unloading a poorly flowing product such as Tapioca, using manual labour to trim the powder to the nozzle, had an average discharge rate of approximately 40 t/hr; whereas a discharge rate of 125 t/hr was obtainable using a loosening device. This device was attached to the end of a nozzle that could be "steered" into the Tapioca by the use of hydraulic rams.

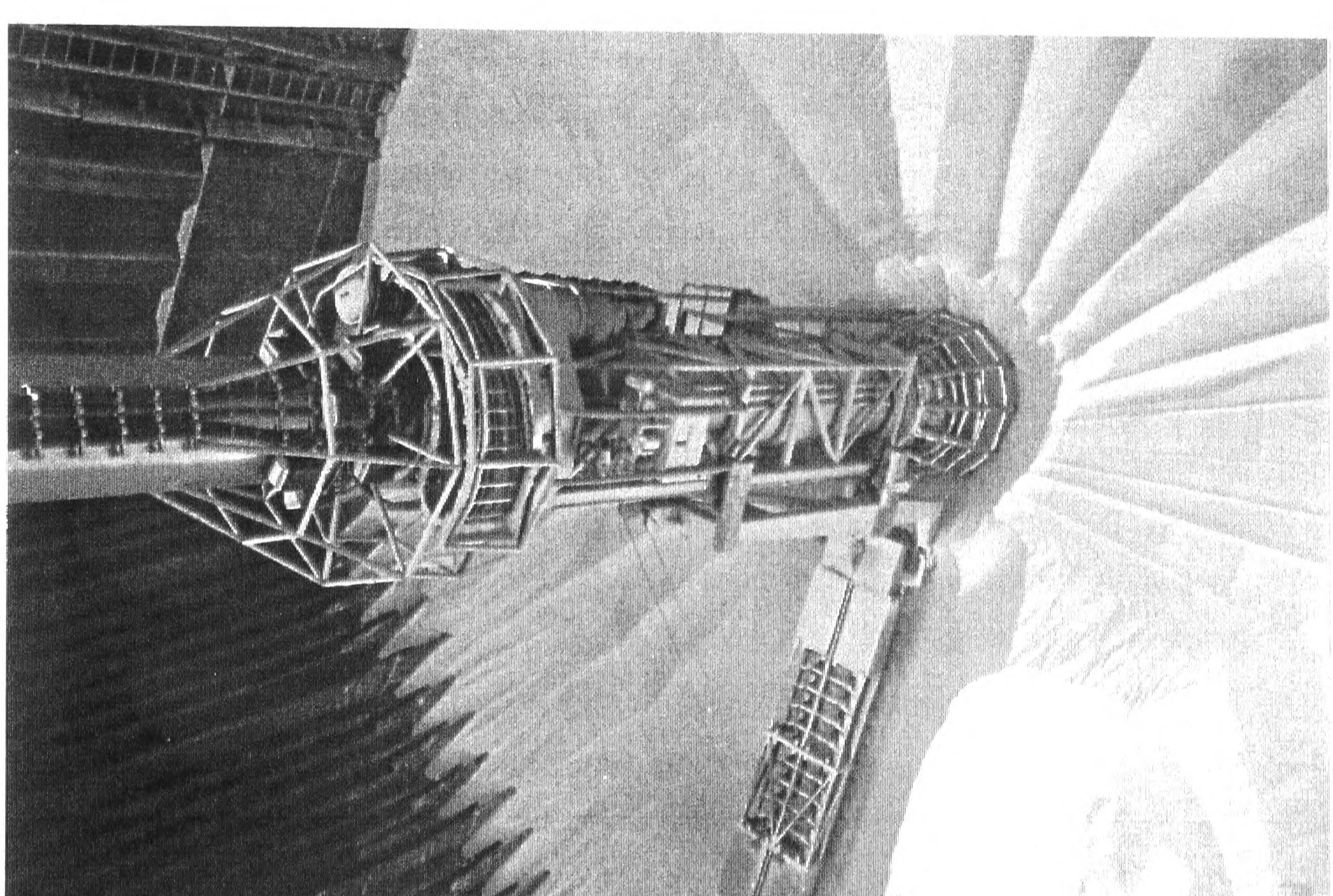


Fig. 2.17 A Mechanical Scraper Used to Feed a Suction Nozzle

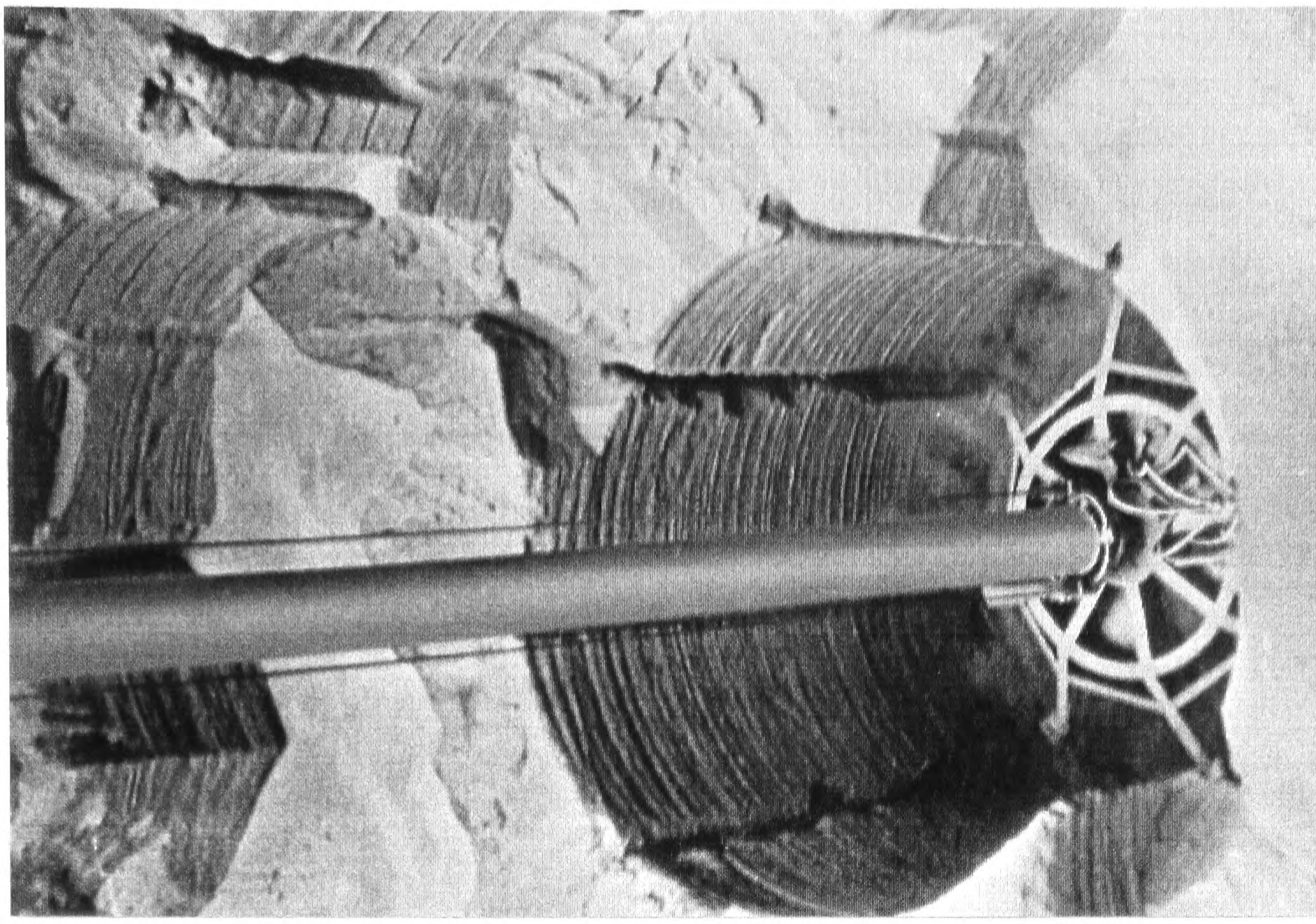


Fig. 2.18 A Bore Hole Created by a Mechanical Agitator in a Cohesive Product

As with all these mechanical agitators, the power required to drive them must be included in the balance between the power consumption versus flexibility. In spite of these devices a large number of 'main sinkings' are required because if the product is very cohesive, they simply create a large bore hole, the size of which is dictated by the size of the attachment, for example see Figure 2.18. Another problem with the use of such devices is that of restraining the vertical riser from twisting since there could be a large turning moment created by the action of the mechanical agitator.

2.7 Suction Nozzles: A Way Forward

The previous sections of this chapter have outlined the historical origins of vacuum pneumatic conveying, with respect to ship unloading, the technical developments and the academic work that has been undertaken over the past century. However, it is clear that many of the technical advances to improve the feeding device of such vacuum conveyors have been made "on-site" by the users of such equipment. Therefore, there is most likely to be a wealth of practical experience which has not been documented. Conversely, it would appear that most of the academic work undertaken in this area has been concerned with studying the performance of suction nozzle feeding devices with respect to a particular product.

Therefore, the purpose of this study is to develop an understanding of the effect of the major variables reported in this thesis on the performance of the co-axial tube type suction nozzle feeders for vacuum conveying systems, for a range of different products. Consequently it is suggested that by taking an experimentally based study it should be possible to develop such an understanding. Therefore, it is envisaged that the outcome of this study will provide the engineer with guidelines for optimising the performance of vacuum pneumatic conveying systems in their wide range of applications.

CHAPTER 3 EXPERIMENTAL PLAN AND TEST FACILITIES

3.1 Introduction

Since the purpose of this study is to provide an insight into how the geometry of the suction nozzle effects the performance of vacuum conveying systems, the design of any experimental facilities must incorporate means by which it is possible to achieve this aim.
The following sections describe, in detail, the test rigs constructed to enable the aims and goals of the test programme to be achieved. These sections also include information on the bulk solid materials used in this study, together with the operational procedures employed.

3.2 Experimental Plan

As with all experimental work a logical sequence of steps, or a plan, should be produced, prior to starting any such work. This enables a clear path to be followed from start to finish, rather than a series of random tests. However, as with all test work the outcome of a particular test or series of tests may necessitate the need to revise the existing experimental plan to accommodate any new tests. Although the basic scheme should be followed quite strictly, it must be flexible enough to permit revisions to be made at any stage, provided such revisions can be justified.
Since this programme of work is to study the effects of entrainment

mechanisms in co-axial tube suction nozzles on the performance of vacuum conveying systems, it is important that the experimental facilities incorporate a means by which it is possible to alter those variables that can be considered to be significant. In terms of the design and operation of suction nozzles it is considered the main variables influencing the performance of vacuum pneumatic conveying systems are,

i). The nozzle geometry, that is, the relative positions of the main conveying pipe and the outer sleeve,

ii). the superficial velocity of the air entraining product into the conveying pipeline,

iii). the throttling of the air flowing into the annulus formed between the inner conveying pipe and the outer sleeve, and

iv). the characteristics of the materials under consideration.

To obtain an initial insight into how these variables influence system performance, it was decided to construct a small scale test rig with appropriate sections fabricated from perspex. The small scale rig was constructed with a sectioned suction nozzle cut in half along its entire length, thereby permitting visual inspection of the entrainment mechanisms and flow patterns within the entrainment region surrounding the nozzle inlet. The four variables listed above are all incorporated into the design.

The actual settings for each of the four variables were chosen to enable the performance envelope of the perspex suction nozzle with a particular product to be determined.

3.3 Small Scale Test Rig

As has been previously mentioned, it was decided to construct a small scale test rig manufactured partially from perspex. The reasoning behind this is that since it is not known exactly how a co-axial tube suction nozzle really functions, it would be sensible to build a small scale test facility first and test this to see which variables are important, from both a visual inspection of the entrainment mechanisms occurring and obtaining quantitative data for differing nozzle configurations and conveying conditions. Based on the outcome of these initial tests a larger scale test rig could then be built to permit the principal variables to be examined with respect to their effect on the overall performance of a vacuum conveying system. The four variables listed in the previous section are all incorporated into this test rig. The actual settings of these variables have been chosen to enable the performance envelope of the perspex suction nozzle to be determined for the various test products under consideration. The basic arrangement of this small scale test rig is as shown in Figure 3.1.

Since it was considered that the geometry of the suction nozzle may have an important role to play in the overall performance of a vacuum conveying system, the nozzle was made to be adjustable so as to permit the inner conveying pipe to be moved relative to the outer sleeve. The suction nozzle, made from perspex, was cut in half along its entire length and attached to a sheet of perspex forming the front of a semi-circular hopper. The inner pipe and outer sleeve of the suction nozzle are made from two pieces of perspex tubing having internal diameters and wall thicknesses of 50mm x 4.5mm and 80mm x 4mm respectively. These sizes were chosen since the annulus formed between them is approximately equal in area to the cross sectioned area of the inner conveying pipe. Thus, the air velocity in the the annulus will be approximately the same as the superficial pick-up velocity of the air at the inlet to the conveying pipe. Figures 3.2.a, b and c. show

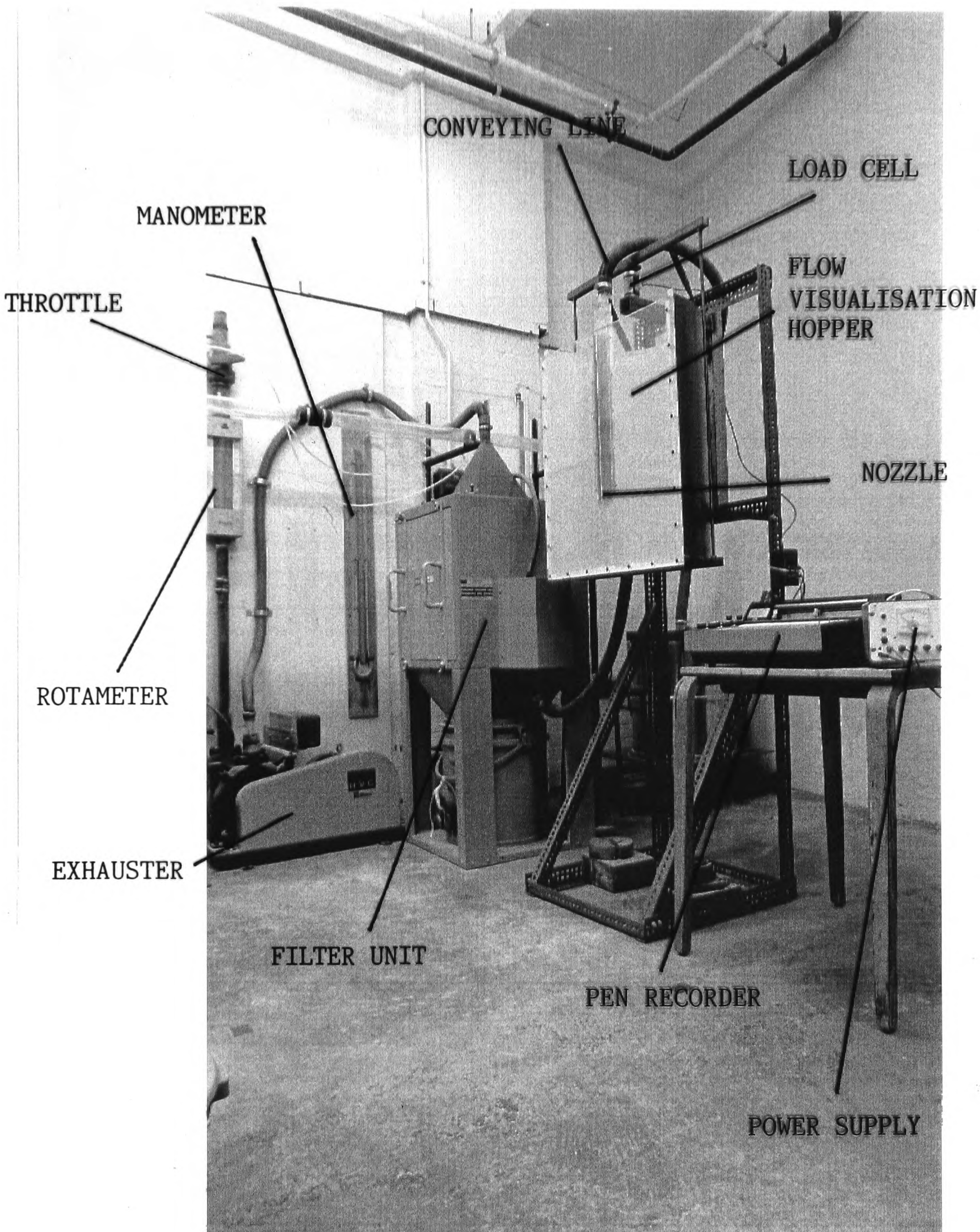
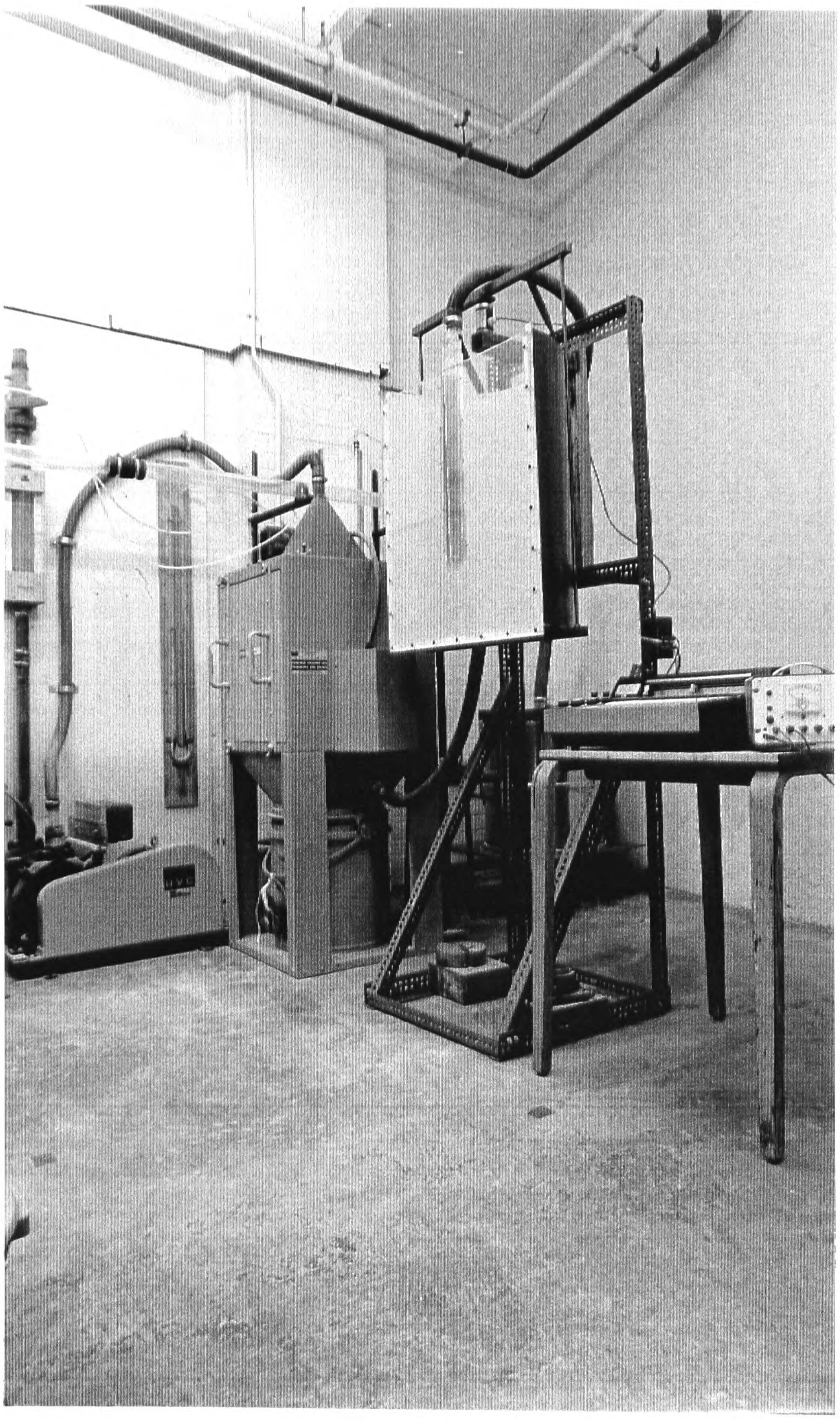
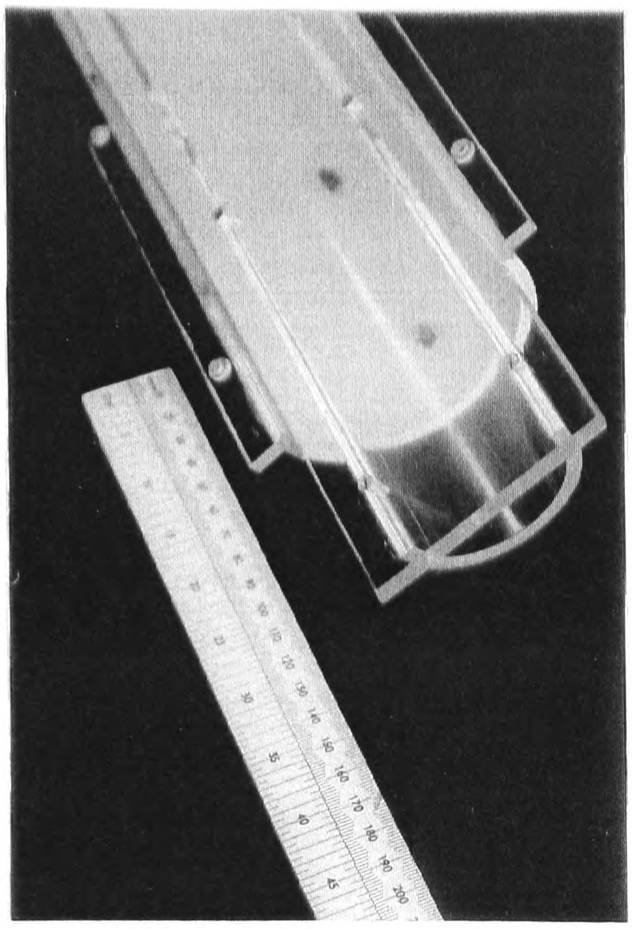
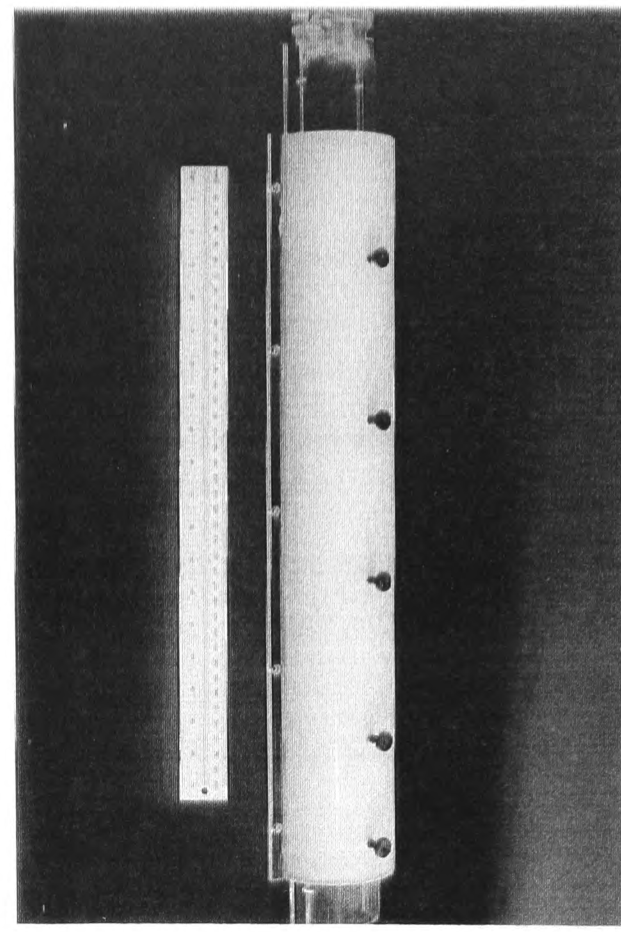


Fig. 3.1 Small Scale Test Rig

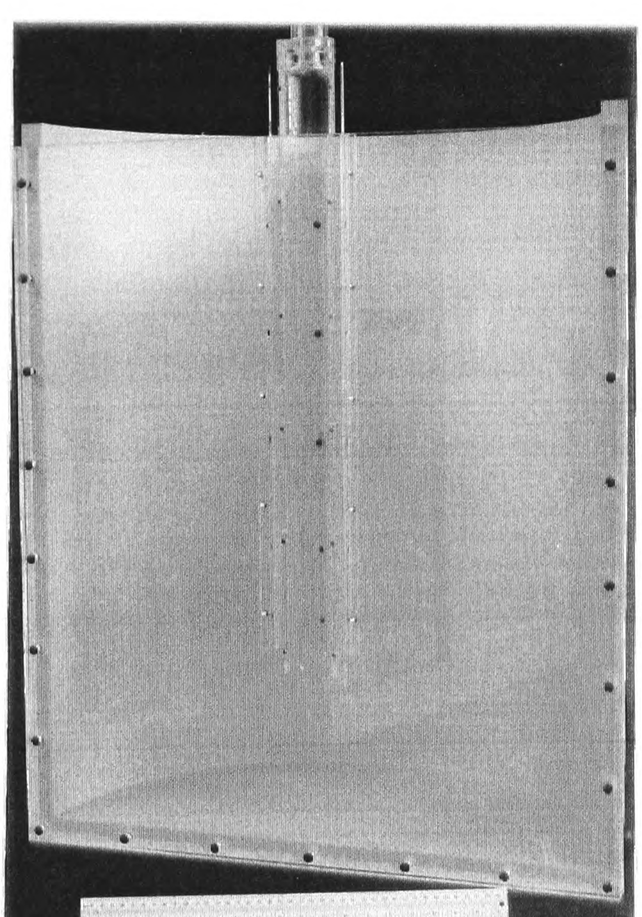




a)



b)



c)

Fig. 3.2 Perspex Hopper and Suction Nozzle Components photographs of the suction nozzle and hopper during manufacture. The hopper itself was suspended on a load cell which was energised from a regulated (5 Volts) power supply. The output of the load cell was fed directly to a y-t chart recorder. The y-axis, (pen displacement) when calibrated represents the variation of mass of the hopper,and the paper speed represents time. Thus, the slope of any trace produced corresponds to the mass flow rate of product being withdrawn from the hopper.

The hopper and load cell are both mounted on a simple supporting framework, with the hopper at a height to allow easy viewing of the flow patterns within the entrainment region, and also the overall flow patterns within the whole hopper as the bulk of the product contained within the hopper moves towards the nozzle.

The conveying line was approximately 2000mm long of 38mm bore flexible hose and was attached to the inlet of the receiving vessel at one end, and to the transition piece of the suction nozzle at the other end. The receiving vessel is a free standing DCE filter unit having an integral hopper mounted beneath the bag type filter elements. The clean air from the filter passes through another length of hose, similar to that of the conveying line, to the inlet of a fixed speed multi-stage axial fan. The exhaust air from the fan passed through a rotameter (flow meter) of suitable volumetric capacity to cope with the expected range of air flow rates, and then through a full bore ball valve which could be gradually closed to vary the air flow and hence pick-up velocity at the mouth of the nozzle. One limb of a mercury filled U-tube manometer was attached to the collection hopper of the filter whilst the other limb is open to the atmosphere, thereby permitting the conveying line pressure drop to be measured. To enable the effects of superficial pick-up velocity on the performance of the overall system to be examined, the performance characteristics of the fan were chosen to give a range of velocities between 0 - 60m/s. Although 60m/s is very high by pneumatic conveying standards, it was considered necessary to be able to ascertain if there might be some form of an upper limit above which no appreciable gains in system performance could be achieved.

3.3.1 Modifications for Throttle Adaptor

To allow the effect of throttling on system performance to be assessed, the small scale rig was modified to include a variable throttling arrangement at the inlet to the annulus formed between the inner conveying pipe and outer sleeve. Pressure tappings were placed in the wall of the outer sleeve to measure the pressure variations at various locations in the annulus. Figures 3.3.a and b show the detail of the throttle arrangement employed whilst Figure 3.4 shows a general view of the rig incorporating this feature together with the inclined multi-tube manometer bank used to measure the pressure distribution in the annular chamber formed by the two tubes comprising the suction nozzle.

3.4 Pilot Scale Test Rig

Having undertaken tests using the small scale rig, outlined in the previous section, a larger rig was constructed, Figure 3.5. It is essentially the same as the smaller rig but has some minor modifications. Firstly, the suction nozzle was circular in cross-section with the inner conveying pipe being fixed in position. The nozzle geometry is altered by adjusting the position of the outer sleeve. The design of the throttle was modified to suit the suction nozzle assembly and to enable the throttle setting to be altered easily.

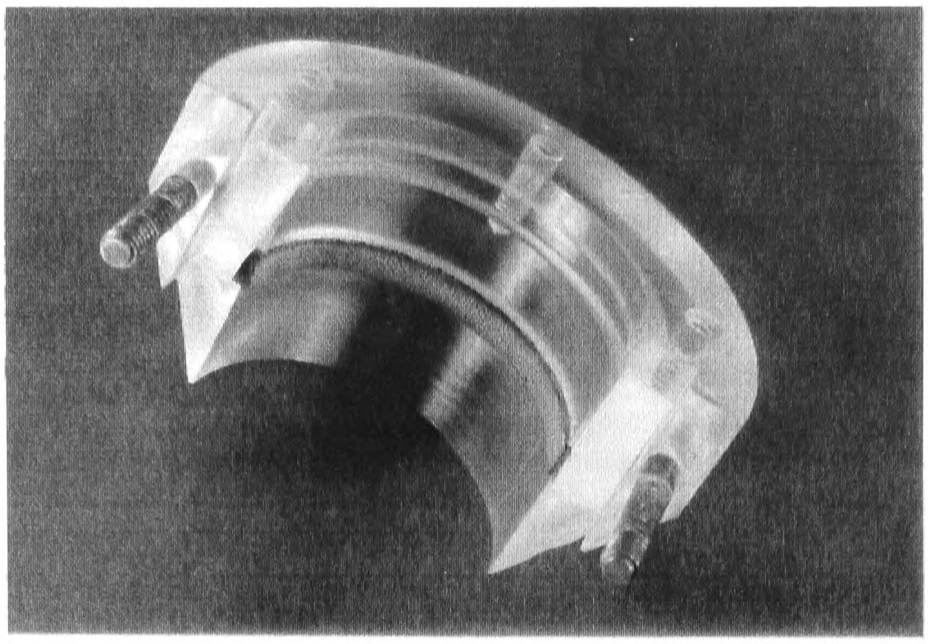
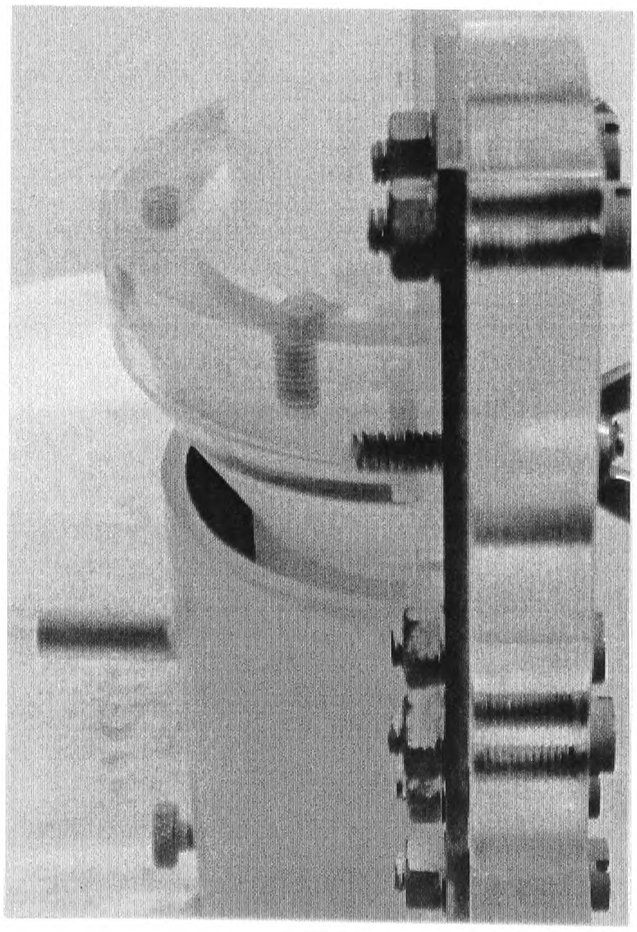


Fig. 3.3.a View of Throttle During Manufacture

Fig. 3.3.b Location of Throttle With Respect to Induced Air Inlet



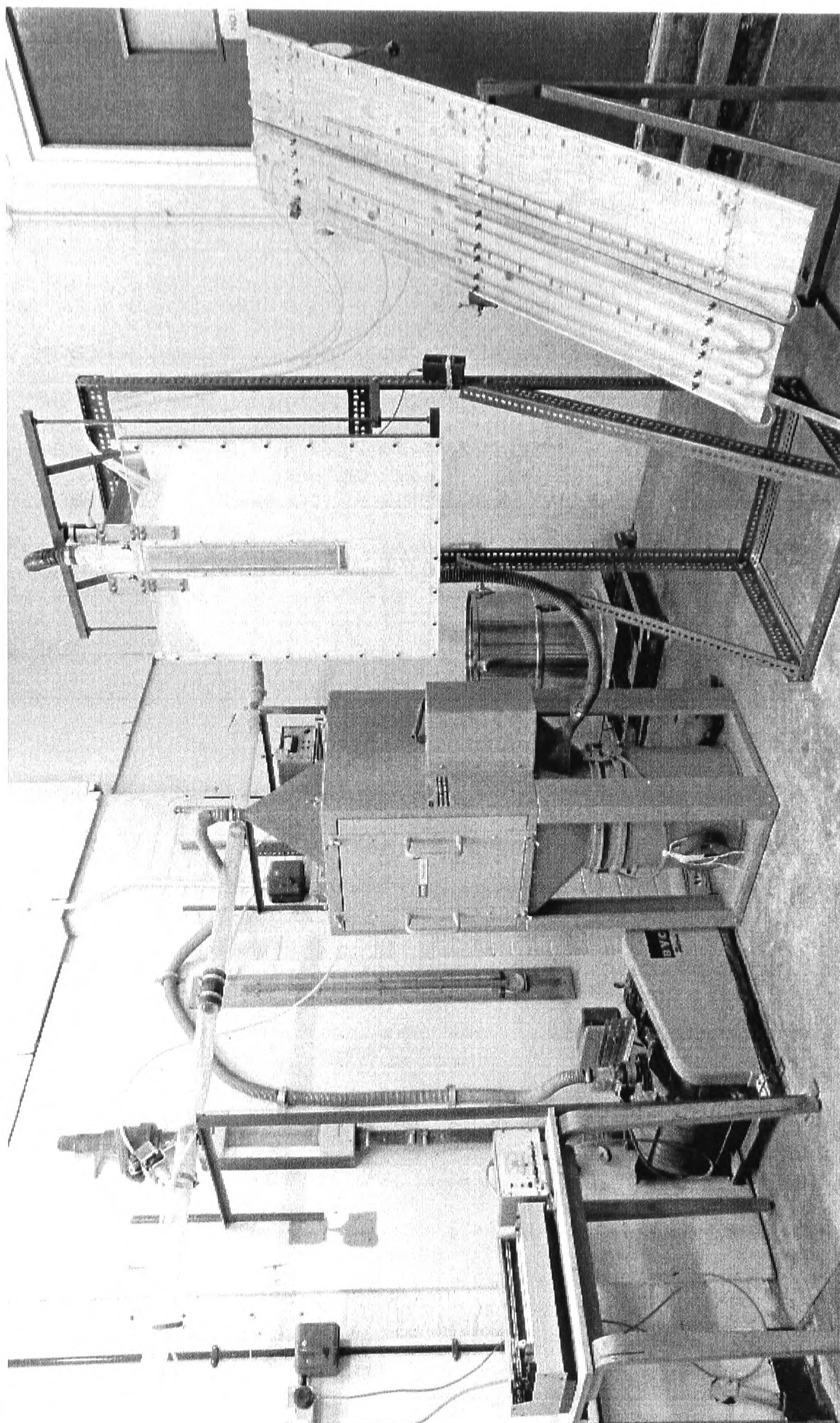


Fig. 3.4 Flow Visualisation Rig With Additional Manometer Bank for the Effects of Throttling Test Programme

3.4.1 Suction Nozzle and Supply Hopper

The supply hopper, shown to the right of the exhauster housing in Figure 3.5 was supported on three load cells. The output of these load cells was fed to an amplifier, the output of which, was linked to a chart recorder, as shown in Figures 3.6.a and b respectively. The inner conveying pipe of the suction nozzle (54mm bore x 3.25mm wall thickness) was supported by a three legged tripod arrangement which straddled the supply hopper. The tripod itself was bolted to the main structure of the test rig, isolating the suction nozzle and conveying pipeline from the supply hopper. This prevented any undue loads being imposed upon the hopper causing possible errors in the load cell readings. Figure 3.7 shows the general arrangement of the suction nozzle, tripod and conveying pipeline. The outer sleeve of the suction nozzle (105mm bore x 4.5mm wall thickness) was adjustable allowing the geometry of the nozzle to be varied. The linear displacement of the sleeve was altered by means of three rods which were attached to the sleeve at one end and passed through holes in each leg of the spider to handles at the top of the rods. The rods have holes, at 20mm centres, along their entire length through which retaining pins are placed enabling the outer sleeve to be raised or lowered as desired. Additional shims of 5mm and 10mm thickness were used to achieve finer adjustments of the sleeve as required. The general arrangement of the inner pipe and outer sleeve together with the adjusting rods is depicted in Figure 3.8.

The air was throttled by placing throttle discs in the throttle disc holder. The throttle disc holder fitted over the atmospheric air inlet to the annular chamber as shown in Figure 3.8. A length of flexible hose was attached to it, to duct clean air in from the surrounding atmosphere. The main reason for the hose was to prevent the accidental filling of the annulus with product due to spillage on

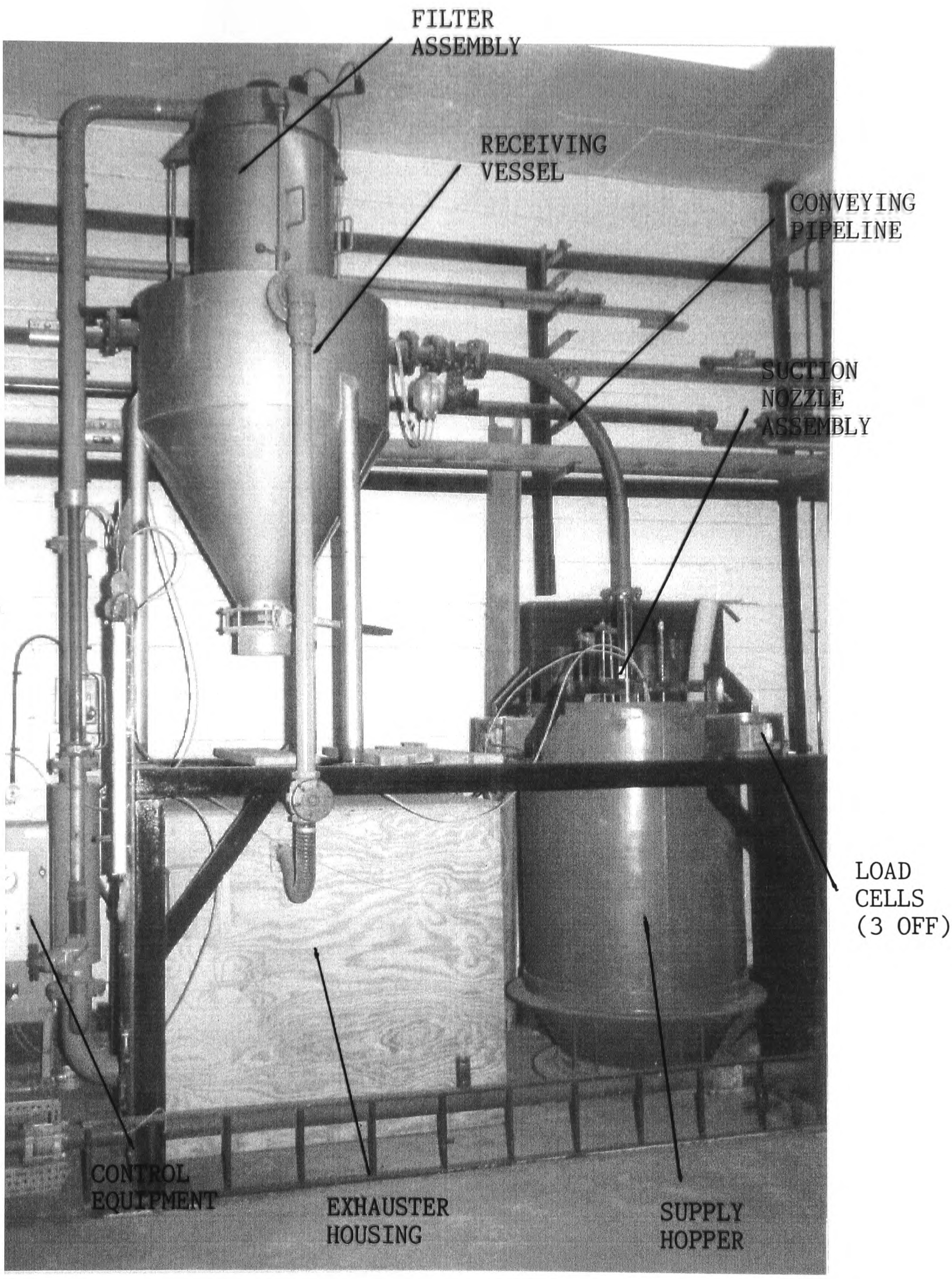
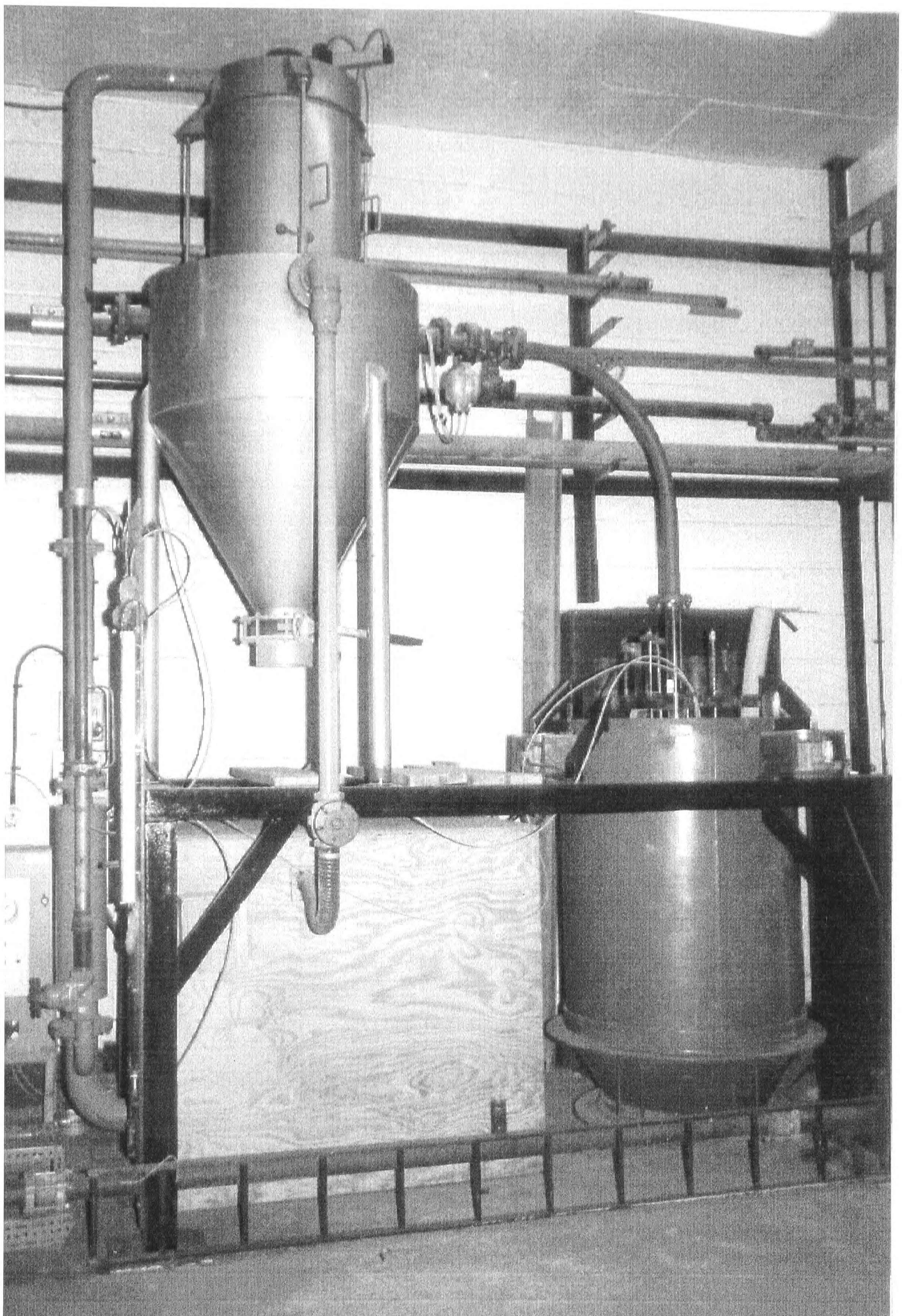


Fig. 3.5 Pilot Scale Test Rig



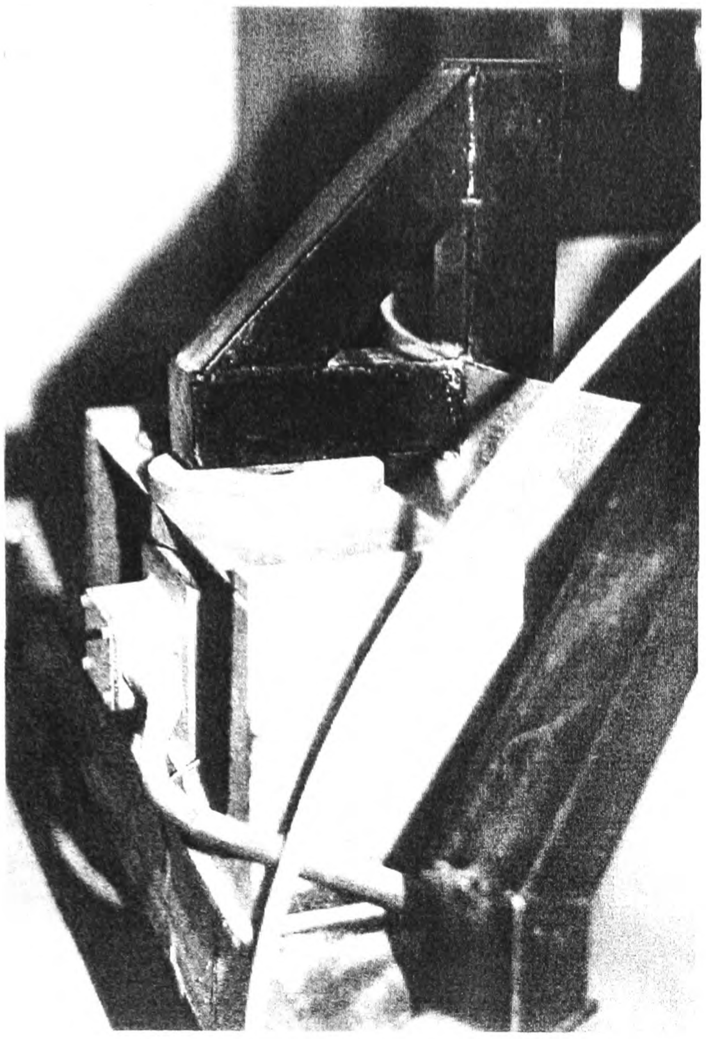
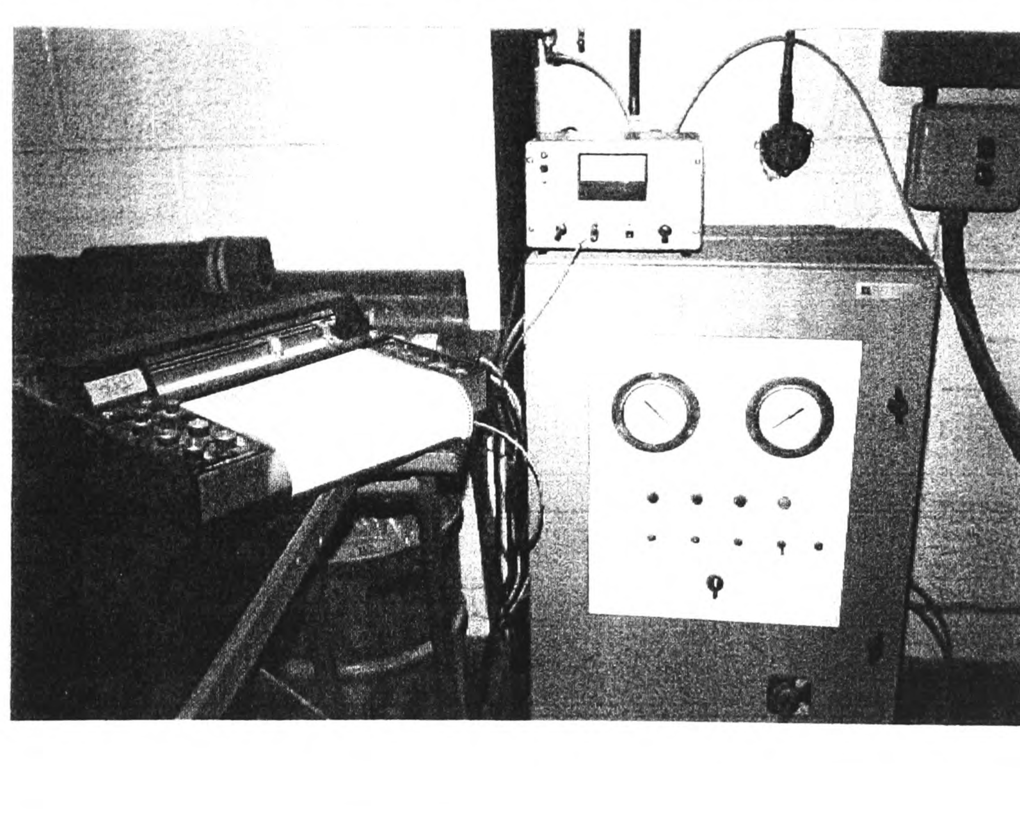


