PREDICTION OF PRESSURE LOSSES
IN
PNEUMATIC CONVEYING PIPELINES

by

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SYNOPSIS

This project arose out of a need to improve the accuracy with which the pressure drop along pneumatic conveying pipelines in process plant could be predicted.

The methods previously available for making this prediction are examined and critically assessed. The need for a different method is shown, and a new approach is developed and tested.

The new approach involves testing of the product to be conveyed, in a test pipeline at the smaller end of the industrial scale, with measurements being made of the pressure drop caused by bends and of the pressure gradients in straight lengths; the data is fed into a storage and retrieval system then extracted and used to predict the pressure drop in a plant pipeline conveying the same product. The method has been developed to the point where it is in current use for the design of pneumatic conveying systems for industrial applications.

The development of a suitable test rig, the data storage and retrieval systems, and the method for predicting the pressure drop in a plant pipeline, are examined in detail. The method is tested against data from pipeline loops and found to give good results.

A quantitative comparison is made against the work of other authors in the field; the results of this show good agreement although the scope of the current work is much wider than anything comparable. An assessment is also made of the areas requiring further work.

A major advantage of the method lies in its use to predict the pressure drop along pipelines having steps up in bore size along their length, which were not amenable to treatment by previous methods. The advantages of such systems and the consequent value of the method are examined in detail.

DEDICATION

This work is dedicated to all the men and women, past, present and future, whose lives have been or will be devoted to the pursuit of the noble art of manufacturing.

It is hoped that through the application and extension of what is contained herein, the continual striving to produce more and better goods from less resources, at lower cost, may be assisted in a small way so that the comfortable, peaceful lifestyle which we lead may be preserved, and shared by more of the people on this planet. If that occurs then the purpose of this thesis will be well served.

M.B.

ACKNOWLEDGEMENTS

First and foremost in these acknowledgements I must cite Mr. I.R. Bittle, previously of Thames Polytechnic (now retired) for the tremendous value of the influence he has had on my development as an engineer during the time I have been working on this project. His input has included continual encouragement and provocation of thought, patient discussion of ideas and strategies, assistance with research into the history and development of engineering technology and finally the reading of the manuscript. He has been a tower of strength, and without his input this work would have been immeasurably poorer. My debt to him is enormous.

Inevitably there are a great many people involved in bringing a research project such as this to fruition. No thought need be given to the relative importance of each, since all are essential; as many as can be brought to mind are listed below, and apologies are due to those who have been missed out:-

Dr. J.S. Mason, Head of School of Engineering at Thames Polytechnic at the time this project was initiated, together with my successive supervisors Dr. D. Mills and later Dr. (now Prof.) A.R. Reed for making it all possible.

H.G. Engineering Ltd. of Canada, particularly in the person of Dr. D. Goodwill, for financial support and encouragement.

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Finally my parents and all my friends, whose encouragement and understanding, especially through the inevitable periods of despair, has given me the strength to bring this work to a useful conclusion.

AUTHOR'S NOTE

All of the work in this thesis is the sole and original work of the author, except where stated otherwise by acknowledgement or reference.

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"If, as is sometimes supposed, science consisted in nothing but the laborious accumulation of facts, it would soon come to a stand-still, crushed, as it were, under its own weight. The suggestion of a new idea, or the detection of a law, supersedes much that had previously been a burden upon the memory, and by introducing order and coherence facilitates the retention of the remainder in memory."

Quotation from Lord Rayleigh's presidential address at the Montreal meeting of the British Association, 1884.

CHAPTER 1

INTRODUCTION

In the 1970's and early 80's there became established at Thames Polytechnic a group of academic staff working on the handling of bulk solid materials, carrying out research and consulting activities particularly related to pneumatic conveying. Pneumatic conveyors have been very widely used in processing, food and chemical industries since before the turn of the century; they offer certain distinct advantages over other types of conveyors, but their design has never been properly understood amongst plant engineers whose first concern, naturally, is with the actual material processing operations in their plants. Even amongst equipment suppliers there has always been something of a lack of proper understanding of the way in which their equipment works. This was recognised by the staff at Thames and efforts were directed at tackling the problem of gaining an understanding of pneumatic conveying, and providing consulting services in this field.

Over a period of several years, a method for predicting the performance of pneumatic conveyors to a degree of accuracy not previously achievable, was evolved at Thames. The development of this was achieved through successive Ph.D. and undergraduate projects. The system which was developed was basically a simple one involving conveying the product for which a plant system was to be designed, in a pilot scale test rig (2, 3 or 4in. pipes) then applying a set of rules, which had been determined empirically, to the data to take account of the differences between the pilot and plant pipelines and thus predict the performance of various options for plant pipelines (often between 4 and 12in. diameter). This process has come to be generally referred to as "scaling". The method was used for several years with some measure of success, but experience showed that it had some serious deficiencies. Some of these were overcome by refinement, but a major problem remained in the scaling procedures; the way in which to account for the differences in the number and positions of bends, between the pilot and plant pipelines, was not satisfactory. This was more significant than may at first be thought, since it was suspected (and has since been shown by the work described herein) that in many systems a major proportion of the pressure drop is caused by the bends. Added to this was the fact that the number of bends on the plant pipeline is usually very different from that on the pilot pipeline, increasing the significance of the inaccuracy. Additional problems arose in that the method could not properly be applied to pipelines with steps up in bore size, then becoming more widely considered for reasons which are explored in full later in this work.

