DEVELOPMENT OF PNEUMATIC STOWING EQUIPMENT AND A STUDY OF THE PRINCIPLES INVOLVED

A Thesis Submitted to

THE UNIVERSITY OF ASTON IN BIRMINGHAM

for

The Degree of Master of Science

by

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SEPTEMBER 1970

SYNOPSIS

In certain methods of coal mining it is necessary to extract rock in order to enlarge the excavations which provide access to the coal seam. This rock must either be transported out of the mine or be deposited into space provided by the extraction of the coal. It is desirable to mechanise the deposition of rock into this space. One means of doing this is to blow the rock through pipes by means of compressed air, feeding the rock and compressed air into the pipes through a unit known as a stowing machine. The operation is known as pneumatic stowing.

The aim of the work described in this thesis is to make an assessment of existing stowing machines used for the disposal of material underground, and to carry out a study of the principles associated with the development of pneumatic stowing equipment.

The faults and failures of existing stowing machines are examined and a Design Theory as a basis for a new development is given.

The manufacture of a test rig and a study of the results of trials carried out with it, is used to design a prototype stowing machine. The stages of development of the prototype machine with the subsequent trials results are given, and an assessment of the performance, efficiency and economic possibilities are made.

A full scale model manufactured in perspex has been used to obtain a clearer understanding of the behaviour of material passing from the stower into the conveying pipeline. The construction of the model is described. Model tests using different types of material are examined as a means of assessing the optimum air flow and rotor speed values for the stowing machine.

The author holds the position of Mechanical Project Engineer at the National Coal Board Mining Research and Development Establishment.

The N.C.B. MK.I Stowing Combine described in chapter 1 was already in existence, and the author's contribution to this development was to carry out design improvements, assess the results of underground trials, and formulate conclusions and recommendations for future development. The author's stower design theory is given in chapter 3, and chapter 4 describes the subsequent investigation and development of a prototype stowing machine.

The author's laboratory investigations using a stower model test rig, the observations made, and the resulting conclusions, are outlined in chapters 5 and 7.

The work carried out for this thesis also includes a search and examination of other investigators research to obtain any available information which could be relevant to the author's investigations. The scope and limitation of the search are outlined in chapter 2, with a more detailed reference to the work of Broadhurst[16] which is given in chapter 6. The work of Broadhurst has been referred to specifically because his investigations were concerned with stowing materials and conveying pipelines in stowing installations within the National Coal Board.

Throughout the thesis, British units are given, followed by the equivalent metric values rounded off to the nearest unit.

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CHAPTER 1

INTRODUCTION

1.1 BACKGROUND

The work described in this dissertation is a study of the principles involved in the design and development of pneumatic stowing equipment used for dirt disposal operations underground.

To clarify more the scope and limitations of this study it is necessary first to provide some background information relating to the need for dirt disposal operations.

There is a system of mining carried out by the National Coal Board known as "Longwall Mining". Basically this consists of mining coal from a long face perhaps 200 yards (183 m) and providing a roadway at each end (Maingate and Tailgate) for the transportation of men and materials; output of coal, and the supply of adequate air for ventilation. (Figure 1). The height of these roadways is determined by the space requirements for transportation of men and materials and ventilation purposes, and is usually greater than the coal seam thickness. This condition is usually unavoidable and necessitates the extraction of stone either above the coal seam (Ripping) or below the coal seam (Dinting) when cutting the roadway, which must keep up with the advancing coal face.

This stone, often referred to as dirt, can be transported out of the pit and dumped as tips on the surface, or it can be more usefully employed for stowing in pack form in the excavated areas by the side of the roadway behind and along the coal face (Figure 2). The disposal of pit dirt into pack areas provides roof support which controls roof movements due to the undermining of the strata.

1.2 PACKING METHODS

Several methods of packing dirt are adopted, for example, hand

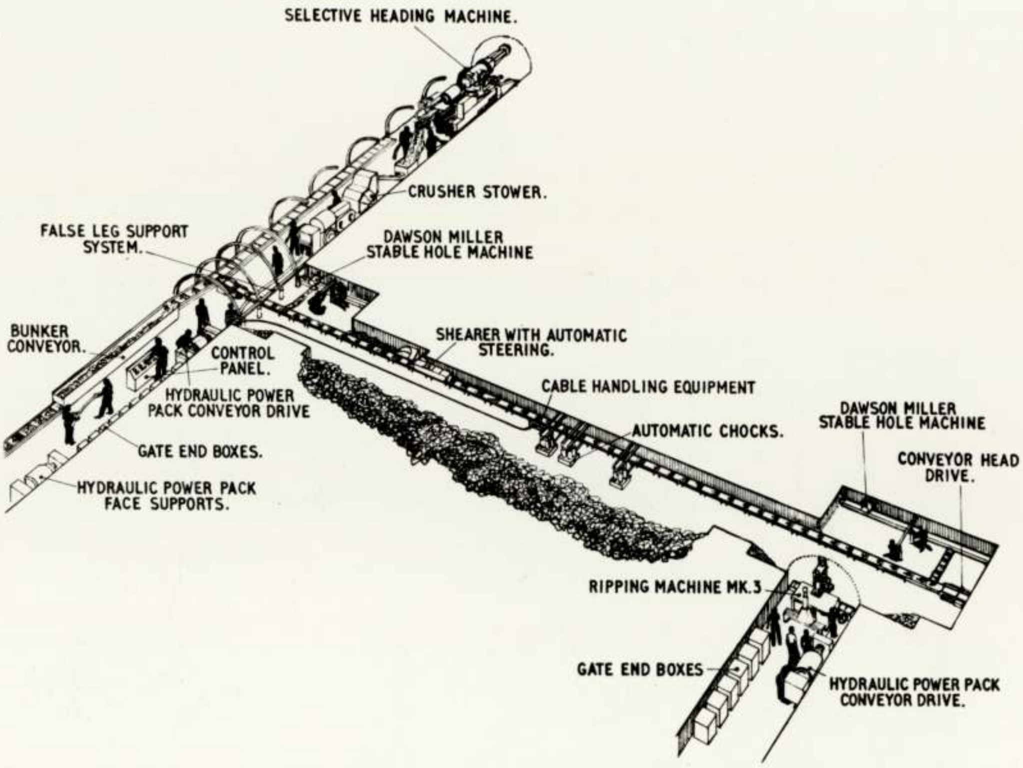

FIG. 1 COALFACE SHOWING MAINGATE AND TAILGATE ROADWAYS

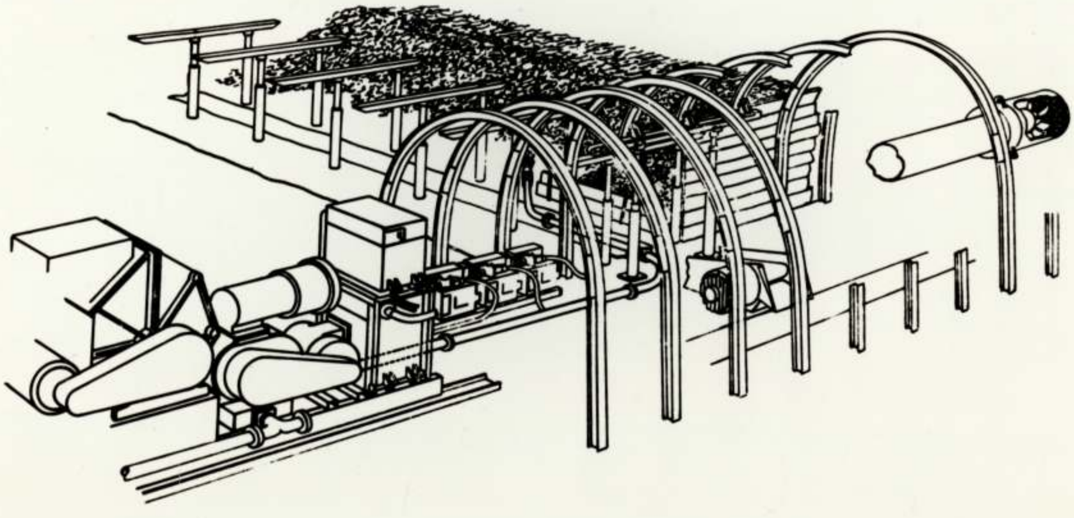

FIG. 2 STOWING OPERATIONS FROM ROADWAY INTO PACK AREA

packing, slushing which makes use of rope propelled buckets, ram packing which uses hydraulic rams, specially adapted conveyors, and other devices which have been developed for particular applications.

Comparisons of the various methods of Mechanical Packing are shown in Figure 3.

1.3 PNEUMATIC STOWING CONSIDERATIONS

Hydraulic conveying systems were also used for stowing operations, but problems of water drainage particularly when mining flat seams presented difficulties, and in the early 1930's nearly all hydraulic stowing systems were replaced by stowing equipment based on pneumatic conveying principles.

The development of stowing machines based on the pneumatic principle has been carried out both in this country and on the continent. The first machines available were the German Torkret, Beien, and Brieden units, and the English "Markham" Blastower.

The commercially available stowing machines, whilst giving reasonably good service, were still large robust pieces of equipment prone to blockages, and difficult to maintain particularly in the confined space available underground.

An assessment of the merits of pneumatic stowing as compared with other methods was carried out by the National Coal Board, and it was decided to instigate a Pneumatic stowing research and development programme taking into account the following advantages and disadvantages.

1.4 ADVANTAGES

1. An effective method of sealing the pack area reduces the danger of pack hole fires caused by spontaneous combustion. Stowing provides a better pack density than other methods.

2. The higher pack density given by pneumatic stowing improves the roof support and provides better control of roof movements brought about by undermining. This control of the strata results in reduced roof and floor

TYPE	SEAM THICKNESS			QUALITY OF PACK	LENGTH OF PACK	MANPOWER INCLUDING RIPPING	CAPITOL COST	MAINTENANCE COST	ADVANCE RATE	MINING CONSIDERATIONS	DEBRIS SIZE
	10X8 ROAD	12X9 ROAD	16X12 ROAD								
STOWING	2'-0" TO 3'-6" (·6 To 1·1m)	2'-0" TO 4'-0" (·6 To 1·2m)	3'-0" TO 6'-0" (·9 To 1·8m)	GOOD	50 YDS (45·7 m)	2 MEN CONT.	£10-12000	HIGH	HIGH	NOT EFFECTED	CRUSHER LIMITATIONS
SLUSHING (EXISTING)	2'-0" TO 3'-6" (·6 To 1·1m)	2'-0" TO 3'-6" (·6 To 1·1m)	3'-0" TO 4'-0" (·9 To 1·2m)	POOR	DEPENDENT ON TIME FACTOR	1 MAN CONT. 1 MAN INTER. 1 ADD MAN IF HAND RIB PACK	£1,500	LOW	LOW	CABLES STAGE LOADER & MK IV M/C ARE EMBARRAS^{MT}	BUCKET LIMITATIONS
SLUSHING (IMPROVED)	2'-6 TO 3'-6" (·76 To 1·1m)	2'-6" TO 4'-0" (·76 To 1·2m)	3'-0" TO 4'-0" (·9 To 1·2 m)	FAIR	DEPENDENT ON TIME FACTOR	2 MEN CONT	£4-5000	HIGH	MODERATE	DITTO	DITTO
CONVEYOR/ RAM PACKING	2'-6" TO 3'-6" (·76 To 1·9m)	2'-6" TO 4'-0" (·76 To 1·2m)	3'-0" TO 6'-0" (·9 To 1·8 m)	GOOD	NO LIMIT EXPECT COST	1 MAN CONT. 1 MAN INTER.	£5,6000	HIGH	VERY HIGH	DITTO	CONVEYOR LIMITATIONS
CHAIN CONVEYOR & PLOUGH	2'-6" TO 3'-0" (·76 To ·9m)	2'-6" TO 4'-0" (·76 To 1·2m)	3'-0" TO 6'-0" (·9 To 1·8m)	MODERATE	NO LIMIT EXCEPT COST	1 MAN CONT. 1 MAN INTER.	£4-5000 LESS RIPP^{G} M/C CONV^{YR}	MODERATE	VERY HIGH	SLIGHTLY REDUCES ACCESS HEIGHT AT RIP	DITTO
RAM PACKING		4'-0" TO 4'-6" (1·2 To 1·4m)	5'-0" TO 6'-6" (1·5 To 2m)	GOOD	5 YD (4·6 m)	2 MAN CONT.	£3000	LOW	MODERATE	UNDER DEVELOPMENT	UNDER DEVELOPMENT
BELT FLINGING	2'-6" TO 3'-6" (·76 To 1·1m)	2'-6" TO 4'-0" (·76 To 1·2m)	4'-0" TO 6'-0" (1·2 To 1·8m)	QUESTIONABLE	5 YD (4·6 m)	1 MAN CONT. 1 MAN INTER.	£2000	HIGH	HIGH	UNDER DEVELOPMENT	DITTO

FIG. 3 MECHANICAL PACKING - COMPARISONS (1964)

convergence in the roadways.

3. The improved roof and roadway conditions provides better ventilation.

4. Surface tipping is reduced or eliminated.

5. A saving in costs of repairing roadways is possible.

1.5 DISADVANTAGES

1. Clay type materials may become sticky causing pipe and stower blockages.

2. The air used for pneumatic stowing may interfere with the underground ventilation system.

3. Problems are created by the airbourne dust produced during pneumatic stowing operations.

Other points to take into account when considering the use of stowing equipment is its bulkiness, which hinders the movement of men and materials, its reliability of operation, and its capital cost. The conditions underground can vary greatly between different collieries and even between different faces in the same colliery. Consequently the system of dirt disposal has to be considered in the light of circumstances appertaining to any particular mining installation. It is often the case that the rate of coal output is restricted by the limitations of the roadway forming activities, an important part of which is pack building, and it is therefore essential that the dirt disposal system adopted is reliable and will meet the face advance rate requirements determined and controlled by the production rate from the coal face.

1.6 EARLY INVESTIGATIONS

In 1960 the National Coal Board made investigations into the various methods of disposing the face rippings, outlining the various means employed. The objective was to speed up roadway forming operations to match the increased face advance rates due to concentration of output and coal face mechanisation. Consideration was given to the severity of the problems associated with roadway forming activities which included, ripping or dinting, various methods of packing, and the setting of roadway supports, whilst avoiding obstruction of coal transport from the face. It was also considered that the application of continuous mining to the formation of face gate roads might well require a different approach to the problem of the disposal of the dirt made during road drivage. This meant that pneumatic stowing equipment should be designed to operate in conjunction with either a continuous ripping machine or road heading machine.

Existing stowing equipment at this time used 6 in (152 mm) diameter pipes which handled material having a maximum dimension of 3 in (76 mm). On 6 in (152 mm) stowing equipment a quantity of 2,250 ft^3/min (1.1^3 m/s) of free air provided reasonably satisfactory stowing over short distances, up to say 60 yards (55 m). It was therefore considered that with a quantity of about 850 ft^3/min (0.4 m^3/s) the correct stowing conditions in a 3.5 in (89 mm) diameter pipe could be maintained.

1.7 CONCLUSIONS

From these initial investigations it was thought that the opportunity to operate efficient stowing with machinery of low pressure, up to 20 lb/in^2 (138 x 10^3 x N/m^2) was attractive, and could greatly reduce the cost of operation as well as capital cost. Furthermore, the equipment being of low capacity would be much smaller in dimensions than existing units and thus most suitable for installation in the restriction of face roadheads or advance headings (roadways driven in advance of the coal face).

1.8 FIRST TRIALS

After a careful study of all the information available at that time, work was started to establish the material size to stow successfully through 3 in (76 mm) or 4 in (102 mm) diameter pipes, and the specific air consumption required. Field trials were carried out using 3 in (76 mm) and 4 in (102 mm) diameter pipes.

Results of these first trials indicated that with 4 in (102 mm) diameter pipes material up to 2 in (51 mm) size could be conveyed along pipes up to 60 yards (55 m) in length including a 90° bend, and the air consumption would be approximately 1,000 ft /min (0.5 m /s) to 1,200 ft /min (0.6 m /s) of free air. It was also established that oversize material must be avoided to prevent blockages.

During these initial investigations a study was made of the additional available equipment which may be suitable for incorporation in a combined crushing and stowing machine, which not only featured the stowing unit but also included the crusher necessary to provide the sized material, a blower for supplying the air, and a conveyor to feed dirt into the crusher.

A final assessment of the investigations and field trials which had been carried out resulted in a specification being prepared for a stowing combine having the following requirements.

1. Stowing pipe size 4 in (102 mm) diameter.
2. Blower capacity 1,000 ft /min (0.5 m /s) free air delivery to operate up to 15 lb/in (103 x 10 N/m).
3. Swing Hammer Crusher - to provide graded stowing material up to 2 in (51 mm) in size.
4. Feed Conveyor to the crusher.
5. Average throughput of 20 ton (20.3 tonnes) to 25 tons (25.4 tonnes) per hour.
6. Mini stower unit - as compact as possible for the stipulated output.

1.9 FIRST PROTOTYPE STOWING COMBINE

A prototype of this machine was built by the Central Engineering Establishment, and during 1962 it was subject to surface trials, followed by underground operations at Glapwell Colliery where some measure of success was achieved. Six more of these machines known as the Bretby MK.I Stowing Combine were built and installed underground at three other collieries. Problems of pipe and stower blockages, difficulties in maintenance, the failure to meet the increasing rates of advance, and the lack of machine reliability, resulted in the gradual withdrawal from service of all the seven MK.I Stowing Combines.

1.10 ASSESSMENT AND RECOMMENDATIONS

The MK.I stowing combines did not meet the dirt disposal requirements at the time they were operated underground. Meanwhile, these requirements have become more severe because faces are advancing even faster. This poses the question as to whether pneumatic stowing equipment can be developed to meet the technical requirements at an economic cost.

Experience with the Bretby MK.I Stowing equipment clearly indicated that its unreliability, failure to meet the required outputs resulting from increased coal outputs, and the time and costs involved in maintenance, ruled out the use of such a machine when planning future mining systems. The necessity to develop a new stowing unit to meet existing and future demands was considered, and in 1964 a research and development programme was started.

CHAPTER 2

SURVEY OF EARLIER WORK

2.1 PNEUMATIC STOWING EQUIPMENT

Pneumatic Stowing Equipment can be divided into three basic components which are :

1. The stowing machine which receives material and meters it into the conveying pipeline.
2. The compressor or blower which supplies the conveying air to the stowing machine.
3. The pipeline which conveys the material to the place of deposit.

The work covered by this thesis is concerned primarily with the development of a stowing machine and the efficiency of metering the material into the conveying pipeline. It is however recognised that the requirements for conveying the material along the pipeline affect the conditions at the stower unit, and it is therefore necessary to give some consideration to the conveying conditions in the pipeline as well.

2.2 STOWING MACHINES

The information available on stowing machine development was found to be limited to the data which could be obtained from the manufacturers of stowing machines both in this country and on the continent. Whilst these machines have been used with reasonable success, their physical sizes are outside the size limitation envisaged for future mining systems where thinner seam mining is inevitable. In all the available stowing machines the air passage through the stower units involves the negotiation of bends, a feature which is undesirable if blockages are to be eliminated and the size of machine is to be limited. There was accordingly no available information on stower development which was relevant to the investigations to be carried out.

2.3 PIPELINE CONVEYING

Over the last 50 years many investigators have carried out research on the subject of pneumatic conveying of solids in a pipeline. It was not possible to cover every source of information therefore the time spent on this search was concentrated on three bibliographical sources, which were :

1. Applied Science and Technology Index (Editions 1958 to 1968 inclusive)
2. Engineering Index (Editions 1955 to 1967 inclusive)
3. These accepted for Higher Degrees (In the Universities of Great Britain and Northern Ireland) (Volumes 1950 to 1965 inclusive)

Additionally the reference libraries at the Mining Research and Development Establishment and the University of Aston in Birmingham have been kept under review.

The search for relevant information on the subject of pneumatic conveying was not limited to the mining industry but was extended to cover a wide field of the manufacturing and chemical industries, where the handling of a considerable number of different types of materials is carried out. The sources investigated covered such industrial activities as the transportation of plastics, powders, grain and foodstuffs, as well as other research carried out over the last 25 years.

Initially a considerable number of references on pneumatic conveying were obtained from various sources, and of these, 43 which were thought likely to contain relevant information, were selected for further investigation. These are listed after Appendix 3.

A study of these selected papers provided some useful background information but none of the theories and formulae proposed were sufficiently general to be of direct use in pneumatic stowing investigations. It was found that, as would be expected, the investigations carried out by others had attempted to establish data applicable to specific materials under specific conveying conditions. The formulae usually involve factors which have to be established experimentally, and direct extrapolation for the conditions under which pneumatic stowing operates is not possible.

To summarise the conclusions to be drawn from the search, it is felt that the direct application of the different theories and formulae to pneumatic stowing is questionable. To substantiate this opinion six papers are briefly outlined; they give a general indication of the field of investigation covered by all the references obtained.

2.3.1. BURK AND PLUMMER$^{(1)}$ - Suspension of Macroscopic Particles in a Turbulent Gas Stream.

BURK AND PLUMMER have proposed the following equation for the suspension of macroscopic particles in a fluid stream

$$v = K \left(\frac{s-\sigma}{\sigma}\, d\right)^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where v = the velocity of the fluid necessary to cause suspension - cm/s

s = specific gravity of material suspended

σ = specific gravity of fluid

d = average diameter of the particles - cm

K = a constant which depends on the shape of the particles. For spherical and cubical particles K varies from 47 to 51.

Equation (1) is simplified if the fluid is air, since then σ is usually negligible compared with s. In this case the formulae becomes :

$$v = K^1\ (ds)^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

where $K^1 = K/\sigma^{\frac{1}{2}}$

Both equations (1) and (2) apply only to the suspension of particles in a fluid stream. For transportation of the particles, velocities in excess of that obtained from the formula are required.

2.3.2. J. M. DALLAVALLE(3) - Determining Minimum Air Velocities for Exhaust Systems.

An attempt is made to determine the velocities required to move particles in a fluid stream. A systematic investigation was carried out using materials of known size and density. The materials consisted of screened cinders, crushed carbon, anthracite, and quartz, having specific gravities within a range of 1.08 to 2.65 respectively. The sizes of material varied between .055 ins (1.4 mm) to .320 ins (8.1 mm).

The results of DALLAVALLE'S work showed that the minimum velocities for suspension of the materials used in his work varied between 16 and 46 ft/s (5 and 14 m/s) the velocities increasing with an increase in size, and specific gravity.

DALLAVALLE proposed the following equation which was found to fit his established data.

For Horizontal Pipes:-

$$v = 6000 \; {}^{s}/(s + 1) \; d^{0.398} \qquad (1)$$

where v = min velocity for suspension - ft/min

s = specific gravity of solids

d = average diameter of particle - ins

In his general discussions DALLAVALLE states that velocities considerably in excess of those that can be obtained from equation (1) were necessary to prevent material build-up at the bottom of the pipes.

2.3.3. HAKON WADELL(4,5,6) - "Volume, Shape and Roundness of Rock Particles", "Sphericity and Roundness of Rock Particles", and "The Coefficient of Resistance as a Function of Reynolds Number for Solids of Various Shapes".

WADELL suggests a formula for expressing the shape of a plane figure in terms of the degree of circularity. The formula is based on the isoperimetric* property of a circle, and reads :

$$\cancel{c} = c/C \quad \ldots\ldots (7)$$

where $\cancel{c}$ = degree of circularity

c = circumference of the circle having the same area as the plane figure - in

C = the actual perimeter of the plane figure - in

The maximum value obtained from the formula is unity which is the numerical value for the degree of circularity of a circle. WADELL also expressed the shape of irregular rock particles by the numerical value of the degree of true sphericity. This is based upon the isoperimetric* property of a sphere and is obtained by the formula :

$$\Psi = As/S$$

where Ψ = degree of true sphericity

As = surface area of a sphere having the same volume as the particle - in^2

S = actual surface area of the particle - in^2

In his expression the maximum value obtained is unity, this being the numerical expression for the degree of true sphericity of a sphere.

In a later paper WADELL attempts to show the usefulness of using the degree of sphericity factor as a means of expressing the shape

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* Of figures having equal perimeters

of rock particles. The resistance of submerged particles is used as a basis for determining the coefficient of resistance as a function of Reynolds number for different degree of sphericity values. This examination considers cube and spherical shaped particles only. Tables are presented illustrating the influence of the degree of sphericity on the settling velocity, and the coefficient of resistance as a function of the sphericity value.

2.3.4. R. F. DAVIES(7) - The conveyance of solid particles by Fluid Suspension.

DAVIES attempts to establish a general principle for the conveyance of solid particles in both fluids and gases. Both spherical and flaky particles are considered, and a general formula is proposed for the velocity of a fluid at a point where particles just commence to rise and disperse in the moving stream.

This velocity which is referred to as the initial mixing velocity, is given as :

$$Vo = \sqrt{\frac{2g}{W} \; \frac{V}{A} \; (Wm - Wf)}$$

where Vo = initial mixing velocity - ft/s

Wf = Mass density of the fluid - lb/ft^3

Wm = solid Mass density of the material composing the particles - lb/ft^3

V = Volume of the particle - in^3

A = horizontal cross sectional area of the particle - in^2

In his general observations DAVIES points out that a flaky or lenticular particle offers a large surface per unit volume of material, whereas a solid spherical particle offers the minimum surface per unit volume of material.

Irregularly shaped particles fall between these two extremes.

2.3.5. SHINZO KIKKAWA(8) - "Research on the Pneumatic Conveyance of Densely Concentrated Solid Particles in a Horizontal Pipe".

In his article KIKKAWA discusses the theory that the pressure loss suffered by conveying a unit weight of particles decreases gradually with an increase of the mixture ratio, after which a steep rise occurs in the pressure loss just before clogging takes place. The use of the gravimetric mixture ratio (Weight of conveyed particles per unit time/weight of air flowing per unit time) and volumetric mixture ratio (Volume of conveyed particles per unit time/volume of air flowing per unit time) as a value for defining mixture ratio is discussed.

KIKKAWA suggests that neither of the two ratios mentioned is suitable for discussing the behaviour of concentrated particles in a horizontal pipe, and defines a new concentration of particles which takes into account the vertical distribution of particles in a horizontal pipe as well as the relative velocity between the air stream and the solid particles. The newly defined concentration of particles (m) is given as :

$$m = \mu \, (W_a \; U_a / W_s \; V_m)$$

where W_a = specific weight of air - lb/ft^3

U_a = Mean velocity of air stream - ft/s

W_s = Specific weight of particle - lb/ft^3

V_m = Mean velocity of a particle in the transport pipe -ft/s

μ = Mean gravimetric mixture ratio

From the results of experiments using soyabeans and tenite* particles, KIKKAWA derives an empirical formula relating the "concentration of particles" (m) to the "rate of decrease of acceleration factor" ($^K/Ko$). The formula states :

$$^K/Ko = \tfrac{1}{2} \left[\exp\,(-424m) + {}^{0.00261}/m + 0.00261 \right]$$

where K = Acceleration factor of concentrated particle

Ko = Acceleration factor of a single particle

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* Thermoplastic consisting of cellulose acetate butyrate

m = Concentration of particles factor

2.3.6. P. H. BROADHURST(9) - "The Pneumatic Transport of Materials for Stowage of Underground Workings".

BROADHURST investigated the general principles associated with the pneumatic conveying of large size solids by pipeline. The relevant information from these investigations falls into three sections, namely :

(1) Stowing Materials

(2) Convyeing material round bends

(3) Material movement in the pipeline.

The work of BROADHURST is referred to in greater detail in Chapter 6 because his investigations whilst limited to material movement in the pipeline and around bends also considered the suitability of material for stowing operations. The information contained in his work must be taken into account when considering stowing equipment as a whole, and his paper therefore offers some contribution to the investigations involving the actual stowing machine development.

2.4 ASSESSMENT OF OTHER INVESTIGATIONS

There have been many attempts to establish formulae for calculating the fluid velocities required to obtain a suspension of particles in a gas stream. The theories and formulae submitted by the many workers are based on investigations which consider specific shapes such as spheres, cubes and prisms. The size of particles considered have in the main, been that associated with sand grains, wheat, and spherical balls up to a maximum size of $\frac{1}{2}$ in (12 mm) diameter.

Floating velocities can be estimated with reasonable accuracy within the limitations of particle size considered.

There seems to be insufficient evidence to show that any of the formulae presented could be directly used to provide with reasonable accuracy a means of assessing the air velocities required for stowing operations. This conclusion is based on the following arguments which are now discussed.

2.4.1. MATERIAL SIZE AND SHAPE

Two factors which influence the conditions for suspension in a gas stream is the size and the shape of the particle. The formulae available vary according to the principles on which they are based, but in all cases a factor related to the particle size and shape is included. Some attempt has been made to identify this factor by relating particle size to spheres having the same volume as the particle. In other cases attempts have been made to obtain the particle size factor by considering the surface areas, perimeters, and degree of circularity of irregular shaped particles. For the conveyance of particles which are basically of the same shape and size, it is acceptable that the degree of circularity or the degree of sphericity can be usefully employed. In the case of stowing materials, where for any particular volume of material being conveyed, particles have an infinite variety of shapes and can be any size from macroscopic to about 2.5 in (63 mm), it is diccicult to evaluate a single particle size factor.

2.4.2. MATERIAL DISTRIBUTION

In stowing operations the percentage of fines [less than $\frac{1}{8}$ in (3 mm)] to rock particles [up to 2.5 in (63 mm)] is dependant entirely on the method of producing the debris ("Machine Cut" or "Blown"). In any unit volume of material passing through the stowing system, the concentration of particles can vary enormously according to the percentage of fines in the stowing material. This condition which exists in pneumatic stowing means that the conveying state of the material is continually chainging during stowing operations.

In the work by KIKKAWA referred to in section 2.3.5. the vertical distribution of particles in a horizontal pipe is considered, and a concentration of particles factor is derived. The usefulness of

KIKKAWA'S formula for pneumatic stowing is again doubtful because the mean velocity of the particles must be known, and in the case of pneumatic stowing, a value for this velocity which covers all the different sizes and shapes would be extremely difficult to assess. The work of KIKKAWA also takes into consideration a particle radius factor which again involves the shape and size of particles to be considered.

2.5 CONCLUSIONS

The many formulae available for determining the flow conditions for transportation of material in horizontal pipes cannot be directly applied to pneumatic stowing for the reasons given previously. It is suggested that there is no single formula available for predicting carrying or floating velocities for pneumatic stowing operations where the solids/gas ratios can vary considerably within any one unit of time. In the case of pneumatic stowing, data must be obtained from experimental investigations if the conditions for efficient stowing operations are to be predicted.

CHAPTER 3

STOWER DESIGN THEORY

3.1 REQUIREMENTS

The first phase of the development was based on the following requirements.

1. Size of material to be handled - up to 2 in (51 mm)
2. Stowing rate 40 tons (40.6 tonnes)/h based on a material density of 90 lb/ft^3 (1442 kg/m^3)
3. Air supply 1000 ft^3/min (0.5 m^3/s) free air delivery
4. Operating pressures up to 15 lb/in (103×10^3 N/m^2)
5. Maximum dimensions of machine to be : height 20 in (508 mm) width 27 in (686 mm) length 72 in (1829 mm)

In addition to the above, the machine was required to be of the paddle wheel type rotating on a horizontal axis and operating in conjunction with a stowing range of 4 in (102 mm) diameter pipes. The selection of a paddle wheel with horizontal shaft was made on the basis that existing developments both inside and outside the National Coal Board had used this principle, and it was considered that there was still considerable scope and potential in this type of machine.

3.2 DESIGN PARAMETERS

The design parameters for a stower test rig were selected after examining the features of existing equipment, and assessing the reasons for the faults and failures already experienced. This design study was divided into the following sections :

1. The air system through the machine.
2. The shape and size of the stower rotor, which includes the number of pockets and their shape.
3. The design of the barrel.

4. The overall arrangement of the machine, bearing in mind the necessity for easy maintenance to be carried out underground, and for the eventual production machines to be produced at an economic cost.

All these parameters were studied within the limitations set by the permissible size of machine required.

3.3 PNEUMATIC PRINCIPLES

One of the faults with previous stower units was the frequent stoppages which occurred due to material blockages within the stower itself. Operational experience had shown that machine utilisation was generally dependant on the type and wetness of the material being handled. The air pressure losses through the system were a fairly high proportion of the total available pressure, resulting in insufficient energy to prevent sticky materials and oversize particles from clogging the stower and conveying pipeline. On closer examination of the Mk.I stower unit, one of the bad features of design was the manner in which the air was directed through a path which was formed by a sweeping arc between bends at the inlet and outlet sides of the stower. The shape of the air passage and design of the rotor prevented a smooth uninterrupted flow through the stower (Figure 4).

As far as the air flow was concerned it was decided that the new development should be based on a straight horizontal air path through the stower, and the rotor pocket was to be designed to pass through the conveying air allowing the material to be carried horizontally through the pocket (Figure 5).

3.4 ROTOR PADDLE

The design of the rotor paddle is dependant on a number of factors which have to be considered together. The size and speed of the rotor, together with the volume and shape of the pockets, have to be matched

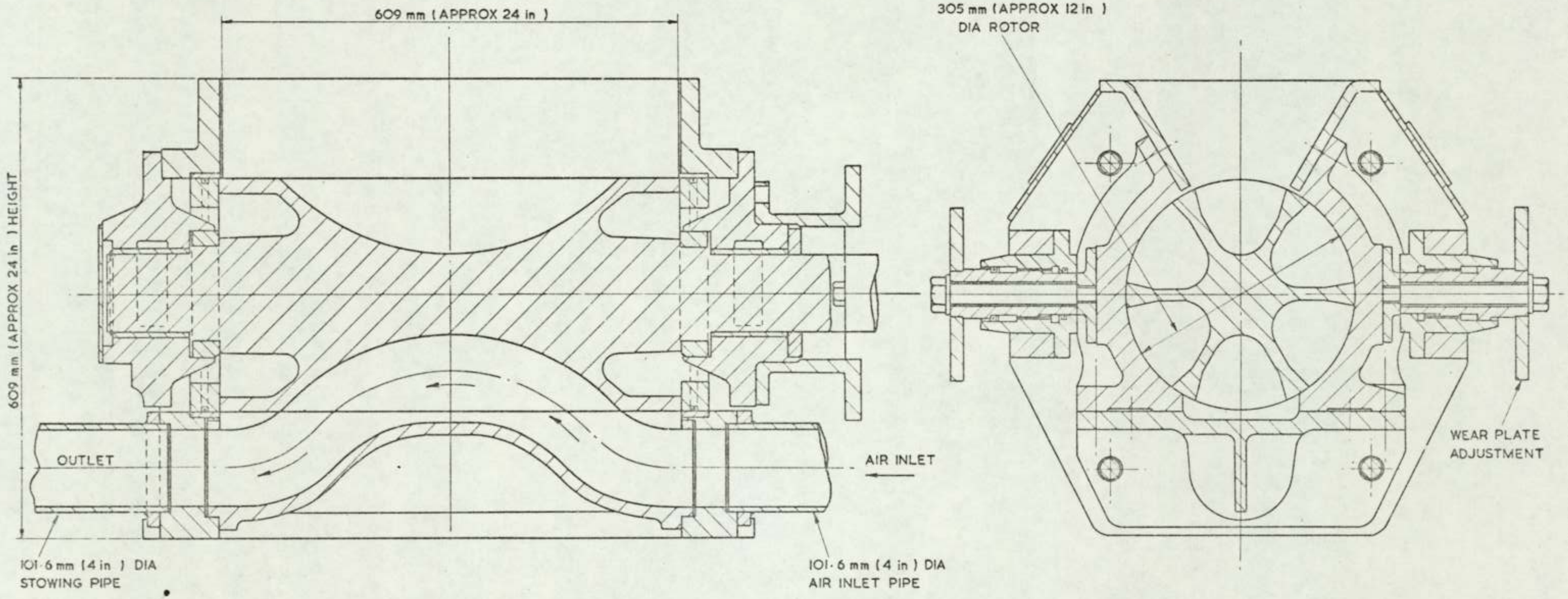


FIG. 4 MK.1 STOWER UNIT

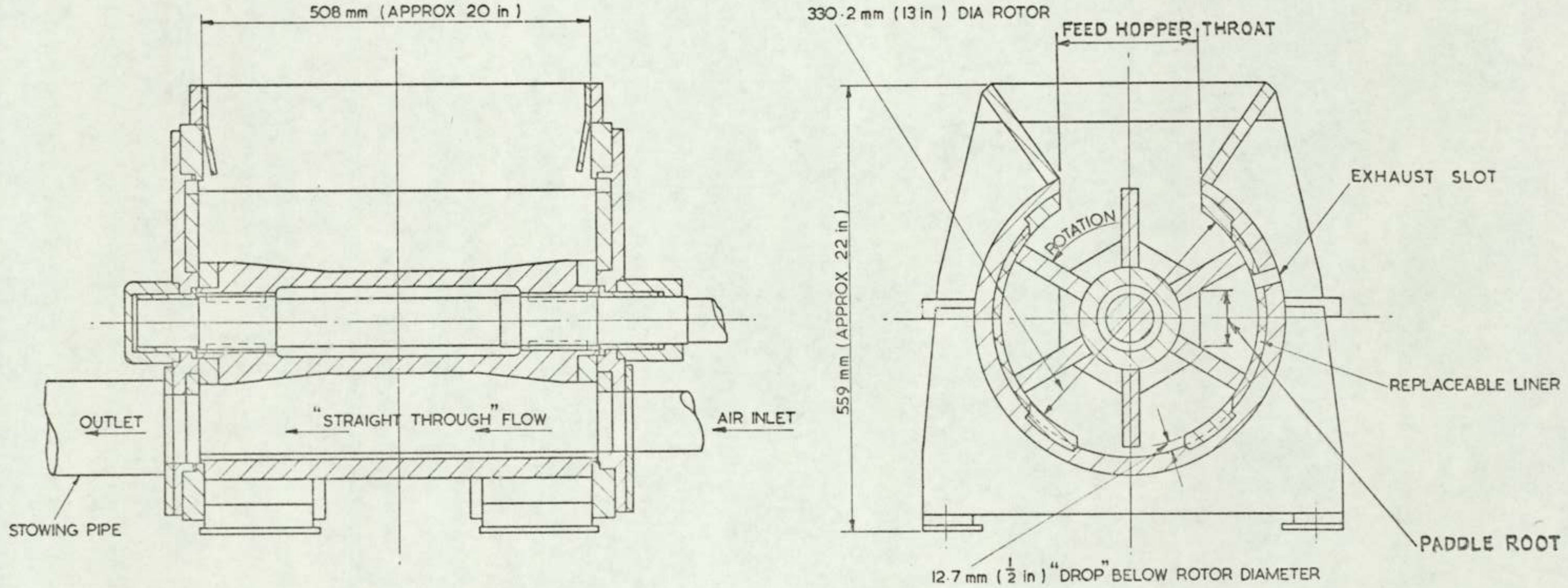


FIG. 5 STOWER UNIT SHOWING STRAIGHT AIR PASSAGE

to the required output and the restricted size of the machine. To form a basis from which the design and subsequent development could proceed it was necessary to establish the relationship between the tip diameter, hub diameter, number of paddles, and the potential output in terms of volume per revolution of rotor. Restricting the height of the machine to approximately 20 in (508 mm) meant that the rotor could not be more than about 14 in (356 mm) diameter. It was also necessary to ensure that whatever the shape and size of the rotor pockets, the width at the throat (Figure 5) of the pockets should be something greater than the maximum piece of material to be handled, this being 2 in (51 mm). With this in mind the theoretical output was calculated over a range of rotor sizes :

Let D = Rotor Tip diameter - in

d = Rotor Hub diameter - in

l = length of rotor - in

W = Width of Rotor Paddle - in

N = Number of Rotor Paddles

n = Rotor speed - rev/min

p = Mass Density of Material - lb/ft^3

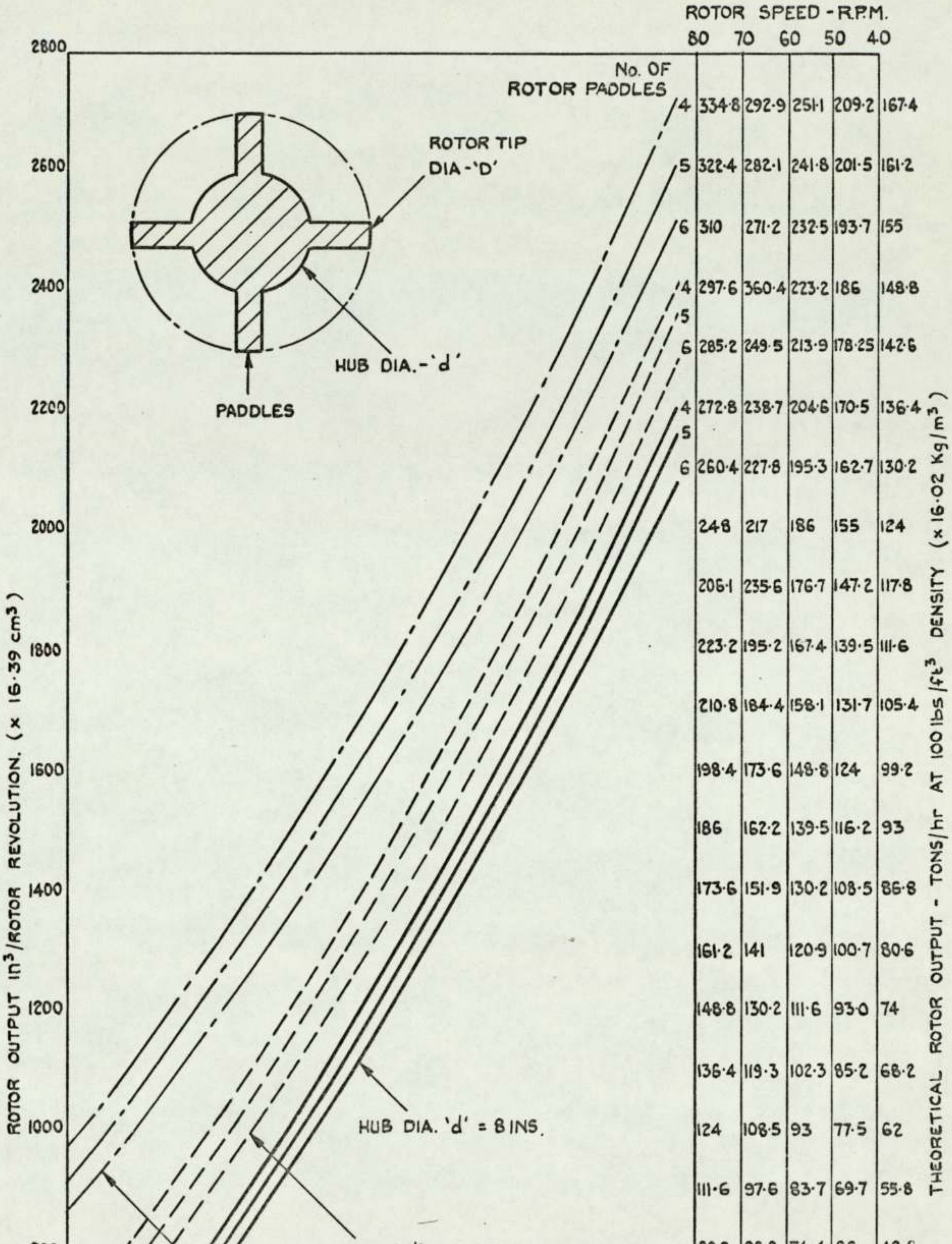
then swept volume of rotor pockets per revolution is given by

$$l\left[\frac{\pi}{4}(D^2 - d^2) - WN\frac{(D - d)}{2}\right] \text{ cu. ft/rev. } \ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

and rotor output

$$= \frac{60\ pnl}{2240}\left[\frac{\pi}{4}(D^2 - d^2) - WN\frac{(D - d)}{2}\right] \text{ Tons/hr. } \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Figure 6 shows the theoretical outputs plotted for a range of rotor sizes and numbers of paddles, the outputs being based on a density of 100 lb/ft^3 (1601 kg/m^3) and for a rotor speed range between 40 and 80 rpm. The thickness of the paddles was fixed at 1 in (25 mm), this being the size necessary to give the rigidity and strength for the severe operating conditions envisaged.



80	70	60	50	40
334·8	292·9	251·1	209·2	167·4
322·4	282·1	241·8	201·5	161·2
310	271·2	232·5	193·7	155
297·6	360·4	223·2	186	148·8
285·2	249·5	213·9	178·25	142·6
272·8	238·7	204·6	170·5	136·4
260·4	227·8	195·3	162·7	130·2
248	217	186	155	124
206·1	235·6	176·7	147·2	117·8
223·2	195·2	167·4	139·5	111·6
210·8	184·4	158·1	131·7	105·4
198·4	173·6	148·8	124	99·2
186	162·2	139·5	116·2	93
173·6	151·9	130·2	108·5	86·8
161·2	141	120·9	100·7	80·6
148·8	130·2	111·6	93·0	74
136·4	119·3	102·3	85·2	68·2
124	108·5	93	77·5	62
111·6	97·6	83·7	69·7	55·8
99·2	86·8	74·4	62	49·6
86·8	75·9	65·1	54·2	43·4
74·4	65·1	55·8	46·5	37·2
62	54·2	46·5	38·7	31

FIG 6 STOWER ROTOR PARAMETERS

Thus for example for a 5 Paddle rotor having a tip diameter of 13 in (330 mm) and a hub diameter of 7 in (178 mm),

From the chart (Figure 6)

Output for a rotor speed of 40 rpm = 97 tons/h. (99 tonnes/h)

Output for a rotor speed of 70 rpm = 171 tons/h. (174 tonnes/h)

These outputs are based on a material mass density of 100 lbs/ft^3 (1601 kg/m^3). For other material densities the output values are directly proportional.

In using the chart to select the rotor parameters, it was necessary to base the required output of approximately 50 tons/h (50.8 tonnes/h) on the minimum material mass density that was considered likely to be handled. This was estimated to be 60 lb/ft^3 (961 kg/m^3) which reduced the output figures on the chart by a factor of 0.6

Another point to be considered was that the actual volume of material metered by the rotor would be something less than the calculated volume. This reduction was dependant on the fines/solids factor, and also the percentage fill of the rotor pockets. It was not possible to evaluate these unknown factors which are entirely dependant on the type of strata being mined, and the sizes of the material to be handled. The sizes are to some extent determined by the means used for winning the material, for example, machine cut with either a ripping, roadheading, or tunnelling machine, or blown by explosives with or without subsequent crushing.

The development of a machine which would meet the required output of about 50 tons/h (50.8 tonnes/h) under the worst possible conditions, meant that the design had to take into account a possible overall efficiency as low as 50% as far as the actual metered volume of material was concerned.

Another limitation on the rotor design was the requirement that the depth of pocket should be not less than 3 in (76 mm), preferably

4 in (102 mm) or 5 in (127 mm), bearing in mind that the maximum material size was set at 2 in (51 mm) with occasional oversize pieces being inevitable.

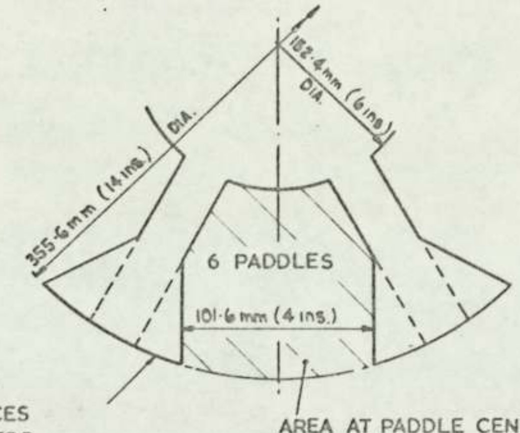
To keep the pocket throat dimension at about 4 in (102 mm) which is twice the material size, the hub diameter could not be less than 6 in (152 mm).

One of the important operating features is the manner in which the material is fed to the stowing equipment. Due to the method of mining the stone, the material is quite often fed in sudden loads at irregular intervals, which requires the stower unit to operate at very high peak outputs for short times, with probably only small amounts between. Because of this, it was considered that the 50 tons/h (50.8 tonnes/h) requirement was defined as the average output. This meant that the stower had to be capable of handling material at much higher rates than 50 tons/h (50.8 tonnes/h), and in fact, past experience has shown that peaks as high as 150 tons/h (152.5 tonnes/h) may well be necessary in order to guarantee a consistent average of 50 tons/h (50.8 tonnes/h).

All the features metioned were finally considered together, taking into account their effects on one another. The selection of rotor size was narrowed to a tip diameter between 13 in (330 mm) and 14 in (356 mm) with a hub size between 6 in (152 mm) and 7 in (178 mm).

3.5 ROTOR POCKETS

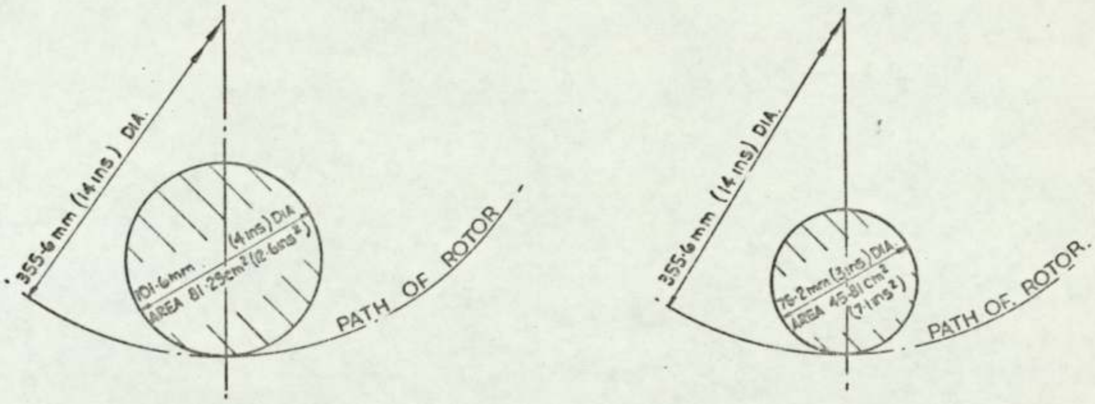
Figure 7 shows the initial design of rotor selected for the stower test rig trials. The rotor had 6 pockets, a hub diameter of 6 in (152 mm), and an outisde diameter of 14 in (356 mm). From the chart (Figure 6) the theoretical output based on a density of 100 lb/ft^3 (1601 kg/m^3) and a rotor speed of 40 R.P.M. is approximately 120 tons/h (122 tonnes/h). The mass density of the material to be handled during the trials varied between 75 and 100 lb/ft^3 (1202 and 1601 kg/m^3), and the actual rotor speed was to be determined by test rig trials.



TAPERED GUIDE PIECES
AT EACH END OF ROTOR
(APPROX. 127mm (5ins) LONG)

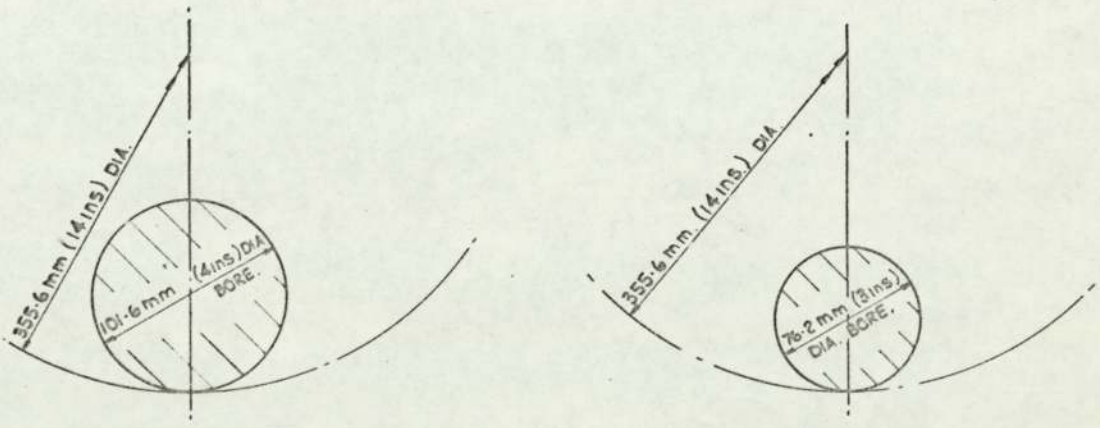
AREA AT PADDLE CENTRE 101·94cm² (15·8ins²) SQ.INS.
AREA AT PADDLE ENDS 69·03cm² (10·7ins²) SQ.INS.

(a) ROTOR.



(b) END COVER (OUTLET.)

(c) END COVER (INLET.)



(d) OUTLET PIPE.

(e) INLET PIPE.

FIG. 7 STOWER DESIGN FEATURES.

3.6 STOWER INLET AND OUTLET PIPES

A study of the manner in which the air is introduced to the stower unit, and the way the material is collected and conveyed out of the unit into the stowing pipe range, was a fundamental part of the development.

Past experience had shown that breakdowns in stowing operations were mainly due to blockages within the stower unit and the first 6 ft (1.8 m) of piping. This particular problem had to be overcome for successful stowing, therefore the design of both the inlet and outlet pipes to the stower was considered critical. There was no available information which could be used to determine the best method of utilising the available air, and it was considered that the point of introduction of the air flow relative to the rotor pockets, and the aperture at the exit side of the stower, would greatly affect the efficiency of metering material into the stowing pipe range.

3.7 INLET PIPE

As far as the stower inlet was concerned, there was no apparent reason why the aperture into the stower, and the shape of the inlet pipe, should be anything other than circular, and because the physical dimensions of the stower drive restricted the available space the air inlet pipe and the aperture through the stower end plate was fixed at 3 in (76 mm) diameter. The pipe was positioned relative to the rotor paddles such that the air supply was directed within the area between adjacent paddles, making direct contact with the material held there. This allowed the rotor paddles to pass through the air passage and to some extent promoted some form of scouring of the rotor pocket (Figure 7).

3.8 OUTLET PIPE

To maintain the straight through air flow, and to avoid as much as possible any restriction or condition which promoted blockages, or increased resistance to flow, it was logical to make the outlet pipe

4 in (102 mm) diameter, which matched the selected size of stowing pipes. The aperture through the outlet end cover plate was also made 4 in (102 mm) diameter and the piping on the outlet side was positioned to keep a straight-through flow. (Figure 7).

3.9 AIR POCKET EXHAUST

One of the features of this type of stowing machine which has an effect on the efficiency of the system, is the problem of exhausting pressurised air from the rotor pocket after the pocket has delivered its material to the air stream and is on its return cycle from the bottom of the stower. The material feed into the rotor pockets relies entirely on gravity, and any opposition to this gravity feed can create a bridging of the material across the throat of the stower feed hopper. Because of this it is important to avoid as far as possible escaping air flowing up into the stower hopper. To achieve this,air exhaust holes have to be provided to allow the pressurised pocket to vent to atmosphere during its rotation from the bottom of the stower to the hopper feed throat. At the same time, in analysing the mechanics of the metering system it is evident that the positioning of the exhaust orifice, whilst satisfying the venting requirements, must not form a bypass for the air flowing through the stower. In other words, the empty pocket on its rotation from the bottom of the stower must be sealed off within the stower barrel before the pocket venting action takes place.

Figure 8 shows the position of the exhaust port relative to the rotor pocket. Figures 9 and 10 gives the available exhaust time.

Consider the rotor pocket after material discharge. (Figure 8)

Let P = Pressure inside rotor pocket at time t.

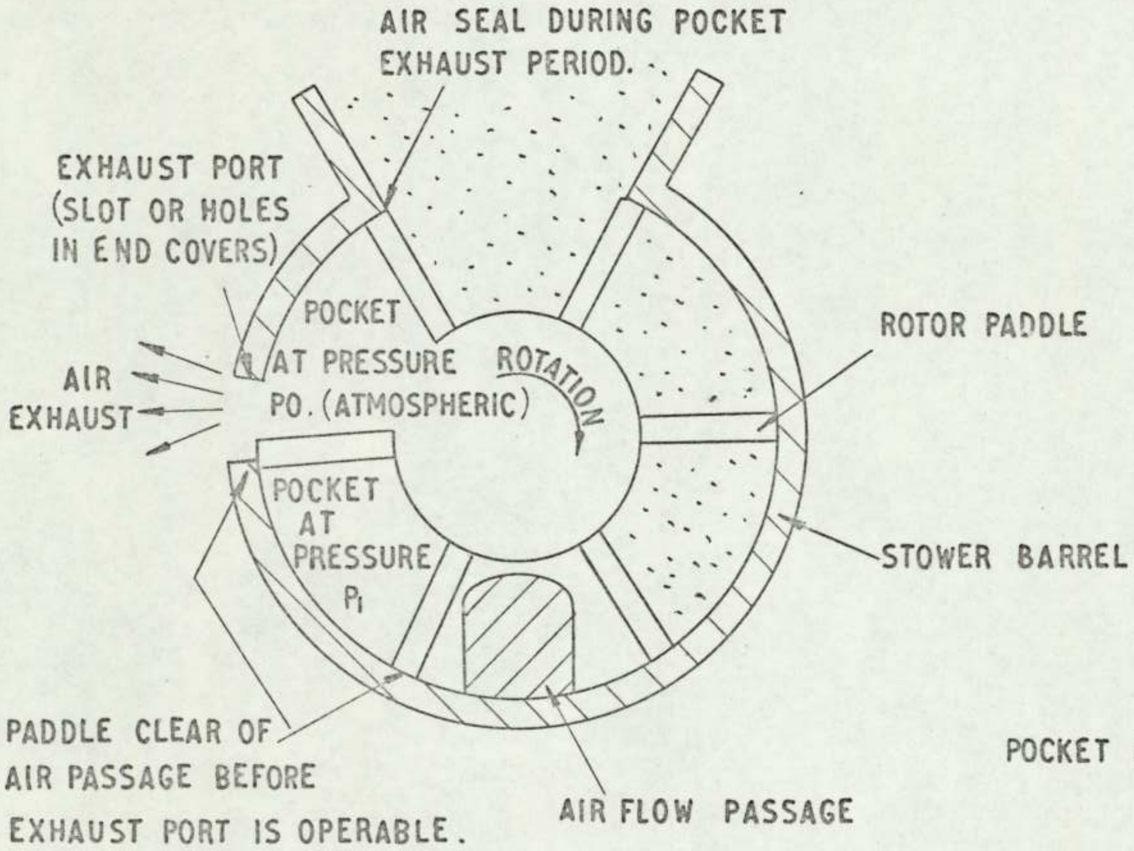
V = Volume of rotor pocket.

t = Instant of time.

Po = Air at atmospheric pressure.

k = Factor derived from air velocity and orifice area.

Assuming isothermal expansion,



POCKET EXHAUST TIME = $^{n}/60\,Pn$. SECS.

WHERE n = ROTOR SPEED (R.P.M.)

Pn = NUMBER OF ROTOR POCKETS.

FIG. 8 ROTOR POCKET EXHAUST CONDITIONS

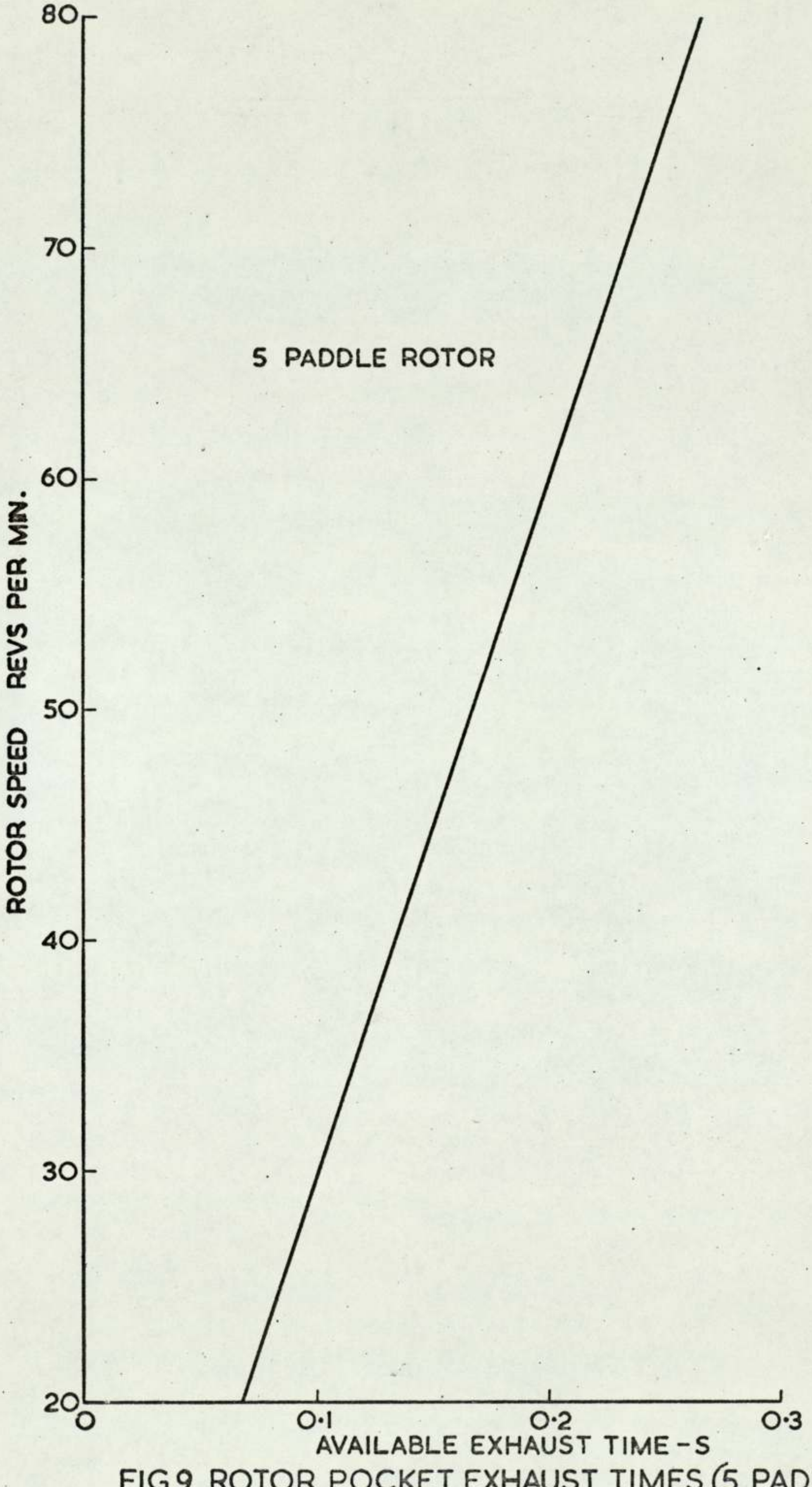


FIG 9 ROTOR POCKET EXHAUST TIMES (5 PADDLE)

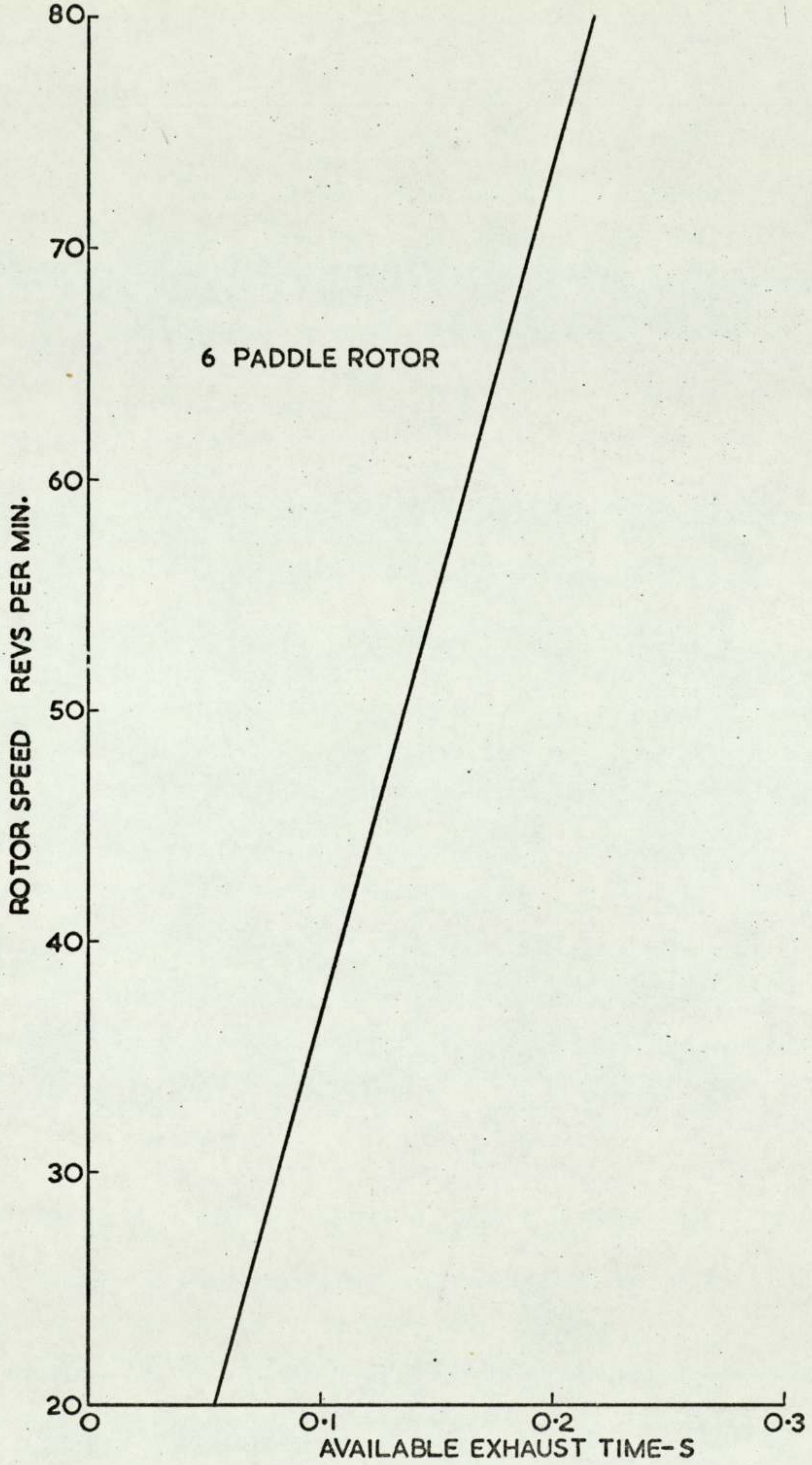


FIG 10 ROTOR POCKET EXHAUST TIMES (6 PADDLE)

At time t we have a volume of air V at pressure P.

At time t + dt we have a volume of air V at pressure P + dP.

$\therefore$ Loss of air at atmospheric pressure $= \frac{PV}{P_o} - \frac{(P + dP)V}{P_o}$

$$\text{Hence } \frac{PV}{P_o} - \frac{(P + dP)V}{P_o} = k\sqrt{P - P_o}\ dt \ \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

$$-\frac{dPV}{P_o} = k\sqrt{P - P_o}\ dt$$

$$\int \frac{-dP}{\sqrt{P - P_o}} = \int \frac{P_o\, k}{V}\ dt$$

$$\therefore\ -2\sqrt{P - P_o} = \frac{P_o\, k}{V}\, t + \text{constant} \ \ldots\ldots\ldots\ldots (2)$$

At time $t = 0$, let pressure be P_1

$$\therefore\ \text{Constant} = -2\sqrt{P_1 - P_o}$$

$$\text{and } \frac{P_o\, k\, t}{V} = 2\left(\sqrt{P_1 - P_o} - \sqrt{P - P_o}\right)$$

when $P = P_o$ the time taken is given by

$$T = \frac{2V}{P_o\, k}\sqrt{P_1 - P_o}$$

Expressing the pressures in terms of ft head,

$$T = \frac{2\ V\ W_o}{144\ P_o\ k}\sqrt{\frac{144\ (P_1 - P_o)}{W_1}} \ \ldots\ldots\ldots (3)$$

where W = density of the air at the corresponding pressure and temperature.