Fig. 3.6.a Load Cell Mounting Point With Supply Hopper Bracket

Fig. 3.6.b Load Cell Amplifier (mounted on control panel), Together With Chart Recorder



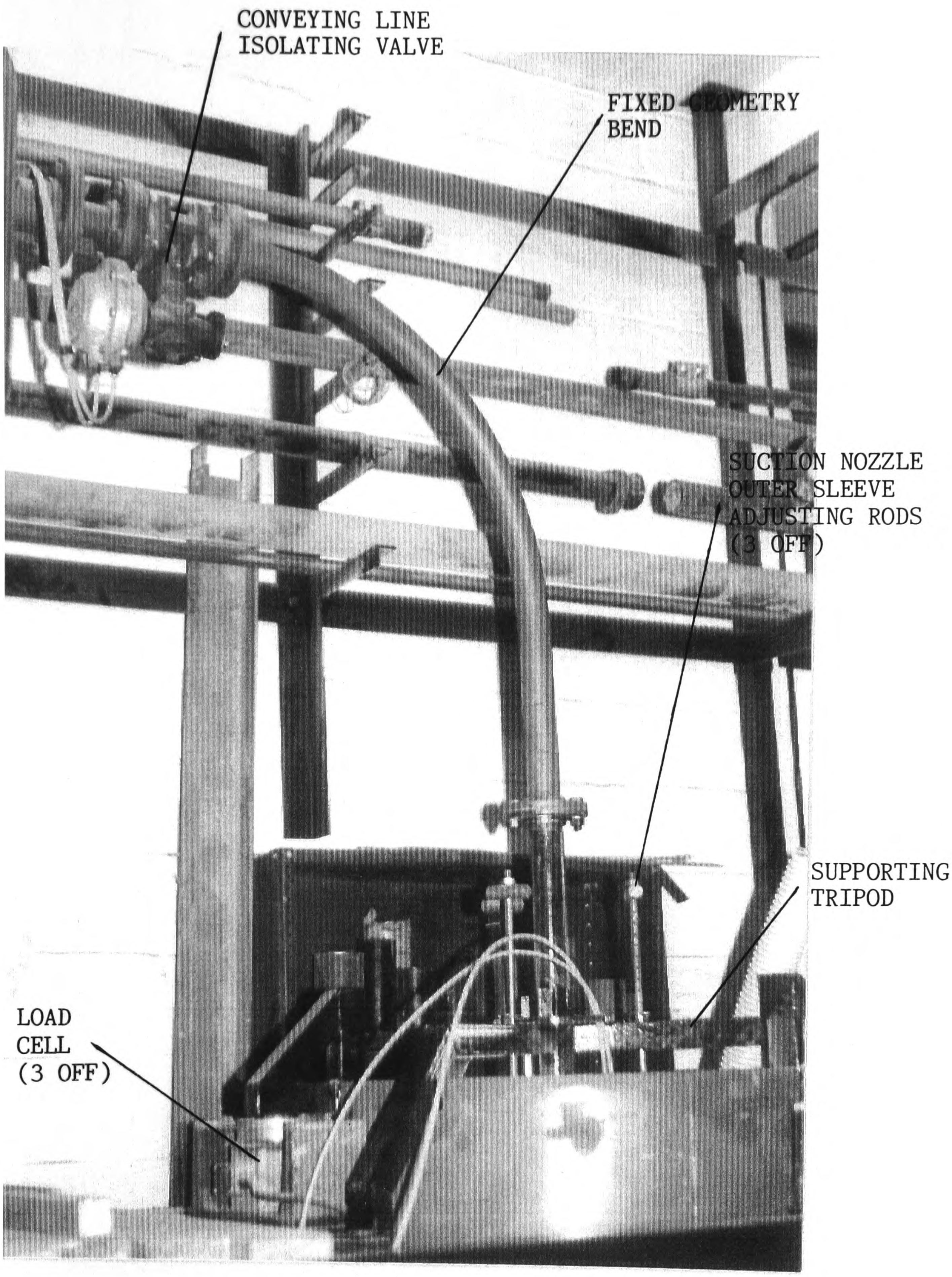


Fig. 3.7 General Arrangement of Suction Nozzle, Tripod and Conveying Line

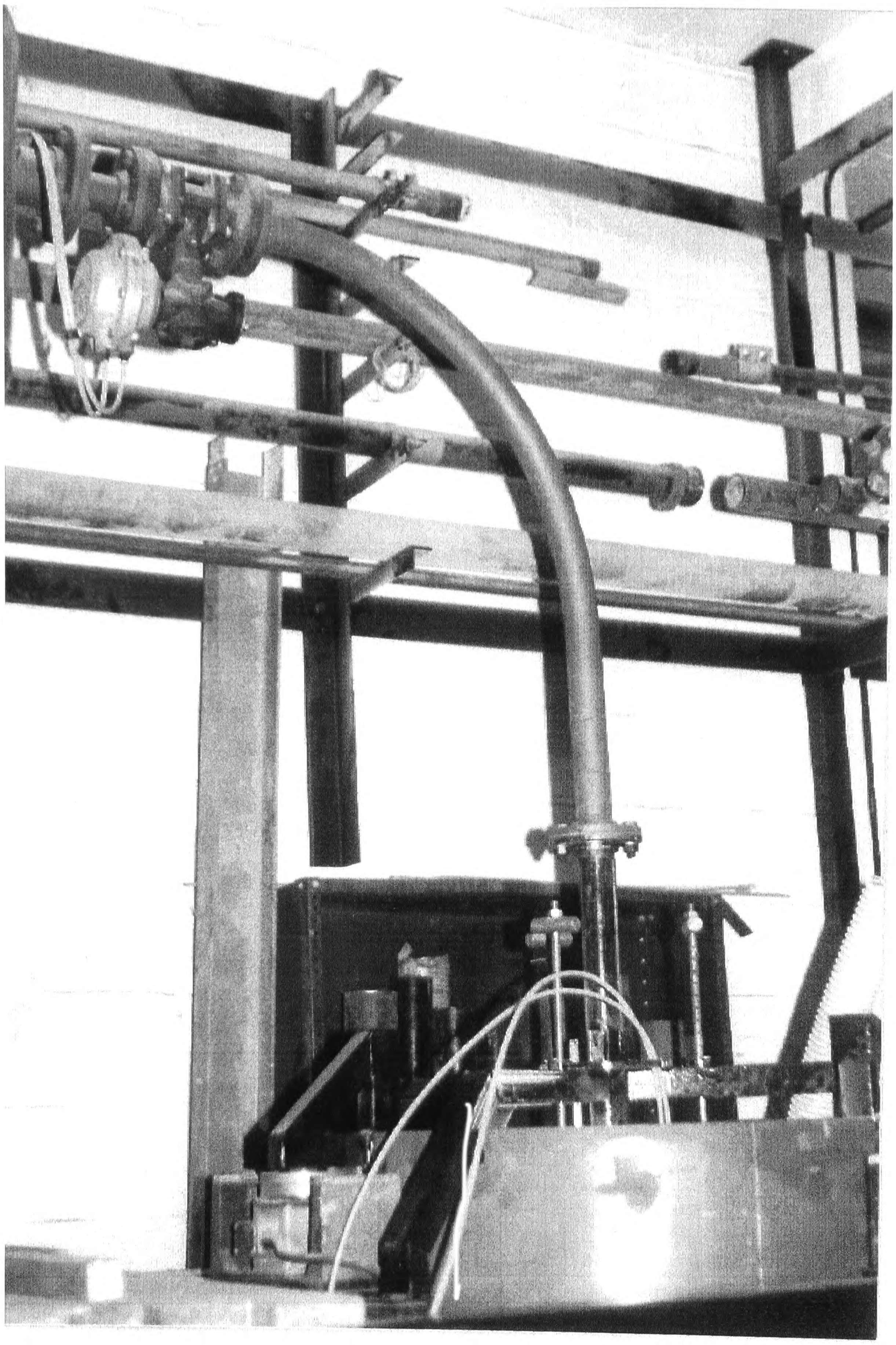
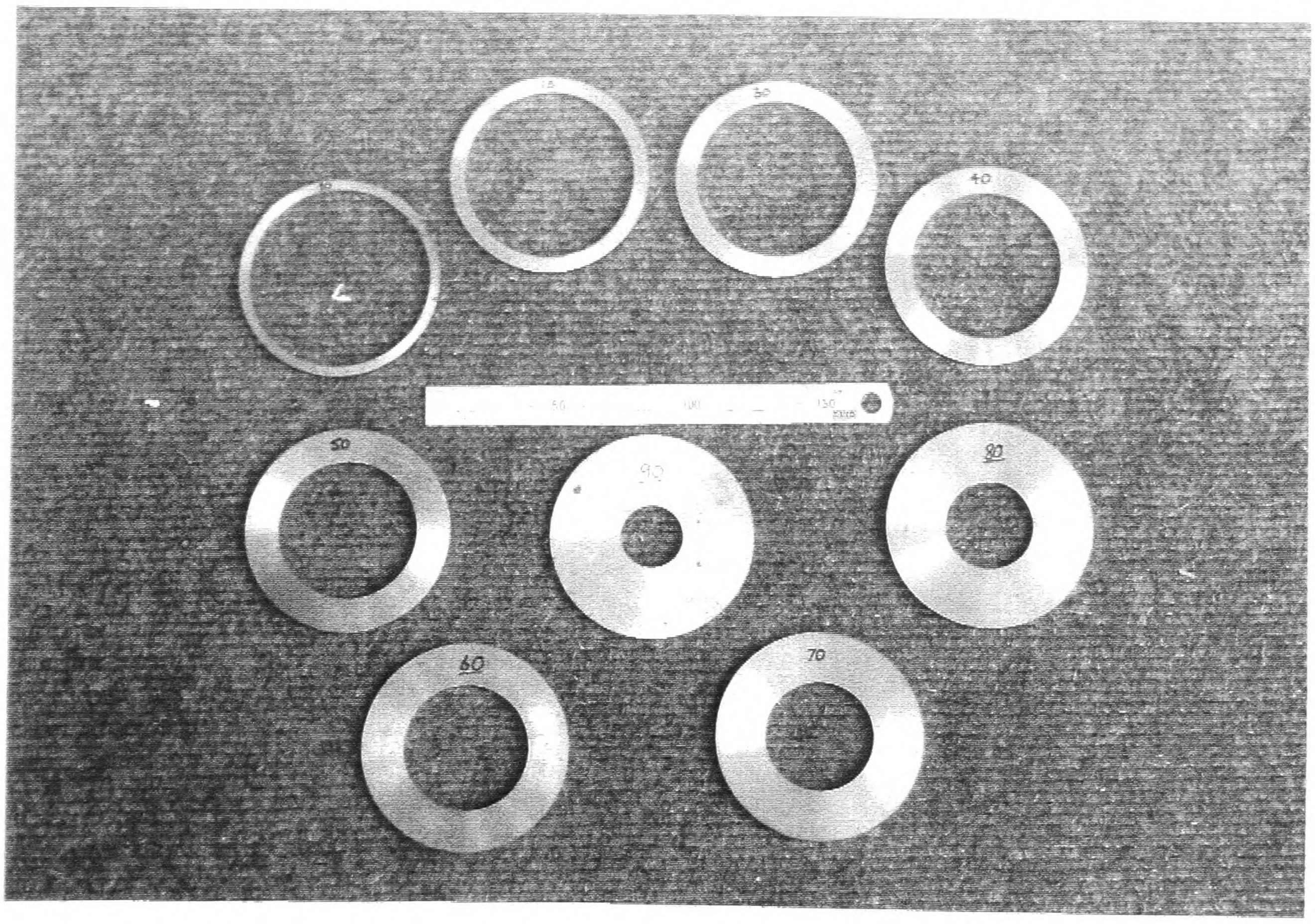


Fig. 3.8 General Arrangement of Suction Nozzle, Adjusting Rods and The Induced Air Inlet (Flexible Ducting)

Fig. 3.9 A Range of Throttle Discs



filling the surrounding vessel with product. The throttle discs themselves were simply mild steel discs with a hole machined in the middle to effect a percentage reduction in the area through which the air could flow. Figure 3.9 shows a range of such throttle discs.

3.4.2 Conveying Pipeline

Since the primary objective of this study was to develop a better understanding of the performance of the suction nozzle itself, it was necessary for the geometry of the conveying line to be fixed, since it is well known that variations in pipeline geometry can influence the overall system performance. This imposed a limitation on the design of the test rig since if the inner conveying pipe was to move up and down, as is the case for the small scale test rig, a flexible bend geometry would be required. Thus, fixing the geometry of the bend meant that the inner pipe would be fixed and the outer sleeve would have to be adjustable as mentioned previously. The pipeline is constructed from 54mm (2 inches) nominal bore mild steel pipe with a wall thickness of 3.25mm (0.125 inch). The radius of curvature of the bend shown in Figure 3.7 was 610mm (24.5 inches).

3.4.3 Receiving Vessel

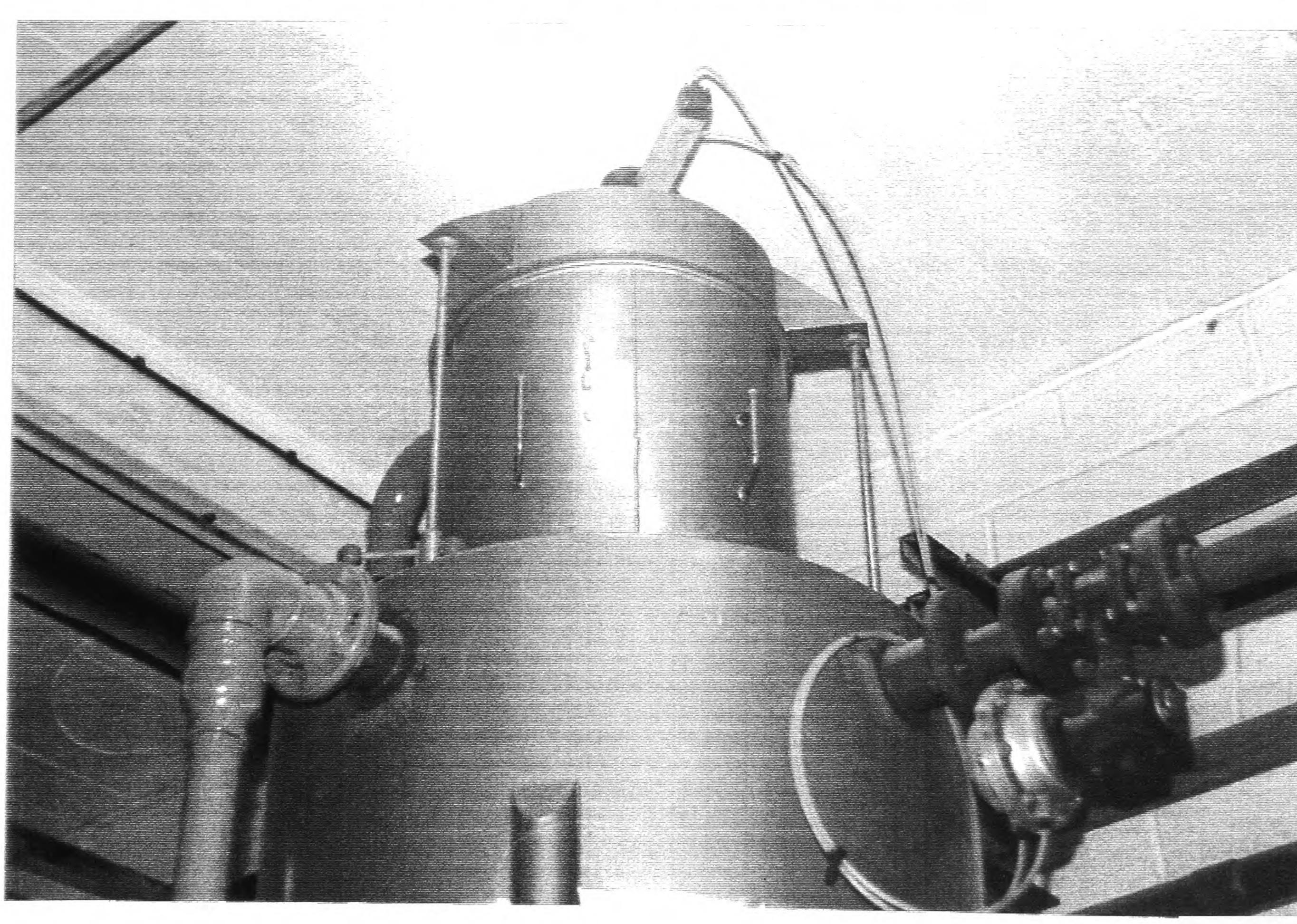
The receiving vessel was manufactured from mild steel with internal stiffening ribs where necessary to prevent the vessel distorting under the operating vacuum. Product was drawn into the vessel in suspension with the conveying air but as the air expanded on entry the product falls out of suspension and settles to the bottom of the vessel. The fines present in the air are carried up into the filters where they are restrained allowing clean air to pass to the main orifice plate and exhauster. To prevent the product impinging directly onto the lower sections of the filter elements, a deflector plate was positioned above the inlet pipe and arranged to run across the full diameter of the vessel. The plate is in the form of an up turned 'V'. A second inlet, mounted on the front of the vessel, as shown in Figure 3.5, enables air to be drawn into the system whilst the main conveying line is closed. Thus, the nominal air flow conditions for a particular test could be set up without conveying any product. This simulates the real system since the air flow is started prior to the nozzle being submerged into the hold of the ship, heap or stockpile. A pressure tapping, hidden from view in Figure 3.5 was fitted to permit the vacuum in the vessel to be monitored. This vacuum level is also equal to the conveying line pressure drop. To enable the material to be transferred back into the supply hopper, a 200mm (8 inches) dia, butterfly valve was fitted to the bottom of the receiving vessel.

3.4.4 Filter

The filter unit formed an integral part of the vacuum receiving vessel rather than being a separate item, since it has to withstand a far greater vacuum than was obtainable with the smaller rig. The filter elements themselves were of the cartridge type and were made from polypropylene having a total media area of $6m^2$. Figures 3.10.a and b show the filter cartridge and housing components and in their assembled position on top of the receiving vessel. A "back wash" filter cleaning unit was mounted on top of the filter cartridge carrier plate and was clamped down by three tie rods, which also served to hold the back wash unit aloft to enable the filter housing to be removed for periodic maintenance. The "back wash" cleaning cycle operated by suddenly reversing the air flow through the filter elements. A vacuum was generated in the receiving vessel by closing the ball valve in the conveying pipeline and restricting the volume of air flowing into the vessel via the relief vent line. When the pneumatic ram attached to

Fig. 3.10.a Filter Cartridge Assembly and Housing

Fig. 3.10.b Filter Assembly Location



the "back wash" housing was actuated, Figure 3.10.b, it opened an air path into the vessel allowing a sudden in-rush of air at atmospheric pressure. This air passed through the filter elements in the reverse direction removing the accumulation of dust particles built up on the surface of the filter media. A secondary in-line filter, placed between the main orifice plate and exhauster inlet, was intended purely as a fail safe device to prevent product entering the exhauster itself should one of the main filter cartridge elements have failed.

3.4.5 Air Mover

A Roots type, positive displacement exhauster package (GMa 11.3) manufactured by Aerzen Machines Ltd. was used as the air mover. This type of machine is currently used in all modern large scale pneumatic ship unloading systems. Since this particular exhauster was a constant speed machine, a method of varying the amount of air drawn into the nozzle was required. One option available was to alter the speed of the exhauster by changing the belt drive pulleys, but this option was not employed since it would be too time consuming and only step changes could be achieved. The option chosen was an air bleed arrangement, whereby atmospheric air could be drawn directly into the system just upstream of the inlet to the exhauster. This then enabled the conveying system to be by-passed, thereby reducing the actual volume of air drawn through the suction nozzle itself.

3.4.6 Air Flow and Pressure Measurements

The amount of air being drawn through the air bleed was adjusted by the use of a diaphragm valve and flow meter. The air bleed flow meter is used as a guide for adjusting the air flow rate prior to a test run rather than for absolute measurements. Figure 3.11 shows the position

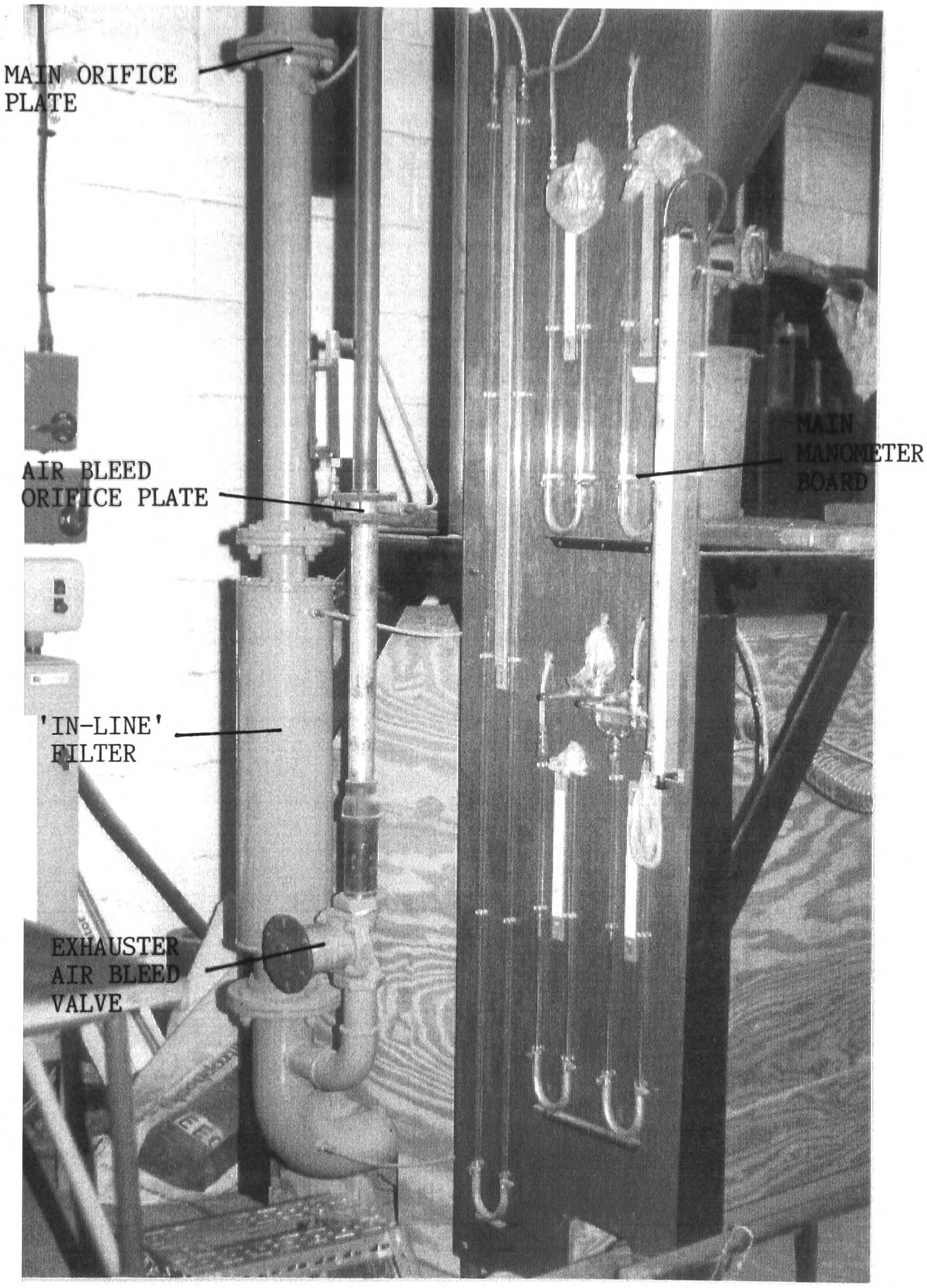
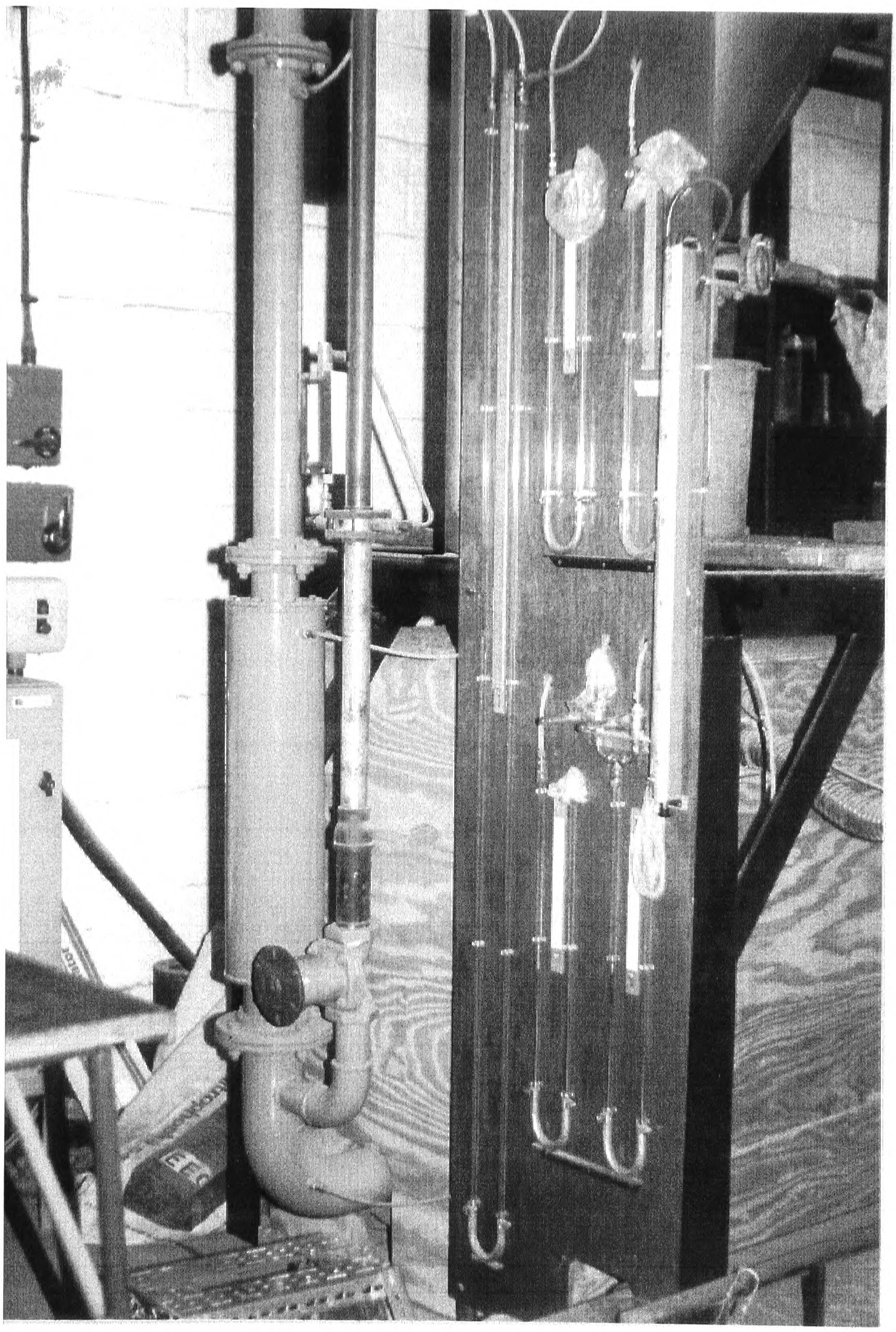


Fig. 3.11 **General Layout of Manometer Board, Exhauster Air Bleed Valve and Location of Main Orifice Plate**



of the air bleed valve and meter relative to the exhauster inlet. The actual air flow through the suction nozzle and conveying pipeline was measured using an orifice plate designed and installed in accordance with British Standards 1042 (Ref 21). This orifice plate was positioned between the main filter and the in-line filter allowing the orifice plate to measure the flow rate of clean air. The differential pressure across the orifice plate and the upstream pressure were measured using U-tube manometers filled with water and mercury respectively. The vacuum in the receiving vessel, in-line filter, suction nozzle throttle and material pick-up point were also measured using mercury filled manometers. The scales on these manometers, together with the upstream orifice plate manometer were directly calibrated in bar gauge, allowing direct measurements to be taken. This also gave the operator an immediate "feel" as to how the system was operating. The scales were mounted on adjustable slides enabling them to be re-set to zero (ie. positioned level with the manometer fluid). The manometers were all mounted on one panel which was positioned close to the switch gear, thereby keeping all operations in one area, as shown in Figure 3.11.

3.5 Test Products and Their Characteristics

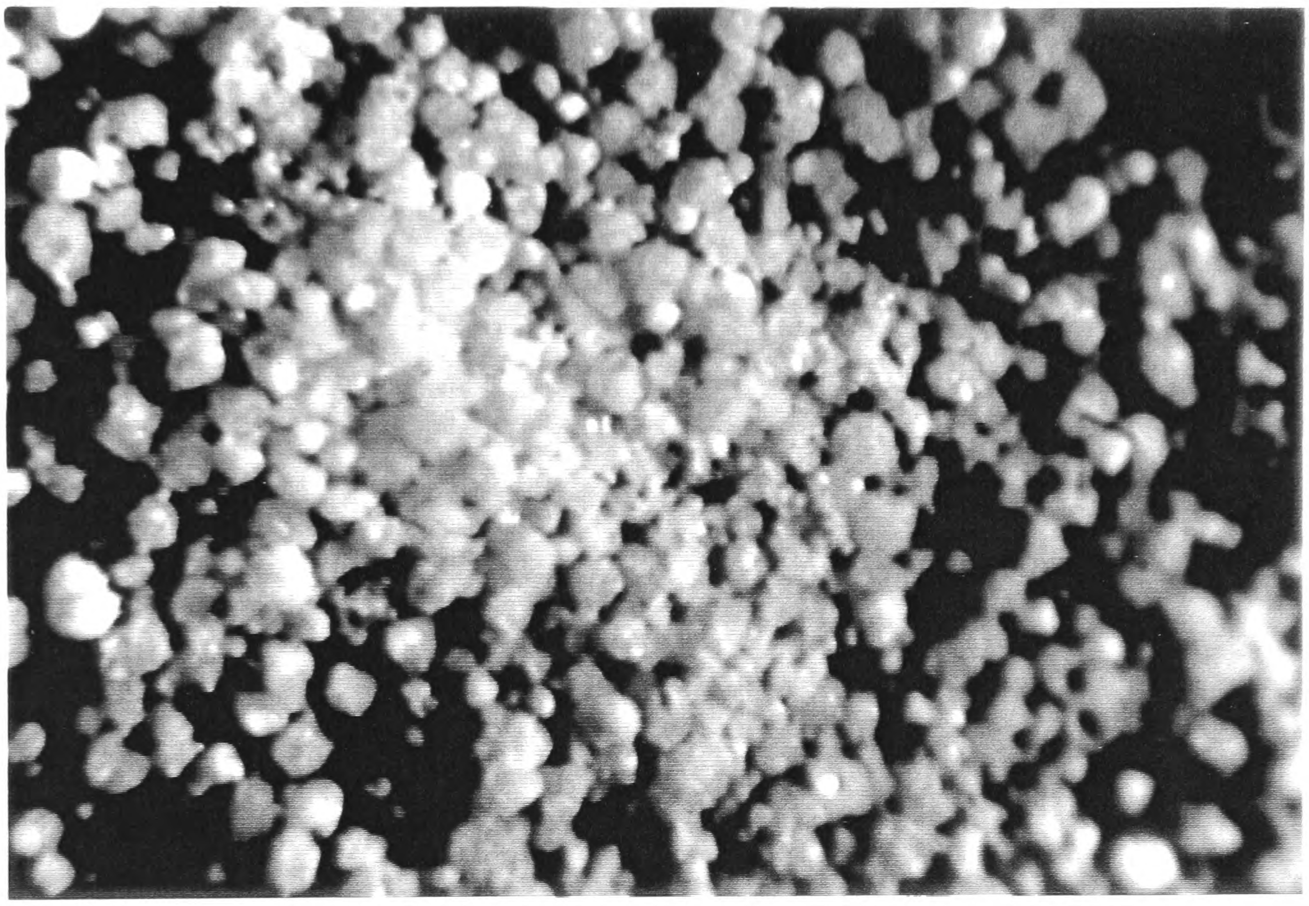
A range of products were chosen for the test programme to enable the performance characteristics of the suction nozzle to be analysed. The products were chosen because they are commonly conveyed in vacuum pneumatic conveyors or off-loaded from ships. Another important consideration was that they had to possess a number of different features, thereby permitting an assessment to be made of product characteristics on the resulting performance characteristics. For the work undertaken on the flow visualisation rig, the following materials were employed,

- Alumina
- Wheat Grain
- Mustard Seed
- Dicalcium Phosphate
- Plastic Pellets

whereas for the work undertaken on the pilot scale rig employed,

- Wheat Grain
- Plastic Pellets
- Aluminium Hydroxide

3.5.1 Alumina



Material Characteristics :-

Particle Density - 2633 kg/m^3 Shape - Irregular

Poured Bulk Density - 1107 kg/m^3

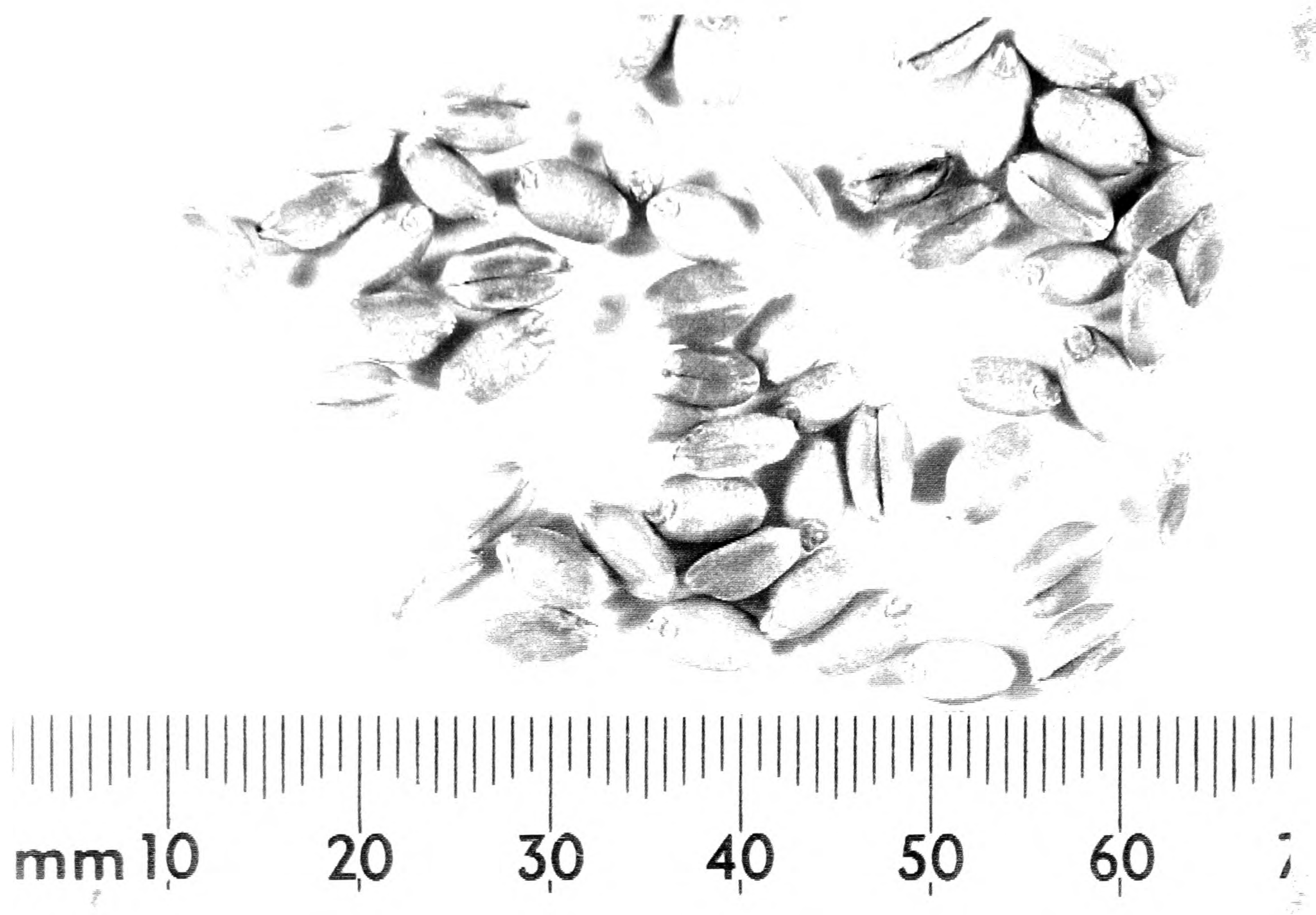
Poured Angle of Repose - 29°

Drained Angle of Repose - 35°

Size Distribution :- (by mass)

D(10%) - 48 μm
D(25%) - 63 μm
D(50%) - 83 μm
D(75%) - 105 μm
D(90%) - 135 μm

3.5.2 Wheat Grain



Material Characteristics :-

Particle Density - 1220 kg/m^3

Poured Bulk Density - 820 kg/m^3

Shape - Oval

Poured Angle of Repose - 35°

Drained Angle of Repose - 30°

Size Distribution :- D(10%) - 2500 μm

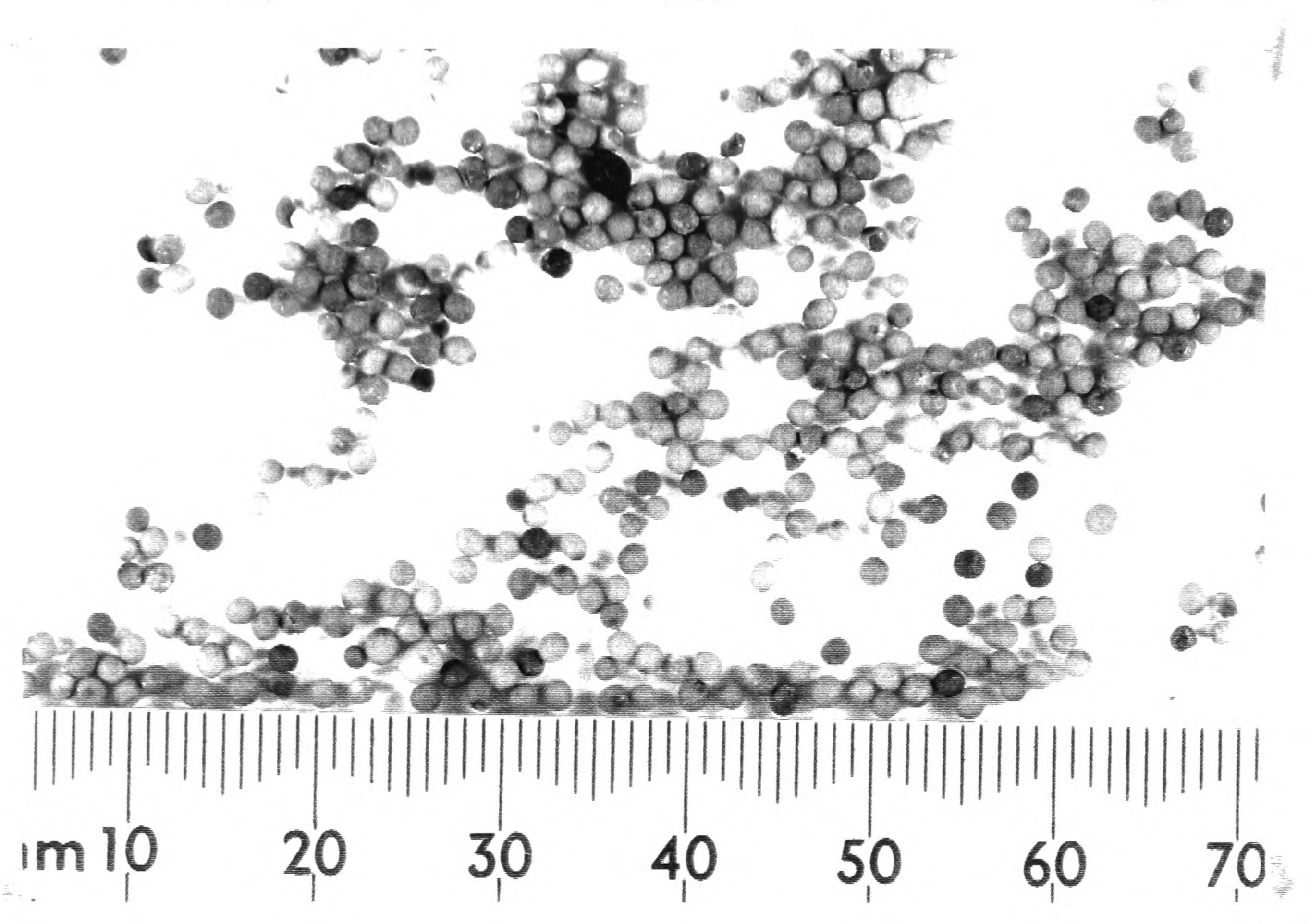
(by mass) D(25%) - 2700 μm

D(50%) - 3200 μm

D(75%) - 3750 μm

D(90%) - 4200 μm

3.5.3 Mustard Seed



Material Characteristics :-

Particle Density - 1120 kg/m^3

Poured Bulk Density - 701 kg/m^3

Shape - Sherical (Mono-sized)

Poured Angle of Repose - 30°

Drained Angle of Repose - 35°

Size Distribution :- (by mass)

D(10%) - 1250 μm

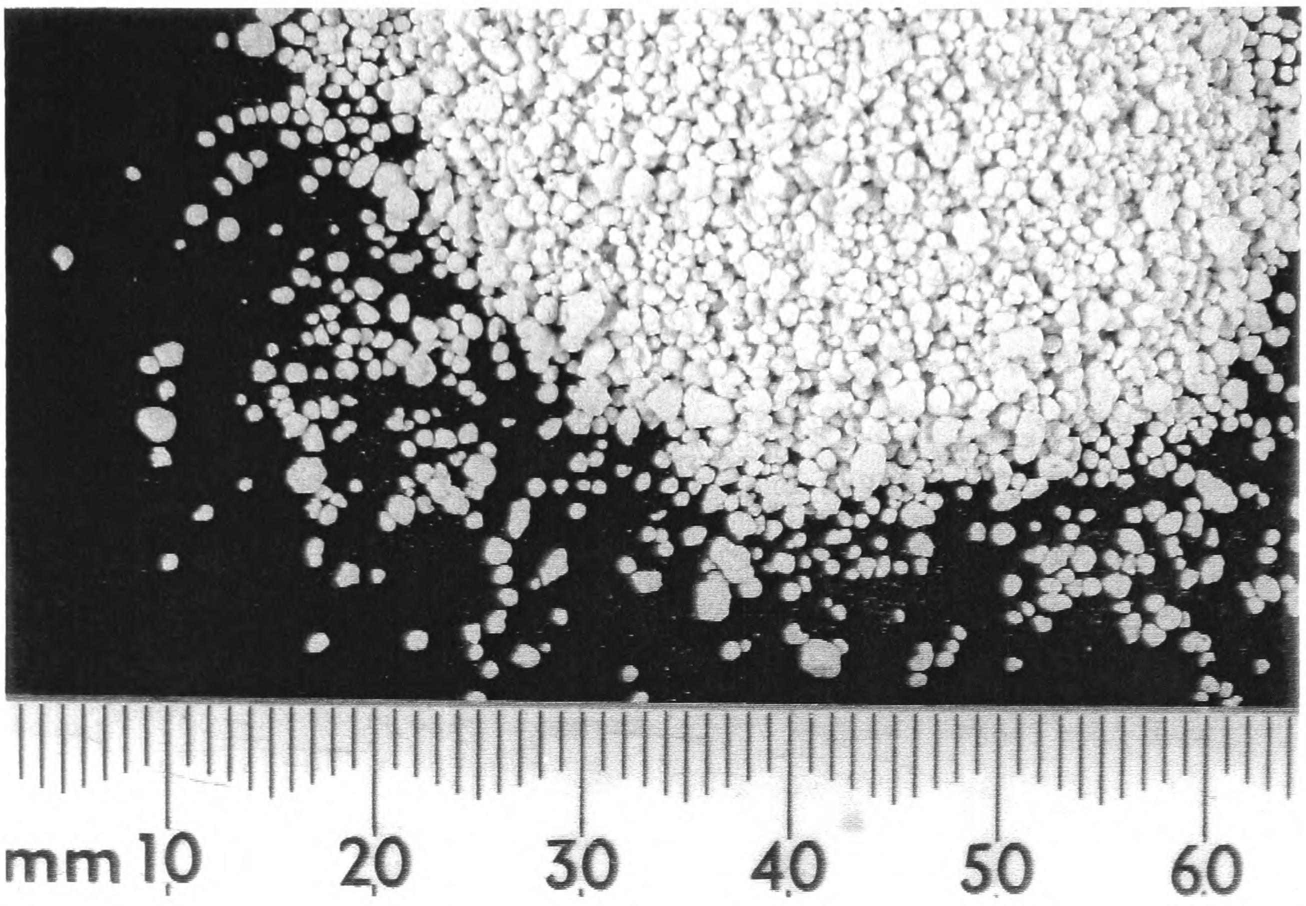
D(25%) - 1400 μm

D(50%) - 1700 μm

D(75%) - 2000 μm

D(90%) - 2200 μm

3.5.4 Dicalcium Phosphate



Material Characteristics :-

Particle Density - 1940 kg/m^3 Shape - Irregular

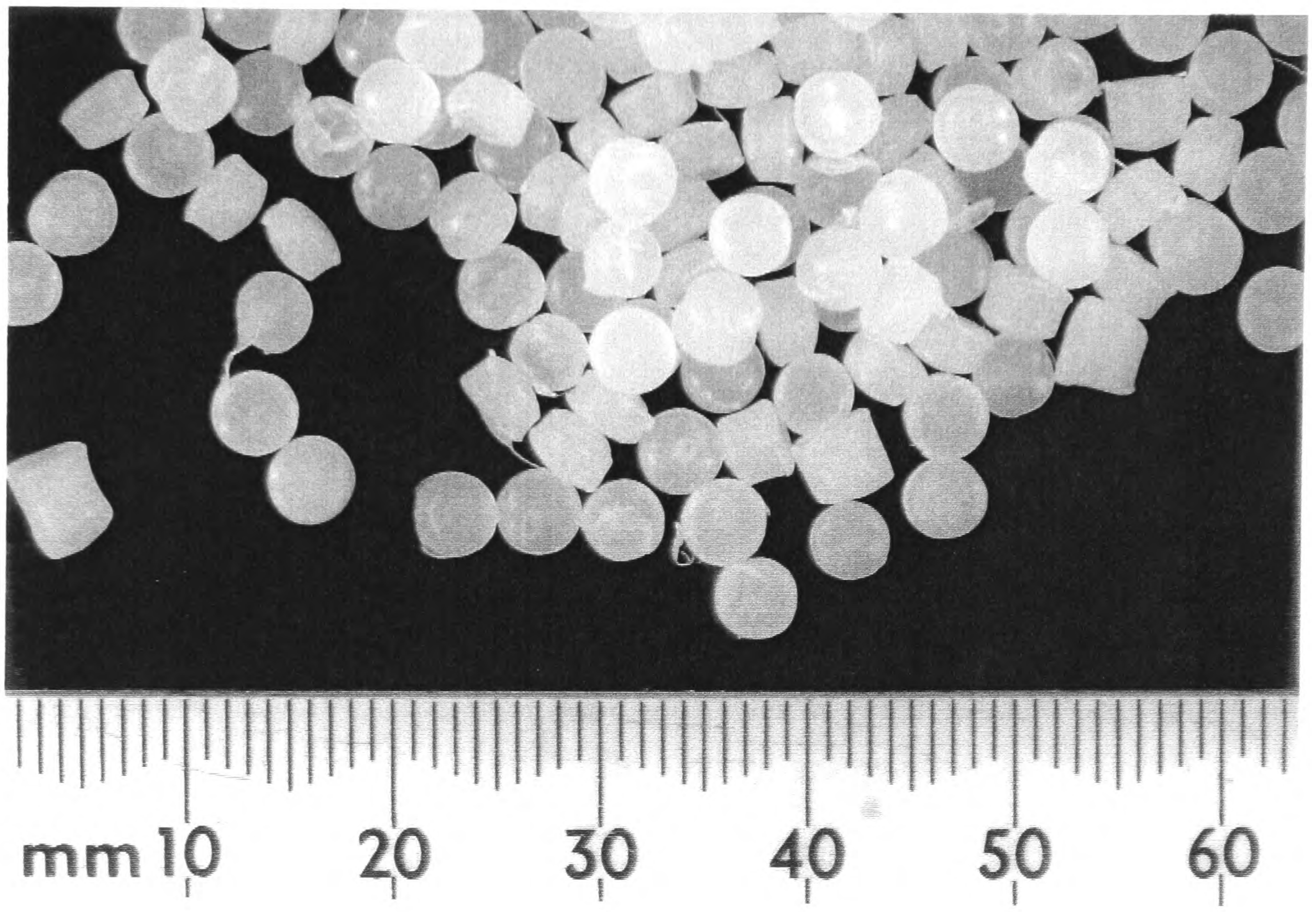
Poured Bulk Density - 840 kg/m^3

Poured Angle of Repose - 30°

Drained Angle of Repose - 38°

Size Distribution :-	D(10%) - 630 μm		D(75%) - 1200 μm
(by mass)	D(25%) - 750 μm		D(90%) - 1500 μm
	D(50%) - 940 μm		