As a result, the need for a project to improve methods of taking account of the effects of bends was perceived, and that is how this project came into being.

The reader will find that as the project began to move forward, the investigation revealed that the original idea of improving the way in which bend effects could be accounted for in the existing method was not an avenue of research which would yield the desired result, for various reasons, and that to achieve the desired result of improving the accuracy of prediction of conveying line pressure drop, a strategy fundamentally different from the Thames testing-and-scaling approach would be required.

What follows in this thesis is essentially an analysis of the reasons why a new approach to the prediction of conveying line pressure drop was needed, followed by the way in which a possible method was identified, a resume of the development of the new method and a detailed description of the method, then some trials to test its accuracy, and finally an overall assessment of its value.

In carrying out this work, the author has become increasingly aware that the strategies which are central to the design of engineering equipment seem rarely to be recognised. The prediction of the behaviour of engineering devices and systems, which enables appropriate designs to be chosen, can be made by a number of strategies or methods, but these methods are rarely identified and described. The community of workers in the pure sciences have evolved a subject known as "The Science of Science", which involves the study of the strategies and philosophy brought to bear in the advancement of these sciences, and yet in spite of the advantages which might be gained by obtaining a similar viewpoint over engineering, a "Science of Engineering" has not yet appeared.

A little consideration shows that Engineering centres around the storage, recall and use of data which has been gathered from experience and experiment. There are several strategies for storing such data, ranging from simple empirical relationships such as Hooke's law or Ohm's law, through mathematical analyses of physical models of mechanisms, such as Poiseuille's law, graphical representations using a curve or family of curves, to complex statistical analyses, with many fine points along the way. In every case, though, the objective is clear; to reduce the mass of disordered data resulting from observations to a compact, ordered and easily handled form which may readily be used to make predictions. The quotation of Rayleigh on the previous page, which summarised the importance of finding some order in the apparently chaotic behaviour of the world about us, was recorded over a century ago but is as relevant now as it was then.

It would therefore seem natural to suppose that an understanding of the various methods, or strategies, used to achieve this objective may be of significant help in attempting to construct suitable means for predicting the behaviour of new mechanisms or devices. Albert Einstein, when congratulated for "seeing further than Newton had" in advancing physics by formulating his General Theory of Relativity, replied that he had been able to see further by standing on Newton's shoulders. Unfortunately such a view does not seem to be widely held amongst engineers working in research, as the reader will find if he examines the literature survey in this thesis; there are cited numerous cases of effort misdirected through a lack of understanding or even awareness of what has gone before.

In recognising the importance of understanding what has gone before, and the strategies which have been adopted by others, one must still bear in mind the importance of maintaining freedom of thought when approaching a problem; the experience gained in working on this project has shown that one should not be unduly swayed by the opinions of others, and should continuously question whether they were moving in the right direction, otherwise it would be possible to become constrained to inappropriate methods. But the importance of understanding the way in which the methods used have come about, and the reasons why, not only in one's immediate field but also in neighbouring fields, has been found to be of far greater importance than is usually recognised. The difficulty of achieving such an understanding is not to be underestimated, since the passage of time and the re-interpretation which occurs when writings are copied from the text-books of one generation to those of the next, distorts and colours the view.

It was in the light of an understanding of what had gone before, not only in the field of pneumatic conveying but also in related fields, that this project progressed. It enabled the limitations of existing methods to be assessed, and led to the recognition of a way in which progress might be made. This led to the discarding of the previous methods and the development of a new, and it is believed better, one.

In concluding this introduction, a few words about the structure of this thesis may be appropriate. The thread of the project and the really significant points are concentrated in the five Chapters, which form a relatively compact central unit; it is hoped that the reader will find it relatively easy to obtain an understanding of the overall project from these, without having the view clouded by too much detail. The essential details of technical work and investigations are distilled and classified into the fifteen Appendices which form the bulk of the pages, and it is felt that this will make the information contained herein more accessible to those who may try to use it. In the same address as that quoted a few pages earlier, Rayleigh said of information which was spoken of as "known", that "....the rediscovery in the library may be a more difficult and uncertain process than the first discovery in the laboratory". It has been the aim of this author to help any reader to avoid such difficulty with this work.

CHAPTER 2

THE INVESTIGATION

2.1 Introduction

The initial brief for this project offered a good deal of freedom of direction, within the area of predicting the performance of pneumatic conveying systems. In order to find a direction, the problems facing a designer, in trying to choose equipment for a pneumatic conveyor, were examined.