Factor k

The discharge losses at the orifice will be dependant on the size and shape of the orifice, and as the factor can be as low as 0.6 it is necessary to consider this.

Air velocity at the orifice $= \sqrt{2\ g\ H}$

Air Volume (Discharge) $= C_d\ A\sqrt{2\ g\ H}$ ....(4)

where C_d = Coefficient of discharge

From equation (1) loss of air $= k\sqrt{P - P_o}\ dt$

$= k\sqrt{\frac{(P - P_o)144}{W}}\ dt$ (Pressure difference expressed in ft-head)

Substituting for k in equation 4,

$$k = Cd\ A\sqrt{2\ g} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(5)$$

Substituting in equation 3,

$$T = \frac{2\ V\ Wo\sqrt{\dfrac{144\ (P\,1 - Po)}{W\,1}}}{144\ Po\ Cd\ A\sqrt{2\ g}}$$

Exhaust time (T)

$$= \frac{V\ Wo\sqrt{144\ (P_1 - Po)\ /W}}{72\ Po\ A\ Cd\sqrt{2g}} \quad \ldots\ldots\ldots(6)$$

where V = Volume of Rotor Pocket - in^3

P = Air Pressure in Rotor Pocket - lb/in^2

Po = Atmospheric Pressure - lb/in^2

A = Area of Exhaust Orifice - in^2

W = Air density at Equivalent pressure - lb/ft^3 (Constant Ambient Temperature assumed)

The available time to exhaust a rotor pocket is dependant on the speed of the rotor. By equating the available time to equation 6 the diameter of the orifice can be determined.

The equation can only give an approximation of the time to exhaust the rotor pocket because the factor 'k' is based on a discharge velocity for a constant pressure head. In the case of the stower the pressure head reduces down to zero during the time exhausting is taking place.

Applying the formula for a range of pocket pressures up to the maximum of approximately 15 lb/in^2 ($103 \times 10^3\ N/m^2$) gives a range of orifice diameters between 1.3 in (31 mm) and 2.5 in (63 mm). This is based on a range of rotor speeds between 30 and 60 R.P.M. and an assumed discharge coefficient of 0.5. The operating condition will in fact be somewhere between these limits.

From the figures quoted above it is concluded that two 2 in (51 mm) diameter exhaust ports (one in each end of the stower) would be sufficient to allow for any discharge losses and still ensure complete exhaust to atmosphere before the pocket reaches the material feed position at the top of the stower.

3.10 AIR SEALING

In order to obtain the maximum effort from the available air, it is necessary to keep the air losses to an absolute minimum. This in itself is not difficult to achieve providing very fine clearances between relative components can be maintained. Initially the running clearances between the rotor and the stower barrel can be obtained by controlling the machined sizes of the rotor tip diameter and the bore of the stower barrel. The air leakage problem arises when wear takes place in the bore of the stower or on the rotor blade tips.

Previous stower designs provided adjustable steel plates inside the stower body. These plates were in the form of segments which were machined to suit the rotor diameter. Adjustment of the plates to reduce rotor tip clearances was carried out by means of large adjusting screws which were fitted through the stower barrel. This design of stower presented problems which affected the reliability and efficiency of stowing operations. These were :

1. The inclusion of adjustable plates complicates the stower design and increases the cost of manufacture.
2. The regular maintenance requirements were extremely difficult to enforce underground where machine operators are not necessarily technicians.
3. After the range of adjustment had been used, it was necessary to return the stower unit back to the suppliers for renewal of the wear plates. It was also necessary to trim the tip diameter of the rotor to suit the radius of the new plates.
4. During the factory reconditioning period, stowing operations were suspended unless an additional stower unit was held in stock as a standby unit. This added to the already high capital expenditure required for stowing operations.

In order to simplify the design of the stower and to reduce manufacturing and operating costs, it became evident that the "wear plate adjustment" feature had to be eliminated. The provision of a simple cheap stower barrel which when worn, could be replaced was first considered. This would be satisfactory providing all the wear took place only on the barrel and not on the rotor tips, a condition which was extremely unlikely unless the bore of the barrel was made of softer material than the rotor. This idea led to fitting the stower barrel with a liner of material softer than that of the rotor tips. This liner could easily be replaced after wear had taken place.

After investigations into the various available materials suitable for this application, Nylon 66 (Polypenco) appeared to meet the requirements. This material was commercially available in bars, sheets and tube. This type Nylon 66 is the most universally used polyamide for industrial and engineering purposes. It is exceptionally tough, strong, resilient, light in weight, resistant to shock, wear and mechanical stresses, non-brittle and has a low coefficient of friction requiring little or no lubrication.

It was thought that the resilience of this material would provide a means of enabling the rotor tip clearance to be kept to a minimum for sufficient time to give a reasonable life to the liners before replacement was necessary.

A major advantage of including polypenco liners was the simplification of the stower design, and the resulting reduction in manufacturing costs. This feature also eliminated any maintenance between liner changes. It was also possible to design for liner changes underground. At this stage there was insufficient evidence to indicate what the life of Polypenco liners would be, this could only be ascertained from the results of trials to be carried out during the development programme. It was however, considered that a change of liners every three or four months could still give an advantage over other systems particularly when considering the savings in labour and time during the maintance period.

CHAPTER 4

TEST RIG DESIGN

4.1 STOWER TEST RIG

Having established the principles on which the development programme was to be based, the first requirement was to build a stower test rig to provide information on,

1. The reliability of the metering system.
2. The rate of output.
3. The rotor design.
4. The pressure losses through the stower unit.

As a first stage, the rig was designed to be as simple as possible. It consisted of a plain steel barrel, with end covers which carried the sleeve bearings for the rotor shaft. At this stage no provision was made for barrel or rotor wear as it was planned that specific limited trials would be sufficient to provide the basic information, and this would then be followed by further investigations and development dependant on the results of the tests.

The design of the stower unit drive, whilst important when considering a complete stowing installation underground, is not within the scope and limitations of this thesis. Therefore, only a brief mention will be made of equipment ancillary to the stower.

The test rig was driven by a 5hp motor through a reduction gearbox, and a chain drive, the latter being the quickest and cheapest method of providing means to run the stower rotor over a range of speeds between 40 and 80 RPM. A coupling incorporating shear pins, was fitted between the gearbox and chain drive to protect the motor and gearbox in the event of the stower becoming blocked. During the initial trials, material was fed to the stower test rig from a feed conveyor which was loaded from a mechanical dumper. The material used during the trials was 'run-of-mine' dirt up to 3 in (76 mm) in size, obtained from local collieries. The conveying air to the stower test rig was supplied from a positive displacement blower having a capacity of 1,00 ft^3/min (0.5 m^3/s) F.A.D. and capable of operating at pressure up to 10 lb/in^2 (69×10^3 N/m^2).

4.2 TEST RIG FEATURES

Examination of the principles of a stowing machine indicated that the efficiency and reliability of the metering system depended on several factors, namely :-

1. Satisfactory material feed to the stower.
2. Control of the material size.
3. Nature of the material, as determined by its stickiness and density.
4. Rotor design and speed.
5. Stower Inlet and Outlet conditions in terms of the size and shape of the aperture in the ends of the stower, and the corresponding features of the stower inlet and outlet pipes.

Items 1, 2 and 3 are determined by factors outside the scope of this work. It is therefore to be accepted that material of limited size and density can be fed at a known rate to the stower rig. The features covered by items 4 and 5, can now be examined to determine the combination giving the best stower design.

The features directly affecting the metering of the material through the stower into the stowing pipeline can be divided into five parameters. These are,

(i) Rotor Pocket Shape and Capacity

(ii) End Cover Outlet orifice

(iii) End Cover Inlet orifice

(iv) Air Inlet pipe size and shape

(v) Stowing Outlet pipe size and shape (i.e. The first pipe in the stowing line)

Any one or combination of these factors affects the stowing operations.

Figure 7 shows the relative designs of the rotor pocket, end cover orifices, inlet air pipe, and outlet stowing pipe, which were selected for the initial test rig investigations.

4.3 TEST RIG INVESTIGATIONS

The objectives of the test rig investigations were to provide information on the problems encountered with this type of stowing machine, to evaluate the results of the tests, and to produce a specification for equipment which would meet the stowing requirements of the coal mining industry.

In order to clarify the investigations carried out and the corresponding development that was found necessary, the programme is divided into sections. Each section is referred to by the particular features of design which were investigated.

4.3.1. ROTOR (Figure 7)

Initially the test rig was manufactured with the 14 in (356 mm) diameter steel rotor operating in direct contact with the steel barrel, and no provision was made at this stage for wear between these two components. It was considered that the first stages of the investigations could be completed before the running clearances between the rotor and barrel were large enough to affect the operations. The ends of the rotor pockets were tapered so that the pocket width at the exit side was equal to the stowing pipe diameter of 4 in (102 mm).

4.3.2. STOWER OUTLET

A 4 in (102 mm) pipe diameter was selected in conformity with one of two standard pipe sizes already used within the coal industry. Previous experience had shown that material up to 3 in (76 mm) in size could be conveyed down a 4 in (102 mm) diameter pipe.

4.3.3. STOWER INLET

The size of the inlet pipe was determined to a large extent by the available space beneath the stower drive equipment. The 3 in (76 mm) diameter pipe was adequate for the available air supply, and by designing the pipe with several bends, the air duct to the rig was accommodated.

4.4 INITIAL TESTS

The first series of tests carried out were abortive because blockages occurred frequently both in the stower and in the stowing pipes. The air supply from the positive displacement blower was not capable of conveying the material through the stower and along the pipe run. None of the tests carried out was of sufficient time to record any output rates, and in each case the maximum blower pressure was reached, resulting in the relief valve blowing off to atmosphere.

Examination of the stower and pipes after each test showed that the causes of the breakdown were attributable to the failure to accelerate the material from the rotor pocket. This had the effect of reducing the air flow through the system which allowed material in the pipeline to settle resulting in material build-up and eventual blockage. In no case was the blockage due to the jamming of oversize pieces.

It was apparent that the tapered guide pieces at the ends of the rotor paddles were preventing the material from flowing freely through the pockets. Because the cross sectional area in the centre of a paddle was greater than that at the exit, material compression was taking place and more energy was required from the available air supply to convey the material out of the stower.

To improve the outlet conditions, it was decided to remove the taper pieces in the pockets. This made the width of the pocket 4 in (102 mm) along its entire length. In introducing this modification it was considered that the outlet cover orifice should also be modified to keep the hole width at 4 in (102 mm); the same as the pocket. Figure 11 shows the new pocket and orifice shapes. It also shows the design of the outlet pipe which was now a transition piece, having the same cross section as the new orifice at one end, and retaining its 4 in (102 mm) diameter at the other.

Further tests were carried out and it was found that flow rates of up to 24 tons/h (24.4 tonnes/h) were achieved, but pipe blockages occurred frequently in the special adaptor pipe on the stower outlet. Another problem was the relatively high pressure of 10 lb/in^2 (69×10^3 N/m^2) was recorded during the first tests. This pressure represented about 70% of the available pressure from the blower.

At this early stage of the work it became apparent that there were two basic problems: outlet pipe blockages, and the comparatively large pressure drop across the air inlet pipe and the stower.

The first approach to overcome these problems was to see what was required to reduce the pressure losses across the stower and inlet pipes, because an improvement in this would result in more pressure energy being made available to transport the material through the stower and adaptor pipe.

4.5 AIR INLET PIPING

Consider the air inlet pipe 3 in (76 mm) diameter

The type of flow in the pipe is dependent on the Reynolds number. (Re)

$$Re = \frac{Vd}{v} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

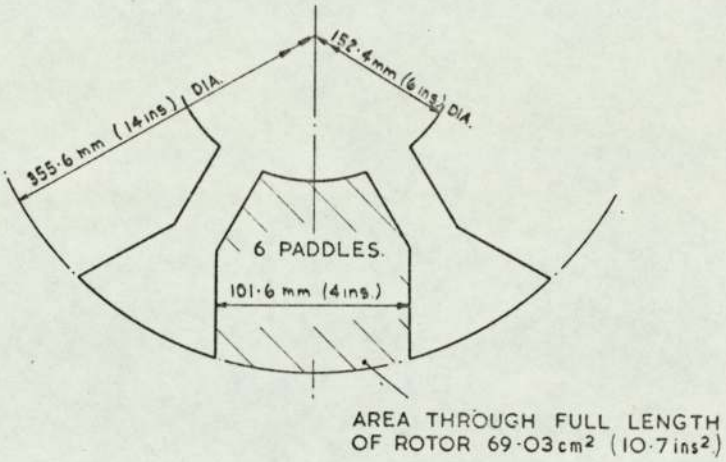
where V = Mean velocity in the pipe (ft/sec)

d = diameter of pipe (ft)

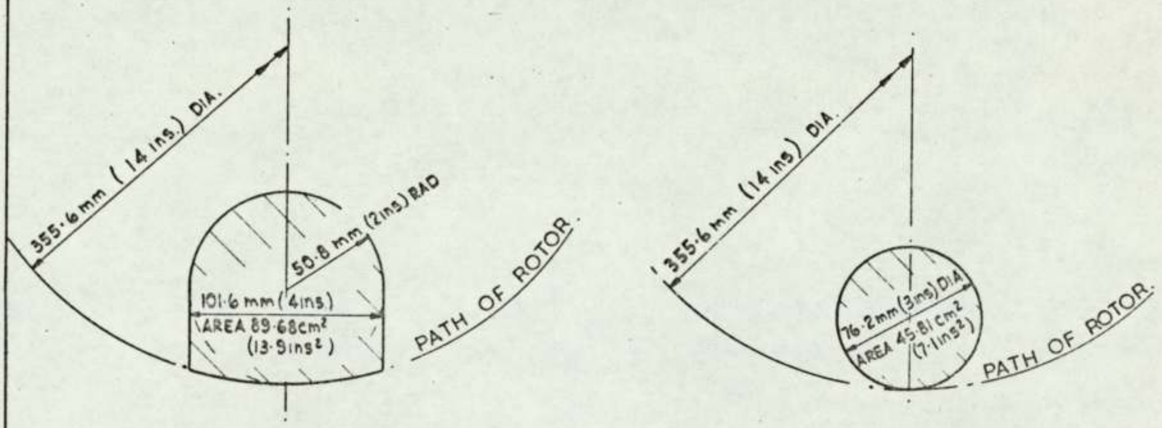
v = Kinematic viscosity of the air (ft^2/sec)

Blower capacity = 1000 ft^3/min - Free air delivery

$\therefore$ for pipe pressure of 1 atmosphere, air volume = 500 ft^3/min

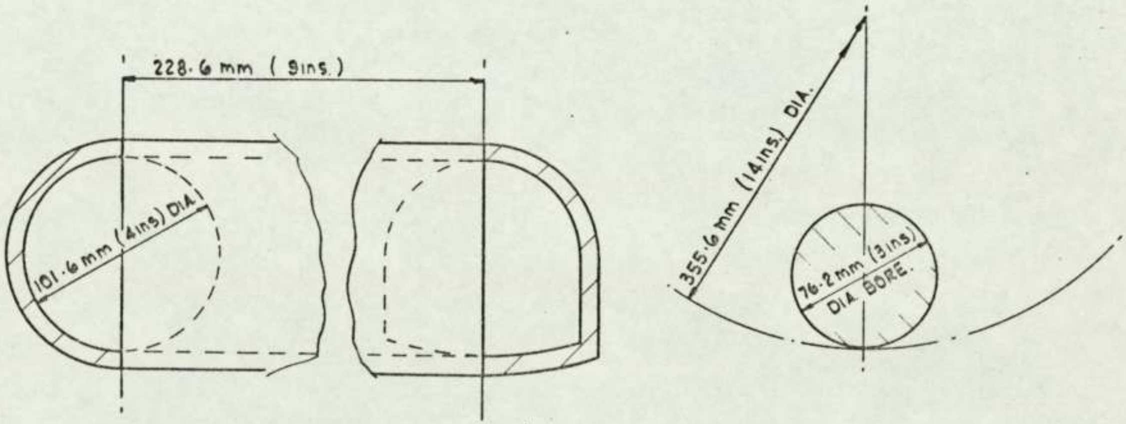


(a) ROTOR.



(b) END COVER (OUTLET)

(c) END COVER (INLET)



(d) OUTLET PIPE TAPERED FROM END COVER HOLE SECTION TO 101·6 mm. (4 ins.) DIA. BORE.

(e) INLET PIPE.

FIG 11 STOWER DESIGN FEATURES

$$Re = \frac{500 \times 4}{60 \times \pi \times d^2} \times \frac{d}{v}$$

$$= \frac{500 \times 4}{60 \times \pi \times d} \times \frac{1}{v}$$

$$= {}^{100}/_{3}\ dv$$

$$\therefore Re = 10.6\ dv \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

From "Knetic Viscosity of air" chart

v for air at 15°C (Ambient Temp.) = 0.8×10^4 ft^2/sec.

Substituting in (2) $Re = 10.6 \times .25 \times 0.8 \times 10^4$

$\therefore$ $Re = 21{,}250$ (Flow is turbulent)

For Turbulent Flow the head loss on a pipe due to friction is given by the Darcy Formulae:

$$h_f = f\,\frac{LV^2}{2gd} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

where h_f	=	head loss	ft fluid
L	=	Length of pipe	ft
d	=	Pipe diameter	ft
V	=	Velocity	ft/sec
f	=	friction coefficient	dimensionless

Applying this formula to the air supply pipe between the blower and stower, the friction coefficient is based on the assumption that it operates in the "Transition" region between smooth and rough pipes. In this region the friction coefficient depends on both the Reynolds number and the relative roughness of the pipe.

The transition formula to cover this region, introduced in 1939 by Colebrook and White (10,11) is defined as:

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left[\frac{2.51}{Re\sqrt{f}} + \frac{k}{3.7d}\right]$$

where k = Hydraulic Roughness

The formula has been plotted by Professor Moody (12) as a graph of f against Re for given values of k/d.

To determine pressure loss in air pipe between Blower and Stower

1. Straight Pipes

For smooth rubber hose or steel pipe,

Hydraulic Roughness (k) = 0.002 in

$$\frac{k}{d} \text{ Factor} = \frac{0.002}{0.25} = 0.008$$

From the "Moody Chart of Colebrook-White Transition Formula":-

$$\text{For Re} = 21{,}250 \text{ and } \frac{k}{d} \text{ factor } 0.008$$

$$f = 0.038$$

Substituting in equation (3)

$$h_f = \frac{.0236L \times 500 \times 4}{2 \times 32.2 \times .25 \times 60 \times \pi \times .25^2} \text{ ft}$$

$$\text{Head loss in Pipe} = 0.4L \text{ ft}$$

This represents a pressure loss of 1.72 lb/in^2 per 10 ft length of straight piping. (11.8 x 10^3 N/m^2 per 3 m length)

This figure will vary according to the air flow in the pipe and could increase as much as 60% at the maximum air velocity.

2. Pipe Bends

It was not possible to provide a straight pipe run between the blower and the stower, and because of this some consideration has to be given to the pressure losses due to pipe bends. In the case of the test rig trials the pipe layout included two bends of about 45°.

Because of the flexibility of the air inlet piping, the precise bend radius was not known but a figure of approximately 12 in has been assumed for the purpose of estimating the pressure losses across the two bends.

The head loss in a pipe due to a fitting or bend can be expressed in terms of an equivalent length of straight pipe of the same nominal diameter.

Using D'arcy's formulae $H = f\frac{L}{D}\frac{V^2}{2g}$

The equivalent length $L = \frac{DK}{f}$ ....(5)

where K = loss coefficient

The results of tests on smooth bends by WASIELEWSKI[(13)] provides the loss coefficient for bends less than 90° for $\frac{R}{d}$ ratios

where R = Bend radius

d = Pipe diameter

For the 45° bends in the air inlet piping

$$\frac{R}{d} \text{ factor} = 4$$

From Wasielewski's graph.

Loss Coefficient K = 0.008

Substituting in (5)

$$\text{equivalent head loss} = \frac{.25 \times .08}{0.038} \text{ ft}$$

$$= 0.53 \text{ lb/in}^2 \ (4 \times 10^3 \text{ N/m}^2)$$

Summing up the study of the air inlet piping, the calculations were made to give a reasonable estimate of the proportion of the available air pressure absorbed due to the resistance within the inlet piping system.

The calculations show that the inlet pipe losses could be as high as 3 lb/in^2 (21×10^3 N/m^2) on the basis that the pipe layout included 10 ft (3 m) of piping and two 45° bends. This represents about 25% of the available air pressure supplied from the blower used in the tests.

The unsatisfactory results of the first test rig trials established that substantial improvements were required if the objectives of the work were to be achieved. Because of influence of the various design features on one another, it is necessary to plan a programme of investigation so that the effects of each individual modification were established by tests. In this way the development of the stowing equipment could be carried out step by step in a logical and orderly manner, and a true assessment of its potential could be made.

4.6 TEST RIG LAYOUT

The equipment assembled to carry out the trials programme consisted of a 12-ton capacity bunker which had an open top and was truncated at the bottom to provide a sliding door through which the material could be fed onto a belt conveyor. The conveyor carried the material from the bunker and fed it directly into the stower unit (Figures 12 & 13). Midway between the bunker and the stower, a weighing machine was fitted which automatically weighed the amount of material being fed to the stower (Figure 14). The rate of flow of material was obtained by checking the duration of each test with a stop watch.

From the stower unit, the pipe run was maintained at a length of 60 yds which included one 90° bend. (Figures 15 & 16) This particular pipe run was chosen because stowing operation underground, using inbye compressors, always included at least one 90° bend, and stowing circuits generally were anything between 20 and 60 yds long.

For convenience the material was stowed into concrete cells specially erected for the purpose. (Figure 17)

During the whole of the investigations the test rig equipment was varied from time to time, but the basic features remained as described above.

4.7 PHASE I TESTS (6 PADDLE ROTOR)

The shape and dimensions of the rotor and air passage selected for the first tests are shown in Figure 18. The trials were carried out using 'run-of-mine' dirt which was sized to -3in (-76 mm), and having a density of 72 lb/ft^3 (1152 kg/m^3). A number of the tests carried out were abortive mainly due to stoppages caused by foreign bodies

FIG 12 BELT FEED CONVEYOR TO STOWER

FIG 13 FEED HOPPER TO STOWER

FIG 14 AUTOMATIC WEIGHING MACHINE BETWEEN BUNKER & STOWER

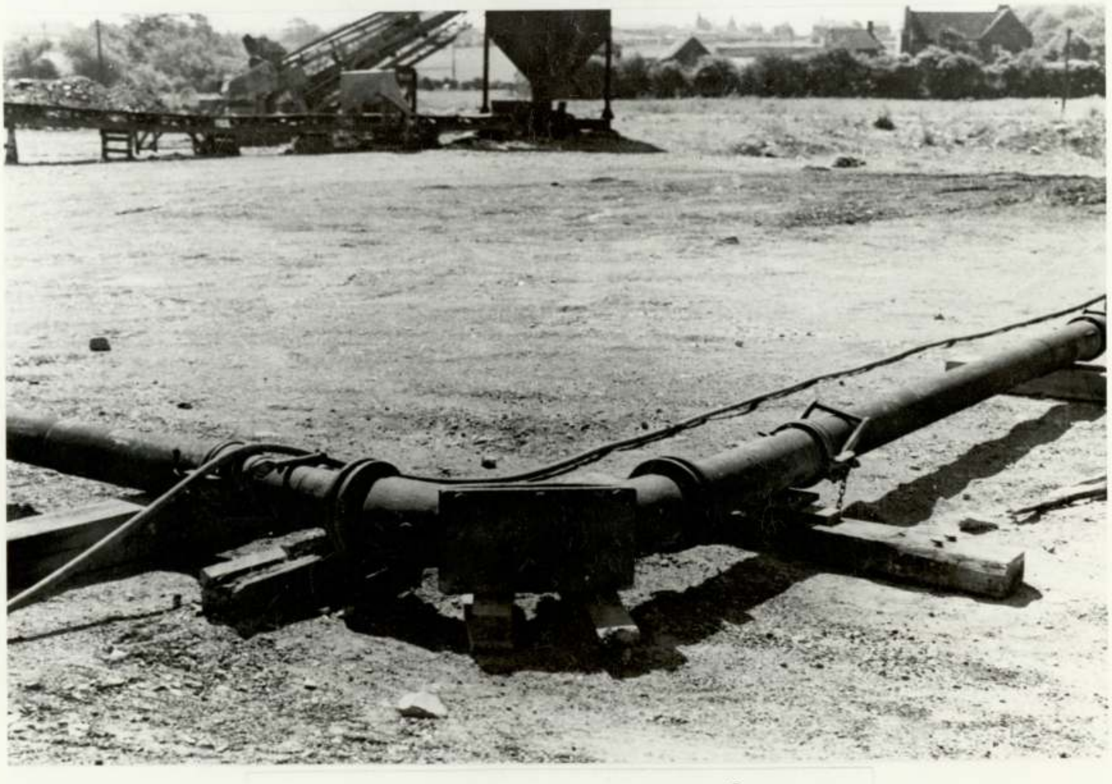


FIG 15 STOWING PIPE 90° BEND

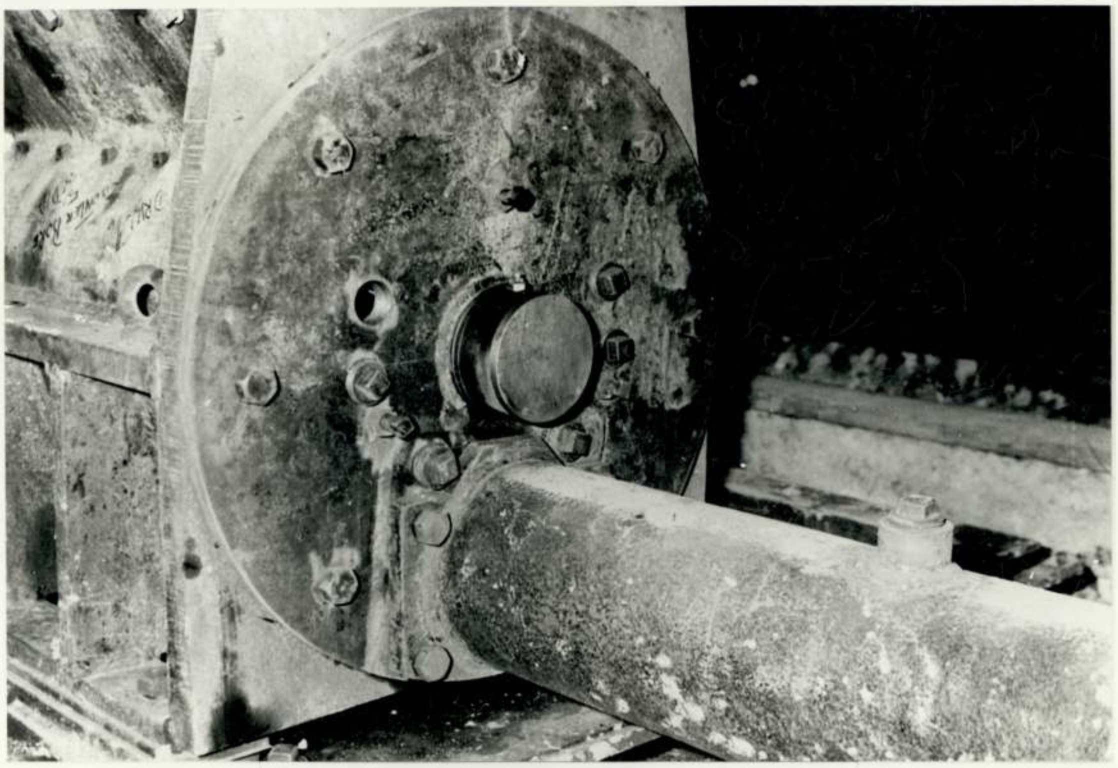
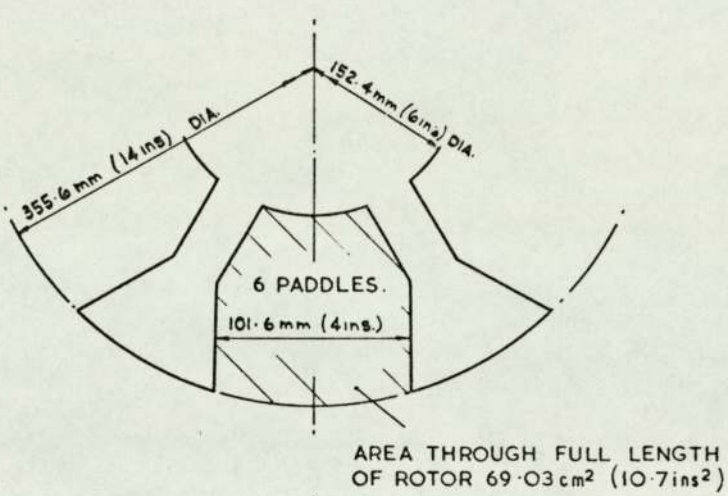
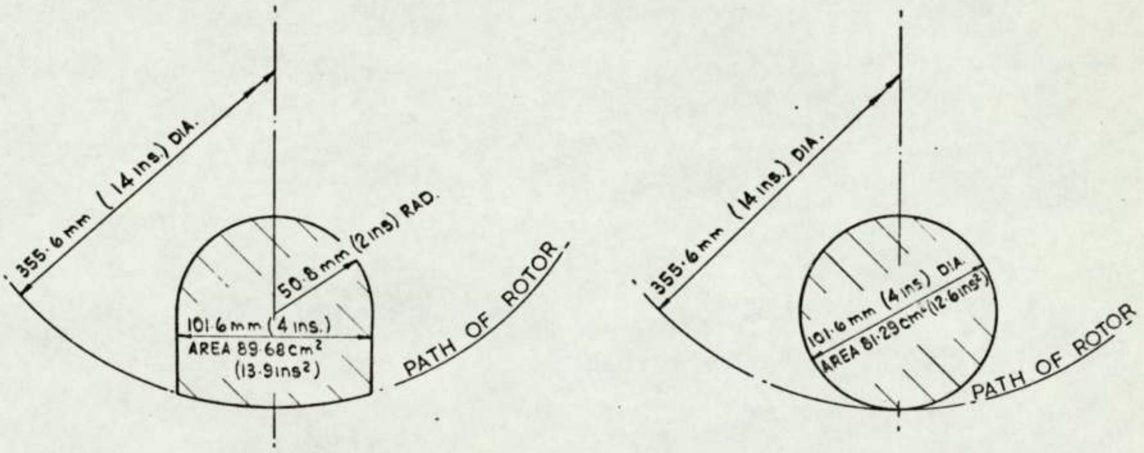


FIG 16   STOWER OUTLET PIPE

FIG 17   CELL FOR STOWING MATERIAL

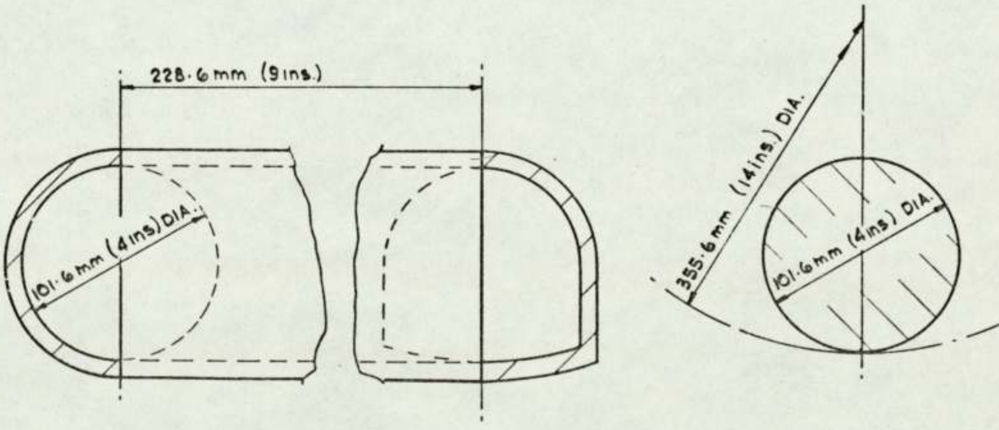


(a) ROTOR.



(b) END COVER (OUTLET)

(c) END COVER (INLET)



(d) OUTLET PIPE TAPERED FROM END COVER HOLE SECTION TO 101·6mm. (4ins.) DIA. BORE.

(e) INLET PIPE.

FIG. 18 STOWER DESIGN FEATURES

(tramp iron, nuts, bolts, steel plates etc.) becoming jammed in the stower unit. These undesirable elements are sometimes present in 'run-of-mine' dirt, and no attempt other than viewing the material travelling along the feed conveyor, was made to eliminate this hazard. The trials which were conducted at rotor speeds of 30 and 34 rev/min showed some improvement and outputs up to 29.6 tons/h (30.1 tonnes/h) were recorded for pressures up to 13 lb/in^2 (90×10^3 N/m^2) (Figure 19).

From the results of the first tests the following conclusions were made :-

(a) Successful stowing was achieved with the "straight through" blow.

(b) The output efficiency of approximately 45% was unsatisfactory.

(c) The recorded back pressure was at the limit of the Blower unit.

(d) Air losses in the stower barrel were too high, mainly due to leakage between the rotor and barrel.

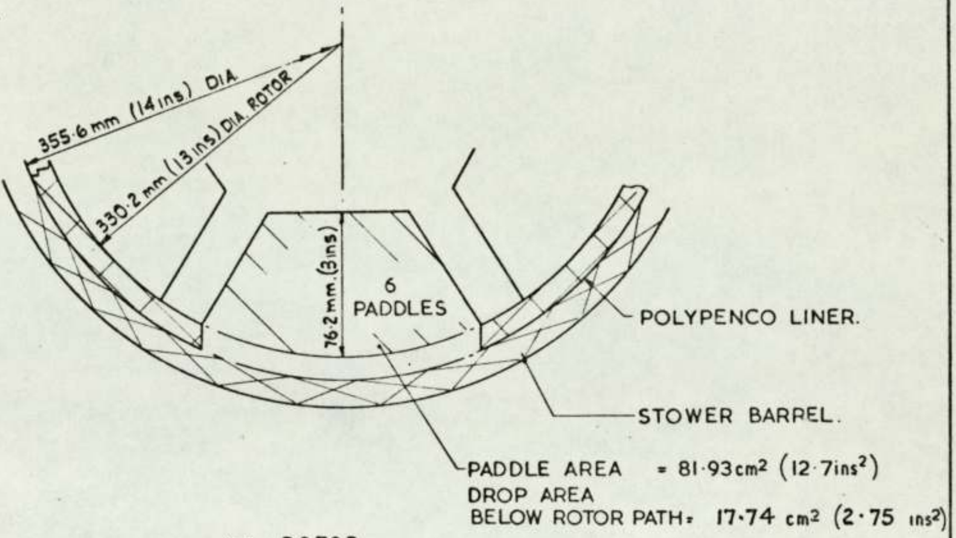
(e) The occasional oversize pieces of material are usually broken by the crushing action of the rotor paddles, and an increase in the power of the stower drive motor would be desirable to cater for the peak loading when crushing does take place.

To improve the passage through the stower and to reduce the air losses between the rotor and barrel, the stower was modified by introducing polypenco liners in the barrel, and eliminating the tapered sections at the rotor tips (Figure 20). The inlet and outlet pipe conditions remained the same.

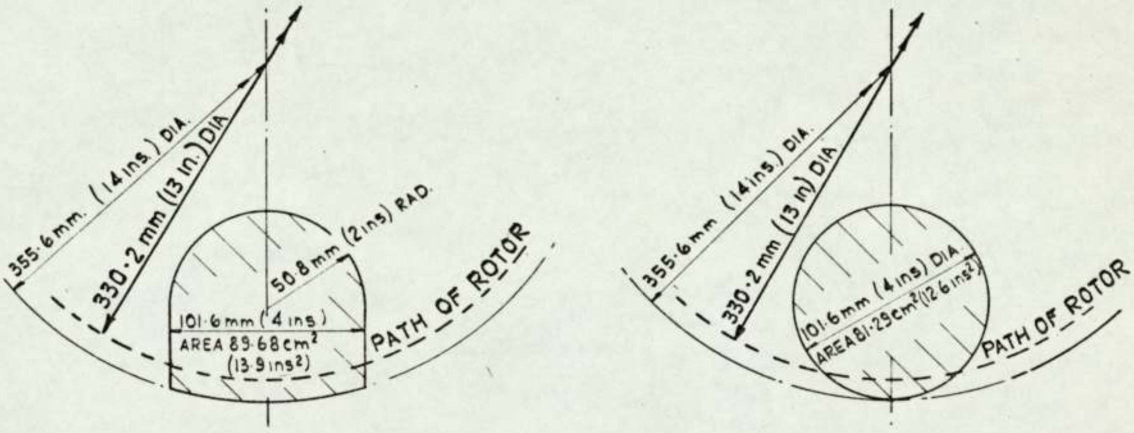
Further trials with this arrangement were unsatisfactory because of continual blockages in the stower and the first stowing pipe. Examinations carried out each time the blockages occurred, indicated that the 12.7 mm ($\frac{1}{2}$ in) step created at the base of the stower by the introduction of the barrel liner, was causing blockage within the stower. It also appeared that the section of the aperture on the stower outlet was not allowing a free discharge of material from the stower in the pipeline.

DESIGN FEATURES	STOWING PIPE CIRCUIT	AIR SOURCE (FREE AIR DELIVERY)	ROTOR SPEED R.P.M.	MEASURED THROUGHPUT		AVERAGE MEASURED THROUGHPUT		THEORETICAL OUTPUT		PRESSURE AT BLOWER OUTLET		OUTPUT EFFICIENCY
				tonnes/hr	tons/hr.	tonnes/hr	tons/hr	tonnes/hr	tons/hr	N/m²	lb/ins²	
REF. FIG. 18	27·4 m (30 YDS.) STRAIGHT PIPE	POSITIVE DISPLACEMENT BLOWER 0·5 m³/sec (1000 ft³/min)	34	20·5	20·16							
				25·7	25·3	28·97	28·51	64	63	89·6X10³	13	45 %
				40·4	39·8							
REF. FIG. 18	54·8 m (60 YDS.) STRAIGHT PIPE + 90° BEND	POSITIVE DISPLACEMENT BLOWER 0·5 m³/sec (1000 ft³/min)	34	24·3	23·9							
				30·1	29·6	27·41	26·98	64	63	89·6X10³	13	43 %
				27·8	27·4							
REF. FIG 18	27·4 m (30 YDS.) STRAIGHT PIPE	POSITIVE DISPLACEMENT BLOWER 0·5 m³/sec (1000 ft³/min)	30	26·2	25·76							
				29·6	29·1	26·67	26·25	54·86	54	89·6X10³	13	47 %
				24·3	23·9							
REF. FIG. 18	54·8 m (60 YDS.) STRAIGHT PIPE + 90° BEND	POSITIVE DISPLACEMENT BLOWER 0·5 m³/sec (1000 ft³/min)	30	30·3	29·8							
				31·5	31·0	30·13	29·66	54·86	54	89·6X10³	13	55 %
				28·7	28·2							

FIG 19 STOWER TRIALS RESULTS

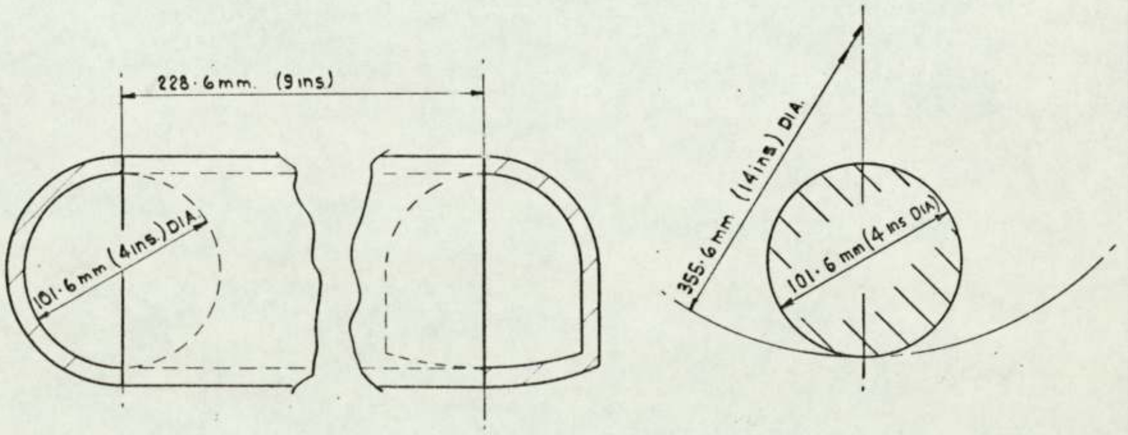


(a) ROTOR.



(b) END COVER (OUTLET)

(c) END COVER (INLET)



(d) OUTLET PIPE TAPERED FROM END COVER HOLE SECTION TO 101·6 mm. (4 ins) DIA. BORE.

(e) INLET PIPE.

FIG. 20 STOWER DESIGN FEATURES

Examination of the blockages that had occurred, suggested that a regrading of the material size from -3 in to -2 in (-76 mm to -51 mm) might reduce the pipe blockages. It was therefore decided to use the smaller material for all future trials.

As far as the air flow was concerned the test results so far indicated that the air consumption was at the limit of the blower capacity. It was therefore decided to carry out the remaining development programme with a Compressor developing 2250 ft^3/min (1.1 m^3/s) free air delivery. This type of compressor was capable of operating at pressures up to 25 lb/in^2 (172 x 10^3 N/m^2).

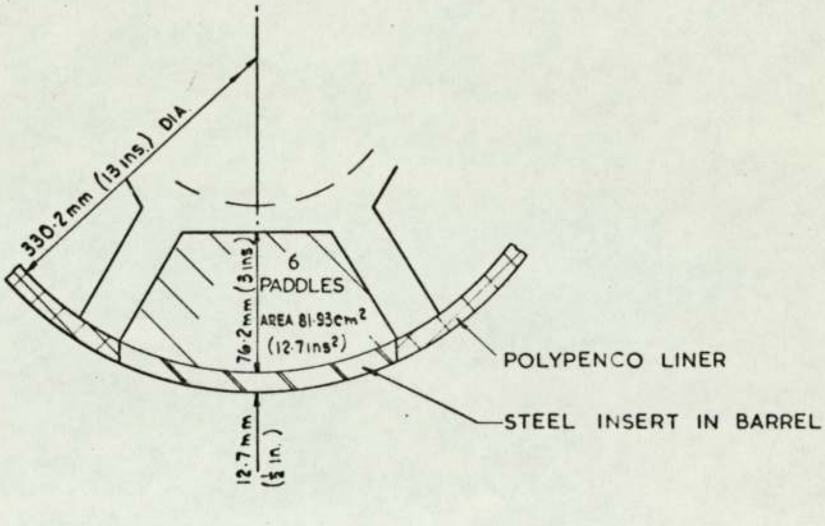
The detailed development and modifications carried out are shown in Appendix 1.

The development concentrated on reducing blockages by making the stower connecting pipe shapes as near as possible to the same shape as the cross section of the rotor pocket.

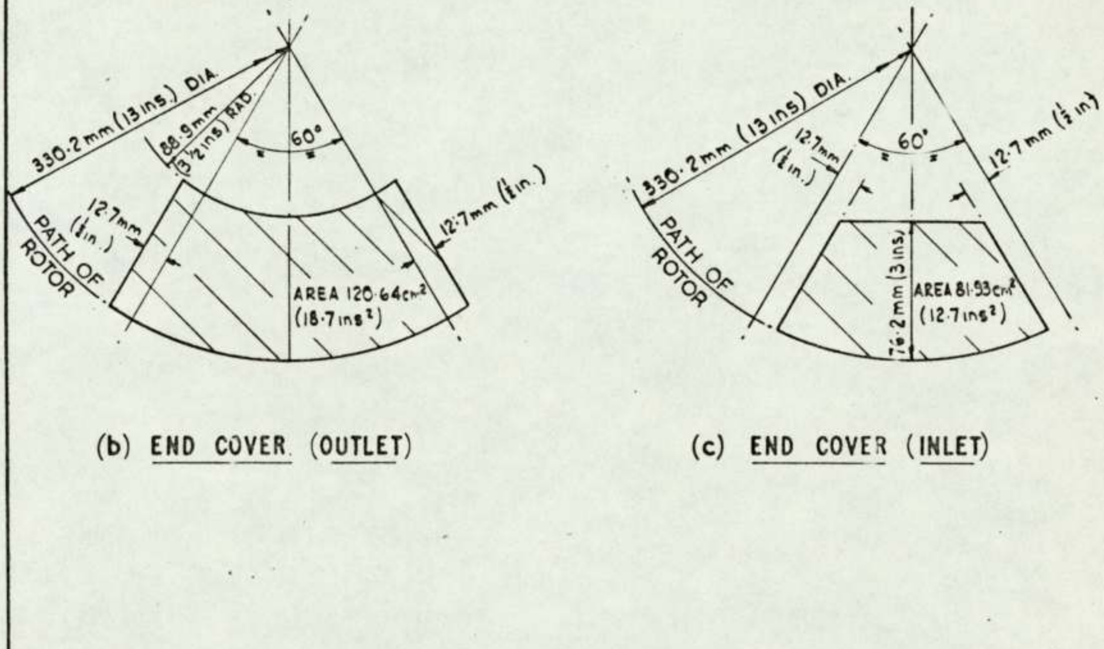
4.8 6 PADDEL ROTOR- ASSESSMENT AND CONCLUSIONS

The final design using the six-paddle rotor (Figure 21) improved the performance of the machine, the output being increased to 50 tons/h (50.8 tonnes/h) with pressures up to 20 lb/in^2 (138 x 10^3 N/m^2). Despite the improvements in output,pipe blockages were still occurring, and it was noticeable that some of the blockages were now found to be in the stowing pipeline clear of the stowing unit. Examination of the blocked pipes did not show any evidence of oversize pieces being the cause. The material was always found to be -2 in (-51 mm) which was packed solid in the pipeline. In theory the blocked pipes could only be caused by a reduction or stoppage of the air flow.

On closer examination of the mechanics of the metering system it can be seen that whilst the air flow to the stower is constant, the flow through the stower and down the outlet stowing pipe range is intermittently cut off each time a full pocket is presented at the

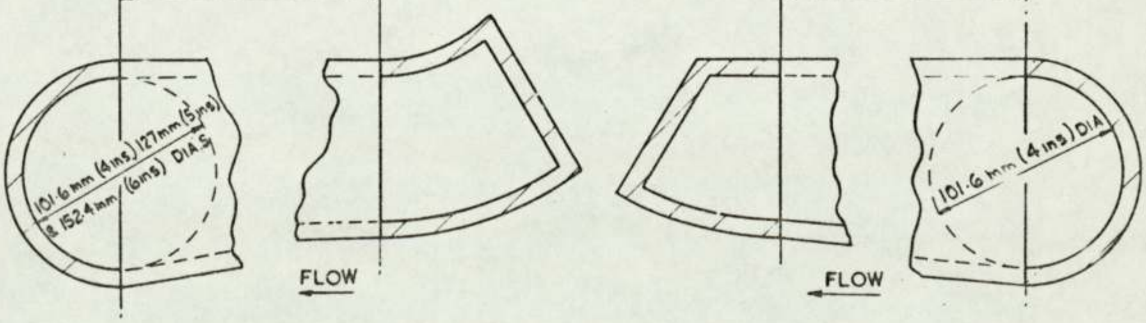


(a) ROTOR



(b) END COVER (OUTLET)

(c) END COVER (INLET)



(d) OUTLET PIPES.

(e) INLET PIPE

FIG. 21 STOWER DESIGN FEATURES

bottom of the stower unit. This theory assumes that the pocket is full and does in effect present a slug of dirt into the full cross section of the air stream. This condition, which prevails each time a rotor pocket passes through the air stream, can result in sufficient reduction in air velocity in the stowing pipe to allow the material to settle in the bottom of the stowing pipe and finally cause complete blockage.

To overcome this problem it was considered necessary to increase the air passage through the system such that the area was sufficiently greater than the rotor pocket to allow a constant air flow through the stowing pipe system.

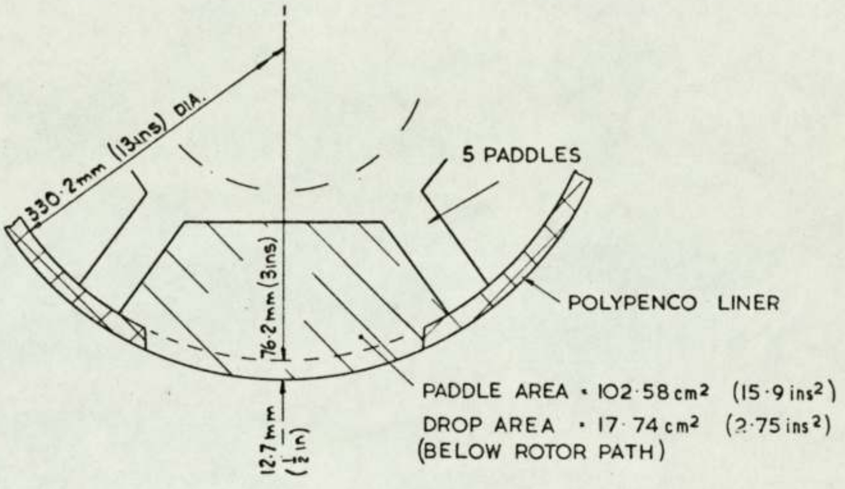
If this principle was correct it was thought that the peak pressures of 20 lb/in^2 (138×10^3 N/m^2) which occurred during the last tests might well be reduced if the pipe build up and blockages could be eliminated.

As far as output is concerned, theoretically, a greater volume of material can be metered by a rotor of the same diameter but with five pockets instead of six. The theoretical increase in material volume is approximately 25%. On this basis the second phase of the investigations were carried out using a five-paddle rotor.

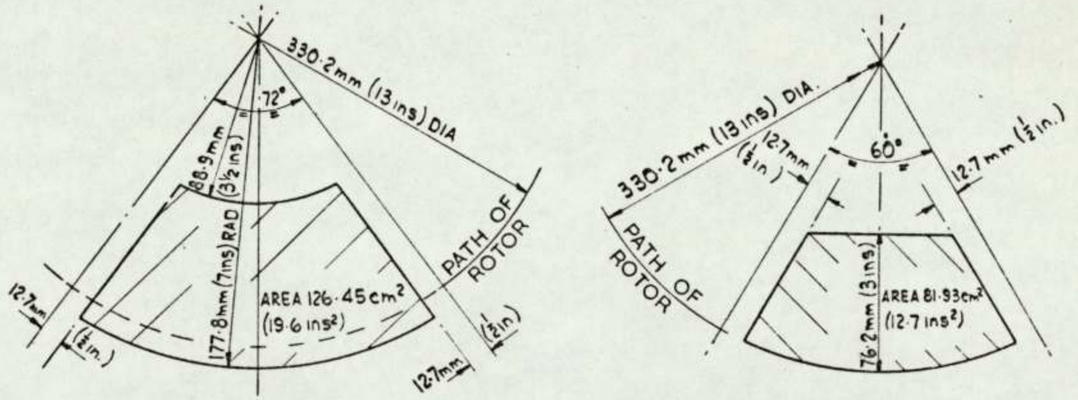
4.9 PHASE II TESTS (5 PADDLE ROTOR)

The step by step development of the stower using the five paddle rotor are given in Appendix 1. The development was started with an additional 0.5 in (13 mm) drop below the rotor periphery (Figure 22). The stower outlet area was also increased to accommodate the larger area of the five-pocket rotor. The size of the stowing pipes was also increased to 6 in (152 mm) diameter to cater for the increased output that it was hoped would be achieved.

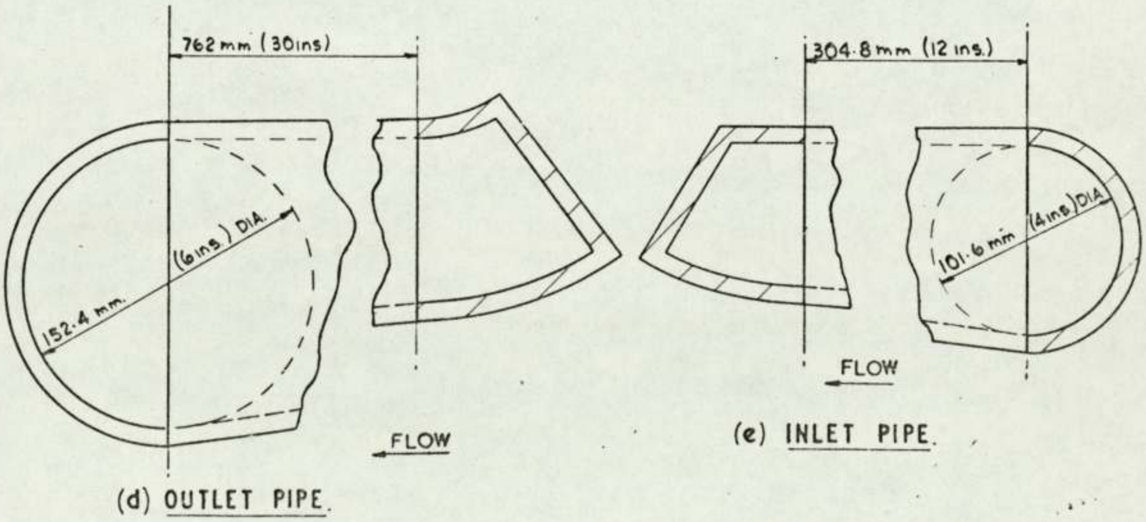
The final design of rotor and pipe shapes which produced the



(a) ROTOR



(b) END COVER (OUTLET) (c) END COVER (INLET)



(e) INLET PIPE

(d) OUTLET PIPE

FIG 22 STOWER DESIGN FEATURES

results which were more acceptable are shown in Figure 23. It can be seen that the circular drop in the stower bottom made it possible to design the inlet and outlet pipes identical to one another and circular in shape throughout their lengths.

4.10 STOWER DEVELOPMENT - ASSESSMENT

The objectives of the development and trials programme was to provide a stower unit which would be free from frequent blockages, have an output rate in excess of 60 tons/h (61 tonnes/h), be of a size within the limitations dictated by the available space underground, and be suitable for operation with an air supply from commercially-available blowers or compressors. This objective was achieved.

The conclusions from the development and trials programme emerge from an analysis of the various features of design which were investigated.

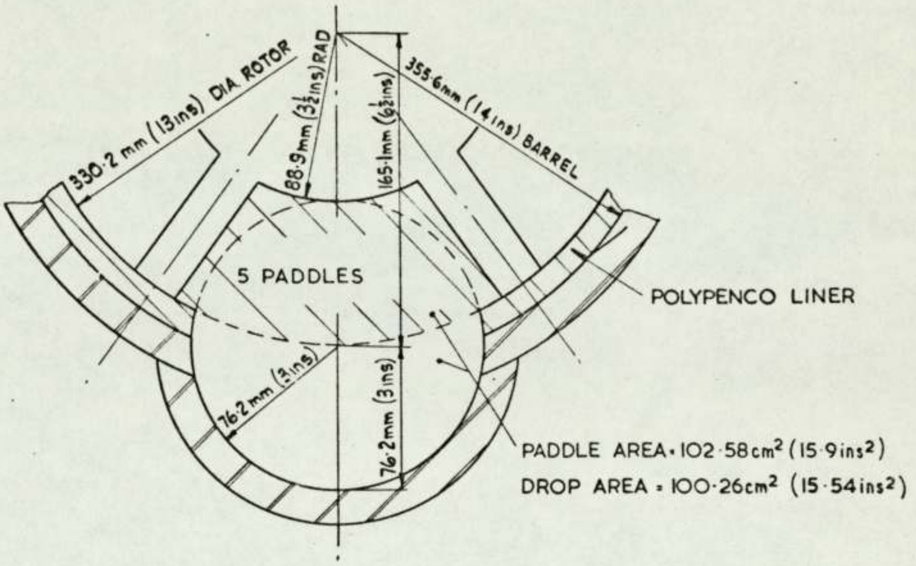
4.11 STOWER ROTOR

The outside diameter and hub diameter of the rotor were virtually fixed by the limitations on size of the stower which is determined by the space restrictions underground. The introduction of Polypenco liners reduced the outside diameter from 14 in (356 mm) to 13 in (330 mm), the latter size being retained throughout the investigations.

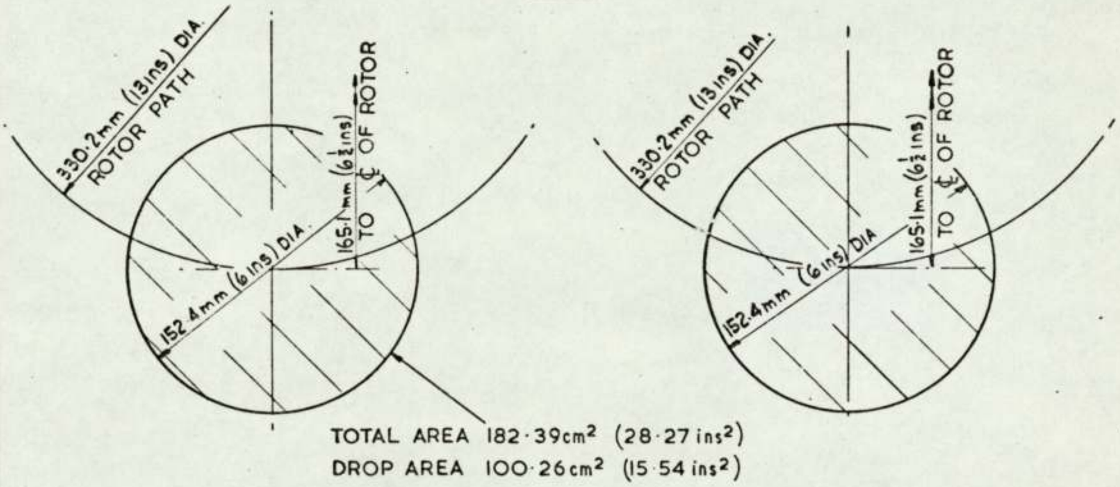
As far as the rotor pocket shape is concerned, this has been varied (Appendix 1) with the object of reducing the pressure drop across the stower, and to provide a good acceleration of material away from the stower. The final shape which is shown in Figure 23 also simplifies the manufacture of the rotor which has a hexagonal shaped hub with straight parallel paddles.

4.12 STOWER INLET AND OUTLET ORIFICE

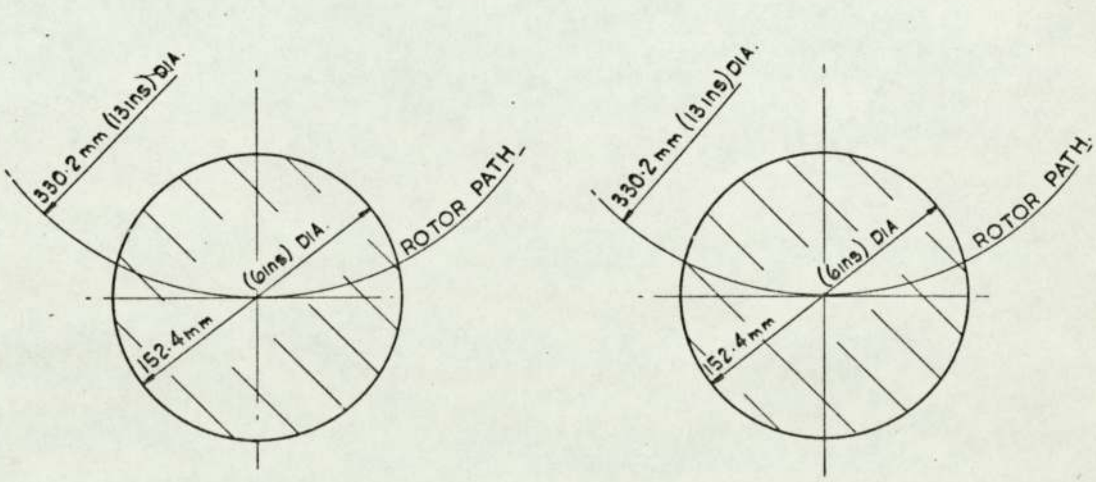
Variation of shapes and sizes of orifices through the stower did not produce any significant improvement until the decision was made to introduce a "drop" in the stower barrel to allow continuous flow of air through the stowing system. The first significant reduction in pressure



(a) ROTOR



(b) END COVER (OUTLET) (c) END COVER (INLET)



(d) OUTLET PIPE. (e) INLET PIPE.

FIG 2.3 STOWER DESIGN FEATURES

drop across the stower was evident when the ½ in (13 mm) "drop" (Figure 22) was introduced. Then further trials with 2 in (51 mm) and 3 in (76 mm) "drops" were carried out.

The results of the fundamental change in the stower design were :-

1. The pressure drop across the stower was reduced from the original 7 lb/in^2 (48×10^3 N/m^2) to 2.5 lb/in (17×10^3 N/m^2)
2. The pneumatic conveying of the material through the stower and along the stowing pipe system was uninterrupted, and no material blockages in the pipes or stower occurred
3. An increase in the capacity of the stower above the target of 60 tons/h (61 tonnes/h) was achieved.

4.13 STOWER INLET AND OUTLET PIPES

The different shapes and sizes of inlet and outlet pipes which were investigated presented severe manufacturing problems. Each pipe was a transition piece which had a continuously changing section from the appropriate orifice shape to a circular section which was the same diameter as the connecting piping.

The introduction of the semi-circular "drop" in the bottom of the stower barrel allowed the stower inlet and outlet pipes to be circular in section throughout their length, as well as being identical. This feature undoubtedly reduces the inlet and outlet pipe production costs, and meant that standard commercially-obtainable piping could be used throughout the stowing pipe system.

4.14 STOWER CAPACITY

The stower trials results have been analysed and a curve plotted showing the throughput against pressure drop (Figure 24). The chart shows the experimental results for the three basic stower barrel conditions which were, (½, 2 and 3 in "drops") (13 mm, 51 mm and 76 mm). The total reduction in pressure drop is of the order of 7 lb/in (48×10^3 N/m^2) and the total back pressure for an output of 80 tons/h

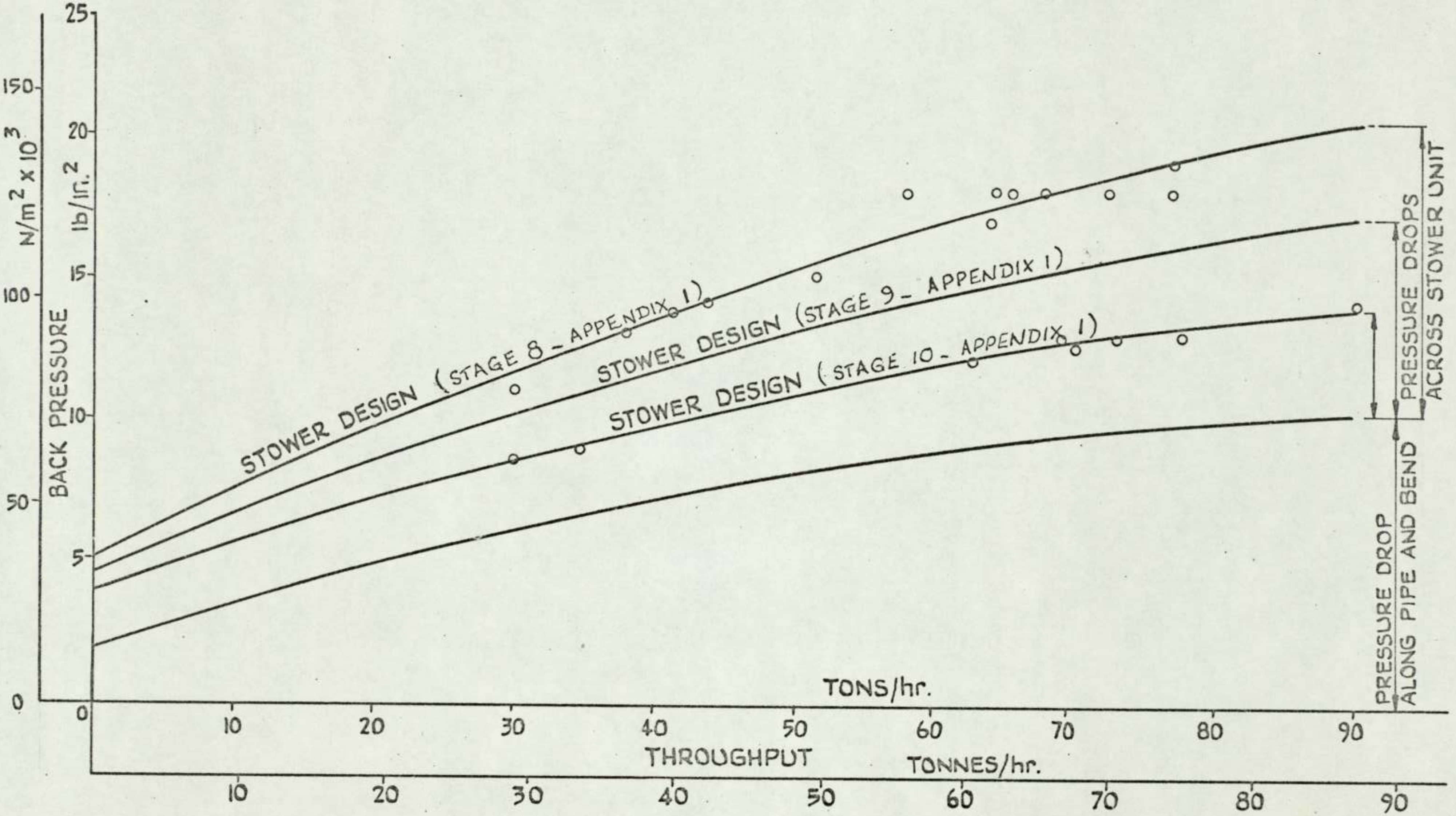


FIG 24 STOWER CAPACITY CURVES.

(81 tonnes/h) is of the order of 13 lb/in^2 (90 x 10^3 N/m^2). Although the test results show some scatter on the graph, the curves have been plotted through the average points and give an indication of back pressures against material output.

It is also emphasised that these results are based on an air supply of 2.250 ft^3/m (1.1 m^3/s) free air delivery from a compressor capable of operating at pressures up to 25 lb/in^2 (172 x 10^3 N/m^2). No attempt has been made in the study to indicate what proportion of the pressure drop along the stowing pipe range was attributable to the 90° bend.

4.15 OUTPUT EFFICIENCY

On closer examination of the results of this work, the measure of success is not only based on the outputs achieved, but must also take into consideration the operating efficiency of the stower unit.

The output efficiency is given by :-

$$\frac{\text{Measured Rotor Throughput}}{\text{Theoretical Rotor Throughput}}$$

The trials have recorded outputs over 100 tons/h (101.6 tonnes/h) but the majority of the results have been between 60 and 80 tons/h (61 and 81 tonnes/h) therefore to obtain a realistic stower efficiency, the calculations are based on the average output which is 70 tons/h (71 tonnes/h). The material density has also varied during the trials programme but on average the material has been of the order of 80 lb/ft^3 (1280 kg/m^3), and this figure has been used in the calculation.

From Figure 6, for a 5 paddle rotor, hub diameter 7 in (178 mm), rotor diameter 13 in (330 mm), and rotor speed 60 r.p.m. :-

Theoretical Rotor output = 143 tons/h (144 tonnes/h) based on a material density of 100 lb/ft^3 (1601 kg/m^3).

For a material density of 80 lb/ft^3

Theoretical Rotor Output = 143 x $^{80}/100$ = 114 tons/h

$$\text{Output Efficiency} = \frac{\text{Achieved Output}}{\text{Theoretical Output}} = \frac{70}{114} \times 100$$

$$= 61\%$$

4.16 AIR REQUIREMENTS

During the early trials, a blower capable of 1000 ft^3/min (0.5 m^3/s) free air delivery was used, but this was found to be incapable of supplying the necessary air flow and pressures for the required outputs. The compressor which was finally used for the majority of the trials operated at pressures up to 25 lb/in (172 x 10^3 N/m) with a capacity of 2,250 ft^3/min (1.1 m^3/s) free air delivery.