3.5.5 Polyethylene Pellets (Rigidex)



Material Characteristics :-

Particle Density - 880 kg/m^3

Poured Bulk Density - 518 kg/m^3

Shape - Cylindrical (Mono-sized)

Poured Angle of Repose - 30°

Drained Angle of Repose - 28°

Size Distribution :- (by mass)

D(10%) - 2500 μm

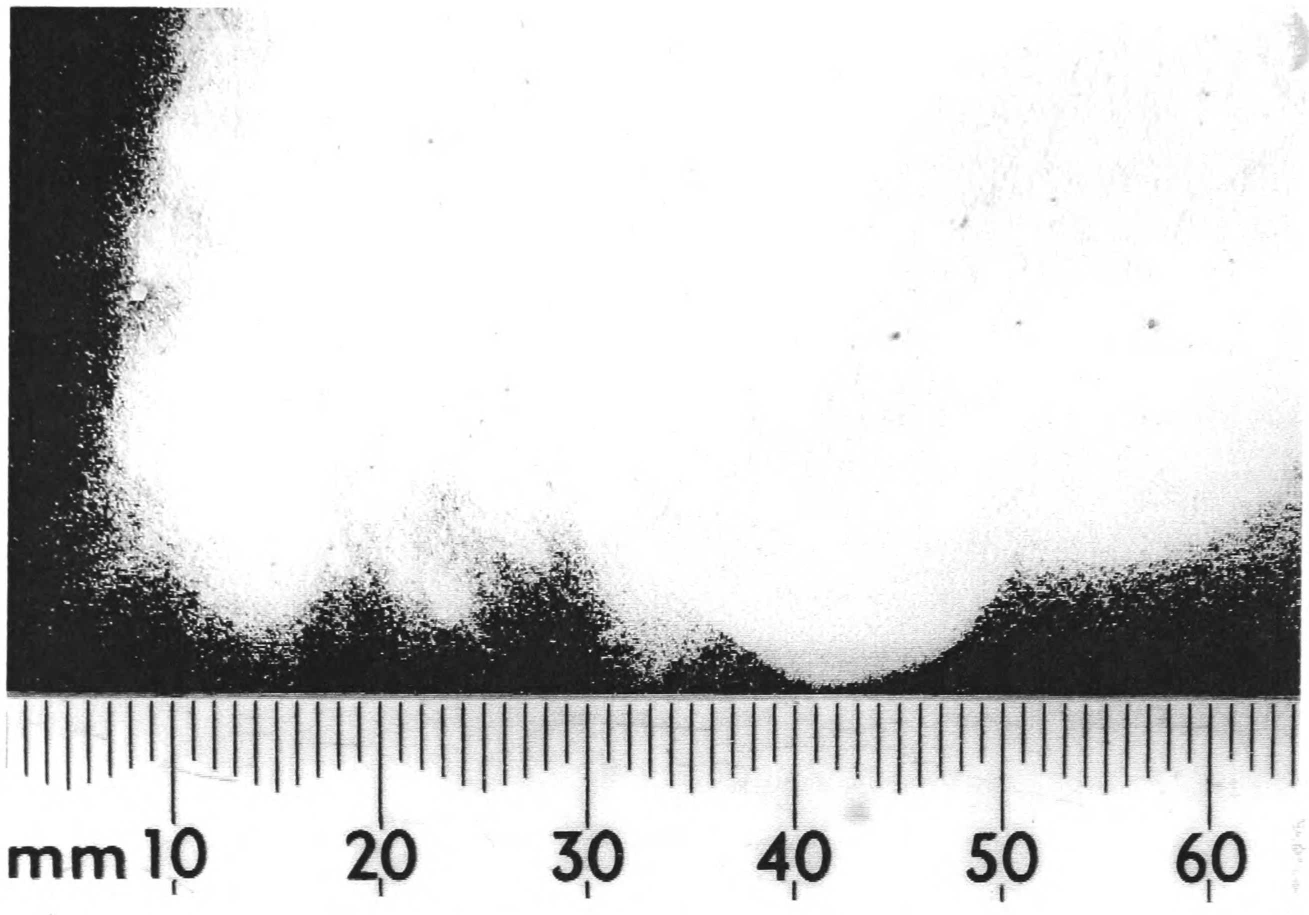
D(25%) - 2700 μm

D(50%) - 3000 μm

D(75%) - 3700 μm

D(90%) - 4000 μm

3.5.6 Aliminium Hydroxide



Material Characteristics :-

Particle Density - 2455 kg/m^3 Shape - Irregular

Poured Bulk Density - 995 kg/m^3

Poured Angle of Repose - 35°

Drained Angle of Repose - 48°

Size Distribution :- D(10%) - 20 μm D(75%) - 58 μm

(by mass) D(25%) - 36 μm D(90%) - 64 μm

D(50%) - 50 μm

3.6 Discharge of Products Through A Circular Orifice

This test rig was constructed to measure the discharge rate of the test products flowing under gravity from a flat bottomed bin, through orifices of different sizes. It also enabled the drained angle of repose to be measured. It was proposed that this data would be used during the analysis of the results of the vacuum conveying trials, as a guide to the rate at which products might flow to the suction nozzle itself.

The rig consisted of a 25 gallon (113 litre) capacity oil drum with a suitable bracket attached to the top allowing it to be suspended from the load cell as shown in Figure 3.12.a. In fact the load cell and supporting framework are those used with the small scale flow visualisation test rig. The facility was arranged such that a large hole in the bottom of the bin could be covered by any one of six plates, each with a differing size orifice. Figure 3.12.b shows the largest and smallest size of orifice plates used. The hole diameters are in 25mm (1 inch) increments from 25mm to 100mm, with two extra orifice plates having 60.5mm and 114mm diameter holes, representing the diameters of the inner conveying pipe and outer sleeve of the suction nozzle respectively. A simple trap door with a quick release mechanism is used to initiate flow from the hopper. The discharge rate was measured using the load cell and chart recorder, with the slope of the resulting trace representing the rate of discharge.

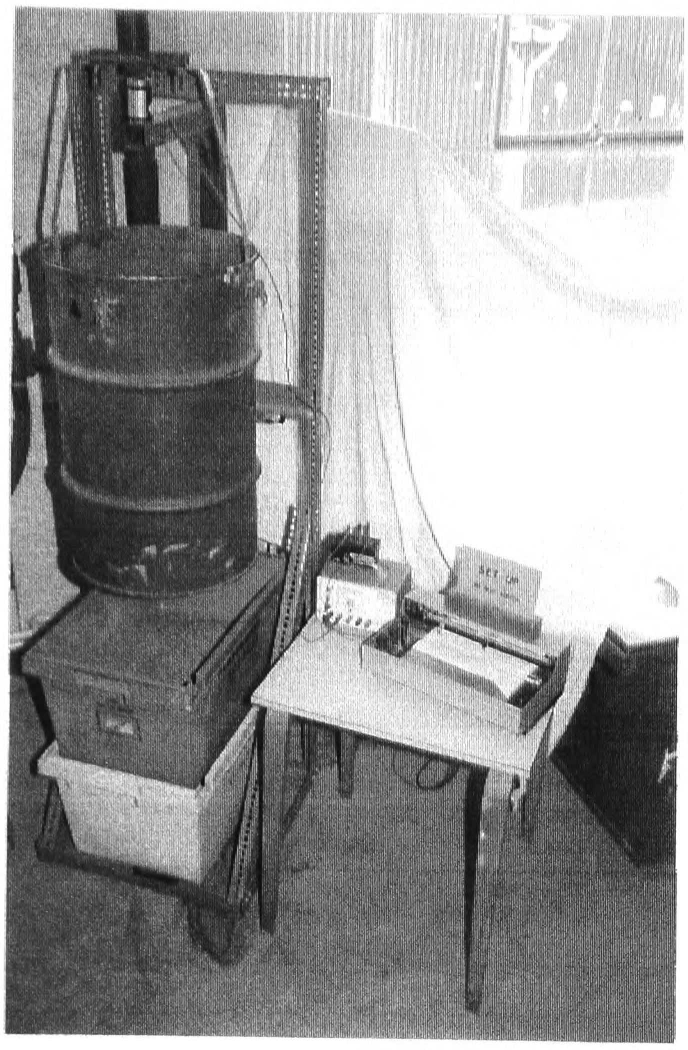
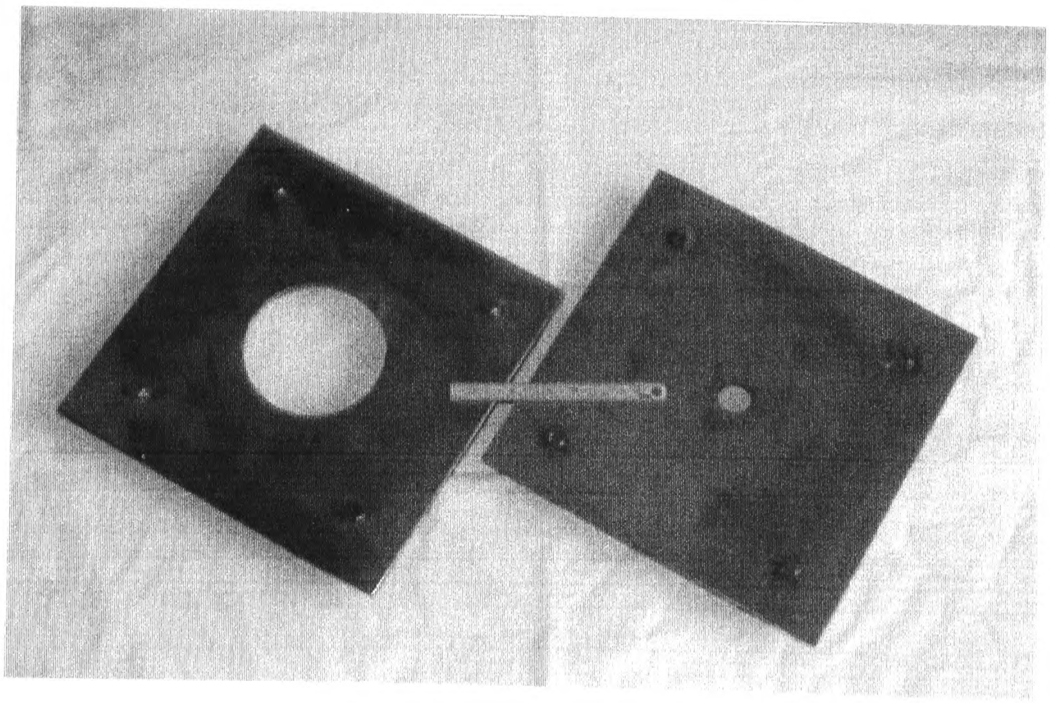


Fig. 3.12.a

Test Rig Used for Determining the Rate of Discharge of Products Through an Orifice

Fig. 3.12.b Typical Orifice Plates Used



3.7 Experimental Procedures

This section outlines the experimental procedures for calibrating and operating the various facilities described in the previous sections.

3.7.1 Load Cell Calibration

The general procedure for calibrating the load cells is exactly the same for both test rigs and the product discharge rate test rig.
The load cells were energised using low voltage regulated power supplies. The output of the cell(s) were connected to a chart recorder, enabling the variation in load applied to the cell(s) to be recorded as a displacement of the pen. The load cells are calibrated by hanging a known mass on the cell (or hopper) and noting the pen displacement on the chart recorder. Since material is being drawn out of the hopper by the suction nozzle, the mass of the hopper will be decreasing as a test proceeds, thus the load cell was calibrated for decreasing loads. By removing the masses incrementally the pen displacement was noted. By plotting a graph of applied load against pen displacement for both increasing and decreasing loads, the slope of the graph was the measuring system calibration constant, the units being kg/mm. As it is the rate of unloading which is important, it is necessary to calculate the time taken to discharge a given mass of product. Thus, using the paper speed of the chart recorder to represent time, and the pen displacement to represent mass of product removed, the slope of the trace drawn will represent the mass flow rate of product being drawn out of the hopper corresponding to a given set of conveying conditions and nozzle geometry.

3.7.2 Procedure for Operating Small Scale Rig

a/. The nozzle geometry was adjusted to an appropriate position such that the inner tube is either retracted within, level with, or protruding from the outer sleeve. A sign convention was adopted, whereupon a negative sign would represent a retracted nozzle configuration, a positive sign would represent a protruding nozzle configuration and a zero would represent a level nozzle. The inner movable nozzle was locked in position by means of pinch screws fitted through the wall of the outer sleeve.

b/. The throttling attachment (when fitted) was adjusted to give the desired throttling effect as related to the test to be undertaken.

c/. The hopper was filled with the test material to the same height for each test, thereby ensuring a constant head of material above the nozzle inlet throughout the entire test programme.

d/. The datum level of the mercury filled U-tube manometer was noted.

e/. The filter was cleaned after each test, by turning on the mechanical shaker attached to the bottom of each filter bag. This ensured, to a certain extent, that the pressure drop across the filter is minimised prior to commencing each test.

f/. For two reasons the ball valve for adjusting the air flow through the system was completely closed prior to turning on the exhauster,

i). the float within the rotameter could be damaged if it hit the top of the rotameter casing due the sudden flow of air through it, and ii). the air flow through the system should be gradually increased from zero.

g/. Once the exhauster was running the globe valve was slowly opened until the desired air flow (pick-up velocity at the nozzle inlet) was achieved. The chart recorder was started after the exhauster was running but before the globe valve was opened.

h/. The manometer was read during the steady state part of the test, with the value indicating the pressure drop in the conveying line. The resulting trace on the chart recorder represented the mass flow rate of product being drawn out of the hopper by the suction nozzle.

i/. The procedure was repeated for each combination of nozzle position, air velocity and throttle setting .

j/. The calibration constant was checked periodically by loading the hopper with known amounts of the test product.

3.7.3 Procedure for Operating Pilot Scale Rig

a/. The nozzle geometry and throttle were adjusted to the desired conditions. As previously described, this was achieved by raising or lowering the outer sleeve and locating it in position by placing the three split pins in the appropriate holes of the adjusting rods. Finer adjustment than the standard 20mm increment could be accomplished by using the 10mm or 5mm shims. The throttle was altered by removing the throttle disc holder from the inlet to the annular chamber, placing the appropriate throttle disc in the holder and then replacing it.

b/. The datum levels of all manometers on the manometer board were either adjusted to give zero readings or noted.

c/. The supply hopper was filled with the test product being used, to the same height each time.

d/. The pneumatically operated ball valve mounted in the conveying line was closed, the vacuum receiver hopper relief valve was fully opened and the air bleed valve was opened.

e/. The Roots exhauster was started under "no-load" conditions, as a result of d/.

f/. The air bleed valve was adjusted to give the desired air velocity for the particular test being undertaken.

g/. The chart recorder was turned on, the conveying line ball valve opened and the vacuum relief valve shut.

h/. All manometer/gauge readings were taken once the steady state conveying rate was been reached. This was denoted by the exhauster inlet vacuum gauge giving a steady reading.

i/. At the end of the test run the system was shut down by turning off the chart recorder, opening the air bleed valve, opening the vacuum relief valve and closing the in-line ball valve.

j/. The "back wash" filter cleaning cycle was performed and finally the exhauster was turned off.

k/. The above steps were repeated as many times as was necessary to complete the test programme for that product.

1/. The load cell calibration was checked periodically by loading the supply hopper with known calibration masses.

3.7.4 Procedure for Operating Gravity Discharge Rate Test Rig

a/. An appropriate orifice was bolted into the bottom of the test hopper, and then the hopper was suspended from the load cell.

b/. The hopper was filled with product, the chart recorder switched on and the orifice outlet uncovered. The trace recorded represented the discharge rate through the orifice.

c/. After the test, the drained angle of repose was calculated.

d/. Prior to the hopper being refilled with product, the sloping free surface remaining in the hopper, forming the boundary between the static and flowing regions of material was dislodged.

e/. This procedure was repeated several times to obtain an average discharge rate and also an average drained angle of repose.

f/. The above steps were repeated for each orifice size and product.

CHAPTER 4. FLOW VISUALISATION STUDIES

4.1 Introduction

This chapter presents the findings of the initial programme of work which was undertaken with the objective of gaining an insight into the effect of the variables influencing the operation of co-axial tube suction nozzles. For this work, the small scale test rig described in section 3.3 was used. The test rig was constructed, having a semi-circular perspex suction nozzle, thus allowing a visual inspection and comparison of the flow patterns resulting for differing operating conditions.

Section 4.2 of this chapter is concerned with the effects of nozzle geometry on the system performance characteristics of what the author refers to as a "standard" suction nozzle, as shown in Figure 4.1.a. This is where there is no throttling of the air as it enters the annular space between the outer sleeve and the inner pipe. Section 4.3 deals with the effects of throttling the atmospheric air at the inlet to the annular chamber of a standard suction nozzle, as shown in Figure 4.1.b, for varying nozzle geometries. Section 4.4 assesses the outcome of these initial test programmes with a view to producing a list of criteria for optimising the performance of a concentric tube suction nozzle.

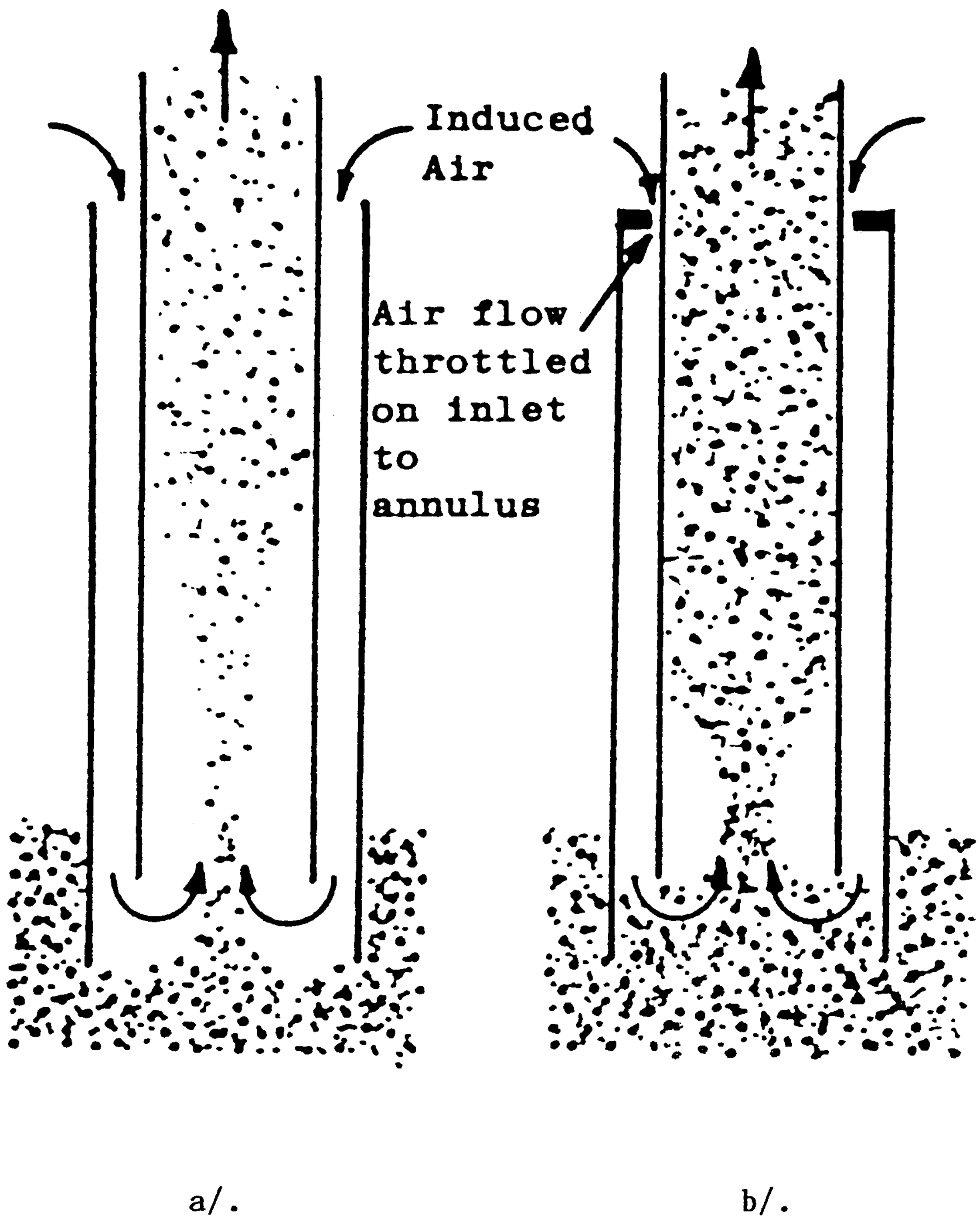


Fig. 4.1 Typical Modes of Operation of The Co-axial Tube Type Suction Nozzle

4.2 Effect of Nozzle Configuration on System Performance

4.2.1 Alumina

During the testing of the alumina powder, it was noticed that all of the air entering the system seemed (but was not verified) to be flowing down the annular chamber to the inlet of the nozzle and thereby not through the bulk of the product itself. It is suggested that this was because the air was taking the path of least resistance due to the permeability of the alumina being low. By referring to Figure 4.2, showing the relationship between the nozzle geometry and the mass flow rate of alumina for varying superficial pick-up velocities, it can be seen that the general trend is for an increase in the mass flow rate of alumina with increasing nozzle geometry, ie. moving from a retracted nozzle position to a protruding nozzle position, for a given superficial pick-up velocity. As mentioned in an earlier section a sign convention was adopted by the author to represent the position of the two tubes relative to one another. With the inner tube retracted inside the outer sleeve, a negative sign was adopted. With the inner tube protruding from the outer sleeve a positive sign was adopted. When the two tubes are level with each other, a zero was adopted.
The photographs shown in Figures 4.3, 4.4, 4.5 and 4.6 correspond to points A, B, C and D on Figure 4.2 respectively. Figure 4.3 (point A) corresponding to the suction nozzle operating in a retracted configuration of X= -12 mm (0.5 inches), clearly demonstrates that it is an inefficient feeding configuration. Even when operating at higher superficial pick-up velocities (typically 60 m/s) the mass flow rate of product is only 0.064 kg/s (0.23 t/hr). The common feature of Figures 4.3, 4.4, 4.5 and 4.6 is that the superficial pick-up velocity was kept constant at 40 m/s and only the nozzle geometry was altered ie. X= -12mm, 0mm, +12mm and +25mm respectively. Although only the geometry of the nozzle was altered, the feeding efficiency of the nozzle increased

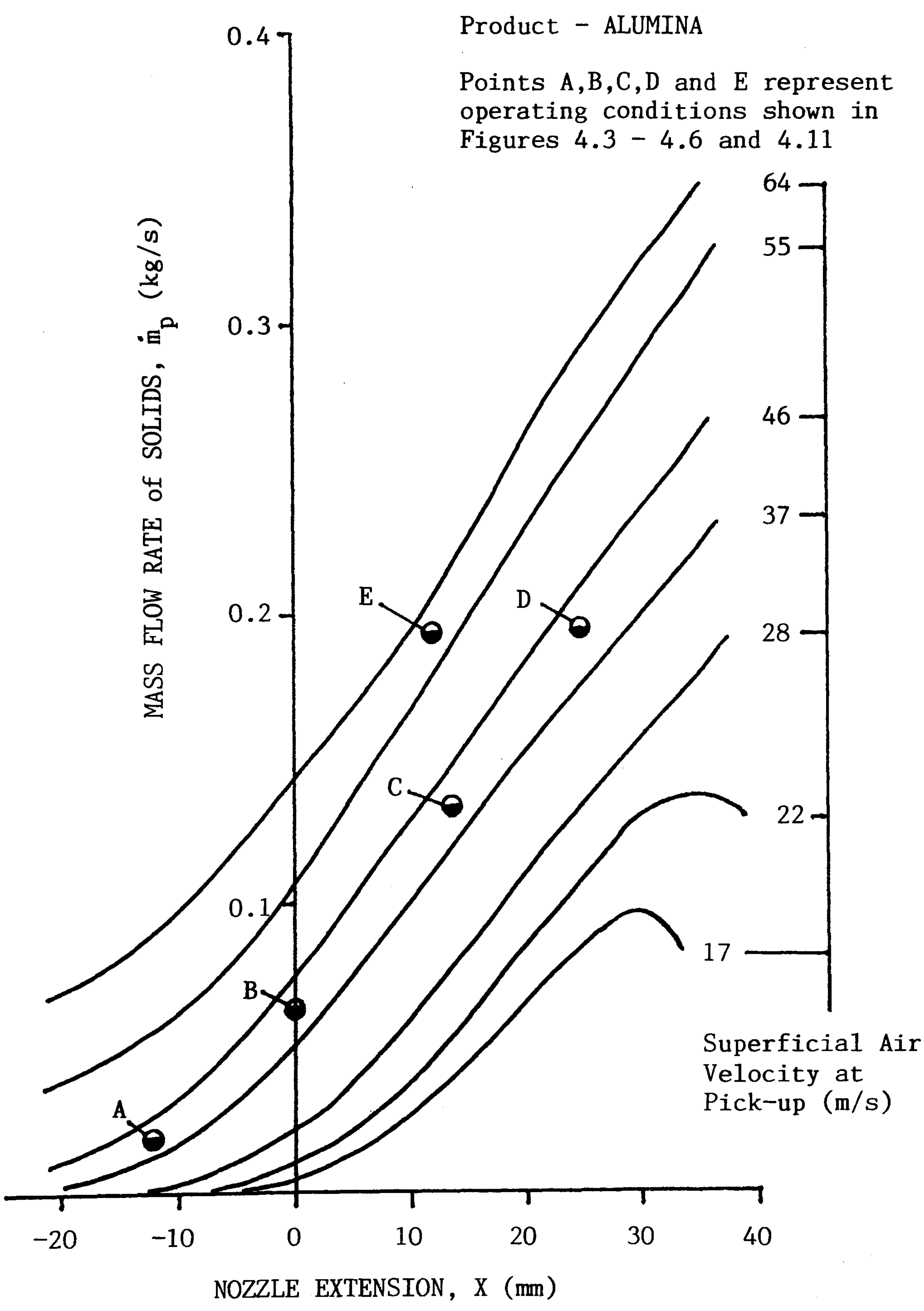


Fig. 4.2 Variation in Mass Flow Rate of Solids With Nozzle Extension for Differing Superficial Air Velocities

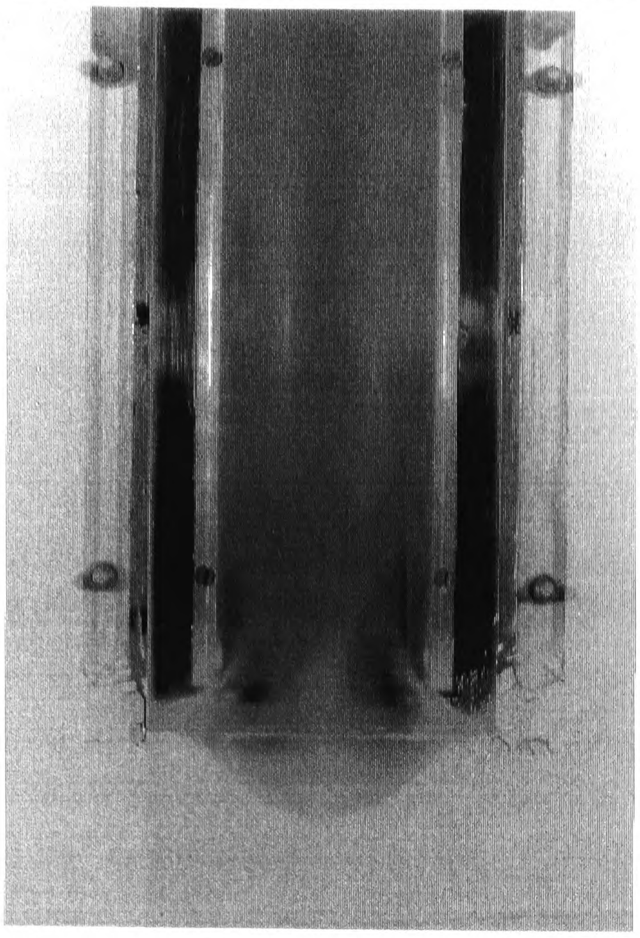


Fig. 4.3 Point A, X= -12.5mm
Superficial Air Velocity = 40m/s

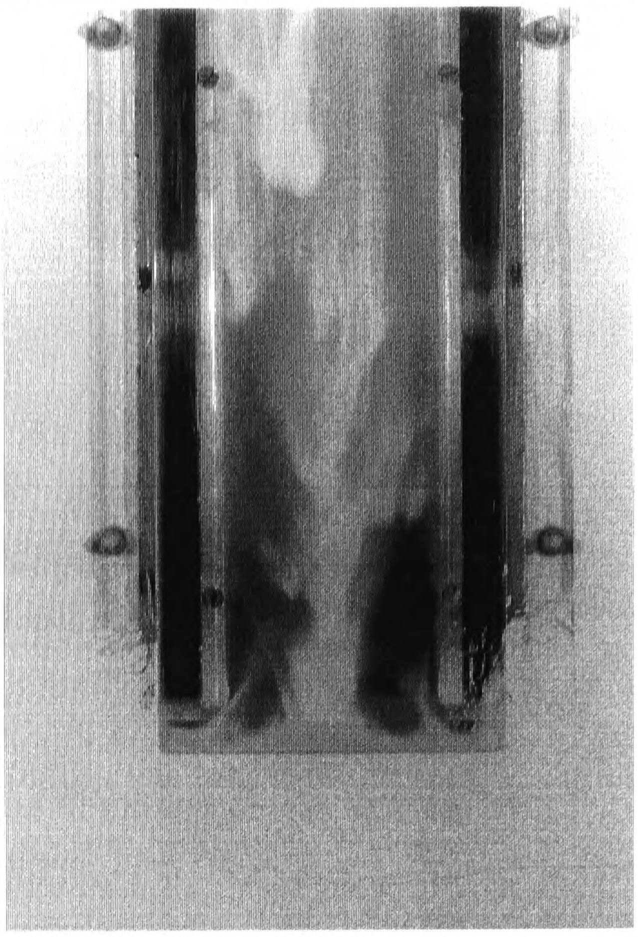


Fig. 4.4 Point B, X= 0mm
Superficial Air Velocity = 40m/s

Product - ALUMINA

Fig. 4.5 Point C, X= +12.5mm
Superficial Air Velocity = 40m/s

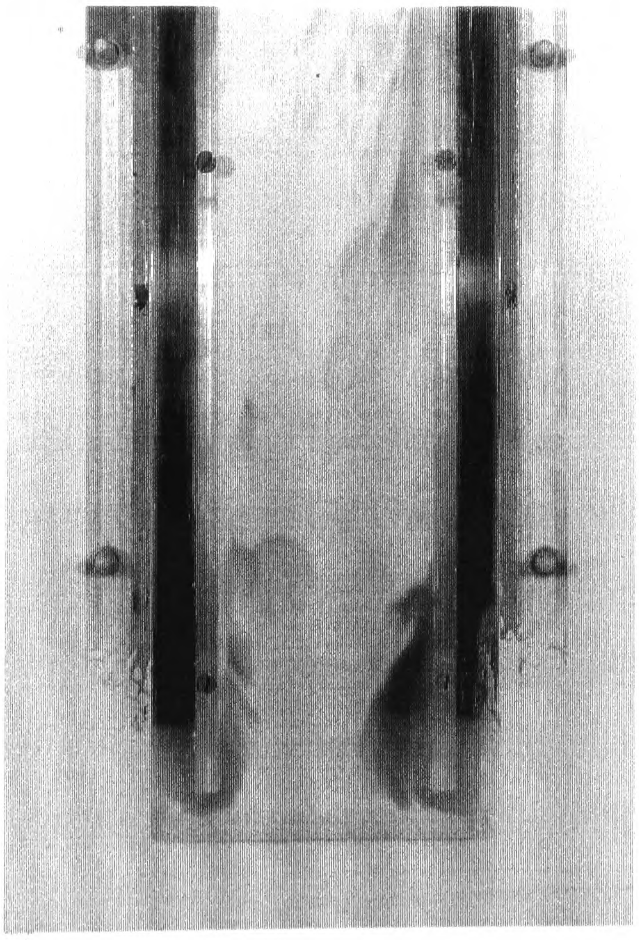
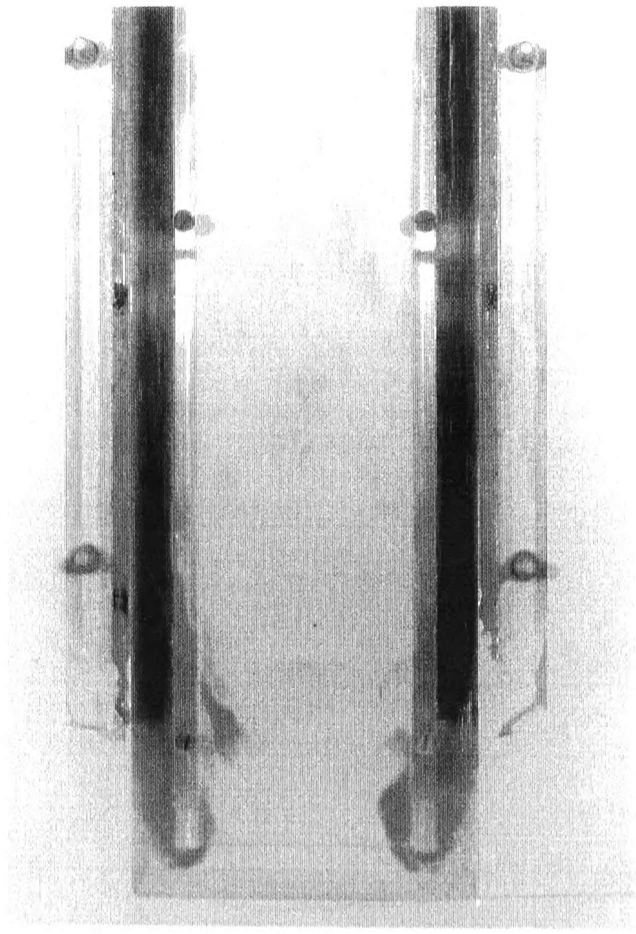


Fig. 4.6 Point D, X= +25mm
Superficial Air Velocity = 40m/s



significantly resulting in an 11 fold increase in the mass flow rate from point A to point D; with Figures 4.3 to 4.6 showing this increase quite clearly. It is suggested that the following is an explanation for the significant increase in the mass flow rate of product with increasing nozzle geometry.

With the nozzle in a retracted position the air flows down the annulus and turns through 180° to enter the inner conveying pipe. By doing so the air scours the free surface of the alumina immediately beneath the suction nozzle, thereby entraining only a small quantity of product into the air stream, as depicted in Figures 4.3 and 4.7. With the nozzle in a level position X= 0mm, the air now has to pass through the material from the annulus to the inner conveying pipe resulting in a greater quantity of alumina being entrained, as shown in Figures 4.4 and 4.8. If the nozzle geometry is increased still further to a protruding position, the material will fill the space immediately beneath the annulus as in Figure 4.9, but since the air is flowing down the annulus the material falls into the moving air stream and is immediately entrained, see Figures 4.6 and 4.10. It is clear to see that for these tests to be successful, the material must be sufficiently free flowing and be capable of flowing to the nozzle at a rate equal to or greater than the take away rate by the pneumatic conveying system. Looking at Figures 4.6 (reprinted for clarity on page 76) and 4.11, corresponding to points D and E respectively, it would appear seem that the mass flow rate of material for Figure 4.6 (X= +25mm, 40 m/s) is greater than that of Figure 4.11 (X= +12mm, 60 m/s). This is because the alumina occupies more of the cross-section in view; but from Figure 4.2 it can be seen that the mass flow rates are equal (0.19 kg/s). This suggests that for a given required mass flow rate of material, a number of nozzle geometry/pick-up velocity combinations are possible.

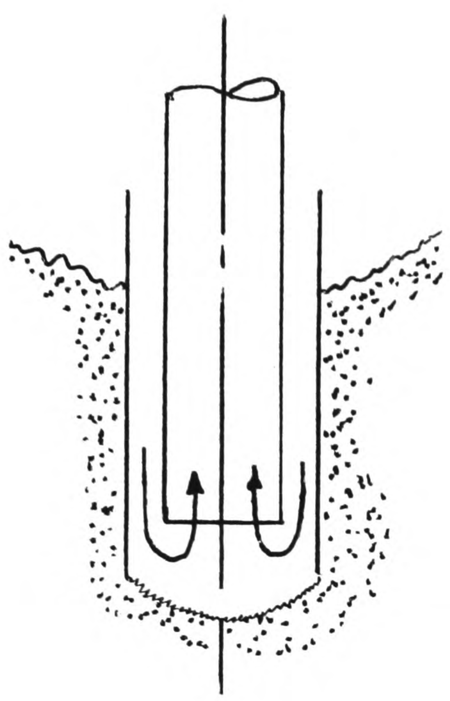


Fig. 4.7

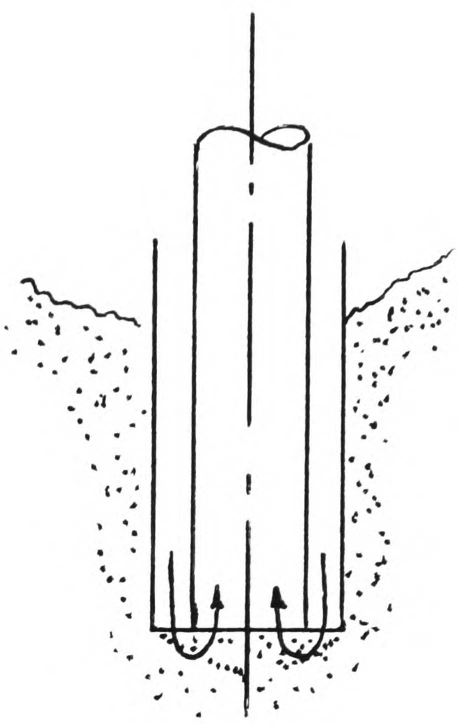
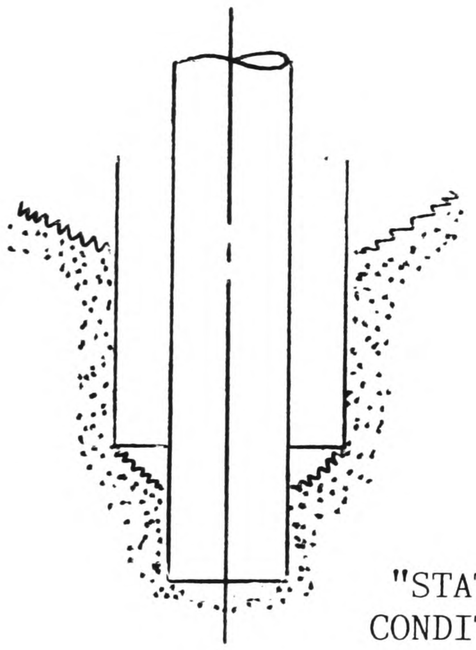


Fig. 4.8



"STATIC"
CONDITION

Fig. 4.9

"LIVE"
CONDITION

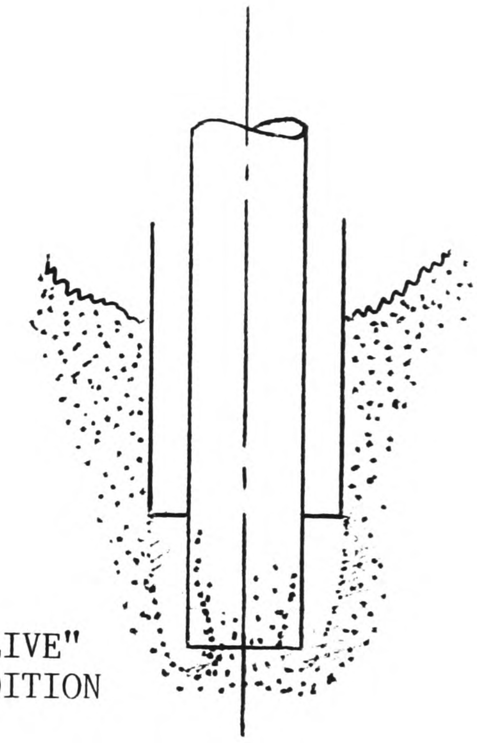


Fig. 4.10

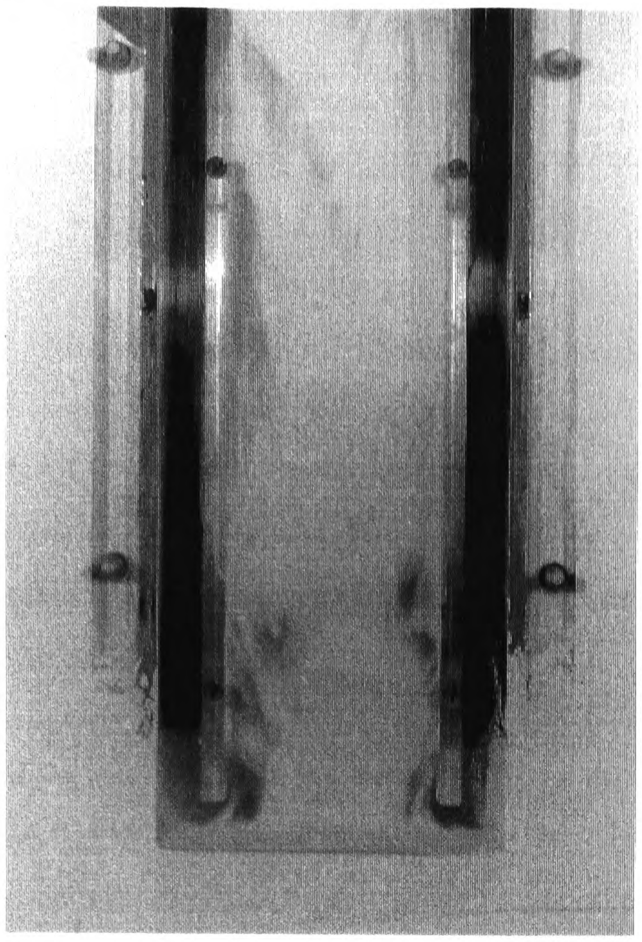


Fig. 4.11 Point E, X= +12.5mm
Superficial Air Velocity = 60m/s

Product - ALUMINA

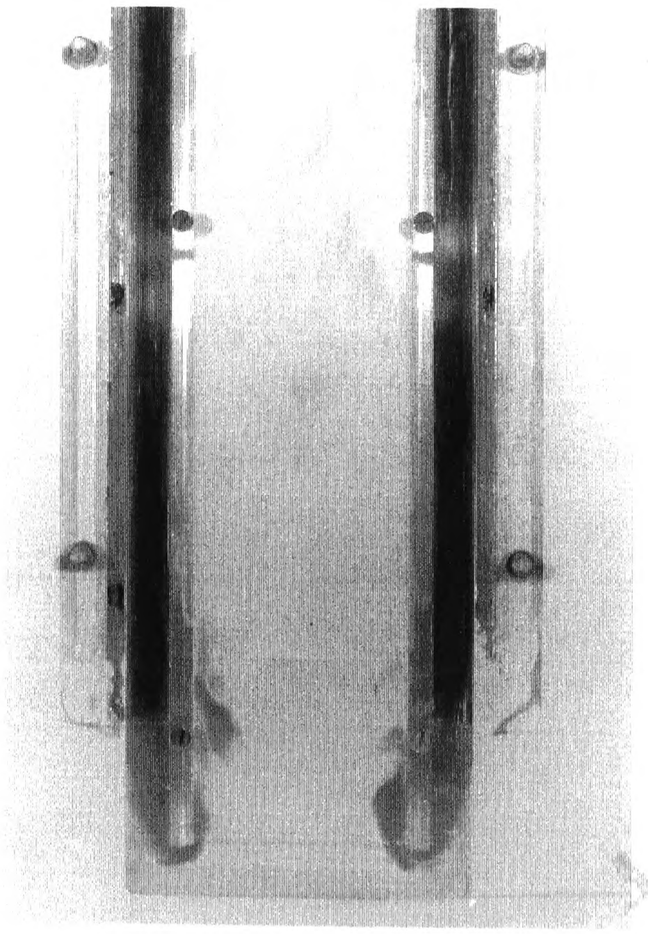


Fig. 4.6 (Reprinted for clarity)
Point D, X= +25mm
Superficial Air Velocity = 40m/s

Product - ALUMINA

This data may be re-plotted as shown in Figure 4.12. This shows the relationship between the mass flow rate of product against the mass flow rate of air for lines of constant nozzle position; with the conveying line pressure drops also being superimposed. Since a number of nozzle geometry/pick-up velocity combinations are possible to achieve the desired conveying rate, it is clear that higher pick-up velocities are necessary when using an inefficient feeding geometry compared with that when using an efficient feeding geometry. The significance of this is important because although points F and G in Figure 4.12 are conveying at the same rate, the overall conveying line pressure drop for point F is lower than point G. This is due to the fact that the "air only" portion of the total conveying line pressure drop will be less for point F, since the air velocity required will be lower to achieve the same overall conveying rate. As the system power consumption is a function of the conveying line pressure drop and volumetric flow rate of air, it is evident that the power consumption or specific power consumption can be reduced. Thus, by increasing the nozzle geometry and reducing the pick-up velocity, the specific power consumption can be reduced from 377.5 W/t/hr corresponding to point G to 260.8 W/t/hr for point F, ie. an approximate reduction in power of 31%.

It would appear from Figure 4.2 that if the nozzle configuration was continually increased in the positive direction, the mass flow rate of product would also continue to increase. However, this did not occur, since with any further increase in the nozzle extension past that required to achieve the maximum mass flow rate, the performance starts to decrease slightly and then the system chokes. This observation was in complete contrast to a paper presented by Foster (Ref 16) where the maximum conveying rate would be reached and then diminish to zero as shown in Figure 2.15, albeit for coal and not alumina.