An initial study of these revealed that the major proportion of the running cost of any pneumatic conveying system is the power consumed by the air mover, which in turn is dependent on the volume flow rate and pressure of the air required. It appeared that in order to select a suitable size of pipe for a system, it is necessary to consider several sizes and determine the air volume and pressure requirements of each, so that capital and running costs of the various options can be compared. Furthermore, the air pressure determines the type of air mover and solids feeder which will be required for a system, from the vast range of types on the market.

Knowing the importance of the prediction of pressure loss along pneumatic conveying pipelines, a detailed study was made of the methods used to make these predictions. Two principal approaches were identified and critically appraised, the outcome being a desire to evolve a new and better method. The first part of this chapter deals with what was found, and the direction which emerged, which appears in section 2.5.

The remainder of the Chapter, from section 2.6 onwards, deals largely with the planning and execution of the analytical and experimental work.

2.2 Examination of methods for predicting pressure drop

It became apparent that two main approaches to predicting pressure drop along a pipeline had been taken by previous workers in the field. One of these approaches, the one about which most papers have been published, consists of trying to develop analytical models for evaluation of pressure drop along straight pipe sections under accelerating flow and steady flow conditions, and occasionally through bends; this will be referred to here as the "analytical approach". The second method was one of carrying out tests using a laboratory rig, conveying the actual product which the final system is to be designed to handle, then scaling the results to predict the pressure drop to be expected in the final system; this will be referred to as the "testing and scaling approach".

2.2.1 The analytical approach

Most attempts at developing analytical models for prediction of pressure drop began by proposing physical models of interactions between the particles, the air and the walls of the pipe within a mixture of solids and air flowing through a pipe, and continued by applying a mathematical analysis to this physical model to yield equations for pressure drop.

A major drawback of most of the results of these analyses (apart from the sheer complexity of the resulting equations in many cases) appeared to be that values for non-measurable quantities were required in order to use the equations; for example, the true velocity of the solid particles in the line is frequently required, but cannot readily be measured or predicted.

The second problem was the limited range of application of all of this work. Most authors claimed good correlation between their analytical results and experimental measurements over a narrow range of flow conditions, but critical appraisals of this work (Appendix N) demonstrated that there was little common ground between authors, with the analysis of one author rarely correlating well with the experimental results of another.

Additionally, the only type of flow regime within the pipe which was analysed was the condition where all solid particles are carried along in the air in a virtually homogeneous suspension; the condition often referred to as 'lean phase' conveying although the phrase 'suspension flow' is probably preferable. No attempts had been made to analyse the condition where a significant proportion of the solid particles are sliding along the bottom of the pipe or moving in waves, dunes, plugs or other discontinuous motions. This is unfortunate, because many modern conveying systems operate with such modes of flow; but given the lack of success achieved in analysing the apparently 'simple' case of suspension flow, it is scarcely surprising.

Finally, characterising the product proves troublesome to such an approach. Whilst most analyses need a figure for size of the particles, most real products have a wide distribution of particle size. Also it has been demonstrated (eg. ref.1) that products with similar median sizes and particle densities can exhibit significantly different pressure drops under similar flow conditions (i.e. similar air velocities and flow rates of solids).

This method did seem to have one positive advantage, however; if the difficulties could be overcome, the effects of changing the pipeline layout (e.g. altering the number and positions of bends) could be examined in detail since each bend and straight section is dealt with separately.

A more complete breakdown of this work is given in the literature survey, but one particular piece of work deserves mention here as being outstanding amongst the area of analytical models; that of P. Mwabe (ref. 50) who developed models for the pressure drop in straight pipes, in vertical sections, in acceleration regions and caused by bends, and reduced them to a series of algorithms suitable for use on a microcomputer. Users of this have reported it to be a useful guide when designing systems, though again only for suspension flow conditions.

2.2.2 The testing and scaling approach

This method, which appeared to be chiefly the product of D. Mills working with various co-authors, accepted the difficulty arising from the unpredictable effects of different products and sought to avoid them by testing the actual product for which the final system is to be designed. The test rig used is normally on a smaller scale than the final system, typically employing 2in. or 3in. nominal bore pipes, and inevitably has a different pipeline route. In these tests, pressure at inlet to the line is measured for a wide range of conveying conditions (i.e. flow rates of solids and air), and the results obtained are scaled according to a set of rules which were determined empirically, to obtain predictions for the performance of the projected pipeline.

This approach has the major advantage that actual data relating to the product to be conveyed is used for the prediction of pressure drop. The method has been extensively used, both by Mills and his co-workers and by vendors of conveyors, for design of systems which have proved to be successful in operation; not only for systems which operate with suspension flow regimes but also those which employ non-suspension flow. This is evidence that such a method is capable of giving useful results, though not necessarily that the resulting designs are ones having lowest costs.

The differences between test and final pipelines, and procedures for scaling the test results to take account of these differences, were examined in some detail. The possible differences are:

- the length of the pipeline,
- the bore of the pipeline,
- the lengths and positions of any vertical sections,
- the number and positions of bends, and
- the type of bends used (for example, short or long radius).

The procedures for coping with each of these differences are outlined in Appendix B, but some attempts at using them showed that the greatest uncertainty came from the number and positions of bends and the type of bends used.