4.17 CONCLUSIONS

The objective of producing a design giving a throughput of 60 tons/h (61 tonnes/h) free of blockages was achieved but it was evident that there was not a full understanding of the behaviour of material passing through the system, and that such an understanding may lead to a further significant improvement in performance by giving some indication of the relationship between material behaviour and air flows.

4.18 FURTHER INVESTIGATIONS

It has already been shown that the stower unit has a metering efficiency of about 60% and could probably be as low as 50%. During all the tests carried out the feed of material to the stower unit was such that the hopper at the top of the stower always remained full. This implies that the low throughput efficiency was due to the rotor pockets either not being filled at the top of the stower or not being emptied at the bottom. If these assumptions are valid, though the reasons for them are not clear, it is apparent that the rotor speed, air velocity and the general behaviour of the material passing through the stower unit into the conveying pipeline affects the movement of material through the system. The size, density, and stickiness of the stowing material must also contribute to these problems.

During each stage of the development of the stower unit the test results and general stowing performance were carefully analysed and the faults diagnosed. This was the basis on which the decision to carry out modifications to the stower unit was based. It is emphasised that the conclusions that emerged during each stage of the development were made partly on assumptions as to how the stowing material behaved during its movement through the stower unit and into the conveying pipeline. It was not possible with the prototype stower to study the behaviour of material in the system.

Before a full assessment of stowing equipment can be made it is clear that the behaviour of material passing through the stower and into the conveying pipeline must be know. To provide some of this basic information it was decided to build a stower model which could be used to carry out controlled laboratory tests in order that a study of material behaviour could be made.

The model was to be a full scale replica of the final design of the prototype stower. Figure 25 shows the basic design features of the stower which could operate with either a 5 or 6 paddle rotor.

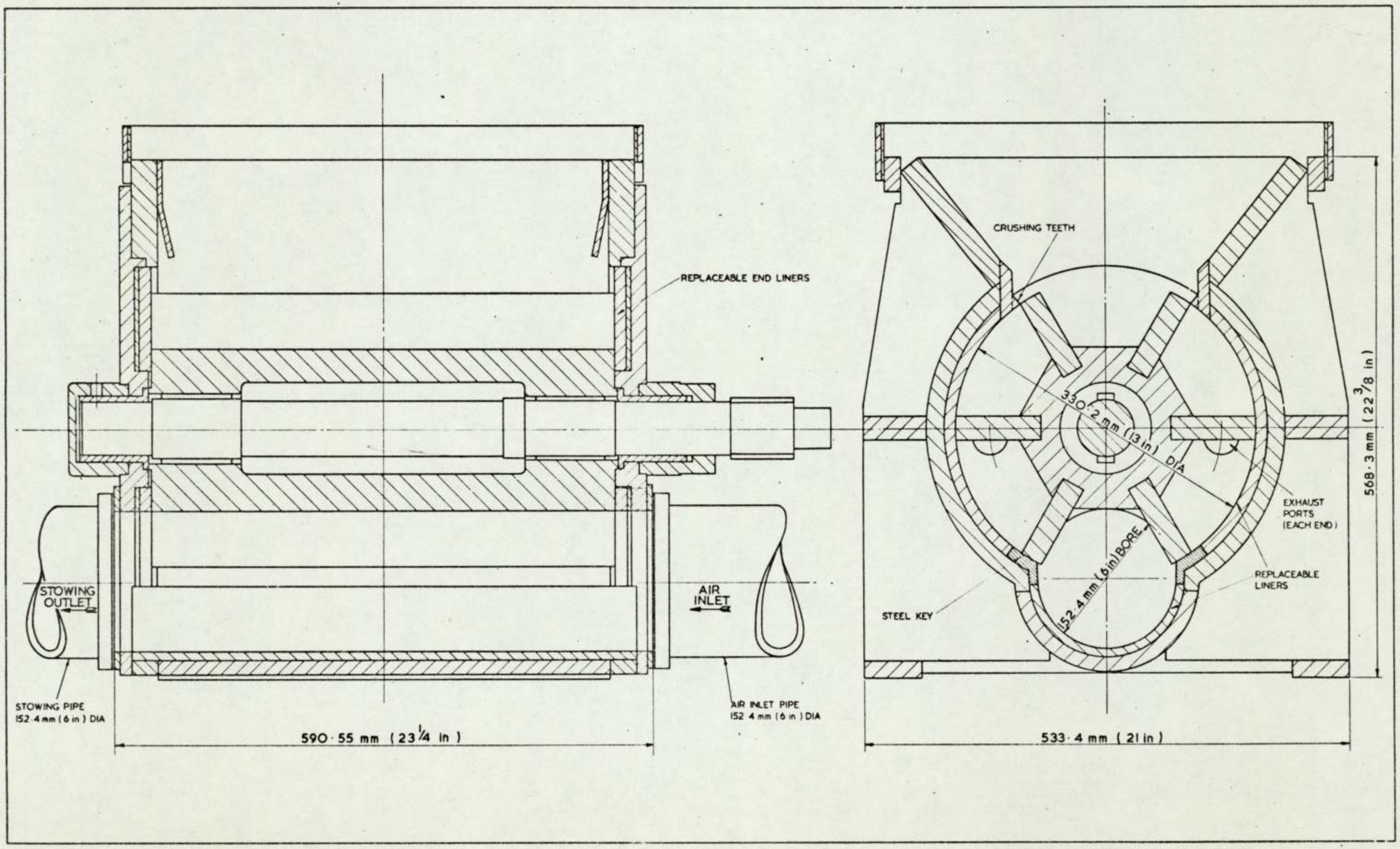


FIG 25 FINAL STOWER DESIGN (5 OR 6 PADDLE ROTOR)

CHAPTER 5

STOWER MODEL INVESTIGATIONS

5.1 STOWER MODEL

The design of the prototype stowing machine which has been discussed in the previous chapter, was the result of a development programme aimed at producing a reliable stowing unit which would have a capacity of not less than 60 tons/h (61 tonnes/h). These requirements were set against limitations on the physical size of the equipment and the air quantity that could be permitted without adversely affecting the ventilation in the mine working. Before any realistic assessment of this work could be made, it was necessary to obtain some knowledge of material movement through the stower and into the conveying pipeline, and to determine whether any further improvements were possible. A study of material behaviour within the stowing system was therefore undertaken. This involved the manufacture of a full scale model of the stower unit which formed the basis of a test rig, and which could be used to conduct tests under controlled laboratory conditions. The model was manufactured in perspex to allow observation of the material passing through the system, and to enable some of the tests to be recorded on film by using a high speed camera.

The stower unit itself was identical in internal details to the prototype stowing machine, and to complete the stowing system 6 ins (152 mm) diameter perspex pipes were used. The stower rotor consisted of a wooden hub 7 ins (178 mm) diameter fitted with five $\frac{1}{2}$ ins (13 mm) thick perspex radial paddles. The pocket shape and tip diameter of the rotor was also a facsimile of the prototype stowing machine. A running clearance of .002 to .004 ins (.05 to .10 mm) between the rotor and barrel was maintained by machining the tip and barrel diameters.

Polishing of the barrel after machining,restored the transparency of the material.

5.2 MODEL TEST RIG REQUIREMENTS

To carry out the study of material behaviour, it was necessary to construct a laboratory test rig in which the perspex stower model formed the basic component. The rig requirements were,

1. To mount the stower model complete with its horizontal stowing pipeline at a convenient height above ground level, to allow high speed filming of the material passing through the system.
2. To provide the means of feeding material into the stower model under controlled conditions.
3. To provide air to the rig and the means of controlling the air volume over a range up to 1800 ft^3/min ($0.8\ m^3/s$) free air delivery.
4. To provide a cell or hopper at the end of the stowing pipeline to receive the stowing material.
5. To include instrumentation for recording air flows, pressures, and temperatures at convenient points both upstream and downstream of the stower model.

5.3 MODEL TEST RIG CONSTRUCTION

The stower unit was mounted on a bench (Figure 26) and was driven by a 0.75 hp (5.6×10^2 W) self contained power unit comprising, motor, reduction gear, variator, and torque limiter. A chain drive was fitted between the power unit and the stower so that the power unit could be offset, providing clear access for the air inlet pipe to the stower. The inclusion of the variator provided an infinitely variable stower rotor speed up to 80 revolutions per minute.

Air to the model stower was supplied through a 6 ins (152 mm) diameter perspex pipeline which consisted of approximately 12 ft (3.6 m) of straight piping and a 90° bend, (Figure 27 and 28). The pipe bend

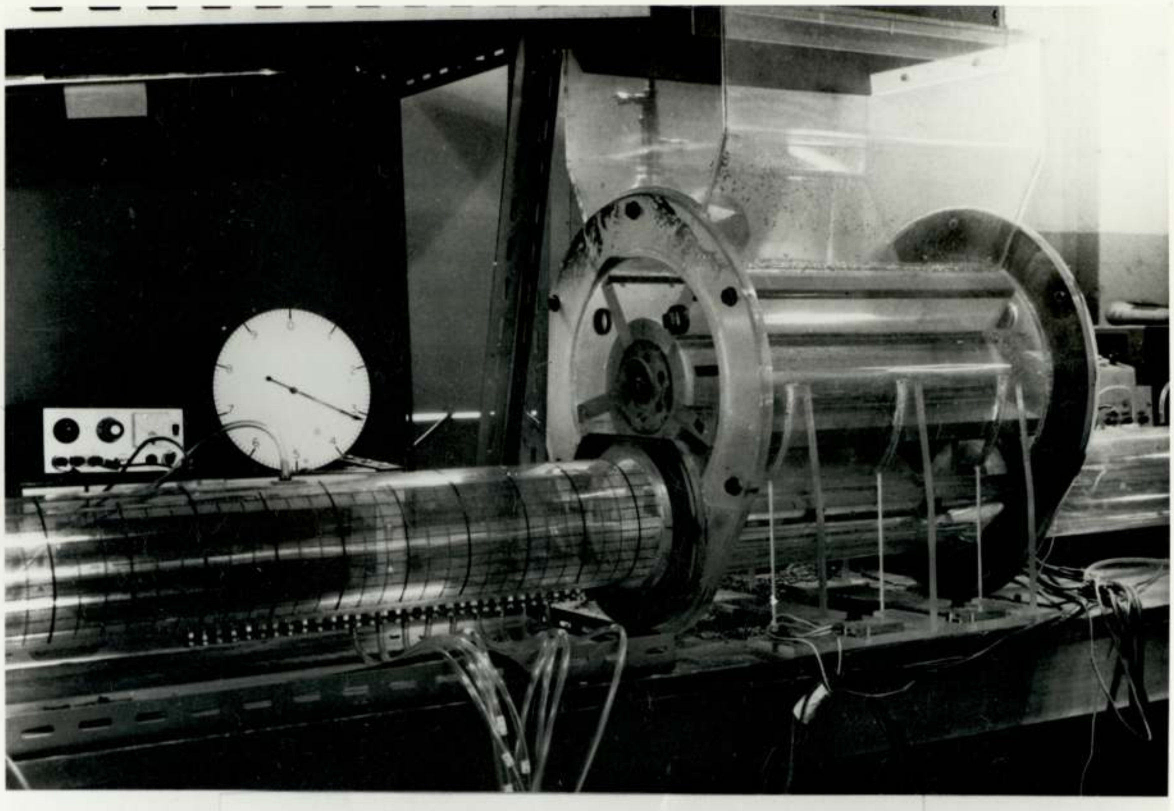


FIG 26 STOWER MODEL SHOWING OUTLET PIPE AND TIMING CLOCK

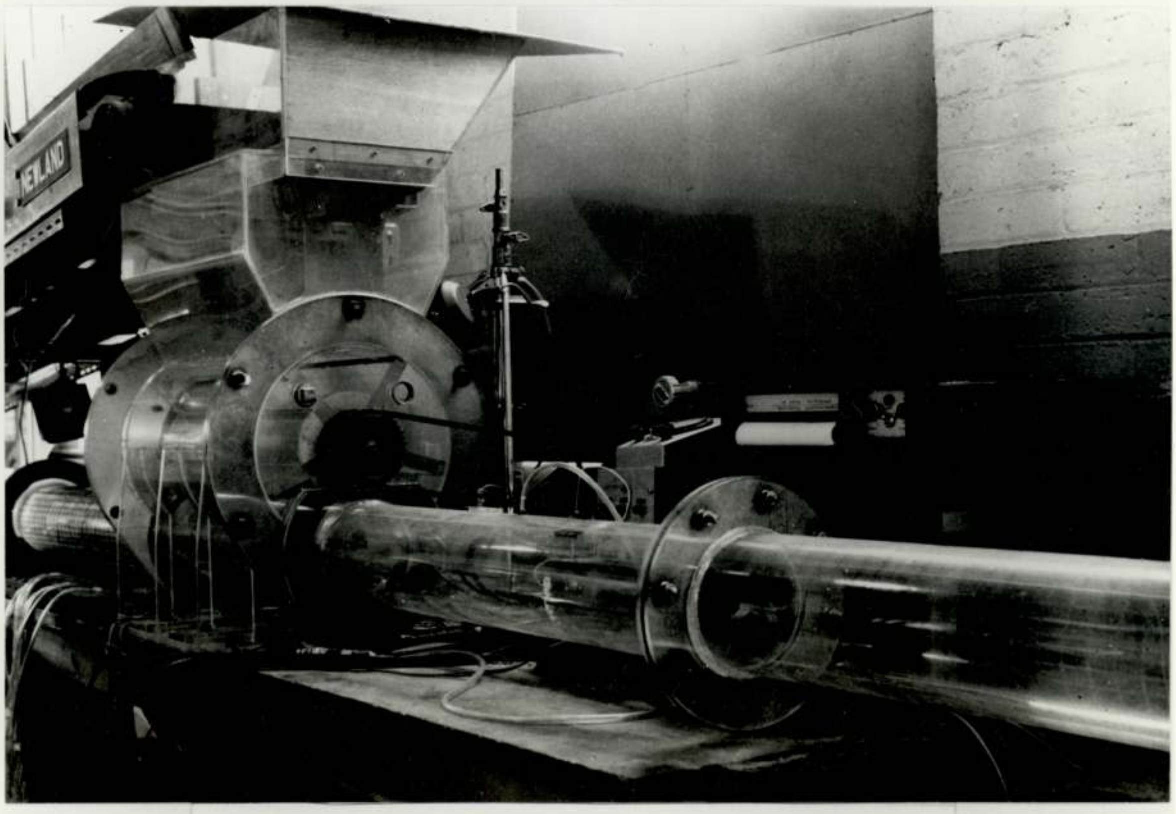


FIG 27 AIR INLET PIPE SHOWING VHF RECORDER FOR TEMPS AND PRESSURES

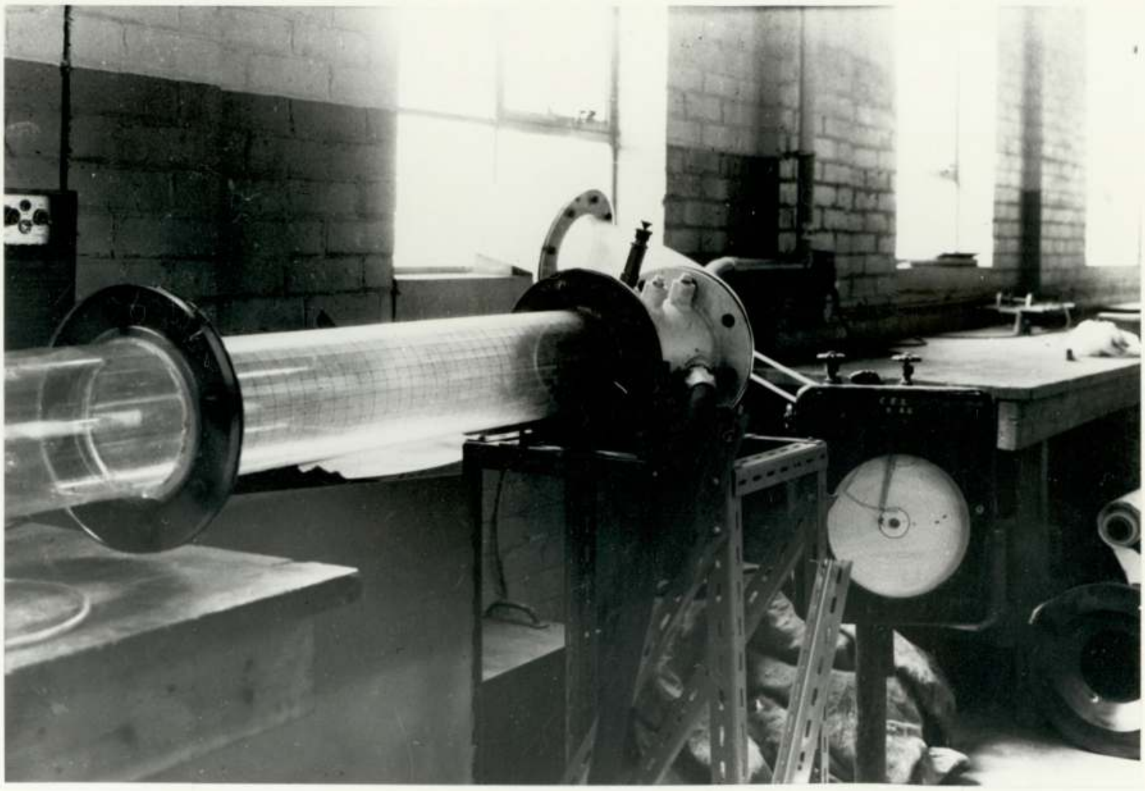


FIG 28 AIR INLET SHOWING RECORDER FOR AIR FLOWS AND PRESSURES

FIG 29 AIR SUPPLY SHOWING POSITIVE DISPLACEMENT BLOWERS AND ELECTRICAL CONTROL UNITS

was necessary to allow the inlet pipe to connect to air supply piping outside the laboratory (Figure 29). The air to the model test rig was provided by two positive displacement blowers which could be operated either independently or together. Each blower had a capacity of 1000 ft^3/min (0.5 m^3/s) free air delivery and was capable of operating at pressures up to 10 lb/in (69 x 10^3 N/m^2). A combination of steel and flexible piping connected the blowers to the perspex pipe range. Between the blowers and the model a valve was fitted in the pipework (Figure 29) to control the air supply to the test rig by bleeding off some of the air to atmosphere.

The stowing pipe range from the model stower outlet consisted of a 40 ft (12 m) length of straight 6 in (152 mm) diameter perspex piping, made up from standard 3 ft (0.9 m) lengths with flange bolted joints (Figure 30). A wooden hopper mounted on an angle iron framework was positioned at the end of the stowing pipeline to receive the material passing through the stowing system (Figure 30). The hopper was designed to receive the last pipe in the line, and a filter which was fitted in the top of the hopper allowed the air to pass through the hopper, and be ducted away to the atmosphere outside the laboratory, the stowing material being retained inside the hopper. By disconnecting the last pipe in the stowing line, the hopper could be easily raised from its supporting frame, and the material removed by means of a sliding hatch situated in its base.

To provide a controlled material feed to the model stower, a wooden container was mounted directly above the stower. One end of the container which formed a vertical sliding door was positioned over the stower hopper, and by fixing the container in an elevated position, the material was gravity fed into the hopper by raising the sliding door. This method of material feed was tried during the initial commissioning period of the test rig, but was found to be unacceptable because control

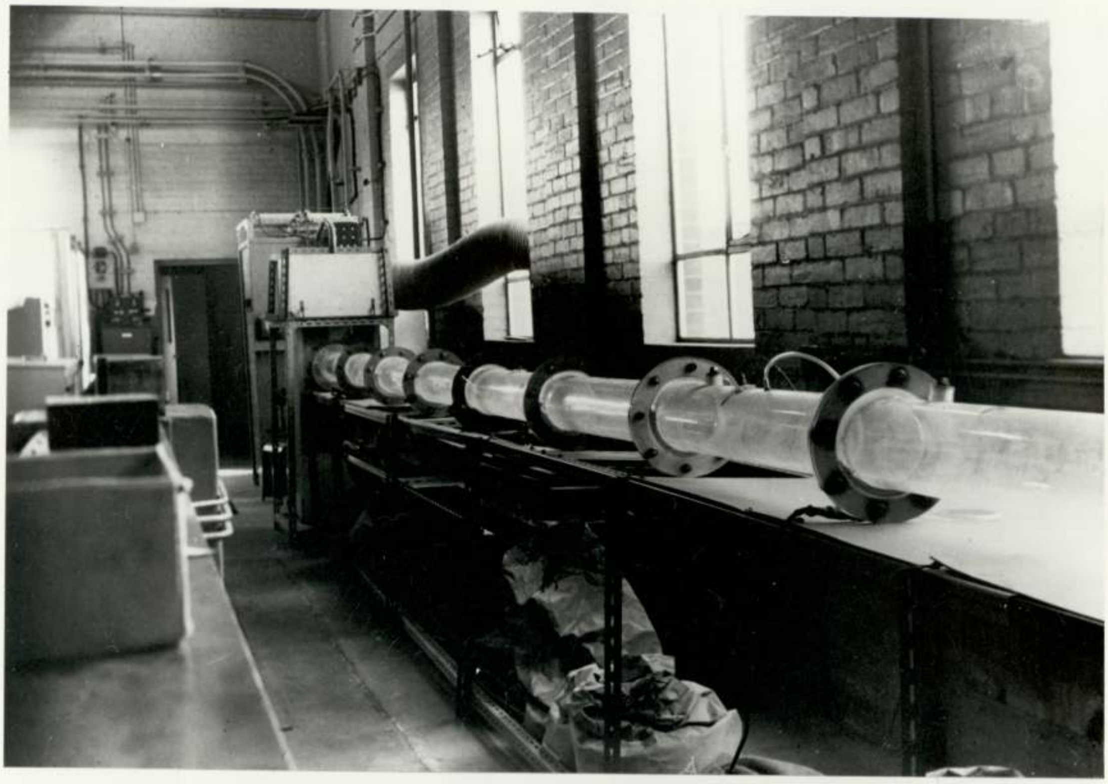


FIG 30 STOWING PIPE RANGE SHOWING RECEIVING HOPPER AND AIR OUTLET

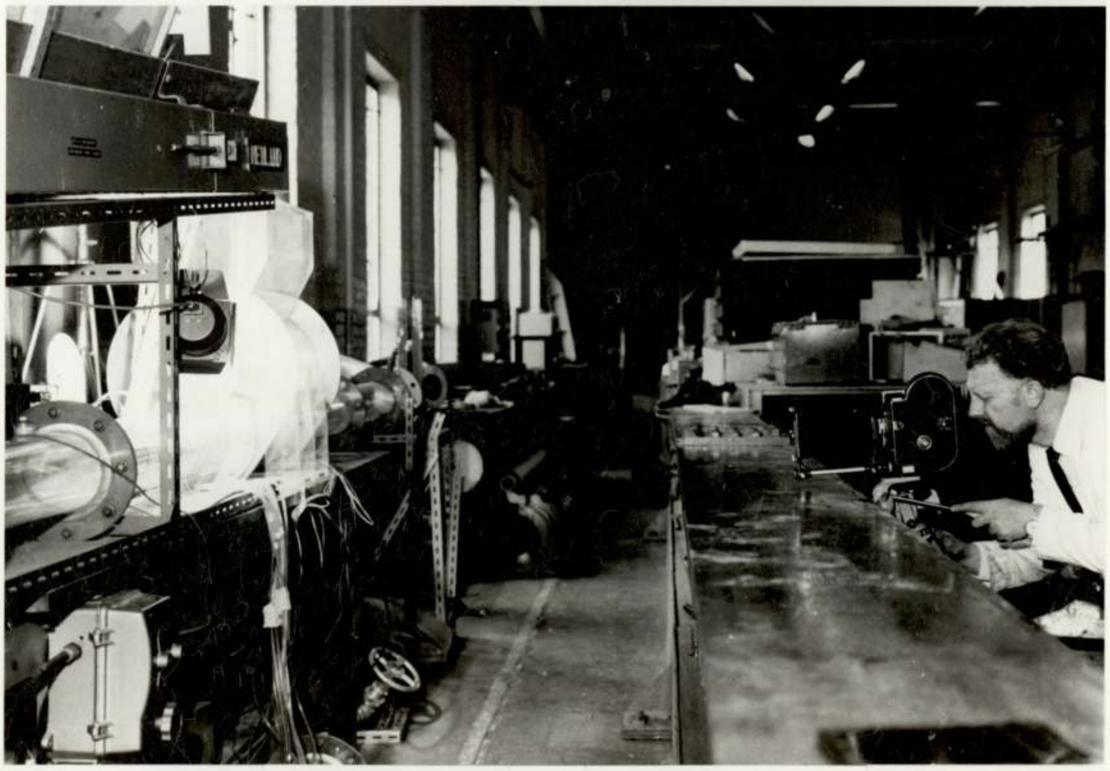


FIG 31 TEST RECORDING USING HIGH SPEED CAMERA

of the amount of material and the rate of feed into the model stower was extremely difficult. After numerous attempts to improve the method of material feed, the container was replaced by a 12 ins (304 mm) wide conveyor (Figure 32) mounted on an angle iron framework directly above and in line with the stowing pipeline. The conveyor was elevated to the stower model height at the feed end, the tail end being at a sufficient height to clear the stowing pipes passing directly beneath. The conveyor was driven by a 0.25 hp (1.9×10^2 W) electric motor through a gearbox and a 'V' belt drive. The 'V' belt drive was designed to provide a variable conveyor belt speed which could be set by an adjusting wheel at the tail end of the conveyor (Figure 33). Extension of the supporting framework at the tail end of the conveyor allowed a removable feed hopper to be suspended above the conveyor, and at the feed end a sliding gate (Figure 32) was fitted to control the bed depth of material passing into the model stower.

5.4 TEST RIG INSTRUMENTATION

During the investigations it was necessary to provide means for measuring air flows, pressures, and pipe wall temperatures. The instruments to supply this information was selected from equipment which was readily available.

5.4.1 AIR FLOW MEASUREMENT

Air flows up to 1800 ft^3/min (0.8 m^3/s) free air delivery were measured by fitting an orifice plate in the steel pipe section between the Blowers and the model stower. Provision was made for the necessary straight pipe run required on each side of the orifice plate to stabilise the air flow at the measuring point. Tappings on each side of the orifice plate measured the differential pressure which was converted to give a direct reading of air flow in ft^3/min (m^3/s). The air flows were registered on a Kent recorder which was positioned inside the laboratory alongside the model test rig (Figure 28). The instrument also measured

FIG 32 BELT FEED CONVEYOR TO STOWER

FIG 33 CONVEYOR TAIL END SHOWING VARIABLE BELT SPEED CONTROL

the air pressure and both readings were recorded on a chart having a linear scale.

By positioning the air bleed valve upstream of the orifice plate position, a range of air flows up to the combined capacity of the two blowers could be selected and recorded by the Kent recorder.

5.4.2 PIPE PRESSURE AND TEMPERATURE MEASUREMENTS

In addition to the air flow measurements, the pressure and temperature condition in the pipes before and after the model stower was considered necessary. During the initial trial runs with the rig, pressure readings using mercury manometers were tried but the comparatively low pressures (below 4 lb/in^2 (28×10^3 N/m^2)) were difficult to obtain with reasonable accuracy. It also became obvious that pressures and temperatures would need to be recorded automatically during the tests that were to take place.

The method of recording the pressures, which was adopted throughout the laboratory tests, consisted of connecting $\frac{1}{4}$ in (6.4 mm) bore plastic tubing from selected tapping points in the pipeline. The tubing was connected to a pressure transducer to convert the air pressure into an electrical signal which was transmitted to a very high frequency recorder (V.H.F. Recorder) (Figure 27). From the signal the pressure was recorded directly on to a chart incorporated in the recorder.

Temperature recordings were also made on the same chart by means of thermocouples which were fitted through the pipe wall and flush with the inside diameter of the pipe.

Pressure and temperature tappings were made at points in the pipeline approximately 15 ins (381 mm) each side of the stower model, and at a position downstream 25 ft (7.6 m) from the stower.

5.5 MODEL TEST RECORDING

To carry out a significant study of the material behaviour it was necessary to make provision for recording on film the events of some of the tests to be carried out. To do this a reflex camera was set up so that the model stower and first 3 ft (0.9 m) of stowing pipe could be filmed (Figure 31). The film was taken at a speed of 64 frames/sec which was the maximum speed at which the camera could operate, and by using a variable shutter the maximum magnification was obtained with this equipment.

To indicate the time factor during each test, an electrically driven rev. indicator was mounted above the outlet stowing pipe within the range of the camera vision (Figure 26). The periphery of the clock dial was graduated into tenth and hundredth digits, and the speed of the indicator finger could be set by means of a variable voltage controller in the minature power pack used to drive the clock.

An analysing projector having a variable projector speed was used to study the test results in detail.

5.6 STOWING MATERIAL REQUIREMENTS

The use of perspex as the material of which the model stower and stowing pipes were manufactured presented difficulties when the selection of stowing materials was considered. The stower model had been produced as a full scale replica of the prototype stower unit, and ideally the model tests should have been carried out with the same stowing materials as that used during the prototype stower trials. This was not possible for the following reasons:

1. Mining debris, which can include Shale, Mudstone, Fireclay and Sandstone, is very abrasive, and would destroy the transparency of the model and piping.
2. The mechanical properties of mining stone was such that any pieces jamming between the rotor blades and the stower barrel would severely damage the model.
3. The general dirty condition of mining debris would have reduced the transparency of the perspex and obsured the study of material behaviour, besides causing a severe dust problem inside

the laboratory.

The search for suitable stowing material for the model test rig was therefore based on the following requirements:

(i) The size of material to be up to approximately 2 in (51 mm)

(ii) The abrasiveness of the material to be such that it would not scratch or mark the perspex and impair the study of material behaviour.

(iii) The strength of the material to be such that if pieces jammed inside the stower, they would either break or allow the torque limiter to become dissengaged without damage to the perspex stower barrel or rotor blades.

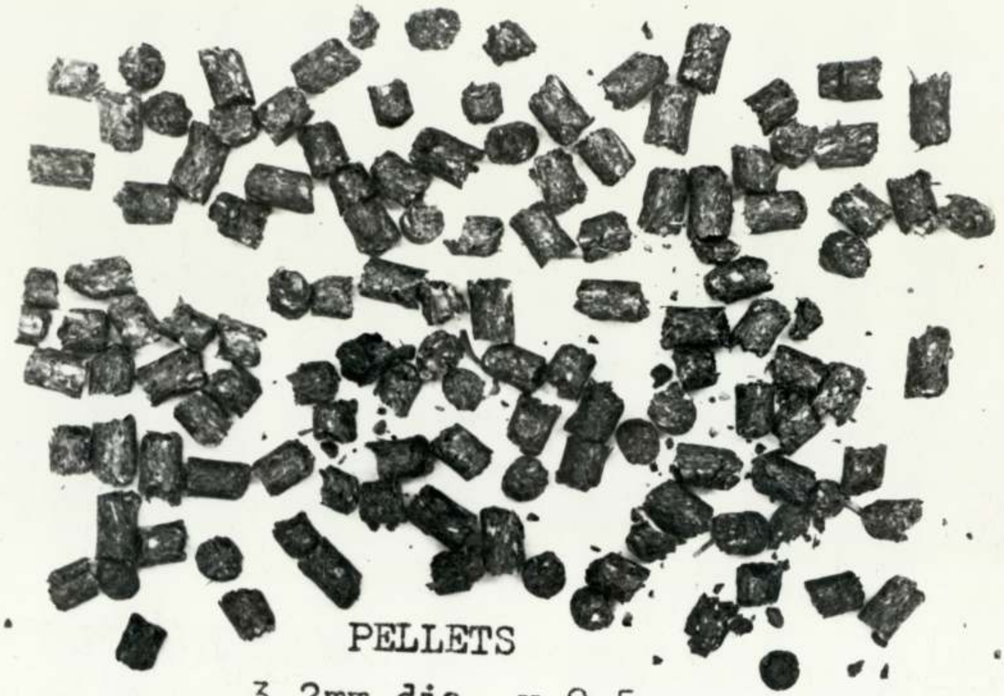
(iv) The selection of several materials of different shapes and densities. The densities being over a range between 45 and 90 lb/ft^3 (721 and 1442 kg/m^3) this being similar to materials used during the prototype stower trials.

The use of rubber and plastic materials were investigated without success, but eventually the materials which came nearest to the requirements were found to be animal foods. The selection of this type of material also offered the advantage of using some materials having known shape and size.

5.7 STOWING MATERIAL SELECTION

Five different materials were finally selected for the model test rig investigations, these were

1. Pellets (Figure 34) $\frac{1}{8}$ in diameter x $\frac{3}{8}$ in long (3.2 mm diameter x 9.5 mm long. Density 44.9 lb/ft^3 (720 kg/m^3).
2. Pencils (Figure 35) $\frac{3}{8}$ in diameter x $\frac{1}{2}$ in long (9.5 mm diameter x 12.7 mm long). Density 80.4 lb/ft^3 (1285 kg/m^3).
3. Pencils (Figure 36) $\frac{1}{2}$ in diameter x $\frac{3}{4}$ in long (12.7 mm diameter x 19 mm long. Density 74.8 lb/ft^3 (1200 kg/m^3).
4. Flat Maize (Figure 37) $\frac{5}{16}$ in diameter x $\frac{5}{32}$ in thick (8 mm diameter

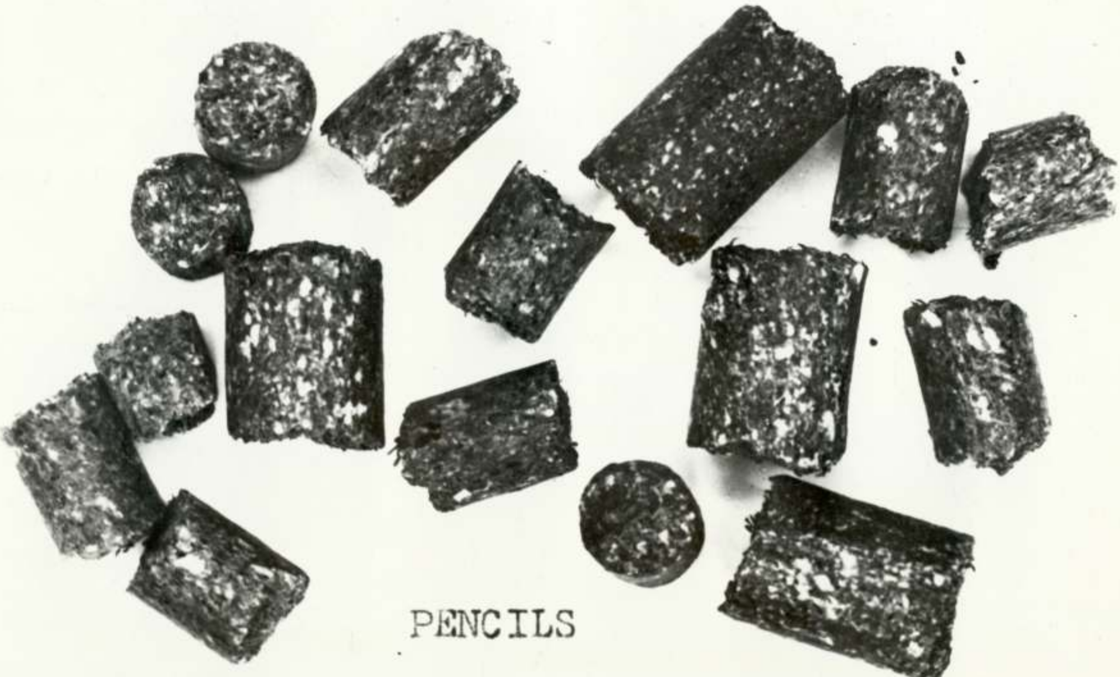


PELLETS

3.2mm dia. x 9.5mm

($\frac{1}{8}$in dia. x $\frac{3}{8}$in)

FIG 34 MATERIALS FOR STOWER MODEL INVESTIGATIONS



PENCILS

9.5mm dia. x 12.7mm

($\frac{3}{8}$in dia. x $\frac{1}{2}$in)

FIG 35 MATERIALS FOR STOWER MODEL INVESTIGATIONS

PENCILS

12.7mm dia. x 19mm

($\frac{1}{2}$in dia. x $\frac{3}{4}$in)

FIG 36 MATERIALS FOR STOWER MODEL INVESTIGATIONS

FLAT MAIZE

8mm dia. x 4mm

(5/16 in dia. x 5/32 in)

FIG 37 MATERIALS FOR STOWER MODEL INVESTIGATIONS

x 4 mm thick). Density 84.2 lb/ft^3 (1346 kg/m^3).

5. Nut Flakes (Figure 38) $\frac{1}{4}$ in (6.3 mm) thick approximately, variable size up to a maximum of 3 ins (76 mm). Density 89.2 lb/ft^3 (1425 kg/m^3).

5.8 MODEL TEST RIG COMMISSIONING

On completion of the test rig it was necessary to carry out some initial trial runs to establish the functioning of the equipment and to organise a system for carrying out each test in the most efficient manner.

5.8.1 MATERIAL LOADING AND RECEIVING

Initially two identical hoppers were built, one for receiving the material at the end of the stowing pipeline, and the other for feeding the material at the tail end of the feed conveyor. Several test runs were made with this system but it was found that the time between finishing one test and starting the next was too long if the proposed test programme was to be completed in the limited time available. Interchanging of the hoppers involved the use of a hand controlled overhead crane, this being operated the full length of the rig each time a change took place. Eventually it was found much quicker to load the conveyor by hand and to use only the receiving hopper at the end of the pipeline. Sufficient material was obtained to allow a number of tests to be carried out before it was necessary to empty the receiving hopper.

5.8.2 AIR FLOW CALIBRATION

A separate electrical control unit was supplied for each of the blowers supplying the air to the model test rig. Each blower had a separate control switch inside the laboratory adjacent to the rig. To provide air flows up to the capacity of the two blowers, two pipe connections (Figure 29) were used to bleed air from the system to atmosphere. A valve was fitted at the end of one of the pipe connections,

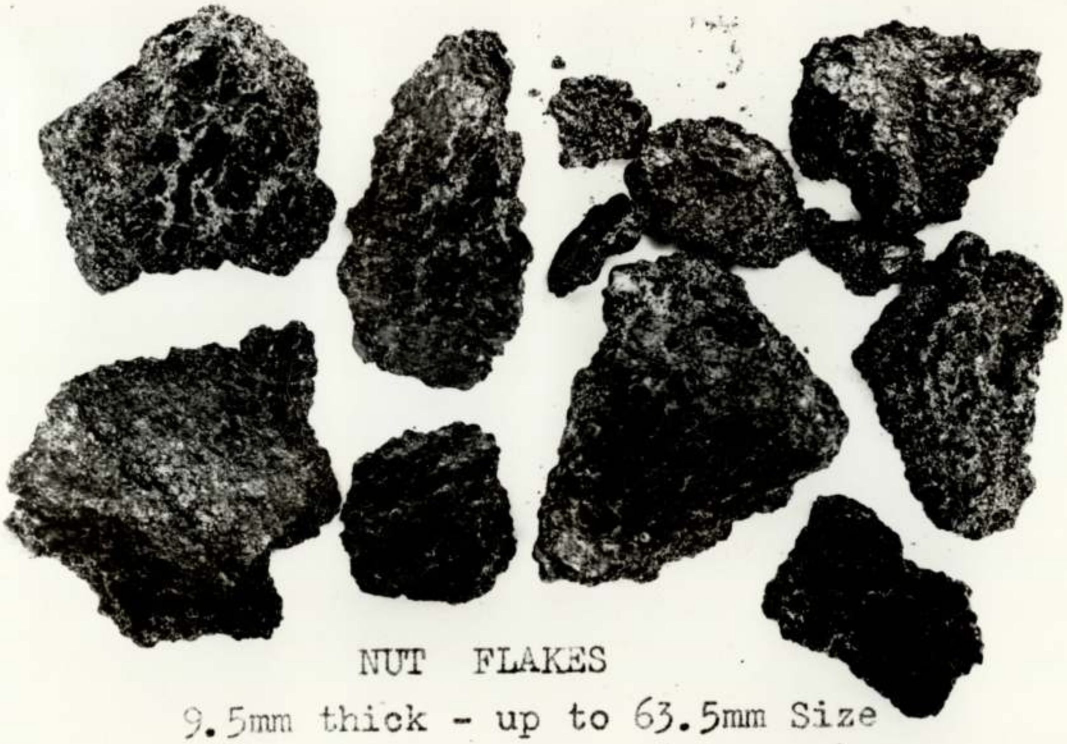


FIG 38 MATERIALS FOR STOWER MODEL INVESTIGATIONS

and bolted onto the other connection was a blanking plate incorporating replacable plugs.

Tests were carried out to obtain the combination of valve and blanking plate setting for air flows between 500 and 1,800 ft^3/min (0.25 and 0.8 m^3/s) free air delivery. This calibration made it possible to pre-set the pipe conditions for any required air flow. The air flow recorder chart had linear scales of

(i) 0 to 2,000 ft^3/min (0 to 1 m^3/s)

(ii) 0 to 50 lb/in^2 (0 to 344 x 10^3 N/m^2)

5.8.3 INSTRUMENT CALIBRATION

The air pressures and temperatures at the pipe wall were recorded on a very high frequency recorder (Figure 27). The recorder chart which was 7 ins (178 mm) wide, printed the three temperatures and pressures directly on a linear scale. The calibration for the temperature and pressure readings were:

1. P_1 (Before Stower)	10 lbs/in^2 = 5 in (69 x 10^3 N/m^2 = 127 mm)
2. P_2 (After Stower)	
3. P_3 (Stowing Pipeline)	
4. T_1 (Before Stower)	18.75°C = 1.0 in (25.4 mm)
5. T_2 (After Stower)	18.75°C = 0.9 in (22.8 mm)
6. T_3 (Stowing Pipeline)	18.75°C = 1.3 in (33.8 mm)

Any fluctuations in the temperature readings on the chart gave the direct fluctuation in air temperature at the pipe wall, the zero setting representing the ambient air temperature.

5.9 STOWER MODEL INVESTIGATIONS

The objectives of the stower model investigations were,

1. To investigate the effects of air flow on material movement from the stower, and to establish the optimum rotor speed for maximum stowing output.
2. To observe and establish the flow pattern at the stower and in the

pipeline downstream and to predict the minimum air flows necessary to avoid material build up and pipe blockages.

5.9.1 MATERIAL MOVEMENT FROM STOWER ROTOR

During the development of the prototype stowing machine the operating speed of the stower rotor was selected at approximately 60 revolutions per minute. This speed was found to produce the best results in terms of stowing output, the tonnage rates obtained being in excess of the original requirements. It was not possible with the prototype stower test rig to carry out any detailed study to establish the optimum rotor speed by examining the factors which limit the output.

The capacity of the stowing unit is dependant on the efficiency of conveying the material through the stower and into the conveying pipeline. This efficiency can be dependant on any one or combination of a number of factors, which are;

1. The volume of material metered per rotor pocket.
2. The weight of material per rotor pocket.
3. The size and shape of the material.
4. The moisture content and stickiness of the material.
5. The air velocity.

Investigations by Broadhurst(9) on the significance and effects of moisture content in pneumatic stowing materials is discussed in chapter 6.

Using the different materials mentioned previously, a programme of tests were carried out with the stower model to measure the rates at which these materials were accelerated from the stower rotor pockets. For each test, known volumes of material were loaded into a rotor pocket at the stower hopper, and film recordings were made of the material being fed into the air stream and conveyed out of the pocket into the conveying pipeline. By observing the time indication, the times to clear the rotor pocket of material was obtained. With the aid of an analysing projector it was possible to determine the time factors to an accuracy of about

0.002 seconds.

Observations were made for each of the selected materials for volumes which varied between that of a single piece and that of a full rotor pocket. For these ranges of material volumes tests were carried out for a number of air flows up to 1800 ft^3/min (0.8 m^3/s) free air delivery.

The recorded results of these tests are given in detail in Appendix 2. The test results which have been analysed and presented in graphical form, are now discussed.

5.9.2 OPTIMUM ROTOR SPEED

Figures 39, 40, and 41 show the times to clear the rotor pocket for weights of material representing single pieces up to a full rotor pocket. The maximum material weight plotted on each curve represents that of a full rotor pocket. The available time to clear a rotor pocket for any given rotor speed has also been plotted for a range of rotor speeds up to 70 R.P.M. By equating the available time to that required to clear the rotor pocket, the optimum rotor speed can be obtained. For example, referring to Figure 39,

Air Flow 1100 ft^3/min (0.5 m^3/s)

Pocket clearance time for full pocket of flakes is 0.4 seconds

From the "Available Time" curve the rotor speed for 0.4 seconds is 30 R.P.M.

This rotor speed of 30 R.P.M. is the optimum speed for metering the maximum weight (Full pockets) of Flakes. Any speed in excess of this would result in failure to clear the pocket of material and would in fact reduce the overall output.

From the curves shown in Figures 39, 40, and 41 the optimum rotor speeds for full pockets of material has been plotted (Figure 42) for a range of air flows between 1100 and 1800 ft^3/min (0.5 and 0.8 m^3/s). For an air flow of 1800 ft^3/min (0.8 m^3/s), the optimum rotor speed varies for the different materials between 43 R.P.M. for the Pellets

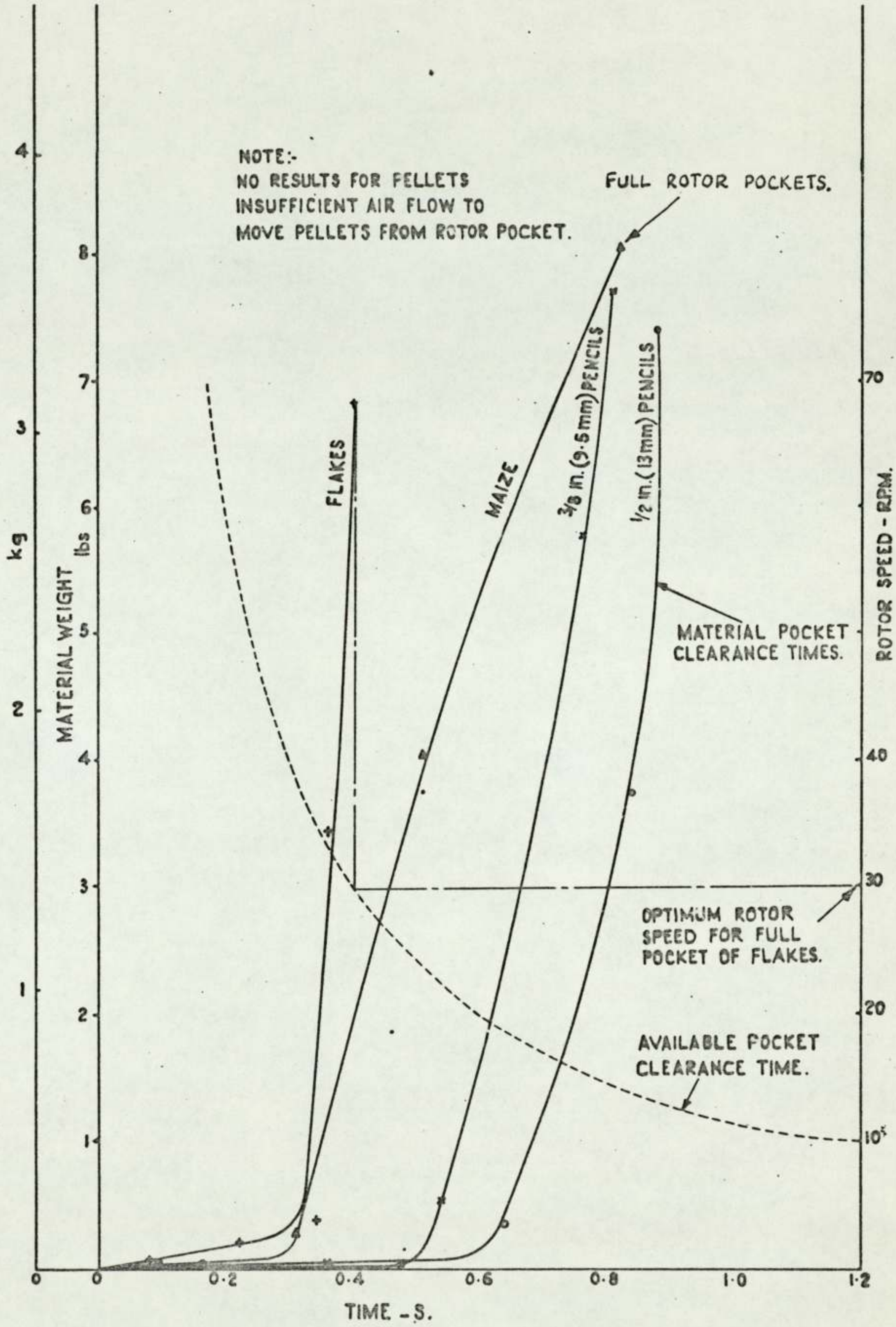


FIG. 39 ROTOR POCKET CLEARANCE TIMES - AIR FLOW 1100 ft^3/min. (0·5 m^3/s)

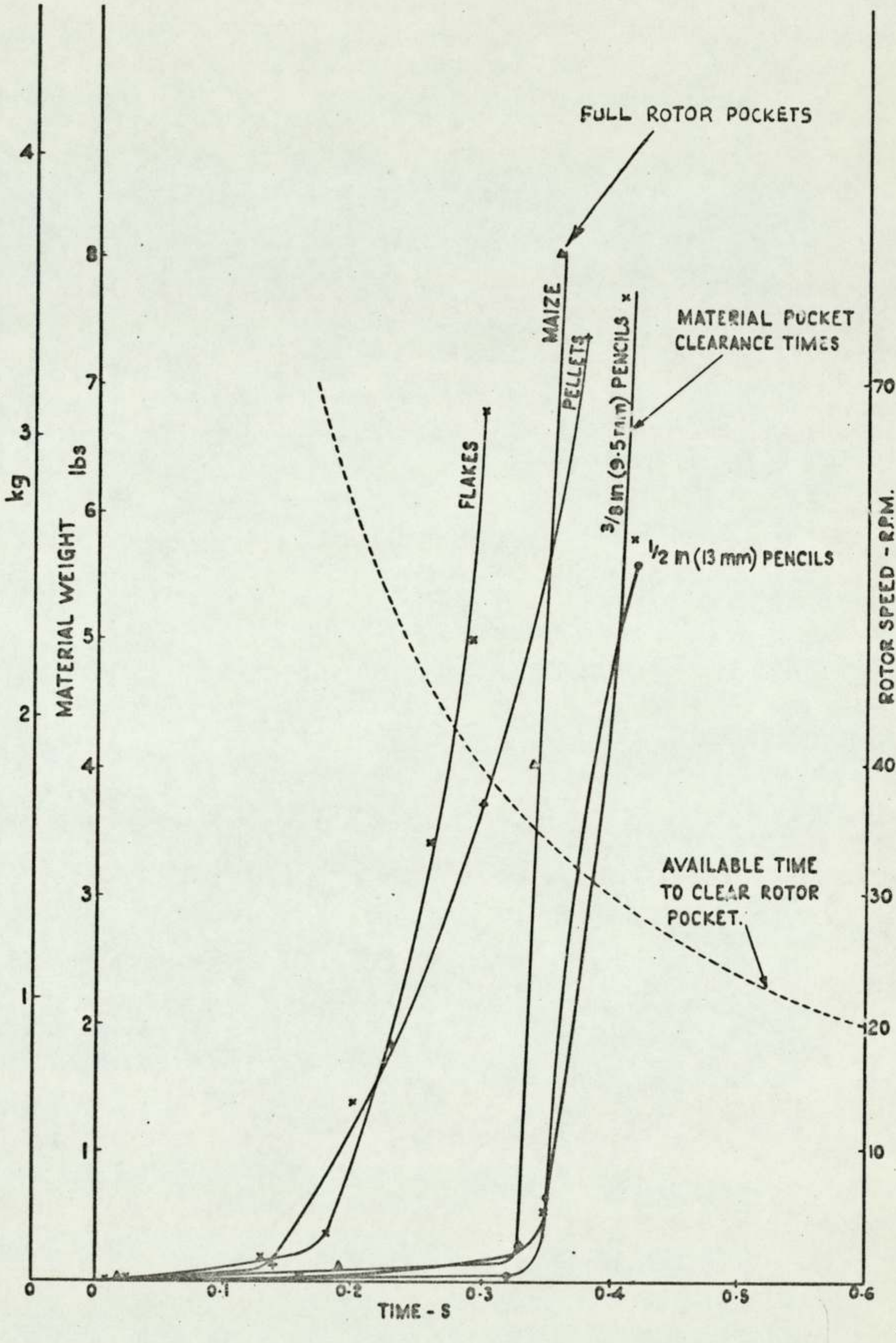


FIG. 40 ROTOR POCKET CLEARANCE TIMES - AIR FLOW 1350 ft.^{3}/min (0·63 m^3/s)

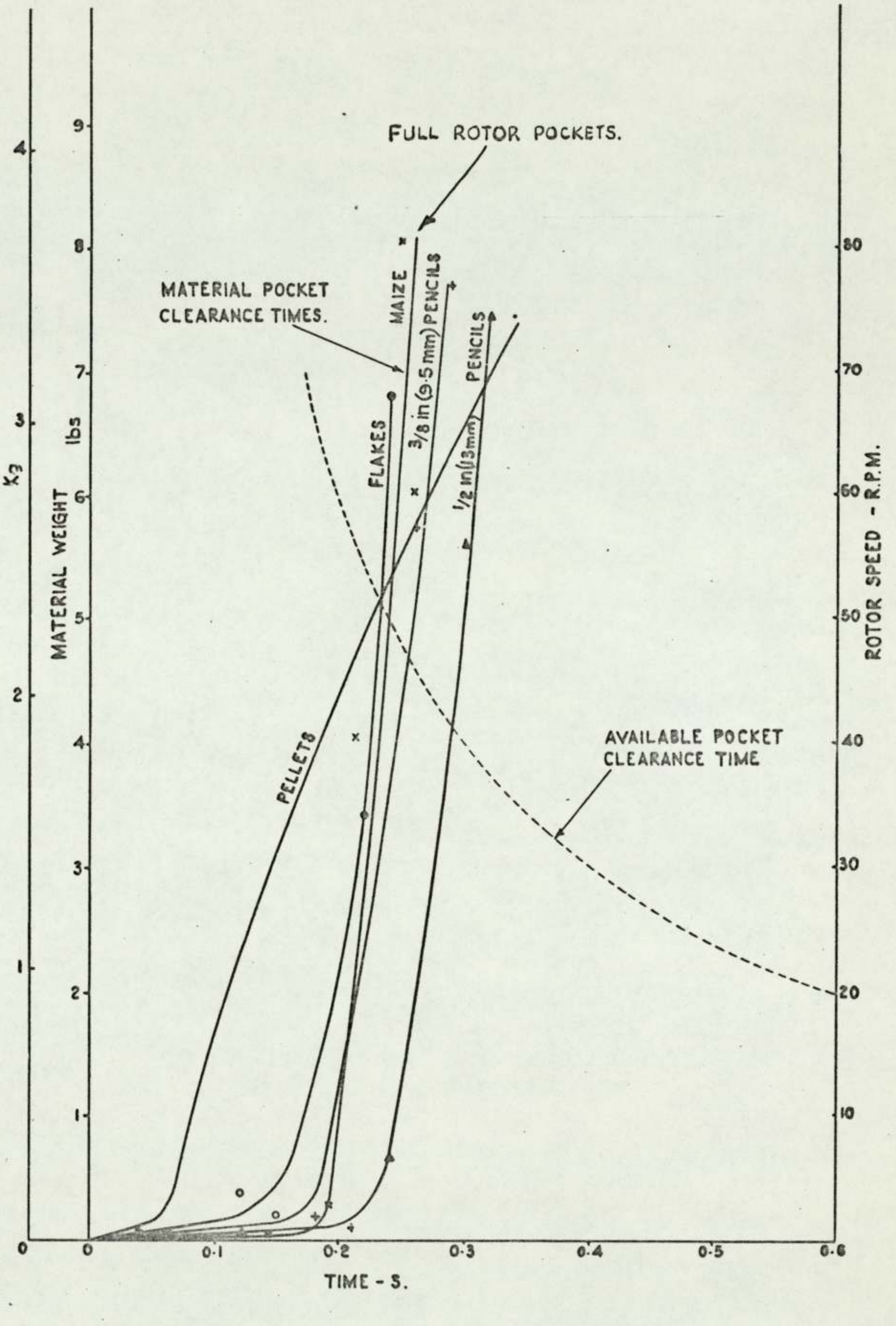


FIG 41 ROTOR POCKET CLEARANCE TIMES. AIR FLOW 1600 ft.^{3}/min.(0·75 ^{3}/s)

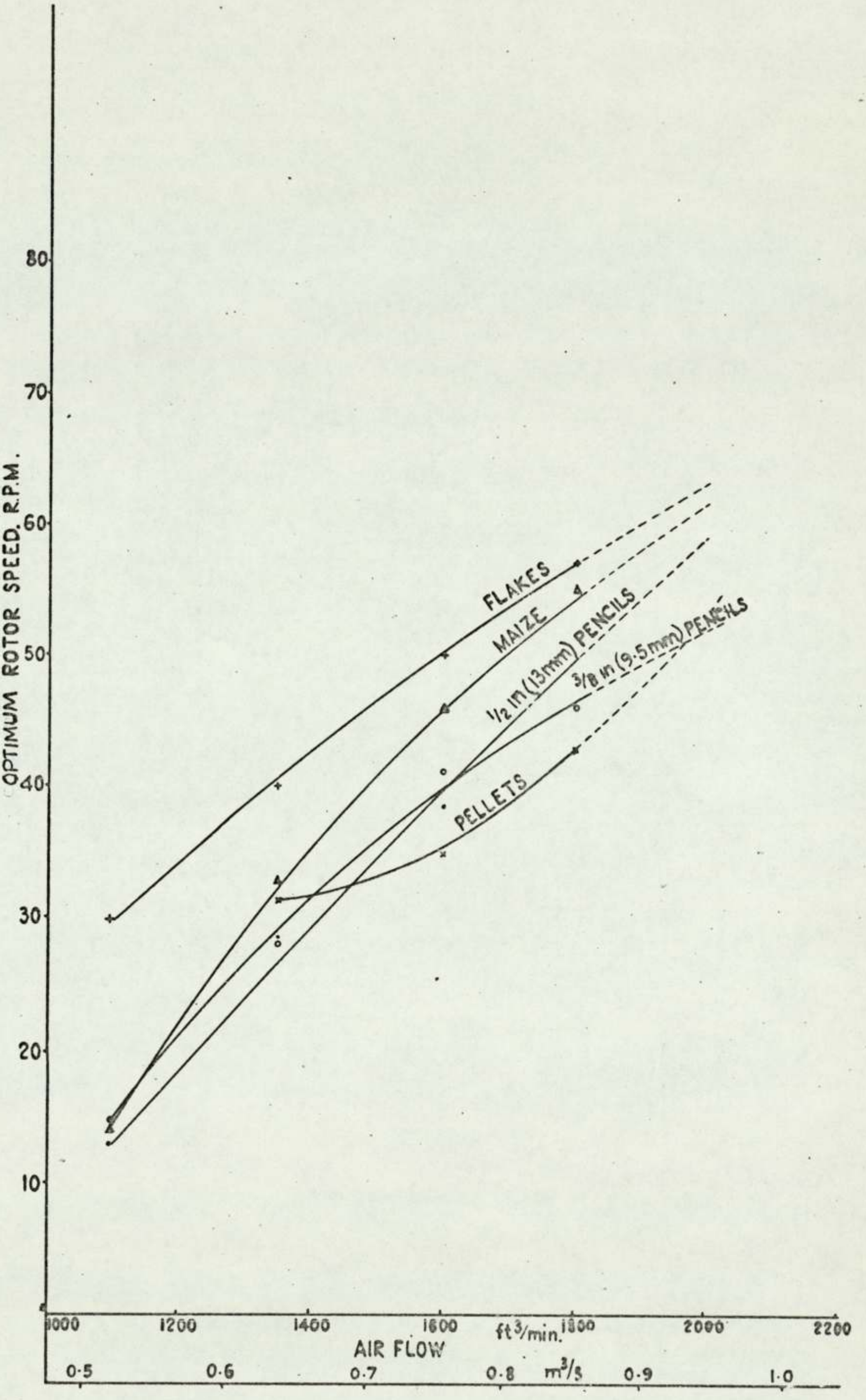


FIG. 42 OPTIMUM ROTOR SPEEDS FOR MAXIMUM MATERIAL WEIGHTS (FULL POCKETS)

and 57 R.P.M. for the Flakes. Tests to determine the optimum rotor speeds for air flows in excess of 1800 ft^3/min (0.8 m^3/s) were not carried out, but if the curves showing the rates of increase in optimum rotor speed against air flow continued as in Figure 42, it is predictable that the optimum rotor speed for air flows around 2200 ft^3/min (1 m^3/s) is within the range of 58 to 68 R.P.M.

Taking into account that run of mine debris because of its moisture content and greater specific weight would move along the pipe away from the stower at a lower acceleration rate than the materials used in the model tests, then the optimum rotor speed for run of mine debris will be in a lower range than that obtained from the model experiments. A more detailed study of material movement from the stower using mining debris could provide a more accurate assessment of the optimum rotor speed, the results of the model tests does confirm that the speed of 60 R.P.M. used during the final prototype stowing trials is of the right order.

Because of the wide variation in the nature, size and shape of run of mine debris it is very likely that further detailed investigations on optimum rotor speeds would not provide information which was any more conclusive than that obtained from the work carried out with the stower model.

5.9.3 MATERIAL BEHAVIOUR

Material blockages within the stower and first 6 ft (1.8 m) of stowing pipe have been the major cause of failure in stowing installations. The development of the prototype stower described in the earlier chapters resulted in the elimination of the blockages, but though a remedy for the failures was apparently found, the way in which blockages occur was not clearly understood. To obtain some knowledge of this a series of tests were carried out.

The tests involved feeding material through the stower into the stowing line and observing the behaviour of the material during its movement from the stower into the pipeline. Each of the different types of material described in paragraph 5.7 were used, and for each material, tests were carried out for a range of air flows between 800 and 1700 ft^3/min free air delivery, (0.4 to 0.8 m^3/s). To analyse the material movement with reasonable accuracy, films were taken of material flow at the stower outlet and at the other observation point 25 ft (7.6 m) downstream from the stower. For each test, measured volumes of material were loaded into the rotor pockets, and flow characteristics for different air flows through the system were observed.

5.9.4 FLOW CHARACTERISTICS

To identify the type of flow conditions observed during these investigations, the test results are analysed by relating the flow patterns to three basic types of flow conditions which can occur. These are,

1. SALTATION (Fig. 43A)

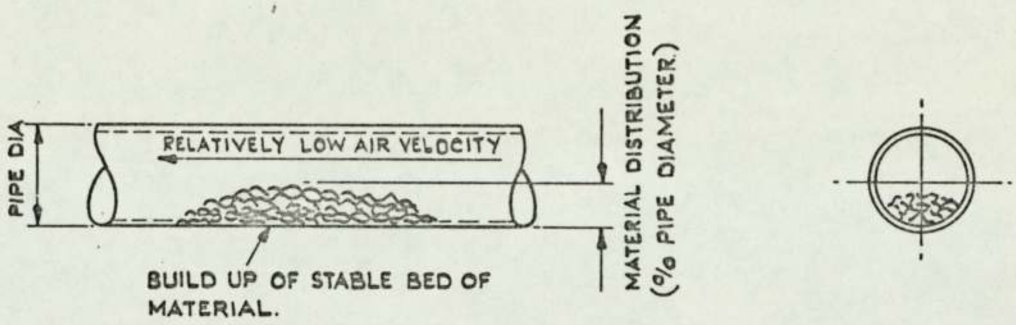
At relatively low air velocities and material feed rate, material settles on the bottom of the pipe with small particles moving over the material bed in short hops. A drop in the air velocity results in a build up of the stable bed of material which eventually results in the blockage of the pipe. This state is known as "SALTATION" and is defined by Bagnold(19).

2. PSEUDO GAS STATE (Fig. 43B)

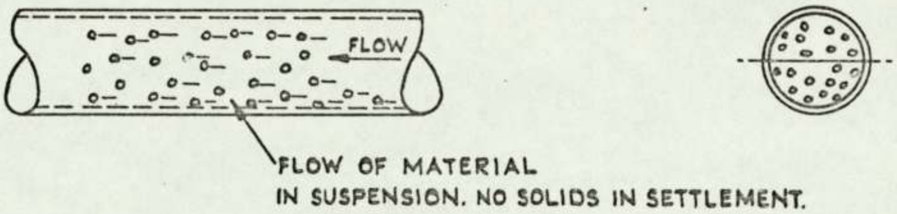
At high air velocities material is distributed evenly across the section of the pipe. The particles move at high speed and no material bed is formed. This flow condition is known as "PSEUDO GAS STATE".

3. INTERMEDIATE STATE (Fig. 43C)

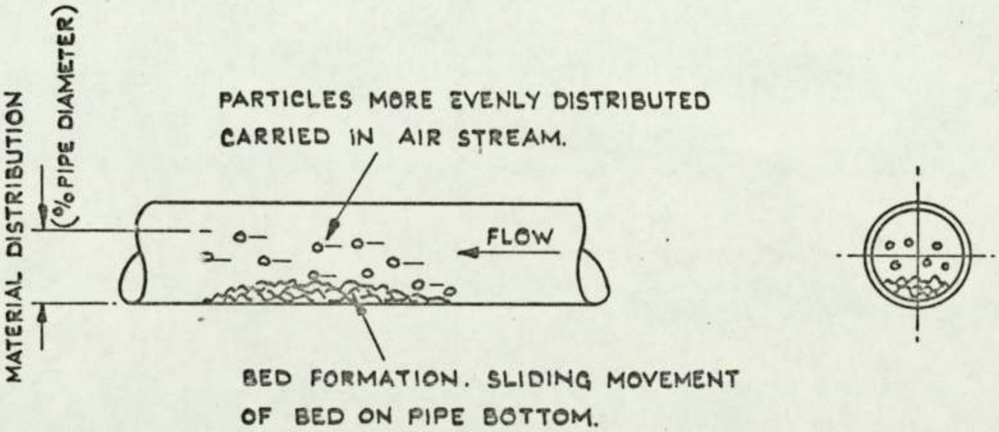
At moderate air flows solids move along the lower half of the pipe with varying solid concentration. The smaller particles are more evenly distributed and are carried in the air stream. In the "INTERMEDIATE STATE" some of the solids settle in the pipe bottom and form beds which



(A) SALTATION



(B) PSEUDO GAS STATE.



(C) INTERMEDIATE STATE

FIG. 43 FLOW CHARACTERISTICS.

slide along the pipe.

To establish the flow characteristics in terms of those defined above, the pipe diameter at the two observation points was divided into ½ in (13 mm) vertical divisions. Each division was marked along the pipe length, and by observation, material distribution in the pipe and the flow characteristics were assessed. With the equipment available it was not possible to measure the proportions of the material which were in settlement and in suspension. It is accepted that this method of assessment is open to criticism as to its accuracy. It is however considered of sufficient value to predict the possibility of material settlement and pipe blockages in 6 in (152 mm) diameter pipes with air flows significantly below 2000 ft^3/min (0.9 m^3/s).

5.9.5 AIR FLOW INVESTIGATIONS - OBJECTIVES

The significance of establishing whether air flows below 2000 ft^3/min (0.9 m^3/s) would be sufficient for stowing operations in 6 in (152 mm) diameter pipes lies in the appreciation of the financial economics of pneumatic stowing operations underground.