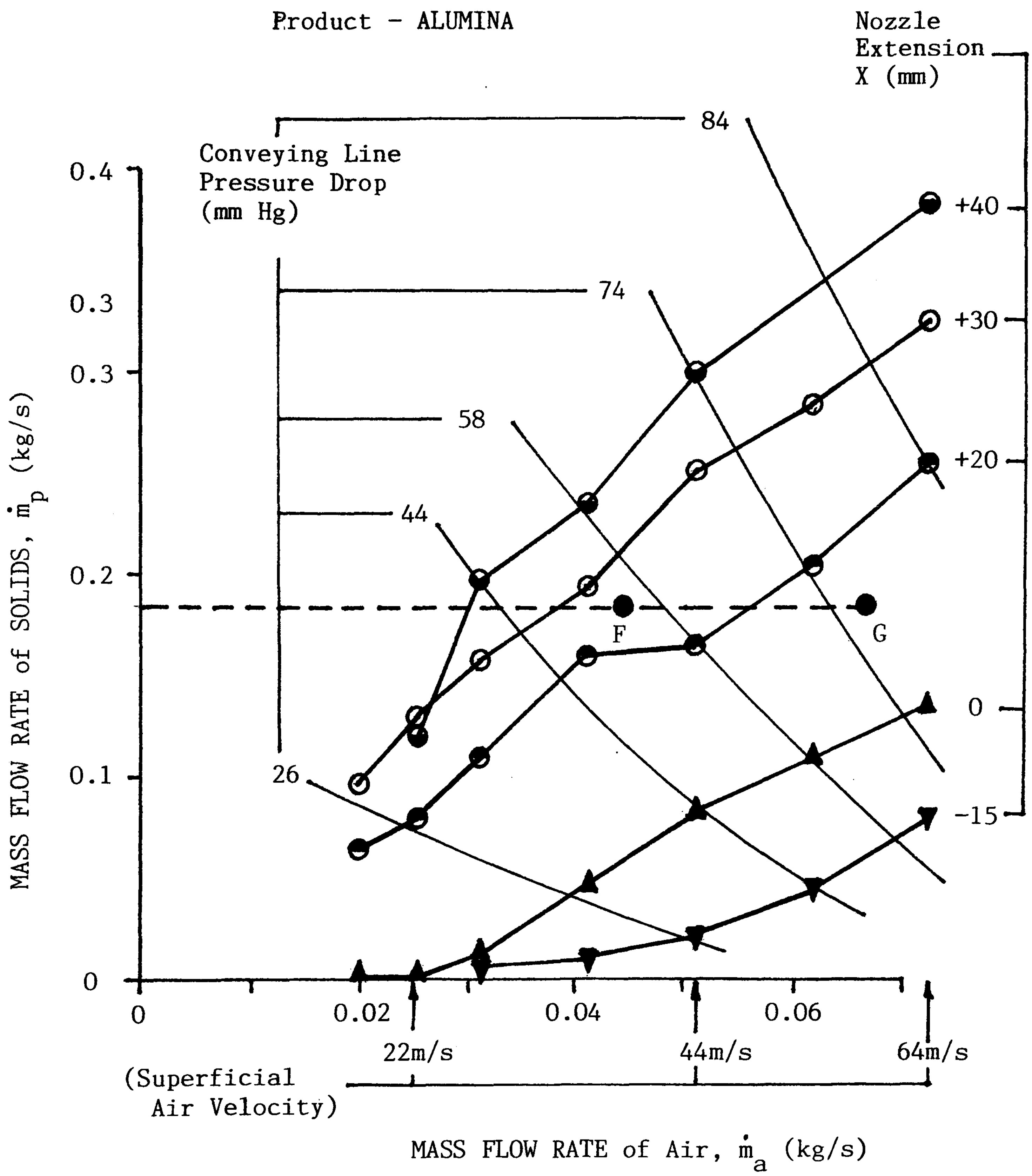


Fig. 4.12 Variation in Mass Flow Rate of Solids With Mass Flow Rate of Air for Differing Nozzle Extensions

4.2.2 Wheat Grain

Wheat grain was tested following the same procedure as that for the alumina. The most obvious difference between the two products was the trend of the curves as shown in Figure 4.13 for wheat grain and Figure 4.2 for alumina. For both products the mass flow rate of material increases with nozzle position until a maximum mass flowrate is achieved. However with the grain, instead of the throughput decreasing, with any further increase of nozzle position to a point when choking occurred for alumina the throughput remains nearly constant. This is again completely different from the results produced by Foster (Ref 16) where increasing the nozzle position past that required for maximum mass flow rate of coal, would cause a gradual decrease in mass flowrate down towards zero. With the grain the lowest pick-up velocity needed for reliable conveying was in the region of 30 m/s compared to only 17 m/s for the alumina. The reason for this is that to lift up a single particle in a vertical tube, the air flow around the particle must be such that the resulting aerodynamic drag forces acting on the particle are greater than the force of gravity. Hence, there is a minimum air velocity required to keep a particle in suspension. Since a single particle of wheat grain is heavier than a single alumina particle, the air entraining velocity should be greater. Another reason is, no doubt, due to the shape of the particles themselves. The grains are more aerodynamic than the alumina particles (see sections 3.5.2 and 3.5.1 respectively) and are, therefore, less likely to be entrained into the air stream because the drag forces will be lower for a given air velocity.

Figures 4.14, 4.15, 4.16 refer to points A, B, C on Figure 4.13 respectively and show again the same trend as for the alumina. Figure 4.17 (point D) shows quite clearly that there is little difference between it and Figure 4.16 (point C). Another interesting feature of the grains, is the way in which they point in the direction they are travelling, highlighting the regions of flow and no-flow of the

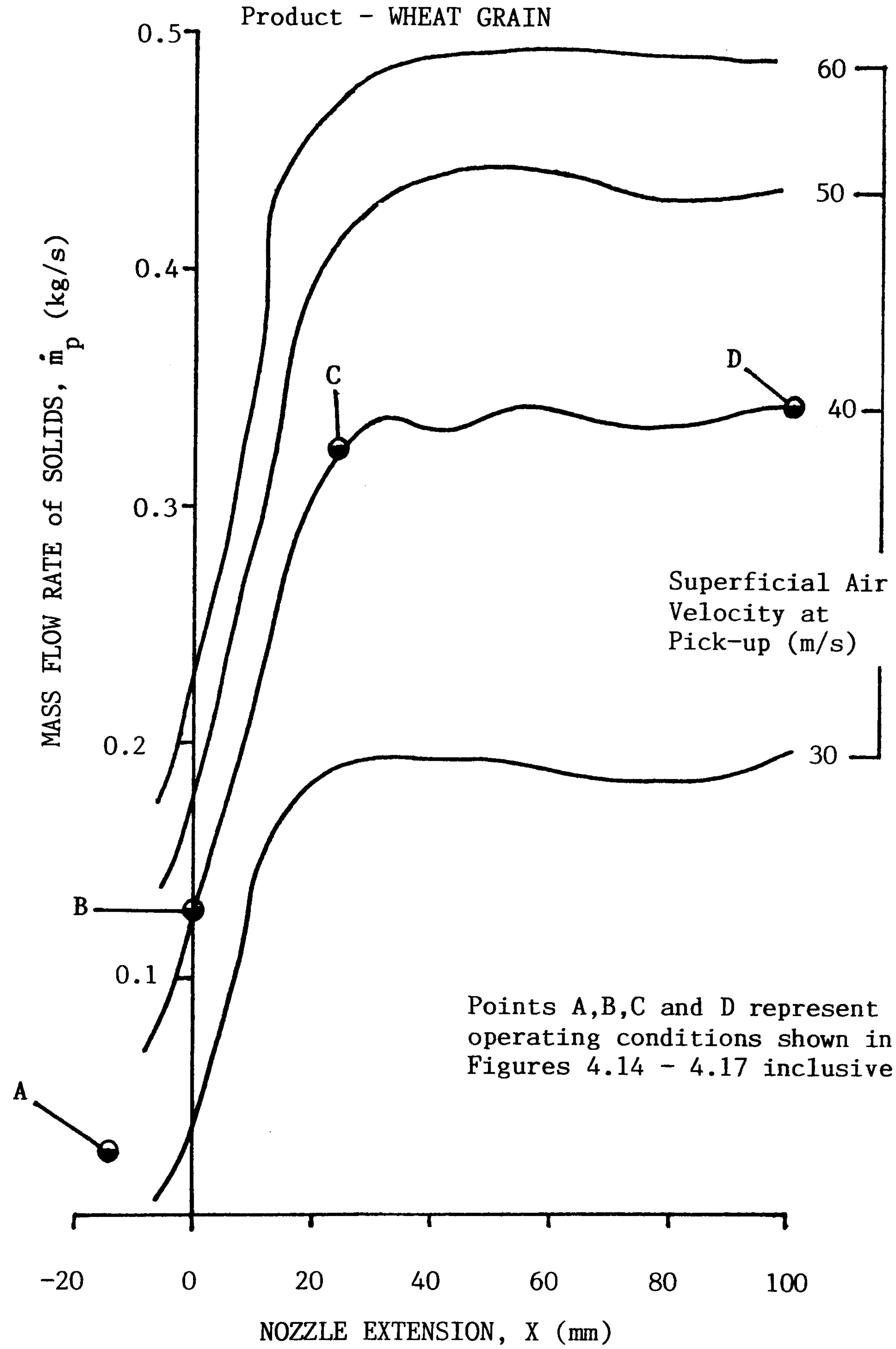


Fig. 4.13 Variation in Mass Flow Rate of Solids With Nozzle Extension for Differing Superficial Air Velocities

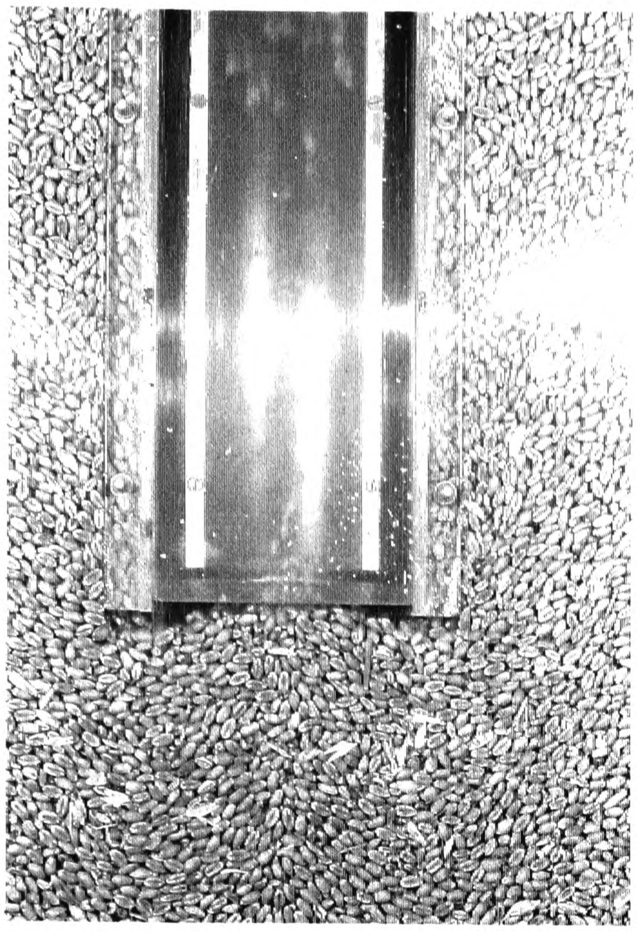


Fig. 4.14 Point A, X= -15mm
Superficial Air Velocity = 40m/s

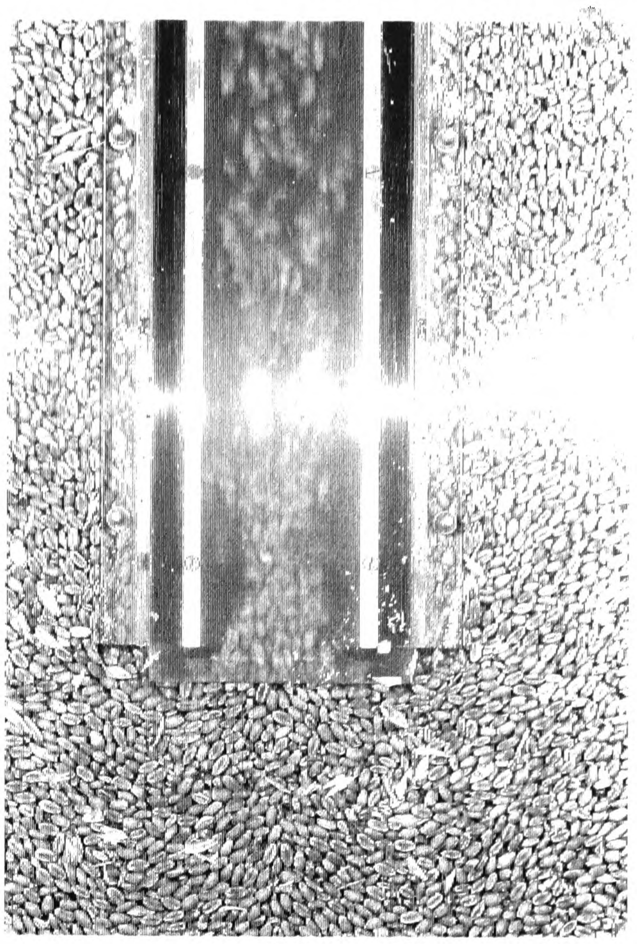


Fig. 4.15 Point A, X= 0mm
Superficial Air Velocity = 40m/s

Product - WHEAT GRAIN

Fig. 4.16 Point A, X= +30mm
Superficial Air Velocity = 40m/s

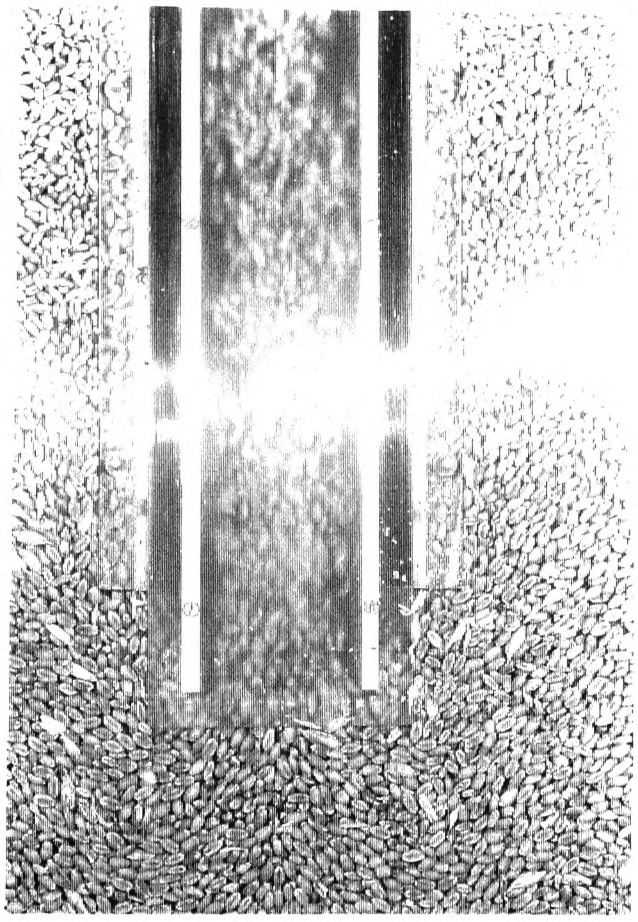
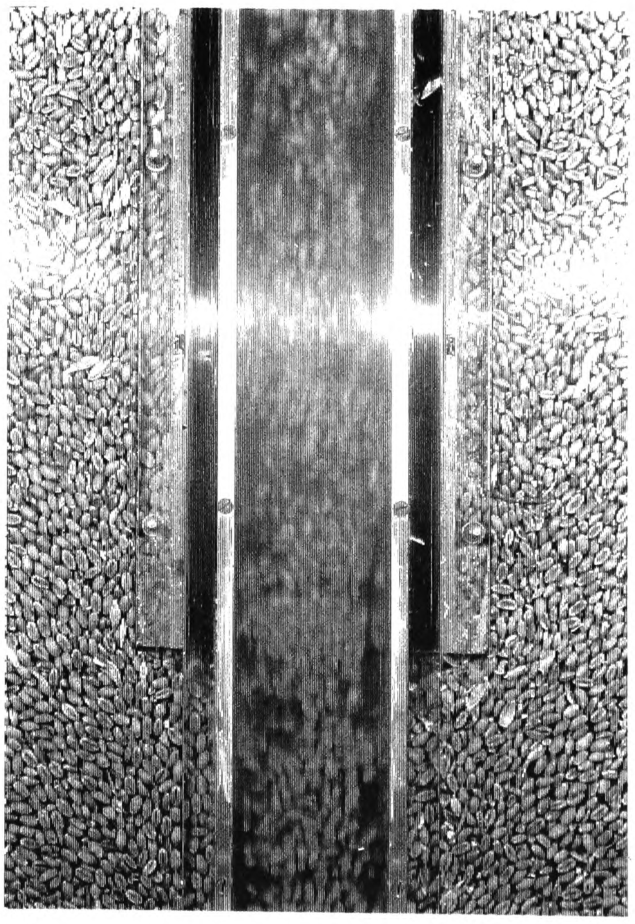


Fig. 4.17 Point A, X= +100mm
Superficial Air Velocity = 40m/s



individual grains within the vessel from which they are being entrained. Figure 4.17 shows that the grain occupies the space immediately beneath the the outer annulus, even when the system is running, whereas this is not so for alumina (Figure 4.10). An explanation for this is that with the alumina, all of the air entering the nozzle flows down the annular chamber because little or no air at all can flow through the interstices within the bulk of the alumina; whereas in the case of the grain, no air was felt flowing down the annular chamber. This suggests that since the interstices of the grain are relatively large, the air can flow through the bulk of the grain as easily as flowing down the annular chamber. This explains why the individual grains flow towards the wall of the inner pipe of the suction nozzle during operation and also orientate themselves in the region surrounding the nozzle.

With the suction nozzle geometry level and low pick-up velocities (typically less than 20 m/s) no grain was conveyed; but it was found, by accident, that if the annulus in the outer sleeve was blocked off, grain could be entrained. This suggests that most of the air must have been flowing through the bulk of the product itself and in doing so, the air flow over the individual grains helps to move them towards the inlet region of the nozzle. Figure 4.18 shows the relationship between the mass flow rate of product against mass flow rate of air. As with the results for alumina (Figure 4.12) the effect of nozzle geometry on air flow rate, conveying line pressure, and hence system pressure drop is clear.

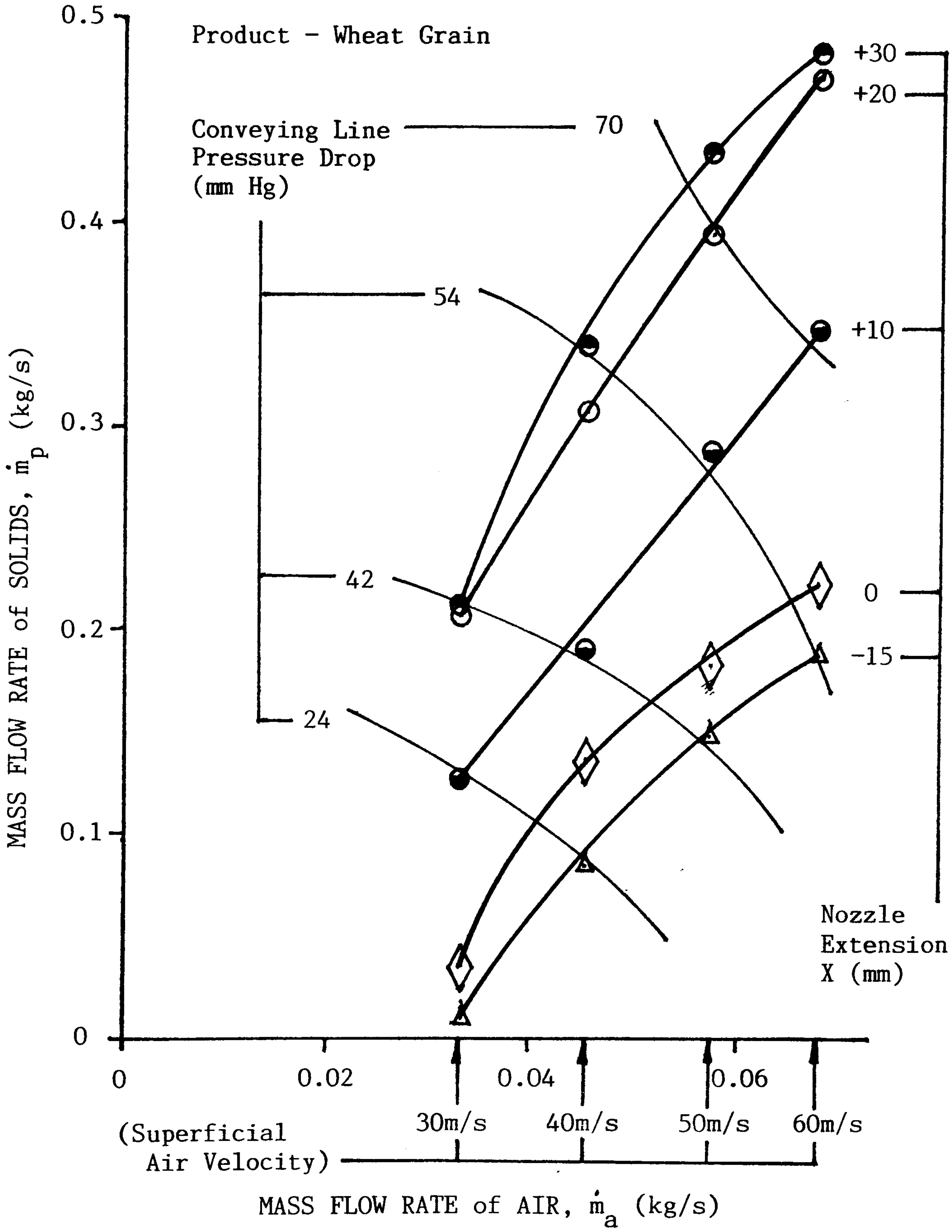


Fig. 4.18 Variation in Mass Flow Rate of Solids With Mass Flow Rate of Air for Differing Nozzle Extensions

4.2.3 Mustard Seed

Mustard seed was used since it is mono-sized and of a regular spherical form. The results have been plotted (Figures 4.19 and 4.20) in the same format as for alumina and wheat grain with Figures 4.21, 4.22, 4.23 and 4.24 corresponding to points A, B, C and D respectively. From Figure 4.19, it may be seen that the mass flow rates are generally higher than those of alumina and wheat grain at a given set of nozzle conditions. However, it may be seen that the mustard seed can be conveyed at lower pick-up velocities than the wheat grain, typically 20 m/s. In common with wheat grain, the mustard seed follows the same general trend of increasing mass flow rate with increasing nozzle position, until a nozzle configuration is reached whereby the mass flow rate remains largely constant irrespective of any further increases in nozzle position.
Two possible inter-related reasons for the observed higher mass flow rates for a given set of nozzle conditions are,

1/. Aerodynamic forces acting on an individual particle.

Since the mustard seeds are smaller in size, the drag forces acting on them will be lower when compared to those acting on the wheat grain.

2/. Flowability of a material

Mustard seed flows more readily than either the wheat grain or alumina. As a consequence the mustard seed is able to feed the nozzle at a rate equal to the rate at which the mustard seed is being drawn into the conveying pipeline.

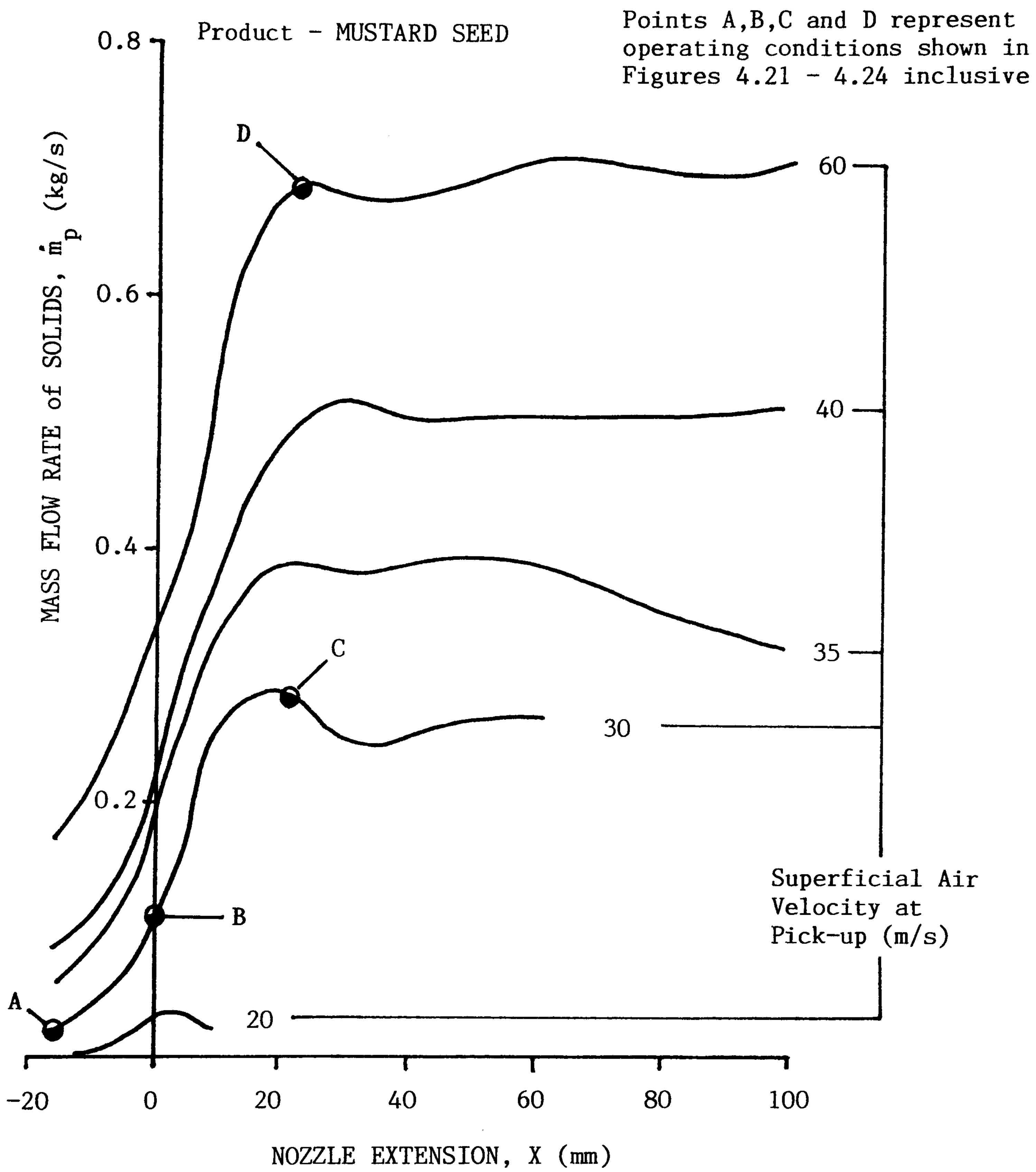


Fig. 4.19 Variation in Mass Flow Rate of Solids With Nozzle Extension for Differing Superficial Air Velocities

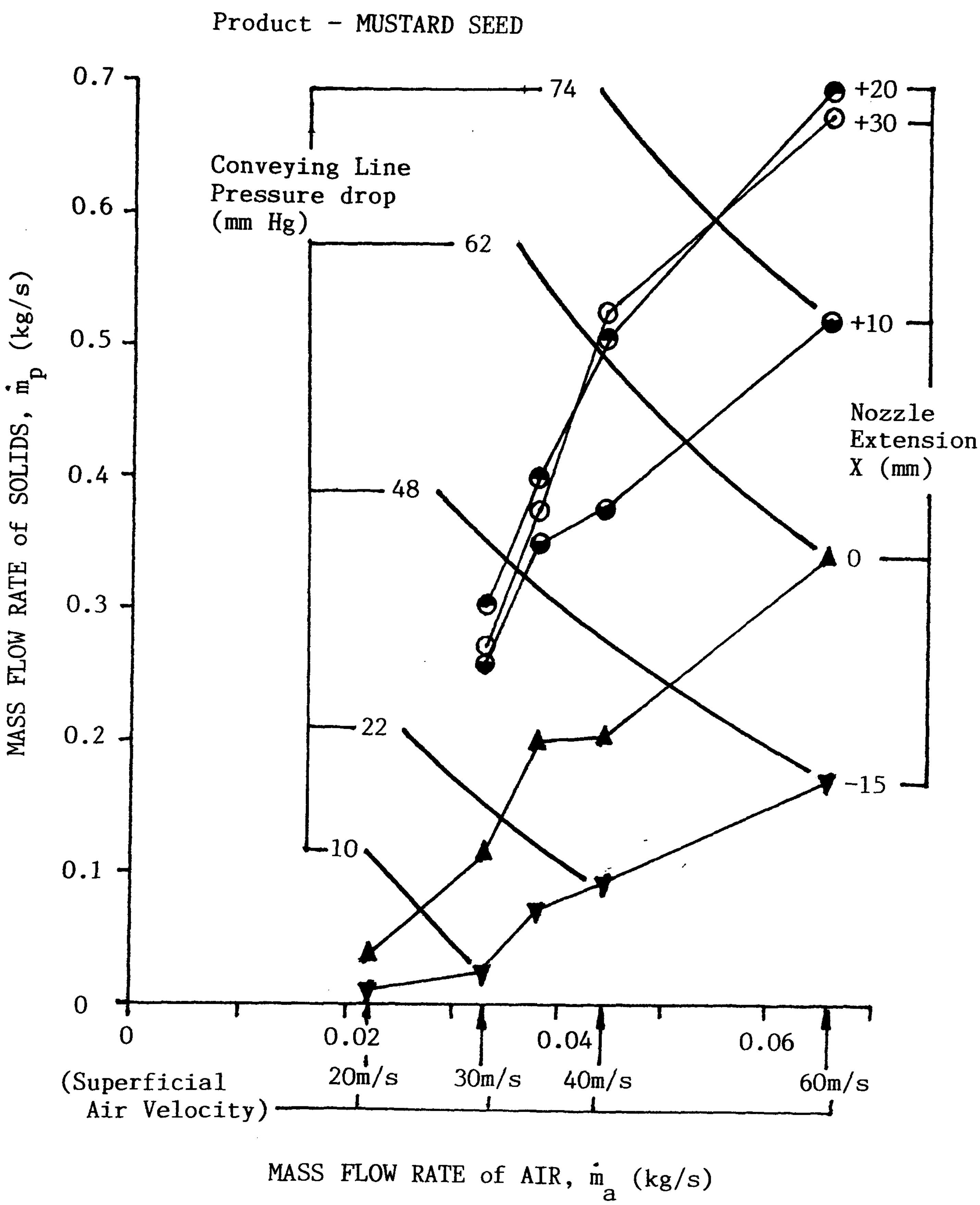


Fig. 4.20 Variation in Mass Flow Rate of Solids With Mass Flow Rate of Air for Differing Nozzle Extensions

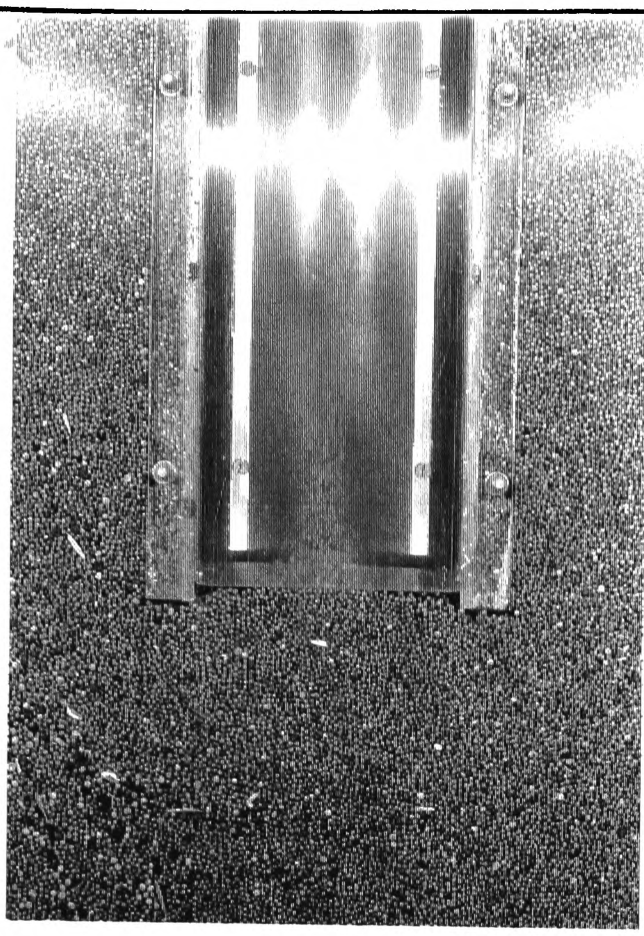


Fig. 4.21 Point A, X= -15mm
Superficial Air Velocity = 30m/s

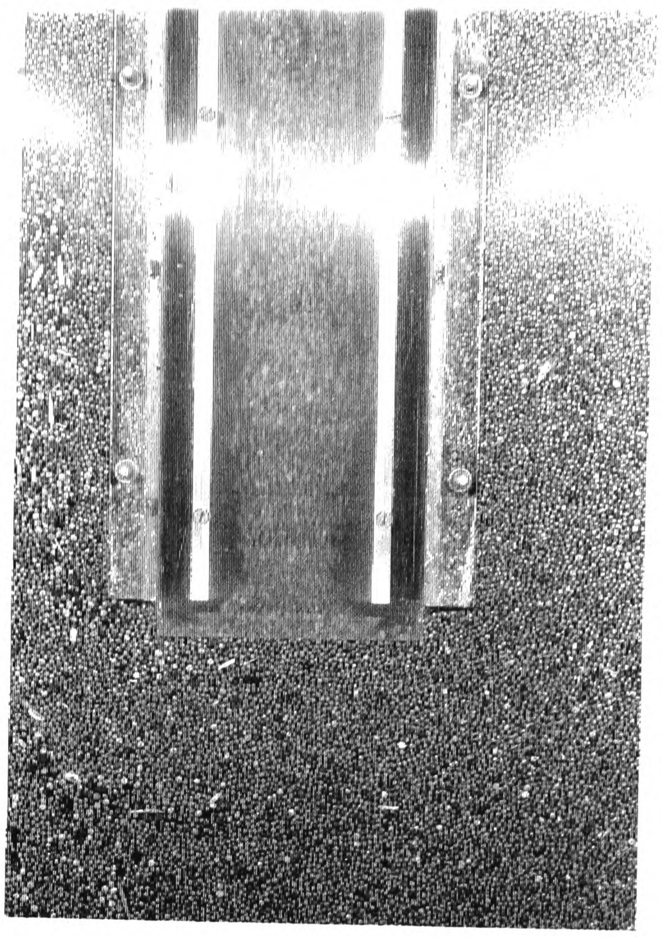


Fig. 4.22 Point B, X= 0mm
Superficial Air Velocity = 30m/s

Product - MUSTARD SEED

Fig. 4.23 Point C, X= +20mm
Superficial Air Velocity = 30m/s

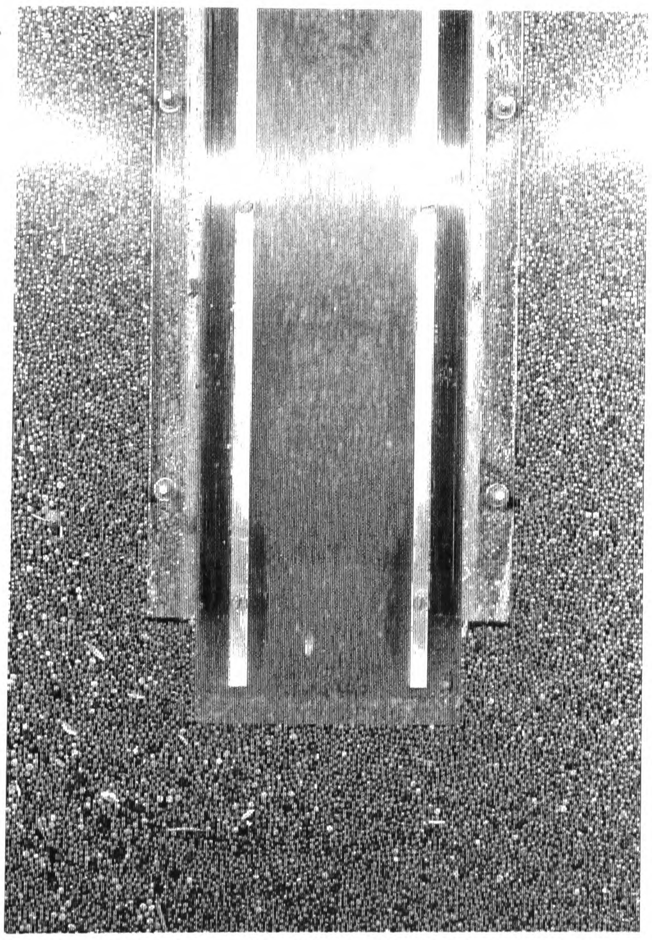
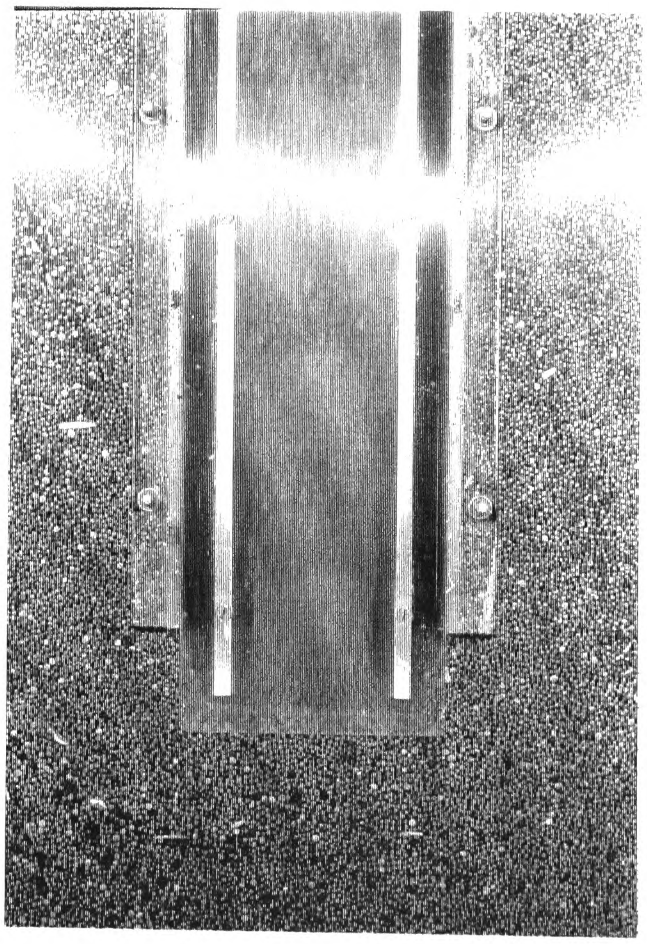


Fig. 4.24 Point D, X= +20mm
Superficial Air Velocity = 60m/s



4.2.4 Dicalcium Phosphate

This product was chosen because it was free-flowing, but of an irregular form having a relatively wide size distribution in comparison with the wheat grain and mustard seed which are virtually mono-sized and regular in shape. Dicalcium phosphate is often shipped in bulk form requiring to be off-loaded from the holds of small coasters of, typically, 1000 tonnes capacity.

The results of these tests are depicted in Figures 4.25 and 4.26 with Figures 4.27 and 4.28 corresponding to points A and B on Figure 4.25 respectively.

Although the performance characteristics for this product follow the same general trend as for the others already tested up to the maximum mass flow rate condition, there would appear to be some important differences. The most marked difference being that, although the product is, relatively speaking, granular in comparison with the alumina it does not follow the general trend for wheat grain, whereby the maximum mass flow rate nozzle configuration is reached and maintained regardless of any further increase in that nozzle position. The actual trend is more in keeping with that for alumina whereby a maximum mass flow rate is reached and then chokes with any further increase in the nozzle position. This tends to suggest that, the performance characteristics of products having a wide size distribution fall into a category of exhibiting some of the typical performance characteristics of both granular and powdered products.

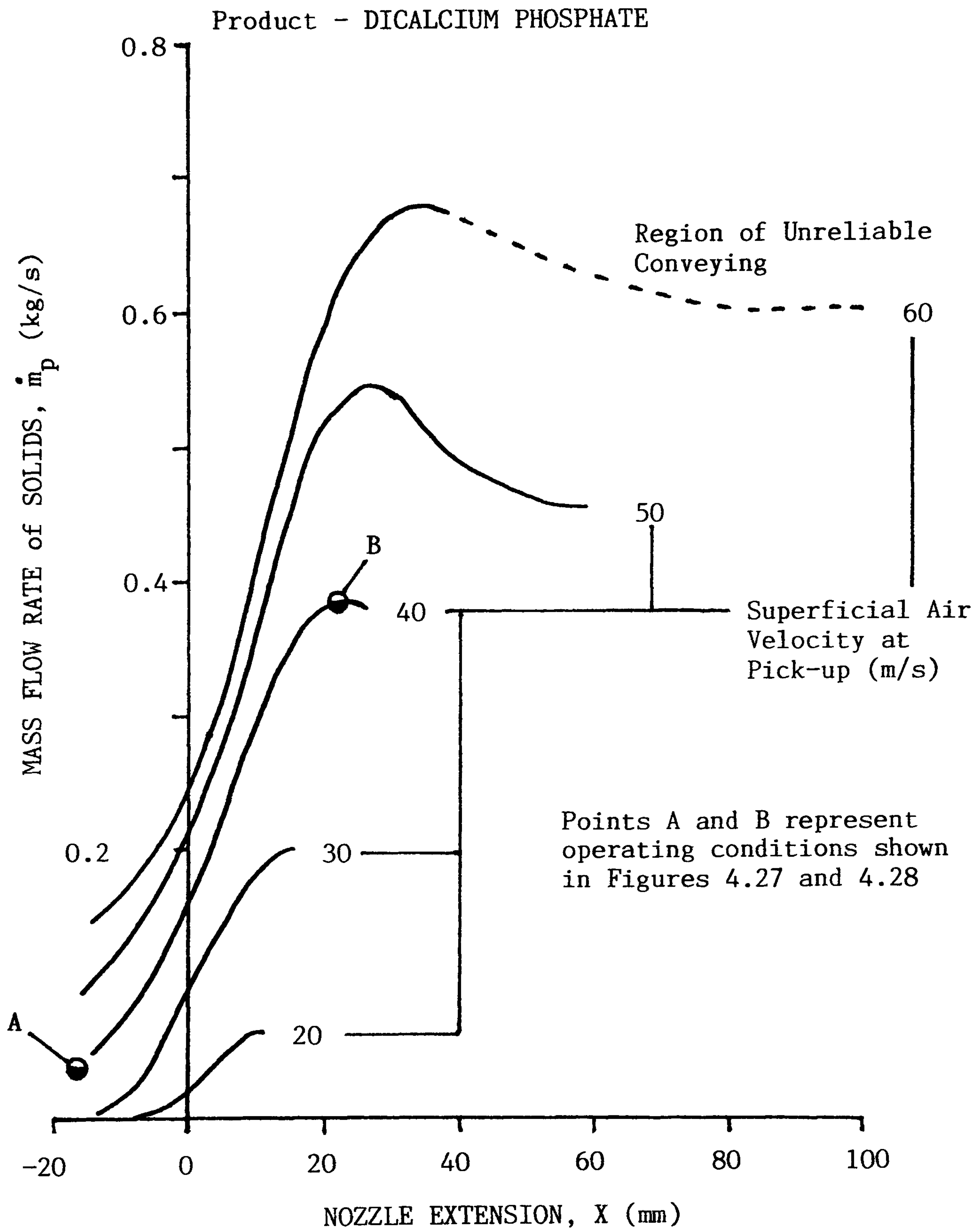


Fig. 4.25 Variation in Mass Flow Rate of Solids With Nozzle Extension for Differing Superficial Air Velocities

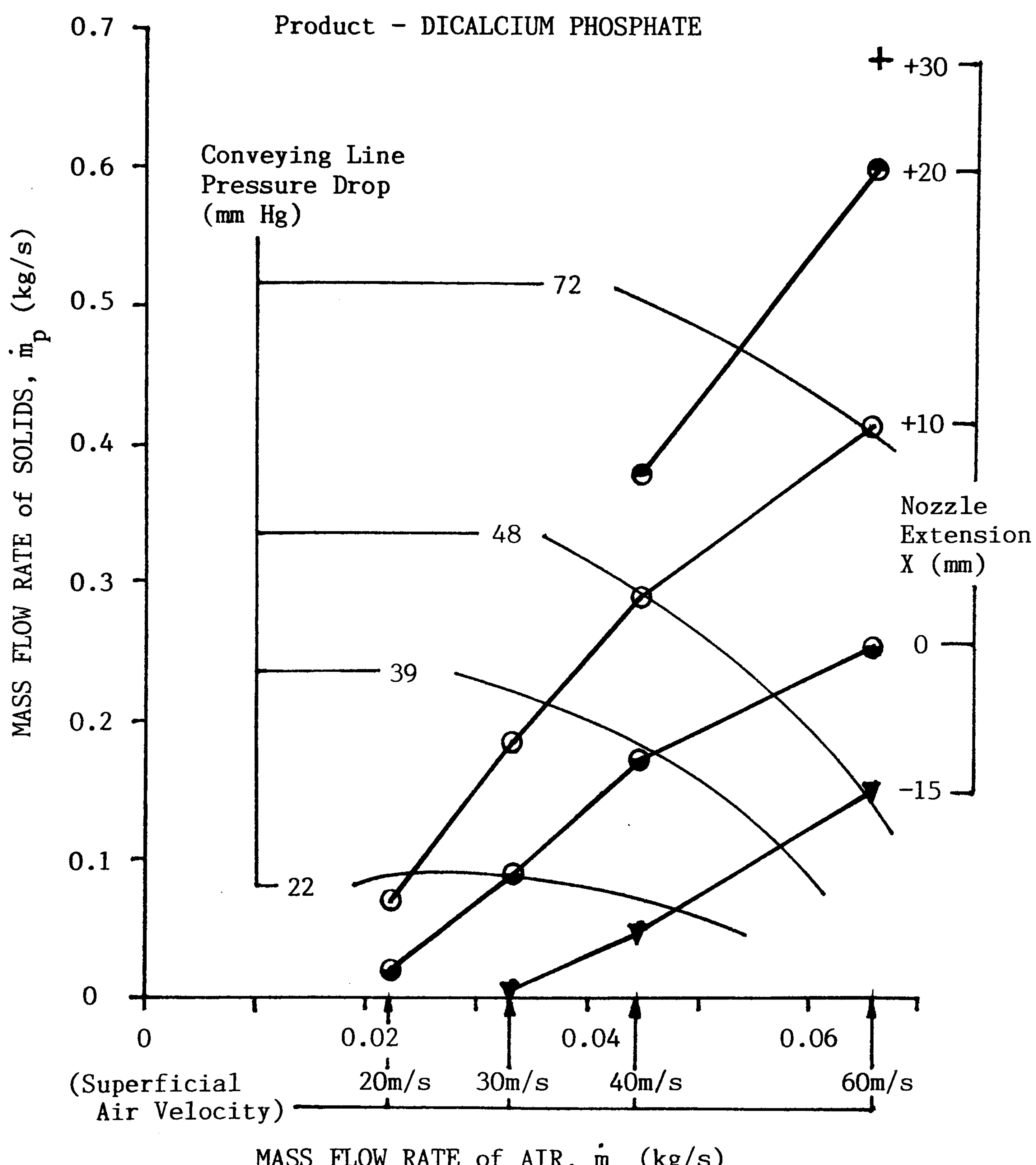


Fig. 4.26 Variation in Mass Flow Rate of Solids With Mass Flow Rate of Air for Differing Nozzle Extensions

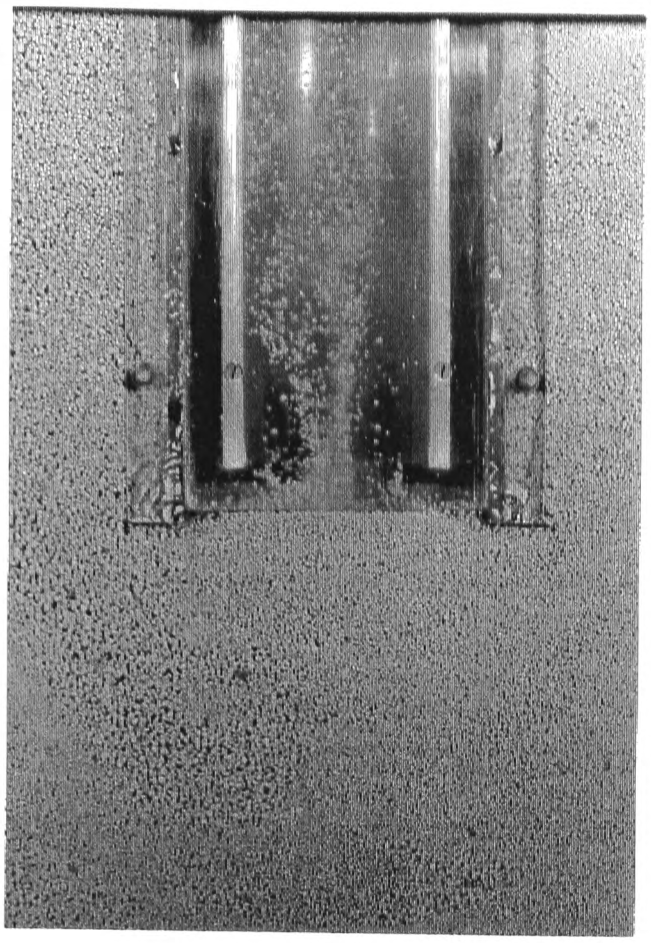


Fig. 4.27 Point A, X= -15mm
Superficial Air Velocity = 40m/s

Product - DICALCIUM PHOSPHATE

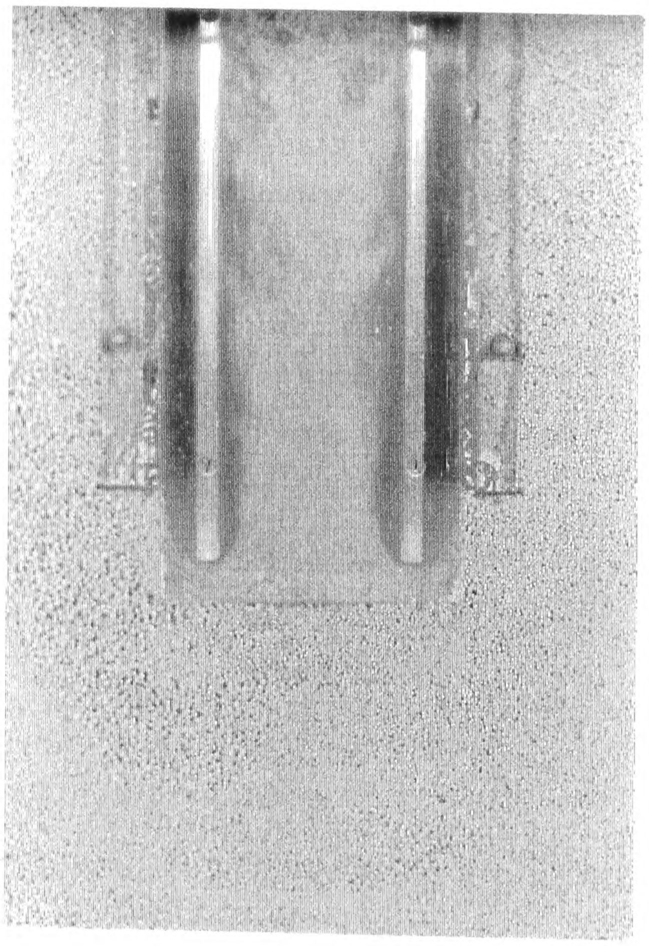


Fig. 4.28 Point B, X= +20mm
Superficial Air Velocity = 40m/s

Product - DICALCIUM PHOSPHATE

4.3 Effects of Throttling on System Performance

These tests have provided information about how the overall system performance of the standard suction nozzle can be affected by throttling the air flowing into the annular chamber. These tests were undertaken using the same test rig as used for the work described in section 4.2, but modified to enable the air entering the annular chamber to be throttled. For this study the description of throttling was based on a percentage reduction in the cross-section of the annulus. The additional throttling components have been previously described in section 3.3.1. The test procedure was exactly the same as that used to assess the performance of the standard suction nozzle, but with the addition of the throttling as an extra variable.