The approach which has been used to scale for the effects of bends is one of trying to obtain a value for a length of straight pipe equal to the sum of all the straight lengths plus all bend equivalent lengths, for the test and final systems, and then applying the rules for scaling with respect to pipeline length. The very simplicity of the method makes it attractive; the idea of each bend in a pipeline being equivalent to so many metres of straight pipe is one which is easily understood, and it fits in well with the scaling techniques.

The difficulty with this, of course, lies in the determination of the appropriate equivalent length values. Mills (ref. 1) carried out an investigation into this by using several pipelines of the same physical length but with different numbers of bends, to isolate the effect of the bends and thus calculate equivalent length values. His results indicated that the bends in a pipeline can be responsible for a considerable proportion of the pressure drop, and that the value of the equivalent length of a bend changes over quite a large range even for a single product conveyed with a range of flow rates through a single type of bend of one bore size. He found a correlation between equivalent length and superficial air velocity at inlet to the pipeline, as shown below:-

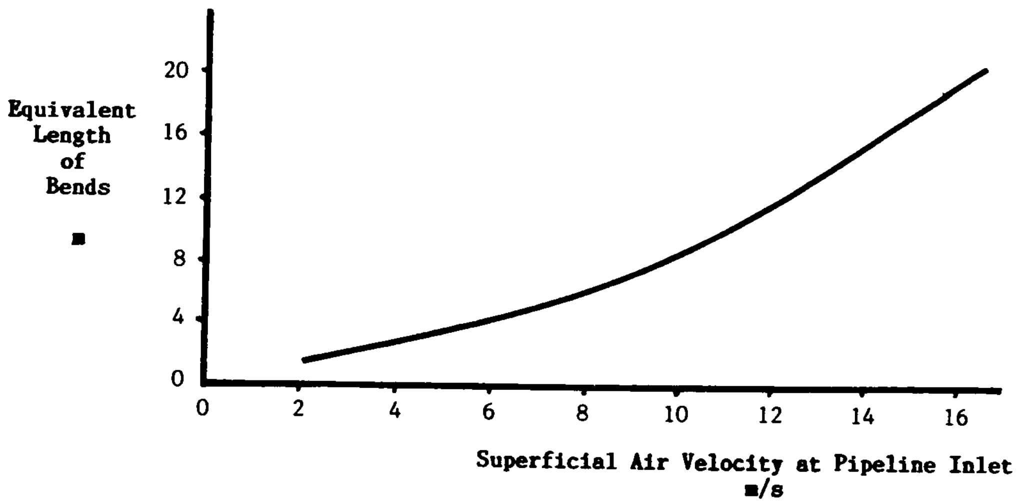


Fig. 2.1

The correlation found by Mills, between Equivalent Length of Bends and Superficial Air Velocity at Inlet to Pipeline, from ref. 1.

This clearly demonstrates that the pressure drop caused by a bend is very dependent on air velocity. However, the pressure drop along a pipeline leads to an expansion of the air and thus an increasing velocity along the line; for example, if a system operates with an inlet pressure of 3 bar gauge and exhausts to atmosphere, the superficial air velocity at outlet will be four times that at inlet. Under such circumstances, the bends towards the end of the line contribute very much more pressure drop than those at the beginning. Whilst the method used by Mills gives a mean equivalent length for all the bends in the test line, most real systems have different distributions of bends. This means that there is an inherent unreliability in using this approach for predicting pressure drop in proposed pipelines.

The results obtained by Mills were re-examined in order to decide whether this difficulty could be overcome, this work being described in Appendix B; however, detailed analysis simply reinforced the view that it could not.

The second drawback of using the scaling approach arises where a "stepped" pipeline is proposed. Pipelines which have an increase in bore size at one or more points along their length have been shown to give very significant energy savings by keeping air velocities down, and are finding increasing favour in long-distance (i.e. over about 300m) conveying applications, as outlined in refs. 54 and 57. The scaling approach cannot be used to predict the pressure drop along such pipelines, nor can it indicate appropriate positions for the steps.

2.3 Consideration of a new approach

At this stage, it was decided to examine the possibility of designing a new method for predicting pressure drop along a pipeline, which would overcome the difficulties inherent in the existing methods.

From the foregoing work, it was concluded that:-

a) The bends in a pneumatic conveying system can be the cause of the major proportion of the pressure drop in a pipeline.

b) The method of testing and scaling used by Mills could not be made to cope properly with the effects of bends; this was shown by the outcome of Mills' work. It seemed likely that any better method would have to deal with each bend and straight length separately, taking account of the increasing air velocity along the pipe, and adding up the contribution made by each in turn to find the total pressure drop.

c) A new approach would have to involve making measurements of pressure drop caused by bends and straight lengths in an actual conveying pipeline, using the product for which the plant system is to be designed, and at the flow conditions to be used in the plant pipeline. To try to predict pressure drop by mathematical analysis of physical models would be most unlikely to provide accurate predictions, even for suspension flow conditions and certainly not for non-suspension flow, for the reasons outlined above.