The need for pneumatic stowing equipment which is both reliable and of sufficient size to operate in the restricted space underground has already been discussed previously, but the essential factor which will finally determine whether pneumatic stowing operations can be usefully employed is the capital cost of the equipment. At the present time the use of a compressor delivering 2250 ft^3/min (1.1 m^3/s) free air delivery for dirt disposal operations rules out pneumatic stowing on economic grounds apart from the serious problem of physical size. A major saving in costs is possible if a Rootes type blower could be used. It has already been established during the prototype stower trials that pressures up to 15 lbs/in^2 (103×10^3 N/m^2) are sufficient for the stowing circuit which was 60 yds (55 m) long. Of equal importance is the air volume which a blower would have to deliver. Air flows substantially below 2000 ft^3/min (1.00 m^3/s) would not only make the air requirements within the scope of a small blower, but would contribute considerably to reducing the size of the equipment, and cause the minimum of interference with the underground ventilation system.

5.9.6 MATERIAL FLOW OBSERVATIONS

Over the range of air flows selected, observations of material behaviour were carried out for full rotor pockets of material and for material volumes which just covered the bottom surface of the rotor pocket. The movement of full pockets of material is of prime importance as far as material stowing rates are concerned, but tests with smaller material volumes were carried out to observe what effects this had on material flow characteristics.

The film recordings of all the tests carried out are shown in Appendix 2. From all the test results an assessment was made of the distribution of material in the pipe, and the percentage of material seen to be in settlement. The observations are tabulated in Figures 44 to 47.

In the tables (Figures 44 to 47) the two volumes for each material and air flow combination represents the full rotor pocket and covered rotor pocket conditions. An assessment of the flow characteristics are shown and at each of the observation points selected, the condition of flow in terms of those defined previously (5.9.4) is given.

5.9.7 MATERIAL DISTRIBUTION

To assess the air flows required for stowing operations it is the flow characteristics for maximum output which is significant. The analyses of the model tests has therefore been mainly concentrated on the results obtained with full rotor pockets of material.

To clarify the effect of air flow on material movement, the assessment of material distribution shown in Figures 44 to 47 has been presented graphically in Figures 48 and 49. The proportions of unit volume of

MATERIAL	VOLUME		DENSITY		AIR FLOW		AIR VELOCITY		STOWER OUTLET			25 ft. (7·6 m) DOWNSTREAM		
	in^3	cm^3	lb/ft^3	kg/m^3	ft^3/min	m^3/s	ft/s	m/s	MATERIAL DISTN. % PIPE DIA	FLOW CHARACTERISTICS	CONDITION OF FLOW	MATERIAL DISTN. % PIPE DIA	FLOW CHARACTERISTICS	CONDITION OF FLOW
FLAKES (FIG. 38)	7·26	119	89·2	1429	800	0·4	67·9	$5{\cdot}74 \times 10^{-3}$	40	BED FORMATION MOVEMENT OF ODD PIECES. NO SUSPENSION (FIG. 82)	SALTATION	25	SETTLEMENT OF ODD PIECES IN PIPE BOTTOM (FIG. 82)	SALTATION
— " —	132	2160	— " —	— " —	— " —	— " —	— " —	— " —	65	SLIDING BED MOVEMENT ODD BROKEN PIECES IN SUSPENSION (FIG. 78)	INTERMEDIATE STATE	30	MATERIAL MOVEMENT IN PIPE BOTTOM (FIG. 78)	INTERMEDIATE STATE
— " —	7·26	119	— " —	— " —	1100	0·5	93·5	$7{\cdot}91 \times 10^{-3}$	70	MATERIAL MOVEMENT IN PIPE BOTTOM. UP TO 40% IN SUSPENSION (FIG. 83)	—— " ——	50	MATERIAL MOVEMENT IN PIPE BOTTOM UP TO 50% IN SUSPENSION (FIG. 83)	—— " ——
— " —	132	2160	— " —	— " —	— " —	— " —	— " —	— " —	80	SLIDING BED MOVEMENT LESS THAN 10% IN SUSPENSION (FIG. 79)	—— " ——	50	MATERIAL MOVEMENT IN PIPE BOTTOM. UP TO 60% IN SUSPENSION (FIG. 79)	—— " ——
— " —	7·26	119	— " —	— " —	1350	0·6	114·5	$9{\cdot}67 \times 10^{-3}$	85	MATERIAL IN SUSPENSION (FIG. 84)	PSEUDO GAS STATE	60	MATERIAL IN SUSPENSION. (FIG. 84)	PSEUDO GAS STATE
— " —	132	2160	— " —	— " —	— " —	— " —	— " —	— " —	90	SLIDING BED MOVEMENT. LESS THAN 10% IN SUSPENSION (FIG. 80)	INTERMEDIATE STATE	70	MATERIAL MOVEMENT IN PIPE BOTTOM UP TO 50% IN SUSPENSION (FIG. 80)	INTERMEDIATE STATE
— " —	7·26	119	— " —	— " —	1700	0·8	144·2	$12{\cdot}18 \times 10^{-3}$	90	MATERIAL IN SUSPENSION (FIG. 85)	PSEUDO GAS STATE	80	MATERIAL IN SUSPENSION (FIG. 85)	PSEUDO GAS STATE
— " —	132	2160	— " —	— " —	— " —	— " —	— " —	— " —	80	SLIDING BED MOVEMENT LESS THAN 20% IN SUSPENSION (FIG. 81)	INTERMEDIATE STATE.	60	MATERIAL MOVEMENT IN PIPE BOTTOM UP TO 50% IN SUSPENSION (FIG. 81)	INTERMEDIATE STATE

FIG. 44 *FLOW CHARACTERISTICS.*

MATERIAL	VOLUME		DENSITY.		AIR FLOW		AIR VELOCITY		STOWER OUTLET			25 ft. (7·6 m) DOWNSTREAM		
	in^3	cm^3	lb/ft^3	kg/m^3	ft^3/min	m^3/s	ft/s	m/s	MATL. DISTN. %PIPE DIA	FLOW CHARACTERISTICS	CONDITION OF FLOW	MATL. DISTN. %PIPE DIA	FLOW CHARACTERISTICS	CONDITION OF FLOW
MAIZE (FIG.37)	5·78	94·7	84·2	1349	800	0·4	67·9	5·74 $\times 10^{-3}$	60	SLIDING BED MOVEMENT. LESS THAN 5% IN SUSPENSION (FIG.66)	INTERMEDIATE STATE	40	MATERIAL MOVEMENT IN PIPE BOTTOM. OCCASIONAL PIECE SUSPENDED (FIG.66)	INTERMEDIATE STATE
—"—	165·2	2720	—"—	—"—	—"—	—"—	—"—	—"—	75	SLIDING BED MOVEMENT. LESS THAN 10% IN SUSPENSION.	—"—	65	MATERIAL MOVEMENT IN PIPE BOTTOM. UP TO 30% IN SUSPENSION.	—"—
—"—	5·78	94·7	—"—	—"—	1100	0·5	93·5	7·91 $\times 10^{-3}$	50	SLIDING BED MOVEMENT. LESS THAN 10% IN SUSPENSION (FIG.67)	—"—	75	MOVEMENT OF MATERIAL IN PIPE BOTTOM. UP TO 50% IN SUSPENSION (FIG. 67)	—"—
—"—	165·2	2720	—"—	—"—	—"—	—"—	—"—	—"—	70	SLIDING BED MOVEMENT. LESS THAN 10% IN SUSPENSION (FIG.)	—"—	90	MOVEMENT IN PIPE BOTTOM UP TO 50% IN SUSPENSION (FIG.)	—"—
—"—	5·78	94·7	—"—	—"—	1350	0·6	114·5	9·67 $\times 10^{-3}$	75	SLIDING BED MOVEMENT. LESS THAN 15% IN SUSPENSION (FIG68)	—"—	95	MATERIAL IN SUSPENSION (FIG. 68)	PSEUDO GAS STATE
—"—	165·2	2720	—"—	—"—	—"—	—"—	—"—	—"—	85	SLIDING BED MOVEMENT. LESS THAN 10% IN SUSPENSION (FIG64)	—"—	95	MOVEMENT IN PIPE BOTTOM UP TO 40% IN SUSPENSION (FIG. 64)	INTERMEDIATE STATE
—"—	5·78	94·7	—"—	—"—	1700	0·8	144·2	12·18 $\times 10^{-3}$	85	SLIDING BED MOVEMENT. UP TO 60% IN SUSPENSION (FIG69)	—"—	95	MATERIAL IN SUSPENSION	PSEUDO GAS STATE
—"—	165·2	2720	—"—	—"—	—"—	—"—	—"—	—"—	75	SLIDING BED MOVEMENT. LESS THAN 20% IN SUSPENSION (FIG65)	—"—	95	MATERIAL IN SUSPENSION	—"—

FIG. 45 FLOW CHARACTERISTICS.

MATERIAL	VOLUME		DENSITY		AIR FLOW		AIR VELOCITY		STOWER OUTLET			25 ft. (7·6 m) DOWNSTREAM		
	in^3	cm^3	lb/ft^3	kg/m^3	ft^3/min	m^3/s	ft/s	m/s	MATERIAL DISTN. %PIPEDIA	FLOW CHARACTERISTICS	CONDITION OF FLOW	MATERIAL DISTN. %PIPEDIA	FLOW CHARACTERISTICS	CONDITION OF FLOW
PELLETS (FIG.34)	4·3	70·5	44·9	719	800	0·4	67·9	5·74 $\times10^{-3}$	15	BED FORMATION. FLOW OF OCCASIONAL PIECE (FIG.74)	SALTATION	15	FLOW OF OCCASIONAL PIECE (FIG.74)	SALTATION
—"—	287	4700	—"—	—"—	—"—	—"—	—"—	—"—	65	BED FORMATION. SLIDING BED MOVEMENT LESS THAN 5% IN SUSPENSION (FIG.70)	INTERMEDIATE STATE	55	MATERIAL FLOW ON PIPE BOTTOM UP TO 20% IN SUSPENSION (FIG.70)	INTERMEDIATE STATE
—"—	4·3	70·5	—"—	—"—	1100	0·5	93·5	7·91 $\times10^{-3}$	50	SLIDING BED MOVEMENT. LESS THAN 10% IN SUSPENSION (FIG.75)	—"—	40	MATERIAL MOVEMENT ON PIPE BOTTOM. LESS THAN 25% IN SUSPENSION. (FIG.75)	—"—
—"—	287	4700	—"—	—"—	—"—	—"—	—"—	—"—	65	SLIDING BED MOVEMENT LESS THAN 10% IN SUSPENSION (FIG.71)	—"—	75	MATERIAL MOVEMENT ON PIPE BOTTOM. LESS THAN 25% IN SUSPENSION (FIG.71.)	—"—
—"—	4·3	70·5	—"—	—"—	1350	0·6	114·5	9·67 $\times10^{-3}$	70	SLIDING BED. MOVEMENT LESS THAN 10% IN SUSPENSION (FIG. 76)	—"—	75	MATERIAL MOVEMENT ON PIPE BOTTOM UP TO 60% IN SUSPENSION (FIG.76)	—"—
—"—	287	4700	—"—	—"—	—"—	—"—	—"—	—"—	60	SLIDING BED MOVEMENT LESS THAN 10% IN SUSPENSION	—"—	90	UP TO 80% OF MATERIAL IN SUSPENSION. SOME MATERIAL ON PIPE BOTTOM (FIG.72)	—"—
—"—	4·3	70·5	—"—	—"—	1700	0·8	144·2	12·18 $\times10^{-3}$	75	SLIDING BED MOVEMENT UP TO 50% IN SUSPENSION (FIG.77)	—"—	95	MATERIAL IN SUSPENSION	PSEUDO GAS STATE
—"—	287	4700	—"—	—"—	—"—	—"—	—"—	—"—	95	SLIDING BED MOVEMENT LESS THAN 15% IN SUSPENSION (FIG.73)	—"—	95	SOME MOVEMENT IN PIPE BOTTOM OVER 50% IN SUSPENSION (FIG.73)	INTERMEDIATE STATE

FIG. 46 FLOW CHARACTERISTICS

MATERIAL	VOLUME		DENSITY		AIR FLOW		AIR VELOCITY		STOWER OUTLET			25 ft. (7·6 m) DOWNSTREAM.		
	in^3	cm^3	lb/ft^3	kg/m^3	ft^3/min	m^3/s	ft/s	m/s	MATERIAL DISTN. % PIPE DIA	FLOW CHARACTERISTICS	CONDITION OF FLOW	MATERIAL DISTN. % PIPE DIA	FLOW CHARACTERISTICS	CONDITION OF FLOW
PENCILS (FIG.36)	14·9	244·5	74·8	1198	800	0·4	67·9	$5{\cdot}74 \times 10^{-3}$	10	BED FORMATION. NO MATERIAL FLOW (FIG. 59)	SALTATION	10	BED FORMATION. NO MATERIAL FLOW (FIG.59)	SALTATION
— " —	172	2818	— " —	— " —	— " —	— " —	— " —	— " —	20	— " — (FIG.39)	— " —	10	— " — (FIG.39)	— " —
— " —	14·9	244·5	— " —	— " —	1100	0·5	93·5	$7{\cdot}91 \times 10^{-3}$	45	BED FORMATION SLIDING BED MATL. FLOW NO MATERIAL SUSPENSION (FIG.60)	INTERMEDIATE STATE.	20	MATERIAL FLOW ON PIPE BOTTOM (FIG.60)	INTERMEDIATE STATE
— " —	172	2818	— " —	— " —	— " —	— " —	— " —	— " —	55	SLIDING BED MOVEMENT. LESS THAN 5% IN SUSPENSION (FIG. 40)	— " —	45	SLIDING BED ON PIPE BOTTOM. LESS THAN 5% IN SUSPENSION (FIG.40)	— " —
— " —	14·9	244·5	— " —	— " —	1350	0·6	114·5	$9{\cdot}67 \times 10^{-3}$	65	SLIDING BED. UP TO 30% IN SUSPENSION (FIG. 61)	— " —	40	SLIDING BED ON PIPE BOTTOM LESS THAN 5% IN SUSPENSION (FIG.61)	— " —
— " —	172	2818	— " —	— " —	— " —	— " —	— " —	— " —	80	SLIDING BED. UP TO 50% IN SUSPENSION (FIG. 41)	— " —	50	SLIDING BED ON PIPE BOTTOM LESS THAN 5% IN SUSPENSION (FIG.41)	— " —
— " —	14·9	244·5	— " —	— " —	1700	0·8	144·2	$12{\cdot}18 \times 10^{-3}$	90	SLIDING BED UP TO 70% IN SUSPENSION (FIG. 62)	— " —		TEST RESULT IN DOUBT (FIG. 62)	— " —
— " —	172	2818	— " —	— " —	— " —	— " —	— " —		70	SLIDING BED. UP TO 50% IN SUSPENSION (FIG.42)	— " —	60	SLIDING BED IN PIPE BOTTOM LESS THAN 5% IN SUSPENSION (FIG.42)	— " —

FIG. 47 FLOW CHARACTERISTICS

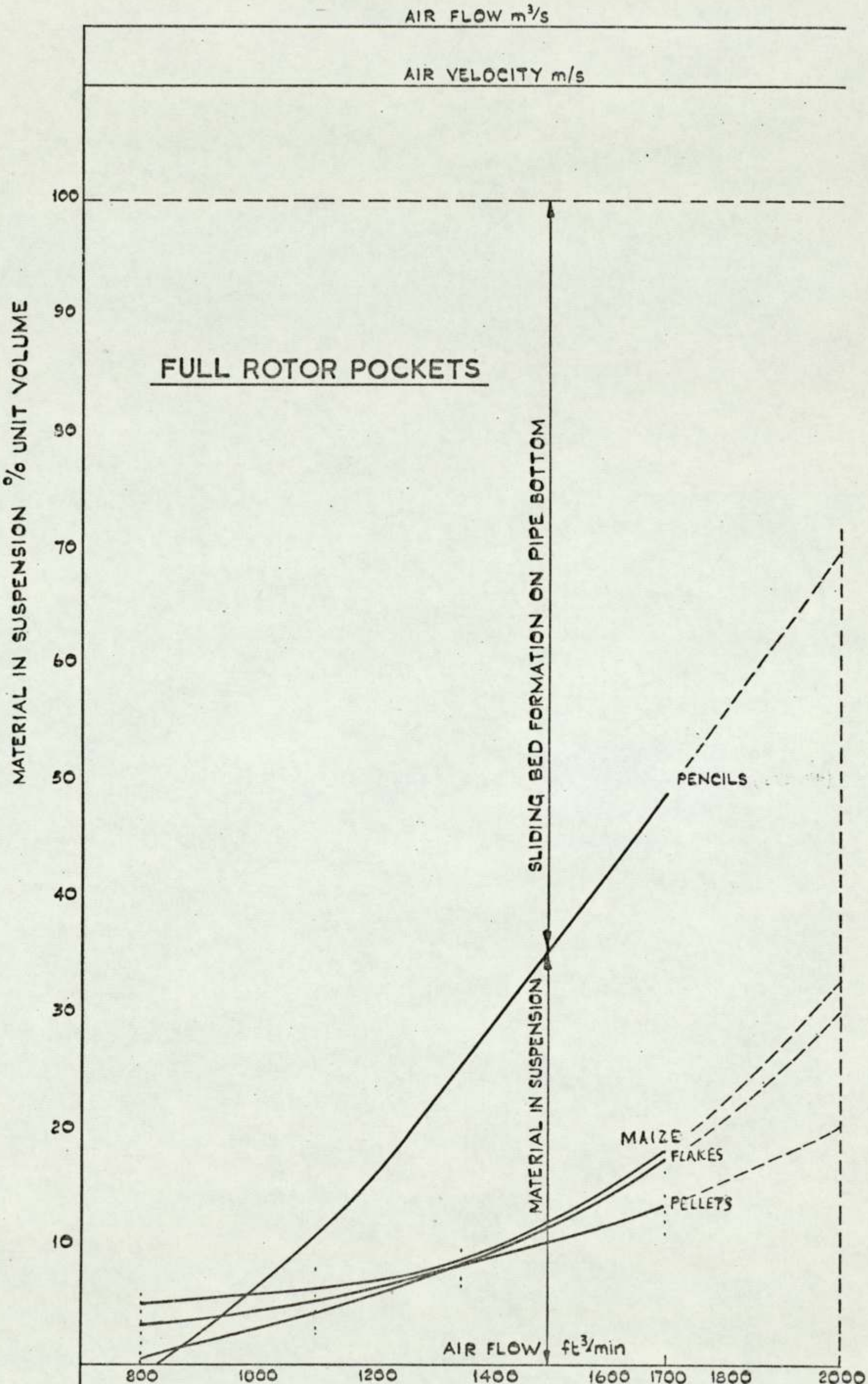


FIG. 48 EFFECTS OF AIR FLOW ON MATERIAL IN SUSPENSION - STOWER OUTLET.

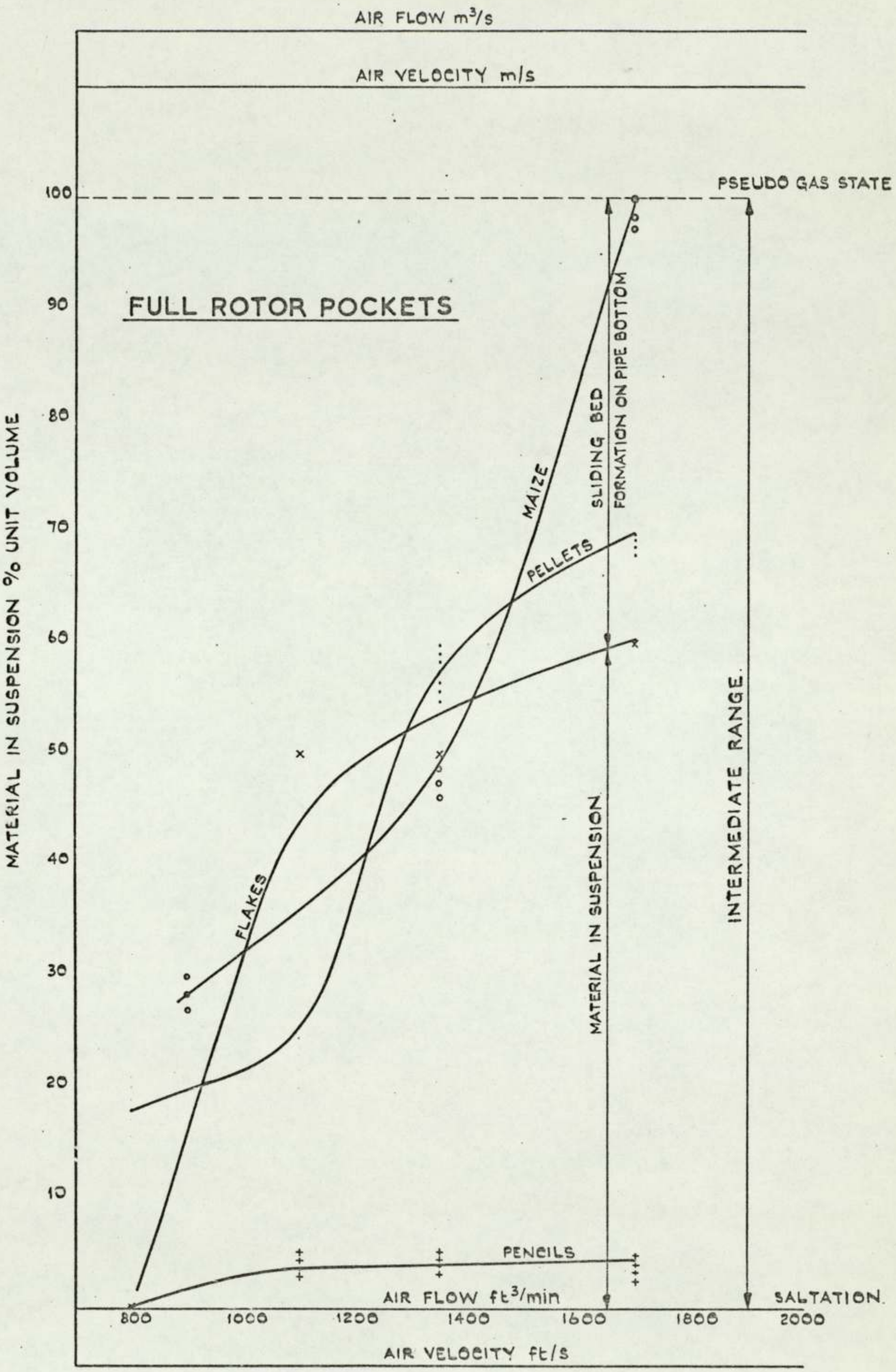


FIG.49 EFFECTS OF AIR FLOW ON MATERIAL IN SUSPENSION - DOWNSTREAM.

material which are in suspension and in bed formation has been plotted against air flow. The tests produced a scatter of results and the curves plotted represent the average values for the different materials examined.

STOWER OUTLET

At the stower outlet the effect of air flow on the percentage of material flowing in suspension is shown in Figure 48. All the materials investigated moved from the stower in the "INTERMEDIATE STATE" (Figure 43C), and the effect of increase in air flow up to 1700 ft^3/min (0.85 m^3/s) was to increase the proportion of material which moved in suspension. The sliding bed still remained.

Examining the curves in Figure 48 the pencils show a distinct difference in the distribution of material than did the other materials which followed similar patterns of behaviour. At the maximum air flow operated during these investigations (1700 ft^3/min 0.85 m^3/s) up to 50% of the pencils were in suspension compared with a figure of 20% for the other materials.

Observations for air flows beyond 1700 ft^3/min(0.85 m^3/s) was not possible but on the assumption that the gradient of the curves shown in Figure 49 continue in the same manner, then it can be seen that with an air flow of 2000 ft^3/min (0.9 m^3/s) the percentage volume of material moving in suspension is approx. 70% for the pencils and between 20 and 30% for the other materials. Considering these values as a whole, an air flow of 2000 ft^3/min (0.9 m^3/s) would maintain up to 50% of the material in suspension, and transportation would still be mainly on the pipe bottom in a sliding bed formation.

To avoid material build-up on the bottom of the pipe, this being the first stage of pipe blockage, the majority of the material must move from the stower in the air stream. On this basis, and from the observations carried out there is no evidence to show that an air flow substantially less than that used during the prototype stower trials (2.250 ft^3/min -

1.1 m^3/s) could be used.

The problems of moisture content, stickiness, and the continual variation in the fines/solids ratio of run of mine debris, will undoubtedly make material movement from the stower more difficult than those observed during the stower model investigations, and air flows below that used during the stower trials would not produce the required flow condition at the stower to prevent material build-up and pipe blockages.

This investigation substantiates the air flow used in the stower trials where the required output and reliable blockage free stowing was achieved.

The significance of this conclusion lies in the necessity to use a Rootes type blower if pneumatic stowing is to be considered an economic proposition. The stower trials confirmed that a maximum operating pressure of about 16 lb/in^2 ($112 \times 10^3 N/m^2$) was required for the stowing circuit investigated. Therefore the development of a Blower supplying 2250 ft^3/min (1.1 m^3/s) free air delivery operating at pressures up to 16 lb/in^2 ($112 \times 10^3 N/m^2$) is necessary before pneumatic stowing underground can be considered economically.

5.9.8 DOWNSTREAM OBSERVATIONS

During these investigations an attempt was made to observe the behaviour of material 25 ft (7.6 m) downstream of the stower.

With the equipment used, the recording on film of the material flow characteristics proved to be extremely difficult. The accuracy of the results relied on the skill of operating the camera by hand at the precise time that material passed through the range of the camera vision. At the stower point where the material was starting from zero velocity, the timing of the camera was much easier and the results obtained produced acceptable results. At the downstream observation point there was a much larger fluctuation in the results obtained. In the first place the material was travelling with greater velocity, since it was accelerating along the pipe, and more important the material tended to spread itself out longitudenally along the pipe. The limited vision of the camera was in some cases only able to record part of the unit of volume of material being conveyed. A concentrated study of the downstream conditions would require a re-examination of the test rig and a more accurate method of recording material behaviour.

The recordings which were obtained have been examined as means of obtaining some indication of the flow characteristics downstream. An assessment of material distribution at the downstream point are included in Figures 44 to 47, and curves representing the average values of material in suspension and that which slithered along the bottom of the pipe is shown in Figure 49.

The results for the pencils shows no appreciable difference in the flow characteristics over the whole range of air flows examined. The reason for this is probably due to the inadequacies of the equipment mentioned previously. The recordings are misleading in as much as the material shown on the film does not represent the full unit volume examined. The values for the pencils are therefore in doubt and are to be disgarded.

The behaviour of the other three materials do appear to follow a similar pattern, and although the accuracy of the values can be questioned there is an indication that the movement of material in suspension at the downstream position is greater than that at the stower exit. Confirmation of this requires further investigation but the problem of pipe and stower blockages experienced in stowing operations does suggest that the air flow requirements for the short stowing circuit considered in this investigation are governed by the conditions at the stowing machine.

5.9.9 PARTICLE SETTLING POSITION

The work of other investigators suggests that with particular shapes such as flat disks, cubes, and cylinders, a settling position in the fluid can be assumed from which the resistance exhibited by the solid can be obtained.

When considering pneumatic stowing where there is a constant flow of material made up from particles of various shapes and sizes, the movement of any particle is greatly influenced by the mechanical interference of particles with one another. Observations of the materials used in the model tests clearly shows that the particles both at the stower outlet and downstream did not show any common settling position. The turbulence within the stowing machine and conveying pipe caused the particles to exhibit spin whilst in motion. The different positions of the particles relative to the direction of flow can be observed in Figure 50. The other materials used during these investigations behaved in a similar manner, and in none of the cases investigated was there any evidence to show that for the stowing circuit examined, particles settled into any particular position relative to the air flow.

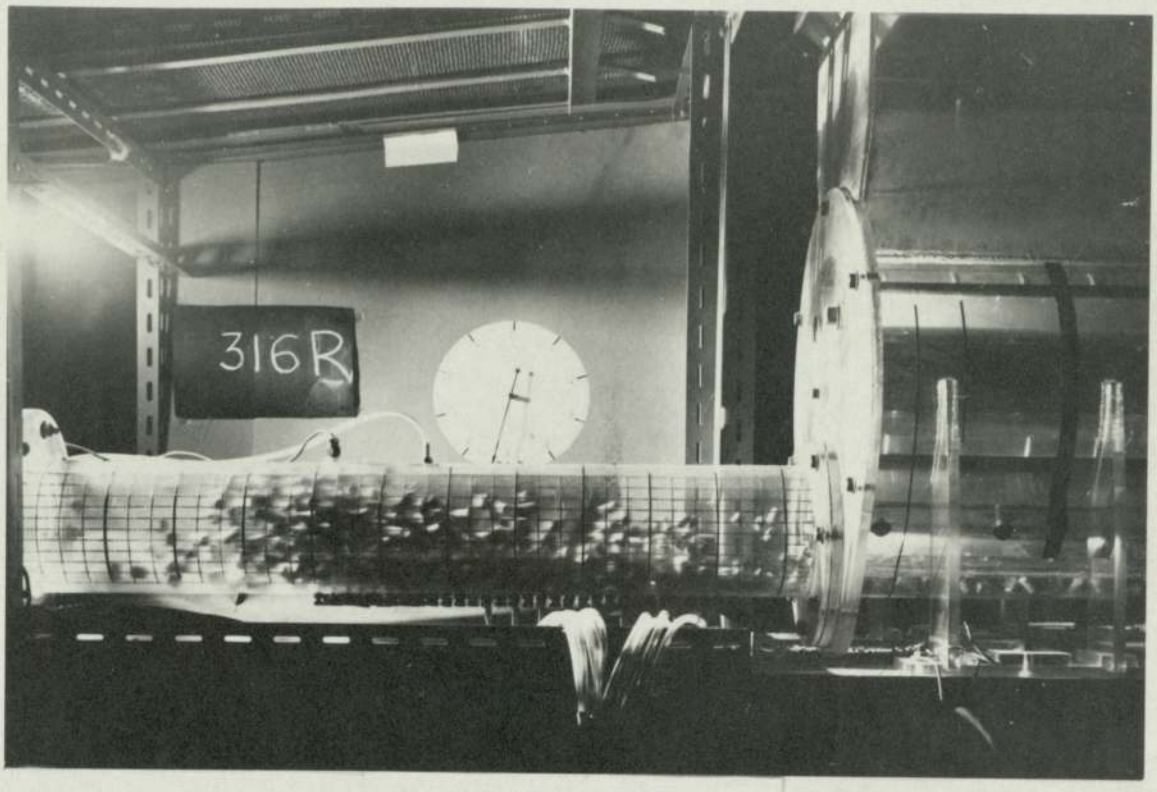
5.9.10 MATERIAL SIZE AND SHAPE

A reliable assessment of the air velocity requirements in the pipeline downstream of the Model stower was not possible because of the difficulties experienced with the test equipment. This has been discussed previously (5.9.8). To obtain additional information the application of existing data to the stower model conditions has been considered.

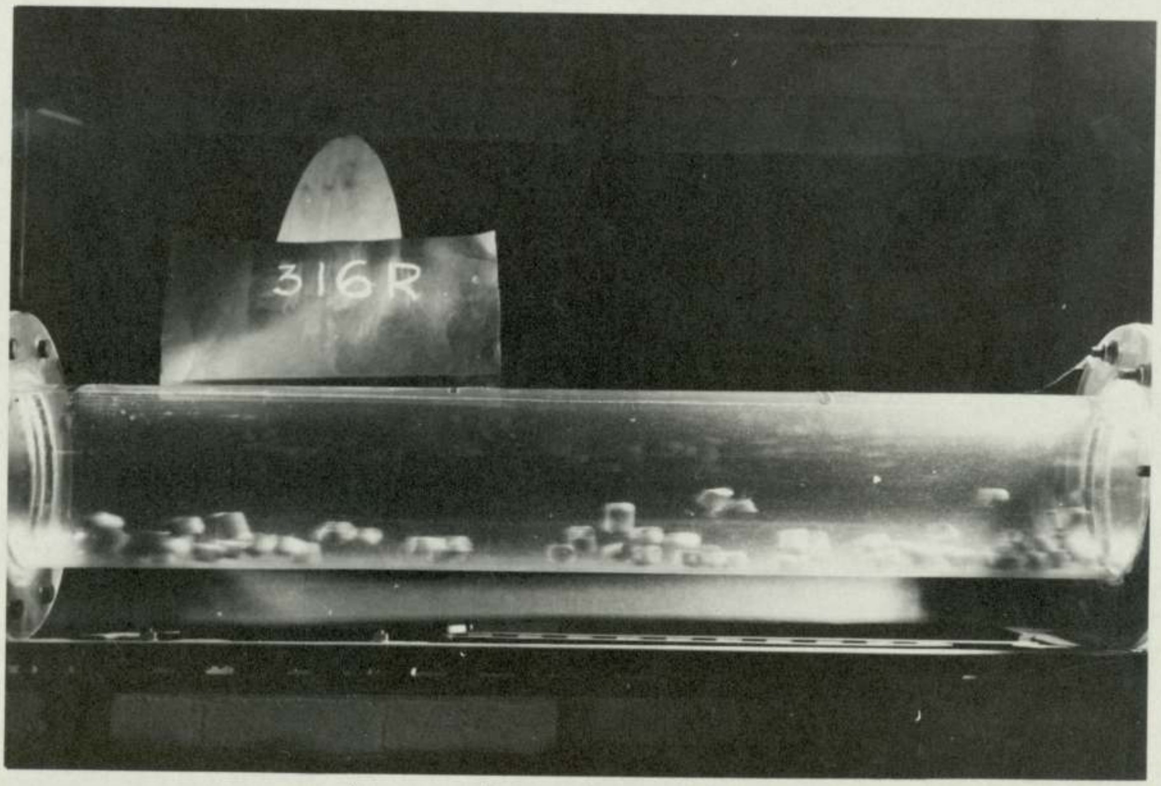
The work of other investigators has already been discussed in chapter 2 and the guides that are available for estimating floating or mixing velocities include amongst other data, factors relating size and shape of particles. Sphericity and circularity factors have been used and the formulae for calculating these values are:

1. Degree of circularity $\not{c}$ $= {}^{c}/_{C}$

where c = circumference of circle having same area as plane figure



a) STOWER OUTLET



b) 25ft (7·6m) DOWNSTREAM

FIG. 50 1/2 IN (13mm) PENCILS - FULL POCKET
AIR FLOW 1350 ft.^{3}/min (0·6m^3/S)

C = actual perimeter of the plane figure

2. Degree of true Sphericity ϕ = $^{s}/_{S}$

where s = surface area of sphere having the same volume as the particle

S = actual surface area of the particle.

It is obvious from these formulae that the circularity factor is applicable to particles which can be classified as plane figures (i.e discs) and the sphericity factor is applied to other shapes which can be more related to spheres (cubes, cylinders etc.).

Sphericity and circularity factors have been considered for the four materials used in the stower model trials. In the case of the flakes, only a maximum value, which is unity, can be derived because the flakes have to be considered as discs which although variable in shape has a maximum size of 2½ in (64 mm) diameter this being fixed as the maximum stower material size for 6 in (152 mm) diameter pipes. Considering the Flat Maize, it is questionable whether particles such as these which have a thickness equal to half their diameter, are classed as plane figures or spheres.

Applying both formulae for the maize, only a circularity factor of unity can be obtained. Cylinders such as the Pellets and Pencils can be related more to spheres, and the sphericity values calculated for these were 0.8 and 0.95 respectively.

The application of sphericity and circularity formulae for the stower model materials shows that any one of a number of factors could be applied to all the sizes and shapes of materials possible in pneumatic stowing. A size factor which is based on values which are common to all particles irrespective of whether they are in the sphere or disc category is necessary for materials such as run of mine debris where size and shape are different and constantly changing.

5.9.11 CONVEYING AND FLOATING VELOCITIES

Other investigators have used factors other than those discussed above for calculating floating or mixing velocities. Dallavalle's(3) formula for the initial mixing velocity states that

$$Vo = 6000 \left[{}^{s}/(s + 1) \right] d^{0.398}$$

where Vo = Initial mixing velocity - ft/min

s = Specific Gravity of Solid

d = Average diameter of Particle - in

Dallavalle's work considered Cinders, Carbon, Anthracite and Quartz particles up to .320 in (8 mm) average diameter. The specific gravities ranged between 1.08 and 2.65, and the average diameter value was based on the average of two screen sizes.

To apply the Dallavalle formula to the stower model materials again presents the difficulty of establishing the average diameter of the particles in question. In Figure 51 the initial mixing velocity obtained from the equation has been plotted as a function of the average diameter for the materials used in the stower model investigations. Although the average diameter value for each material is questionable, the curves do show the initial mixing velocity over a range of average diameters up to a maximum value for the respective materials.

The curves show how the Pellets and Maize come within the range over which the Dallavalle equation is established, whereas the Pencils and Flakes are outside the range. The curves for the Flakes and Pencils have been plotted beyond the established range of the equation as a means of showing how the velocities may vary considerably according to particle size and shape.

Within the range of the equation the initial mixing velocity lies between 24 and 40 ft/s (7 and 12 m/s) for the materials considered.

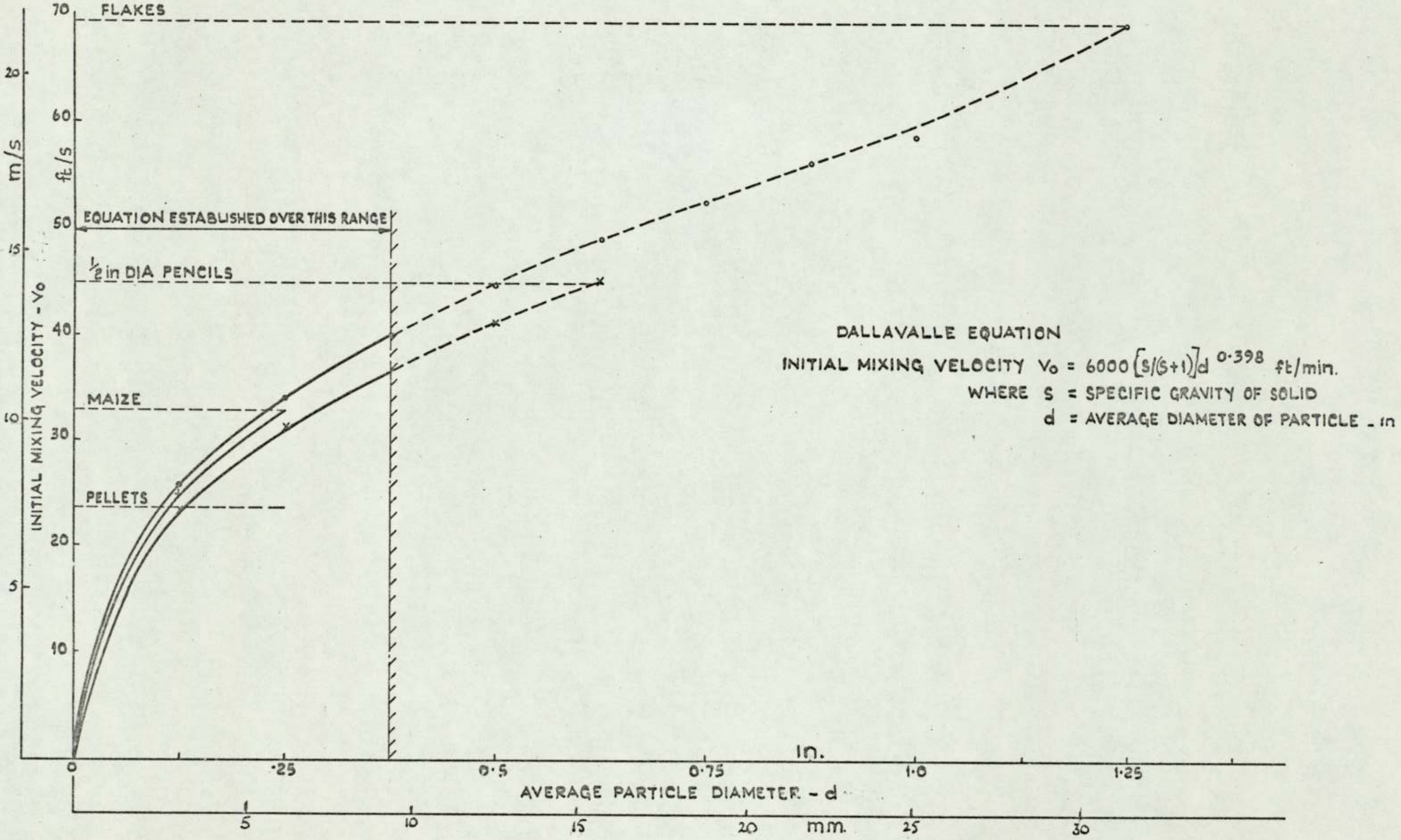


FIG.51 APPLICATION OF DALLAVALLE'S EQUATION FOR STOWER MODEL MATERIALS.

Dallavalle's equation gives the air velocity for material suspension and velocities considerably in excess of those that can be deducted from the equation are necessary in order to transport the material. The formula does not consider material concentration such as that found in the stower model tests. These factors account for the apparent low values of air velocity suggested from the curves in Figure 51 and when compared with the air velocities predicted from the model tests it appears that the conveying velocity would be higher than the initial mixing velocity by a factor as high as three or four.

Consideration of Dallavalle's equation for predicting the air velocity requirement downstream of the stower model is unreliable due to its limited range of application and its inability to relate floating velocity to that required for conveying.

The work of Davies(7) referred to in chapter 2 offers a more reliable method of assessing the air velocity requirements downstream of the stower model. In his work Davies considers Flaky and Spherical particles and gives a formula for initial mixing velocity which states,

$$\text{Initial mixing velocity} - V_o = \sqrt{\frac{2g}{W_f}\,\frac{V}{A}\,(W_m - W_f)}$$

where W_f = Fluid Density lb/ft^3
W_m = Solid Density lb/ft^3
V = Volume of Particle in^3
A = Cross Sectional Area of Particle in^2

The application of the formula again depends on the ability to place a value on the cross sectional area of the particle. With the material being considered it is again questionable which cross sectional area should be taken. This depends to some extent on the settling position of the particle in the air stream a feature which has not shown to be consistent as far as the stower model investigations are concerned (5.9.9).

Considering the Davies equation, the mixing velocity has been plotted over a range of volume to area ratios for the stower model materials (Figure 52).

The air velocities for the maximum possible values of $^V/A$ varies between 100 and 210 ft/s (30 and 64 m/s). Considering the probability that the true value of the particle factor lies somewhere below the maximum, then a predicted value of mixing velocity for the stower materials can be said to be within a range of 80 to 180 ft/s, (24 to 55 m/s). In general terms, this supports the conclusions that air flows less than 2250 ft^3/min (1.1 m^3/s) which gives an air velocity of 191 ft/s (16.15×10^{-3} m/s) in a 6 in (152 mm) diameter pipe would not be sufficient for reliable and blockage free pneumatic stowing.

5.9.12 STOWER OUTPUT

Comparing the movement of materials from the stower there is no single corelation between rate of movement from the stower and the material specific weight. Examination of Figures 39 to 41 shows that the Flakes which have the greatest specific weight did in fact take less time to clear the stower than did the other materials investigated. The reason for this is that the volume of material passing through the stower is dependent on the ability of the material to compact together and occupy the maximum of space available in a rotor pocket. For any particular material, the maximum volume that can be metered through the stower is dependent on the combination of size, shape, and specific weight. The unreliability of assessing outputs on specific weight alone, is shown when examining the stower model throughput rates which are shown in Figure 53. The curves show the stower model throughput rates for the range of air flows investigated. The output in tons per hour have been based on the solid mass density of the materials and the measured weights of materials used. The results shows that the maximum outputs were found to be that of the Pellets with the Maize throughput being the minimum. Comparing the outputs against the specific weight of the materials for a

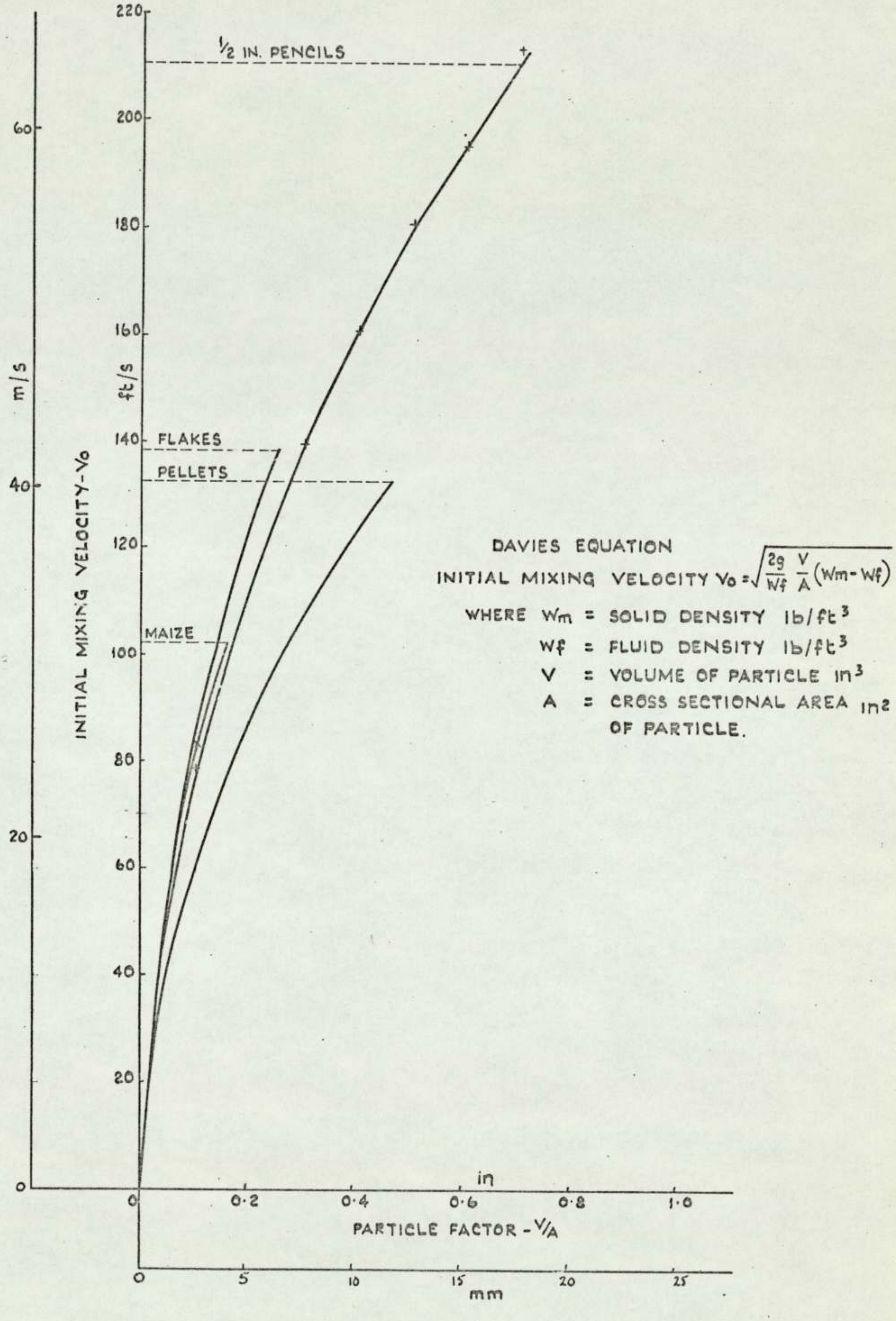


FIG. 52 APPLICATION OF DAVIS'S EQUATION FOR STOWER MODEL MATERIALS.

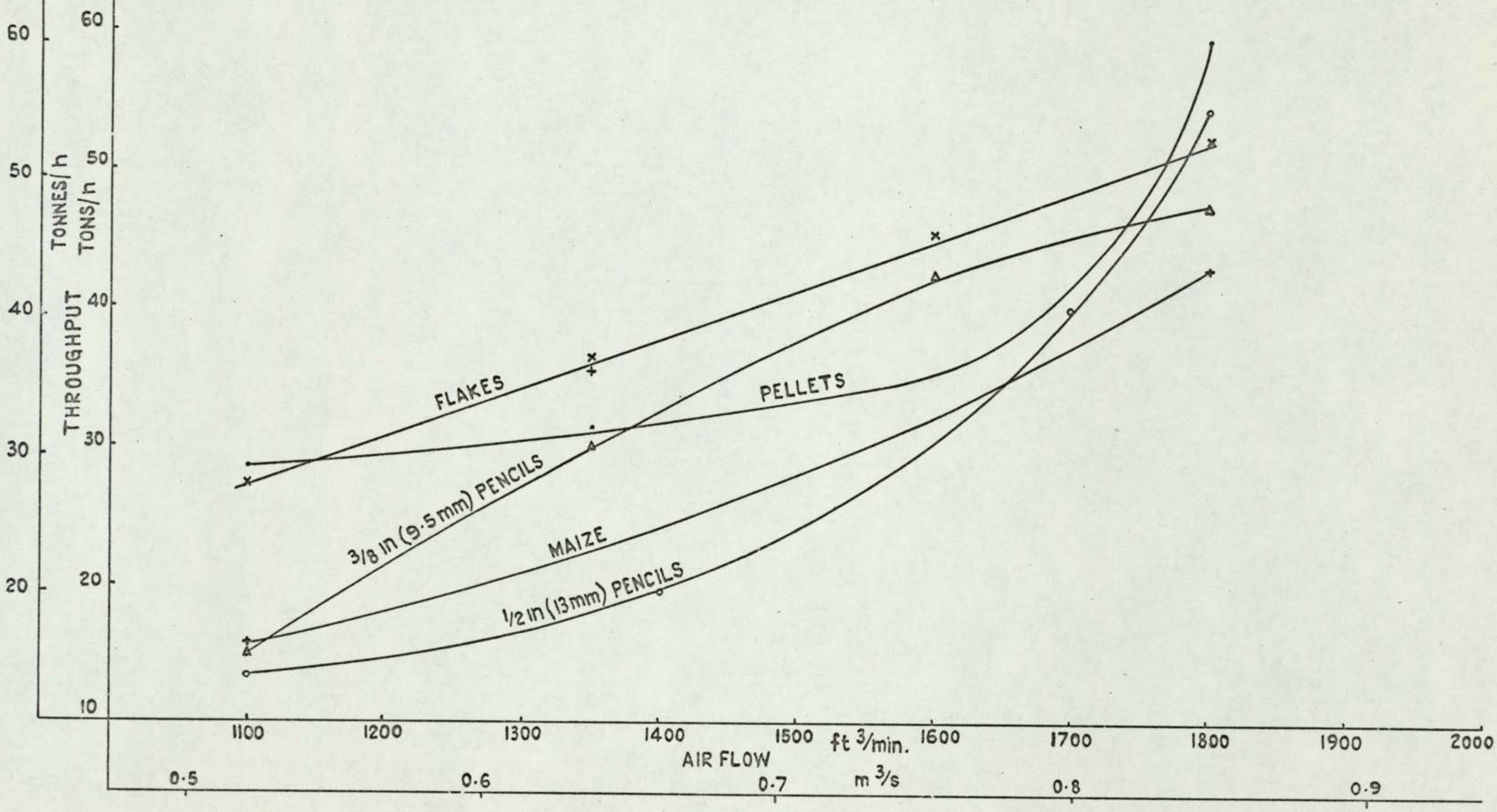


FIG. 53 STOWER MODEL MATERIAL THROUGHPUT RATES - (FULL ROTOR POCKETS)

particular air flow, there does not appear to be any consistency. This again suggests that the criteria in assessing material behaviour and outputs, will be the ability to assess the combined effects of the various characteristics of the materials to be conveyed.

The true value of stowing output can only be specified by its association with the type, size and condition (wet, dry, sticky) of the material to be conveyed.

CHAPTER 6

OTHER PNEUMATIC CONVEYING INVESTIGATIONS

6.1 STOWING PIPE RANGE

The work described in the thesis is limited to a study of the principles associated with the design and development of the stowing unit and a study of the behaviour of material passing through the stower into the stowing pipeline.

No attempt has been made to investigate the dynamics of pneumatic transportation in the stowing pipeline generally. Considerable work has already been carried out by other workers on the pneumatic conveying of material such as grain, seeds and other relatively small particles. These have been discussed in chapter 2.

Many formulae exist, and a study of some of the work has been made in an attempt to understand the behaviour of larger material (Run of Mine debris up to 3 in (76 mm) in size) used for stowing operations underground.

The layout of the stowing pipe range varies according to the particular mining system being operated, but generally the material is produced at the roadway, and is stowed into the pack area which is at right angles to the roadway.

For the purposes of this work the stowing pipe range has been limited to a maximum length of 60 yds (55 m) which includes one 90° bend. This range would normally cover most stowing installations being carried out at the present time.

P. H. Broadhurst(9) Mining Department, Leeds University, investigated the general principles associated with the pneumatic conveying of large size solids by pipeline. Reference to this work is made because the investigations were carried out in the field on full scale stowing installations.

The relevant information from Broadhurst's work falls into three sections. These are;

(1) Stowing materials

(2) Material conveying round bends

(3) Material Movement in the pipeline.

6.2 STOWING MATERIALS

During stowing operations difficulties occur due to the nature of the material being handled. Blockages in the stowing unit and in the pipes are often due to the stickiness of the material.

Broadhurst investigated the stickiness of a variety of materials which were representive of material likely to be handled throughout the areas of the National Coal Board. The conclusions drawn from these tests indicated that the order of stickiness changes with different moisture content, and it was considered that the cause of stickiness is due to the nature rather than the amount of fines in the stowing material. From a study of the properties of soil, it also appeared that the stickiness of some of the wet stowing materials is due to the stickiness of the minute clay particles in the fines.

It can be generally accepted that the stickiness of a stowing material depends on the proportion of clay particles as well as the moisture content.

From Broadhursts study, the method of estimating the stickiness of stowing materials is based on the determination of the clay content by means of plasticity tests. The plasticity of the materials is identified by a Plasticity Index (P.I.) and is obtained from the equation:

Plasticity Index (P.I.) = Liquid Limit (L.L.) - Plastic Limit (P.L.)

where

1. The liquid limit is the moisture content at which the materials passes from the plastic to the liquid state. This is the point

where the fines start to flow when lightly vibrated.

2. The plastic limit is the lowest moisture content at which a -36 mesh B.S. portion of soil can be rolled into a $\frac{1}{8}$ in (3 mm) diameter thread.

The plasticity index is the range of moisture contents over which the soil is plastic and is closely related to the clay content in the solid.

Krynine(20) states that the sticky point which characterises the capacity of a given soil to adhere to metallic surfaces lies between the liquid and plastic limits. Figure 54 shows the suggested relationship between stickiness and moisture content. The graph shows that at the 'Liquid Limit' and at low moisture content, stickiness is negligable. If a material with 15% moisture content is very sticky, then reading the graph (Figure 54) it can be seen that the stickiness will be noticeably less with a moisture content of 5 or 25%. This theory on the cohesiveness of material can be used to advantage when assessing the likelihood of stowing materials sticking to the metallic surfaces of the stowing machine and conveying pipes.

The practical use of this theory is its application to determine the nature of the "Fines" in stowing materials by carrying out "plastic limit" and "Liquid Limit" tests.

The information on stowing materials presented in the previous paragraphs represents a brief summary of the detailed investigations carried out by "Broadhurst", and is intended to indicate the importance of obtaining knowledge of the probable stickiness of materials and the effect it can have on the operating efficiency of stowing installations. Details of the "Liquid Limit" tests, and the procedures for considering the suitability of a material for pneumatic stowing, are included in BS. 812: 1967 104P,(14) BS. 1377: 1967 236P.(15)

Figure 55 shows the results of tests carried out by Broadhurst on

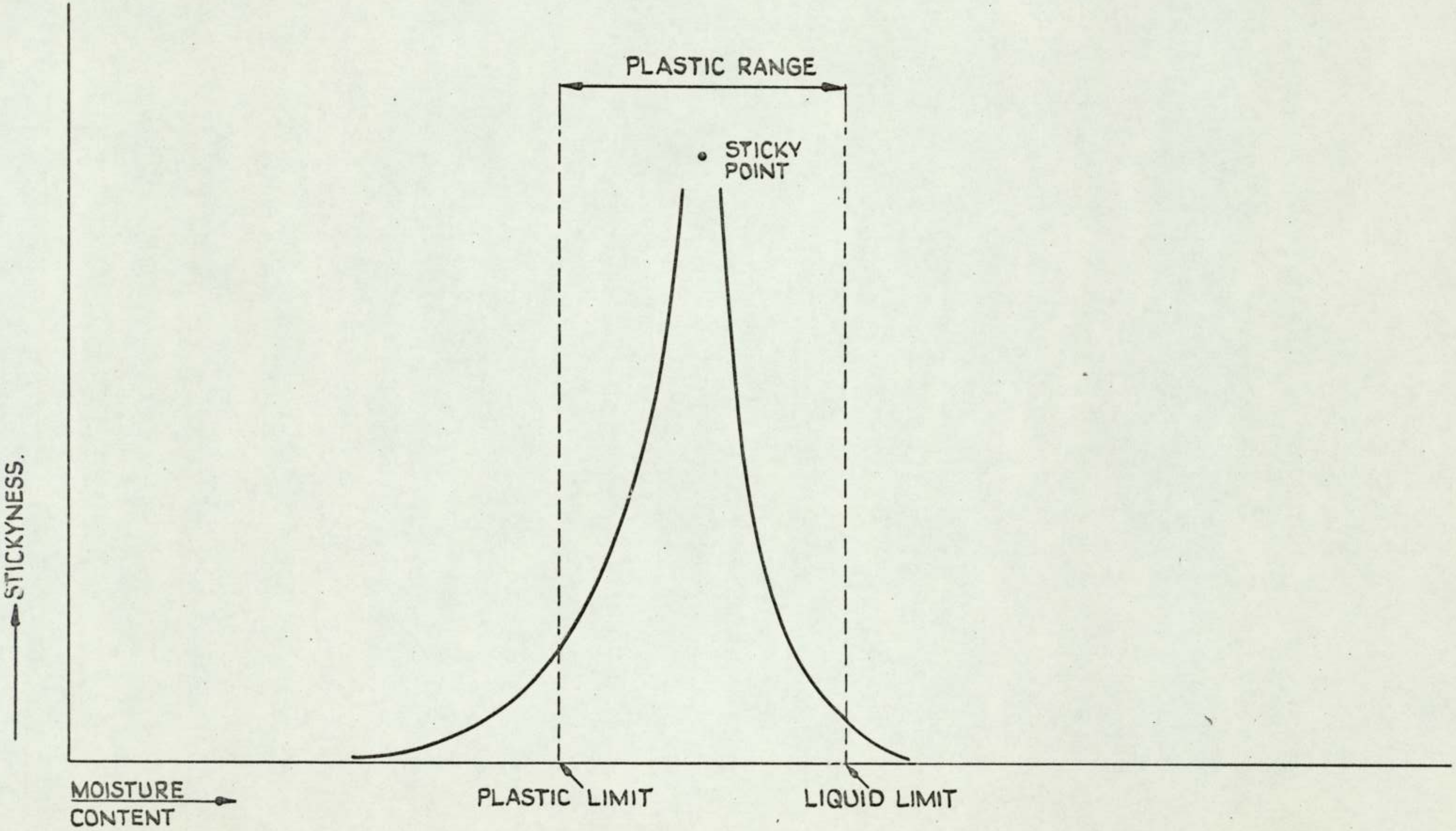


FIG. 54 STICKYNESS/MOISTURE RELATIONSHIPS

	Bullcroft Colliery Yorks.	Upton Colliery Yorks.	Ryhope Colliery Co.Durham
Percentage Fines (-36 Mesh B.S.)	2	6	5
Remainder (Course Sand and Gravel Size)	98	94	95
Plastic Limit (P.L.)	20	19	18
Liquid Limit (L.L.)	31	29	27
Plasticity Index (L.L. - P.L.)	11	10	9
Percentage Water Added in wetting test (Lower Limit of Plastic Range)	4	7	9
Weight of Water for fines (Per 100 lbs of material)	0.62	1.74	1.35
Weight of Water for Gravel (Per 100 lbs of material)	3.92	6.58	8.55
Total weight of water to saturate fines	4.54	8.32	9.9
Practical % Moisture (Upper limit of Plastic Range)	5	8	10
Plus 2% to allow for degradation effects	7	10	12
Plastic range of moisture contents	4-7%	7-10%	9-12%

FIG. 55 TEST RESULTS OF THREE CLAYEY STOWING MATERIALS

three clayey stowing materials. The plastic range of moisture contents is derived for each type of material. Material with moisture contents within these ranges is to be avoided for stowing operations.

6.3 EFFECTS OF PIPE BENDS

Any deviation from a straight stowing pipe affects the performance and overall efficiency of pneumatic stowing operations. For this reason it is important to have some knowledge of the effects of introducing a 90° bend in a stowing pipeline.

Broadhurst, in his study of pneumatic transport of material for stowing operations, carried out a series of experiments to obtain information on the effects pipe bends.

(1) Losses due to Flow of Air round bends

Previous work carried out had shown that the disturbance to air flow at a bend can be carried downstream as much as 50 pipe diameters, and this disturbance produces additional pressure drop separate from that created by the bend itself. For this reason the results obtained by Broadhurst are based on experiments using a pipe range which had 150 ft (46 m) of straight pipes before the bend and 40 ft (12 m) after the bend. The pressure losses due to the introduction of the bend was taken as the sum of the loss at the bend and the loss due to the air disturbance in the straight pipe downstream of the bend. Figure 56 shows the total pressure losses for various bend radii over a range of flows up to 2000 ft^3/mm (0.9 m^3/s). These results indicate that the greatest losses occurred with the small bend having a radius of 1.5 ft (0.5 m) and the lowest losses were produced with bends at 4 and 5 ft (1.2 and 1.5 m) radius. Pressure losses within a 6 in (152 mm) diameter pipe having different pipe bend radii are given in Figure 57. The pressure losses which are related to an air flow of 2000 ft^3/min (0.9 m^3/s) shows that there is an increase in pressure loss for bends beyond 5 ft (1.5 m) radius.

The investigation showed that for the smallest bend the losses

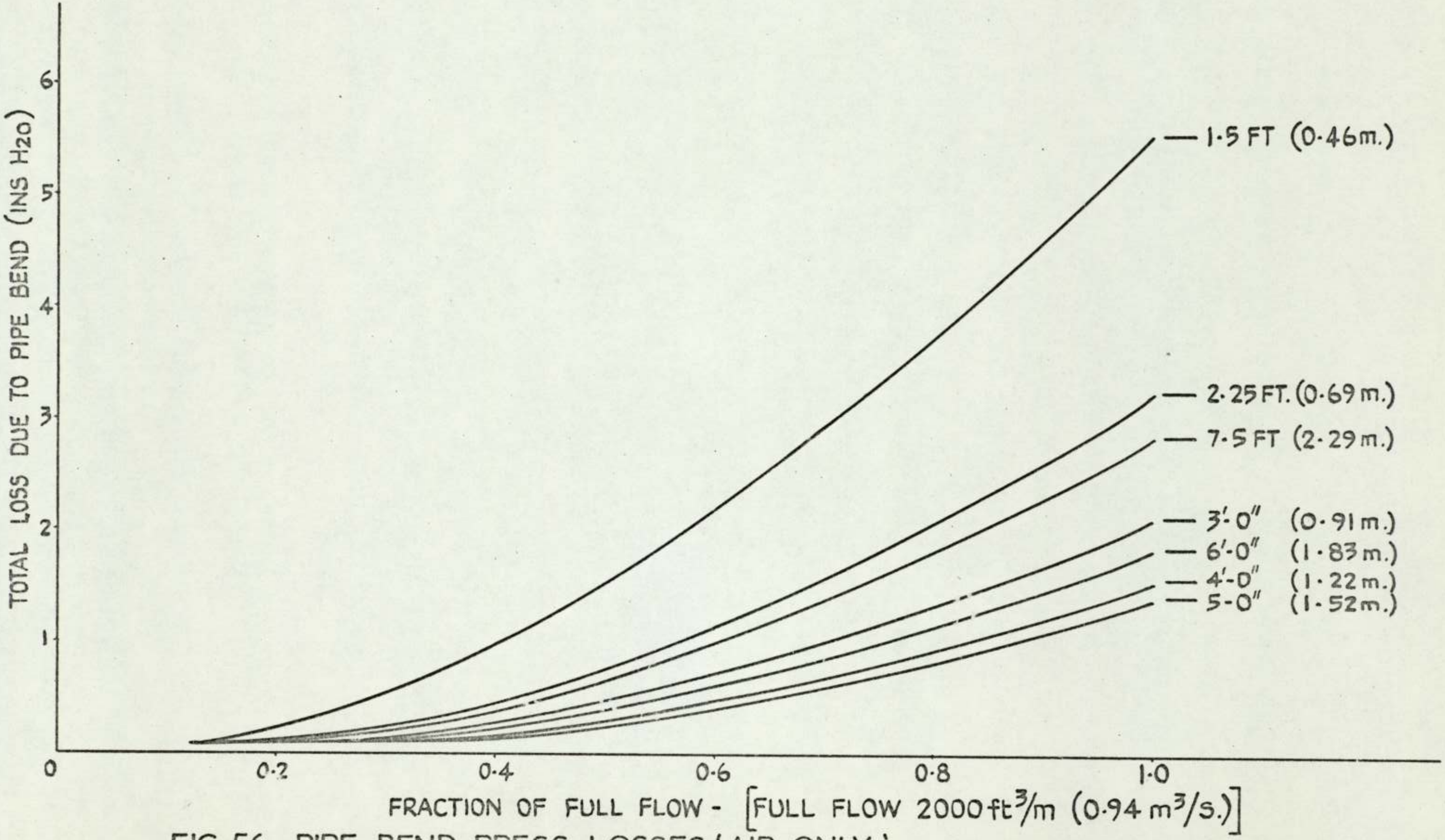


FIG.56 PIPE BEND PRESS. LOSSES (AIR ONLY.)

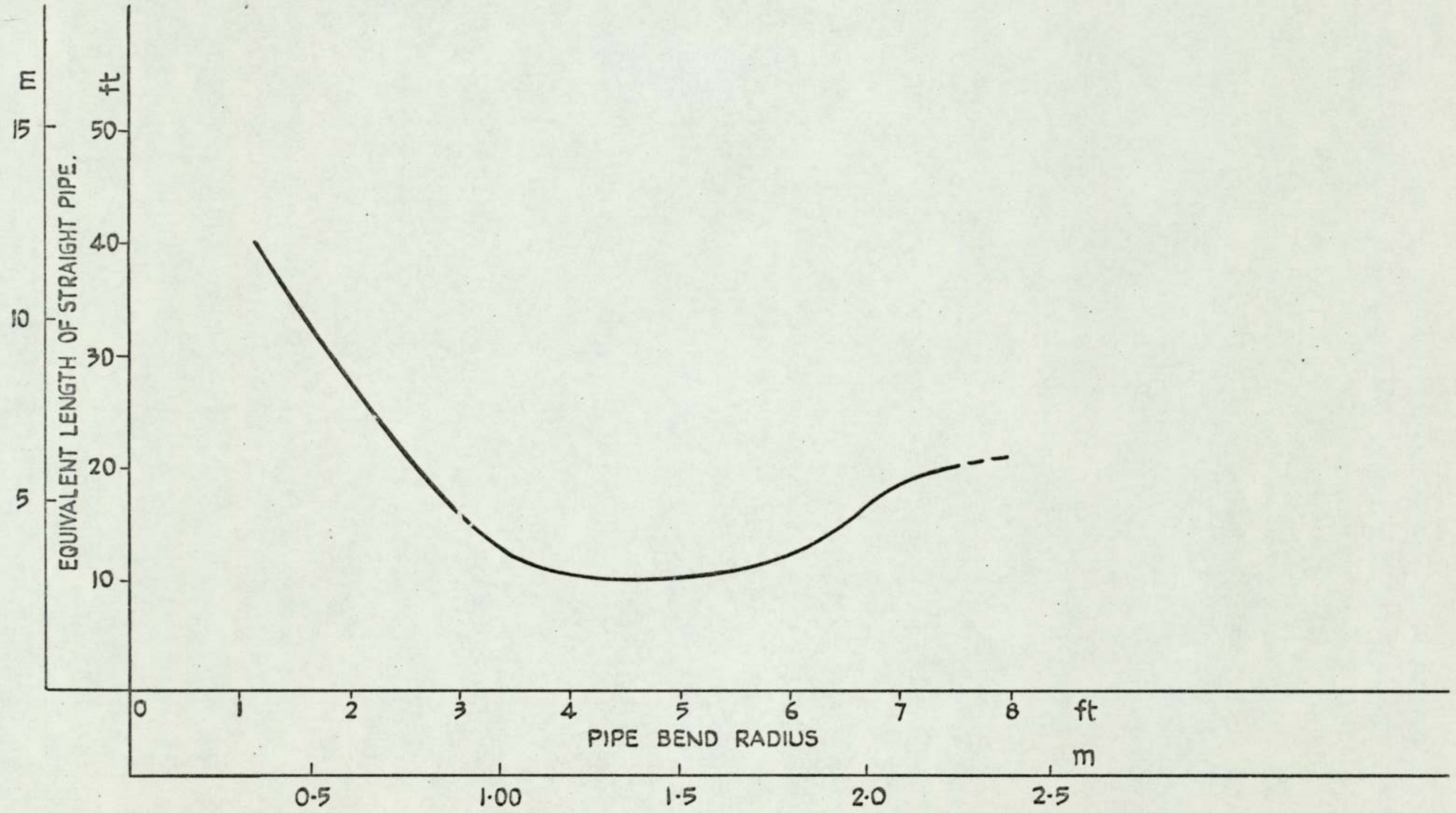


FIG. 57 PIPE BEND PRESSURE LOSSES IN 6ins (152·4mm) DIA. PIPE AT 2000ft^3/min. (0·94m^3/s) AIR FLOW.

at the bend were the same as the downstream loss. On the larger bend the losses downstream were less than the loss at the bend.