4.3.1 Alumina

The effects of increasing the degree of throttle on the mass flow rate of alumina for a nozzle position of X= 0mm, ie. where the inner and outer pipes are level, can be seen in Figures 4.29 and 4.30. Figure 4.29 shows quite clearly that the mass flow rate can be increased for any given set of conditions by simply increasing the percentage throttle. The most marked increases occurred at higher pick-up velocities and with throttle settings of between 60% and 90%. Any further increase in throttling above 90% caused the system to choke and cease conveying. A six fold increase in mass flow rate could be achieved by simply changing the conveying conditions from :-

nozzle position - 0 mm, pick-up velocity - 50 m/s, throttle - 0%

to:-

nozzle position - 0 mm, pick-up velocity - 50 m/s, throttle - 90%

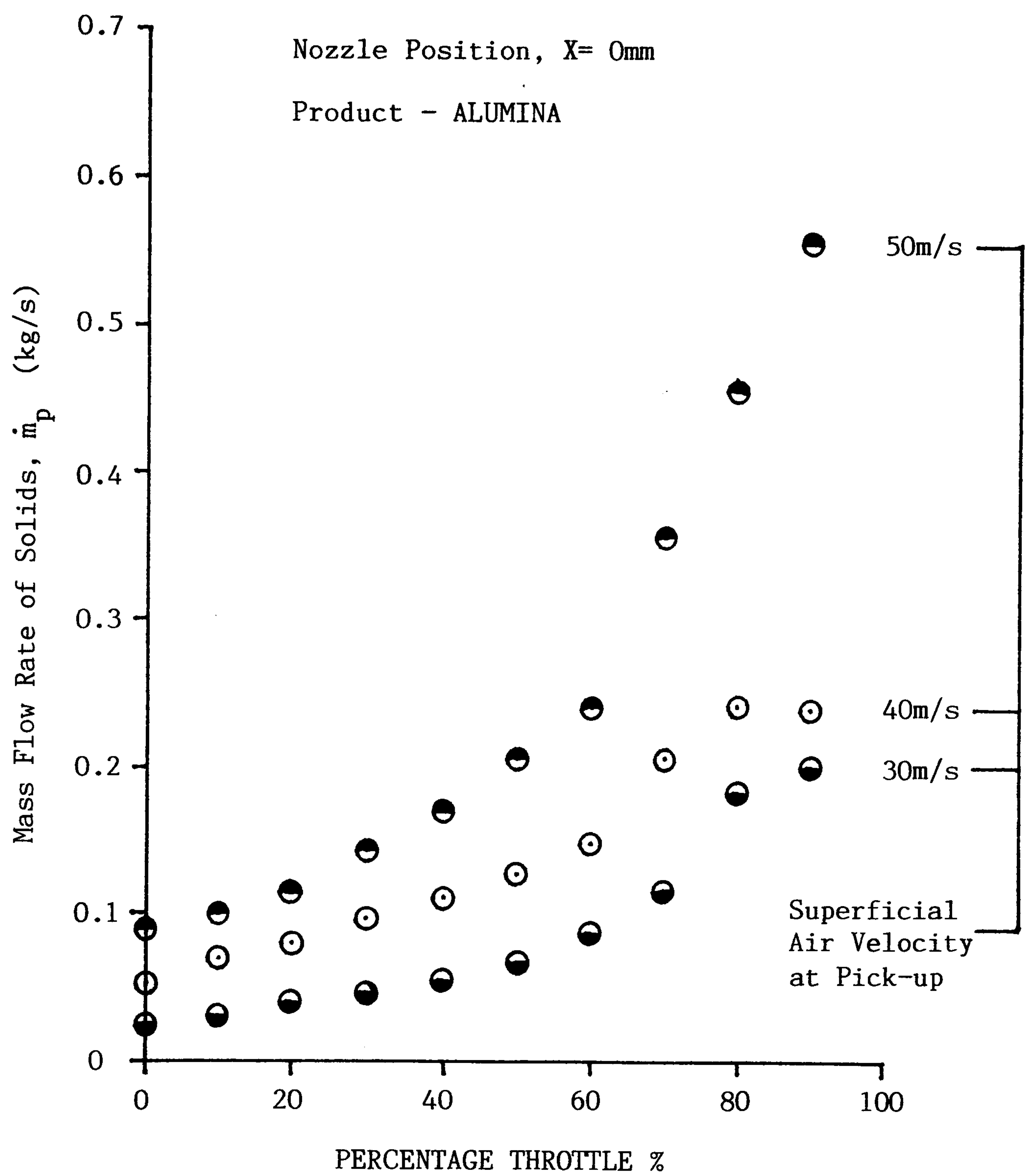


Fig. 4.29 Effect of Throttling on Mass Flow Rate of Solids for Varying Air Velocities

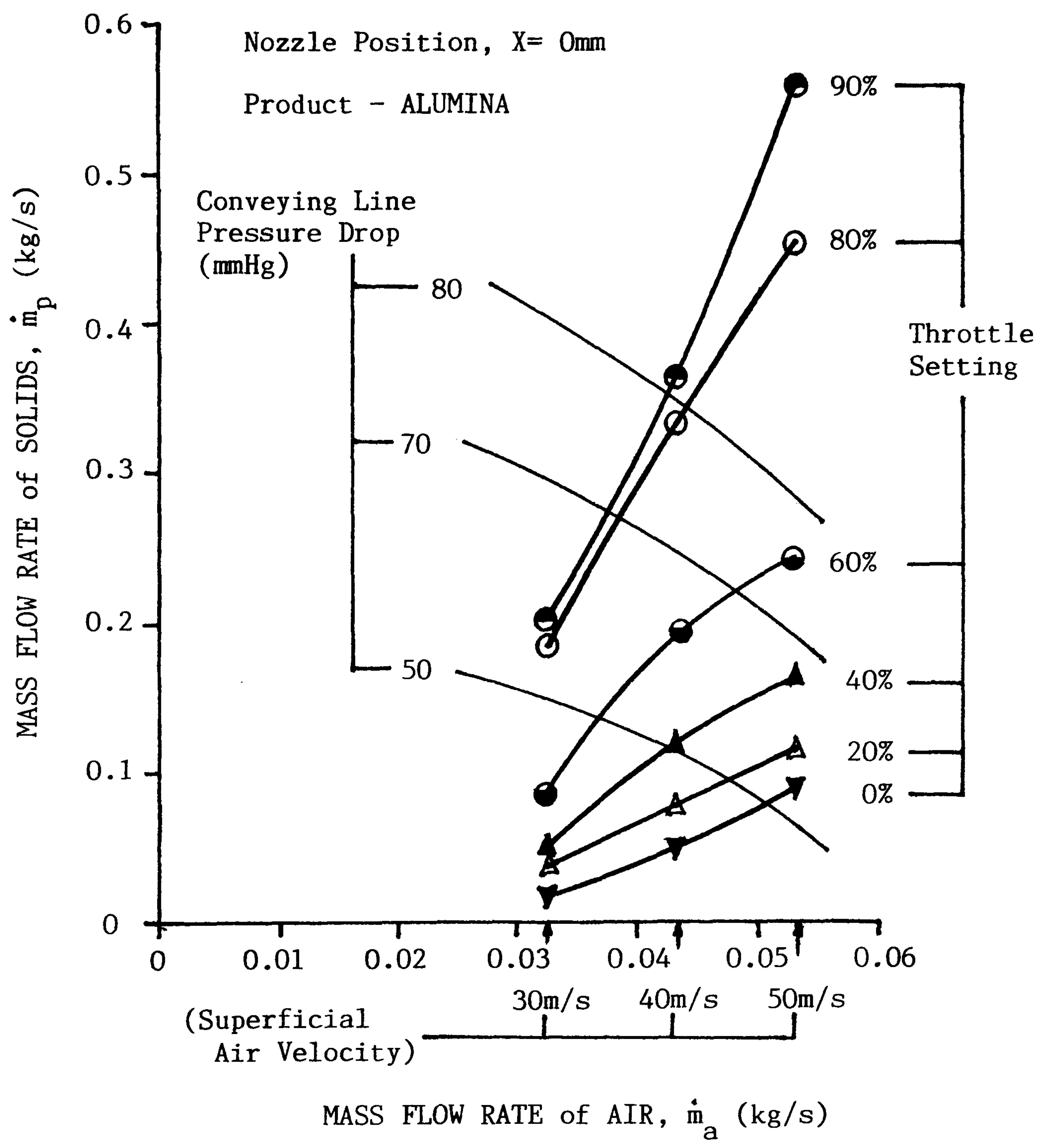


Fig. 4.30 Variation in Mass Flow Rate of Solids With Mass Flow Rate of Air for Differing Throttle Settings

The photographs shown in Figures 4.31 to 4.39 illustrate the effects of throttling the air flow into the annular chamber for a constant pick-up velocity and three nozzle configurations, ie. retracted, level and protruding.
Figures 4.31 to 4.33 are for the nozzle in a retracted position (X= -20mm). Figure 4.31 is typical of the entrainment pattern found when operating a "standard" suction nozzle in a retracted configuration. This can be verified by studying the photograph shown in Figure 4.3. If the throttle is increased to 60%, the alumina is lifted into the space formed beneath the inner pipe, see Figure 4.32, and as a result the alumina is conveyed, albeit at relatively low conveying rates. However, if the throttle is increased still further the mass flow rate increases dramatically as shown in Figure 4.33. Figures 4.34 to 4.36 show the effects of throttling for a level nozzle (X= 0mm), whilst Figures 4.37 to 4.39 again show the effects of throttling, but this time for a protruding nozzle (X= +20mm).

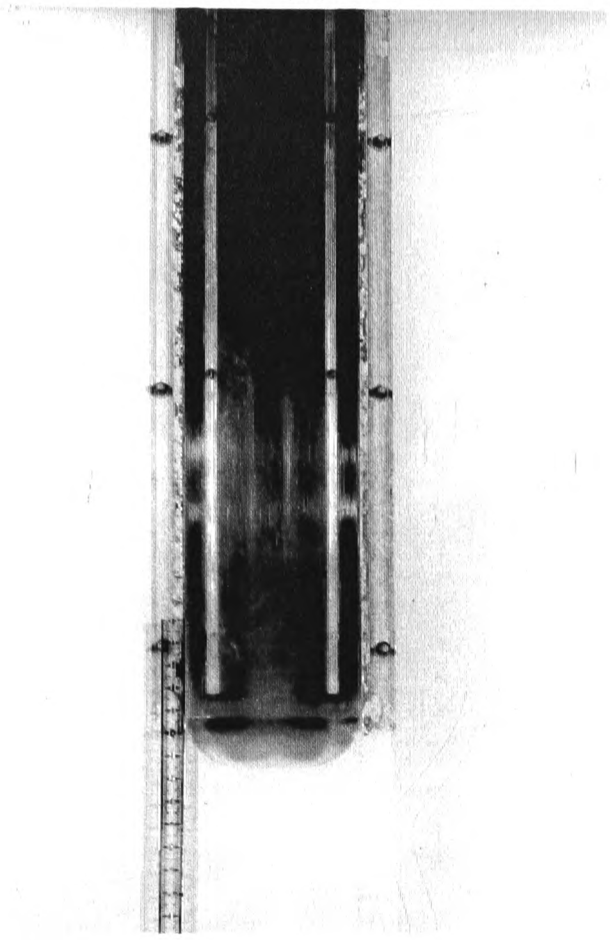


Fig. 4.31

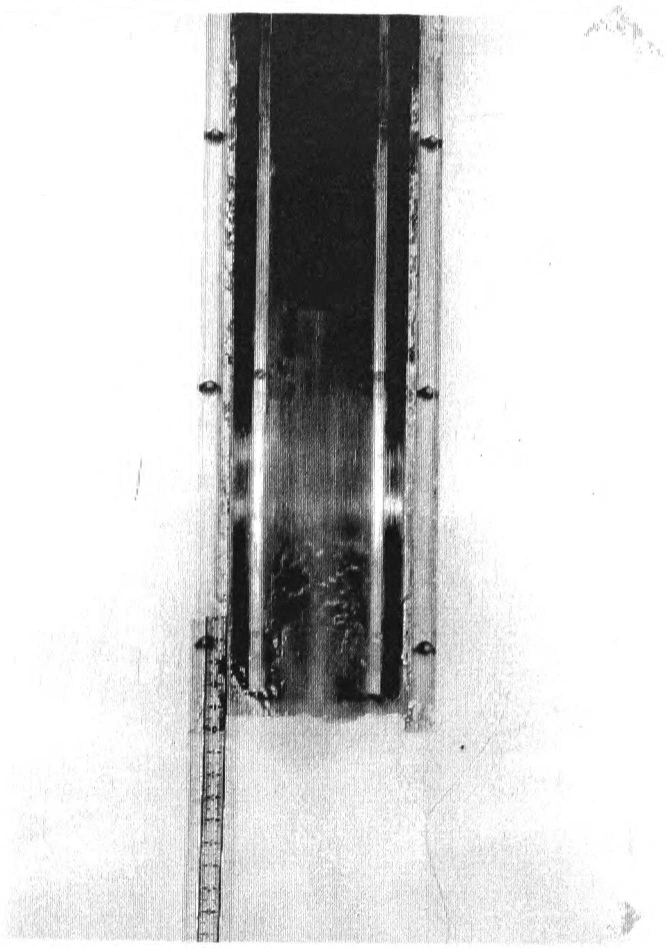


Fig. 4.32

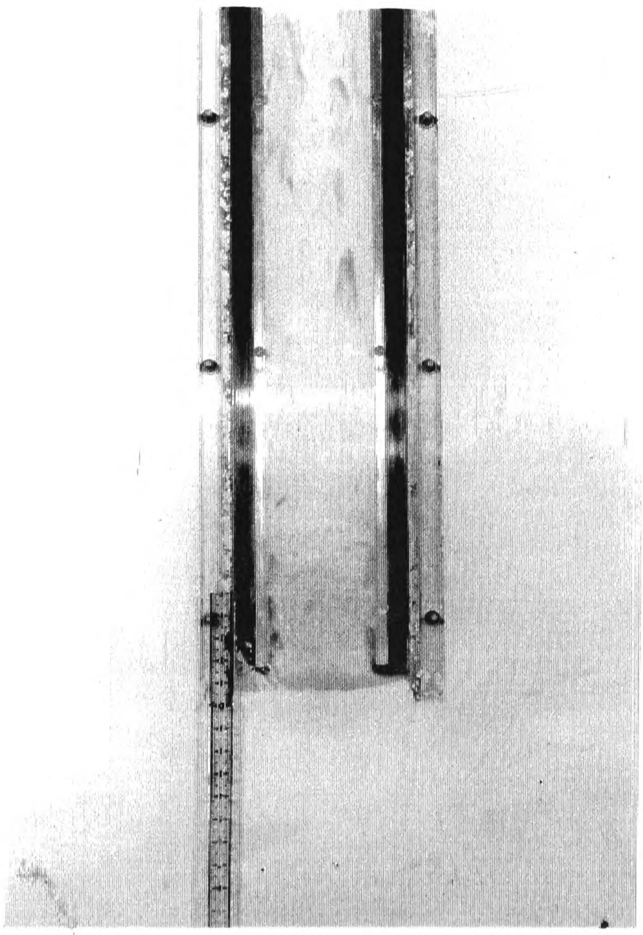


Fig. 4.33

Product - ALUMINA
Superficial Air Velocity = 30m/s

Fig. 4.31 X= -20mm, Throttle = 0%
Fig. 4.32 X= -20mm, Throttle = 60%
Fig. 4.33 X= -20mm, Throttle = 90%

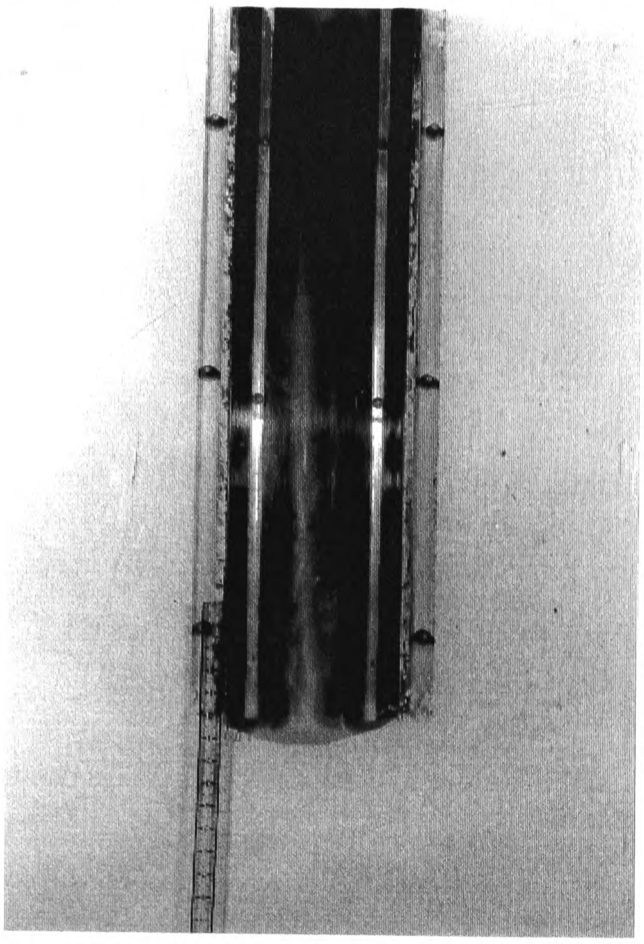


Fig. 4.34

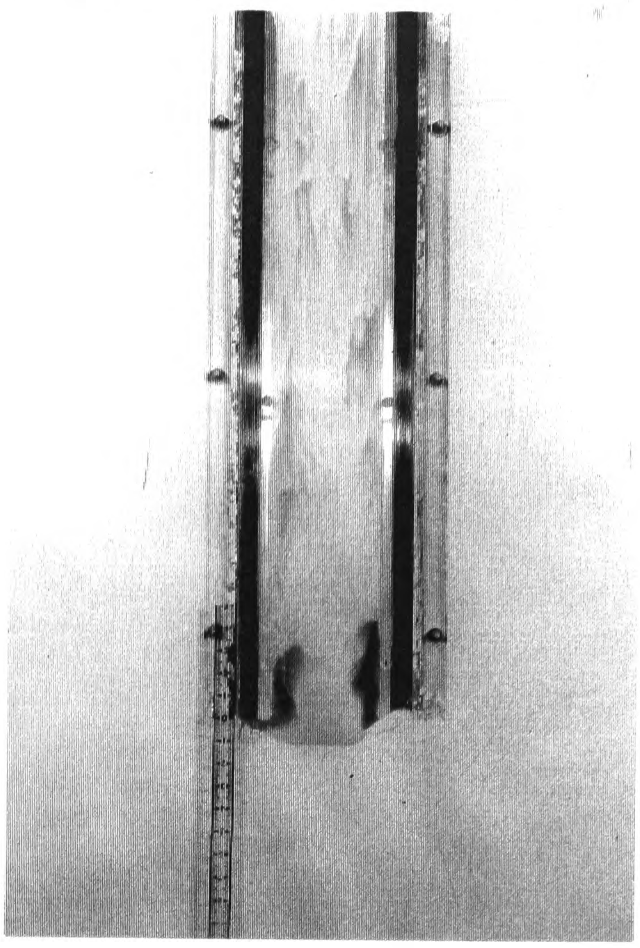


Fig. 4.35

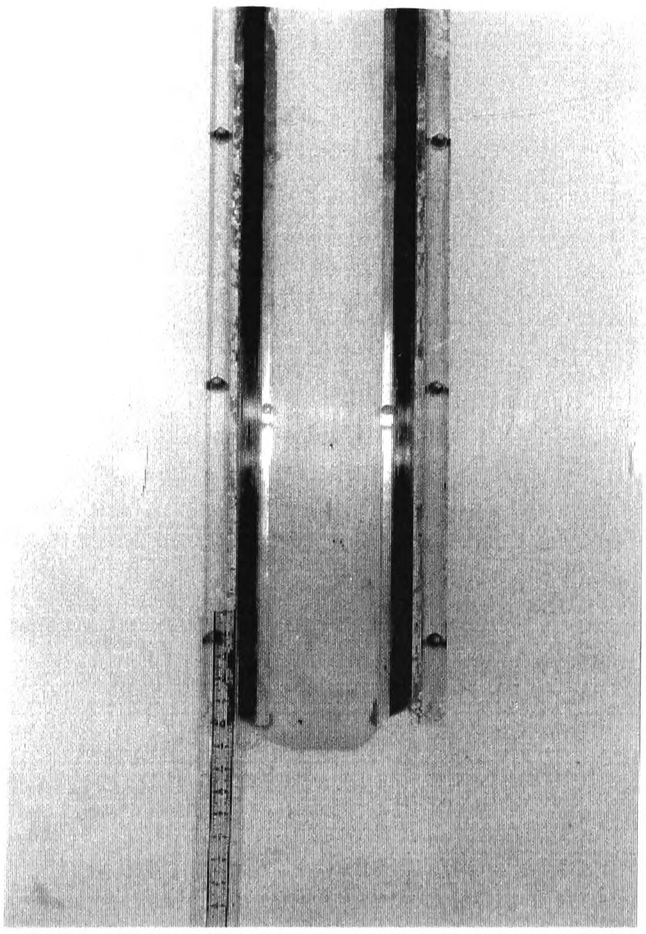


Fig. 4.36

Product - ALUMINA
Superficial Air Velocity = 30m/s

Fig. 4.34 X= 0mm, Throttle = 0%
Fig. 4.35 X= 0mm, Throttle = 60%
Fig. 4.36 X= 0mm, Throttle = 90%

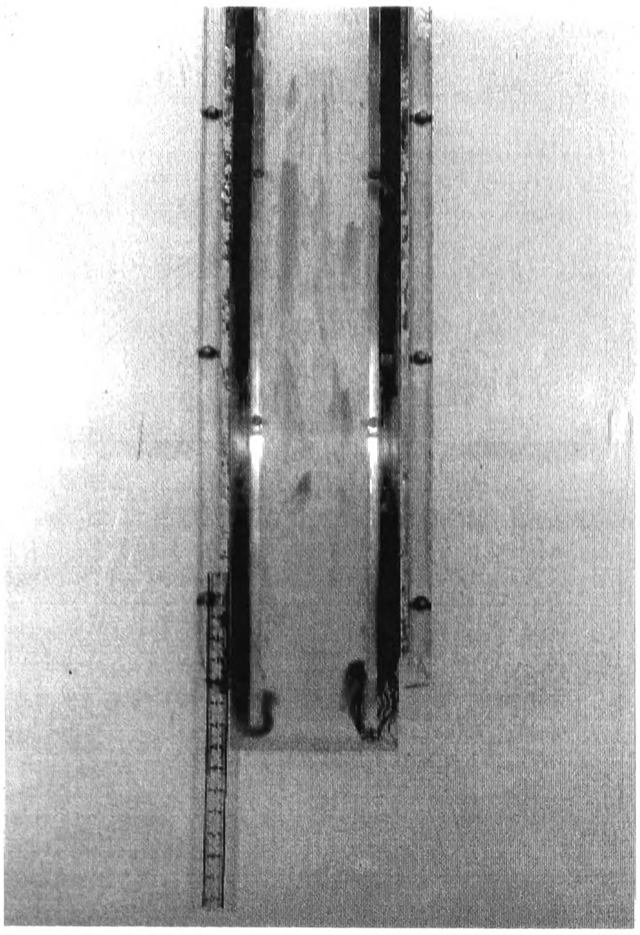


Fig. 4.37

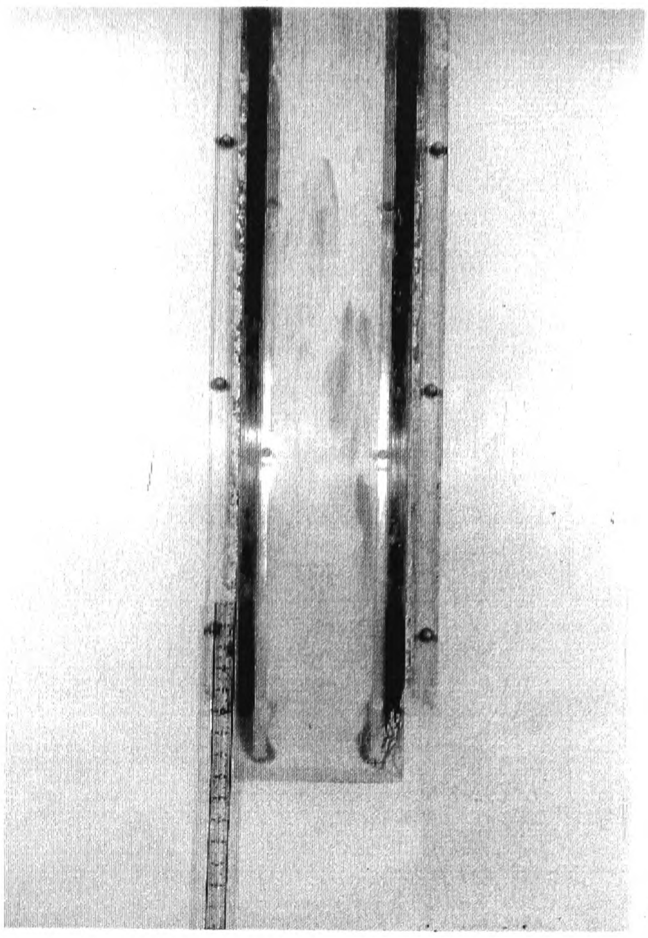


Fig. 4.38

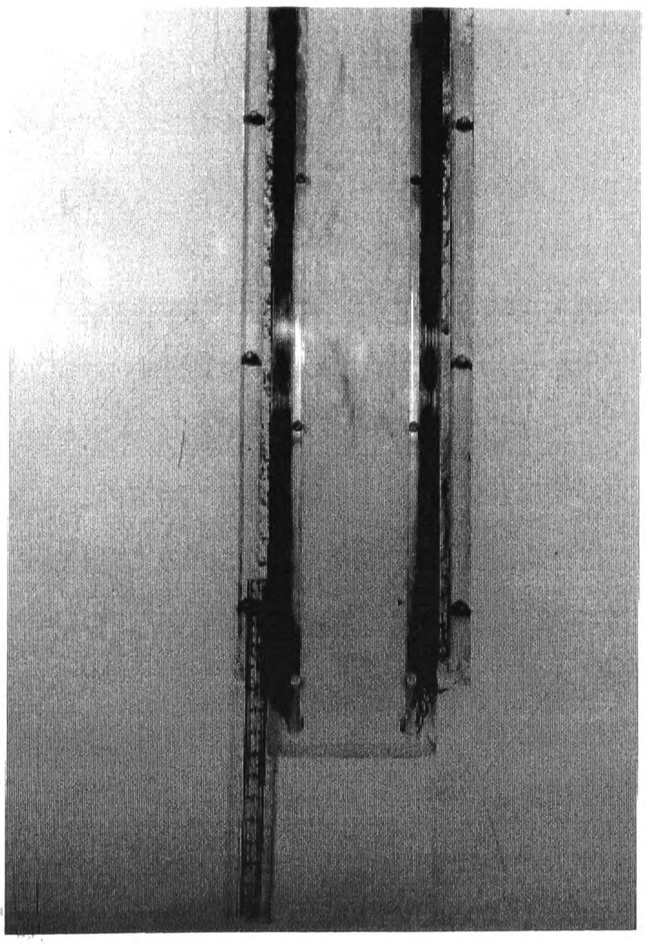


Fig. 4.39

Product - ALUMINA
Superficial Air Velocity = 30m/s

Fig. 4.37 X= +20mm, Throttle = 0%
Fig. 4.38 X= +20mm, Throttle = 60%
Fig. 4.39 X= +20mm, Throttle = 90%

4.3.2 Plastic Pellets (Rigidex)

The effects of throttling on the performance characteristics for plastic pellets when operating with a level (X= 0mm) and protruding (X= +50mm) nozzle can be seen in Figures 4.40 and 4.41 respectively. Figure 4.40 depicts a trend that is similar to that found with alumina, but Figure 4.41 shows that the mass flow rate of the plastic pellets remains fairly constant regardless of throttle setting for a given pick-up velocity. The photographs in Figures 4.42 to 4.50 give a visual indication of the effect of throttling on the plastic pellets for a range of differing nozzle geometries. As indicated in section 4.3.1 for alumina, whereby a "non-flow" retracted nozzle configuration could be turned into a "flow" condition by simply increasing the throttle, this is also the case for the plastic pellets. The reason for this is that with very low pick-up velocities and no throttling, the air can flow down the annular chamber of the suction nozzle and percolate through the interstices of the product positioned immediately beneath the nozzle, thereby entraining only a few particles into the nozzle itself. However, if the annular chamber of the nozzle is throttled, the air will take the path of least resistance. Should the pressure drop across the throttle be greater than that across the bed of product surrounding the nozzle, the air will flow through the bed of product moving it into the entrainment region. Although this can be beneficial, if the throttle setting is increased too much the nozzle will become choked and conveying will cease. The increase in feeding efficiency resulting from increasing the throttle setting for a protruding nozzle configuration is negligible, because, the suction nozzle is already an efficient, self-feeding mechanism. When the nozzle is in a protruding position there is already a bed of particles between the underside of the annular chamber and the inlet plane of the inner conveying pipe. Thus, the pressure drop across the throttle will not have such a significant effect on altering the path the air takes in reaching the entrainment region of the nozzle.

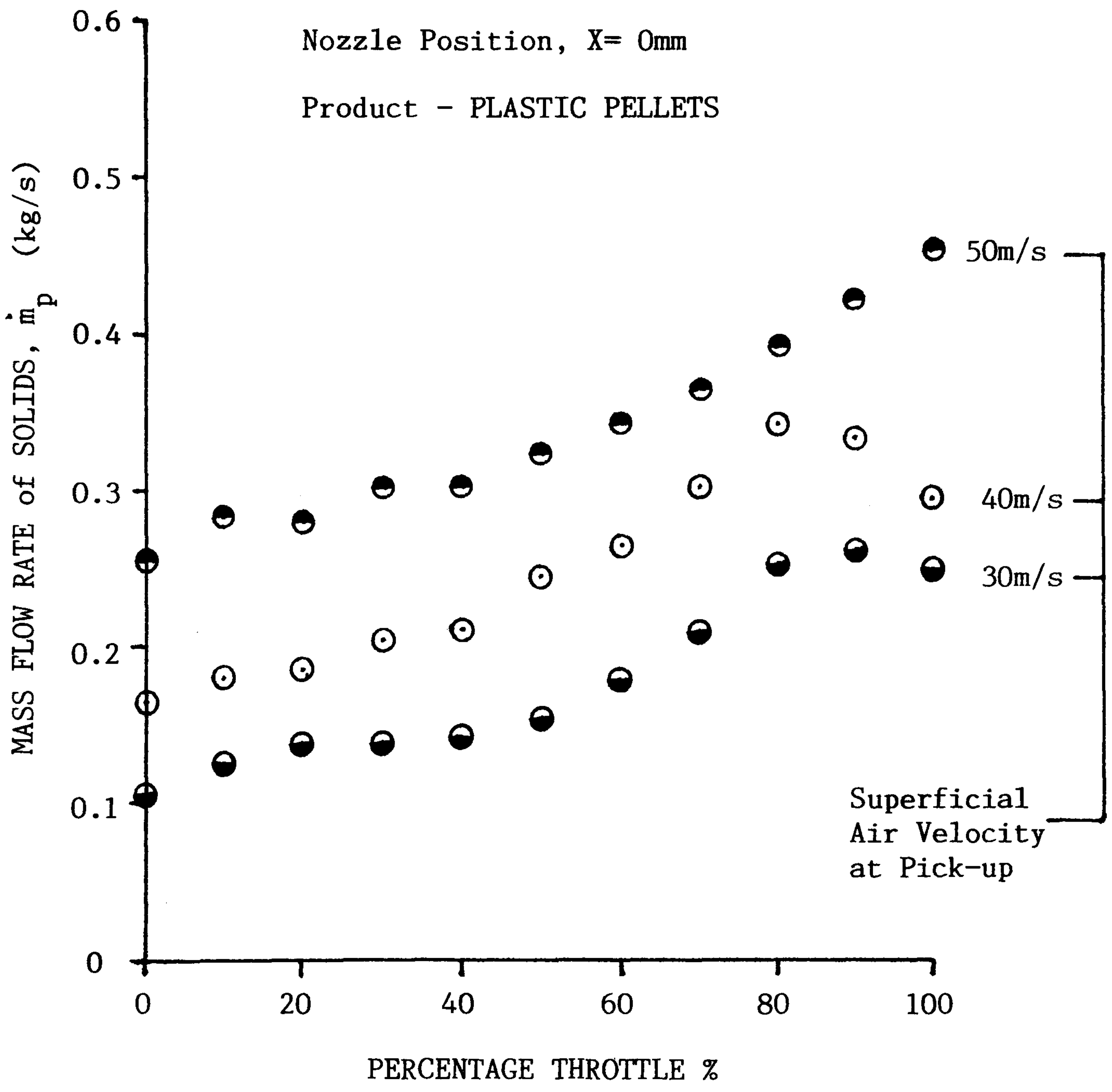


Fig. 4.40 Effect of Throttling on Mass Flow Rate of Solids for Varying Air Velocities

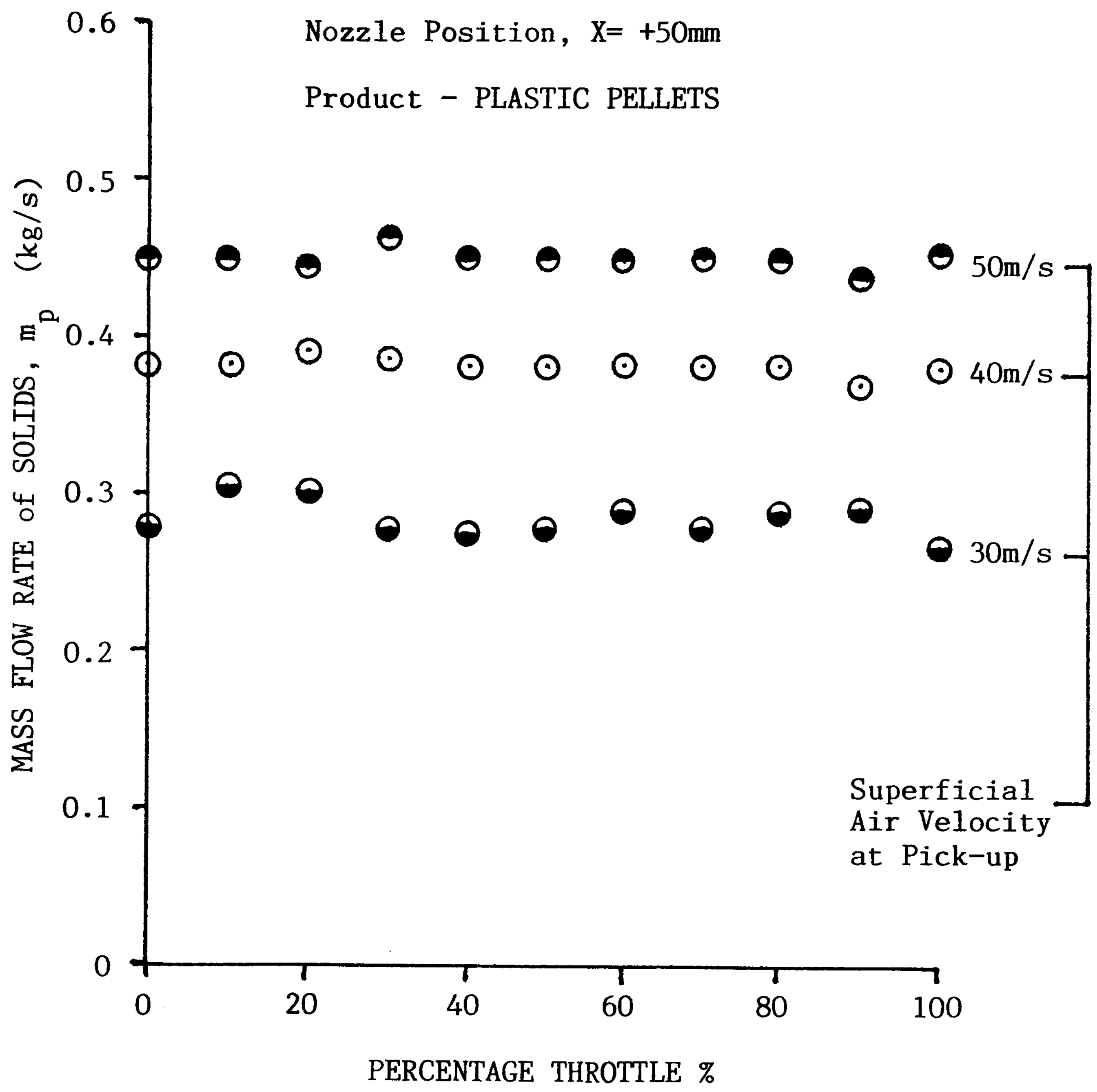


Fig. 4.41 Effect of Throttling on Mass Flow Rate of Solids for Varying Air Velocities

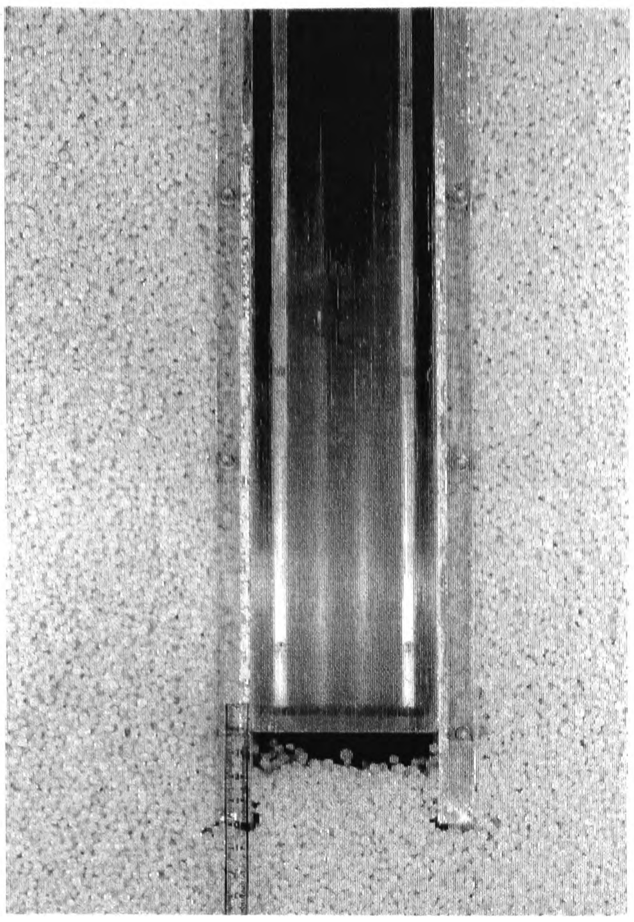


Fig. 4.42

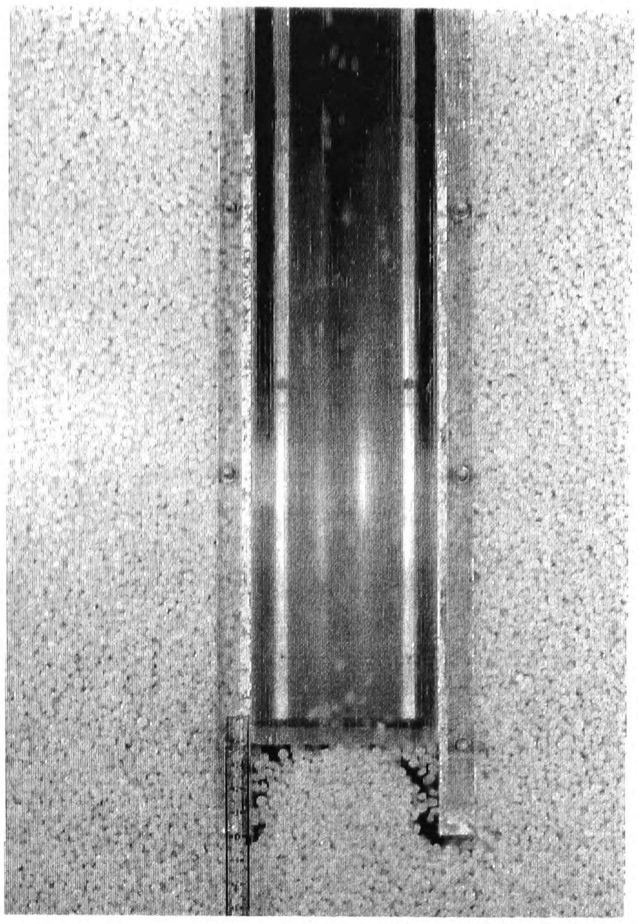


Fig. 4.43

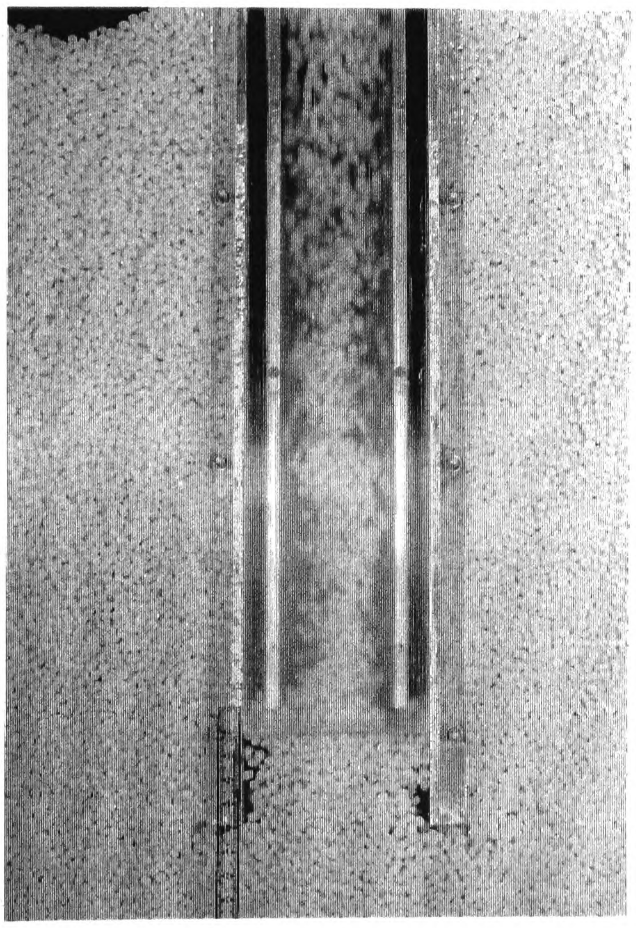


Fig. 4.44

Product - PLASTIC PELLETS
Superficial Air Velocity = 30m/s

Fig. 4.42	X= -50mm,	Throttle = 0%
Fig. 4.43	X= -50mm,	Throttle = 40%
Fig. 4.44	X= -50mm,	Throttle = 100%