2.4 Examination of methods used for single phase flow

Whilst the work described above had been progressing, some attempts had been made to look in detail at the methods commonly used for predicting pressure losses in pipelines carrying only single liquids and gases. Only a broad summary of the findings is given here; for more detail, the reader is referred to Appendix C.

One obvious difference between single phase flow and pneumatic conveying was that in single phase flow, the major proportion of the losses arises from the straight pipes with the bends and other fittings contributing little, except in systems where the total length of straight pipe is short compared with the total length of fittings.

It was plain also that the case of laminar flow in pipes is sufficiently ordered to be amenable to analysis using physical models, as shown by Poiseuille (Appendix C refers). It was the method used for dealing with turbulent flow which was really of interest, as explained below.

The means used for predicting pressure losses in straight pipes consists of an equation and a chart. The equation is the Darcy equation:-

$$H_f = \frac{4fl}{d} \cdot \frac{c^2}{2g}$$

This gives the loss H_f in terms of head of the fluid in the pipe, as a function of pipe bore d, length l, and fluid superficial velocity c, with g being the acceleration due to gravity and f being a coefficient, the "friction factor", found empirically. The apparently illogical presentation of the equation in the form shown above (e.g. the 2 not being cancelled with the 4) was considered, and it was seen that there were clear reasons for this, again Appendix C details these.

The coefficient f appeared to be the most interesting part of this system; values for this appear on the Moody diagram as a function of Reynolds number and the relative roughness of the pipe (based on an "equivalent sand grain roughness" of the pipe wall material, itself an interesting concept, enlarged upon in Appendix C). The Moody diagram is reproduced overleaf.

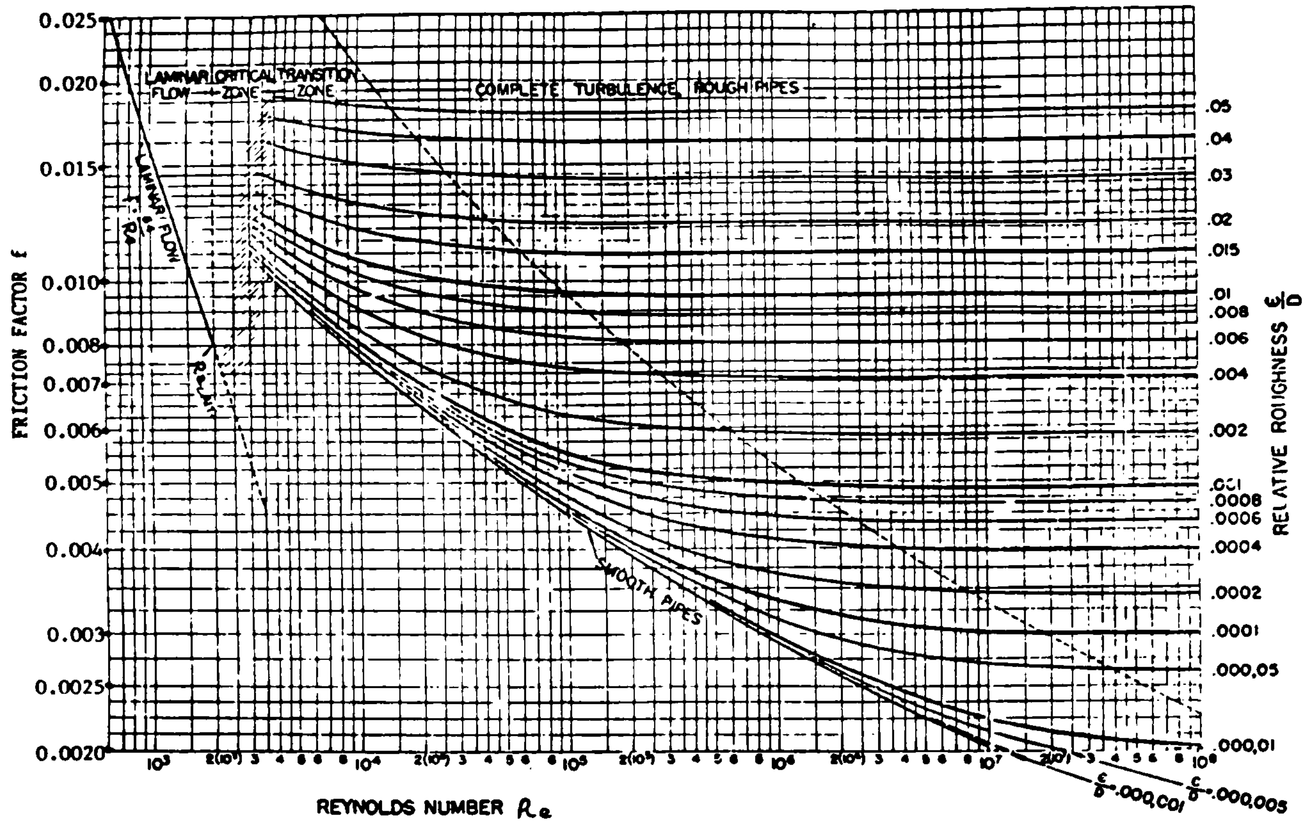


Fig. 2.2
The Moody diagram, from ref. 102

The development of the Moody diagram is detailed in Appendix C. It relates values of the friction factor, f, to the Reynolds number of the flow and the "relative roughness" of the pipe, based on an equivalent sand grain roughness of the pipe wall material.