(2) Losses at Bends during Stowing (Air and Material)

Because of the large variation of conditions namely, air flow, stowing rate, type of material, pipe bend radius, and roughness of pipe, there is no reliable information which can give the pressure losses at a bend during stowing operations. Information from various sources have suggested that a 90° bend can give pressure losses equivalent to a straight pipe up to 55 yds (50 m) long. To provide some information relating to bend losses, Broadhurst carried out experiments from which pressure losses were obtained for stowing ranges approximately 110 yds (101 m) long including a 90° bend which was located about 30 yds (27 m) along the pipe run. Tests were carried out at stowing rates of 86 to 111 tons/h (87 to 112 tonnes/h) using air flows between 2230 and 4340 ft^3/min (1.1 and 2.1 m^3/s).

For the particular stowing range chosen the equivalent straight pipe losses due to the 90° bend of 2 ft (6.03 mm) radius was found to vary between 25 and 80 yds (23 m and 73 m) according to the air flow and stowing rates.

These tests were selective and only give some idea of the relatively high losses that can occur due to the introduction of a 90° bend in a stowing line.

6.4 THE FLOW OF COMPRESSED AIR IN STOWING PIPES

Broadhursts study of air flow in stowing pipes gives the basic expression for compressible flow with friction as,

$$Vs\ dp + V\ dv/g + 4\ fv^2\ dx\ /2\ g.d. = 0 \ldots\ldots\ldots\ldots\ldots\ldots(1)$$

Because the total pressure drop in stowing is usually greater than 10% of the initial pressure, the compressibility effects which mean variations in density and velocity cannot be neglected. By rearranging equation (1) for the purpose of integration, and since the expression

$PV^n = K$, an expression which contained an unknown index of expansion 'n' was derived and given as:

$$P_1/V_1 - P_2/V_2 = \frac{n+1}{2} W^2 (2 \log_e V_2/V_1 + 4fl/d)/2gA^2 \text{ ....(2)}$$

Previously it has been assumed that the flow in stowing operations is isothermal.

Making this assumption, $n = 1$ and $P_1 V_1 = P_2 V_2 = RT$ and equation (2) becomes

$$P_1^2 - P_2^2 = W^2 RT/gA^2 (2 \text{ Log}_e V_2/V_1 + 4fl/d) \text{ ....(3)}$$

Using Pigott's(22) Curves, Broadhurst obtained an average value of .004 for the friction coefficient (f) for a 6 in (152 mm) diameter stowing pipe, and calculated the pressure drop from equation (3) for a known stowing range. The calculated pressure drops were sufficiently in excess of the experimental results to provide evidence that the expansion was not truly isothermal, and did in fact lie between adiabatic and isothermal expansion.

For experiments carried out by Broadhurst a single variable (f^1) was obtained which took into account:-

(i) Deviation from true isothermal flow

(ii) Change of velocity of the air

(iii) The pipe wall friction.

Concluding this particular investigation Broadhurst submitted a suitable equation which could be used for calculating the pressure drop likely to occur when compressed air flows along a 6 in (152 mm) diameter stowing pipe and discharges to atmosphere.

The equation states that:

$$P_1^2 - P_2^2 = 3.1 \times 10^{-5} \text{ T.L.V.}_A{}^{1.62} \text{ ....(4)}$$

Where P_1 = Initial absolute pressure in lb/ft^2

P_2 = Final absolute pressure in lb/ft^2

T = Average Air Temperature in $^\circ$F Absolute

V_A = Quantity of free Air Flowing in ft^3/min

L = Length of Pipe in ft plus the equivalent length for each bend.

6.5 PRESSURE DROPS (SOLID/AIR MIXTURE) IN PNEUMATIC STOWING PIPES

In his study of Pneumatic Transport of Stowing Materials Broadhurst carried out underground trials at several collieries to obtain some information on the pressure losses during stowing operations.

In his analysis of pressure loss the total pressure drop along a straight horizontal pipe is divided into three elements (Figure 58).

Δ_1 = Pressure Drop required to accelerate the material

Δ_2 = Pressure Drop required to overcome material friction

Δ_3 = Pressure Drop required for air flow at the same rate

To obtain a suitable formula which could be applied to pneumatic stowing, reference is made to the work of GASTERSTADT(16) who shows that for horizontal conveying:

For any given air velocity $\propto - 1/_r$ = Ga constant(k)

Where $\propto$ = pressure drop ratio $= \frac{\Delta P_s}{\Delta P_a}$

$= \frac{\text{Pressure Drop Solid Air Mixture}}{\text{Pressure Drop Air only}}$

and r = solid/air mass ratio $= \frac{W_s}{W_a}$

From this,

Total Pressure Drop $\Delta P_s = \Delta P_a + k \frac{W_s}{W_a} \Delta P_a$ ....(5)

Gasterstadt's findings, which have been confirmed by investigations from other works, shows that the constant (k) approaches a constant value for high air velocities of about 90 ft/sec (27 m/s).

Constants obtained by Hillyar Russ(17) (between 0.095 and 0.14) for high air velocities, and the calculated value of "k" from the results of Broadhurst's pressure drop tests (0.21) were sufficiently in agreement with Gasterstadt's work to justify the use of Gasterstadt's theory for pneumatic stowing operations.

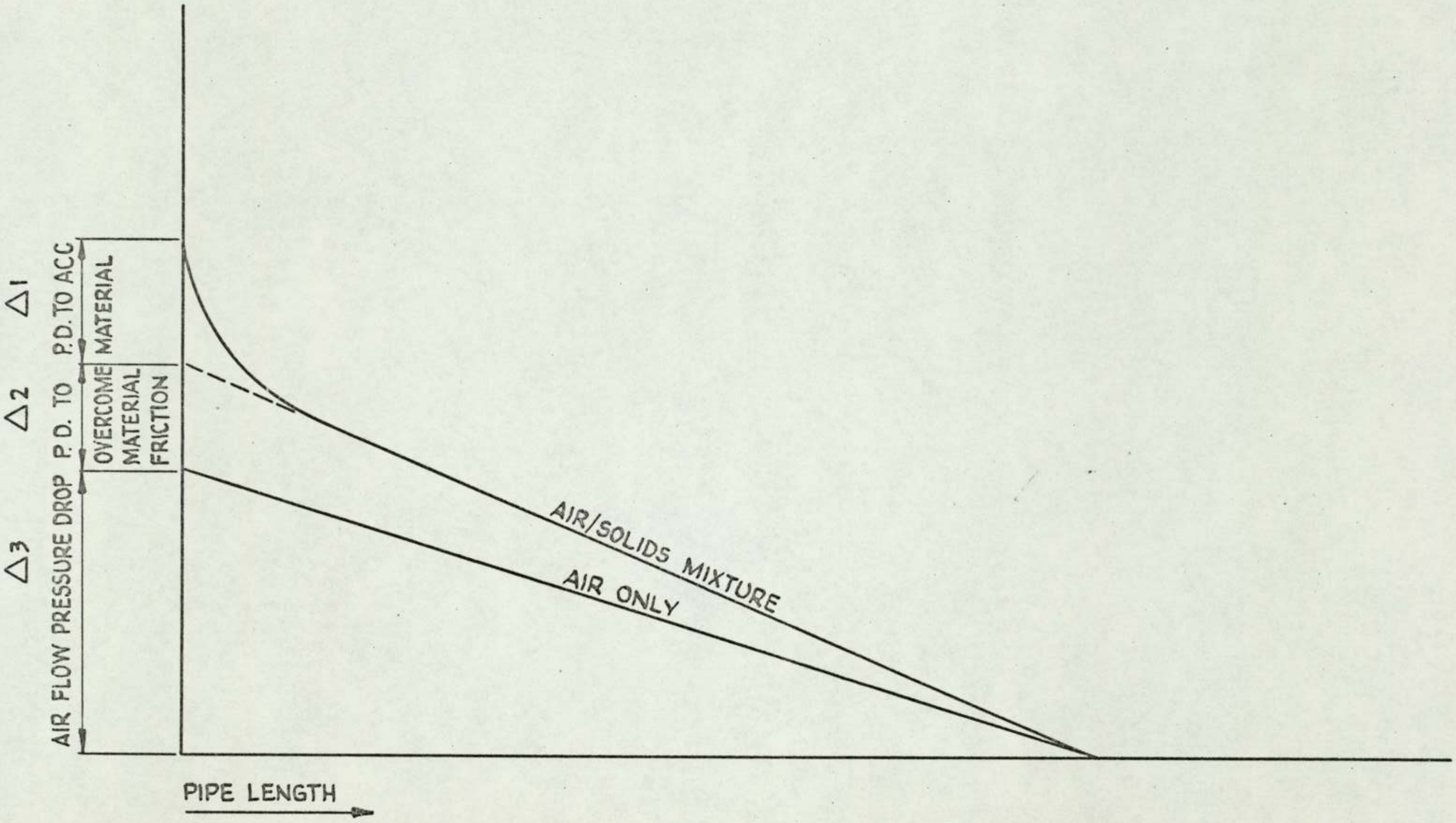


FIG 58 ANALYSIS OF PRESSURE LOSS.

From equation (4):

$$\text{Pressure Drop } P_1^2 = (3.1 \times 10^{-5}\ T.L.V._A^{1.62}) + P_2^2$$

$$\therefore P_1 - P_2 = \Delta P_s = (3.1 \times 10^{-5}\ T.L.V._A^{1.62} + P_2^2)^{\frac{1}{2}} - P_2$$

and substituting in Gasterstandt's equation

$$\Delta P_s = 1 + k\frac{W_s}{W_a}\left[(3.1 \times 10^{-5}\ T.L.V_A^{1.62} + P_2^2)^{\frac{1}{2}} - P_2\right] \quad .....(6)$$

Equation (6) was submitted by Broadhurst as a possible formula for stowing.

6.6 CONVEYING VELOCITY IN STOWING PIPE

From tests carried out by "Broadhurst" on the angle of friction for various stowing materials in 6 in (152 mm) diameter pipes, it was estimated that the angle of friction was probably between 31° and 39°, which represents a coefficient of friction of 0.65. From this investigation Broadhurst suggests that the force required to just move stowing material along a horizontal pipe is about 65% of its weight. To move the material up a gradient, or to accelerate it, requires additional force.

Croft(18) suggests that the horizontal conveying velocity is equal to or greater than the terminal velocity, and Davis(7) gives a conveying velocity known as the "mixing velocity" which exceeds the terminal velocity by about 10%.

Terminal Velocity

The terminal velocity is defined as the maximum velocity of a free falling particle at a point where the air drag force is equal to the gravitational force on the particle.

Floating Velocity

In his work, Broadhurst suggests that the terminal velocity is approximately equal to the floating velocity, which is defined as the velocity of an upward stream which floats a particle and keeps it at a constant level.

Broadhurst conducted a series of tests to determine the floating velocity experimentally. The results of these tests showed that an air velocity of 80 ft/sec (27 m/s) was sufficient to float most particles up to 3 in (76 mm) in size, but velocities as much as 100 ft/sec (30.5 m/s) or more are required for the large pieces which occupy a greater area of the pipe than do the smaller pieces.

6.7 AIR DENSITY EFFECTS ON FLOATING VELOCITY

From equations defined by "Burk and Plummer"(1) and Davies(7) (chaper 2) it can be seen that the floating velocity is approximately proportional to the square root of the particle density to air density ratio. This means that the floating velocity of 'x' atmospheres pressure will be $^1/x$ times that for atmospheric pressure.

In his conclusions, Broadhurst offers the following equation for a good approximation of floating velocity in a pipe with air at atmospheric pressure, for stowing materials.

$$V_A = 28.6\left[{}^{S}/(S+1)\right]^{\frac{1}{2}} V_W \text{ ft per sec. } \ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where S = specific gravity of the material

V_W = Terminal velocity in water-filled pipe,

and in this case the quantity of air required for conveying can be found from:-

$$V_A = 336\left[{}^{SP}/(S-1)\right]^{\frac{1}{2}} V_W \text{ STD. ft /min}$$

where P = air pressure at stower outlet in atmospheres.

6.8 CONCLUSIONS

It is not the object of the work described in this Thesis to include investigations of material and air behaviour in the stowing pipeline and pipe bends. It is however considered important to refer to the work carried out by Broadhurst and others, because any pneumatic stowing operations involve the technology of flow through pipes and bends together with the stower unit which provides the means of metering the material into the stowing pipeline. It is necessary to have some understanding of the flow characteristics in the pipeline in order to appreciate the requirements of the metering system.

The work of Broadhurst has been referred to in particular because some of his investigations have been based on full scale stowing trials carried out underground with 6 in (152 mm) diameter pipes and bends.

Reference has been made to equations submitted by Broadhurst as a means of assessing within reasonable limits the conditions for pneumatic conveying through pipes and bends. It is considered that the work referred to in this chapter offers a good contribution to the subject in question, and the formulae submitted by Broadhurst and others are offered as a means of providing information which is relevant to the investigations and development of pneumatic stowing equipment.

CHAPTER 7

ASSESSMENT AND CONCLUSIONS

7.1 FIRST AVAILABLE STOWING EQUIPMENT

The first machines, developed for mining debris disposal operations underground, used 6 in (152 mm) diameter pipes and for their air supply Rootes type positive displacement blowers having a capacity of 2000 ft^3/min (0.9 m^3/s) free air delivery were used. These machines which had a limited air pressure of about 12 lb/in^2 (83 x 10^3 N/m^2) operated with reasonable success but problems with pipe and stower blockages particularly when working with sticky materials, reduced machine utilisation and overall output to levels below that required in mining systems where rates of coalface and roadway advance were increasing with the increase in mechanisation. In addition to the operating problems being experienced, the physical size of these machines presented problems at the face end where good access to the face for men and supplies is essential.

The unreliability of the equipment mainly due to pipe and stower blockages, and the very difficult maintenance and operational problems due to its bulkiness, resulted in the need to investigate the design of smaller more reliable stowing equipment.

7.2 STOWING EQUIPMENT WITH 4 in (102 mm) DIAMETER PIPES

The Bretby Mk I Stowing Combine was developed for use with 4 in (102 mm) diameter pipes. The purpose of this was to reduce the size of the equipment, and because of the smaller pipe area a Rootes type positive displacement blower having a capacity of 1000 ft^3/min (0.5 m^3/s) free air delivery was selected for the air requirements. Previous investigations had shown that mining debris up to 2 in (51 mm) in size could be transported along a 4 in (102 mm) diameter pipe.

Initial trials with a prototype machine proved reasonably successful and outputs up to 30 tons/h (30.3 tonnes/h) were achieved. Seven of these machines were manufactured but attempts to operate these underground where the mining debris was particularly sticky, proved unsatisfactory because of the constant blockages of the stower unit and stowing pipes. It was also found that the machines were in general only achieving outputs up to 25 tons/h (25.4 tonnes/h), which was completely inadequate particularly for the rates of debris disposal envisaged for future mining systems.

7.3 RECOMMENDATIONS FOR FURTHER DEVELOPMENT

Investigations into mining debris disposal requirements for future mining systems and an assessment of the operating experience gained with the machines operating with both 4 in (102 mm) and the 6 in (152 mm) diameter pipes concluded that if pneumatic stowing was to form part of a mining system, a stowing machine capable of continuous stowing at a rate of 60 tons/h (60.3 tonnes/h) and having the ability to handle peak loadings up to 100 tons/h (101.6 tonnes/h) was necessary. Moreover the size of such a stowing machine would have to be limited to a maximum height of 24 in (603 mm) this being the same height as the small stowing unit developed for the Bretby Mk.I Stowing Combine.

7.4 STOWER DEVELOPMENT

7.4.1 STOWER OUTPUT

To meet the limitations on size and to eliminate the causes of stower blockages, the stower test rig was designed with a straight through air passage which eliminate the internal bends which were a feature of existing stowing machines. This major change in design together with the introduction of a "DROP" at the bottom of the stower greatly improved the performance of the stowing trials which were carried out. During the investigations with the test rig it was also found that the shape and size of the inlet and outlet pipes to the stower greatly affected the performance and reliability of the stower.

After subsequent trials and development during which various shapes and sizes of inlet and outlet pipes were investigated, stowing reliability was achieved and outputs up to 100 tons/h (101.6 tonnes/h) were recorded. Taking into account the stoppages that did occur during the trials due to jamming of the stower by foreign metallic bodies in the mining debris, and the varying rates of outputs recorded it is concluded that the design of stower finally submitted (Figures 23 and 25) has a capacity of 60 tons/h (60.6 tonnes/h) with the capability of dealing with peak outputs up to 100 tons/h (101.6 tonnes/h). These figures are based on a stowing pipe range of 60 yds (55 m) including one 90° bend, and mining debris having a density between 75 and 90 lbs/ft^3 (1202 and 1442 Kg/m^3).

7.4.2. PRESSURE LOSSES ACROSS STOWER

Reduction in pressure across the stower unit was also the aim of the trials and development programme. The straight through design of stower and the uniform shape of inlet and outlet pipes made substantial improvements in the pressure losses across the stower. The first trials which were carried out with inlet and outlet pipes having cross sections which changed between pipe ends produced a stower pressure loss of 7 lbs/in^2 (48×10^3 N/m^3). This was considered too high and after the trials and development discussed in detail in chapter 4, the pressure drop across the stower unit was finally reduced to a maximum of 2.5 lb/in^2 (17×10^3 N/m^3). This was considered of paramount importance and was achieved by the streamlining of the air passage through the stower, and the lowering of the air passage below the periphery of the rotor at the stower bottom.

7.4.3 AIR FLOW

It has been mentioned previously that the use of a Rootes type blower in stowing operations has a great advantage over a compressor on the grounds of physical size and cost. During the trials programme the air supply was obtained from a compressor having a capacity of 2,250 ft^3/min

(1.1 m^3/s) free air delivery and capable of operating at pressures up to approximately 25 lb/in^2 (172×10^3 N/m^2). As far as the air requirements are concerned it was found that the stower capacity discussed previously (7.4.1) was obtained with pressures up to 13 lb/in^2 (90×10^3 N/m^2) with occasional peaking at 15 lb/in^2 (103×10^3 N/m^2). This pressure is on the limits of a Rootes type blower.

As far as air flow is concerned all the investigations using the prototype stower test rig were conducted at flows of 2,250 ft^3/min (1.1 m^3/s) free air delivery. To provide some information on the necessary air velocities to prevent material build up and eventual pipe blockages it was decided to obtain information on air flows below 2,000 ft^3/min (0.9 m^3/s) by building a full scale model of the stower and carrying out a study of the behaviour of material metered from the stower into the stowing pipeline.

7.5 STOWER MODEL

7.5.1 MATERIAL FLOW OBSERVATIONS

Material movement at the stower outlet pipe and at a point 25 ft (7.6 m) downstream indicated that for air flows up to 144 ft/s (44 m/s) material flows in the intermediate region, where the majority of the solids settle on the bottom of the pipe and form beds that slide along the pipe. Only a small percentage of the particles are conveyed in the air stream. This flow condition applied to all the types of materials used during the stower model tests. Generally speaking material conveyance took place on the lower half of the conveying pipe with the solid concentration varying with the air flow and type of material used. Measurement of material distribution and material concentration was not possible with the equipment, but the assessments made from the model tests indicate that conveying conditions are more difficult at the stower outlet than downstream, and air flows above 144 ft/s (12 x 10 m/s) will be necessary to avoid material settlement and eventual pipe blockage. The application of existing equations for obtaining floating velocities can only be used in a limited capacity because the formulae available include particle factors whose value for stowing debris is difficult to define. Particle factor values for stowing debris could only be assessed over a fairly wide range covering all shapes and sizes up to the limit set by the pipe diameter.

7.5.2 MATERIAL CHARACTERISTICS

In an attempt to obtain information on the effects of material characteristics on the behaviour and movement of solid particles, a study of research carried out by other workers was investigated and is discussed in chapters 2 and 6. It was found that the theories and formulae submitted were based on known particle size and shape, and in some cases particle factors related to spheres having the same volume have been suggested. In other cases particle size factors have been derived by considering the surface areas, perimeters, and circularity of irregular shaped particles. Other factors used to obtain floating velocities include values for the average diameter and cross sectional area of the particle.

The difficulty of selecting a size factor which would satisfy the constantly changing conditions during stowing operations make it difficult to apply directly the various formulae available for estimating air flow requirements. The consideration given to the equations of Davies and Dallavalle for estimating floating velocities of the stower model materials was only able to provide data which was proven over a much smaller range of material size and concentration than those being examined.

7.5.3 PARTICLE SETTLING POSITION

Other workers have carried out investigations to establish the settling position of solid shapes and the influence it has on fluid motion. Flat discs, cubes and cylinders have been considered and it has been suggested that with shapes such as these a settling position can be assumed.

Observations made during the stower model tests clearly shows that there was no common settling position which could be assumed. The particles travelled from the stower and along the pipes in complete turbulence due to the mechanical interference of particles with another. When considering stowing materials where shapes and sizes vary and the proportion of fines to solids also varies, any assessment of settling position would be extremely difficult. It is therefore concluded that the resistance exhibited by stowing material and an assessment of the energy requirements for conveying can only be established from data obtained by experimentation using mining debris.

7.5.4 OPTIMUM ROTOR SPEED

Data to assess the optimum speed of the stower rotor was based on equating the times to clear material from a rotor pocket to the available rotor pocket clearance time determined by the rotor speed. These investigations which were conducted for air flows up to 1800 ft^3/min ($0.83\ m^3/s$), indicated that the optimum rotor speed for air flows around 2,000 ft^3/min ($0.9\ m^3/s$) is somewhere between 58 and 68 RPM. These conclusions are in general agreement with the results obtained with the stower test rig which produced the best results with a stower rotor speed of 60 RPM.

7.5.5 METERING EFFICIENCY

The output efficiency of the stower test rig was assessed from the relationship between the measured output, and the calculated output which was based on the volume of the rotor pockets and the specific weight of the material. Reasons for the apparently low efficiency of the stower test rig (about 60%), and in other stowing operations underground as low as 50%, were never fully understood.

During the stower model investigations it was found that the maximum volume of material which filled a pocket varied considerably with the materials used. An attempt was made to obtain data which would provide a more accurate assessment of the metering efficiency.

The measured weights and volumes of material together with the percentage volume of pocket occupied by the material were compared. The results show that the actual volume of the rotor pocket occupied by material varied between 52 and 90%.

It was not possible to carry out a full scale investigation to study in detail the effects of material size, shape, and specific weights on the metering efficiency, but the observations made during the stower model investigations indicates that the apparent overall efficiency of stowing operations is greatly affected by the characteristics of the mining debris, and in assessing the efficiency of stowing the maximum quantity of material that can fill a rotor pocket should be the basis of efficiency and not the calculated rotor pocket volume as used in the past.

An assessment of the efficiency of a stowing unit can only be based on experimental data which evaluates full rotor pockets with material volumes.

7.6 PRESSURE LOSSES IN PNEUMATIC STOWING PIPES

The useful application of Broadhurst's formulae for calculating the pressure losses due to material and air flow in 6 in (152 mm) diameter stowing pipes was assessed by comparing the pressure requirements recorded during the trials with the stower test rig with the calculated figures obtained from Broadhurst's formulae.

The complete stowing circuit used during the trials consisted of 60 yds (55 m) of 6 in (152 mm) diameter stowing pipes which included one 90° bend having a radius of 18 in (457 mm). Air was supplied to the stower from the compressor through 6 in (152 mm) flexible piping approximately 18 ft (456 mm) long. Due to the layout of the equipment the flexible piping could not be straight in its entire length, and for purposes of calculation the bends in the line have been assessed at the equivalent of two 90° bends.

(1) Pressure Losses in Stowing Pipe Range

Considering the stowing circuit operated during the final stower test rig trials.

Stower output = 70 tons/h
Length of straight 6 in (152 mm) diameter pipes = 180 ft
Equivalent length of straight pipe for the 90° bend (ref. 6.3) = 80 ft
Average air temperature (absolute) during stowing(t) = 75 + 460°F
= 535°F
Constant 'k' for pneumatic stowing (Ref. 6.5) = 0.21
Quantity of free air flowing (V_A) = 2250 ft^3/min
Initial absolute pressure = P_1 lb/ft^2
Final absolute pressure (Atmospheric) = (P_2)
= 211.8 lb/ft^2
Solid/Air Mass Ratio W_s/W_a = 15.4

Broadhurst's Equation states:

$$P_1 - P_2 = 1 + k\frac{W_s}{W_a}\ (3.1 \times 10^{-5}\ T.L.V_A{}^{1.62} + P_2{}^2)^{\frac{1}{2}} - P_2$$

Solid/Air Mass ratio W_s/W_a = 15.4
Substituting values in the formula $P_1 - P_2$ = 23.2 lb/in^2 Abs.
Pressure Loss (Stowing pipes) = 8.5 lb/in^2 (Gauge)

(2) Total Pressure Loss in Stowing Circuits

The loss of air pressure in the flexible piping between the compressor and the stower was estimated at approximately 3 lb/in^2 (21×10^3 N/m^2) (reference 4.5).

∴ Total Pressure drop in circuit.

= Loss in stowing pipe + Loss in air supply pipe + Loss across the stower

= 8.5 + 4.5 + 3

= 16 lb/in^2 (110×10^3 N/m^2)

From the average curves shown in Figure 25 it can be seen that the total back pressure measured during the stower trials for an output of 70 tons/h (71 tonnes/h) was about 12 lb/in^2 (83 x 10^3 N/m^2) with the pressure along the pipe and bend approximately 9 lb/in^2 (41 x 10^3 N/m^2). Taking into account the assumptions made regarding losses in bends, and the relatively high value of 'k' suggested by Broadhurst the results are sufficiently in agreement to justify the use of Broadhurst's formulae for estimating pressure losses in 6 in (152 mm) diameter pneumatic stowing pipes.

7.7 AIR FLOWS

The selection of the minimum air requirements to provide efficient conveying of material where material settlement and pipe blockages are non-existant, presents a more difficult problem. The work of Broadhurst and others has suggested formulae for calculating the floating velocities or irregular shaped particles, but their direct application to pneumatic stowing is not practical because a single value of a particle size factor cannot be determined for run of mine debris.

Broadhurst has suggested that air velocities up to 100 ft/s (30.5 m/s) was sufficient to float most particles up to 3in (76 mm) in size. The results of the trials carried out with the stowing test rig and the subsequent investigations conducted with the stower model indicates that the air requirements for pneumatic stowing using 6 in (152 mm) diameter pipes should not be less than the 2,250 ft^3/min (1.1 m^3/s) free air delivery used during the stower test rig trials. This air supply gives air velocities in the stowing pipe up to 180 ft/s (55 m/s).

There is sufficient variation between Broadhurst's results and those obtained from the studies carried out with the stower model to suggest that further investigations into conditions necessary for efficient conveying, is required before the formulae submitted by other investigators for calculating floating velocities can be used with reliability.

7.8 PNEUMATIC STOWING CONSIDERATIONS

The research and development which has been discussed in this thesis has shown that the selection of stowing equipment for mine debris disposal underground must be based first on the stowability of the material in question. Simple tests should be carried out to determine the stickiness of the material (Ref. 6.2). The possibility of increasing the moisture content of the material by the addition of water may have to be considered. The general size of the material and the proportion of fines that may be present can greatly affect the stowing output in terms of tons/h (tonnes/h) and some assessment has to be made on the ability of the material to settle and compact into the stower rotor pocket before any realistic output can be estimated. Pressure losses in a stowing system can be determined by the use of formulae submitted by Broadhurst, but at the present time experimental data suggests that the air flow requirements for 6 in (152 mm) diameter stowing pipes cannot be less than 2,250 ft^3/min (1.1 m^3/s) free air delivery. This means that for the stowing system considered during these investigations, the use of a Rootes type blower is questionable, and the inclusion of a compressor as used in the stower trials is necessary.

The use of pneumatic stowing equipment for mining debris disposal underground will probably be determined initially by the economics of such a system. At the present time the necessity to use a compressor for the air supply will probably dismiss pneumatic stowing operations because of the high capital cost involved. It is suggested that if stowing operations are to be competitive economically with other methods of mining debris disposal, it will be necessary to develop a Rootes type blower capable of operating at pressures up to about 17 lb/in^2 (117×10^3 N/m^2) with a free air delivery around 2,200 ft^3/min (0.9 m^3/s).

The work described in this Thesis has produced the design and operating conditions of a reliable pneumatic stowing machine for consideration in future Mining Systems. The limitations of the machine and the air requirements have been established from the trials and model investigations carried out. The work of other investigators, and the observations made with the stower model has shown the importance of establishing the stowability of mining debris, and the limited investigations on material behaviour has been used to establish the flow conditions necessary to avoid material build up and blockages.

Consideration of the stower for use in pneumatic stowing operations underground will ultimately depend on the availability of a Rootes type blower which will meet the air flow and pressure requirements. The low pressure drop across the stower which has been achieved through these investigations, makes it possible to consider the development of a blower which would meet the requirements of the stowing circuit considered.

ACKNOWLEDGEMENTS

The invaluable help, guidance, and supervision of Professor A.J. Ede of the University of Aston in Birmingham, Mechanical Engineering Department, during the investigations and writing of this thesis is gratefully acknowledged.

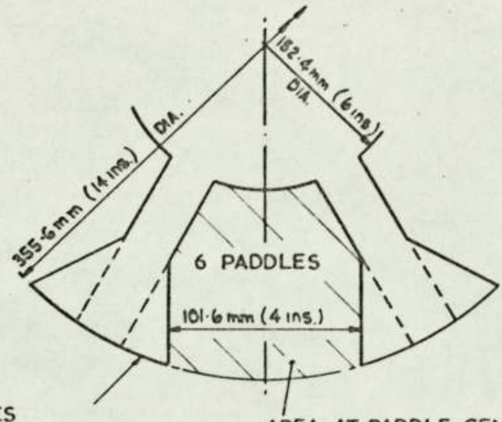
The author also wishes to thank the National Coal Board for their generosity in granting facilities and finance without which this project would not have been possible.

The valuable assistance of collegues at the National Coal Board Mining Research Establishment is acknowledged, and of particular value has been the information so readily made available by Dr. P.H. Broadhurst of the Academic VI Form Centre, Devonshire, Bermuda.

The author thanks the National Coal Board for its permission to present this thesis. The views expressed in it however, are not necessarily those held by the National Coal Board.

APPENDIX 1

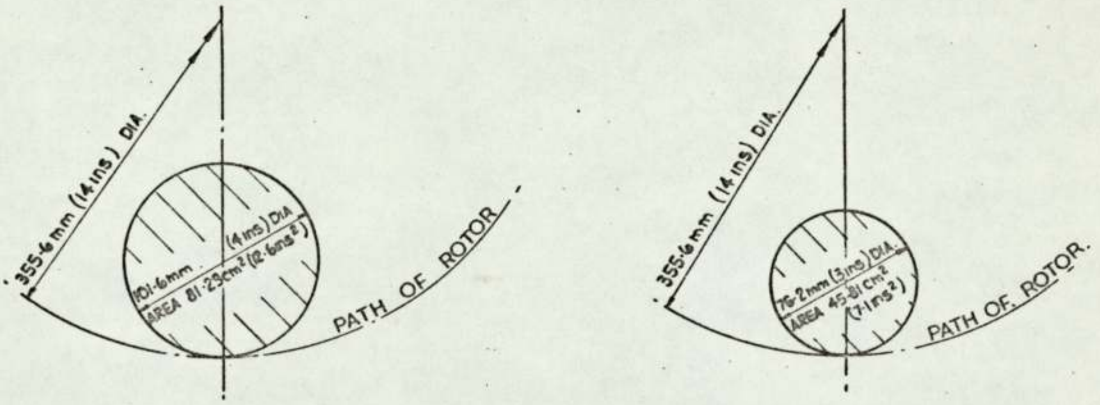
DEVELOPMENT STAGES OF PROTOTYPE STOWER



TAPERED GUIDE PIECES
AT EACH END OF ROTOR
(APPROX. 127mm (5ins) LONG)

AREA AT PADDLE CENTRE 101·94cm² (15·8ins²) SQ.INS.
AREA AT PADDLE ENDS 69·03cm² (10·7ins²) SQ.INS.

(a) ROTOR.



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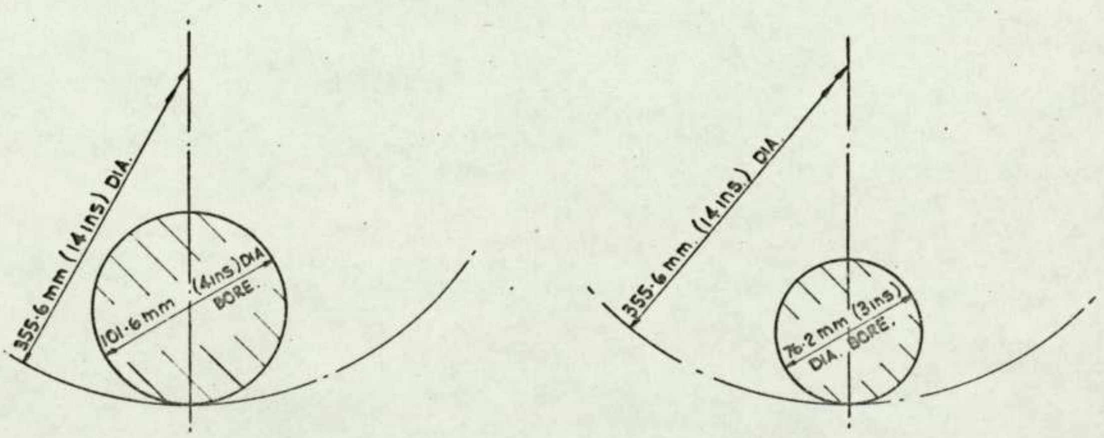
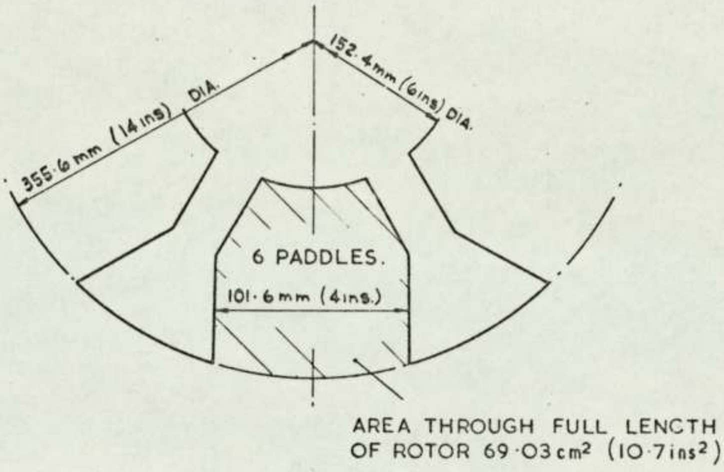
(c) END COVER (INLET.)



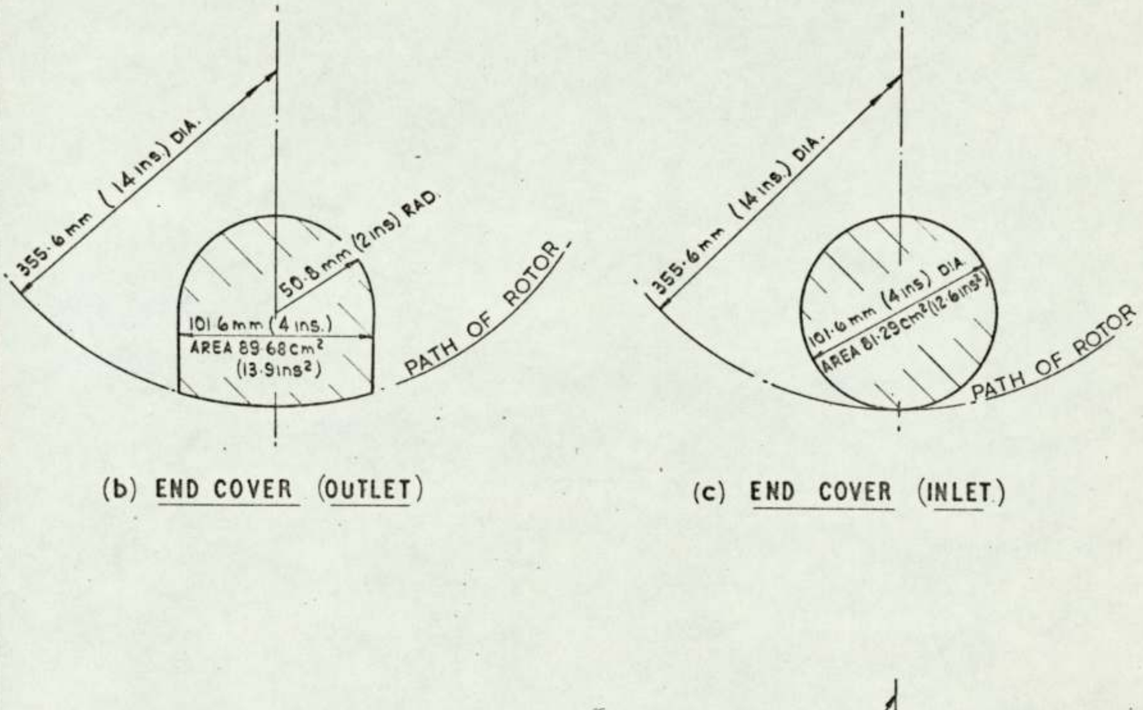
(d) OUTLET PIPE.

(e) INLET PIPE.

STAGE 1. STOWER DESIGN FEATURES.



(a) ROTOR.



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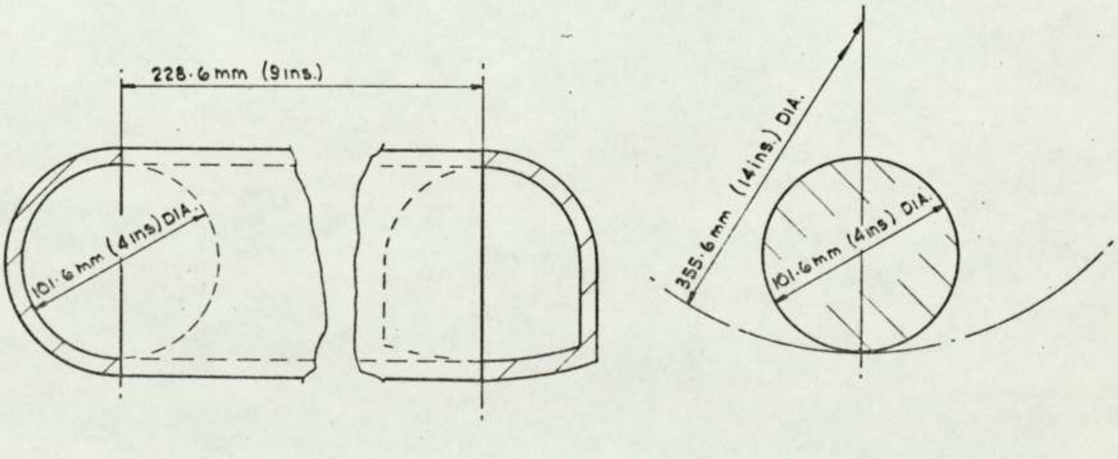
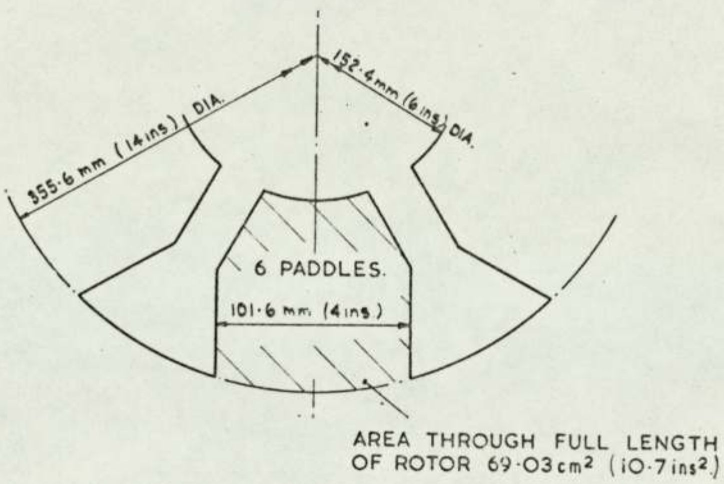


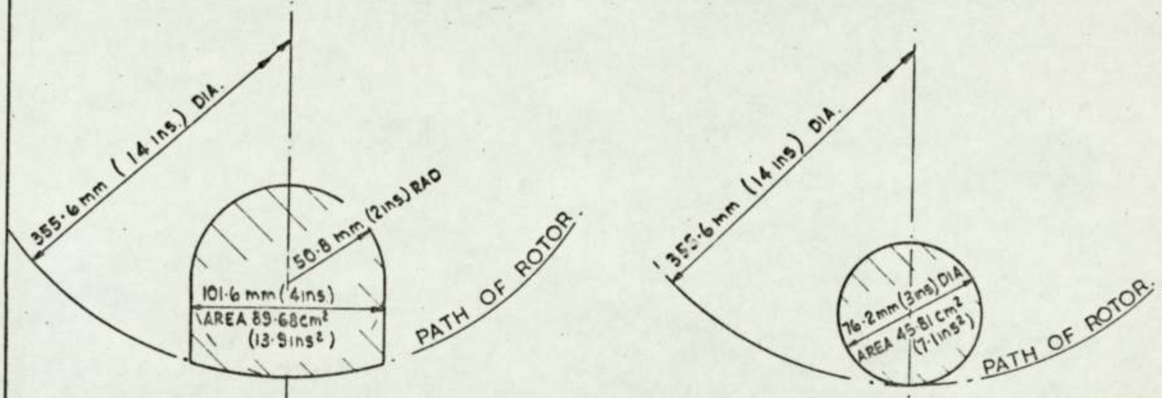
(d) OUTLET PIPE TAPERED FROM END COVER HOLE SECTION TO 101·6 mm. (4 ins.) DIA. BORE.

(e) INLET PIPE.

STAGE 2. STOWER DESIGN FEATURES



(a) ROTOR.



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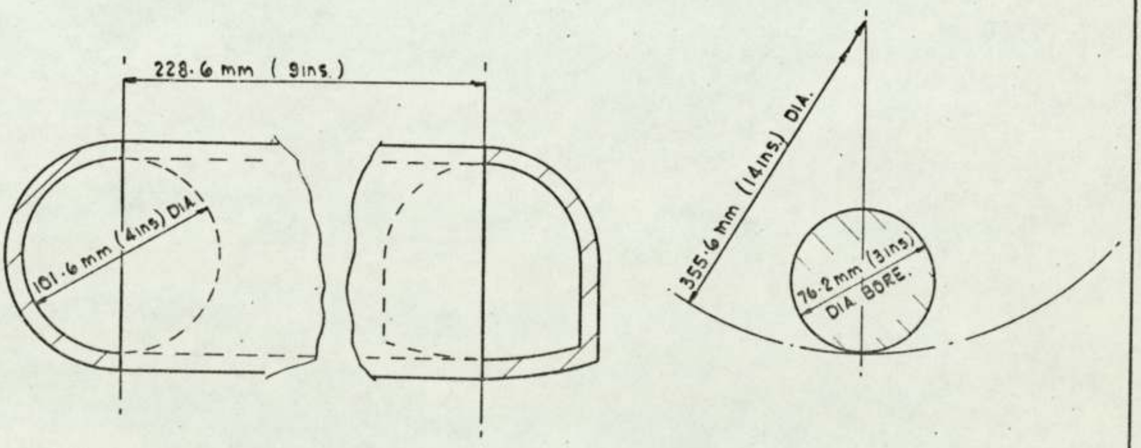
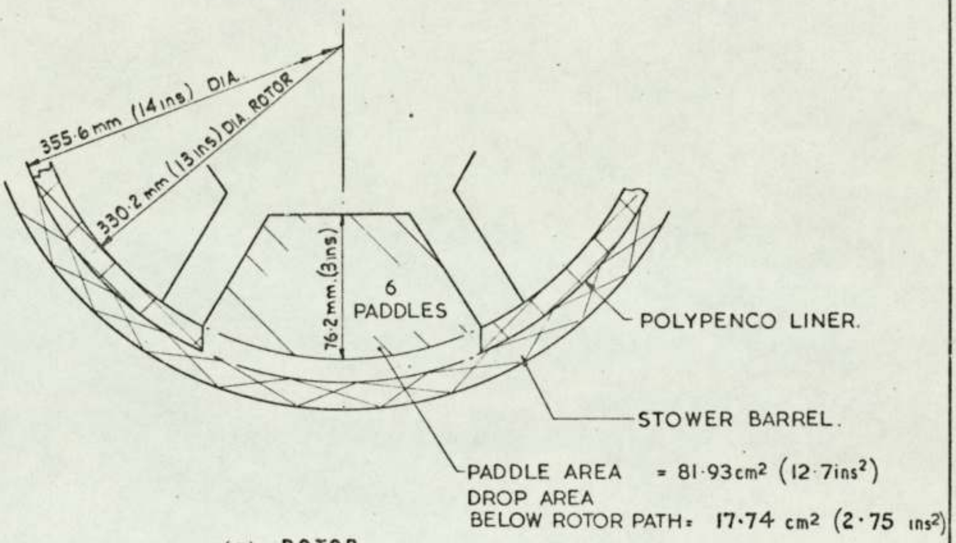
(c) END COVER (INLET)



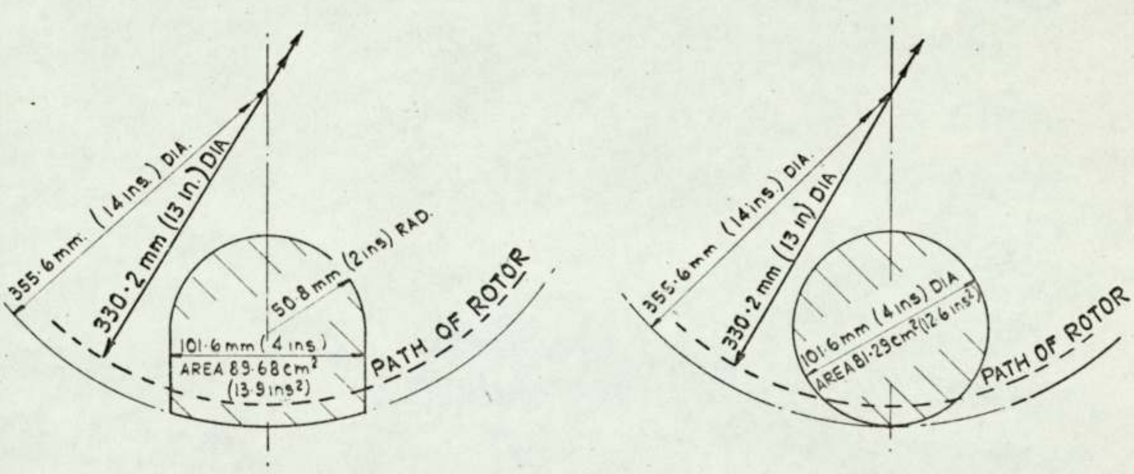
(d) OUTLET PIPE TAPERED FROM END COVER HOLE SECTION TO 101·6 mm. (4 ins.) DIA. BORE.

(e) INLET PIPE.

STAGE 3. STOWER DESIGN FEATURES



(a) ROTOR.



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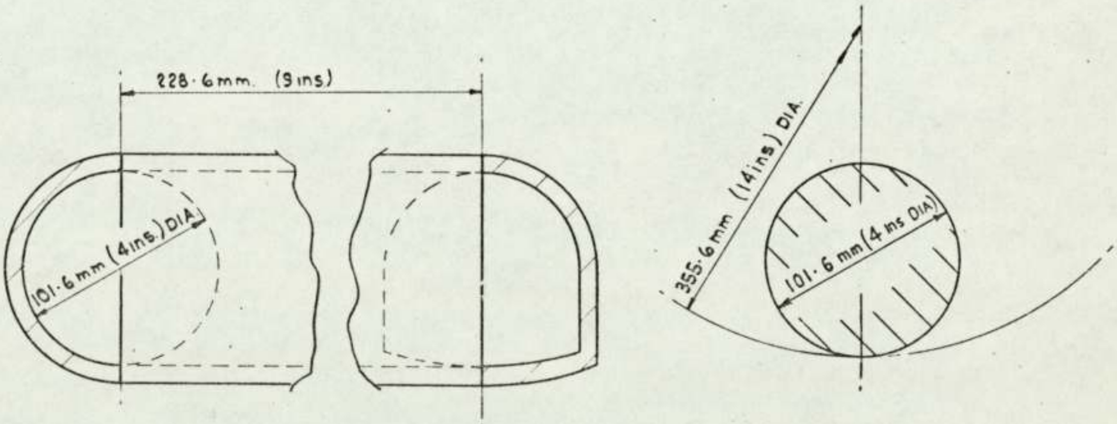
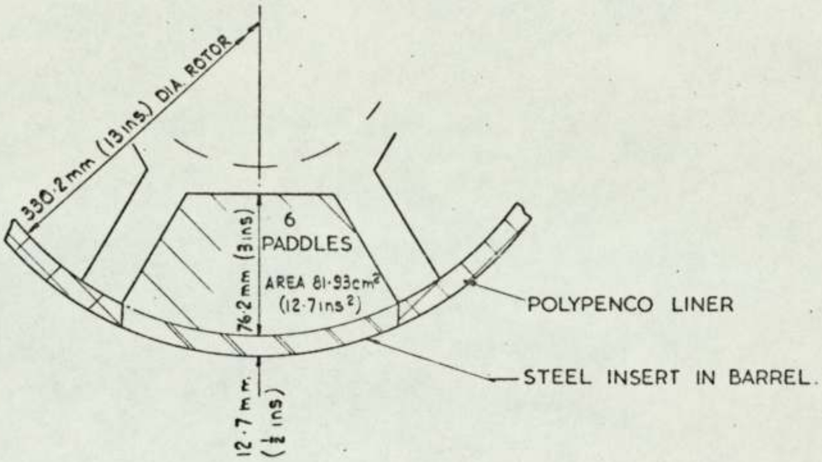
(c) END COVER. (INLET)



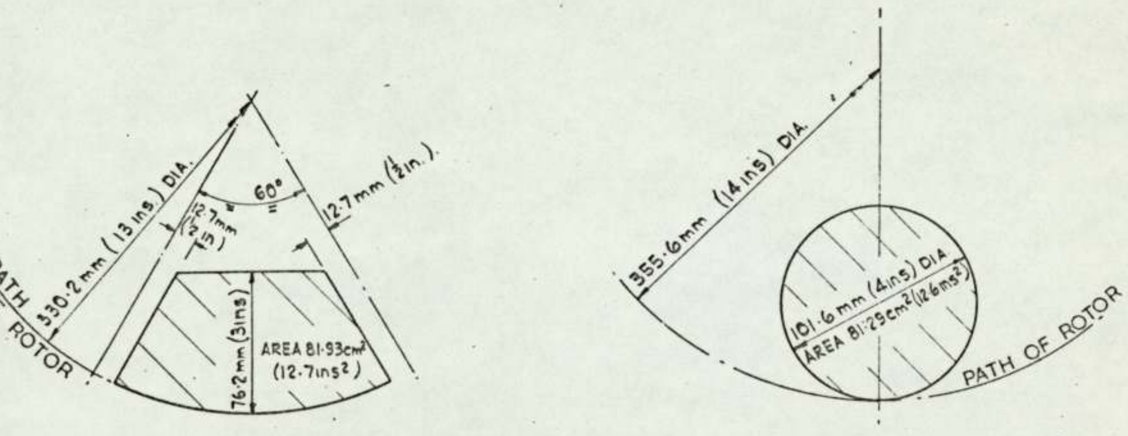
(d) OUTLET PIPE TAPERED FROM END COVER HOLE SECTION TO 101·6 mm. (4 ins) DIA. BORE.

(e) INLET PIPE.

STAGE 4 STOWER DESIGN FEATURES



(a) ROTOR.



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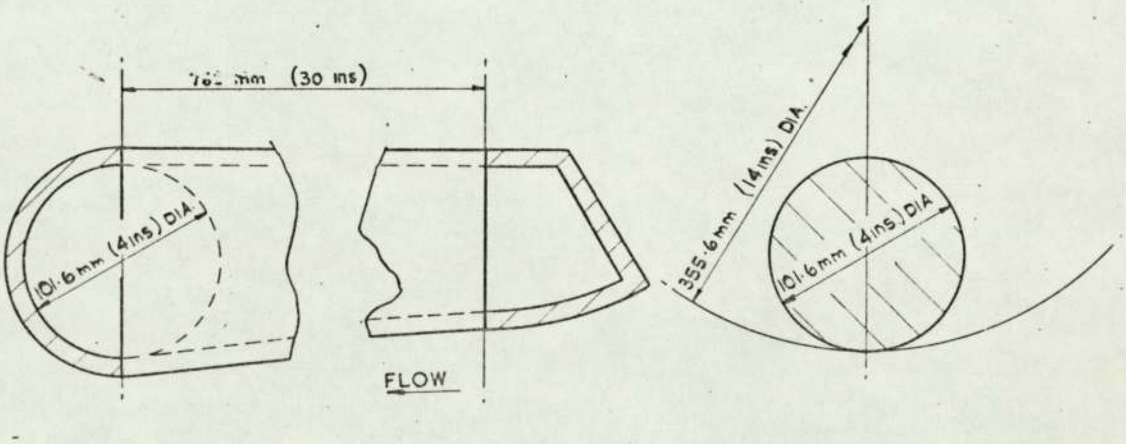
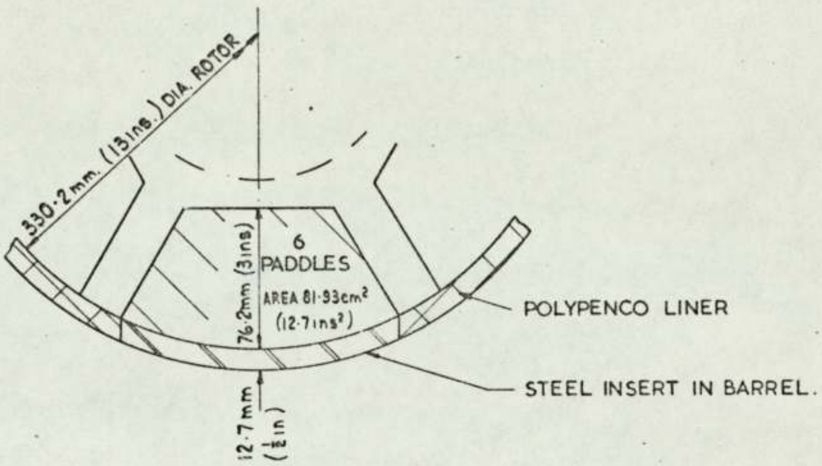


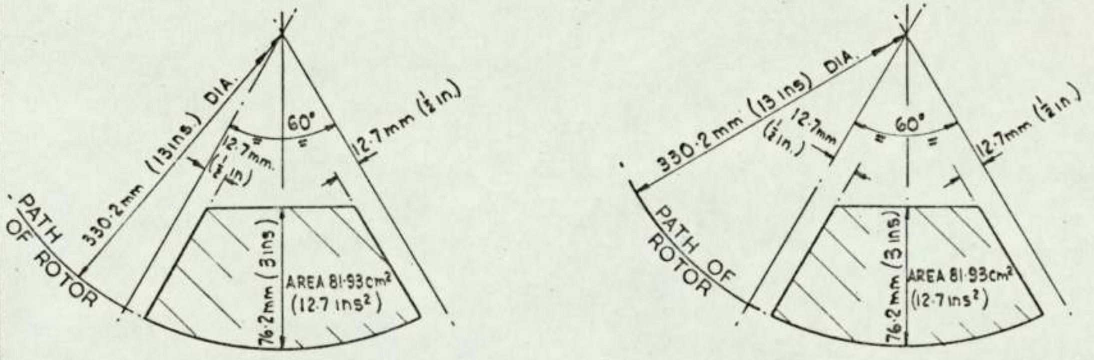
(d) OUTLET PIPE

(e) INLET PIPE

STAGE 5. STOWER DESIGN FEATURES.



(a) ROTOR



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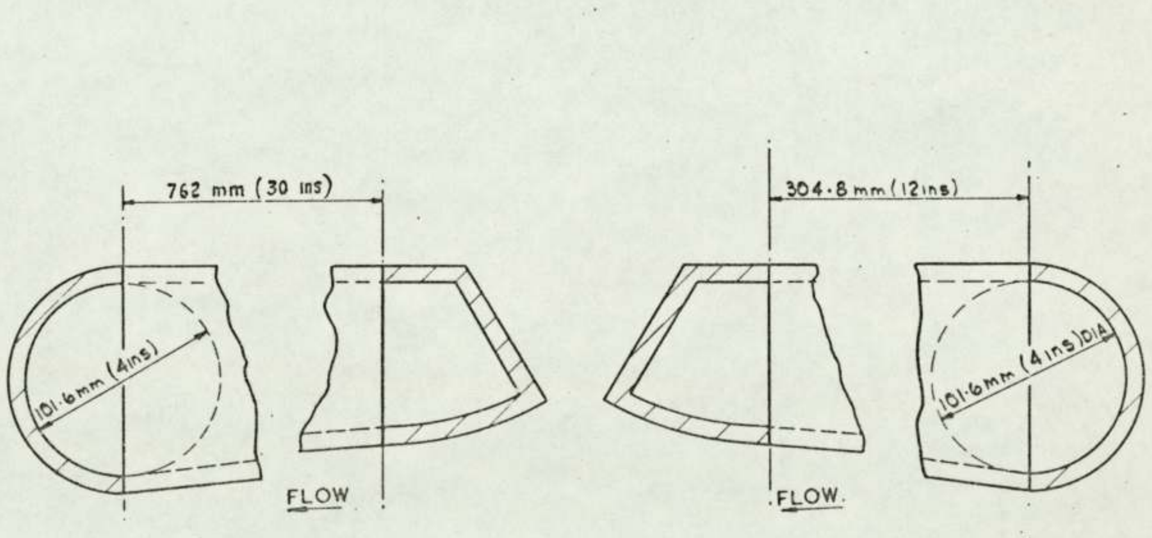
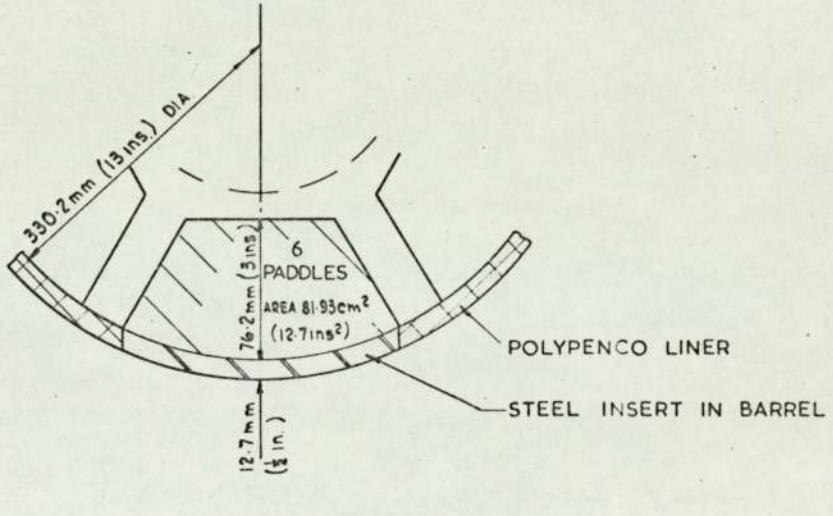


(d) OUTLET PIPE (e) INLET PIPE.

STAGE 6. STOWER DESIGN FEATURES



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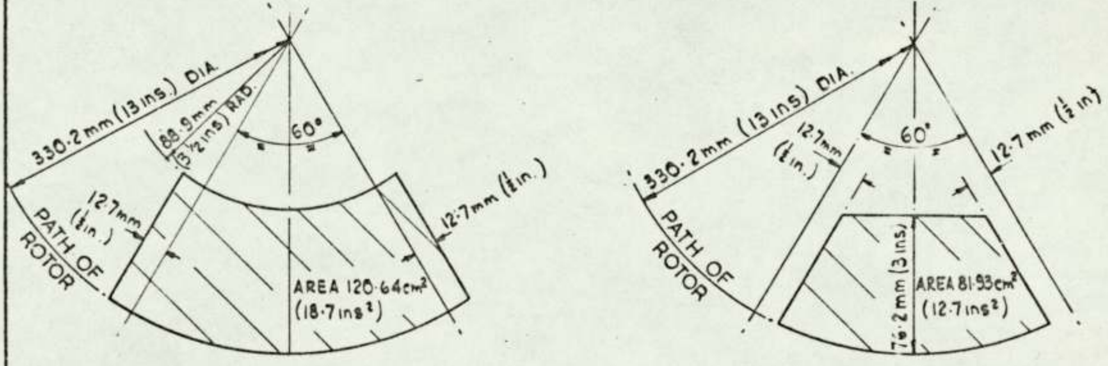

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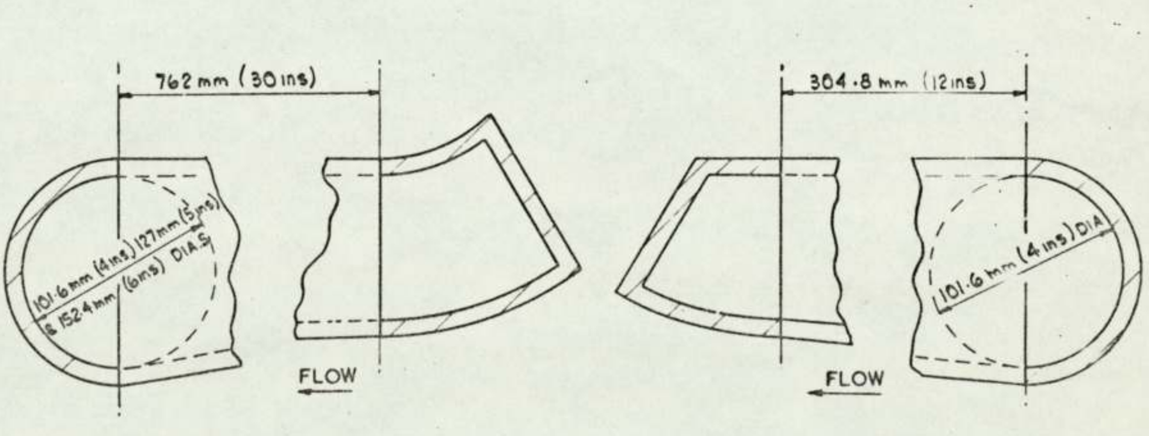
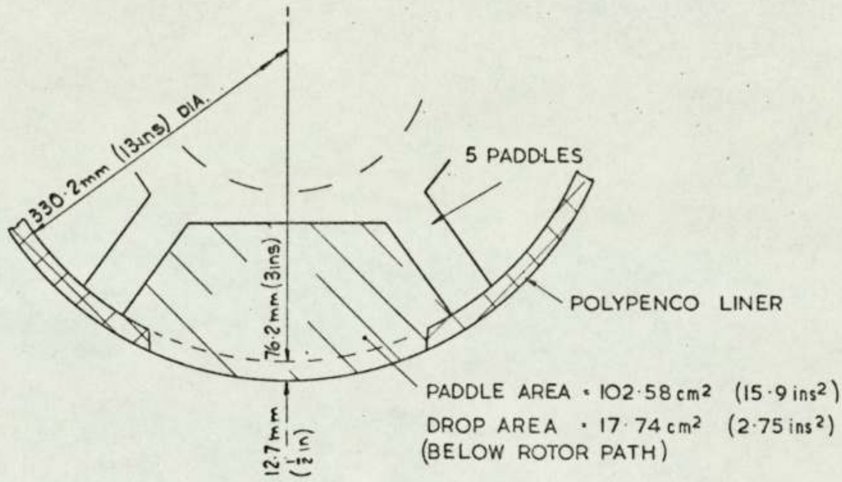
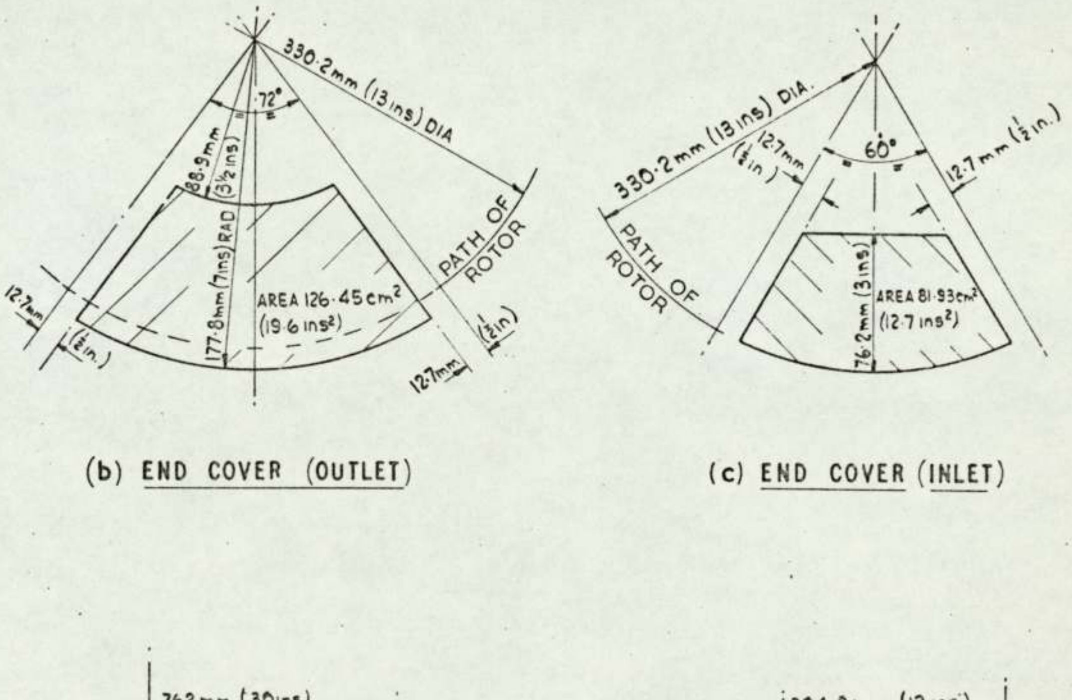


(d) OUTLET PIPES. (e) INLET PIPE

STAGE 7. STOWER DESIGN FEATURES.