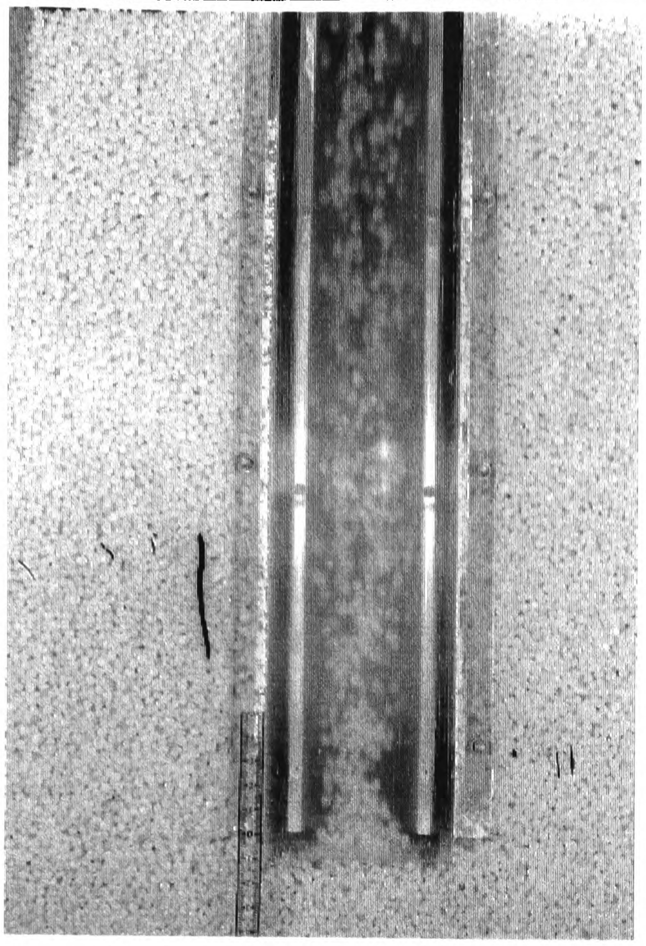


Fig. 4.45

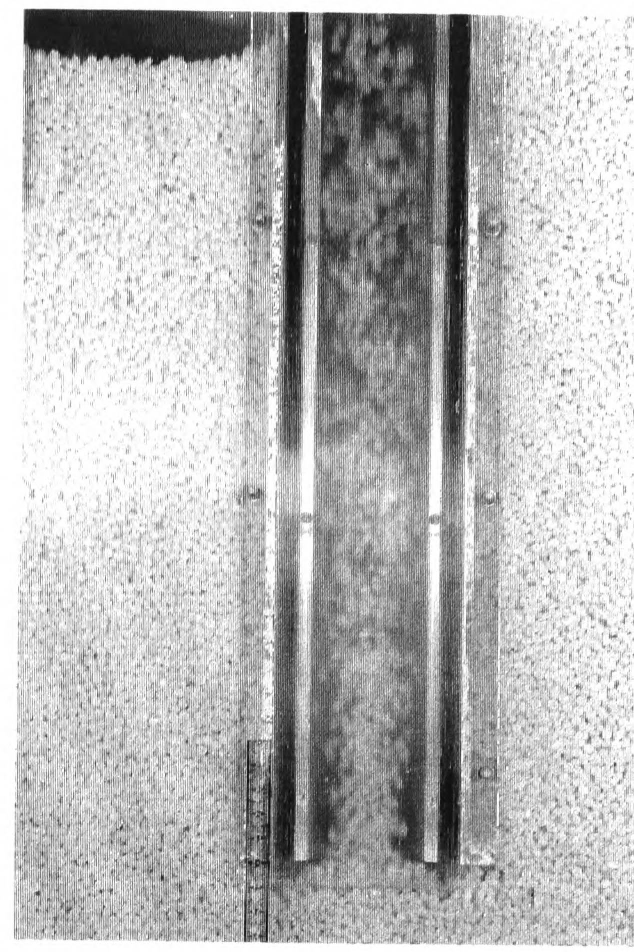


Fig. 4.46

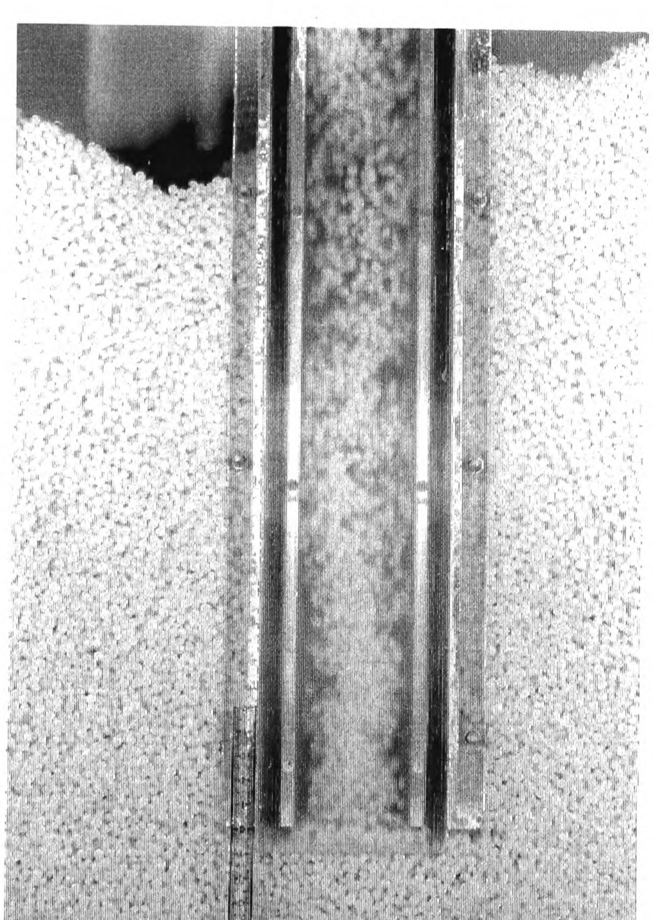


Fig. 4.47

Product - PLASTIC PELLETS
Superficial Air Velocity = 30m/s

Fig. 4.45 X= 0mm, Throttle = 0%
Fig. 4.46 X= 0mm, Throttle = 40%
Fig. 4.47 X= 0mm, Throttle = 100%

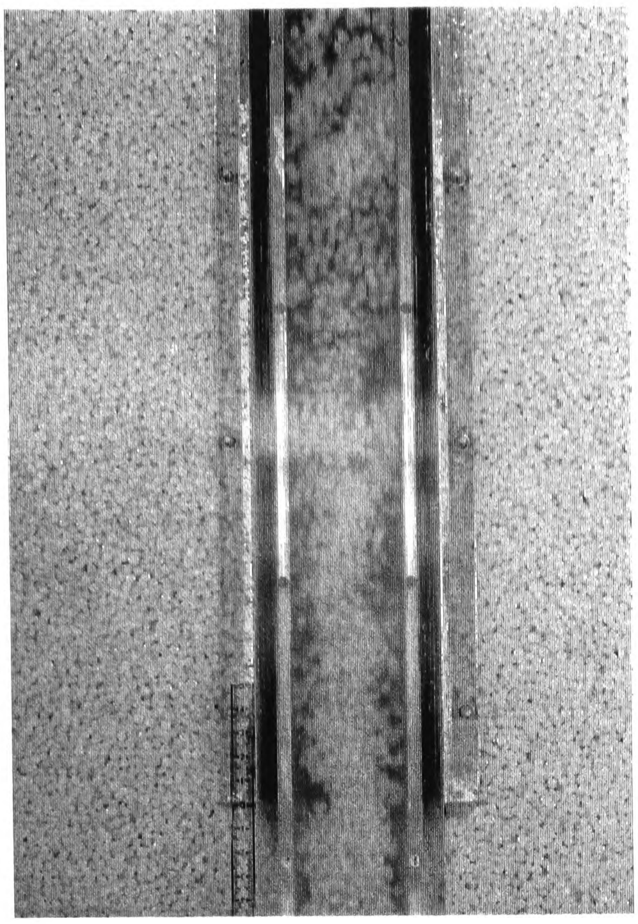


Fig. 4.48

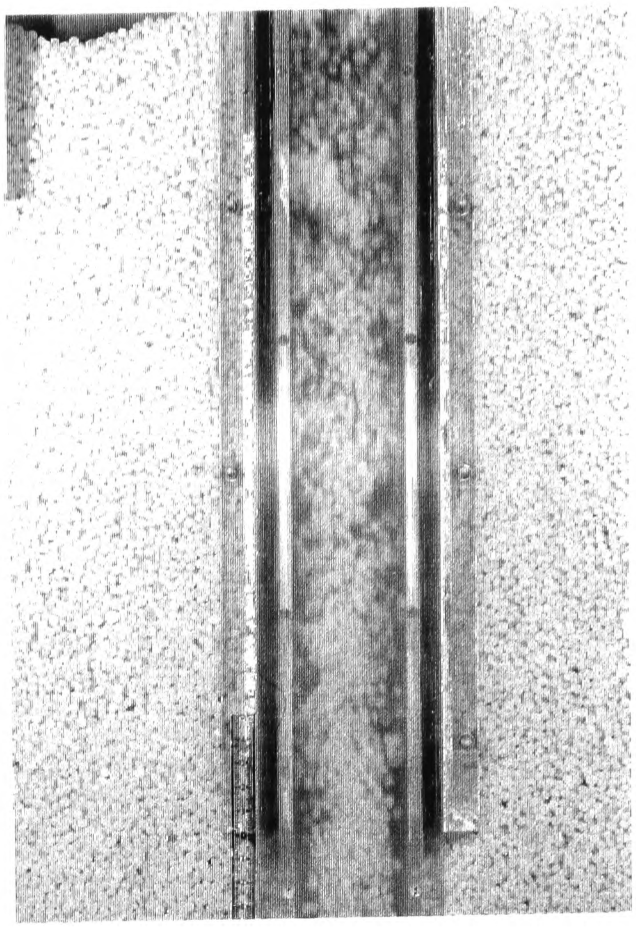


Fig. 4.49

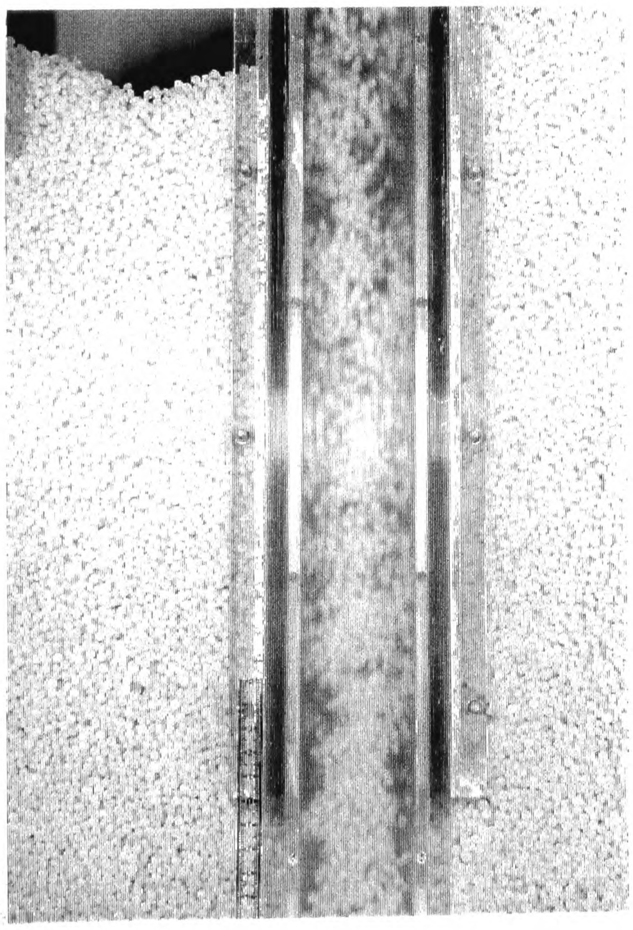


Fig. 4.50

Product - PLASTIC PELLETS
Superficial Air Velocity = 30m/s

Fig. 4.48	X= +50mm,	Throttle = 0%
Fig. 4.49	X= +50mm,	Throttle = 40%
Fig. 4.50	X= +50mm,	Throttle = 100%

4.3.3 Dicalcium Phosphate

The graph shown in Figure 4.51, is of the variation in mass flow rate with the degree of throttling for a level nozzle configuration and varying pick-up velocities. The graph depicts a similar trend as for the alumina. However, the point of maximum mass flow rate occurs at a throttle setting of approximately 80%, whereas the maximum for alumina occurred at approximately 90%. The increase in mass flow rate is not affected by throttling at relatively low pick-up velocities, but at higher pick-up velocities (typically 50 m/s) the mass flow rate of dicalcium phosphate can be doubled by applying a 90% throttle setting. The photographs in Figures 4.52 to 4.60 again show the effect of the variation in feeding efficiency for the suction nozzle when operating under various configurations.

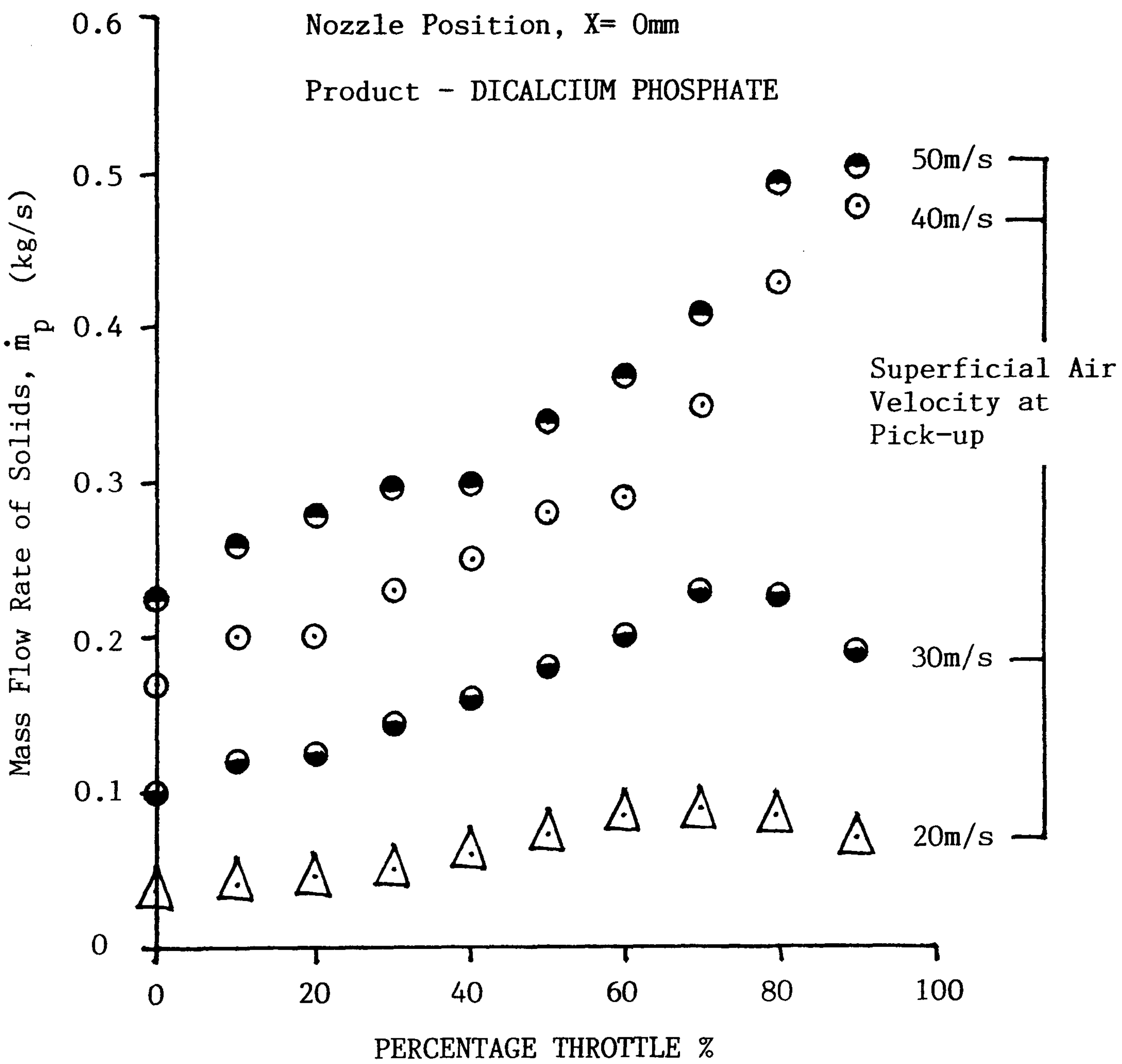


Fig. 4.51 Effect of Throttling on Mass Flow Rate of Solids for Varying Air Velocities

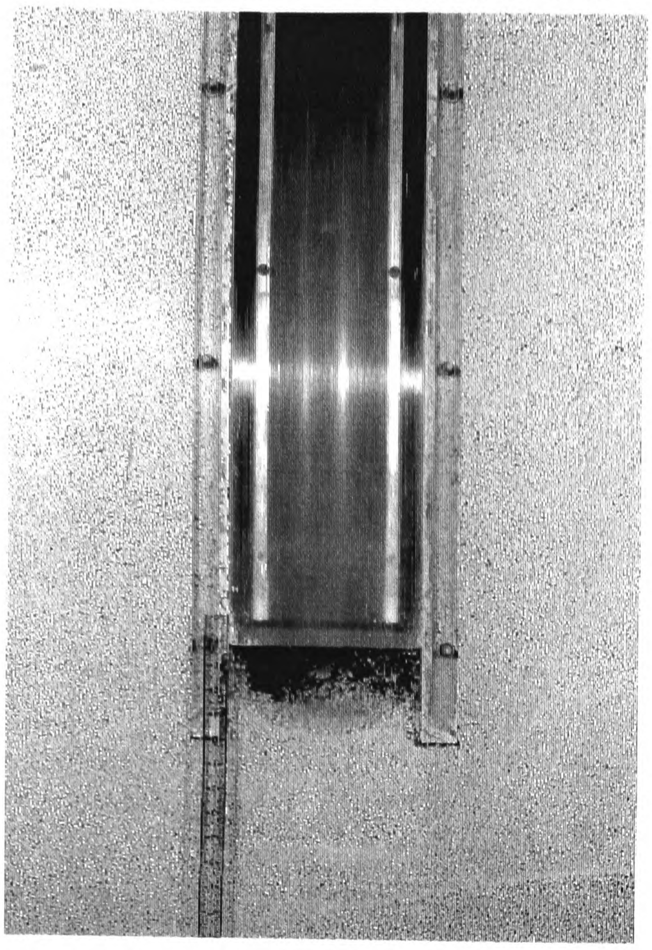


Fig. 4.52

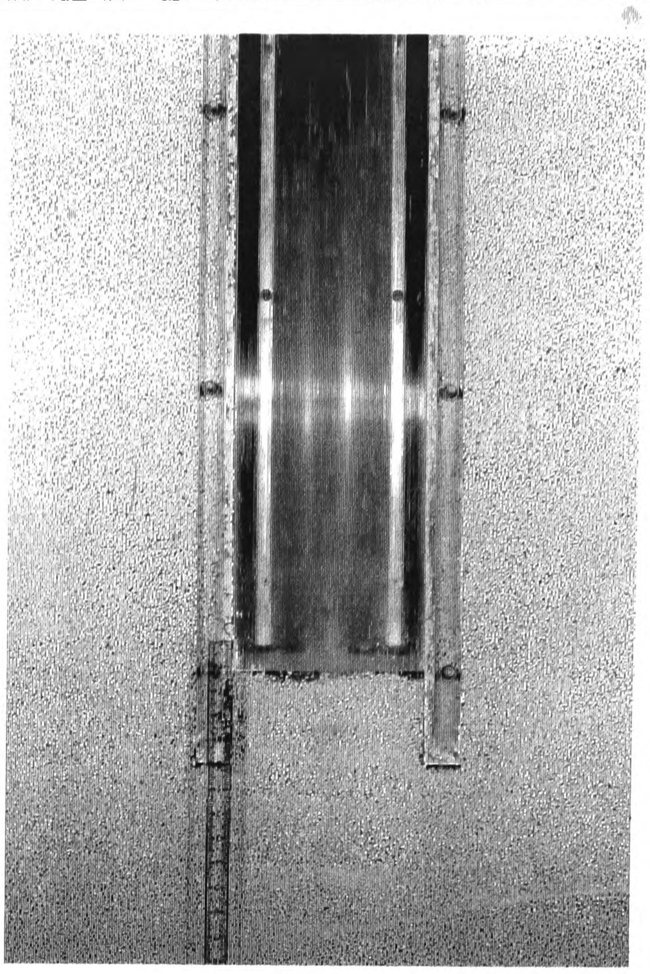


Fig. 4.53

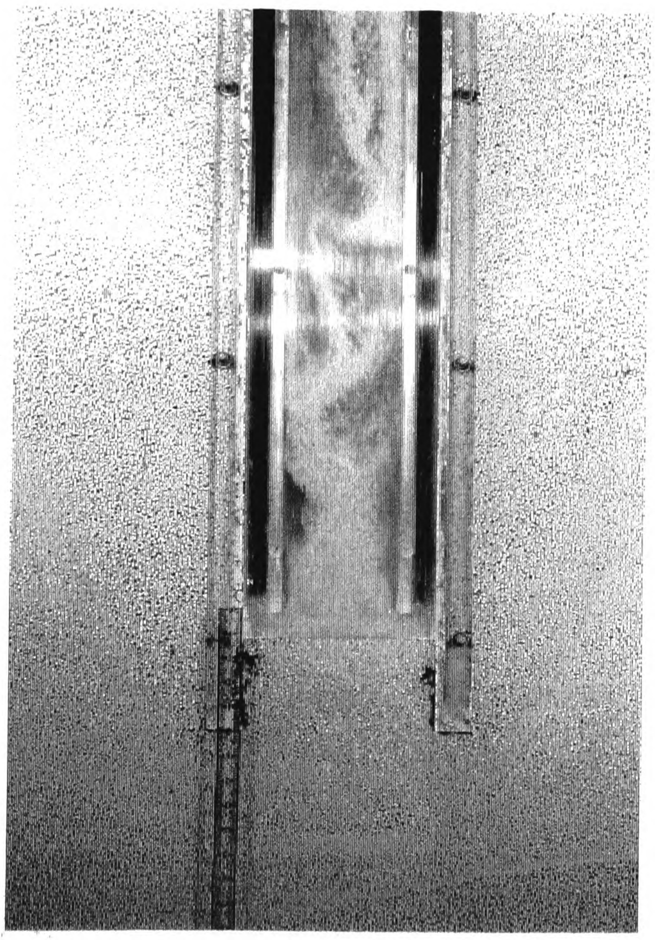


Fig. 4.54

Product - DICALCIUM PHOSPHATE
Superficial Air Velocity = 30m/s

Fig. 4.52 X= -50mm, Throttle = 0%
Fig. 4.53 X= -50mm, Throttle = 40%
Fig. 4.54 X= -50mm, Throttle = 90%

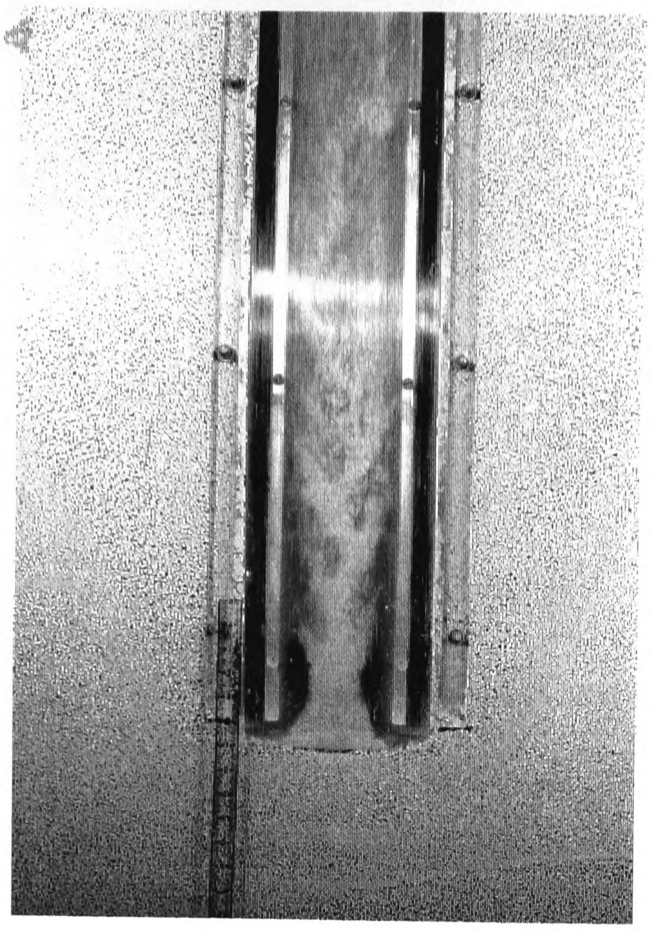


Fig. 4.55

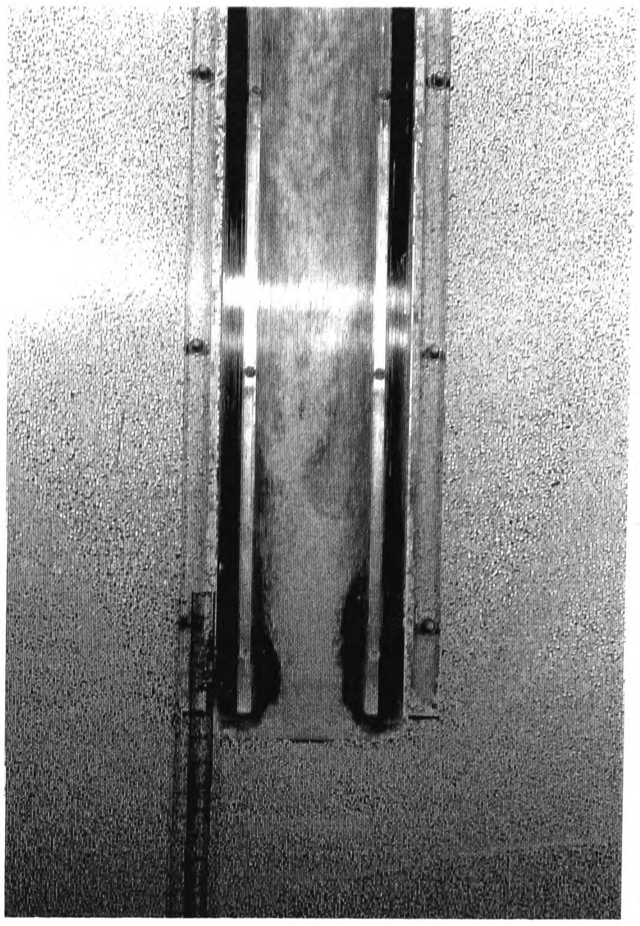


Fig. 4.56

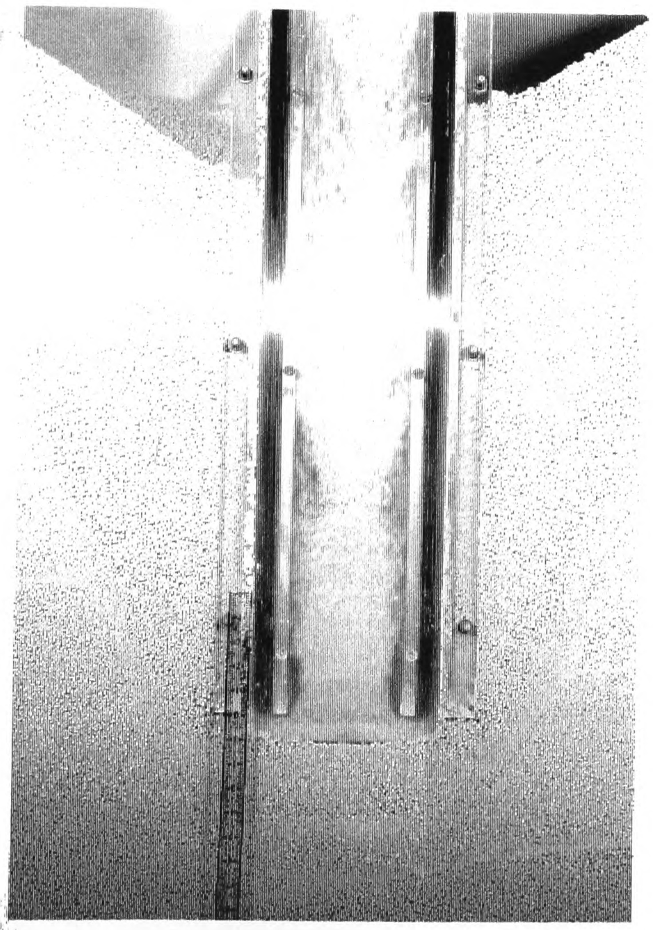


Fig. 4.57

Product - DICALCIUM PHOSPHATE
Superficial Air Velocity = 30m/s

Fig. 4.55 X= 0mm, Throttle = 0%
Fig. 4.56 X= 0mm, Throttle = 40%
Fig. 4.57 X= 0mm, Throttle = 90%

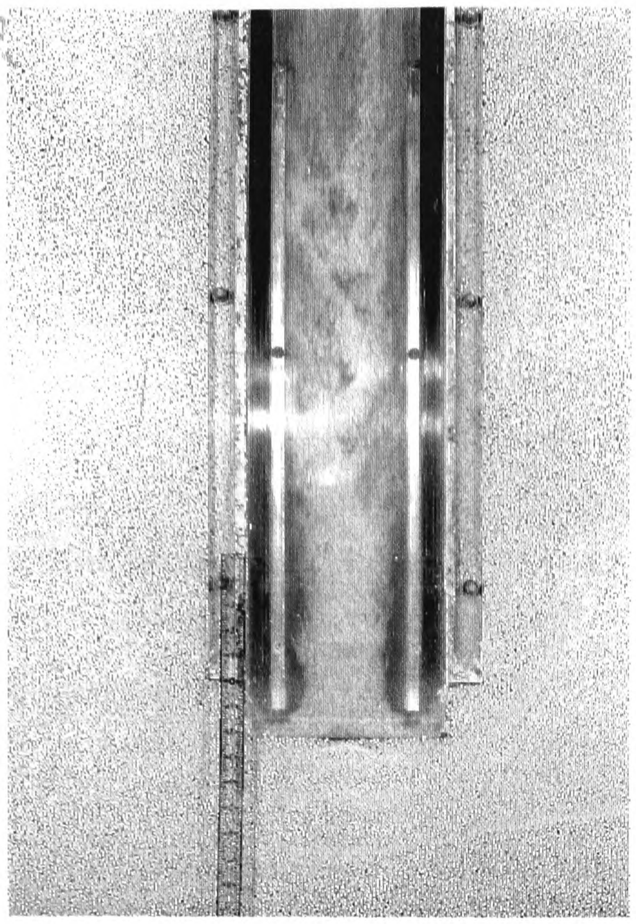


Fig. 4.58

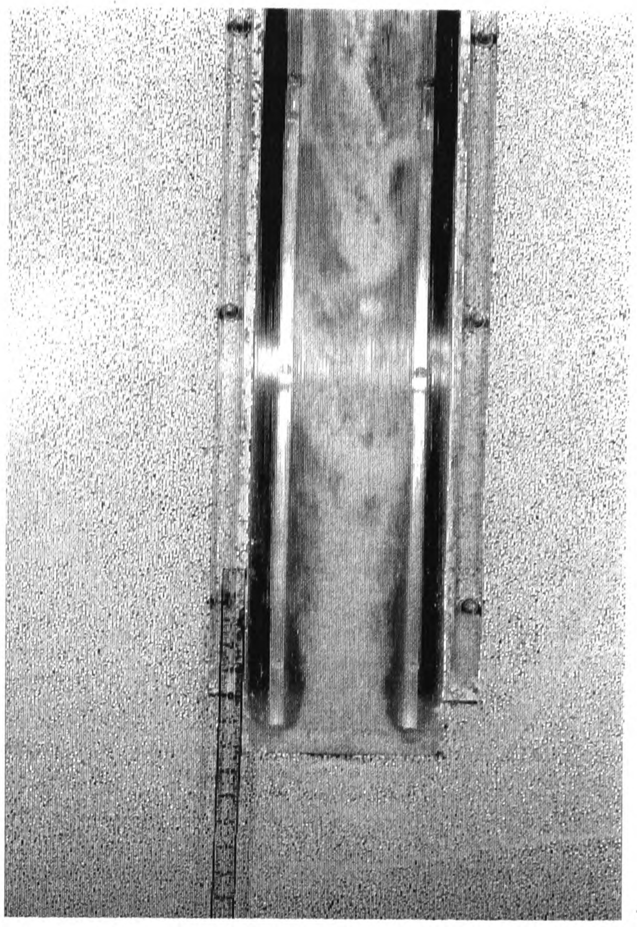


Fig. 4.59

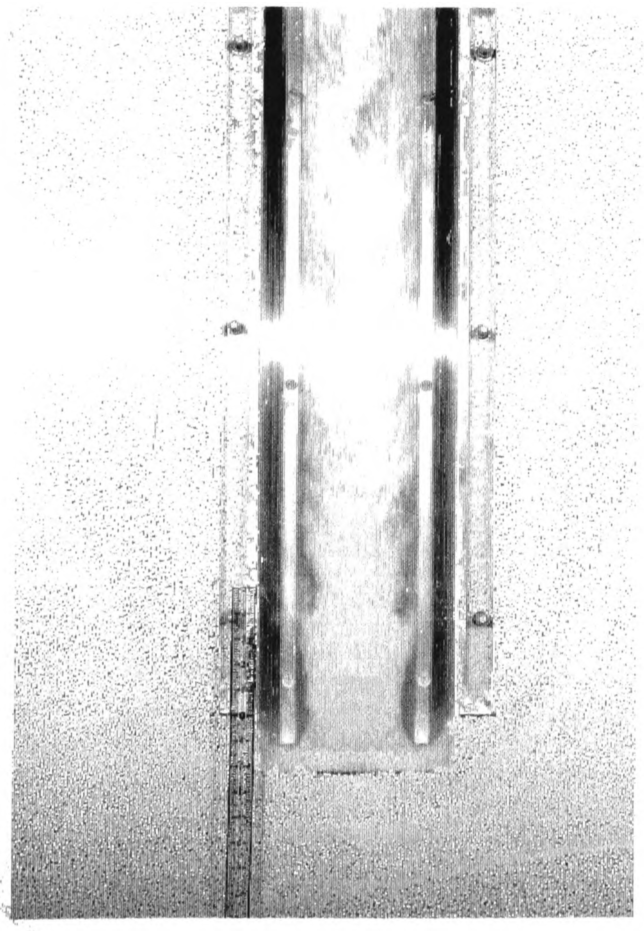


Fig. 4.60

Product - DICALCIUM PHOSPHATE
Superficial Air Velocity = 30m/s

Fig. 4.58 X= +10mm, Throttle = 0%
Fig. 4.59 X= +10mm, Throttle = 40%
Fig. 4.60 X= +10mm, Throttle = 90%

4.4 Criteria for Maximising the Performance of Vacuum Pneumatic Conveying Systems Through Nozzle Design and Operation

This initial test programme has provided an insight into entrainment mechanisms of co-axial tube suction nozzles, and given an indication of how subtle changes in the geometry of and/or throttling the suction nozzle can bring about significant changes in the performance of a vacuum pneumatic conveying system as a whole. A significant observation is that the specific power consumption of such systems can be decreased by adopting one of two possible approaches.
These are:-

1/. Maximising the overall mass flow rate of product commensurate with the power consumption of the existing prime mover, or

2/. Maintaining the same mass flow rate of product and reducing the overall power required to achieve this.

It was found that by making slight changes in the geometry of the inner and outer pipes of a standard suction nozzle relative to one another, the mass flow rate of product could be increased significantly. However, there were factors limiting the extent to which the nozzle position (protrusion) could be increased before either the nozzle choked, or no further increase in performance could be achieved. It was also found that by varying the throttle setting of the modified suction nozzle, the mass flow rate could be increased for any given set of conditions. However, the general trend was, with respect to the benefits of throttling, for a much greater increase in mass flow rate of product for a retracted nozzle configuration than for a protruding nozzle configuration. The reason for this, is that the work has shown that a protruding nozzle configuration is already an efficient self feeding system, and hence the addition of a throttle will have little extra beneficial effect. With a retracted configuration, the standard nozzle is an inefficient feeding system, where the surface of material formed beneath the outer pipe is some distance below the inlet plane of the inner pipe. Therefore, the addition of the throttle creates a pressure drop down the annular chamber tending to "lift" the product into the inner pipe, thereby enabling more to be entrained into the conveying pipeline.

CHAPTER 5. PILOT SCALE STUDIES

5.1 Introduction

Having undertaken a series of tests using the flow visualisation rig, as detailed in the previous chapter, a further series of tests were undertaken, but this time using a pilot scale test rig as described in section 3.4. Apart from its size, this test rig features the same basic components as the flow visualisation test rig, but has a fully circular co-axial tube type of suction nozzle and a Roots type exhauster as the air mover. By undertaking these tests, it would enable a comparison to be made of the results from the two test rigs, to see if there are any marked differences in the performance characteristics of the co-axial tube type of suction nozzle as a result of,

1/. the physical size of the test rig,

2/. using a different design of suction nozzle, and

3/. the effects of using a different type of air mover.

The following sections discuss the results of these tests.

5.2 Wheat Grain

The wheat grain was tested, following the operational procedures detailed in section 3.7.3. for a range of air velocities, nozzle geometries and throttle settings. Figure 5.1 depicts the performance of the "standard" suction nozzle with no throttling of the induced air. It shows that although the actual rates are different from the semi-circular test rig, the general trend of increasing mass flow rate of product with increasing nozzle position is similar. By studying the graph it may be seen that, for a given superficial air velocity, there is no advantage to be gained by extending the nozzle geometry beyond the X= +50mm setting; ie. the mass flow rate remains constant for nozzle extensions up to X= +200mm (the maximum nozzle extension tested). The maximum mass flow rate achieved for wheat grain during these tests was approximately 4 kg/s. The throttling test data is superimposed on these results as shown in Figure 5.2 where it can be seen that for both retracted nozzles and protruding nozzles in the range X= -100mm to +50mm the effect of throttling is quite marked, whereas for a protruding nozzle configuration greater than X= +50mm the effect of throttling is not noticeable. For a throttle setting of 100%, ie. the annular chamber is completely closed, the mass flow rate of product is independent of the nozzle position between the limits of X= -100mm to +200mm, but is dependent upon the superficial air velocity at the inlet to the inner conveying pipe of the suction nozzle. For a completely closed annular chamber, none of the air being drawn into the conveying pipeline flows down the chamber. Therefore the air has to flow through the bed of the product surrounding the nozzle, implying that the use of an outer sleeve may not be necessary.

The implications of these results, and the effects of suction nozzle geometry and throttling on the performance of vacuum conveying systems, will be discussed in the following chapter.

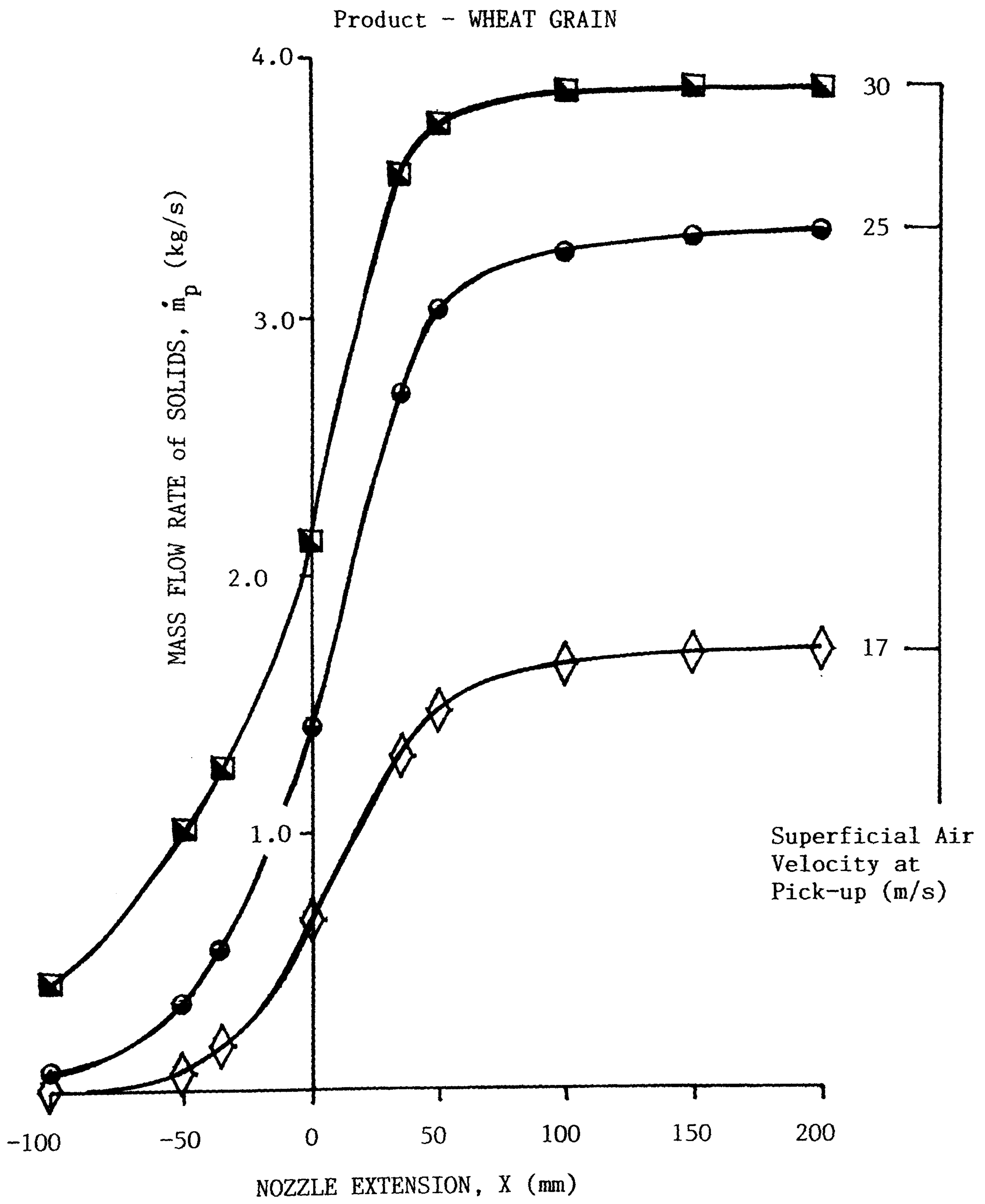


Fig. 5.1 Variation in Mass Flow Rate of Solids With Nozzle Extension for Differing Superficial Air Velocities

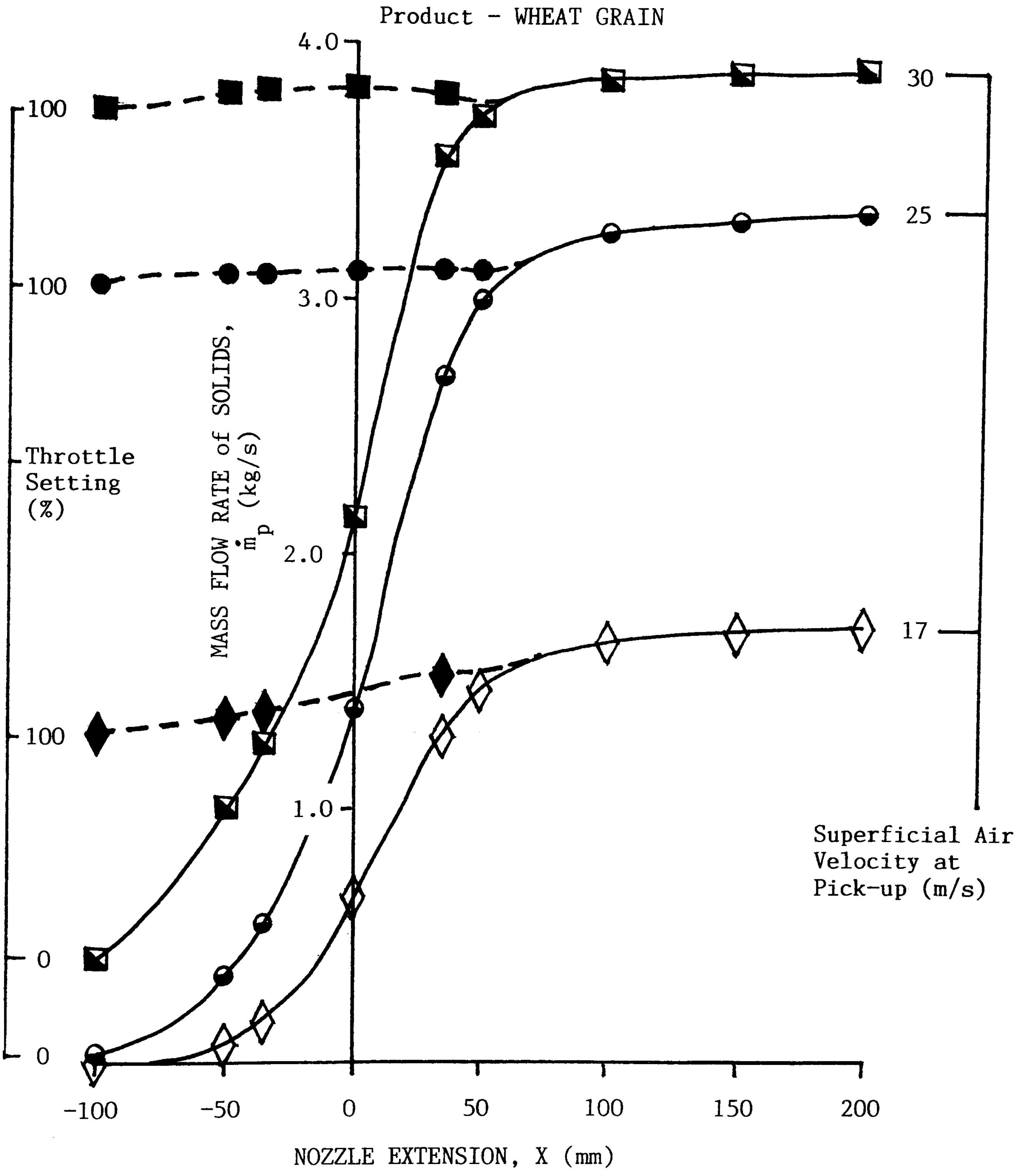


Fig. 5.2 Variation in Mass Flow Rate of Solids With Nozzle Extension for Differing Superficial Air Velocities and Throttle Settings

5.3 Plastic Pellets (Rigidex)

The plastic pellets were tested following the same operational procedures as those used for the wheat grain. The results of these tests for the standard suction nozzle, are shown graphically in Figure 5.3, with the effects of throttling depicted in Figure 5.4.
The general trends are similar to the grain tests, except that the optimum nozzle geometry, where the mass flow rate of product remains constant for any further increases in nozzle extension, is typically X= +40mm and the maximum mass flow rate achieved was approximately 3 kg/s. The maximum mass flow rate associated with each of the three superficial air velocities chosen remained constant for nozzle extensions up to X= +200mm. The effect of throttling on the mass flow rate of plastic pellets was more significant for nozzle configurations in the range X= -100mm to +40mm than for protruding configurations greater than X= +40mm, where the effect was negligible. Once again, for a completely closed annular chamber, the mass flow rate of product is independent upon the nozzle configuration, but is dependent upon the superficial air velocity at the inlet to the inner conveying pipe.
These results also indicate that the outer sleeve may not be necessary, since sufficient quantities of air needed for conveying the product can be drawn through the bed of product surrounding the nozzle.
A further analysis of these results and their implications on system performance will be discussed in the following chapter.