The lines on the area of the diagram which deals with turbulent flow come from two distinct origins. The "smooth pipes" curve, and the horizontal lines towards the right for "complete turbulence, rough pipes" are lines which have simply been drawn through the experimental data of many workers over a period of many years. The transition curves, however, are from a mathematical expression which was designed to provide a smooth transition between an empirical expression fitted to the "smooth pipes" curve and the horizontal lines to the right. This transition equation was subsequently shown to give a good representation of experimental data in this region.

Thus the Moody diagram, although not derived directly from experimental data, has been built to represent the collected data of many experimenters, and in reading a value from it, the user is effectively recalling that data in order to predict pressure drop in a pipe operating under conditions similar to those under which the data was collected.

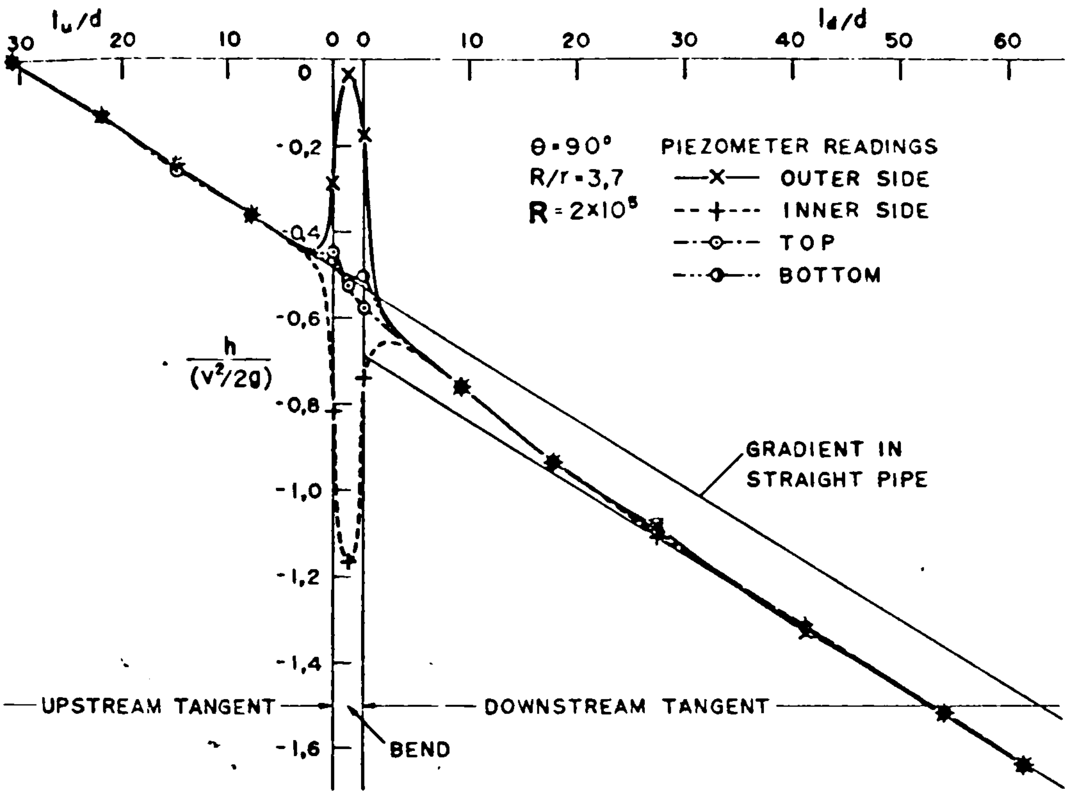
The Darcy equation is dimensionally homogeneous, with the coefficient (the friction factor) being a pure number, features which are clearly desirable; and furthermore, the value of the friction factor is largely independent of the variables in the equation - not completely so because Reynolds number depends upon fluid velocity and pipe bore, but this turns out not to be a major drawback in use.

In order to predict losses caused by bends and other fittings, a system of a simple equation and a coefficient is used. The equation is

$$H_1 = k. \frac{c^2}{2g}$$

with k being a coefficient. This really expresses the loss caused by a fitting as a fraction of the velocity head of the flow, which is not only quite easily understood but also ties in with the Darcy equation above. The coefficient k is listed in tables for bends of various radius of curvature, pipe size, and construction, as well as other fittings such as reducers, tapered transition pieces and so on. Consequently in using this, the engineer is again simply recalling the results of measurements which have already been made.