(a) ROTOR



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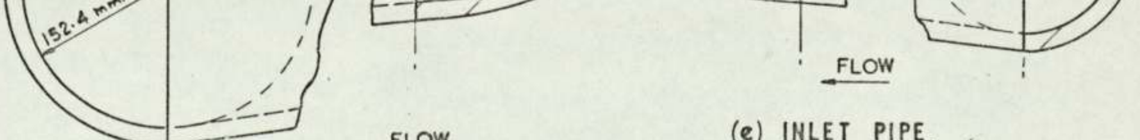
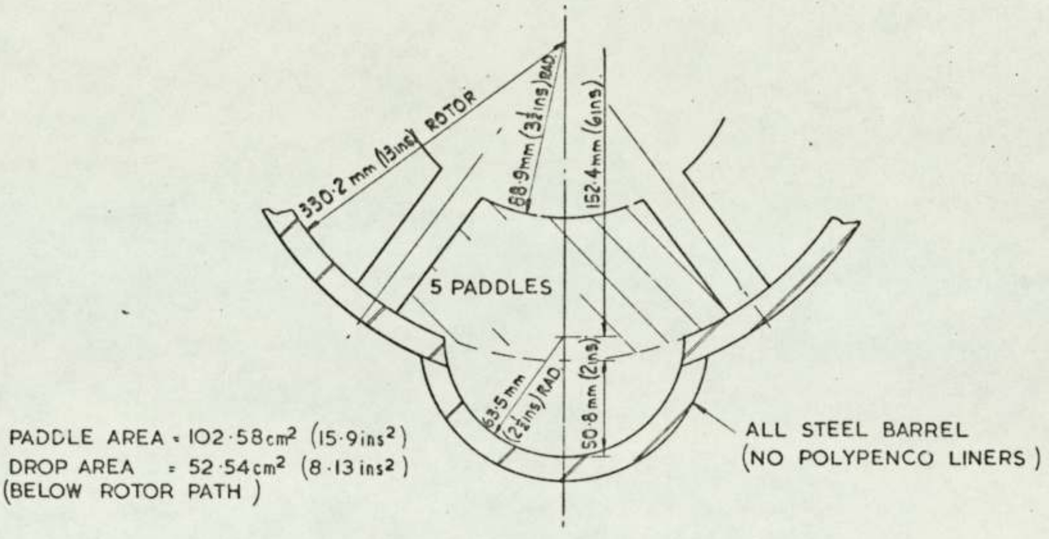


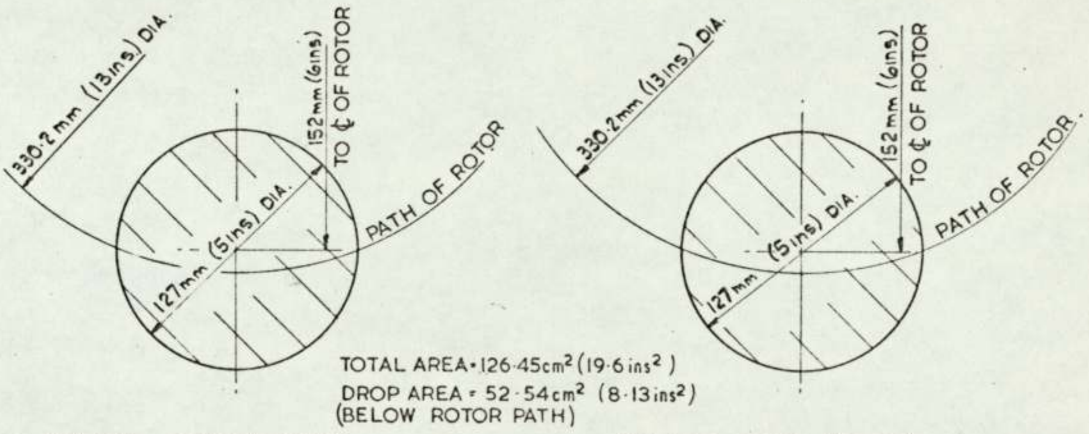
(e) INLET PIPE.

(d) OUTLET PIPE.

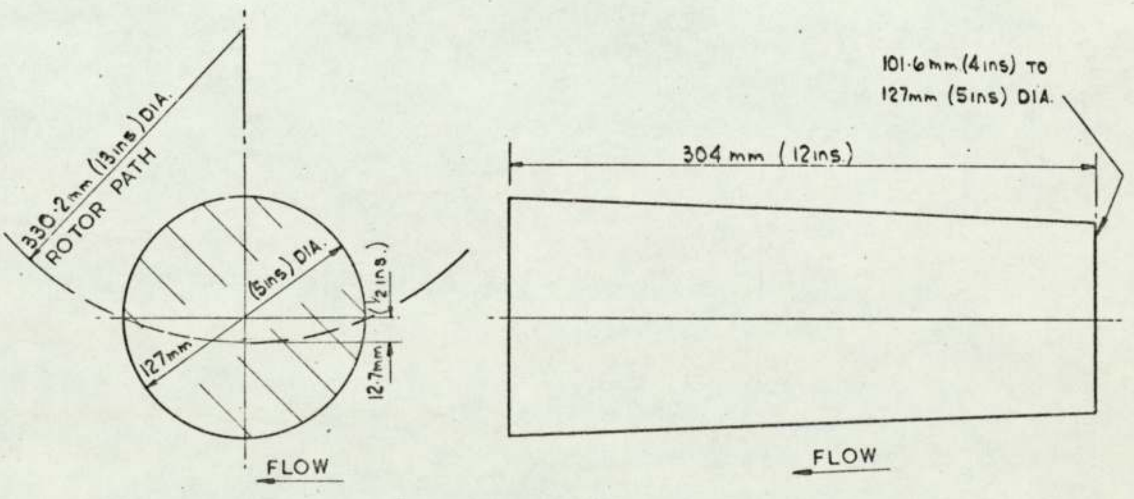
STAGE 8. STOWER DESIGN FEATURES



(a) **ROTOR**

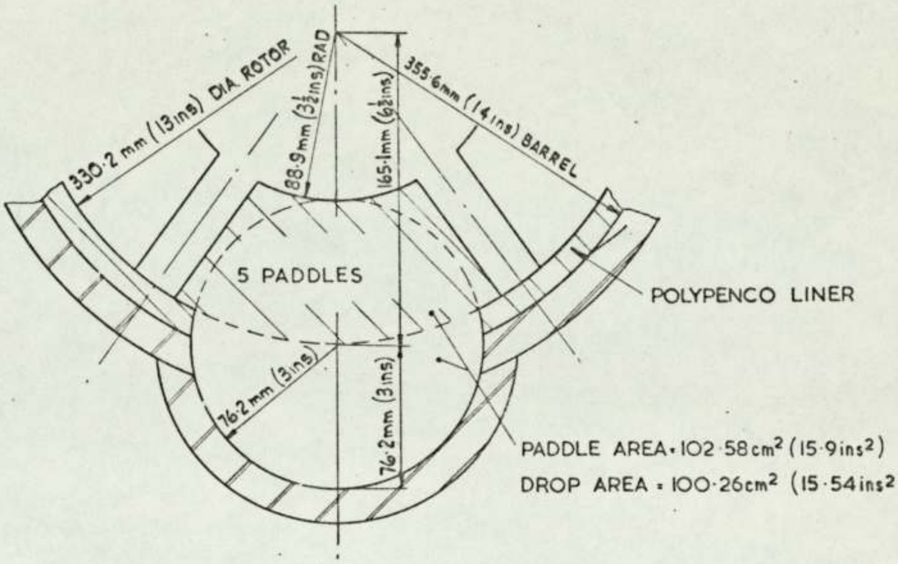


(b) **END COVER (OUTLET)** (c) **END COVER (INLET)**

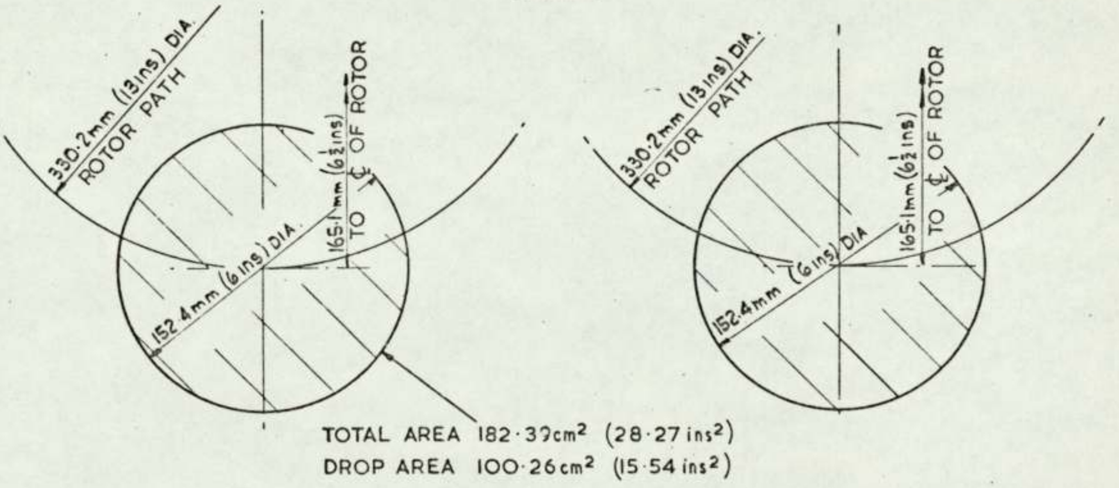


(d) **OUTLET PIPE** (e) **INLET PIPE**

STAGE 9. STOWER DESIGN FEATURES



(a) ROTOR



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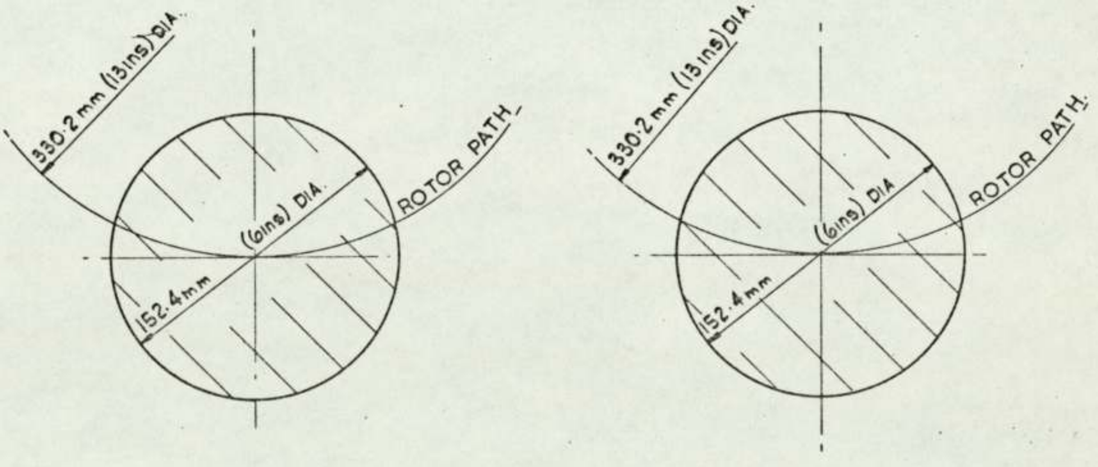


(d) OUTLET PIPE.

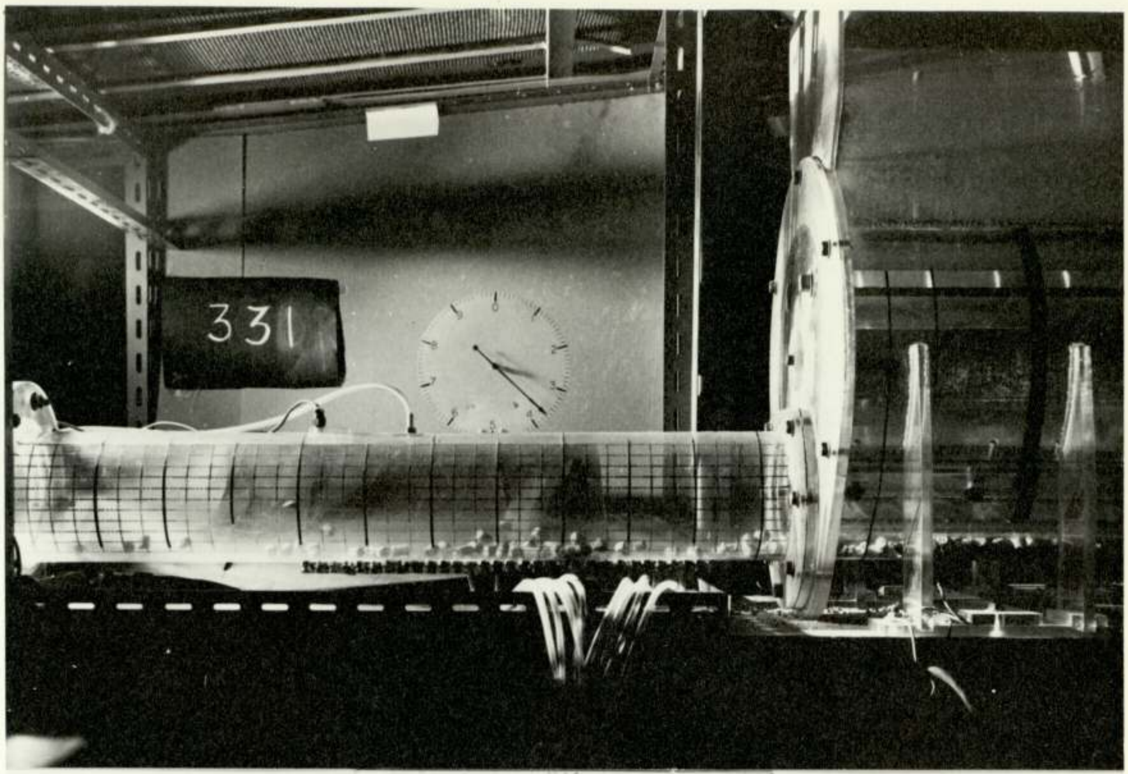
(e) INLET PIPE.

STAGE 10. STOWER DESIGN FEATURES

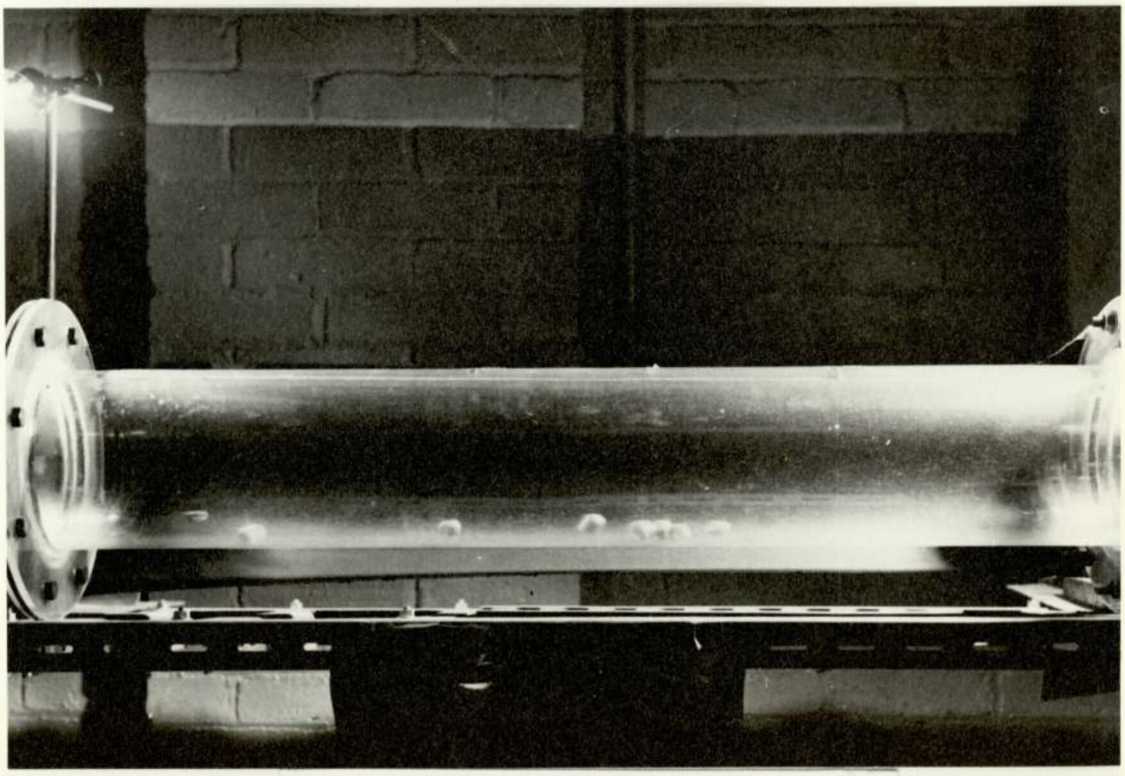
APPENDIX 2

STOWER MODEL TEST RIG

MATERIAL FLOW OBSERVATIONS

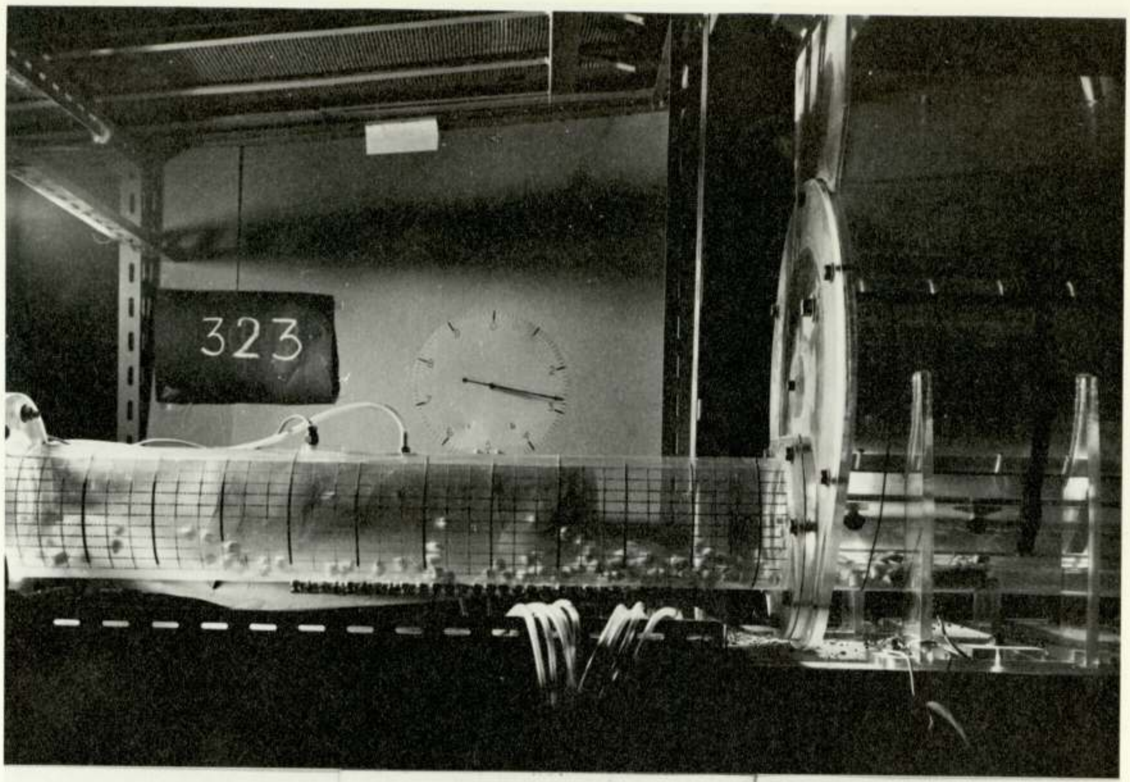


a) STOWER OUTLET

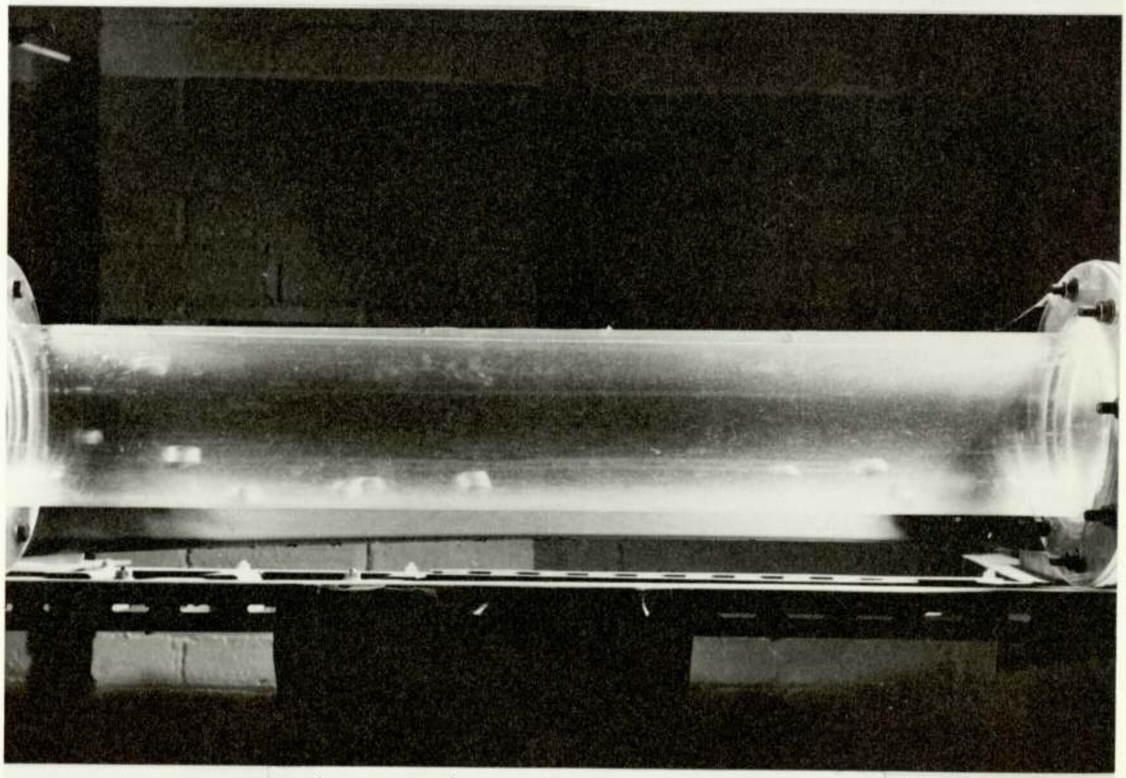


b) 25ft (7·6m) DOWNSTREAM

FIG. 59 1/2 in. (13 mm.) PENCILS
COVERED POCKET (8·6% FULL)
AIR FLOW 800 ft ^{3}/min. (0·4 m ^{3}/s)

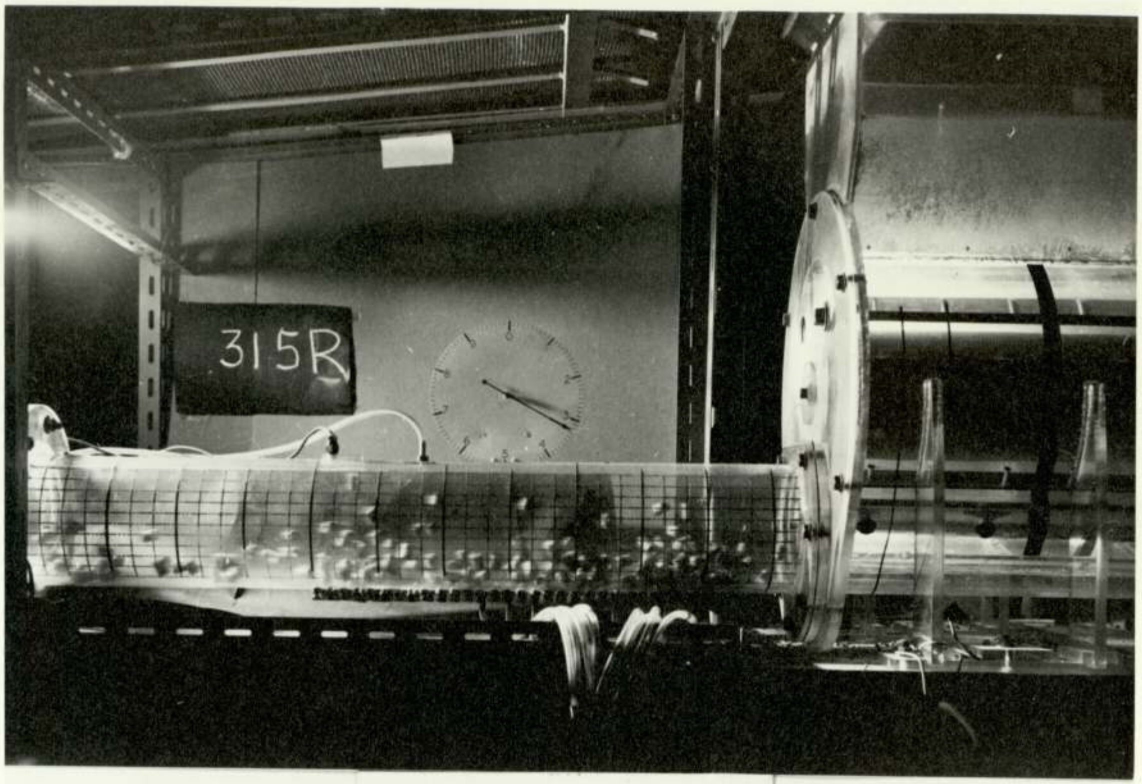


a) STOWER OUTLET

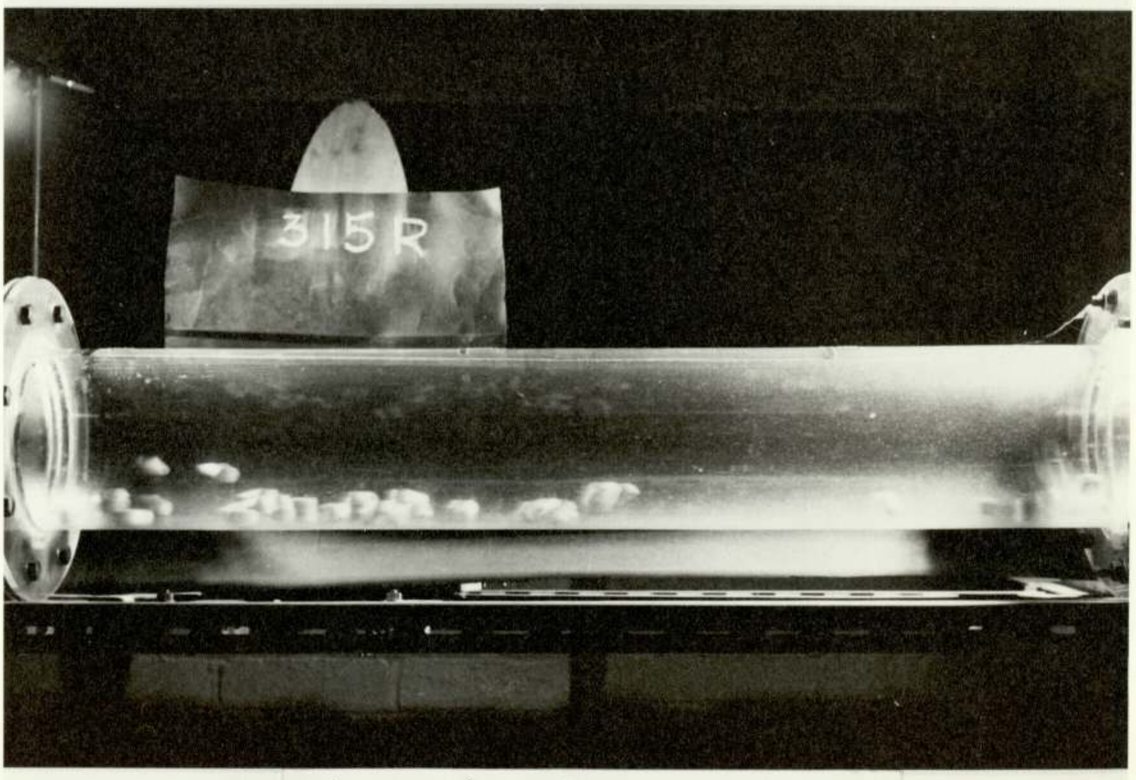


b) 25ft (7·6m) DOWNSTREAM

FIG. 60 1/2 in. (13 mm.) PENCILS
COVERED POCKET (8·6% FULL)
AIR FLOW 1100 ft.^{3}/min. (0·5 m.^{3}/s)

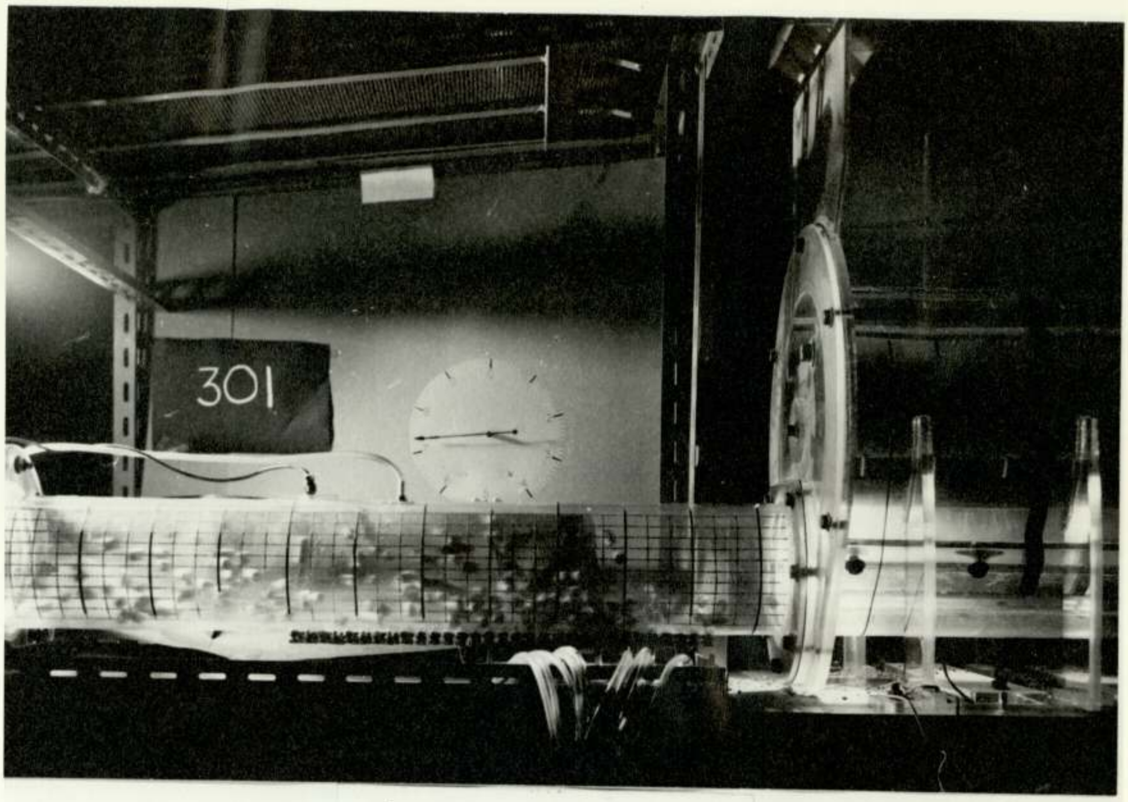


a) STOWER OUTLET

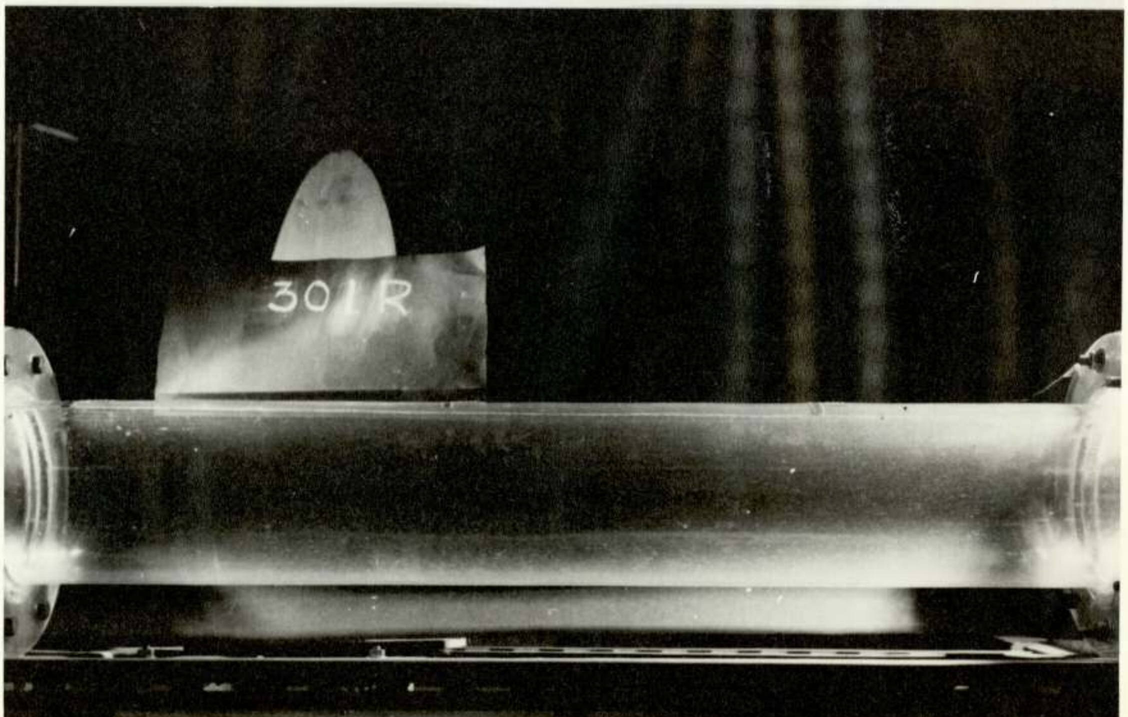


b) 25ft (7·6m) DOWNSTREAM

FIG. 61 1/2 IN (13mm) PENCILS
COVERED POCKET (8·6% FULL)
AIR FLOW 1350 ft.^{3}/min (0·6m.^{3}/S)

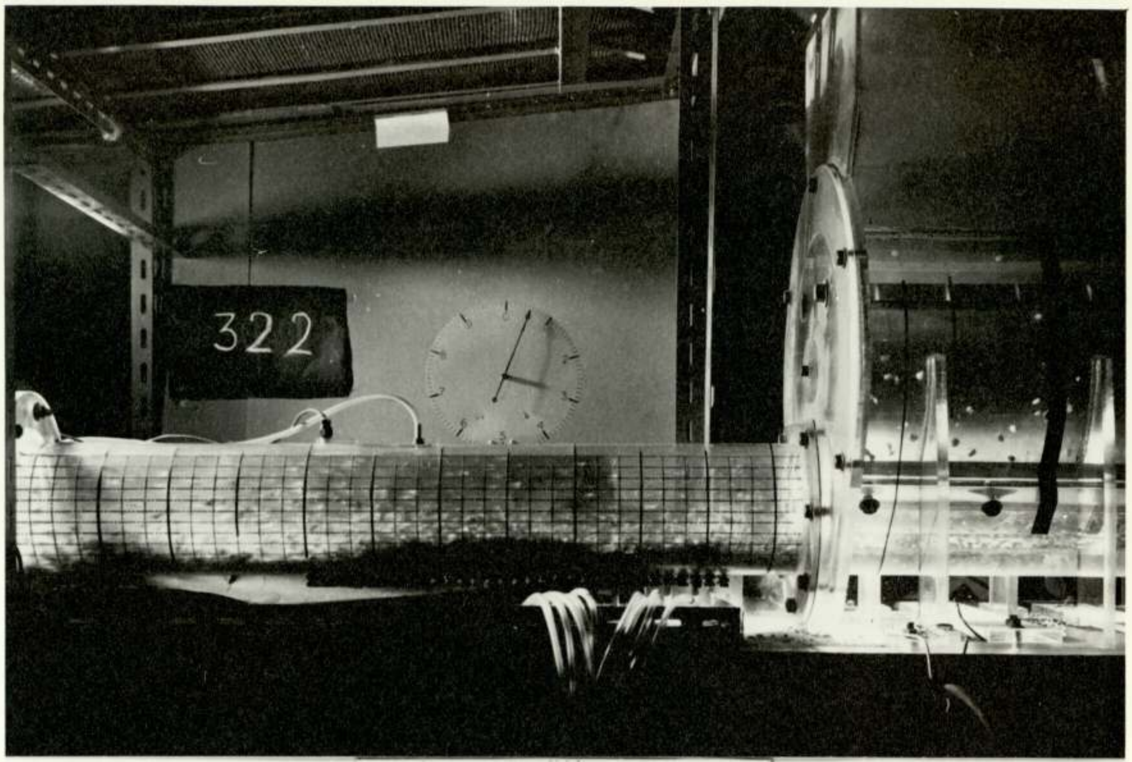


a) STOWER OUTLET

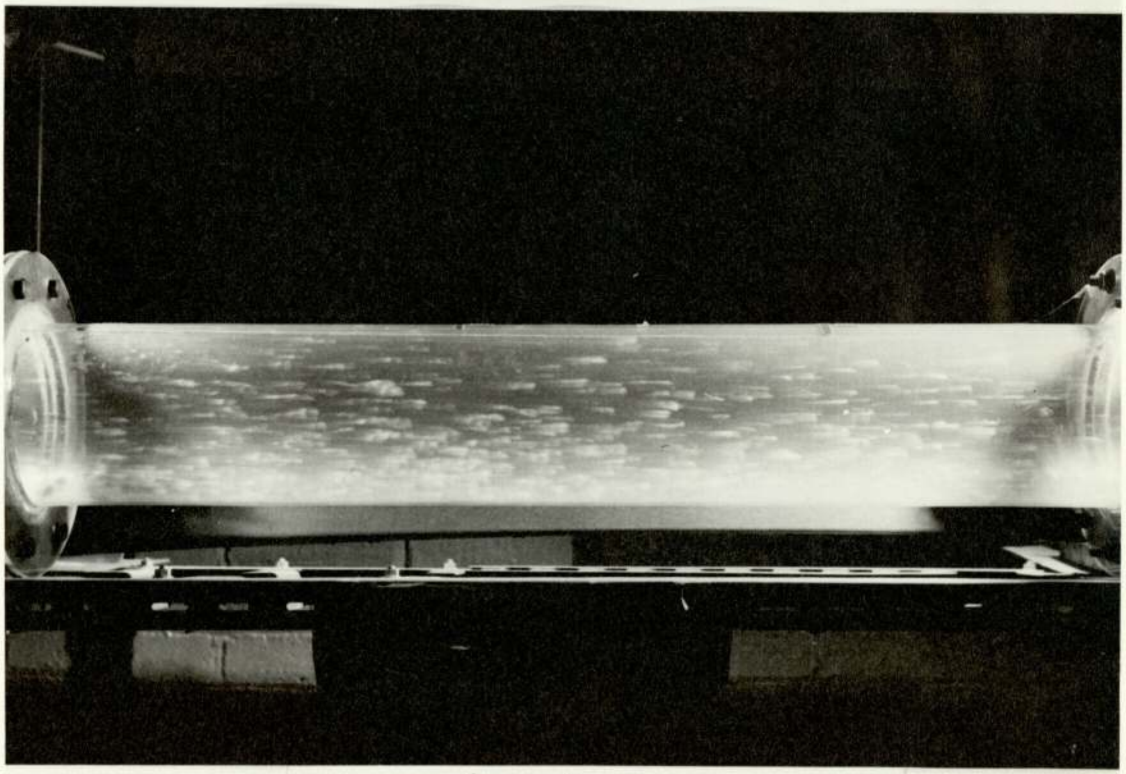


b) 25ft (7·6m) DOWNSTREAM

FIG. 62 1/2 in (13 mm.) PENCILS
COVERED POCKET (8·5% FULL)
AIR FLOW 1700 ft.^{3}/min (0·8 m^3/s)

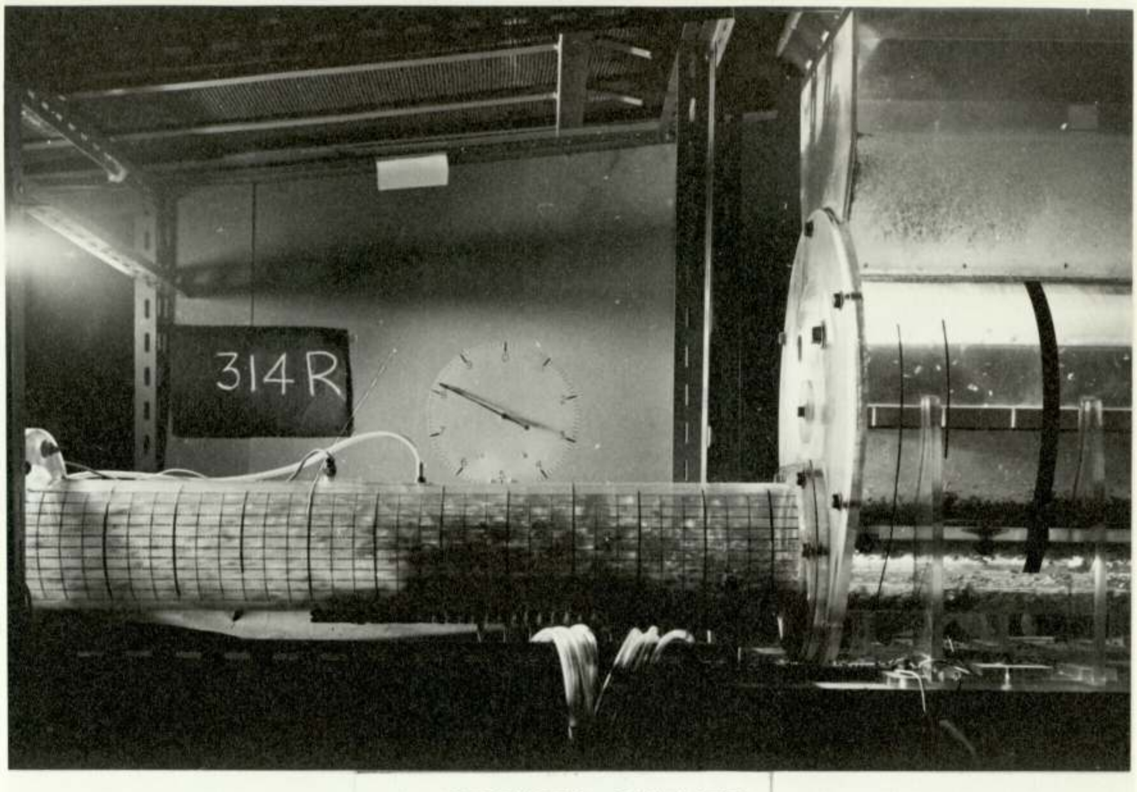


a) STOWER OUTLET

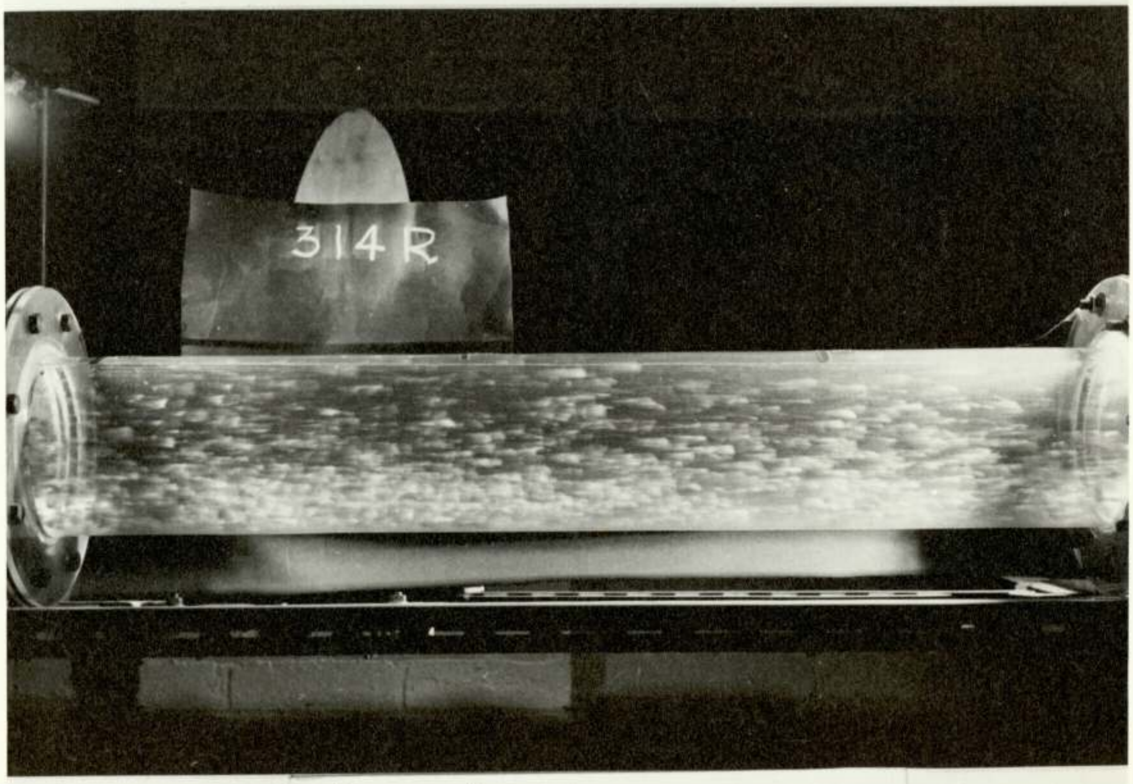


b) 25ft (7·6m) DOWNSTREAM

FIG. 63 MAIZE – FULL POCKET
AIR FLOW 1100 ft.^{3}/min. (0·5 m.^{3}/s)

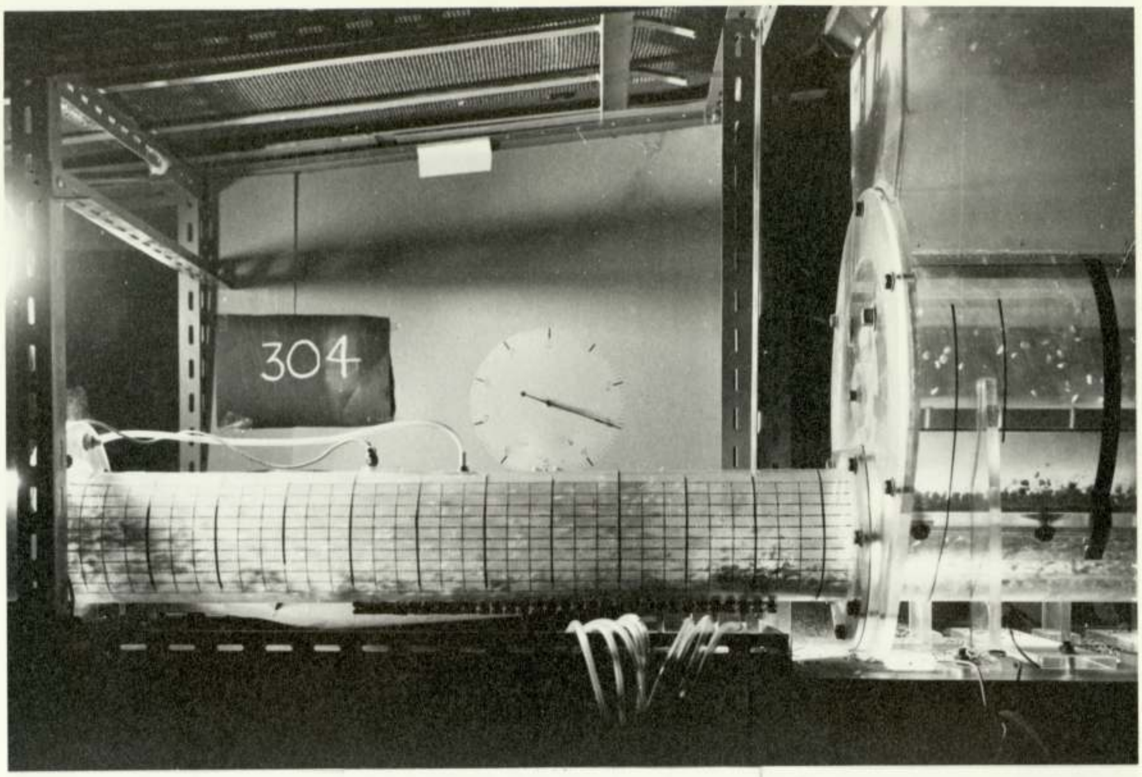


a) STOWER OUTLET

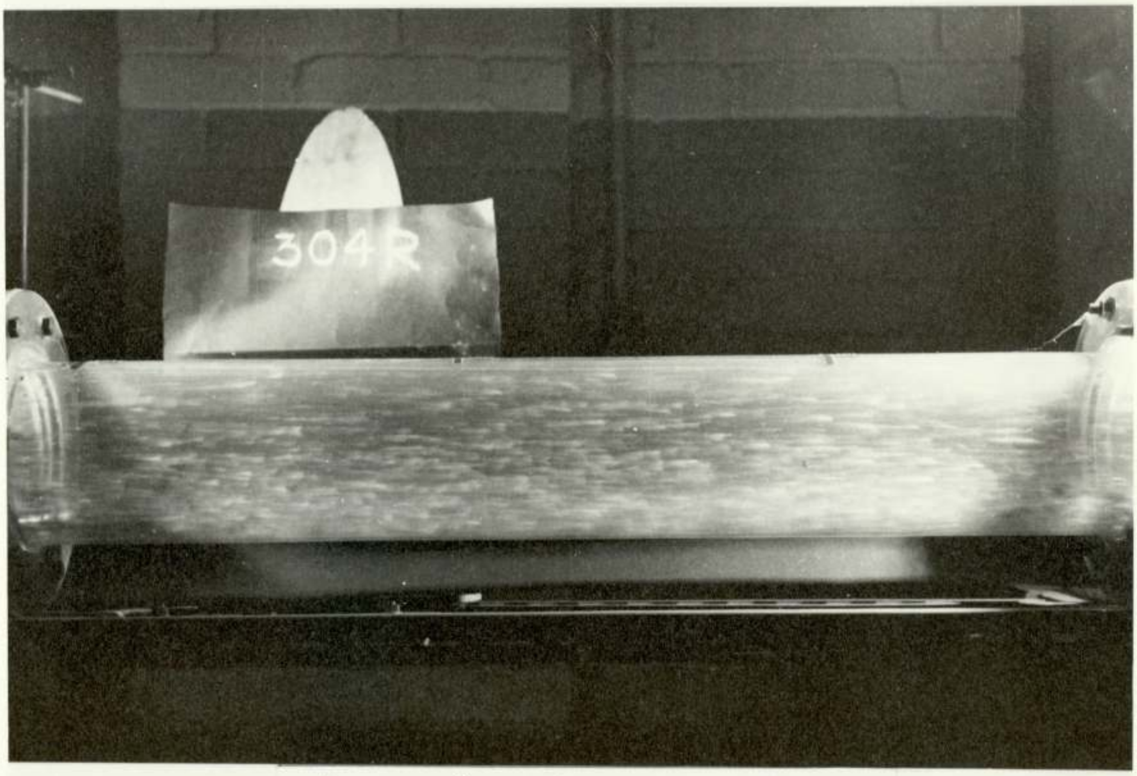


b) 25ft (7·6m) DOWNSTREAM

FIG. 64 MAIZE – FULL POCKET
AIR FLOW 1350 ft.^{3}/min. (0·6 m.^{3}/s)

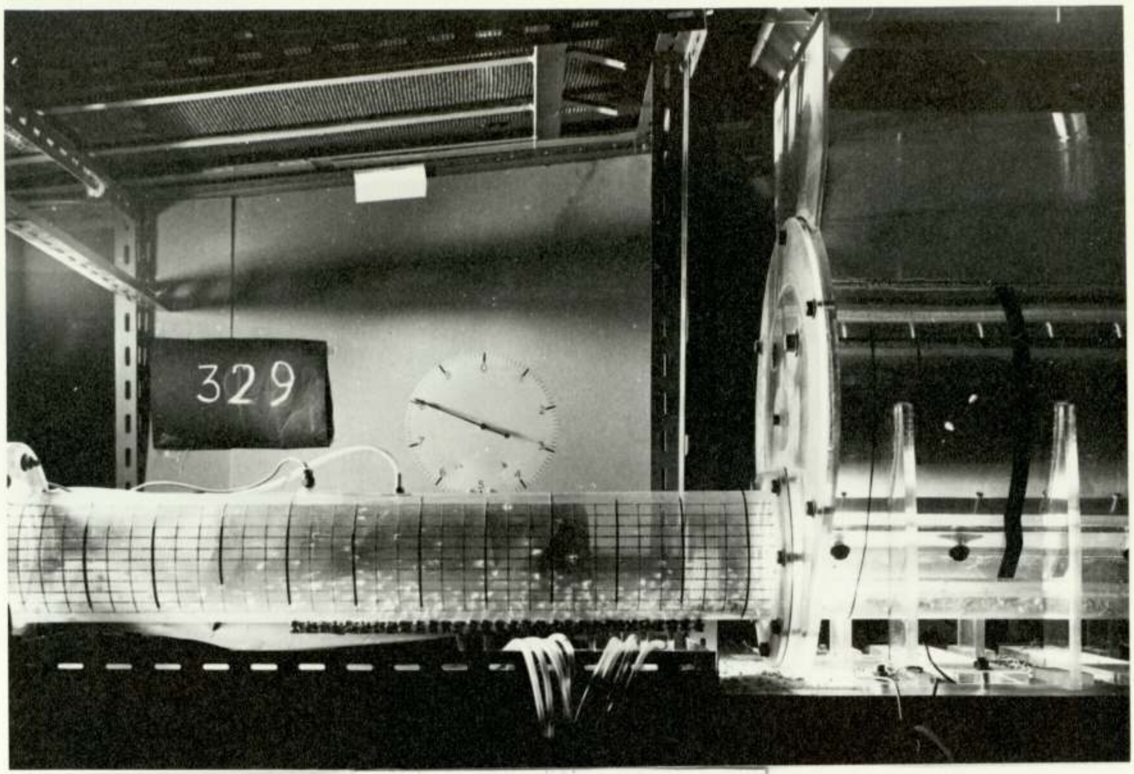


a) STOWER OUTLET

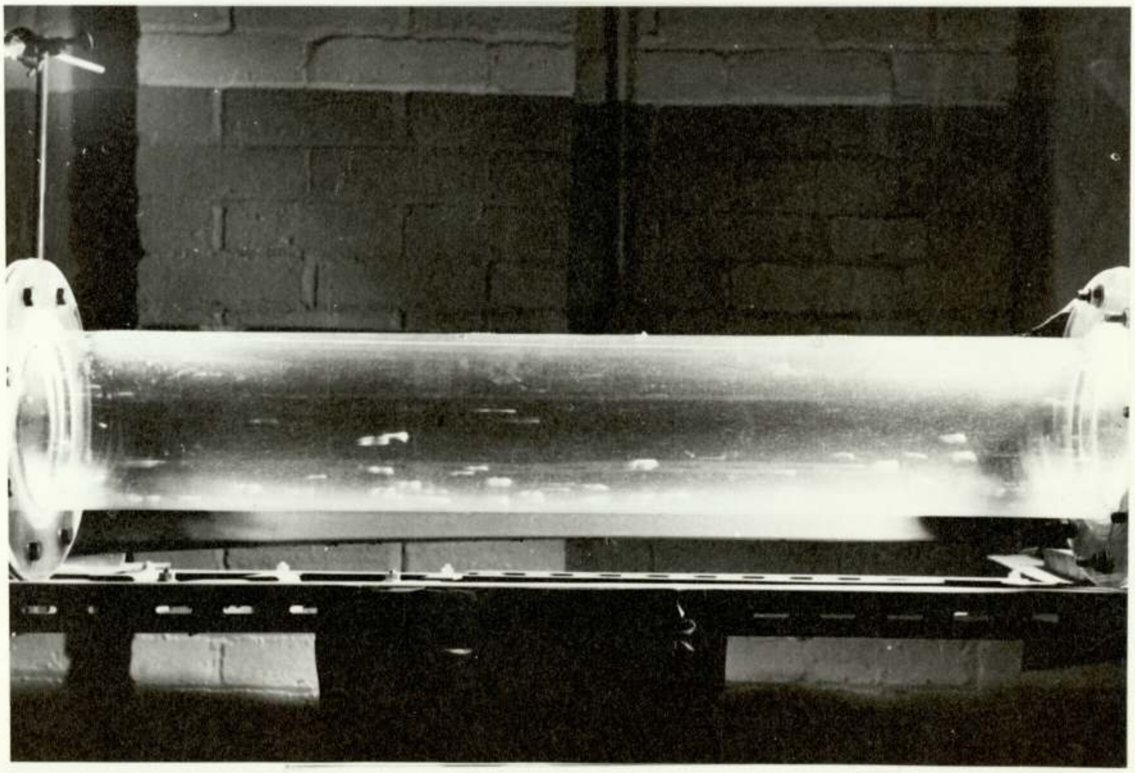


b) 25ft (7·6m) DOWNSTREAM

FIG. 65 MAIZE – FULL POCKET
AIR FLOW 1700 ft.^{3}/min. (0·8 m.^{3}/s)

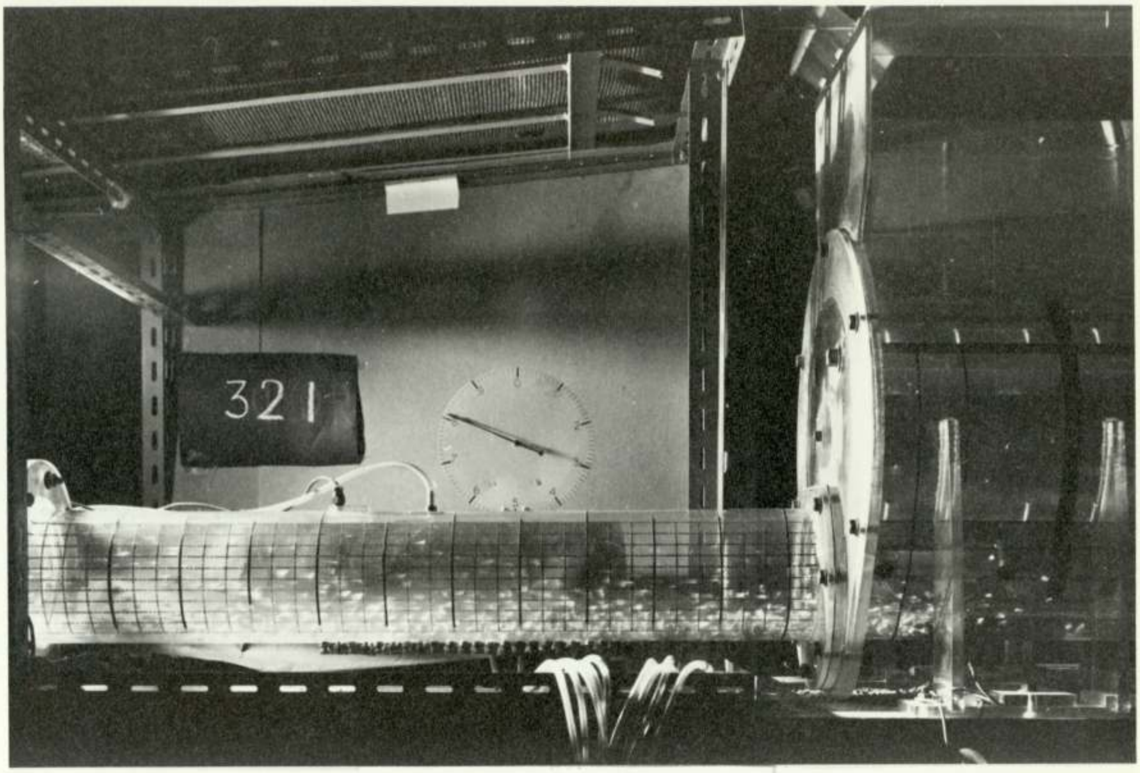


a) STOWER OUTLET

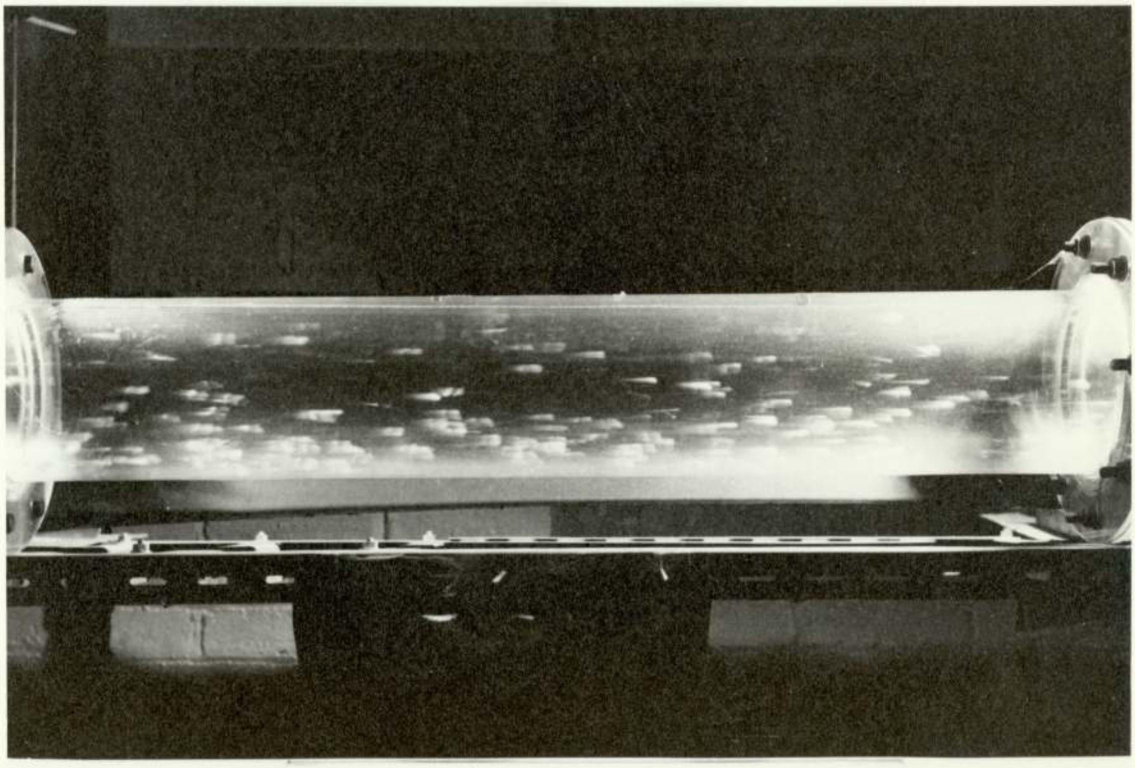


b) 25ft (7·6m) DOWNSTREAM

FIG. 66 MAIZE – COVERED POCKET (3·5% FULL)
AIR FLOW 800 ft.^{3}/min. (0·4 m.^{3}/s.)