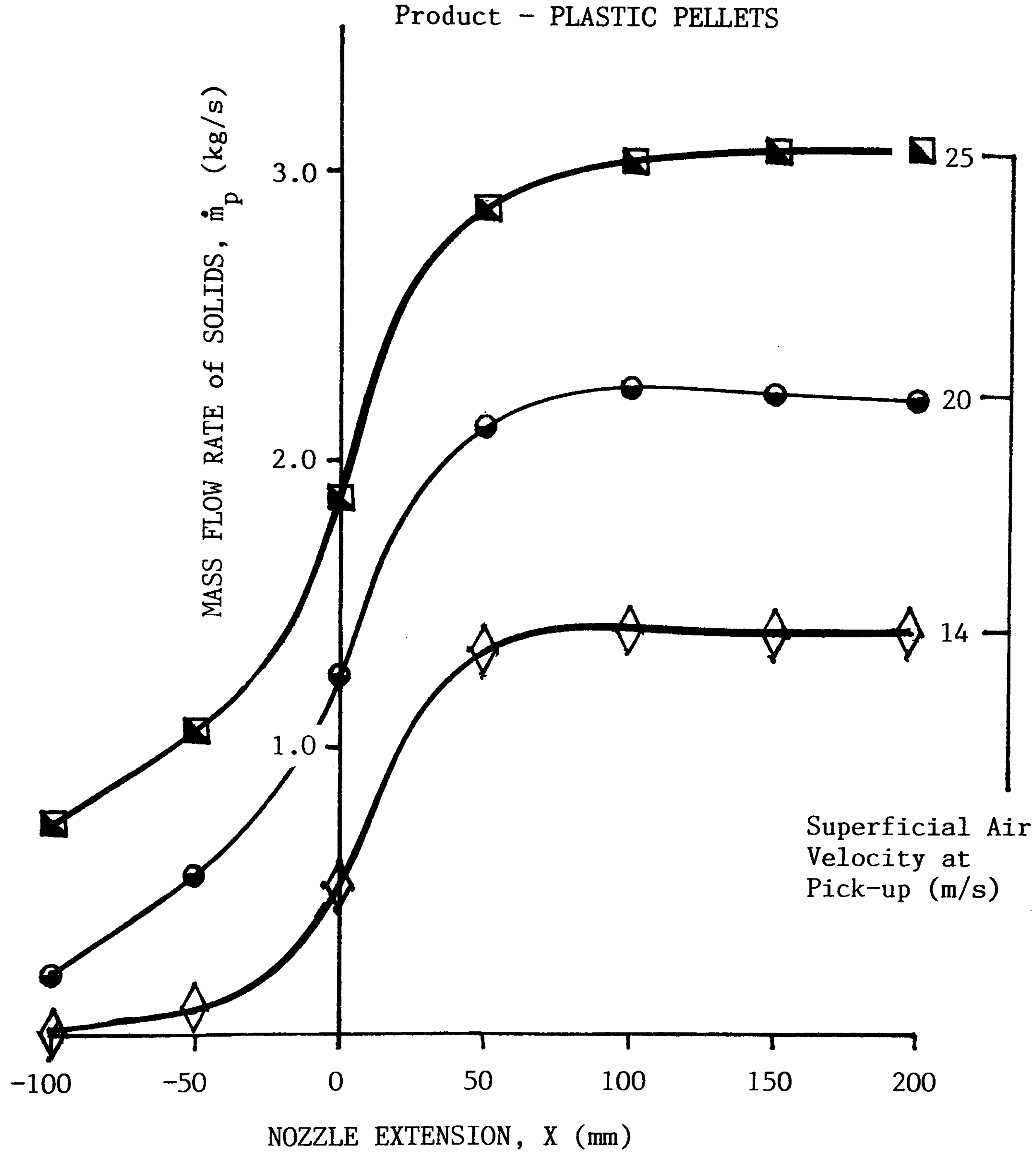


Fig. 5.3 Variation in Mass Flow Rate of Solids With Nozzle Extension for Differing Superficial Air Velocities

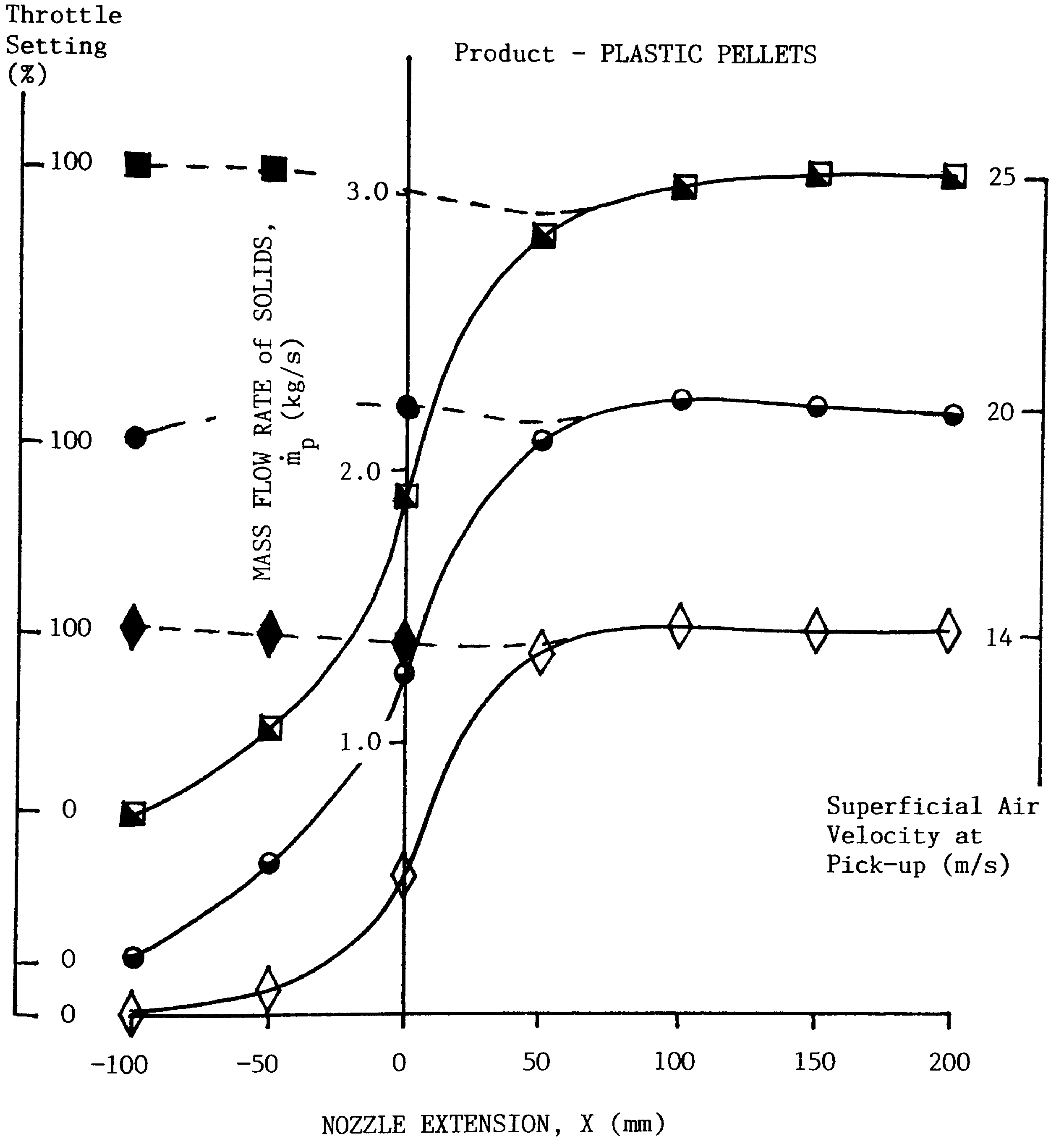


Fig. 5.4 Variation in Mass Flow Rate of Solids With Nozzle Extension for Differing Superficial Air Velocities and Throttle Settings

5.4 Aluminium Hydroxide

The aluminium hydroxide was tested in a similar manner to the other products, but the test programme was revised slightly as a consequence of the results obtained for alumina using the flow visualisation test rig. The results of the flow visualisation work showed that all of the air entering the system passed down the annulus to the entrainment region. Since the pilot scale test rig uses a Roots type exhauster, a 100% throttle setting would not be desirable. The reason being, for throttle settings of 100%, the exhauster inlet vacuum would rise to -0.5 bar gauge causing it to heat up due to the lack of air flowing through it, thereby putting an unnecessary strain on the transmission and electric motor. For multi-stage axial fans, such as the one used on the flow visualisation test rig, no damage will result if the air flow is restricted. Therefore, considering these facts, the throttle settings were adjusted in smaller increments with the maximum throttle setting limited to 90%. The results of these particular tests are shown in Figures 5.5 and 5.6 respectively. The most obvious difference between these results and those for wheat grain and plastic pellets, is that once the optimum nozzle configuration has been reached, any further increases in nozzle extension simply lead to choking. These results, for the standard suction nozzle, have similar trends to those obtained from the small flow visualisation test rig. However, the effects of throttling were not so well defined as compared with the grain and plastic pellets. During some of the tests, the mass flow rate of product would suddenly increase to a new steady state, with a corresponding increase in the conveying line pressure drop. Attempts were made to reproduce the conveying conditions at the "switch" position but were not successful.

Again, the implications of these results will be discussed in the following chapter.

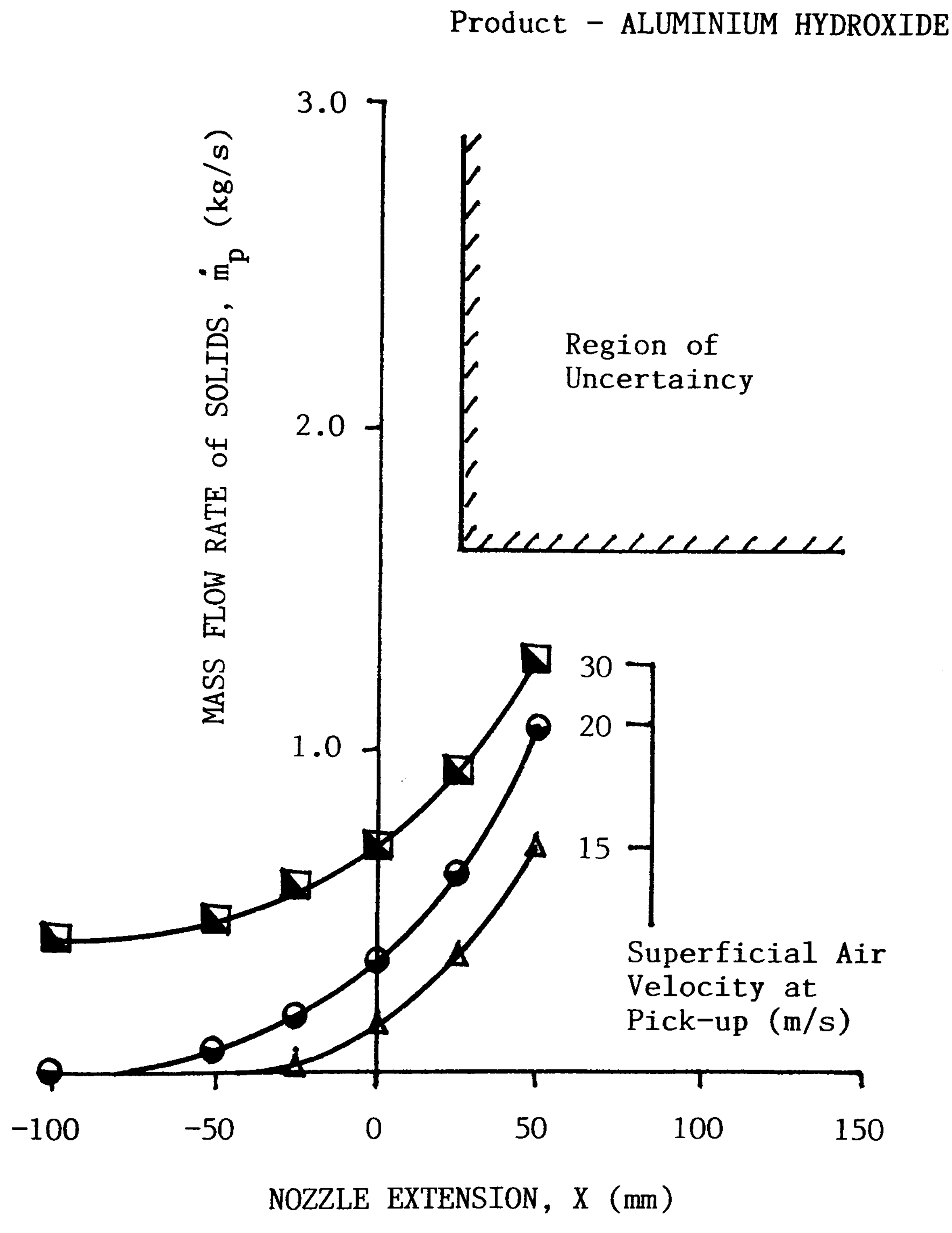


Fig. 5.5 Variation in Mass Flow Rate of Solids With Nozzle Extension for Differing Superficial Air Velocities

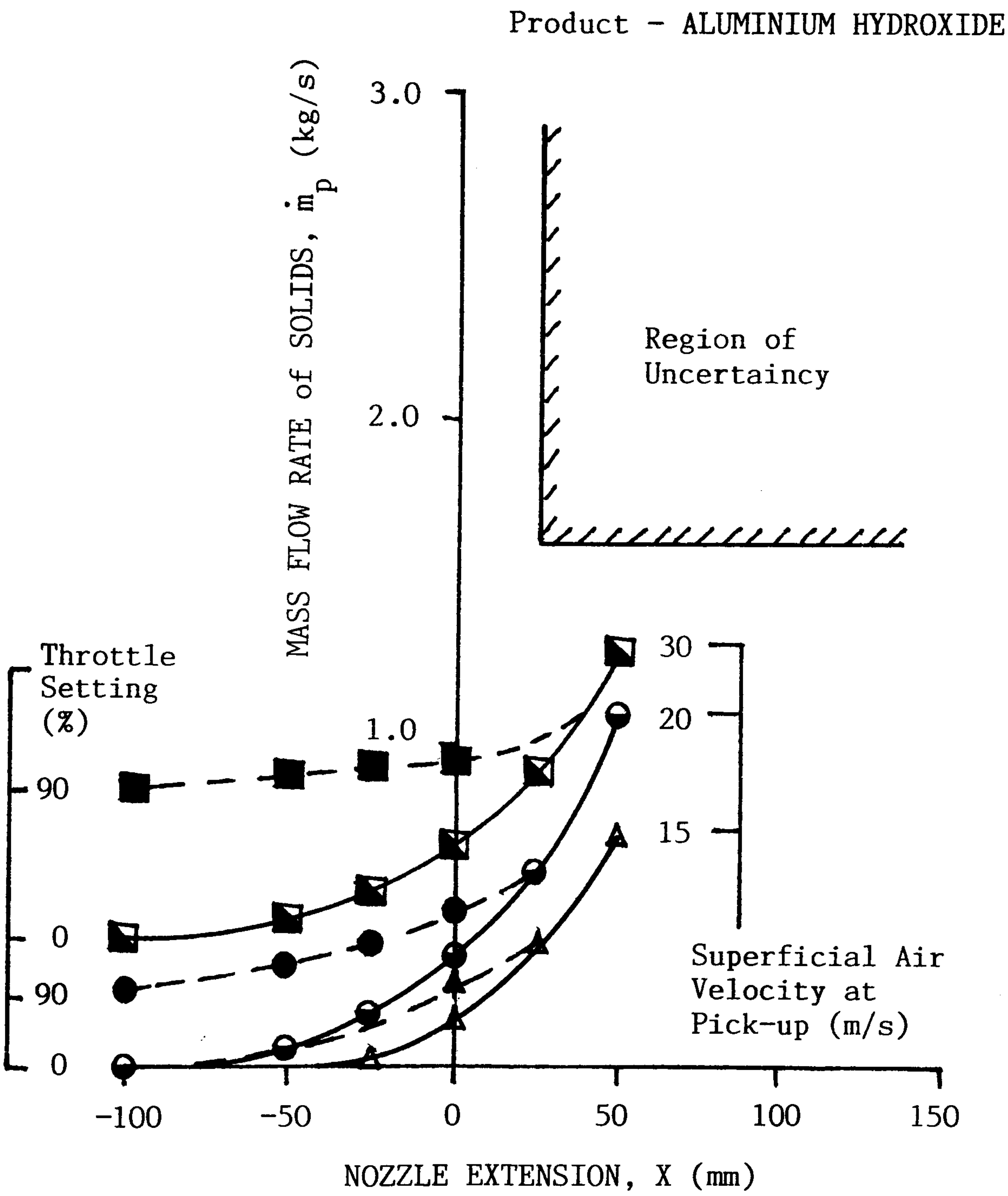


Fig. 5.6 Variation in Mass Flow Rate of Solids With Nozzle Extension for Differing Superficial Air Velocities and Throttle Settings

CHAPTER 6. DISCUSSION and CONCLUSIONS

6.1 Introduction

The purpose of this chapter is to discuss the findings of the tests undertaken with the flow visualisation and pilot size rigs, with a view to establishing the mechanisms by which a co-axial tube suction nozzle entrains product into a vacuum pneumatic conveying pipeline and their effect on system performance. Criteria for predicting the optimum nozzle configuration commensurate with maximising the performance of vacuum conveying systems, together with the effects of different product characteristics are also presented.

6.2 What is the Purpose of the Outer Sleeve ?

The implications of the results presented in the previous chapters is that for the wheat grain, mustard seed and plastic pellets (ie granular and approximately mono-sized products) the outer sleeve is not generally required for a standard suction nozzle, and the position of the outer sleeve, relative to the inner conveying pipe, can have a detrimental effect on the overall performance of the system.

The outer sleeve is not essential for granular products because the air required to convey the product through the conveying pipeline to the receiving vessel can flow through the interstices of the product from the free surface to the entrainment region of the nozzle just as easily as passing down the annular chamber. This has been proven by simulating the removal of the outer sleeve by completely closing the atmospheric air inlet to the annular chamber. This confirms a statement made by Cramp (Ref 11) many years ago that the outer sleeve was of little use when conveying wheat grain.

However, the use of and position of the outer sleeve can be used to purposely limit the mass flow rate of product being drawn into the nozzle. For instance, this study has indicated that for a simple pick-up tube (ie with no outer sleeve) a fixed quantity of product (granular, mono-sized) will be entrained corresponding to a given superficial pick-up velocity. Thus, by adding the outer sleeve it is possible to allow a fixed speed exhauster (fixed pick-up velocity), to entrain varying quantities of product (up to the maximum for that superficial air velocity) into the conveying line by simply altering the position of the inner and outer tubes relative to each other. The sleeve can also be used to maximise the mass flow rate of product induced into a system, whilst limiting the level of vacuum generated to suit the performance characteristics of the air mover being used.

If the foregoing is accepted it is important to realise that the outer sleeve is an essential requirement for entraining fine powders, because the air can not percolate through the interstices of the product and hence must flow along the annular chamber to the entrainment region of the nozzle. This can be confirmed by studying the graphs for alumina and aluminium hydroxide (ie. Figures 4.3 and 5.5) respectively. It demonstrates that, with the standard suction nozzle, there is a limit to the nozzle geometry to prevent choking occurring, irrespective of the superficial pick-up velocities tested (ie. 0 to 60m/s). This implies that if the nozzle is extended too far, for a given air flow, too much material is fed into the system causing the suction nozzle and following vertical riser to choke.

6.3 Entrainment Mechanisms in Standard Suction Nozzle

From the flow visualisation work described in chapter 4 it is clear that depending on nozzle configuration there are a number of different mechanisms by which products can be entrained into co-axial tube suction nozzles. For the purposes of developing a better understanding of these mechanisms the reader is referred to Figure 6.1

With the nozzle in a retracted configuration and at low air velocity, the air flows down the annular chamber and into the conveying pipeline without entraining any product, Figure 6.1.a. As the air flow rate, and hence air velocity, is increased, in turning through 180° this air will scour the surface of the product and may dislodge and entrain particles. As a consequence the conveying rate will depend on the air flow rate employed. From this it may be seen that this is a very inefficient method of achieving the required throughput.

As the nozzle tubes are brought towards the level condition, Figure 6.1.b more product is presented to to the scouring air flow and as a consequence more product is entrained into the conveying pipeline. At the level condition Figure 6.1.c even more product is entrained and the conveying rate will increase accordingly. With the inner tube protruding from the outer tube, if the product is sufficiently free flowing it will flow into the space formed by the tubes Figure 6.1.d. It will then be entrained by the air flowing down the annulus. For this entrainment mechanism any increase in air flow rate will simply remove the product flowing into the region more quickly and the product conveying rate will rise accordingly.

Since the mechanisms corresponding to Figure 6.1.d is, in effect, a product into air situation rather than an air into product mechanism of Figures 6.1.a to 6.1.c it is clear that the former should be a more efficient method of feeding vacuum pneumatic conveying systems and the results presented in earlier chapters confirm this to be the case.

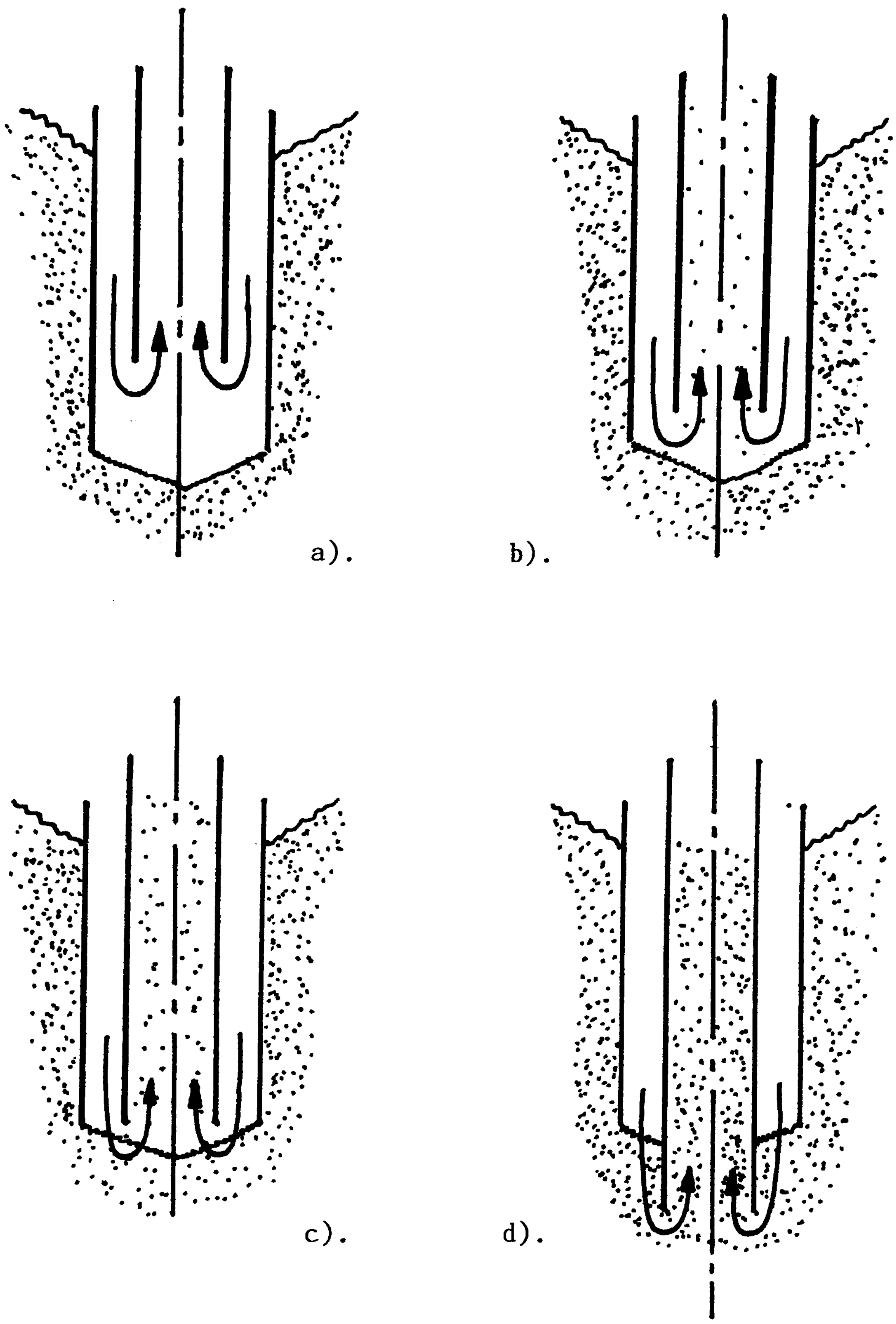


Fig. 6.1 Entrainment Mechanisms of a Standard Suction Nozzle

6.4 Optimum Nozzle Geometry for a Standard Suction Nozzle

As already stated, the results for the standard suction nozzle show that the mass flow rate of product increases with increasing nozzle extension until a "cut-off", or optimum point is reached. Increasing the nozzle position beyond this cut-off point, results in no further increase in mass flow rate for the range of granular products tested.
The cut-off points for the wheat grain and plastic pellets, tested in the pilot scale rig, are X= +50mm and +40mm respectively. Knowing the geometry of the inner and outer pipes, the nozzle extension can be expressed as an angle rather than a linear displacement. Simple geometry can express these dimensions, as the angle made between the horizontal and an imaginary line drawn through the outer edges of the inner and outer pipes as shown in Figure 6.2. This makes it possible to assess the feeding efficiency of the suction nozzle with respect to a material characteristic, such as the drained angle of repose of the products being tested. Appendix B. shows the relationship between the nozzle angle and linear displacement for both test rigs.
The linear dimensions of X= +50mm and +40mm correspond to angles of 62° and 56° for the flow visualisation and pilot scale rigs respectively. If we now consider the values of the drained angle of repose for the wheat grain and plastic pellets (determined from the orifice discharge rate test rig) we find that the angles are 30° and 28° respectively. Thus the cut-off angle is between 1.5 to 2.0 times the drained angle of repose for the granular products tested. This also holds true for the mustard seed and wheat grain tested in the small scale flow visualisation rig. As already mentioned in the previous chapter, the actual mass flow rate of product being entrained is unaffected by throttling once the cut-off point is reached, and the superficial pick-up velocity then governs the rate of entrainment.
Unfortunately this is not the case for the aluminium hydroxide or for the alumina tested in the small scale test rig; as it was found for both of these products that conveying ceases completely, beyond the

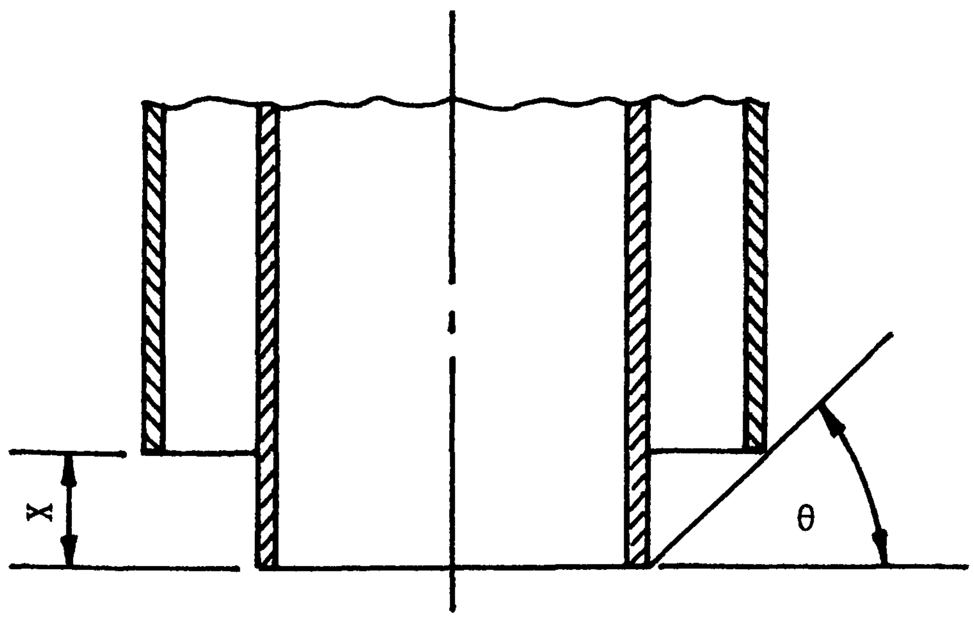


Fig. 6.2 Relationship Between Nozzle Extension and Angle

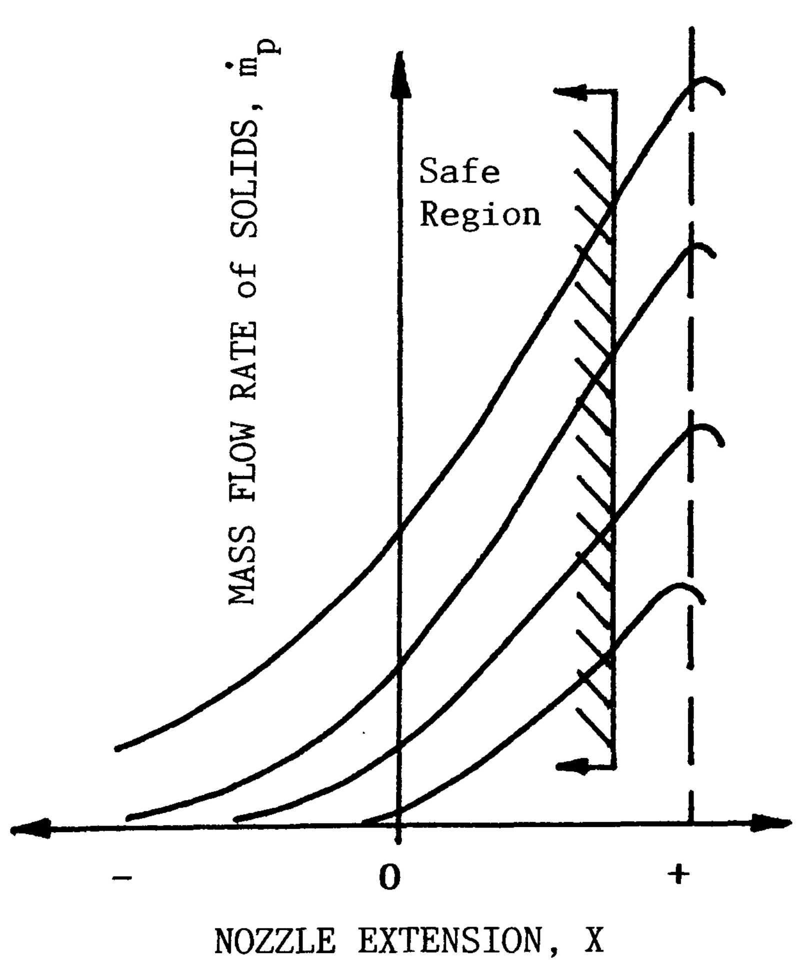


Fig. 6.3 Optimum Geometry for Safe Reliable Conveying

cut-off position. Although the cut-off point is a desirable position to operate with respect to maximising the mass flow rate of product for a given air flow, the conveying conditions are not stable. Thus, for fine powders, the optimum nozzle configuration is in the area to the left of the cut-off point where the conveying conditions are more reliable, as generalised in Figure 6.3. This is of great importance to the users of such pneumatic conveying equipment, because the costs associated with lost production resulting from pipeline blockages etc. often far outweigh the extra energy costs of operating with a reliable but slightly less efficient conveying system.

6.5 Effects of Throttling on System Performance

The effects of throttling the atmospheric air entering the annular chamber can be very significant in terms of increasing the mass flow rate of product. If we consider an operating condition, for a granular product such as wheat grain, corresponding to nozzle position of X= -50mm, superficial air velocity of 30m/s and no throttle, as shown on Figure 5.2, the mass flow rate is typically 1 kg/s. If the throttle is partially closed (say, 50% throttle), thereby restricting the air flowing down the annular chamber, the mass flow rate doubles (approximately). However, the most significant increase is when the throttle is completely closed, causing the mass flow rate to increase by almost 4 times. As the nozzle configuration is moved towards a positive position, the effect of throttling (regardless of the actual throttle setting), diminishes. Once the nozzle geometry is equal to, or greater than the cut-off point the effect of throttling is not discernible. Thus, the general trend, regardless of superficial pick-up velocity, is for the greatest improvement to be made for retracted nozzle configurations and the least improvement for protruding nozzle configurations. Figure 6.4 shows the effect of throttling on mass flow rate for wheat grain tested in the pilot scale rig. Perhaps this is

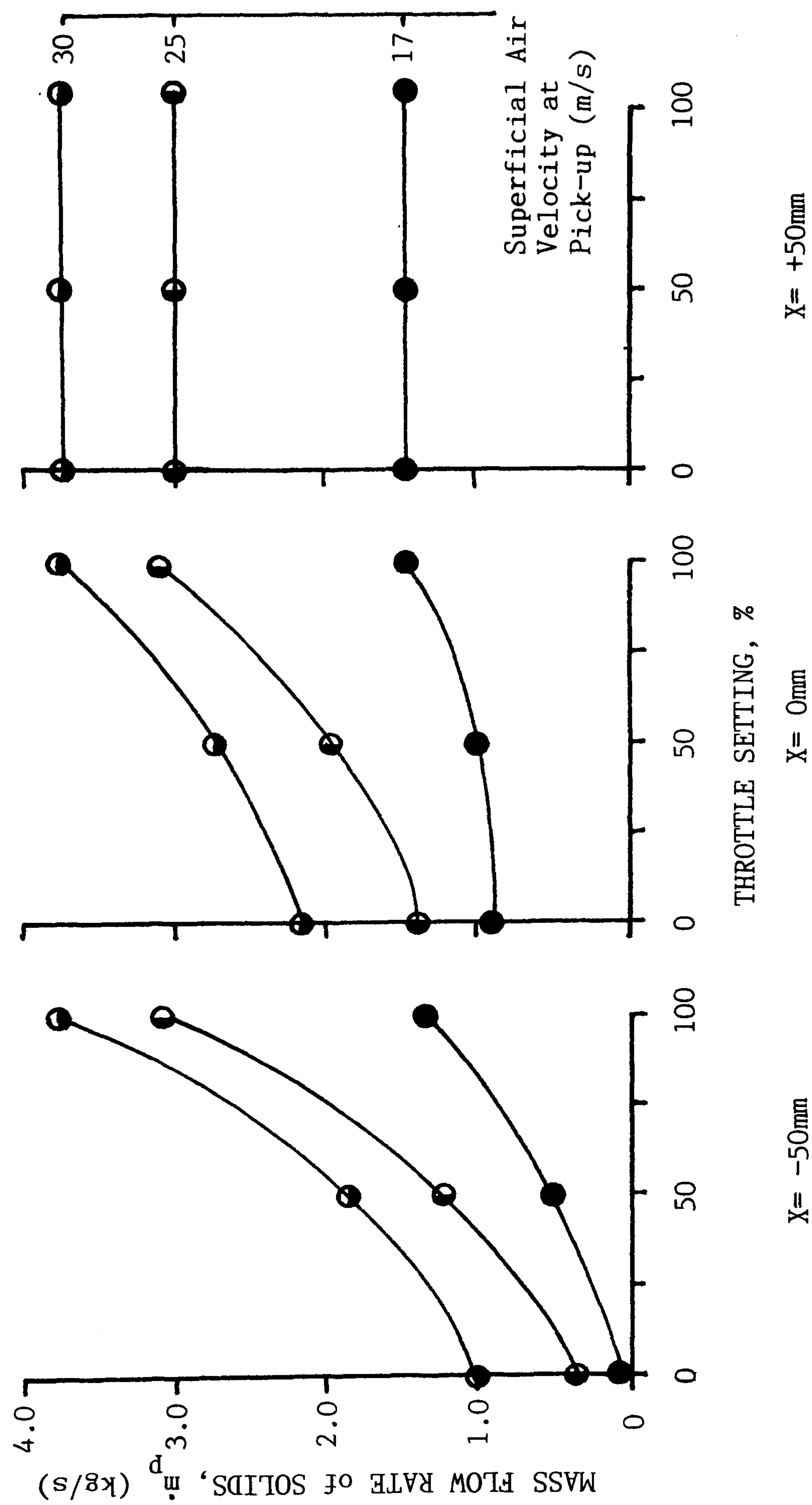


Fig. 6.4 Effect of Throttling on Nozzle Performance for Wheat Grain

not surprising, because the latter is already a considerably more efficient configuration than the former.

6.5.1 How Does Throttling Improve the Mass Flow Rate of Product ?

Based on the preceding paragraph it is clear that throttling can have a positive effect on system performance. However, it is not immediately obvious the mechanism by which this is achieved. There is a general misconception within the pneumatic conveying industry as to the cause of the increase. The general view is, that the throttle controls the rate of air flowing into the conveying pipeline, thus controlling the solids loading ratio (phase density) in the system. This concept is also expressed in the British Standard of Appendix A.

The variation of the flow rate of air, will not only depend upon the effective reduction in the area of the annulus but will also depend upon the performance characteristics of the air mover (exhauster) being used. The performance characteristics for radial fans, axial fans and Roots type exhausters are completely different as shown in Figure 6.5. The radial and axial type fans have flat performance curves as distinct from the Roots type exhauster which has a very steep curve. Thus, the volumetric flow rate of air induced is very dependent upon the inlet pressure or overall system pressure drop. Fans are mainly used for light duty applications where the system pressure is unlikely to change. However, if a restriction is applied to the inlet of the suction nozzle, there will be a pressure drop across it, causing the vacuum level to rise slightly. As a result, the volumetric flow rate of air drawn through the system will reduce. From the general relationships of nozzle performance, the mass flow rate of product drawn into the system is velocity dependent. Therefore, under such circumstances, there is likely to be a reduction in the conveying rate and not an increase. By contrast, the Roots type exhauster has a steep

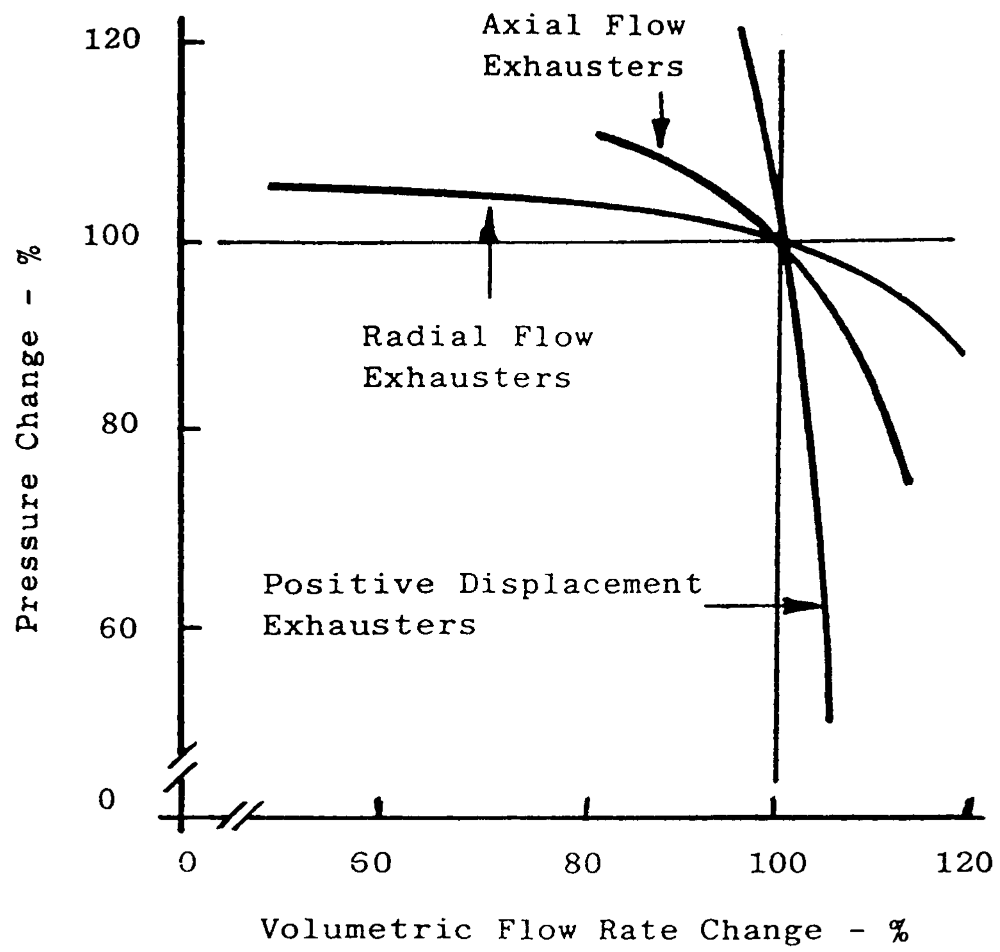


Fig. 6.5 Constant Speed Characteristics of Aerodynamic and Positive Displacement Exhausters

performance curve and can, therefore, maintain a near constant air flow rate through the system, regardless of the vacuum level. Thus, if a restriction is applied to the suction nozzle of a system incorporating a Roots type exhauster, the pressure drop will still rise slightly, but the volumetric flow rate of air drawn into the system will remain relatively unchanged. Therefore, if this argument is to be adopted, there must be another mechanism effecting the system performance for varying throttle settings.

As may be seen from the various photographs relating to section 4.3

throttling the air flow on entry to the annular chamber with a retracted nozzle configuration, produces an effect that 'lifts' the product towards the entraining airstream. The reason for this may be better appreciated by referring to Figure 6.6. This demonstrates the effect of throttling on the measured static pressure of the air as it flows down the annular chamber of the nozzle arranged with a basic configuration corresponding to X= -20mm when conveying alumina in the flow visualisation rig. In essence, for a given entrainment velocity, increasing the degree of throttle increases the partial vacuum of the entraining airstream as it flows down the annular chamber. Since the pressure of the air in the interstices of the product surrounding the nozzle will be close to atmospheric pressure, the difference in pressure in this region will then result in a force that effectively lifts or elevates the material in the direction of the the pressure gradient. By this way material is then presented to the entraining airstream.

Whilst this is a plausible explanation with respect to the operation of suction nozzles that have the inner tube retracted inside the outer tube, for nozzles of the reverse geometry it is clear that the effect will be less marked, if at all, since this study has already shown this to be a more effective feeding configuration.

In conclusion it is evident, therefore, that the throttle is simply a device used to create a pressure gradient across the bed of product to lift the product into the entrainment zone, but is far more effective at creating pressure differentials down the annular chamber than using the velocity of air alone.

6.6 What Dictates the Mass Flow Rate of a Granular Product?

We have seen from the previous sections, that there is no advantage to be gained by increasing the nozzle configuration beyond twice the drained angle of repose for granular mono-sized products, and that

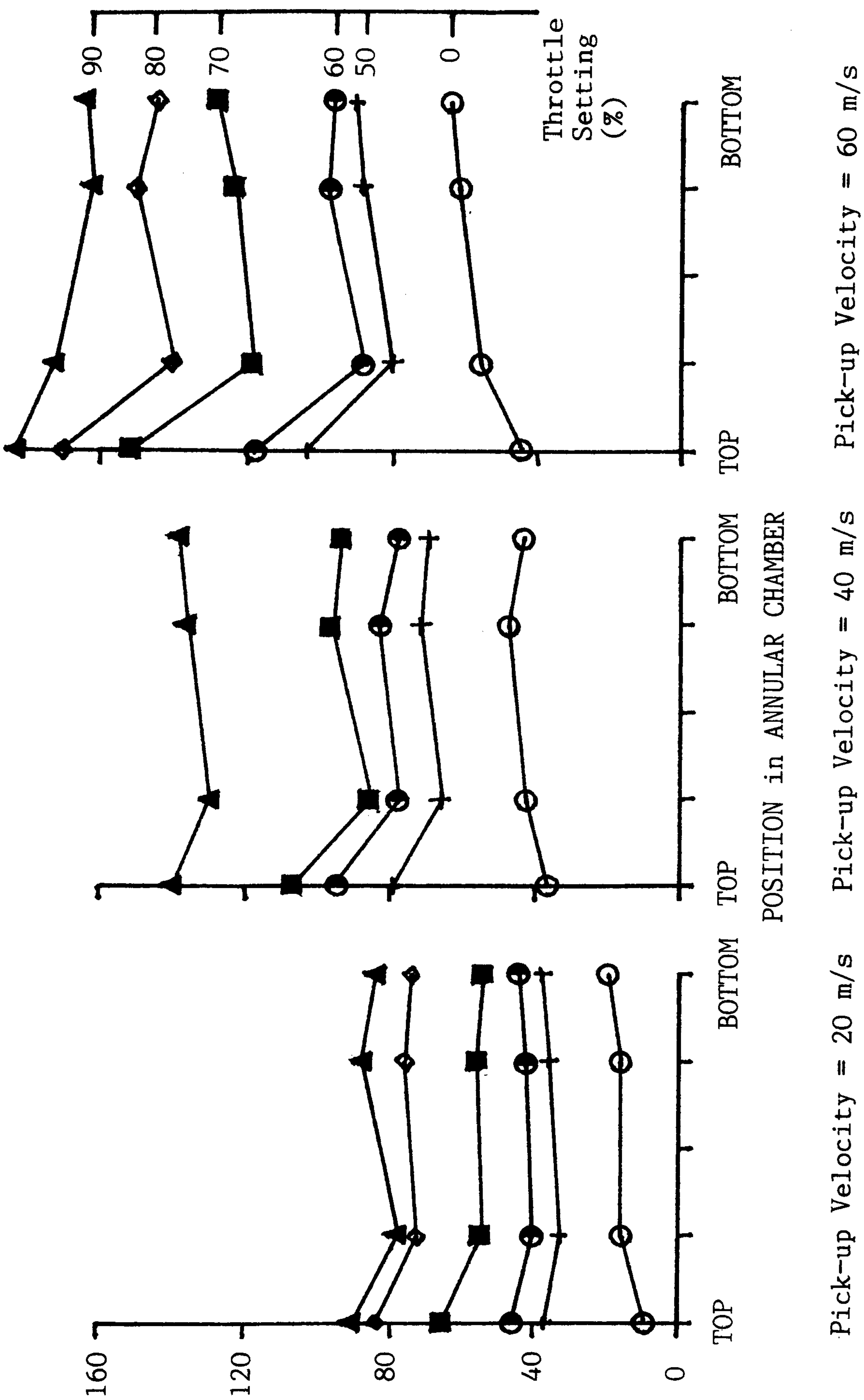


Fig. 6.6 Effect of Throttling on The Pressure Distribution in The Annular Chamber - for a Nozzle Extension of X= -20mm

throttling has no effect beyond the cut off position. Thus, there must be other factors influencing the rate of entrainment into the nozzle when operating at the cut off point and, more importantly, how these factors are related.