There is another interesting aspect to this method of dealing with fitting losses, in that it makes use of a technique which might be called "lumping". In reality the fitting loss occurs not actually in the fitting itself, but in the straight pipe downstream, where the disturbed flow is settling back down to its normal profile. But the loss is treated as though it occurred as a single lump, i.e. a step change at the fitting, followed immediately by a return to the normal steady gradient in the downstream pipe. Fig. 2.3 below illustrates this. This removes any problem of modelling the curved shape of the pressure profile along the downstream pipe, and also gives the benefit that when designing a pipeline of constant bore, the engineer simply has to sum the coefficients for all of the fittings and add the total to the 4fl/d term in the Darcy equation, almost as though all the fittings were being grouped together into one lump, i.e. one large step loss.



Pressure distribution along pipeline containing a 90-deg pipe bend ($R/r = 3.7$)

Fig. 2.3

An example of a true pressure profile and step change model, from ref. 103

From the foregoing analysis of the systems used for prediction of pressure loss in single phase flow, it seemed clear that those who were involved with its development must have recognised that physical modelling of turbulent flow was not likely to be useful, because of the complication of the flow patterns (as demonstrated by Reynolds - see Appendix C). Therefore the strategy which was adopted was to simply collect data on pressure losses, and to devise a means for storing this data in a compact form from which it could conveniently be recalled, so that the user would be able to find out what pressure losses have already been measured in a system like the one he is concerned with.

In order to make the data storage system as compact and convenient as possible, use has clearly been made of correlations which must have been found empirically. For example, on the Moody diagram it is seen that the friction factor f can be found from simply the Reynolds number and pipe relative roughness, and appears to be independent of say pipe size or fluid velocity as such, although in practice one might expect it to be affected to some extent by these quantities; this does not prove to be a problem in practice because of the nature of the relationship between head loss and pipe bore as outlined below. Another example is the fitting losses being expressed as a proportion of the velocity head, which would imply that for any given fitting, loss caused is directly proportional to velocity head; the coefficients for the losses are rarely given to more than one significant figure, but errors in these would hardly be noticed in the context of a real pipeline system where such losses are normally small compared with the losses in the straight pipe sections.

Some experience in using these systems for pipeline design showed that because of the limited range of pipe sizes available to the designer, the one suitable size for an application becomes clear when pressure loss (and hence running cost) is compared for that pipeline system in a range of pipe bores. The position of the turning point between uneconomic and economic bore sizes is so clearly defined, and the gap between bore sizes so large in comparison, that even if the pressure loss predictions were in error by as much as, say, 20%, then the same choice of pipe bore would be made. Thus it was seen that in devising the data storage system, even fairly poor correlations could justifiably be used since the inaccuracies they introduce would not change the final choice of hardware whereas the simplifications they lead to would be worthwhile. This is largely why the Moody diagram has been so successful - not because it is accurate, but because it leads to satisfactory design. Again this is expanded upon in Appendix C.

From this, certain important points emerged:-

a) For single phase flow. (apart from the ordered case of laminar flow, modelled successfully by Poisseuille for isothermal flow) the approach of physical modelling appeared to have been abandoned as far as making serious attempts at predicting pressure drop were concerned; although some qualitative understanding of certain effects had been obtained through modelling of velocity gradient effects near surfaces. This reinforced the thought that it was not a promising avenue for dealing with two-phase flow.

b) The approach used for turbulent flow was one of pure empiricism, involving the gathering of data and devising of a system for storing this, by fitting curves and making use of correlations where they could be found. Over the years, the system has been improved and the volume of data within it has increased so that today, a designer has available a diagram representing a set of data which enables him to predict pressure loss in pipelines for virtually any conceivable practical situation - and all this is contained in just an equation and one diagram.

c) The development of this method has taken of the order of a century to complete, with no doubt many false trails having been explored and abandoned along the way (although evidence of these is hard to locate now, having been abandoned for so long). Therefore it should be expected that to develop any comparable system for gas-solid flow would take a great deal of effort even given a clear strategy and modern aids to calculation and data gathering.

d) Accuracy of the predictions resulting from the use of such a system must be commensurate with the use to which the results are to be put; i.e. if a certain error in the predictions would not lead to a change in the hardware selected, then that error is acceptable, but if it would lead to a wrong choice of hardware and consequently an uneconomical design then that is not acceptable.

2.5 A strategic decision

At this stage, the sum of the work so far undertaken was analysed, with a view to finding a direction in which to proceed. The points summarised in 2.3 and 2.4 above were considered, and it was decided that:-

a) To attempt physical modelling of gas-solid flow for quantitive prediction of pressure drop would almost certainly be unrewarding, although some qualitative understanding of processes may be obtained.

b) To improve the testing and scaling approach to enable better account to be taken of pipeline layout would not be possible.

c) Testing of the product to be conveyed would always be necessary to enable accurate predictions of pressure drop to be made.

d) The straight sections and bends which make up pipelines must be examined separately in order to account for their effects properly.

e) Losses caused by the bends should be treated as step changes although it was expected that they would actually occur mainly in the downstream straight pipes.

f) It was unlikely that all bend losses in a pipeline could be lumped together, because of the effect of increasing velocity along the pipe as mentioned above. Therefore each bend and straight would most likely have to be dealt with in turn, working along the pipeline from one end to the other, to predict pressure losses.