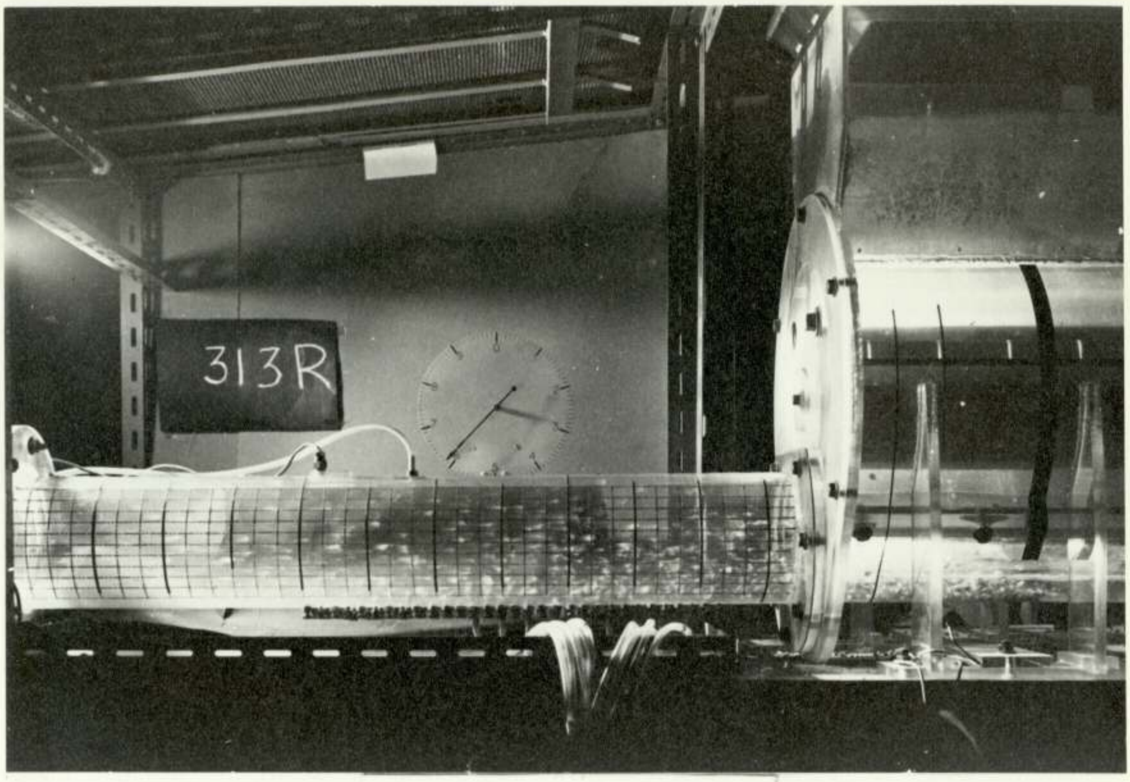


a) STOWER OUTLET

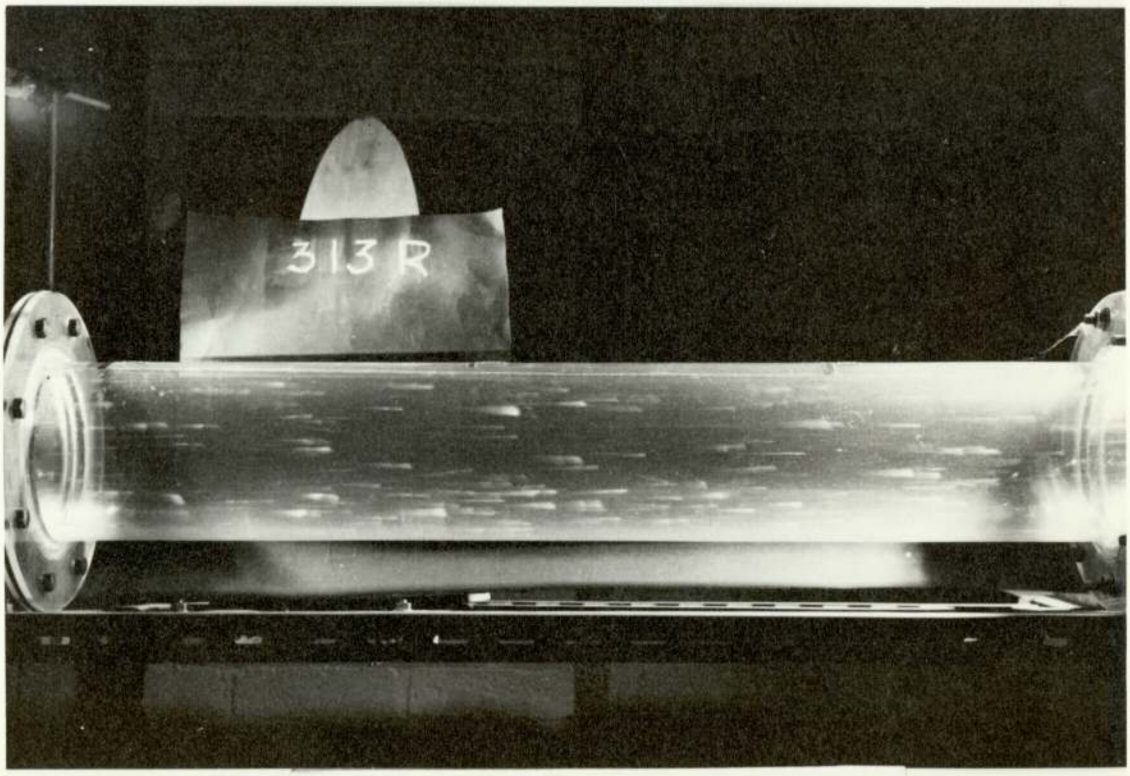


b) 25ft (7·6m) DOWNSTREAM

FIG. 67 MAIZE - COVERED POCKET (3·5% FULL)
AIR FLOW 1100 ft.^{3}/min. (0·5 m.^{3}/s)

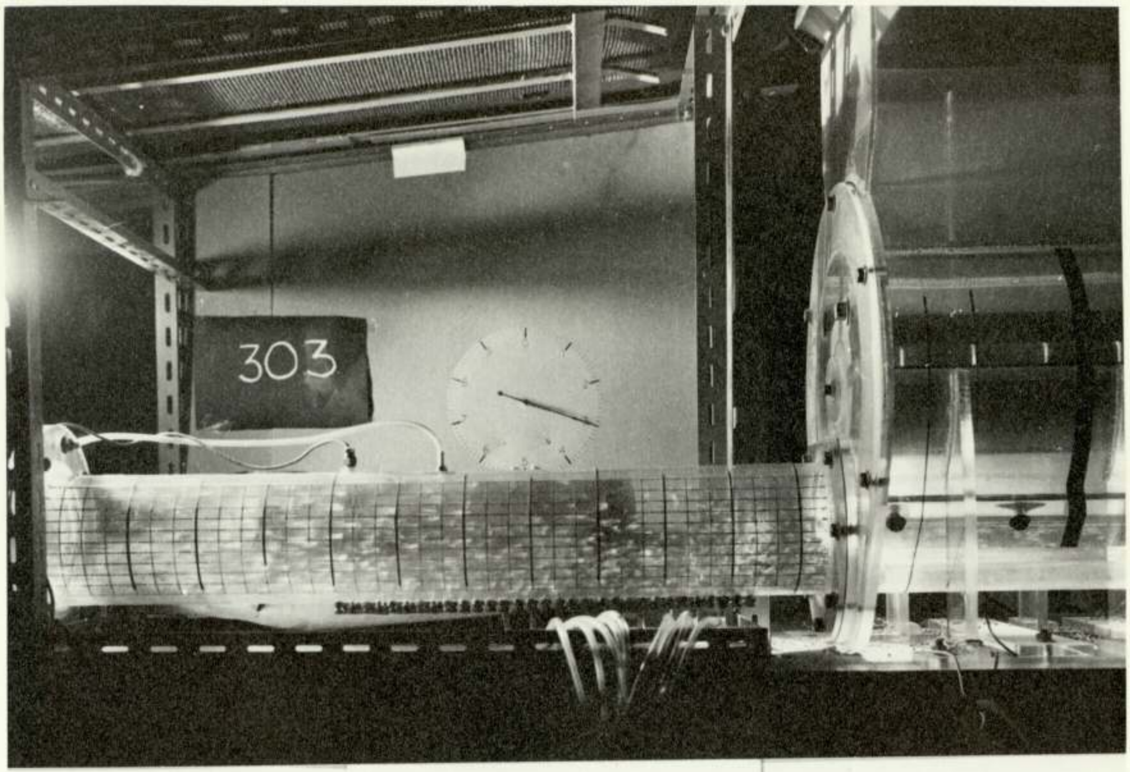


a) STOWER OUTLET

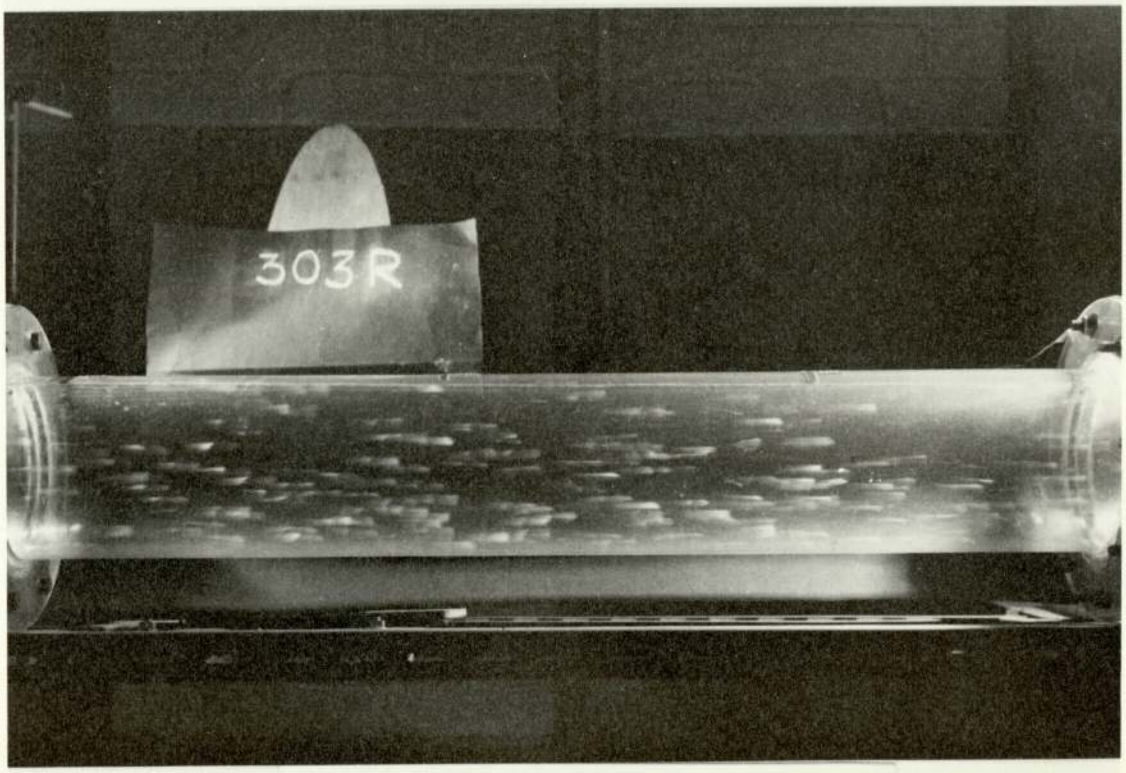


b) 25ft (7·6m) DOWNSTREAM

FIG. 68 MAIZE – COVERED POCKET (3·5% FULL)
AIR FLOW 1350 ft.^{3}/min. (0·6 m.^{3}/s)

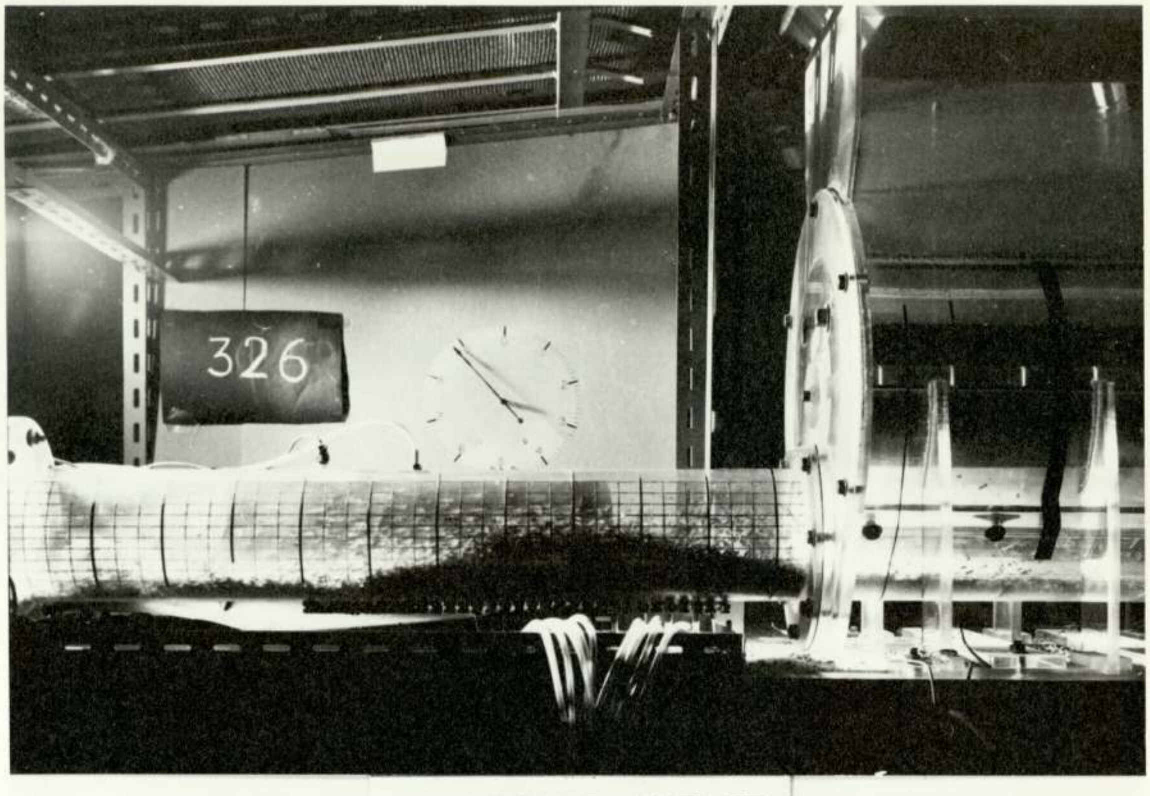


a) STOWER OUTLET

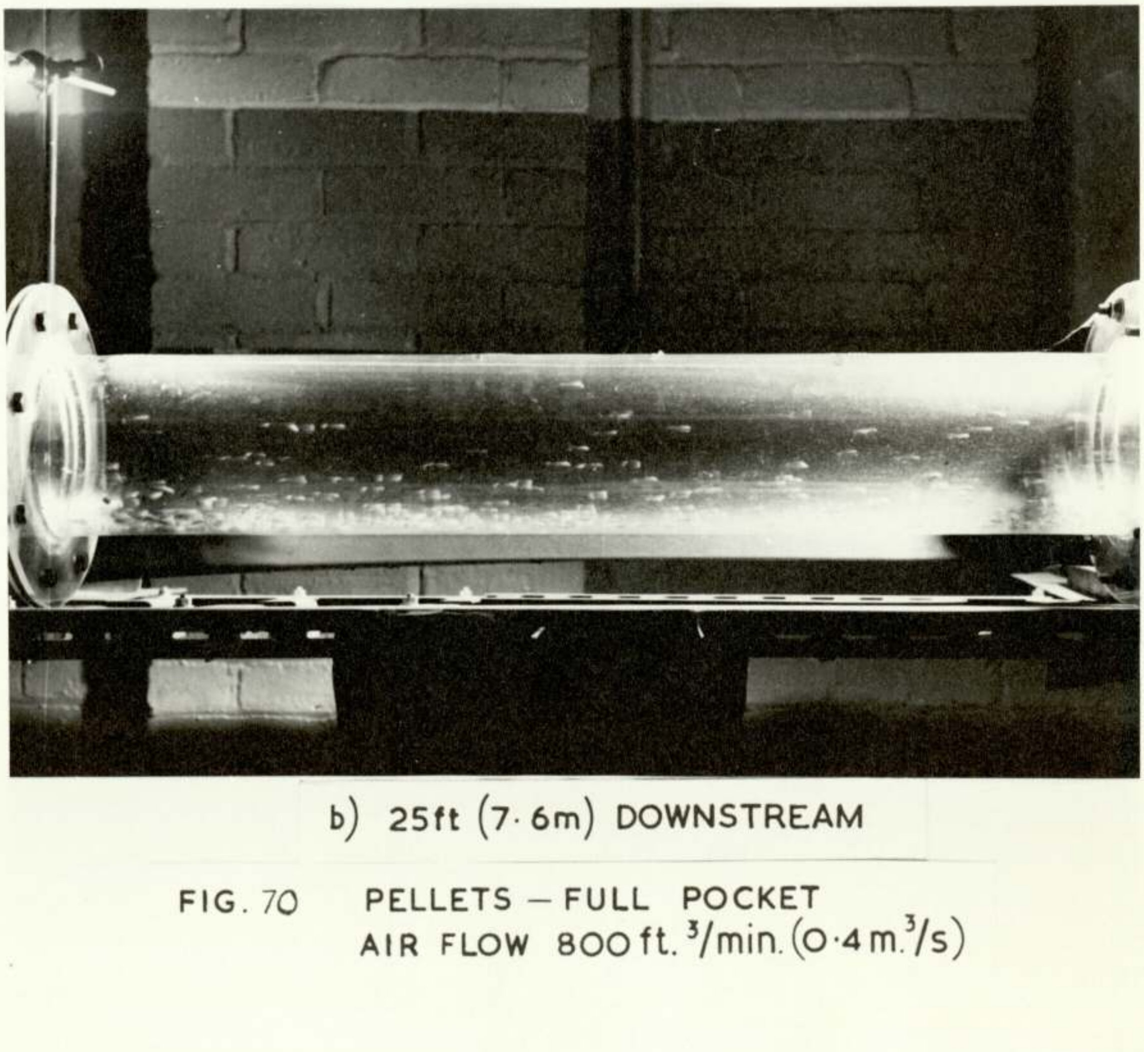


b) 25ft (7·6m) DOWNSTREAM

FIG 69 MAIZE – COVERED POCKET (3·5 % FULL)
AIR FLOW 1700 ft.^{3}/min. (0·8 m.^{3}/s)

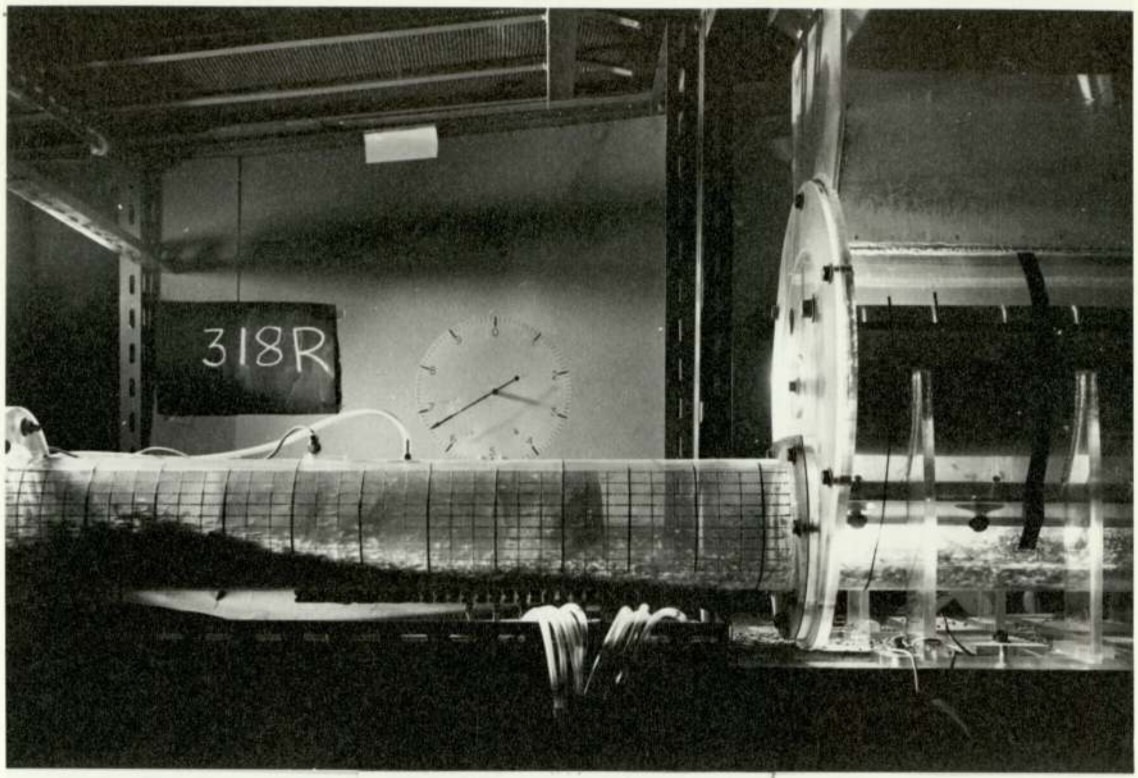


a) STOWER OUTLET

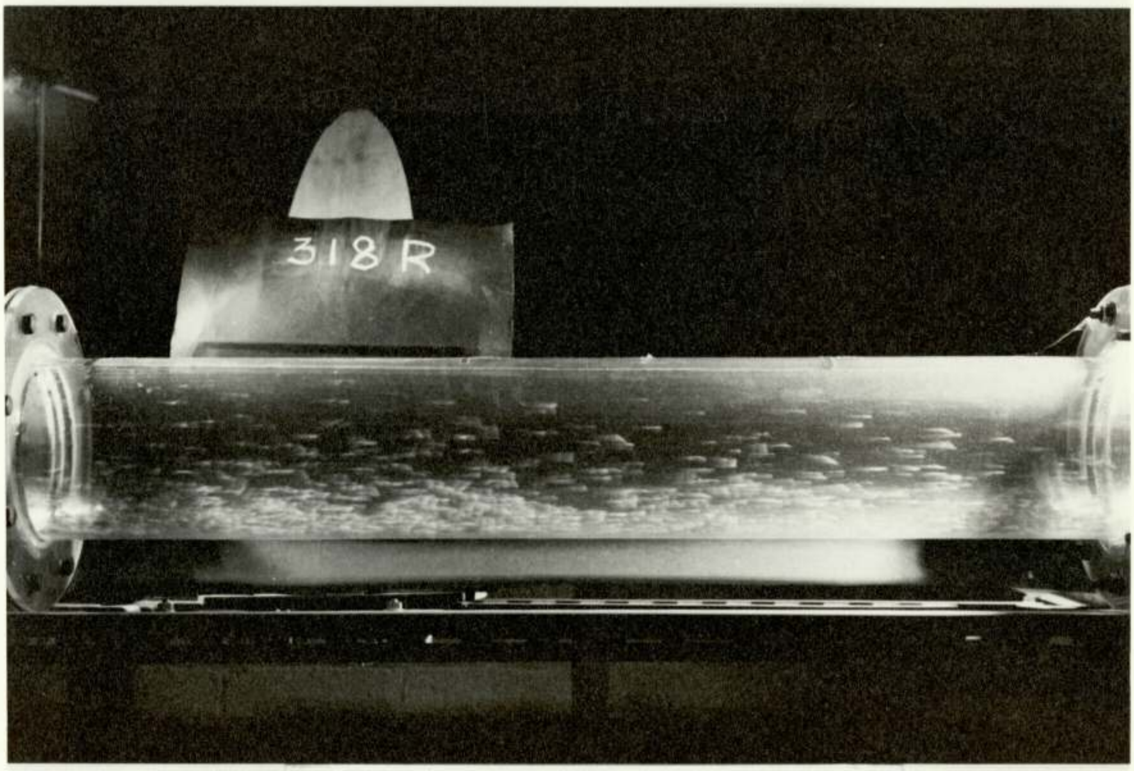


b) 25ft (7·6m) DOWNSTREAM

FIG. 70 PELLETS – FULL POCKET
AIR FLOW 800 ft.^{3}/min. (0·4 m.^{3}/s)

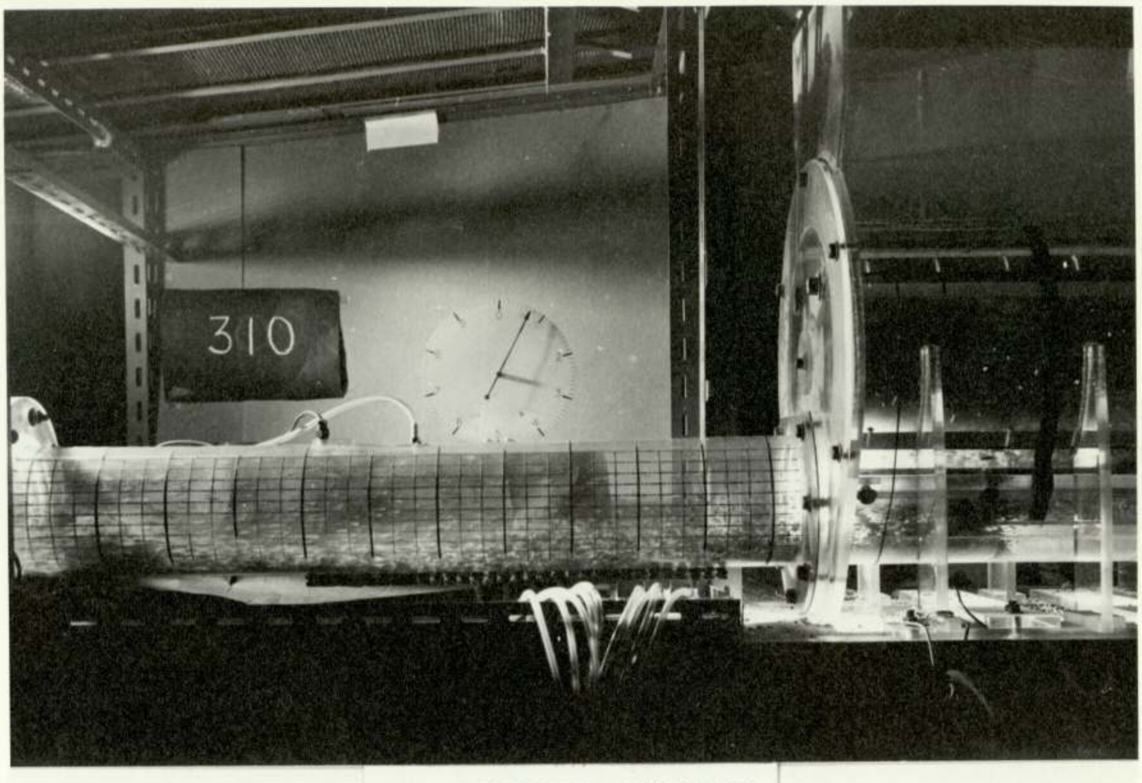


a) STOWER OUTLET

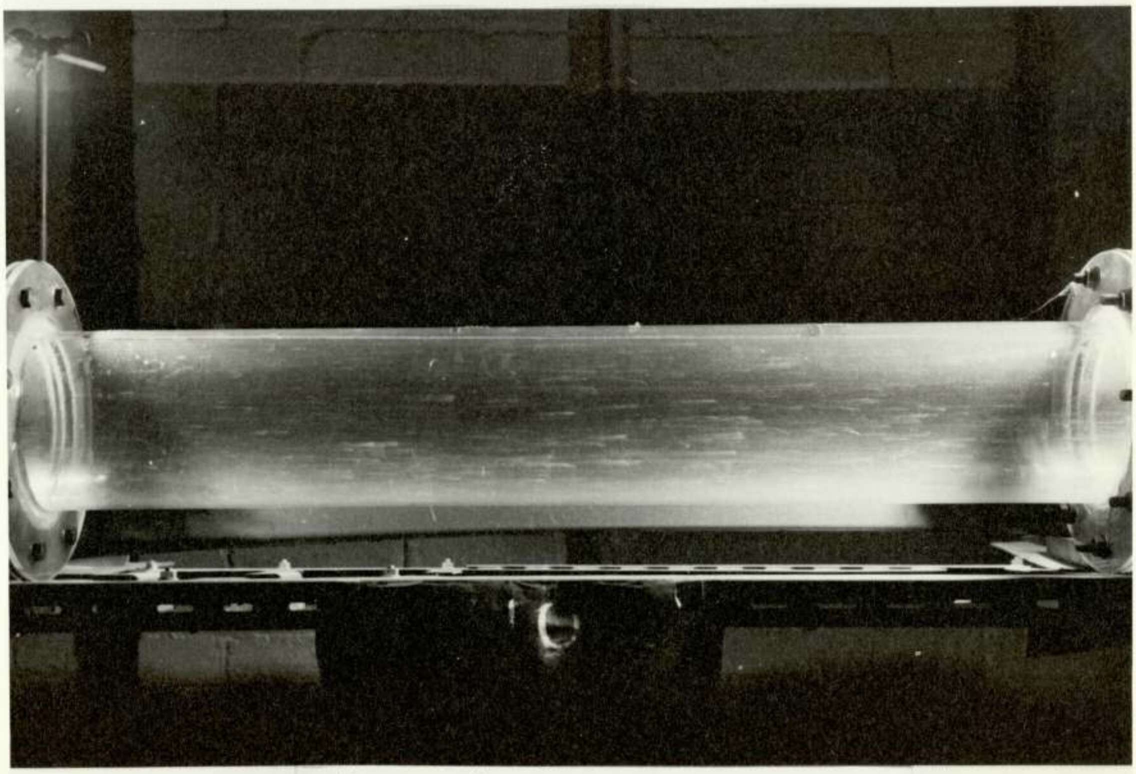


b) 25ft (7·6m) DOWNSTREAM

FIG. 71 PELLETS – FULL POCKET
AIR FLOW 1100 ft.^{3}/min. (0·5 m.^{3}/s)

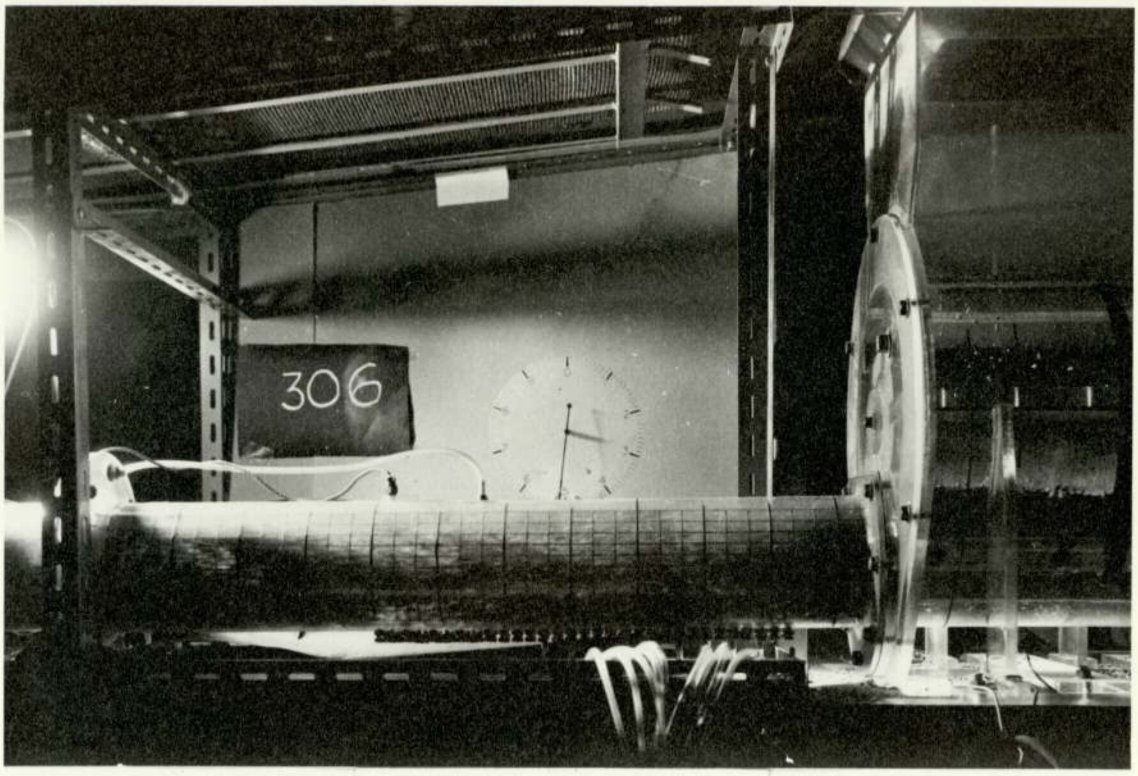


a) STOWER OUTLET

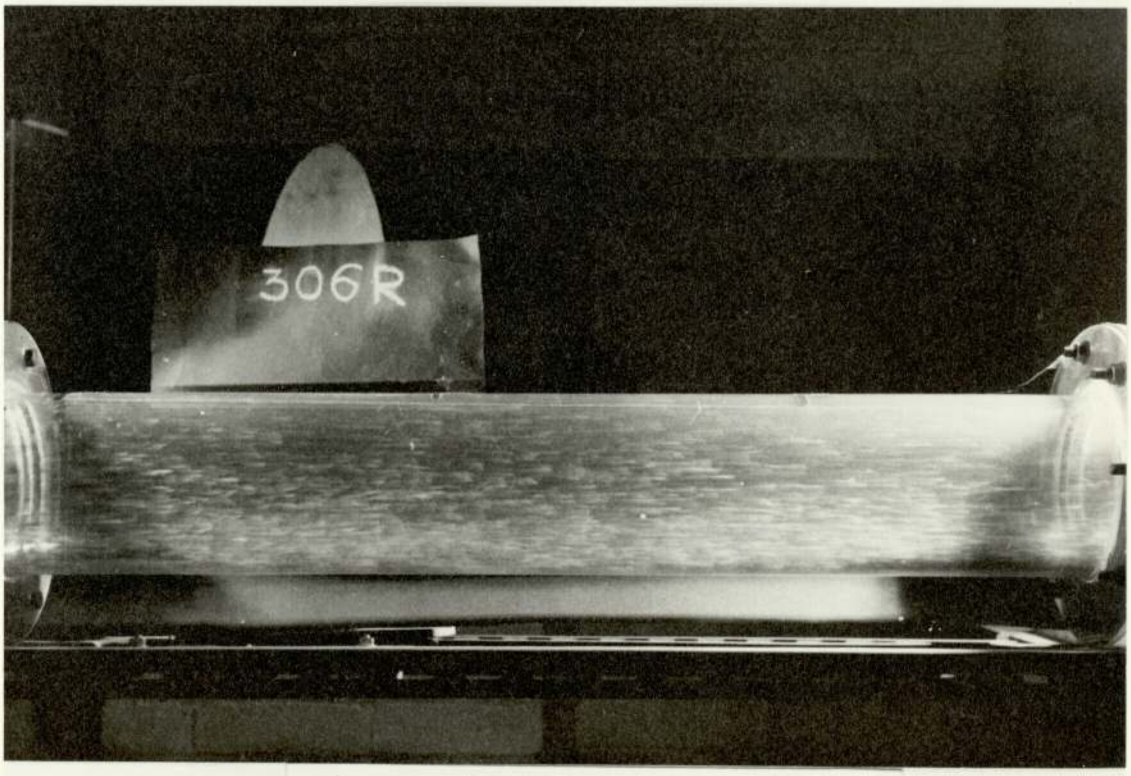


b) 25ft (7·6m) DOWNSTREAM

FIG. 72 PELLETS – FULL POCKET
AIR FLOW 1350 ft.^{3}/min. (0·6 m.^{3}/s)

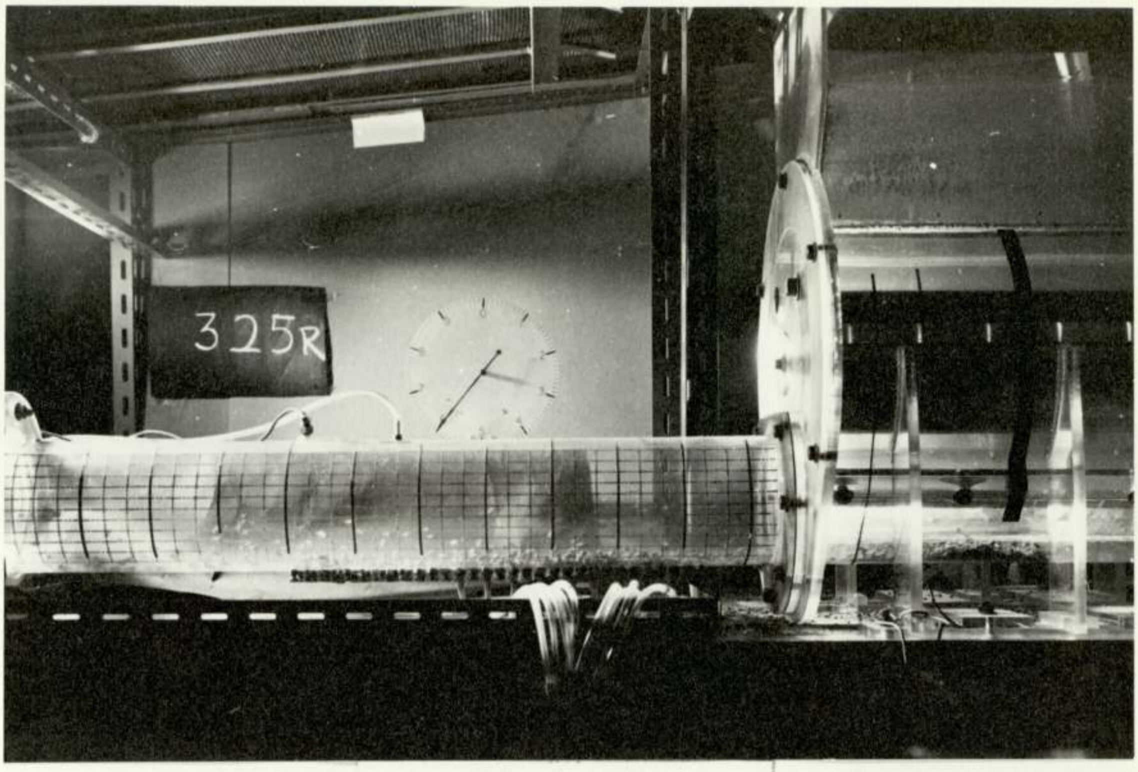


a) STOWER OUTLET

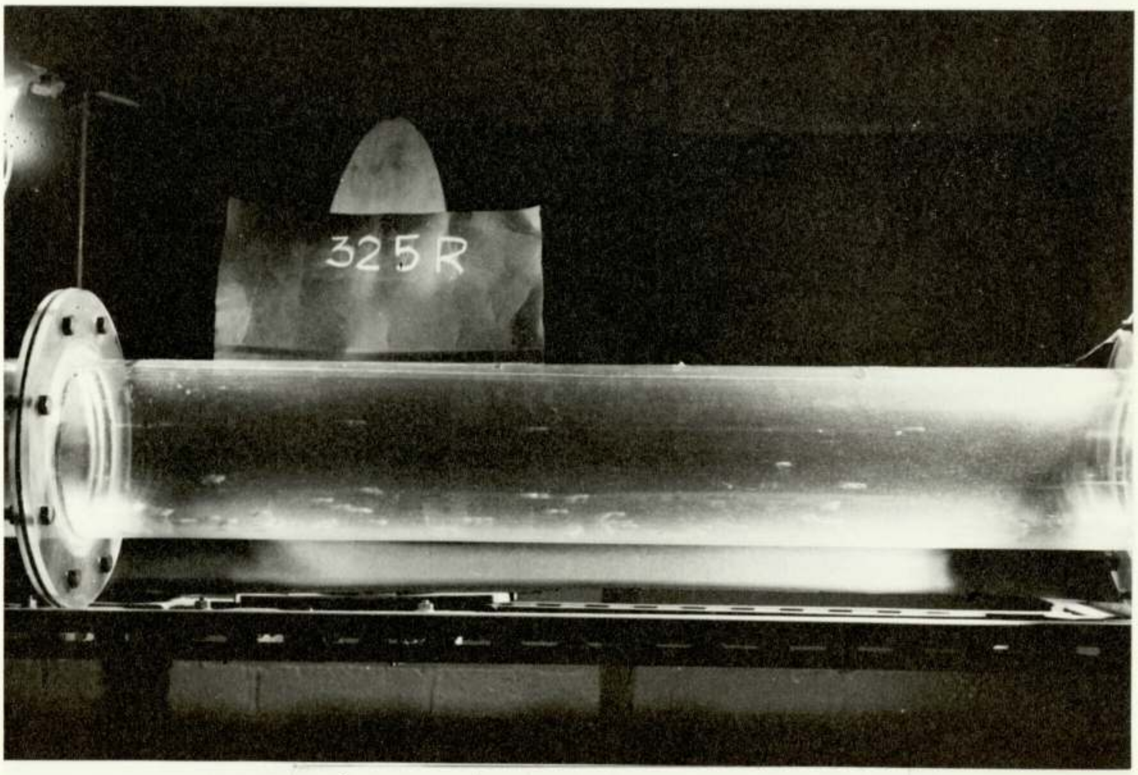


b) 25ft (7·6m) DOWNSTREAM

FIG. 73 PELLETS - FULL POCKET
AIR FLOW 1700 ft.^{3}/min. (0·8m^3/s)

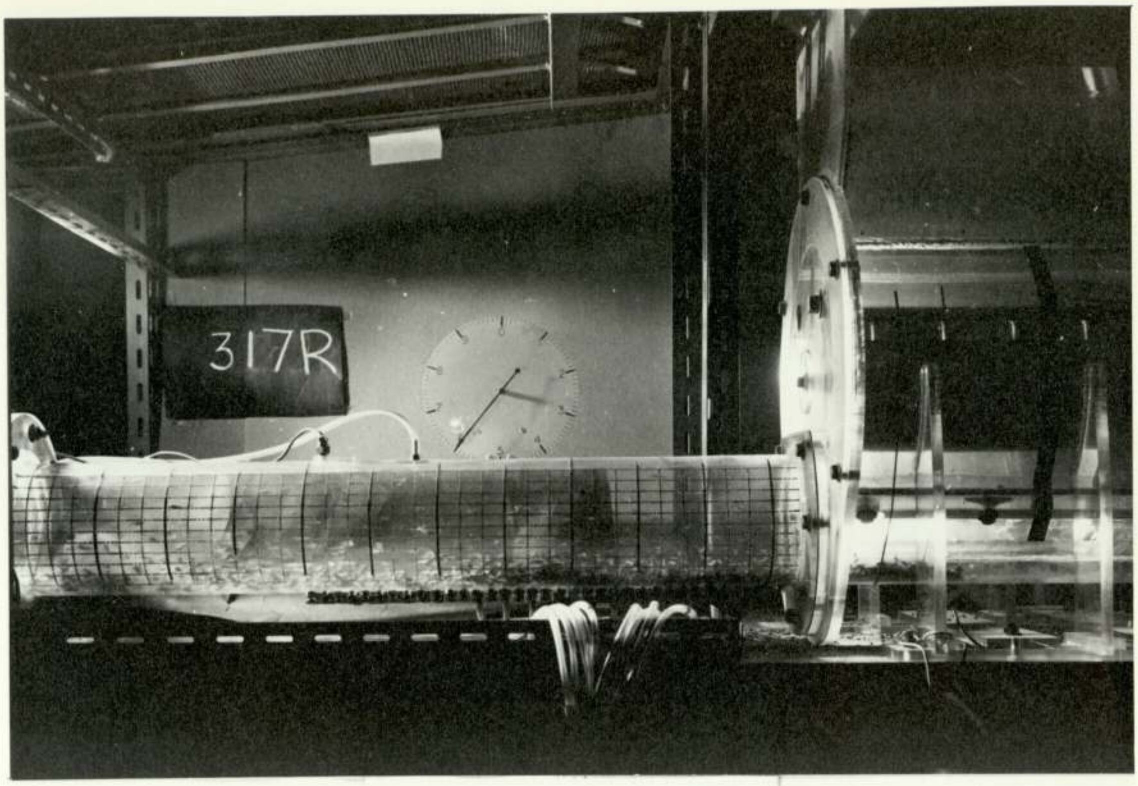


a) STOWER OUTLET

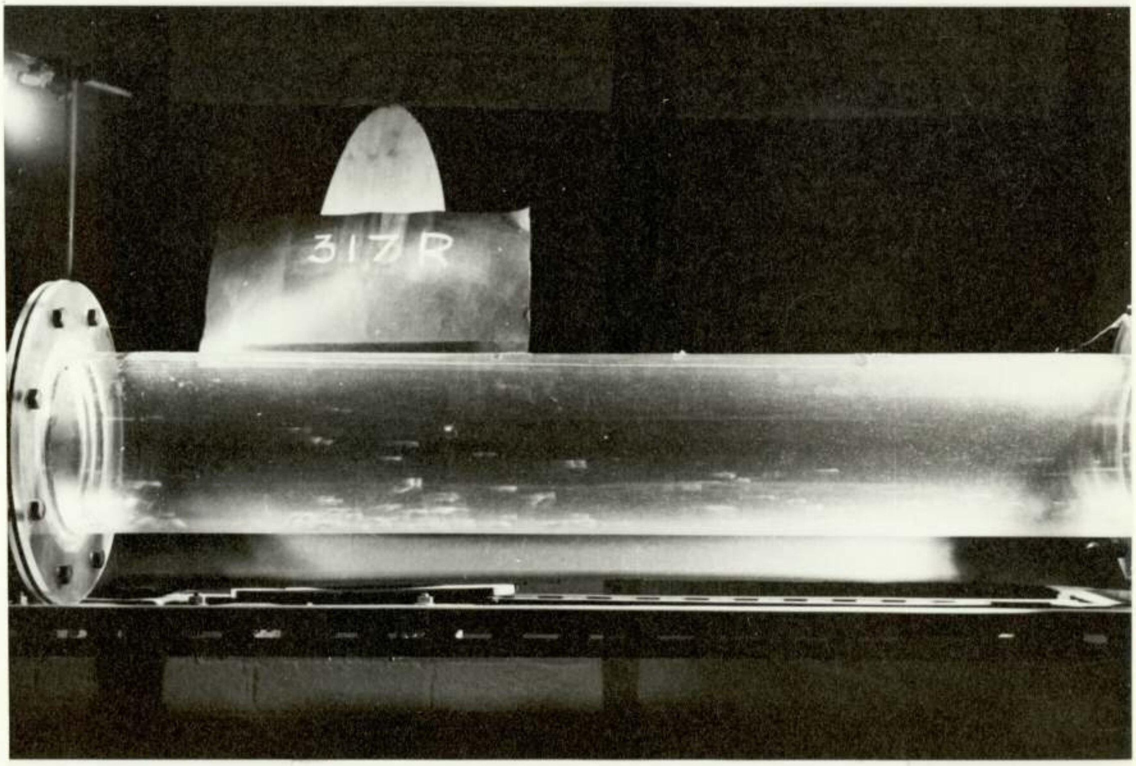


b) 25ft (7·6m) DOWNSTREAM

FIG 74 PELLETS – COVERED POCKET (1·5% FULL)
AIR FLOW 800 ft.^{3}/min. (0·4 m.^{3}/s)

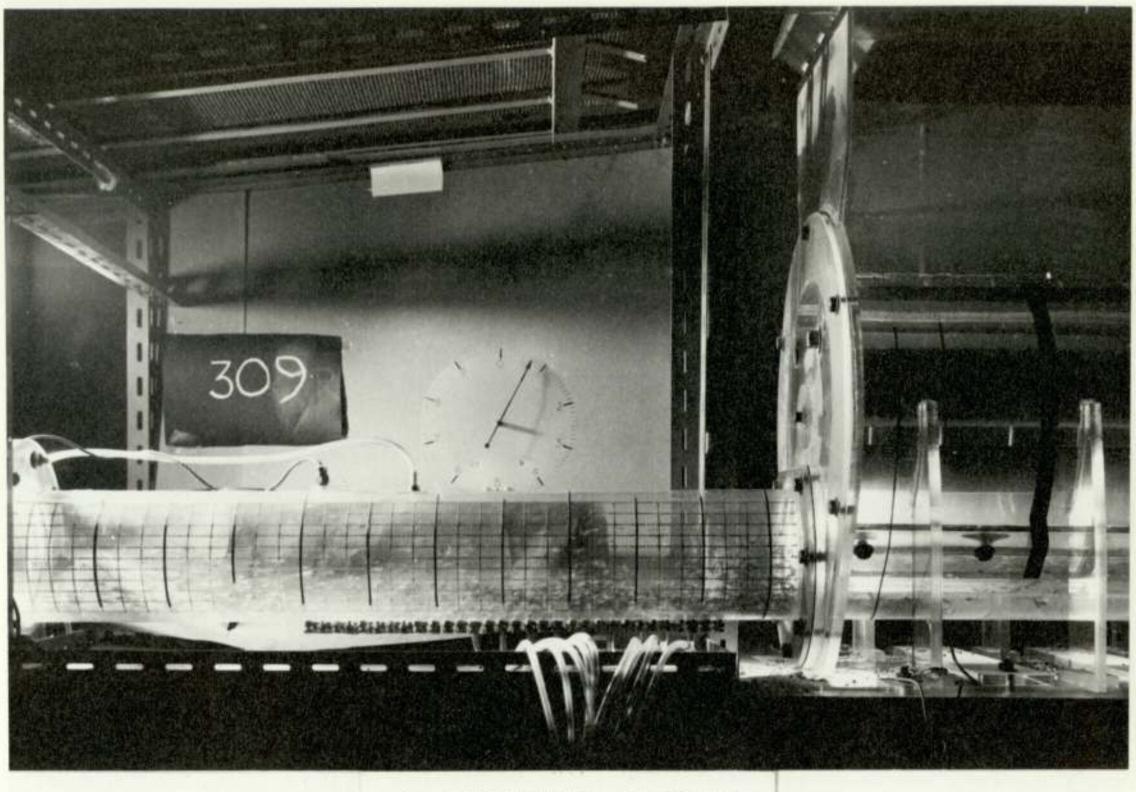


a) STOWER OUTLET

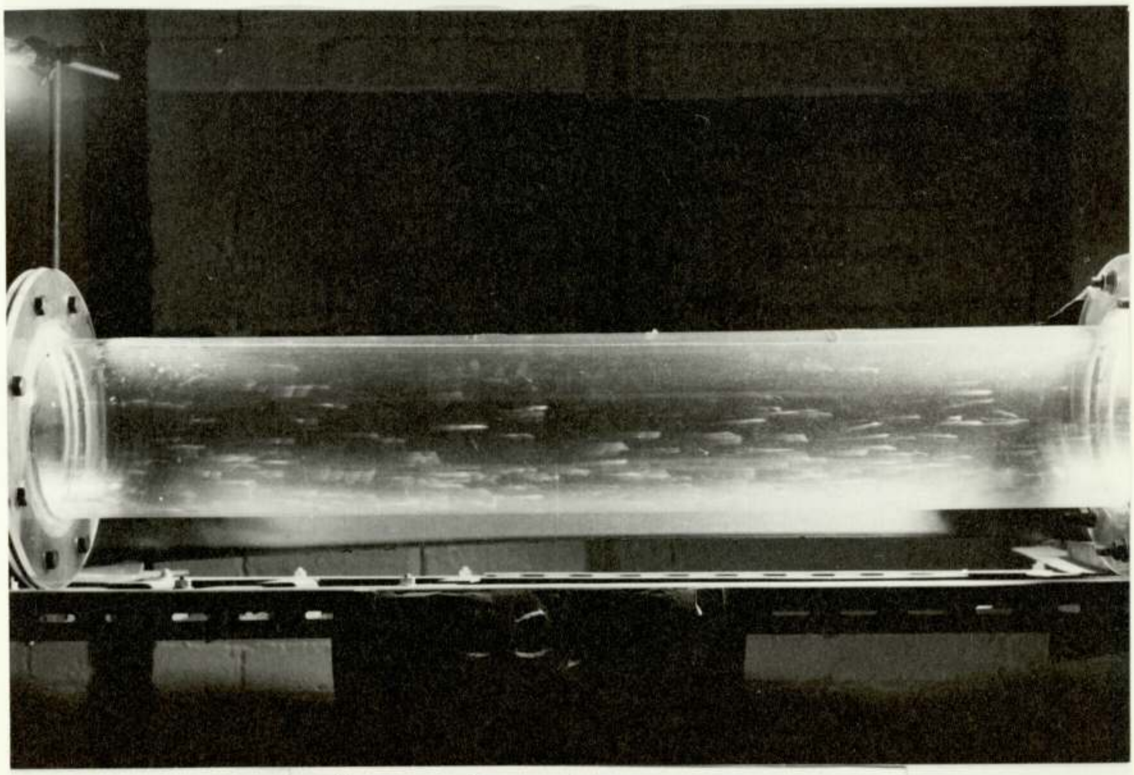


b) 25ft (7·6m) DOWNSTREAM

FIG.75 PELLETS – COVERED POCKET (1·5% FULL)
AIR FLOW 1100 ft.^{3}/min.(0·5m.^{3}/s)

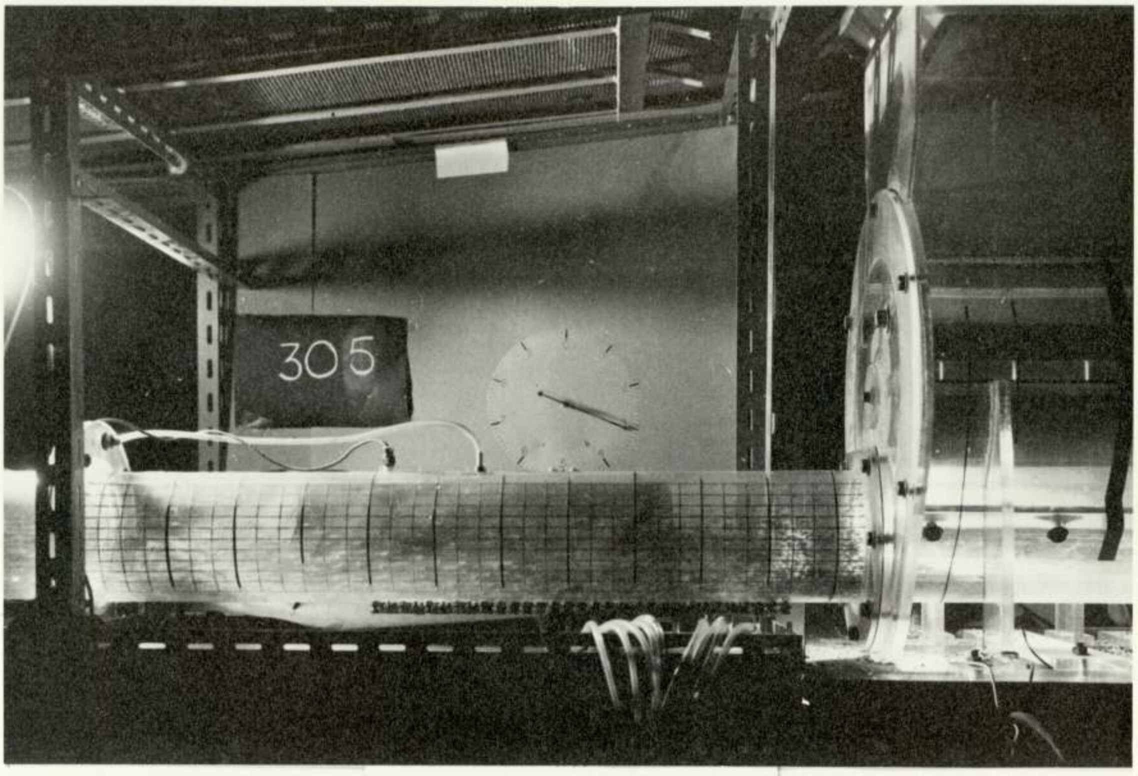


a) STOWER OUTLET

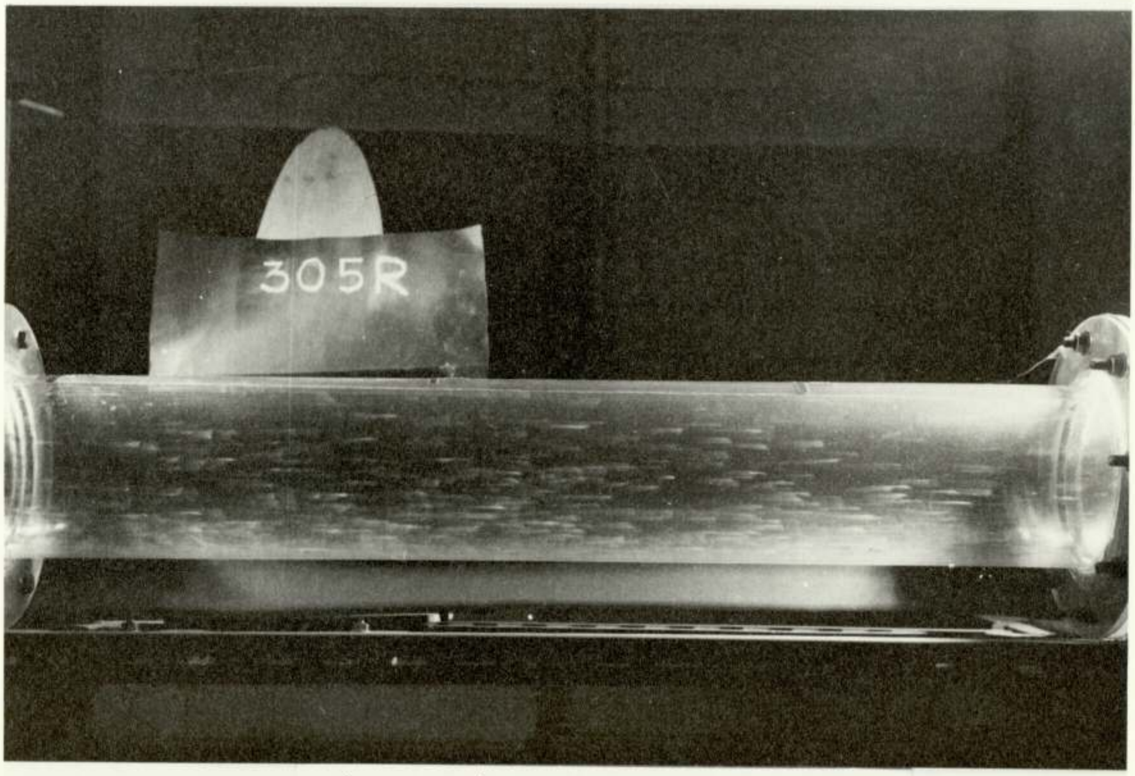


b) 25ft (7·6m) DOWNSTREAM

FIG. 76 PELLETS – COVERED POCKET (1·5% FULL)
AIR FLOW 1350 ft.^{3}/min. (0·6 m.^{3}/s)

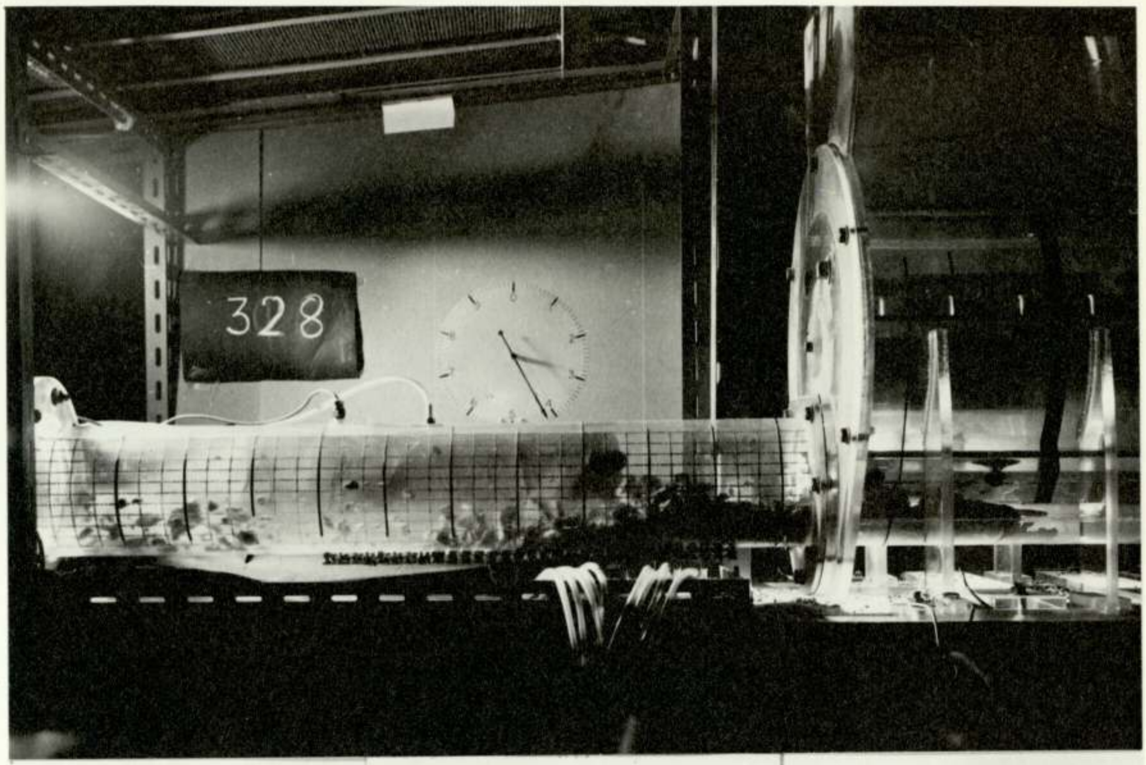


a) STOWER OUTLET

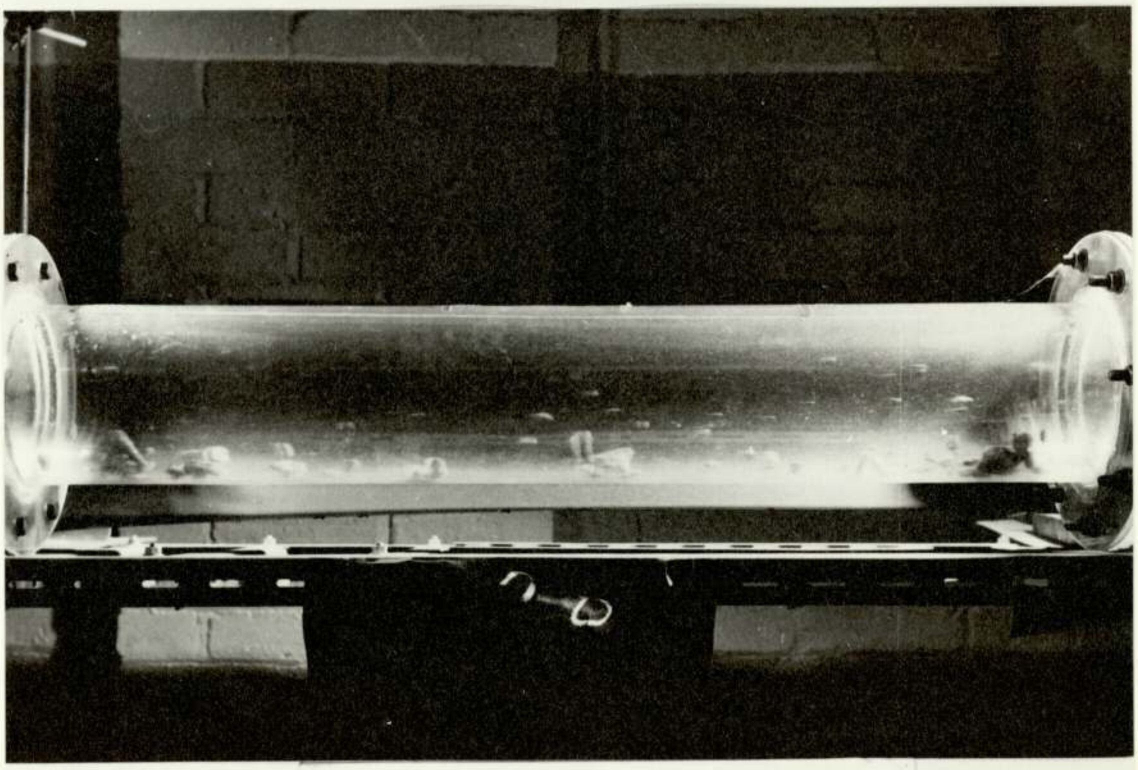


b) 25ft (7·6m) DOWNSTREAM

FIG 77 PELLETS – COVERED POCKET (1·5 % FULL)
AIR FLOW 1700 ft.^{3}/min. (0·8 m.^{3}/s)

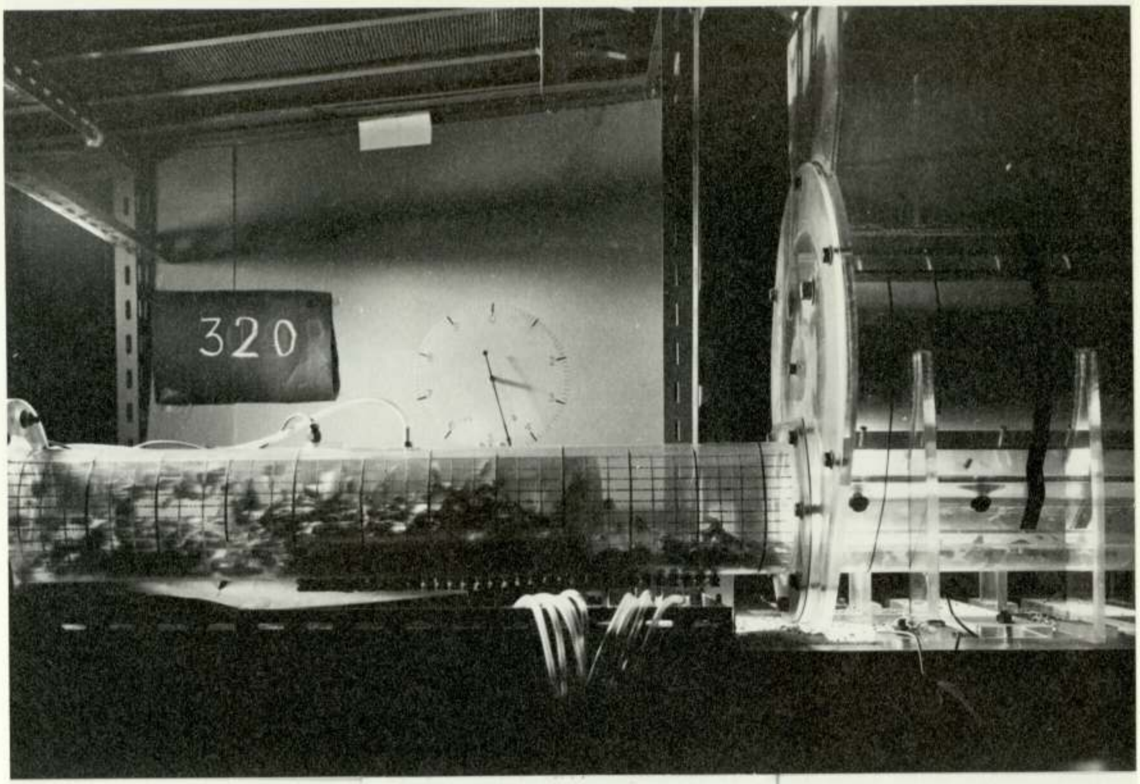


a) STOWER OUTLET

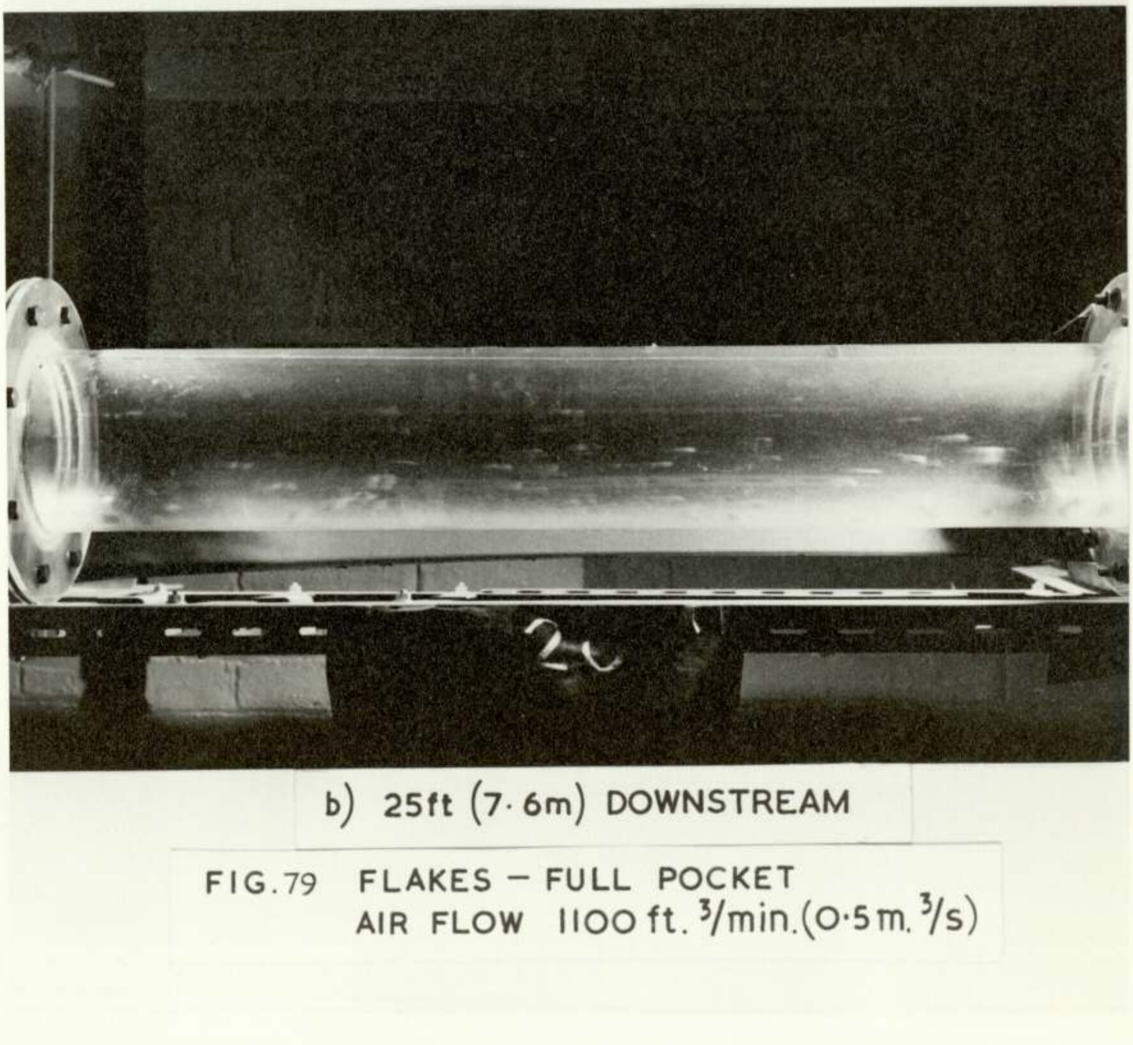


b) 25ft (7·6m) DOWNSTREAM

FIG. 78 FLAKES – FULL POCKET
AIR FLOW 800 ft.^{3}/min (0·4 m.^{3}/s.)