6.6.1 Limiting Material Discharge Rates

It has been noted by the author that there might be an upper limit to the discharge rates that can be achieved for a given nozzle diameter regardless of its configuration. The basis for this statement is that the product has to flow down to the nozzle entrainment zone under the influence of gravity and also by the drag effect caused by the air flowing through the interstices. Neglecting the drag effect for the time being, it is possible to liken the flow to that of product discharging from a circular flat bottomed hopper having a circular outlet. The important feature to note is that there will be a "rough wall" (funnel flow) condition at the interface between the "live" product flowing over the material forming the "dead" region. It was, therefore, decided to construct a small test rig (as detailed in section 3.6) to enable some measurements to be taken of discharge rates through differing size orifices.
Figure 6.7.a. shows the variation in discharge rate with orifice area for the those products tested in the pilot scale test rig. If we consider a nozzle geometry equal to or greater than the cut-off point, the outer sleeve can be ignored. This means that any of the actual pneumatic conveying rates achieved can be assessed with respect to the dimensions of the inner pipe alone. Since the product initially "sees" the outer edge of the conveying pipe, as it is drawn in by the air stream, the outside diameter of 60.5mm should be used. This corresponds closely to the 62mm diameter orifice plate. The discharge rates, through this size of orifice, for the plastic pellets and wheat

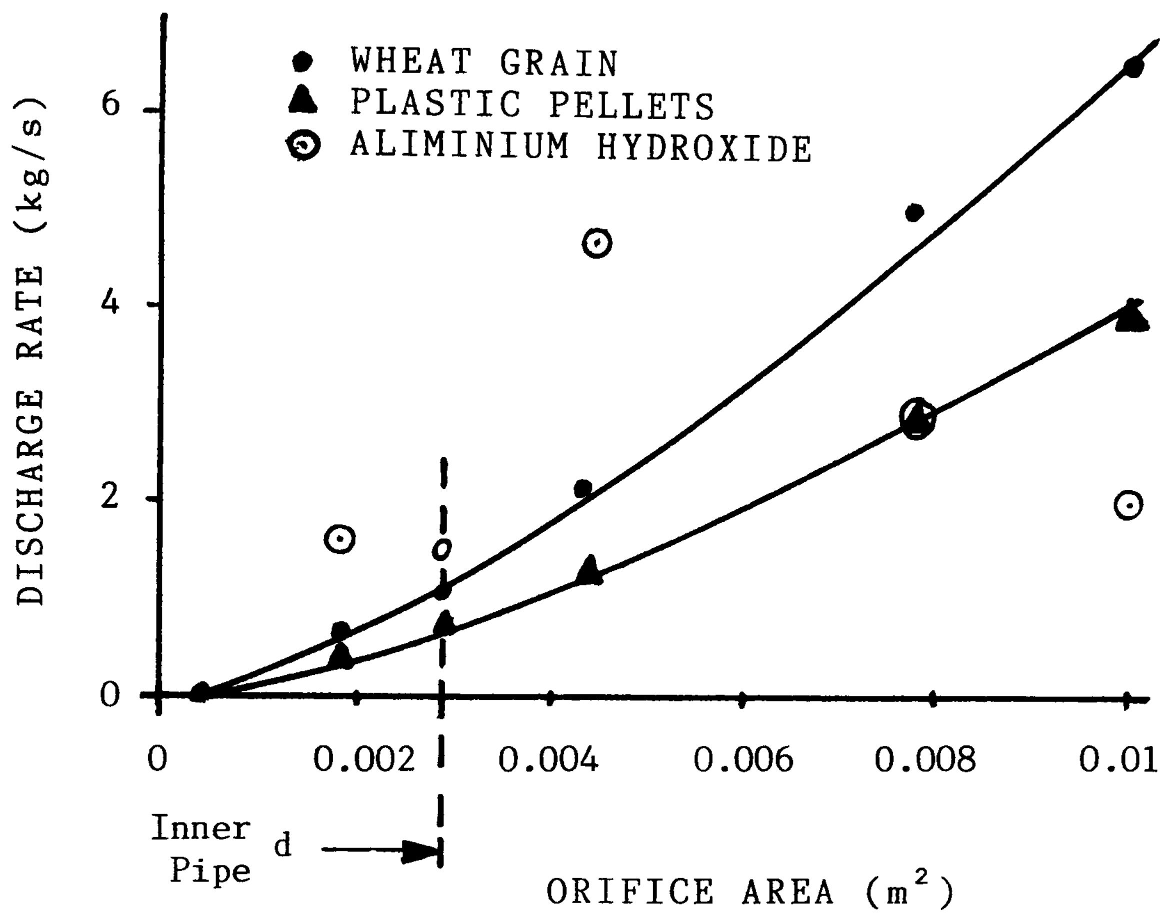


Fig. 6.7.a Variation in Discharge Rate with Orifice Area

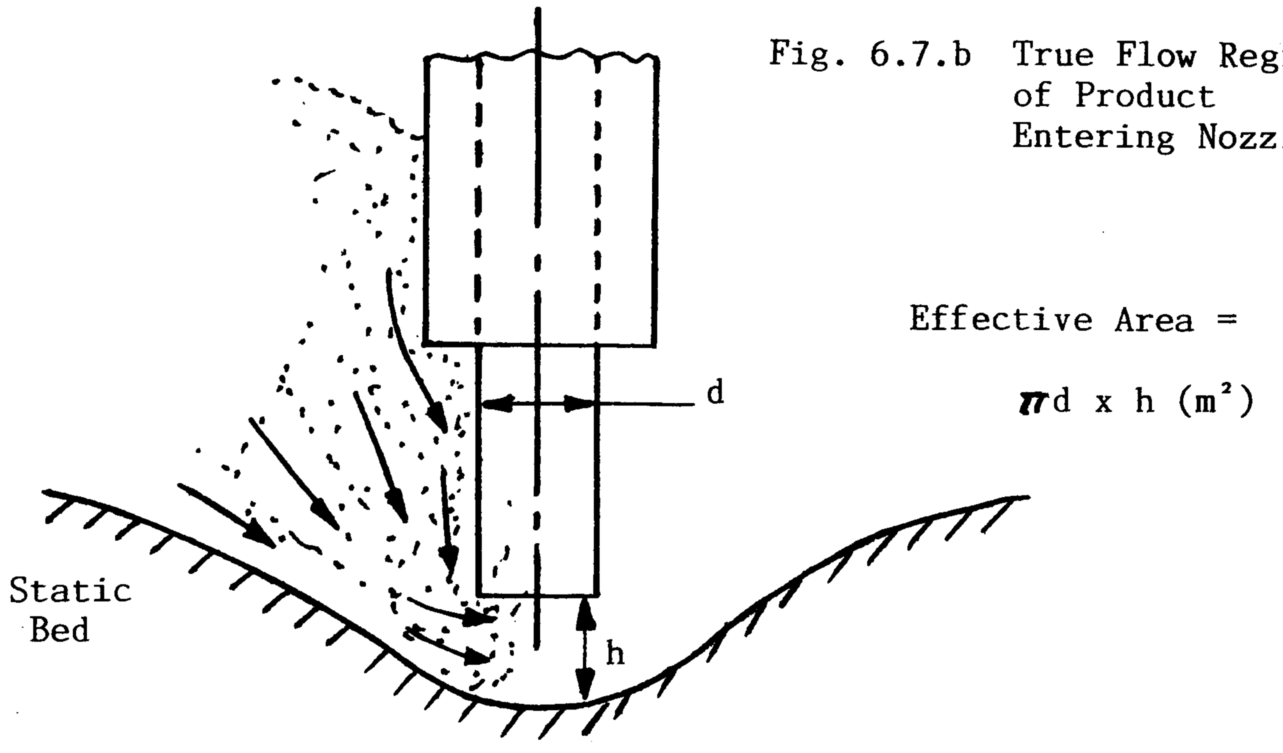


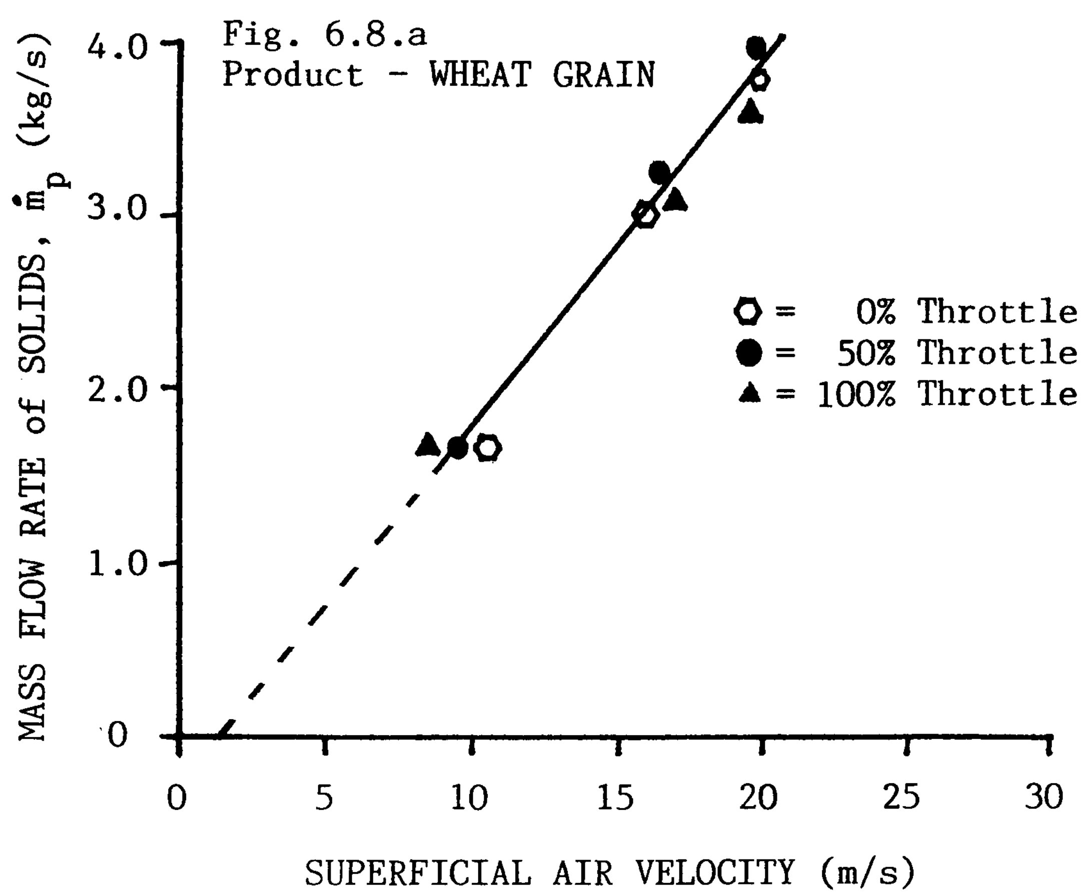
Fig. 6.7.b True Flow Regime of Product Entering Nozzle

grain are 0.7 kg/s and 1.15 kg/s respectively. By comparing these results with corresponding data in Chapter 5, it may be seen that the conveying rates are far higher than the orifice discharge values (typically 3.4 to 4.4 times greater). However, the maximum mass flow rates plotted are for the pick-up velocities attained in this series of tests and, therefore, are not necessarily indicative of the ultimate conveying rates that might be achieved for even higher air velocities. Thus, it is possible for a situation to arise where, the flow of product to the nozzle has a limiting effect, assuming the nozzle can entrain and convey any quantity of product that is presented to it. However, the present set of results suggest that the difference between the conveying rates and orifice discharge rates is attributable to the air flowing through the interstices of the product, thus dragging the individual particles towards the nozzle inlet at a higher velocity than by gravity alone.

If the actual flow patterns of a granular product flowing into a protruding nozzle configuration are observed, the flow is as depicted in Figure 6.7.b. It is clear to see that the product flows horizontally through an annular "slot" (nominally the same diameter as the outer wall of the conveying pipe) prior to being raised into the conveying line. Thus if discharge rate tests were undertaken in a rig simulating the actual flow patterns rather than the orifice test rig used, the results may be more indicative of the true discharge rates.

6.6.2 Pick-Up Velocity

It has already been stated that the mass flow rate of product is constant for a given pick-up (superficial air) velocity regardless of the nozzle geometry and throttling, provided that the nozzle geometry is equal to, or greater (protruding further) than the cut-off point. Thus, the graphs shown in Figures 6.8.a. and 6.8.b. for wheat grain and plastic pellets respectively can be plotted showing the variation in



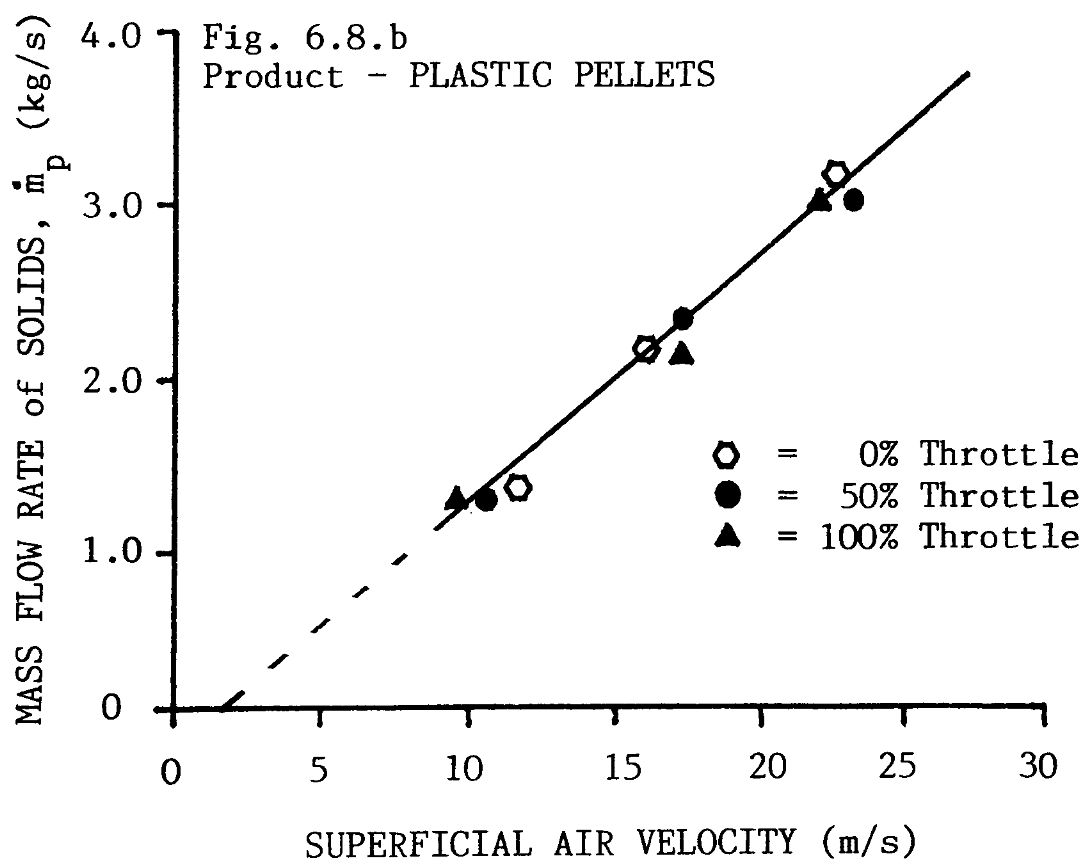


Fig. 6.8.a and b Variation in Mass Flow Rate of Solids With Superficial Air Velocity

mass flow rate of solids with superficial air velocity. The mass flow rate of solids can also be expressed in the form of a Nozzle Loading term, ie. the mass flow rate of solids per unit area of inner conveying pipe. The significance of this will be presented shortly. It can be seen that straight line relationships exist for both products. For reasons of clarity, only the data representing the cut-off point of X= +50mm for the wheat grain and plastic pellets have been plotted, since their respective data for the X= +100mm and X= +200mm nozzle configurations lie on the X= +50mm points.

The straight lines, have been extrapolated to intersect the superficial air velocity axis. The intersection points for the wheat grain and plastic pellets are 1.5 m/s and 2 m/s respectively, implying that these velocities are their respective minimum pick-up velocities. However the lowest pick-up velocities achieved in the pilot scale test rig were, typically, between 10 m/s and 15 m/s for the wheat grain and plastic pellets. However, these were not practical minimum conveying velocities. The minimum velocity required to keep a single particle just "floating", will be dependent upon the weight, drag and the buoyancy effect for the product.

A Nozzle Loading term has been introduced rather than simply calculating an actual mass flow rate of product, because it enables different size suction nozzles of differing cross sectional areas to be assessed. The only other data that fits this relationship is that for a retracted nozzle which is fully throttled, ie. the inlet to the outer annulus is completely closed. This is to be expected since this configuration can be considered to be exactly the same as for a protruding nozzle with or without throttling.

6.6.3 Other Relationships

Since the system design engineer is likely to know the type of air mover to be used for a proposed new system, there will be a limit to the maximum vacuum that can be generated by the air mover chosen. Thus, although the pick-up velocity relationship gives an indication of the nozzle loading for a given pick-up velocity, it does not take into account the system pressure drop ie. equivalent to the vacuum within the receiving vessel. Therefore, it is necessary to consider other ways of presenting the data gathered from the test rig. As already mentioned, the engineer is likely to know the maximum vacuum that can be generated, the typical mass flow rate of product required and, perhaps, a pick-up velocity commensurate with the product in question. The information that will be required is the internal diameter of the conveying line and the volumetric flow rate of air required. The volumetric flow rate of air will enable the air mover to be specified for the required duty.

Graphs can be plotted as depicted in Figures 6.9. and 6.11 for wheat grain and plastic pellets respectively, showing the variation in nozzle loading against a System Factor (system pressure/pick-up velocity) term. This term has been developed, since the system pressure and pick-up velocity are inter-related by the performance characteristics for the air mover. Thus, system pressure drop, pick-up velocity and mass flow rate of product are inter-related. An important feature of these graphs is that the data points for all nozzle configurations, ie retracted with and without throttle, protruding with and without throttle can be plotted depicting a single general trend. Unfortunately, these graphs do not portray the nozzle configuration required or the specific power consumption, to achieve the desired set of operating conditions. However, if we now plot the nozzle loading ratio against specific power consumption, as depicted in Figures 6.10 and 6.12 for wheat grain and plastic pellets respectively, we find that a family of curves results. Each of these curves depicts a particular nozzle geometry/throttle setting combination and from this it is possible to choose the best combination (ie most energy efficient) for the throughput under consideration.

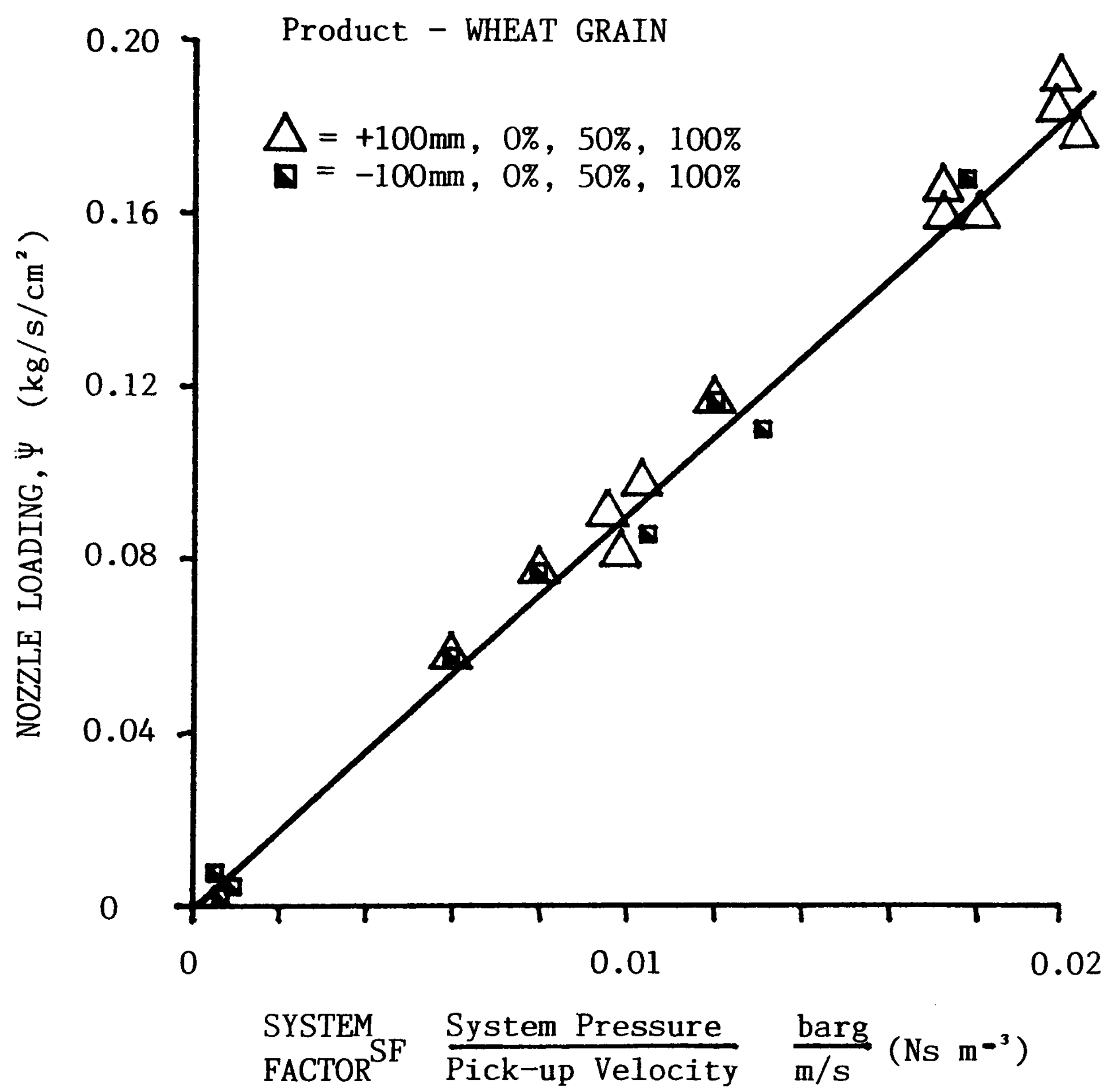


Fig. 6.9 Relationship Between Nozzle Loading and System Factor

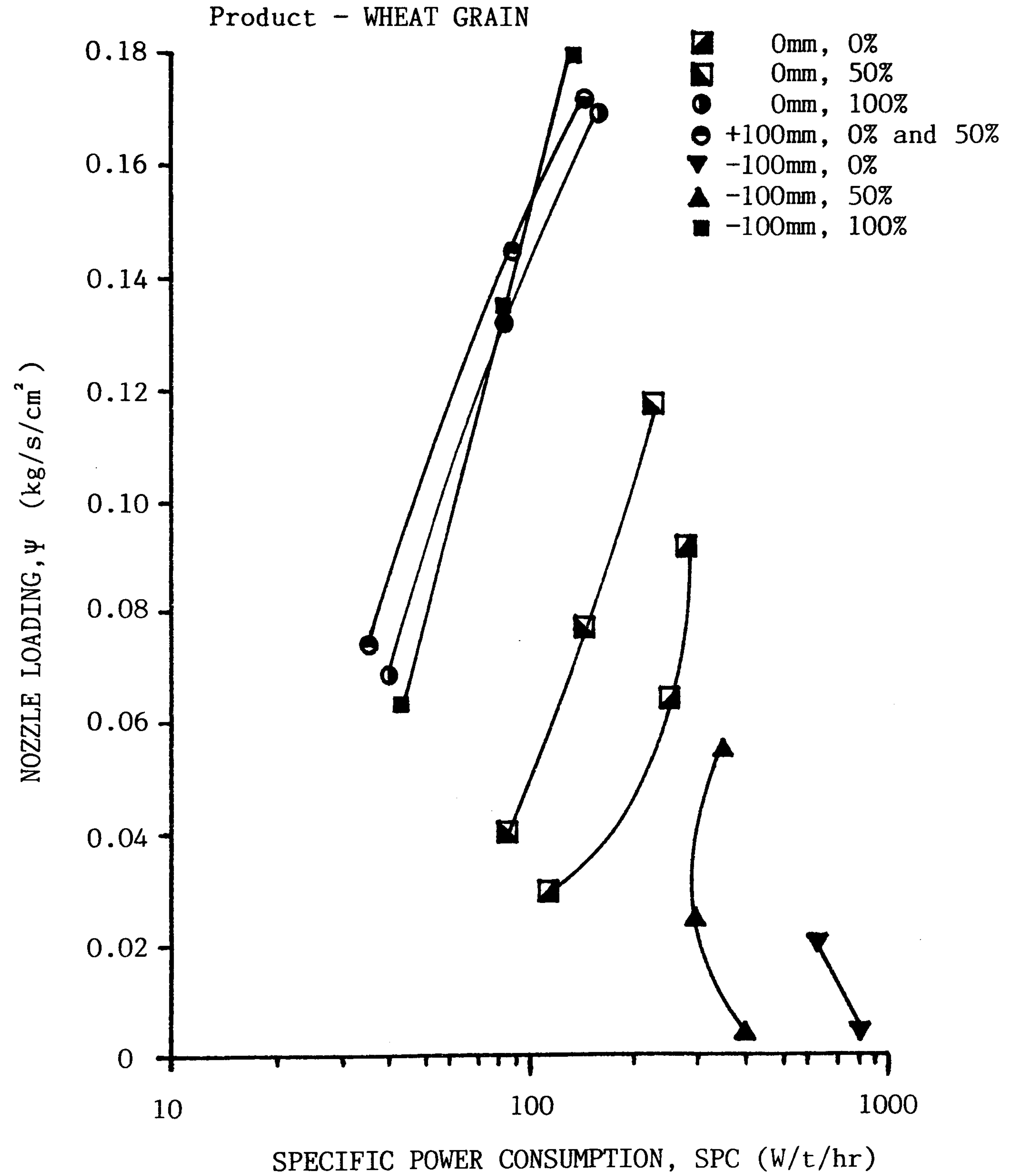


Fig. 6.10 Relationship Between Nozzle Loading and Specific Power Consumption

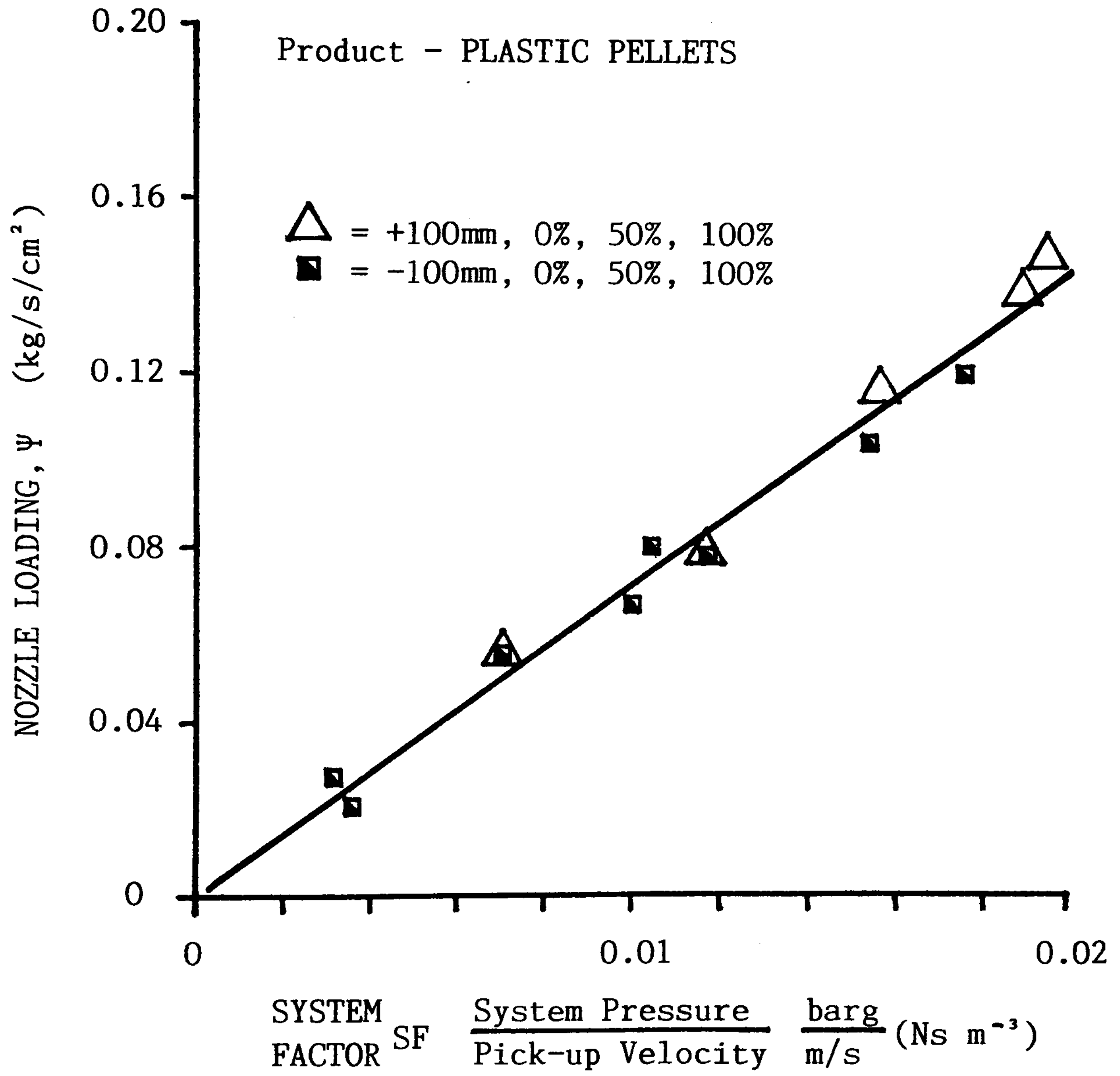


Fig. 6.11 Relationship Between Nozzle Loading and System Factor

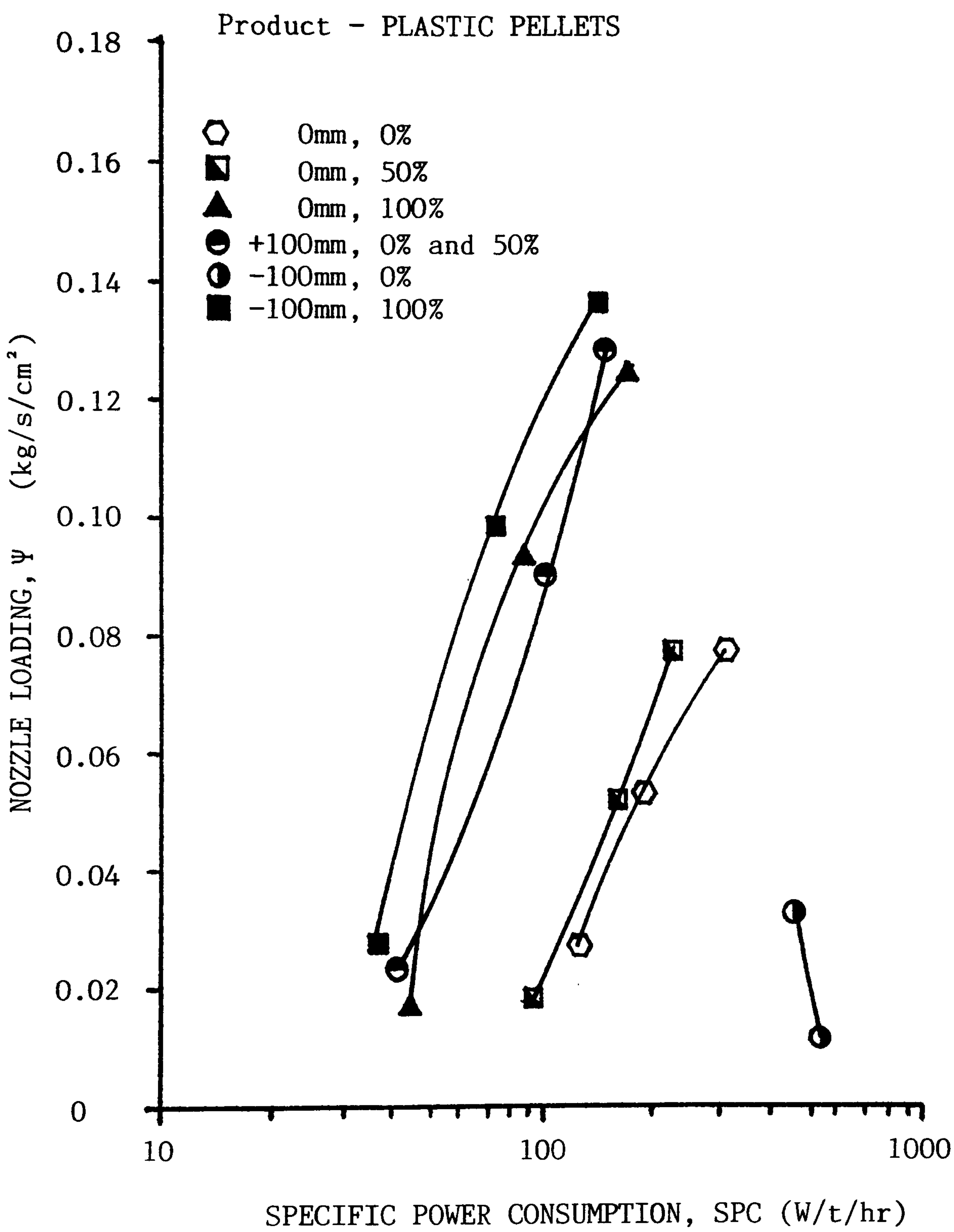


Fig. 6.12 Relationship Between Nozzle Loading and Specific Power Consumption

6.7 What Dictates the Mass Flow Rate of a Fine Product?

As has already been stated, for alumina and aluminium hydroxide, there is a maximum nozzle position beyond which the system chokes and ceases to convey. This is in complete contrast to the granular products, where the outer sleeve can be removed making little difference to the operation of the system. If the nozzle position is extended from a level position to a maximum protruding point, the air flowing down the annulus has to pass through the powder that is flowing into the space formed immediately beneath the outer sleeve. The range of nozzle extensions tested varied from X= -100mm (-2d), where d is the internal diameter of the inner conveying pipe, to X= +100mm (+2d) for the alumina and aluminium hydroxide. If we compare the full size operational system unloading alumina described by Davies et al. (Ref 18) the suction nozzle geometry varied between X= -60mm (-0.1d) to -100mm (-0.17d). These figures equate to X= -5mm and -10mm for the pilot scale suction nozzle. The actual difference in mass flow rate of product for such small nozzle movements could easily be less than the experimental errors. Graphs similar to those of Figures 6.9 and 6.10 can be plotted for the aluminium hydroxide as shown in Figures 6.13 and 6.14.

6.8 Material Characteristics

Having proposed a graphical means of predicting the parameters for the design of a vacuum conveying system for different products (wheat grain and plastic pellets); it would now be sensible to consider which material characteristics, if any, have a significant effect on the overall system performance.

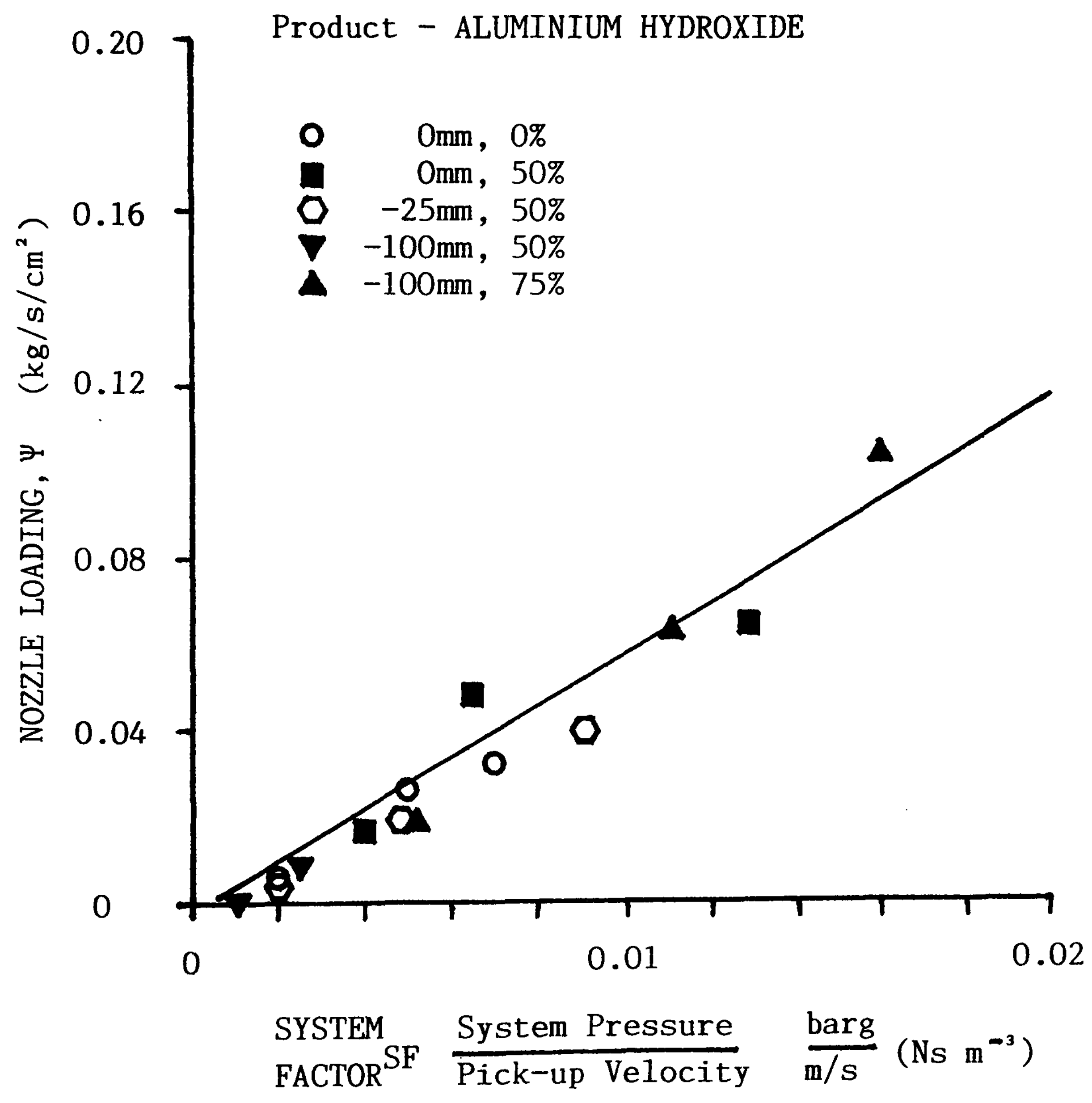


Fig. 6.13 Relationship Between Nozzle Loading and System Factor

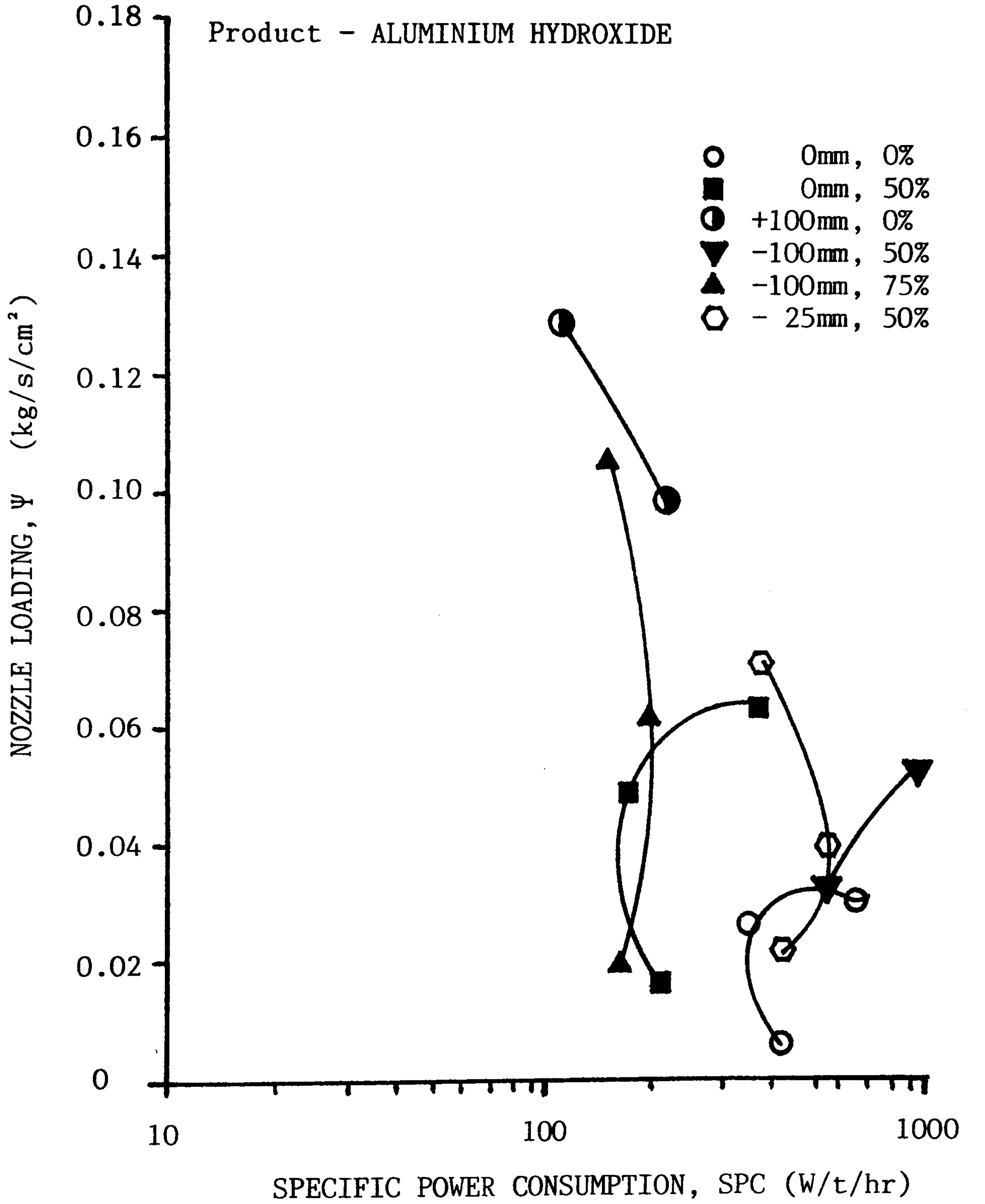


Fig. 6.14 Relationship Between Nozzle Loading and Specific Power Consumption

6.8.1 Bulk Density

It was discussed, in section 5.2.2, that the maximum mass flow rate of wheat grain conveyed was approximately 3.88 kg/s whereas for a similar pick-up velocity, the maximum mass flow rate of plastic pellets was approximately 3.05 kg/s, giving a ratio of 1.27. If we compare the poured bulk densities for the two products, a ratio of 1.58 results (820/518). Although these ratios are not similar, it might be a reasonable basis for obtaining an approximation of the performance of a new material in relation to the performance of a known product.

6.8.2 Permeability

It is clear from the results of all of the tests undertaken that the boundaries of the performance characteristics of the co-axial tube suction nozzle are dependent upon the type of product being entrained into the conveying pipeline by the nozzle. The coarse granular products have shown that with a throttled suction nozzle the mass flow rate of product is independent of the nozzle position. In the extreme it is possible to remove the outer sleeve without any adverse effect on the maximum flow rate. This would not be possible with the fine powdered products. This tends to suggest that the permeability of the product will influence the extent of the performance envelope. Thus, the permeability factor, U, for some of the products tested was determined according to the procedures outlined in Ref 22. The following results were obtained,

$$\left.\begin{array}{lcl} \text{Alumina} & = & 2.77 \times 10^{-6} \\ \text{Dicalcium Phosphate} & = & 3.0 \times 10^{-5} \\ \text{Plastic Pellets} & = & 1.96 \times 10^{-4} \\ \text{Wheat Grain} & = & 1.36 \times 10^{-4} \end{array}\right\} (m^4\ Ns^{-1})$$

The order of these figures tend to match the general trend of, the finer the product is, the more problematic it will be and the more confined the operating range of nozzle configurations will be. However, apart from this general conclusion this parameter cannot be used for the purposes of design.

6.9 Concluding Remarks and Suggestions for Further Work

This study has shown that, based on the evidence of the test data gathered, the following conclusions can be drawn :-

- For granular products having a near mono-size distribution, the maximum mass flow rate of product will occur with a protruding nozzle configuration equivalent to twice the drained angle of repose of the product,

- No further advantage can be gained by increasing the nozzle geometry beyond this optimum (or cut-off point) for granular products,

- The mass flow rate of product is proportional to the superficial pick-up velocity for nozzle geometries equal to or greater than the cut-off point,

- Throttling the air entering annular chamber can have a significant effect on the performance of a co-axial tube suction nozzle when operating in a retracted configuration, whereas the effect is not so marked for the protruding, more self feeding configurations,

- A suction nozzle loading factor for granular products ma:[assessed from a knowledge of the system pressure drop (in this c the conveying line pressure drop) and the superficial pi velocity for granular products, irrespective of any combinatio nozzle geometry and throttle setting.

- From a knowledge of the nozzle loading factor the optimum n geometry and throttle setting commensurate with minimum consumption may be determined.

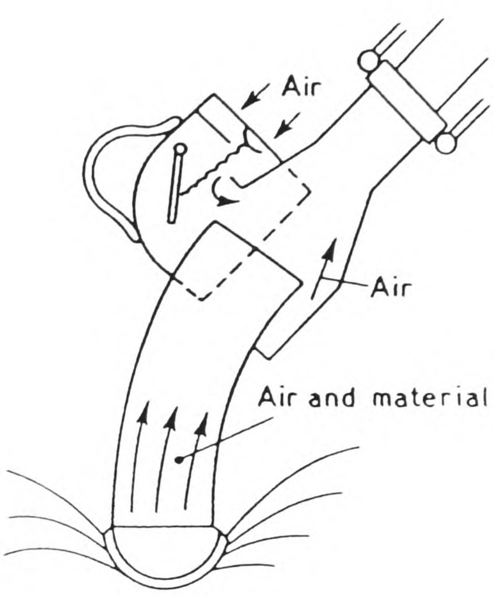
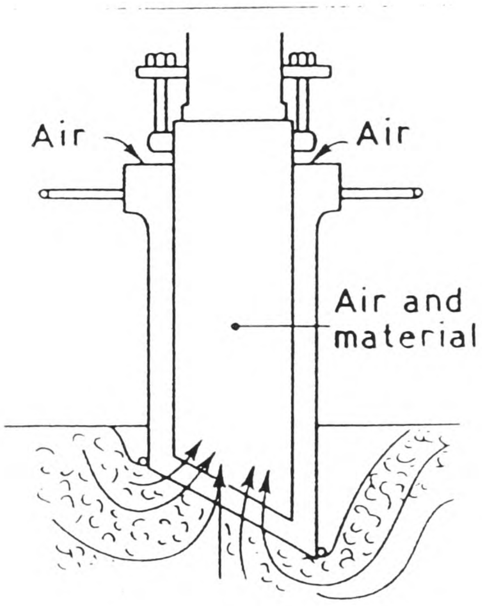
- For systems containing fixed speed exhausters, the mass flow n of product can either be up-rated to the maximum possible, or energy efficient or even down-rated by altering the configur of the suction nozzle and the actual pick-up velocity at the n inlet plane.

At the time of writing the author has been unable to compare t evidence gathered with actual operating conditions for full operational vacuum systems unloading grain and other gra products. This would enable the level of confidence and limita of the extent of scaling from the small scale test data to pro system size to be assessed. Thus, one area of further work could compare the test data with full size systems be they, 100 - 500 ship unloaders or 5 t/hr "in-plant" conveying systems.

To present a more complete picture, a range of products shoul tested having various particle and bulk characteristics. This enable specific product characteristics to be assessed, and determine which of these are most dominant.

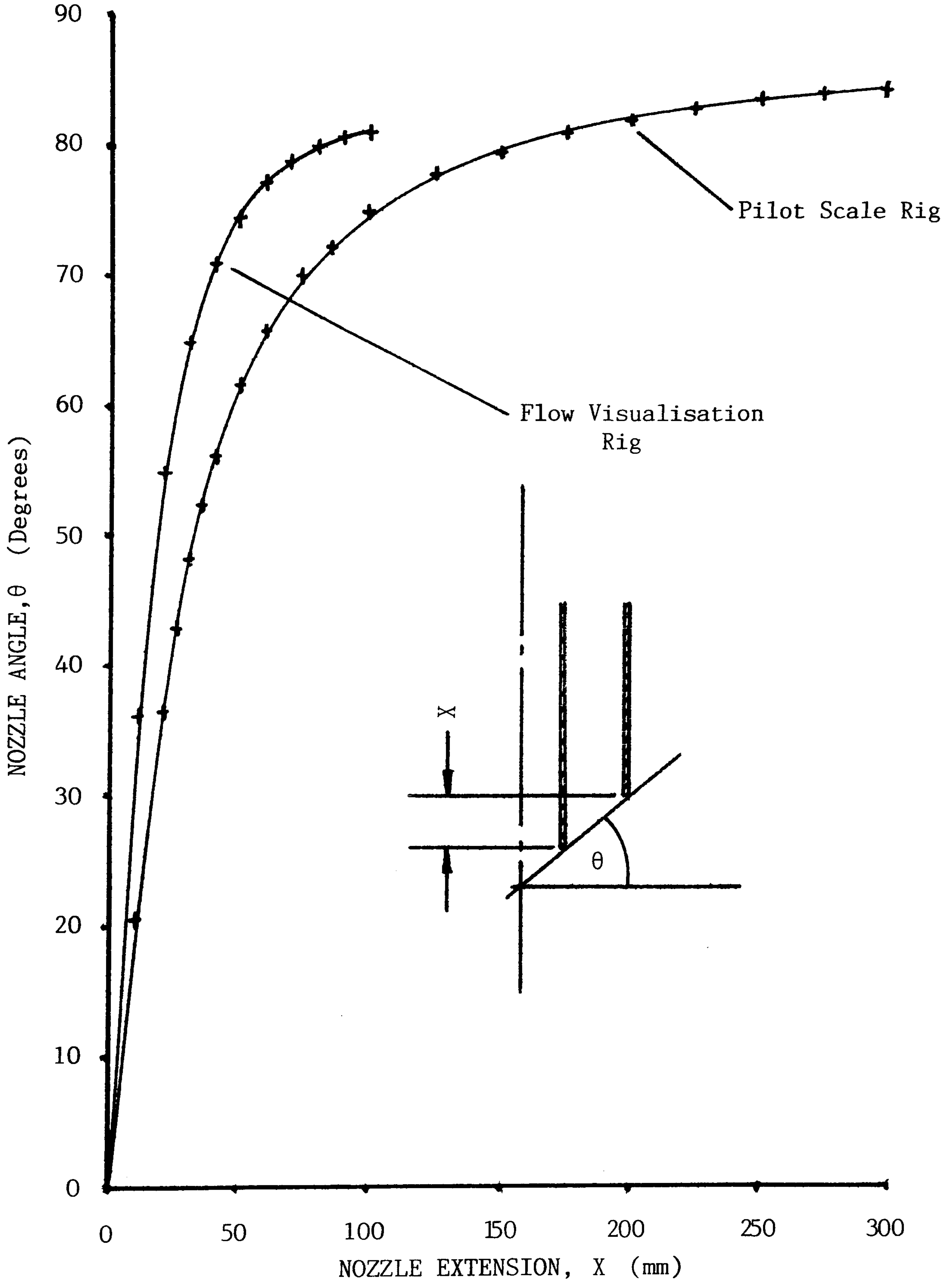
APPENDIX A

BRITISH STANDARD DEFINITION OF A SUCTION NOZZLE

Term No.	Term and definition	Typical example
32139	**Suction nozzle.** A device to introduce material into a suction line.	
	a. Camel-back nozzle. A unit with hand-controlled valve for regulating the intake of air to ensure correct air-material mixture.	 Fig. 43
	b. Sleeve nozzle. A unit with an adjustable outer sleeve for regulating the intake of air to ensure correct air-material mixture.	 Fig. 44

APPENDIX B

NOZZLE GEOMETRY RELATIONSHIP


90
80
70
60
50
40
30
20
10
0
NOZZLE ANGLE,θ (Degrees)
0
50
100
150
200
250
300
NOZZLE EXTENSION, X (mm)
Pilot Scale Rig
Flow Visualisation
Rig
X
θ

APPENDIX C

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