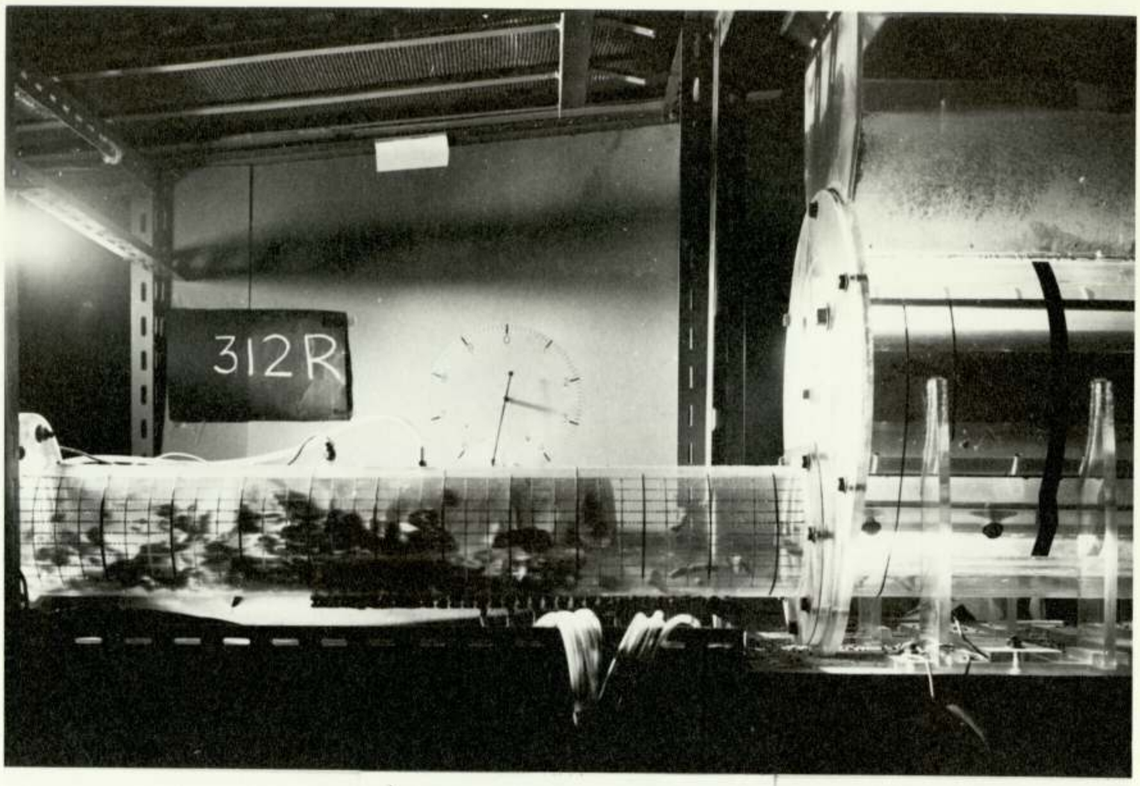


a) STOWER OUTLET

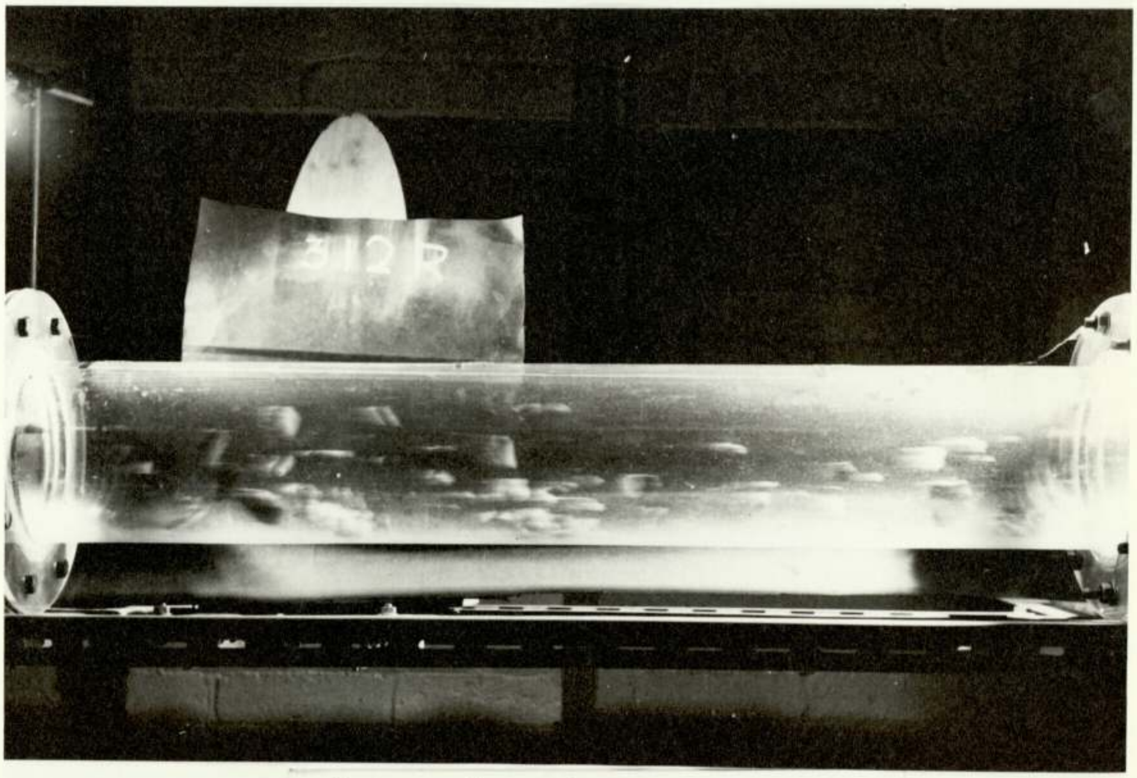


b) 25ft (7·6m) DOWNSTREAM

FIG.79 FLAKES — FULL POCKET
AIR FLOW 1100 ft.^{3}/min.(0·5 m.^{3}/s)

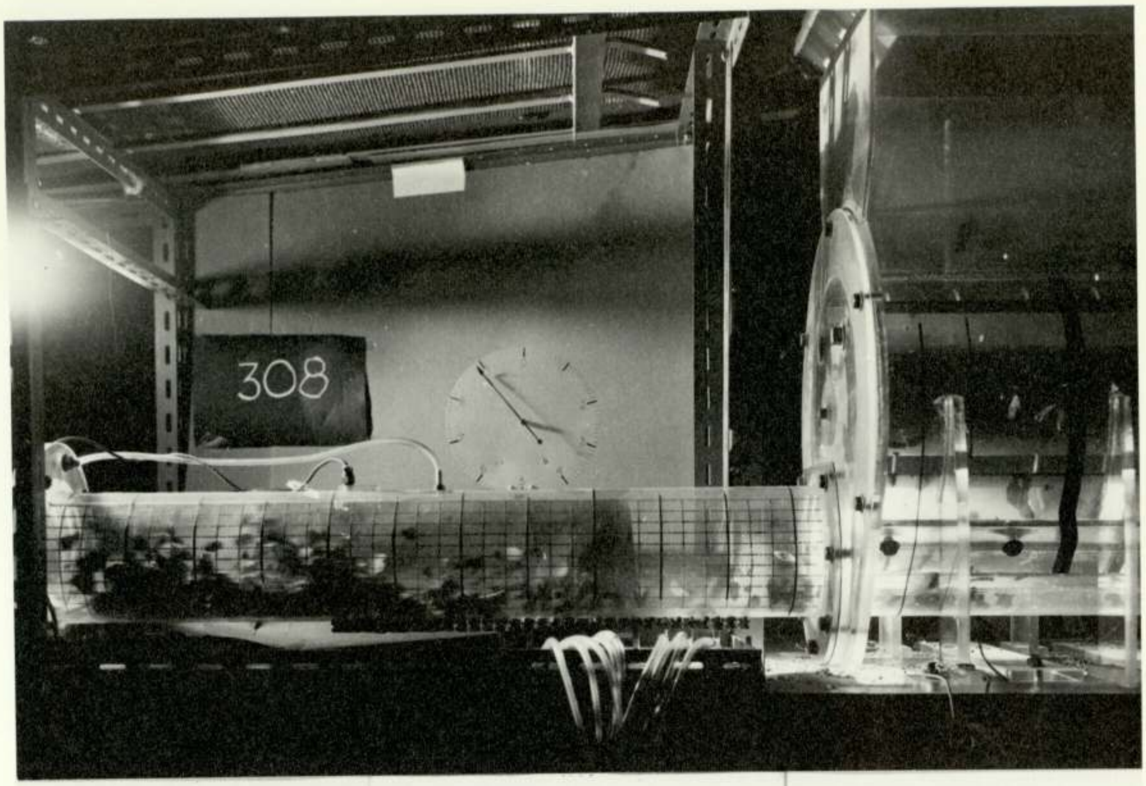


a) STOWER OUTLET

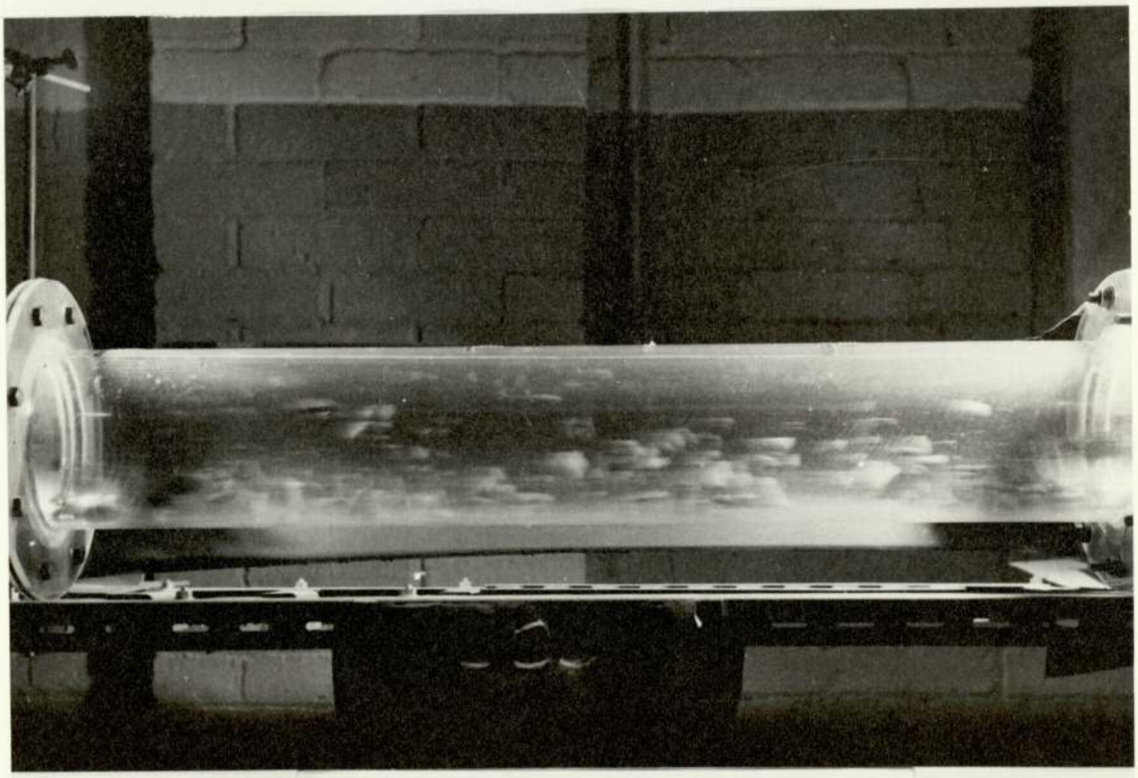


b) 25ft (7·6m) DOWNSTREAM

FIG. 80 FLAKES – FULL POCKET
AIR FLOW 1350 ft.^{3}/min. (0·6m.^{3}/s)

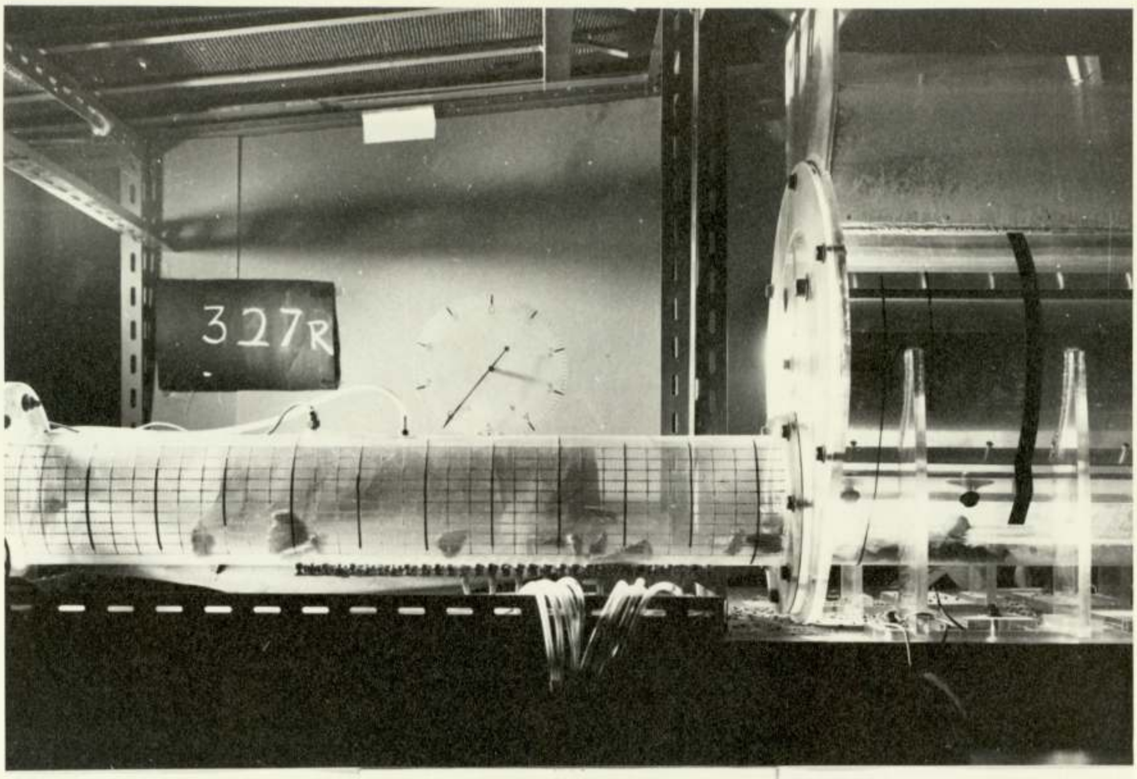


a) STOWER OUTLET

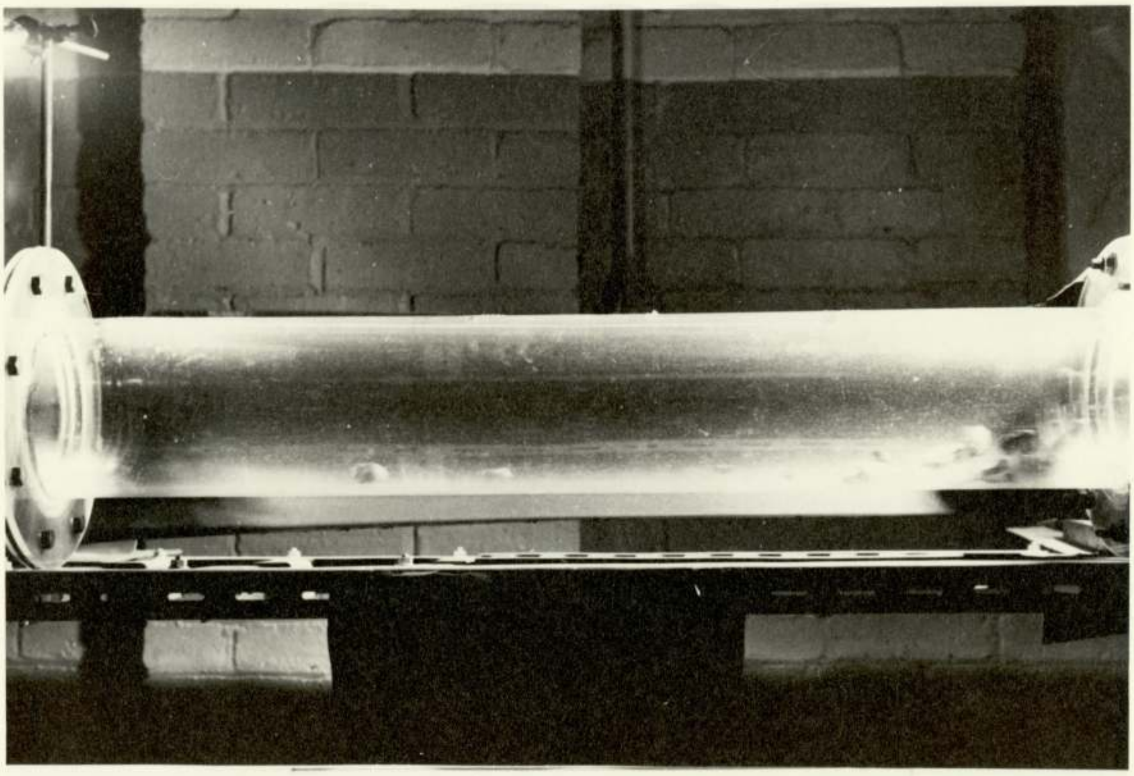


b) 25ft (7·6m) DOWNSTREAM

FIG. 81 FLAKES — FULL POCKET
AIR FLOW 1700 ft.^{3}/min. (0·8 m.^{3}/s)

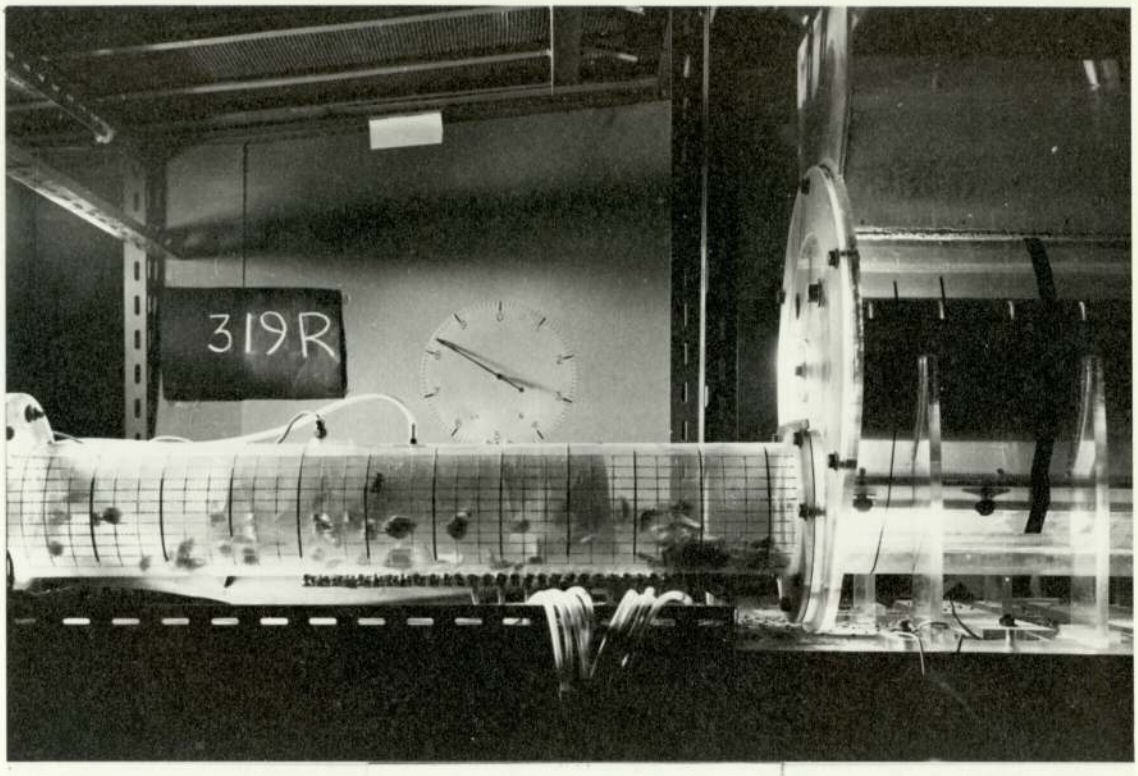


a) STOWER OUTLET

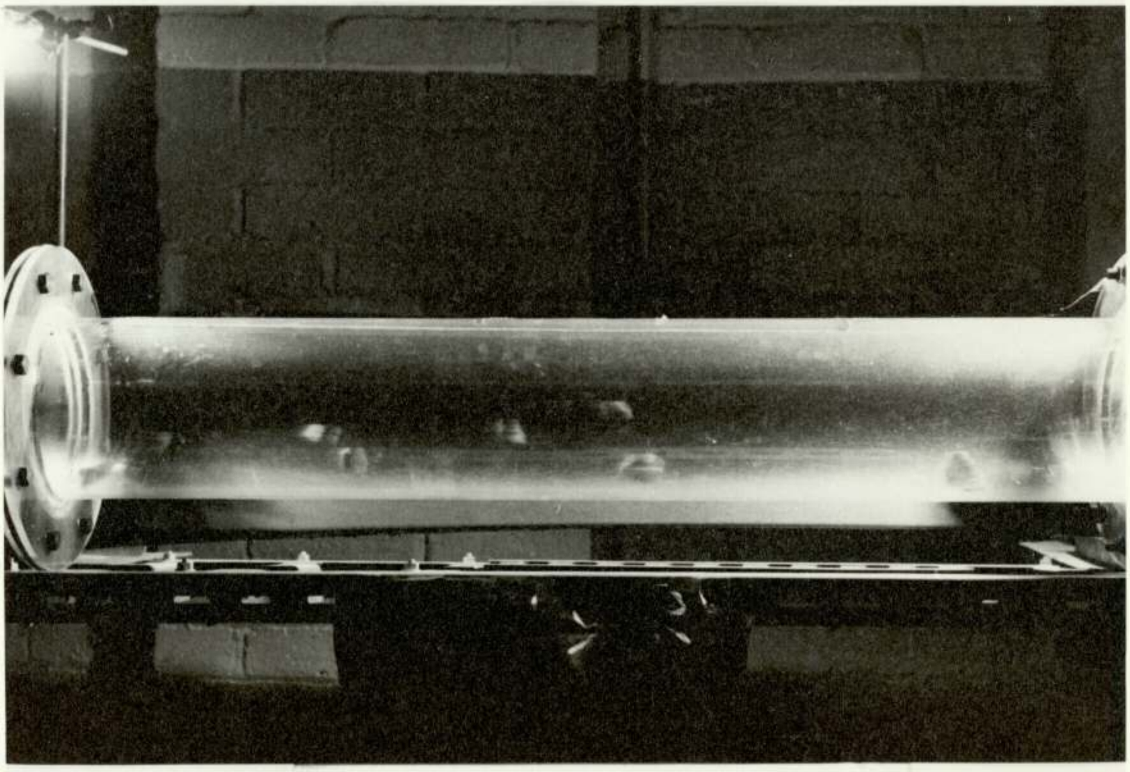


b) 25ft (7·6m) DOWNSTREAM

FIG.82 FLAKES – COVERED POCKET (5·5% FULL)
AIR FLOW 800 ft.^{3}/min 0·4 m.^{3}/s

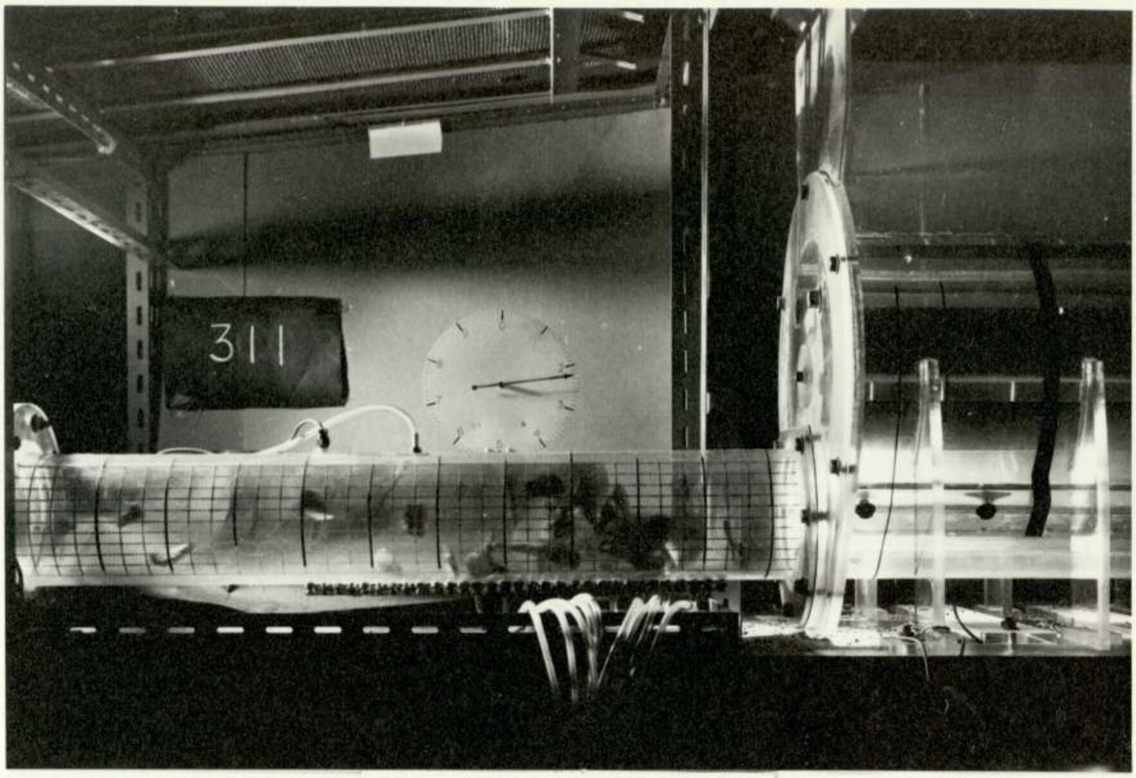


a) STOWER OUTLET

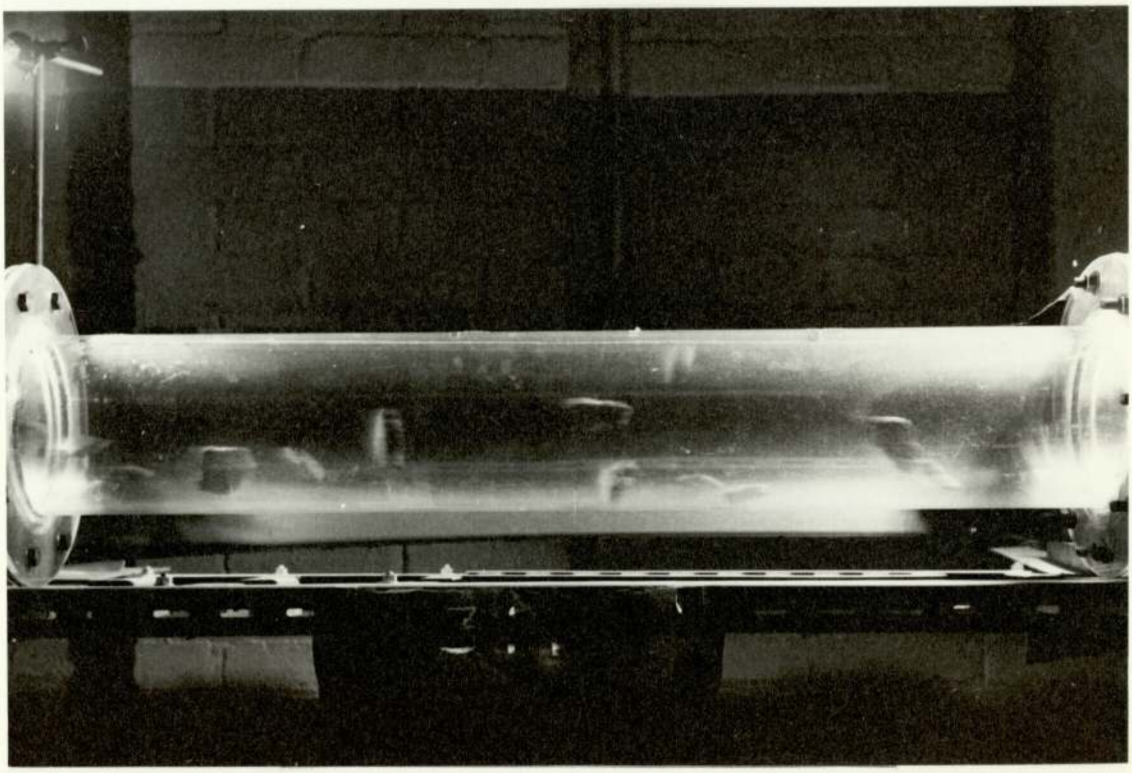


b) 25ft (7·6m) DOWNSTREAM

FIG. 83 FLAKES – COVERED POCKET (5·5% FULL)
AIR FLOW 1100 ft.^{3}/min. (0·5m.^{3}/s)

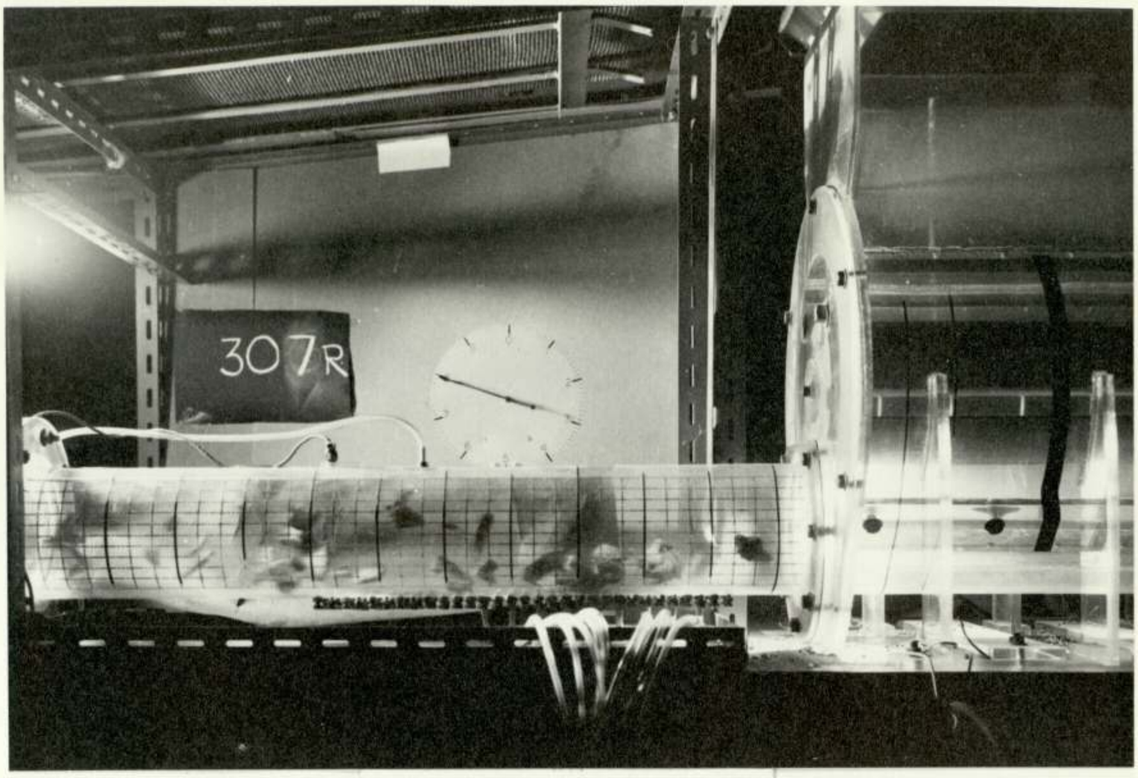


a) STOWER OUTLET

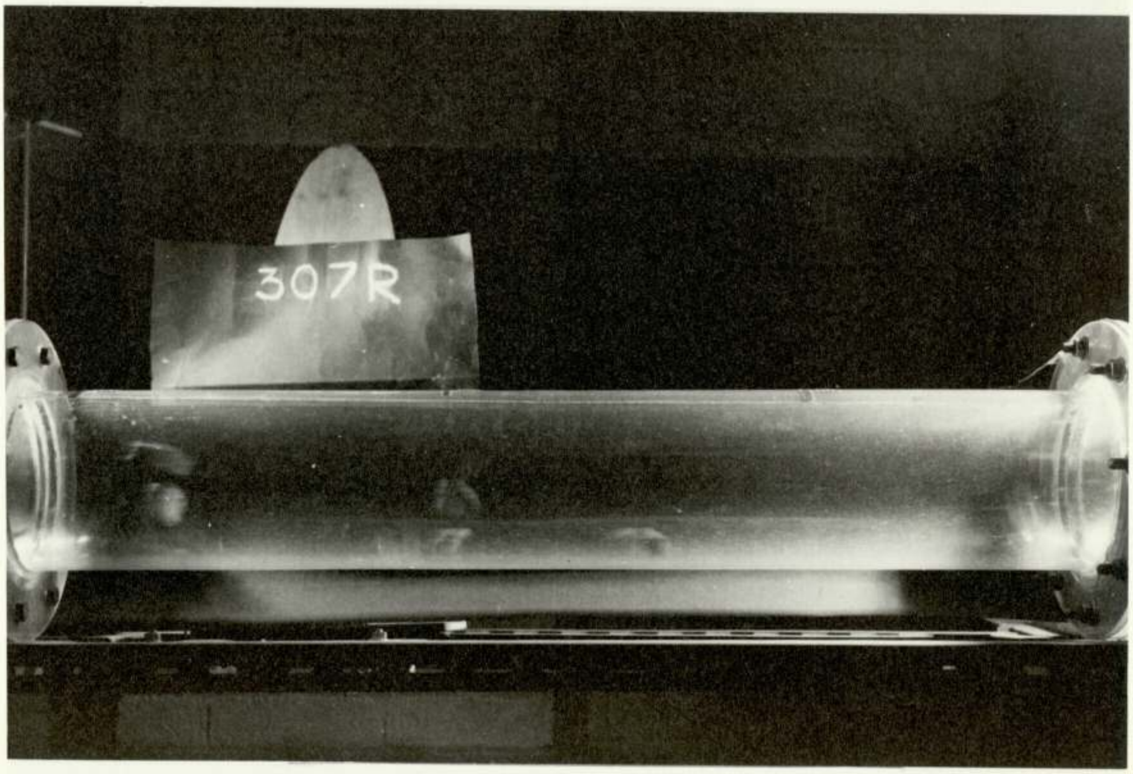


b) 25ft (7·6m) DOWNSTREAM

FIG.84 FLAKES – COVERED POCKET (5·5 % FULL)
AIR FLOW 1350 ft.^{3}/min (0·6 m.^{3}/s)

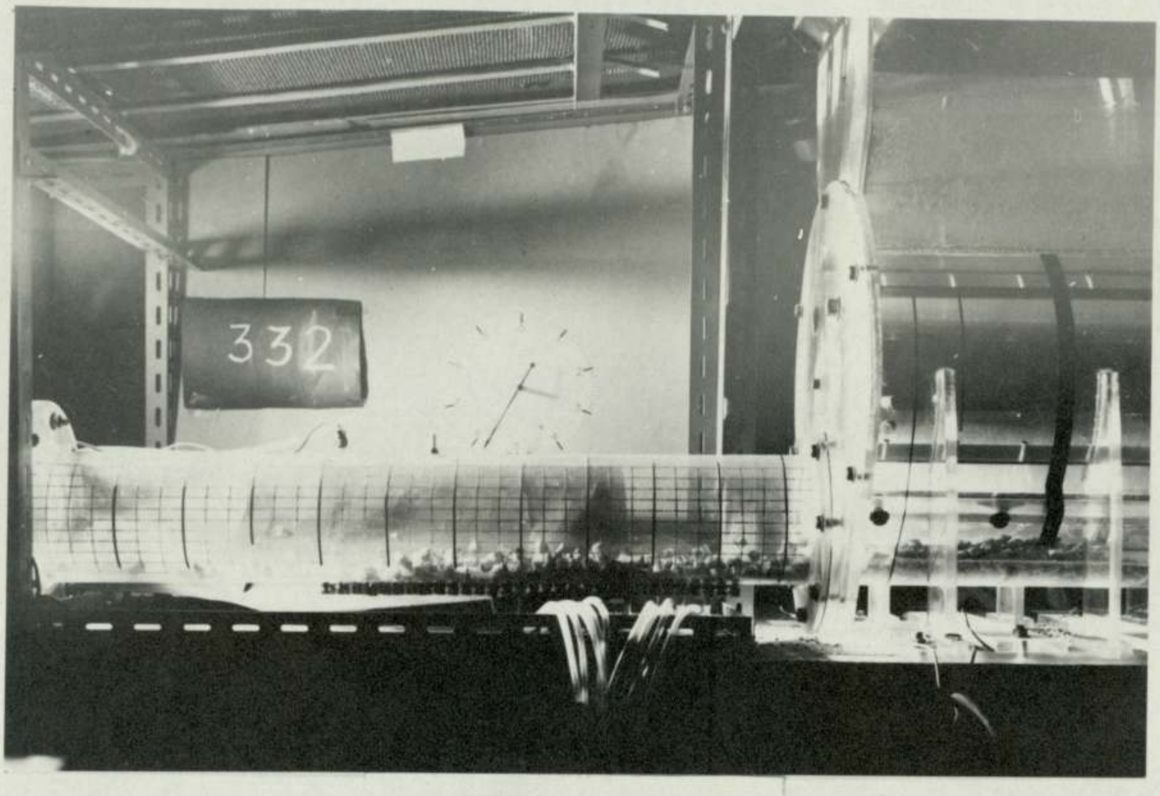


a) STOWER OUTLET

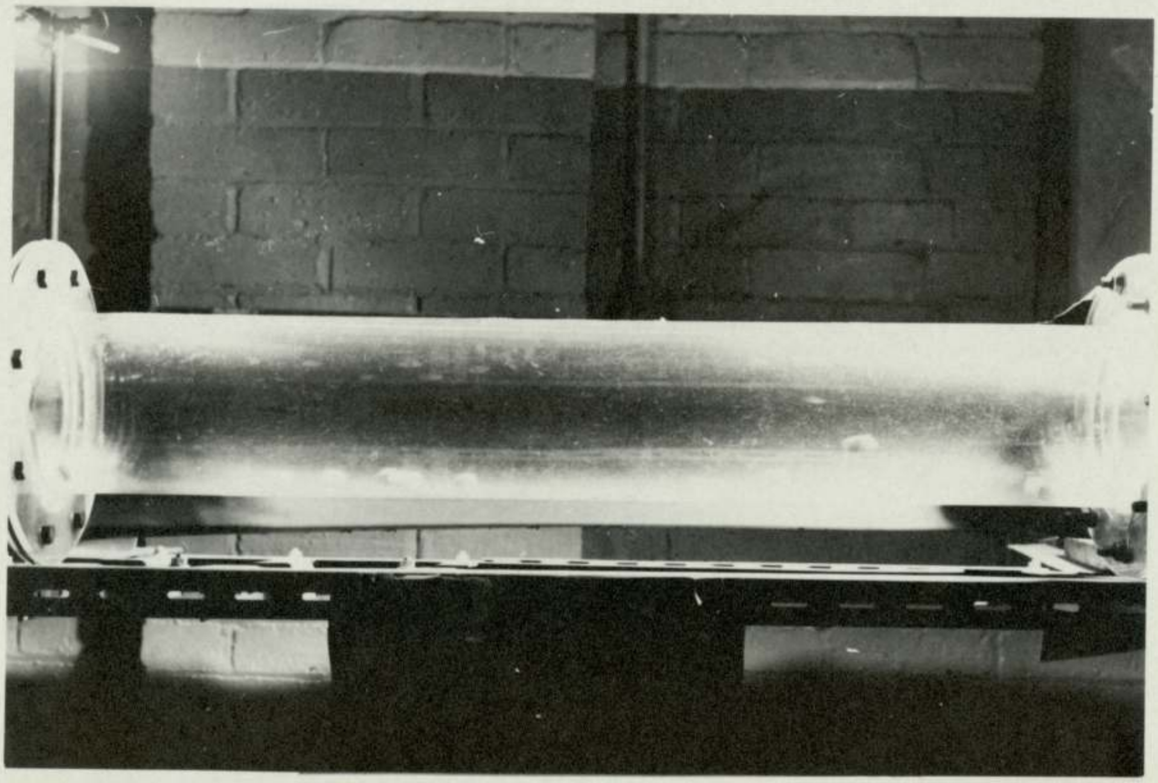


b) 25ft (7·6m) DOWNSTREAM

FIG 85 FLAKES – COVERED POCKET (5·5% FULL)
AIR FLOW 1700 ft.^{3}/min (0·8 m^3/s)

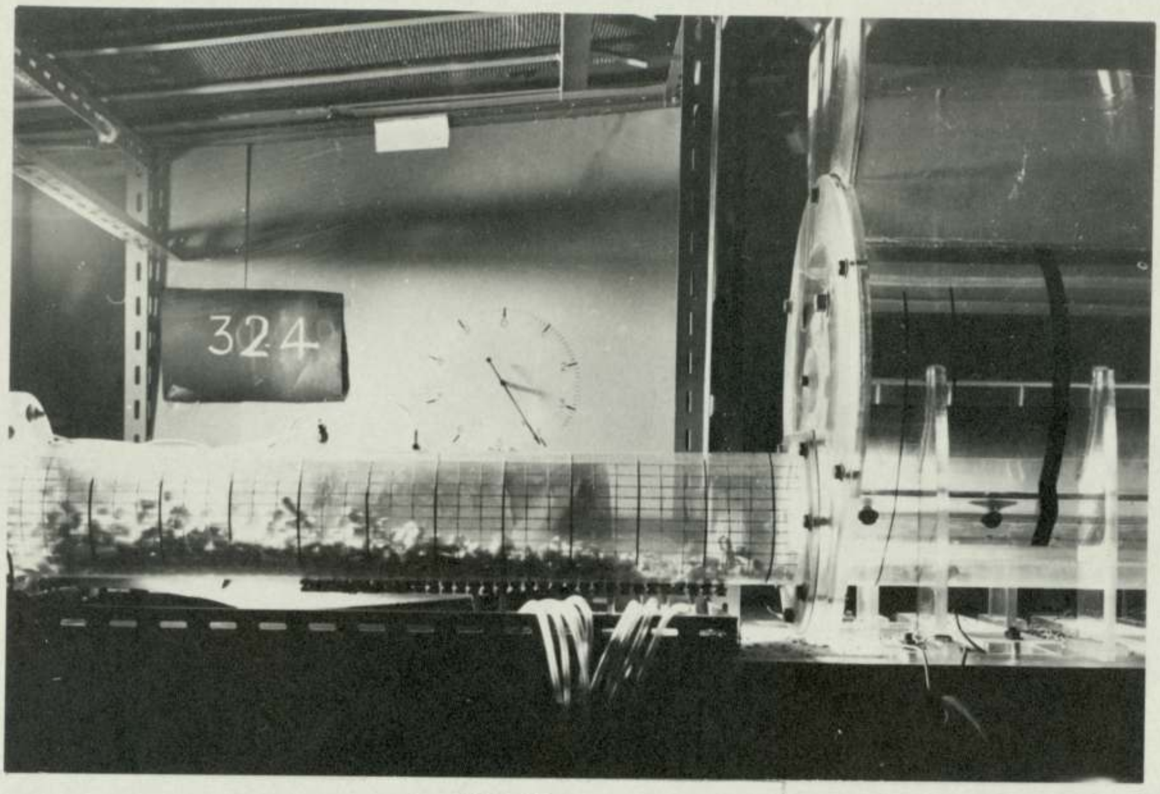


a) STOWER OUTLET

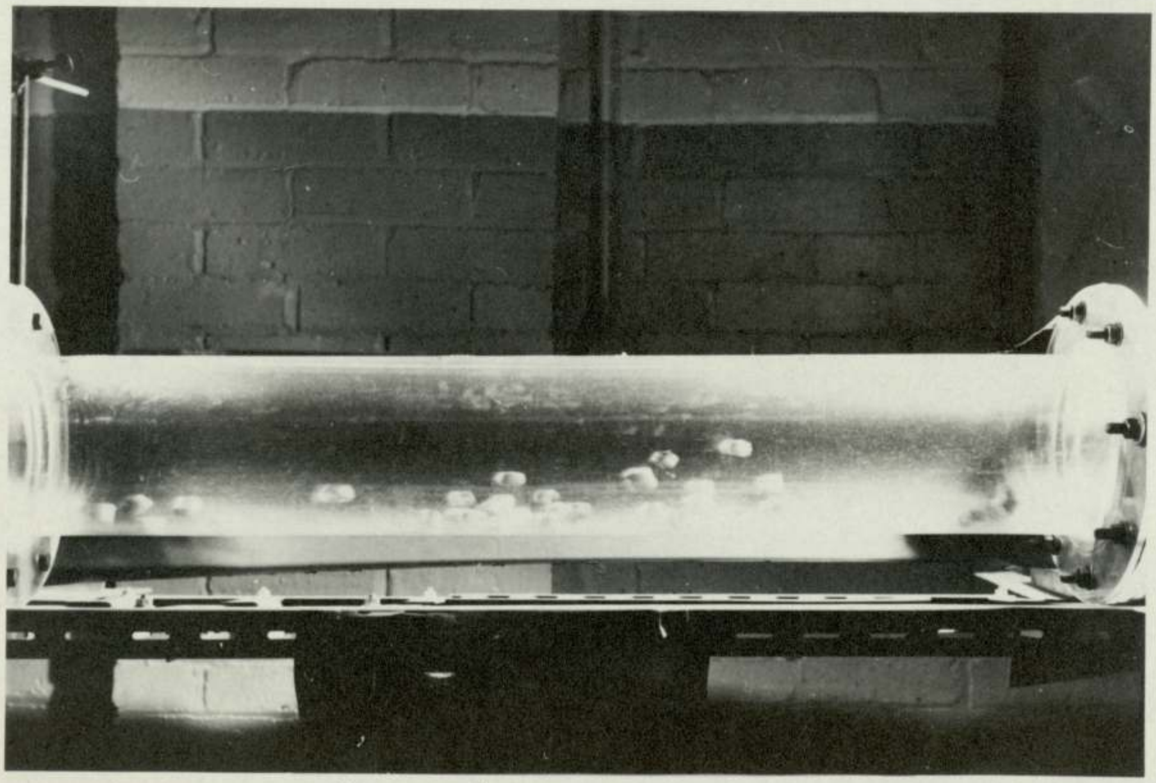


b) 25ft (7·6m) DOWNSTREAM

FIG 86 1/2 IN (13mm) PENCILS - FULL POCKET
AIR FLOW 800 ft.^{3}/min (0·4m^3/S)

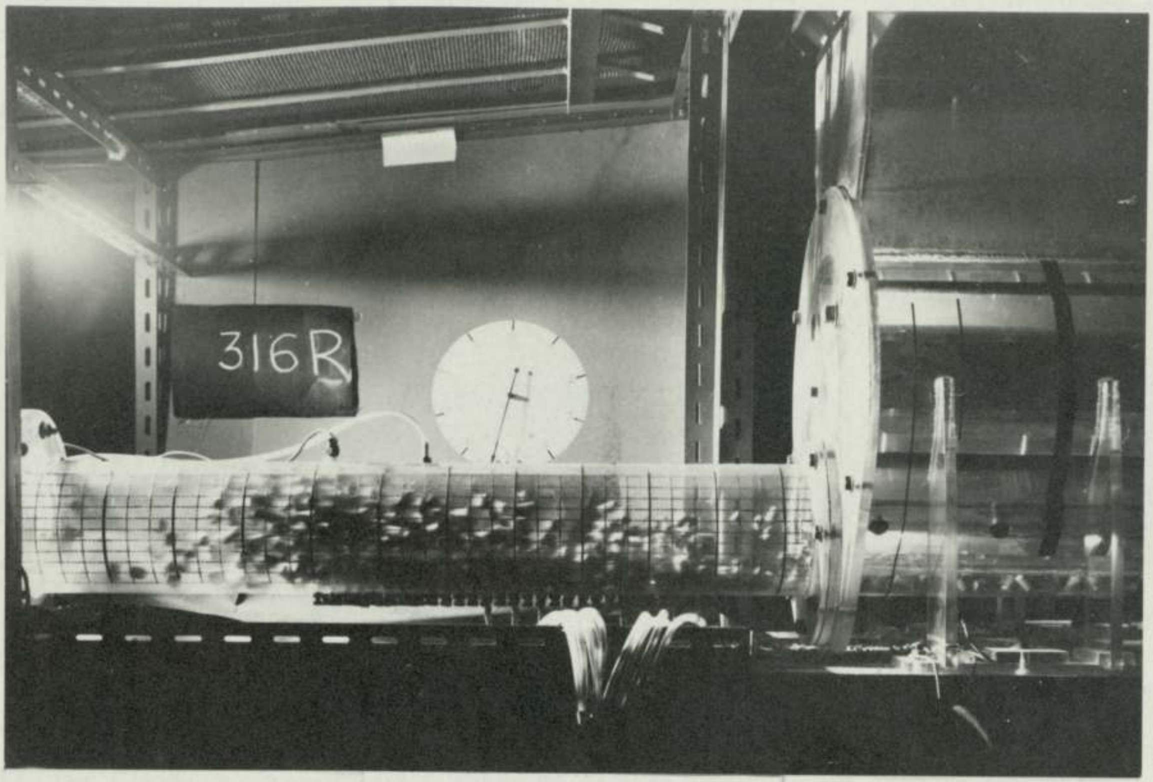


a) STOWER OUTLET

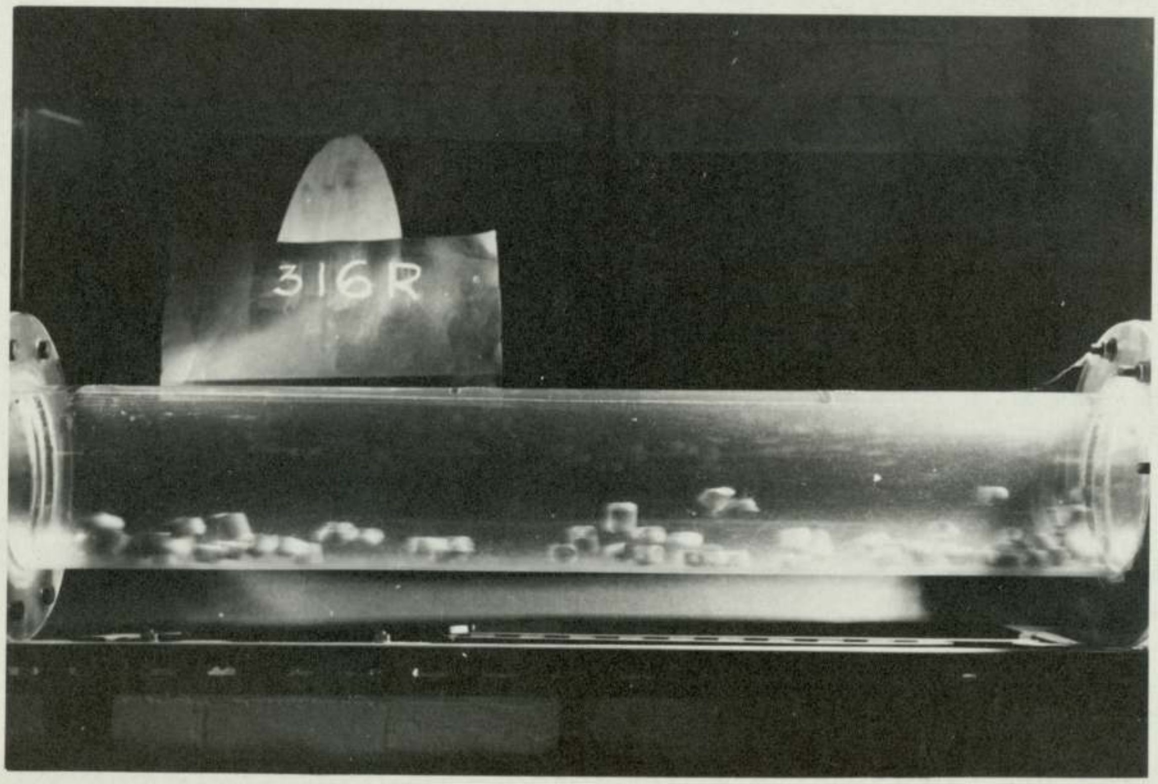


b) 25ft (7·6m) DOWNSTREAM

FIG. 87 1/2 IN (13mm) PENCILS - FULL POCKET
AIR FLOW 1100 ft.^{3}/min (0·5m^3/s)

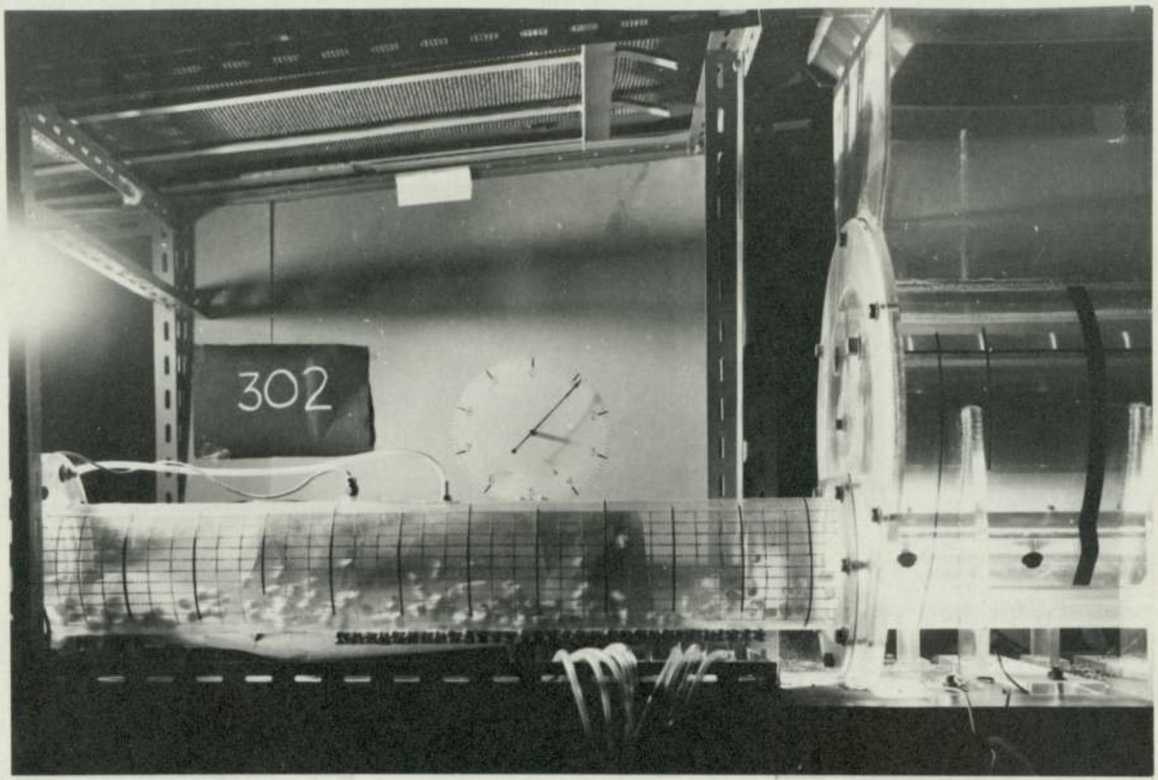


a) STOWER OUTLET

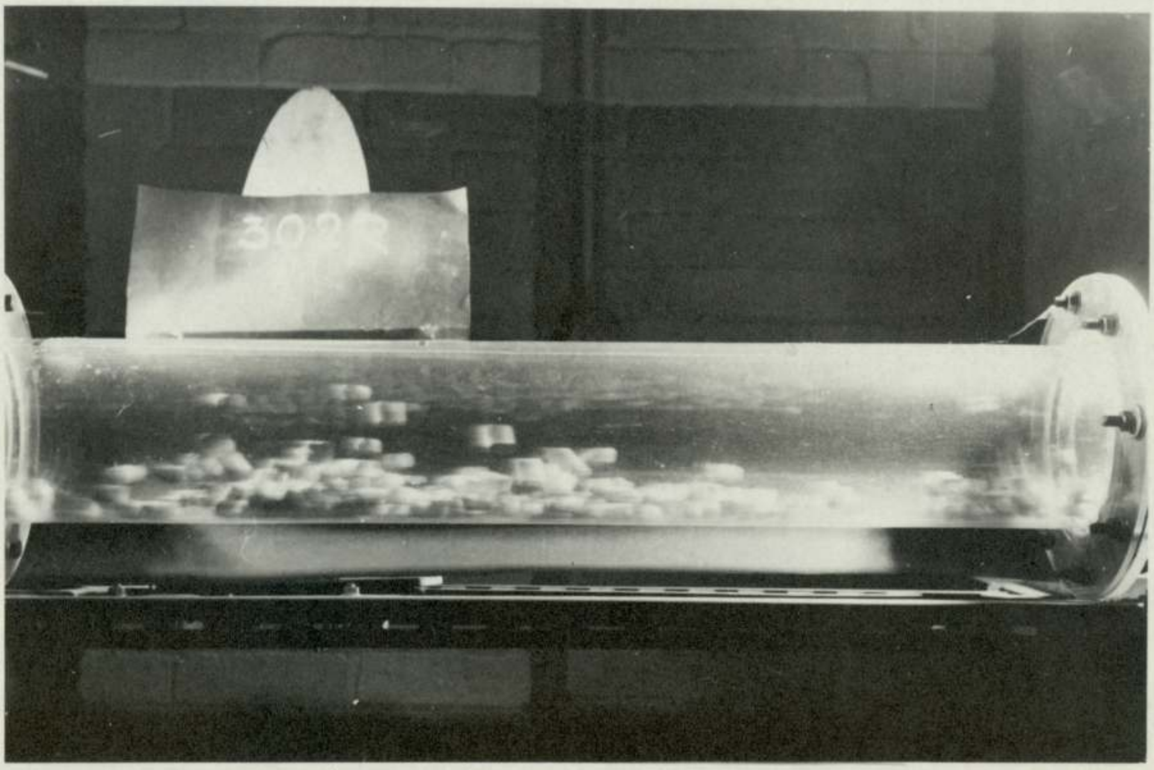


b) 25ft (7·6m) DOWNSTREAM

FIG. 88 1/2 IN (13mm) PENCILS - FULL POCKET
AIR FLOW 1350 ft.^{3}/min (0·6m^3/S)



a) STOWER OUTLET



b) 25ft (7·6m) DOWNSTREAM

FIG. 89 $1/2$ IN (13mm) PENCILS - FULL POCKET
AIR FLOW 1700 ft.^{3}/min (0·8m ^{3}/S)

APPENDIX 3

STOWER MODEL TEST RESULTS

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft^3	Kg / m^3	ft^3/ min	m^3/ S	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
PELLETS (FIG 34)	44·9	720	700	0·35	47	36	18	21

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET S.	AIR TEMPERATURE °C			REMARKS.
	in^3	cm^3	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
BASE COVERED	4·3	70·5	0·11	0·05	>4	18	18	18	MATERIAL NOT CLEARED FROM STOWER & PIPE BOTTOM
1/4 POCKET	71·9	1179	1·87	0·9	"	"	"	"	" "
3/4 "	215·6	3523	5·6	2·5	"	"	"	"	" "
FULL "	287	4703	7·45	3·24	"	"	"	"	" "

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft^3	Kg / m^3	ft^3/ min	m^3/S	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
PELLETS (FIG 34)	44·9	720	1100	0·55	47	36	12	21

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET. S.	AIR TEMPERATURE °C			REMARKS.
	in^3	cm^3	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
BASE COVERED	4·3	70·5	0·11	0·05	0·48	21	21	21	
1/4 POCKET	71·9	1179	1·87	0·9	0·46	"	"	"	
1/2 POCKET	143·5	2351	3·73	1·7	0·51	"	"	"	
FULL POCKET	287	4703	7·45	3·4	0·42	"	"	"	

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C	
	lb / ft³	Kg / m³	ft³/ min	m³/s	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
PELLETS (FIG 34)	44·9	720	1350	0·68	58	42	12	21

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET S.	AIR TEMPERATURE °C			REMARKS.
	in³	cm³	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
COVERED BASE	4·3	70·5	0·11	0·05	0·14	21	21	21	
1/4 POCKET	71·9	1179	1·87	0·9	0·23	"	"	"	
1/2 POCKET	143·5	2351	3·73	1·7	0·30	"	"	"	
FULL POCKET.	287	4703	7·45	3·4	0·38	"	"	"	

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft^3	Kg / m^3	ft^3/ min	m^3/s	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
PELLETS (FIG 34)	44·9	720	1600	0·80	47	36	16	21

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET S.	AIR TEMPERATURE °C			REMARKS.
	in^3	cm^3	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
BASE COVERED	4·3	70·5	0·11	0·05	0·19	21	21	21	
1/2 POCKET	143·5	2351	3·73	1·7	0·16	"	"	"	
3/4 POCKET	215·6	3523	5·6	2·5	0·28	"	"	"	
FULL POCKET.	287	4703	7·45	3·4	0·34	"	"	"	

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft^3	Kg / m^3	ft^3/ min	m^3/s	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
PENCILS (FIG 36)	74·8	1200	850	0·43	80	67	15	22

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET S.	AIR TEMPERATURE °C			REMARKS.
	in^3	cm^3	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
SINGLE PIECE	0·15	2·4	0·01	0·003	0·12	22	22	22	
7 PIECES	1·03	16·9	0·04	0·02	0·2	"	"	"	
BASE COVERED	14·92	244·5	0·64	0·29	—	"	"	"	MATERIAL NOT CLEARED.
3/4 POCKET.	129·1	2114.	5·6	2·55	—	"	"	"	" " "

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft^3	Kg / m^3	ft^3/ min	m^3/s	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
PENCILS (FIG 36)	74·8	1200	1100	0·55	80	67	15	22

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET. S.	AIR TEMPERATURE °C			REMARKS.
	in^3	cm^3	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
2 PIECES	0·29	4·8	0·013	0·01	0·06	22	22	22	
7 PIECES	1·03	16·9	0·04	0·02	0·18	"	"	"	
BASE COVERED	14·92	244·5	0·64	0·29	0·64	"	"	"	
½ POCKET	86·1	1409	3·73	1·6	0·84	"	"	"	
FULL POCKET	172	2819	7·45	3·4	0·88	"	"	"	

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft^3	Kg / m^3	ft^3/ min	m^3/s	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
PENCILS (FIG 36)	74·8	1200	1350	0·68	80	67	15	22

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET. S.	AIR TEMPERATURE °C			REMARKS.
	in^3	cm^3	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
7 PIECES	1·03	16·9	0·04	0·02	0·24	22	22	22	
BASE COVERED	14·92	244·5	0·64	0·29	0·35	"	"	"	
1/2 POCKET	86·1	1409	3·73	1·6	0·84	"	"	"	
3/4 POCKET.	129·1	2114	5·6	2·55	0·42	"	"	"	

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft^3	Kg / m^3	ft^3/ min	m^3/s	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
PENCILS (FIG 36)	74·8	1200	1600	0·8	87	65	14	21

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET S.	AIR TEMPERATURE °C			REMARKS.
	in^3	cm^3	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
7 PIECES	1·03	16·9	0·04	0·02	0·12	21	21	21	
BASE COVERED	14·92	244·5	0·64	0·29	0·24	"	"	"	
3/4 POCKET	129·1	2114	5·6	2·55	0·3	"	"	"	
FULL POCKET	172	2819	7·45	3·4	0·32.	"	"	"	

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft^3	Kg / m^3	ft^3/ min	m^3/S	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
MAIZE (FIG 37)	84·2	1346	800	0·4	55	50	14	19

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET S.	AIR TEMPERATURE °C			REMARKS.
	in^3	cm^3	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
12 PIECES	0·47	7·66	0·02	0·01	0·18	19	19	19	
BASE COVERED	5·78	94·7	0·28	0·13	2·2.	"	"	"	
½ FULL	/	/	/	/	/	"	"	"	RESULT IN DOUBT.
FULL POCKET.	165·2	2704.	8·06	3·7	>5	"	"	"	

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft³	Kg / m³	ft³/ min	m³/S	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
MAIZE (FIG 37)	84·2	1346	1100	0·55	55	50	14	19

POCKET FILL CONDITION	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET. S.	AIR TEMPERATURE °C			REMARKS.
	in³	cm³	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
12 PIECES	0·47	7·66	0·02	0·01	–	19	19	19	TEST RESULT IN DOUBT.
BASE COVERED	5·78	94·7	0·28	0·13	0·31	"	"	"	
1/2 FULL POCKET	82·7	1372	4·03	1·85	0·51	"	"	"	
FULL POCKET	165·2	2704	8·06	3·7	0·82	"	"	"	

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft^3	Kg / m^3	ft^3/ min	m^3/s	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
MAIZE (FIG 37)	84·2	1346	1350	0·68	58	42	12	21

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET S.	AIR TEMPERATURE °C			REMARKS.
	in^3	cm^3	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
12 PIECES	0·47	7·66	0·02	0·01	$<$ 0·05	21	21	21	
BASE COVERED	5·78	94·7	0·28	0·13	0·33	"	"	"	
½ FULL POCKET	82·7	1372	4·03	1·85	0·34	"	"	"	
FULL POCKET.	165·2	2704	8·06	3·7	0·36	"	"	"	

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft³	Kg / m³	ft³/ min	m³/s	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
MAIZE (FIG 37)	84·2	1346	1600	0·8	87	54	16	21

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET S.	AIR TEMPERATURE °C			REMARKS.
	in³	cm³	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
12 PIECES.	0·47	7·66	0·02	0·01	0·14	21	21	21	
BASE COVERED	5·78	94·7	0·28	0·13	0·19	"	"	"	
1/2 FULL POCKET	82·7	1372	4·03	1·85	0·20	"	"	"	
3/4 FULL POCKET	124·2	2032	6·05	2·76	0·26	"	"	"	
FULL POCKET	165·2	2704	8·06	3·7	0·25	"	"	"	

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft^3	Kg / m^3	ft^3/ min	m^3/s	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
FLAKES (FIG 38)	89·2	1425	920	0·46	58	50	17	24

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET S.	AIR TEMPERATURE °C			REMARKS.
	in^3	cm^3	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
SINGLE PIECE	1·23	20·16	0·07	0·03	0·11	24	24	24	
3 PIECES	3·69	60·5	0·2	0·09	0·22	"	"	"	
BASE COVERED	7·26	119	0·38	0·17	0·65	"	"	"	
½ FULL POCKET.	66	1081	3·41	1·55	0·68	"	"	"	

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft^3	Kg / m^3	ft^3/ min	m^3/S	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
FLAKES (FIG 38)	89·2	1425	1100	0·55	35	50	20	21

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET. S.	AIR TEMPERATURE °C			REMARKS.
	in^3	cm^3	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
SINGLE PIECE	1·23	20·16	0·07	0·03	0·08	21	21	21	
3 PIECES	3·69	60·5	0·2	0·09	0·22	"	"	"	
BASE COVERED	7·26	119	0·38	0·17	0·34	"	"	"	
1/2 FULL POCKET	66	1081	3·41	1·55	0·36.	"	"	"	
FULL POCKET	131·9	2163.	6·82	3·1	0·4	"	"	"	

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C.	
	lb / ft^3	Kg / m^3	ft^3/ min	m^3/S	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
FLAKES (FIG 38)	89·2	1425	1350	0·68	58	50	17	24

POCKET FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET S.	AIR TEMPERATURE °C			REMARKS.
	in^3	cm^3	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
SINGLE PIECE	1·23	20·16	0·07	0·03	0·024	24	24	24	
3 PIECES	3·69	60·5	0·2	0·09	0·13	"	"	"	
BASE COVERED	7·26	119	0·38	0·17	0·18	"	"	"	
1/2 FULL POCKET	66	1081	3·41	1·55	0·23	"	"	"	
FULL POCKET	131·9	216·3	6·82	3·1	0·3				

STOWER MODEL TEST RESULTS.

MATERIAL	DENSITY		AIR FLOW		AIR HUMIDITY %		AMBIENT TEMP. °C	
	lb / ft³	Kg / m³	ft³/ min	m³/s	AT BLOWER	AT STOWER	AT BLOWER	AT STOWER
FLAKES (FIG 38)	89·2	1425	1600	0·8	46	50	18	21

POCKET · FILL CONDITION.	MATERIAL VOLUME		MATERIAL WEIGHT.		TIME TO CLEAR ROTOR POCKET S.	AIR TEMPERATURE °C			REMARKS.
	in³	cm³	lb	Kg		STOWER INLET	STOWER OUTLET	DOWNSTREAM	
SINGLE PIECE	1·23	20·16	0·07	0·03	0·04	21	21	21.	
3 PIECES	3·69	60·5	0·2	0·09	0·14	"	"	"	
BASE COVERED	7·26	119	0·38	0·17	0·12	"	"	"	
½ FULL POCKET	66	1081	3·41	1·55	0·22	"	"	"	
FULL POCKET	131·9	216·3	6·82	3·1	0·24.	"	"	"	

STOWER MODEL TEST RESULTS.

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