The course of action proposed was therefore:-

1) To examine the possibility of designing some experiments to obtain data on pressure losses caused by a single bend in a pneumatic conveyor, and losses along horizontal straight lengths. It was felt that this would not be too difficult to do, and that a fairly large volume of data could be obtained in the time available.

2) To look for correlations in this data which might suggest a suitable system of equations and/or graphs for representing such pressure loss data, broadly along the lines of the system described for single-phase flow.

3) To develop a suitable storage system, feed the data into it, then test the system by predicting pressure drop for some real pipelines from which actual pressure drop figures could be obtained for comparison.

4) A decision on how to answer questions about the effects of type and radius of bend, pipe bore, product, and vertical sections, was deferred until some progress had been made towards obtaining some reliable data.

2.6 Examination of the mechanisms of pressure drop in gas-solid flow.

It had by this stage become necessary to try to understand the mechanisms by which pressure drops along straight pipes and around bends in pneumatic conveyors might occur. Detailed descriptions of the mental models developed appear in Appendix D so only a brief resume of the main points will be given here.

Whilst the work so far described had been progressing, some mental models of such mechanisms had been forming in the mind of the author. It was first thought that any pressure drop over and above that for air alone could only arise as a result of relative motion between the particles and the air causing transfer of momentum (and thus energy) from air to particles. If there was no relative motion then there would be no forces between the air and the particles so the pressure drop would be as for air only. This train of thought relies on there being no relative motion at all, eliminating even the random velocity fluctuations which exist in the air, in order to remove all forces between the air and the particles. An alternative train of thought would be to allow the random velocity fluctuations in the air but with the local velocities of air and particles equal, the effect of which would be to increase pressure drop purely in proportion to the increased density of the flowing suspension compared with that of the air alone (as a result of increased momentum transfer associated with the increased density of the fluid).

Relative motion giving additional pressure drop could arise in two ways, firstly by the air and particles having different speeds and secondly by them having different directions of motion; these will be dealt with separately below.

Although both particles and air travelling in a pipe must have mean velocities in a purely axial direction, different instantaneous directions of motion could arise from the random element of turbulence in the air, from the collisions of particles (with each other and with the wall) causing them to be deflected from a purely axial direction, and from the effect of gravity tending to pull particles to the bottom in a horizontal pipe. Another mechanism was also thought possible, that of particles spinning as a result of entering the region of high velocity gradient near the pipe wall and tending to move across the air stream because of this. It was thought that these effects would be impossible to model in any useful way.

Different average speeds between the particles and the air were thought to arise initially from the introduction of the solids at low speed, then from collisions between particles and pipe wall causing the particles to slow down (because of both friction and the low air velocity near the wall). Thus the solids would always be travelling more slowly than the air, so momentum would be continuously transferred from air to particles. Collisions with the pipe wall would occur particularly where the pipe changes direction, so the re-acceleration of slow moving particles in the straight pipe after a bend was thought to be the main mechanism causing the pressure drop associated with bends. This was thought to explain the observation that the pressure drop caused by a bend occurs mostly downstream of the bend and not in the bend itself.

It was thought that it may be possible to analyse a physical model based on conservation of mass, momentum and/or energy to obtain a value for pressure drop expected as a result of accelerating solids in a straight pipe, which might indicate the order of pressure drop to be expected after a bend. An attempt was made to do this but it did not succeed. Another attempt at a later date was to prove more fruitful, as described in Appendix D; this will be discussed later on.

2.7 Planning of experimental work.

The work so far described had shown the need for direct measurements of pressure drop attributable to bends and straight pipes, and the methods by which these could be made. To deal with bends, measurements of the pressure gradients along the adjacent straight pipes would be needed so that the equivalent step change in pressure could be determined by drawing diagrams similar to fig. 2.3. This method would yield values for the pressure gradients in the straight pipes at the same time.

The re-examination of Mills' work had shown the range of values to be expected for pressure drop in straight pipes and that caused by bends. For the straight pipes, gradients of the order of 10 to 30 mbar/metre would be expected whilst for the bends, equivalent step changes of between 0.04 and 0.14 bar would be expected. Some experience of running a pneumatic conveying test rig had shown that continual fluctuations of line pressure occurred, often of the order of 0.1 bar; this poor signal to noise ratio would mean that averaging of readings over a period of time would be needed to obtain useful data.

It was decided that the necessary data could be obtained from a pipeline having two straight lengths of pipe with a bend between, with pressure tappings along the straight pipes and some means of measuring and recording the pressures at these tappings. The question of the lengths of the straight pipes before and after the bend was considered; a sufficient length of fully developed flow to obtain a reliable pressure profile before the bend would be needed, with another similar length downstream of the bend beyond the region where the bend pressure drop occurs. Values for the length of such a re-acceleration region were sought from literature, the most reliable appearing to be a figure of about 4 metres from ref. 55, which described the results of measuring pressure profiles downstream of a point where solids were introduced into a pipeline. Ref. 39 gives similar figures. From this it was decided that a straight length of at least twelve metres would be desirable downstream of the bend.

The accuracy of the pressure gradient measurements would clearly improve with increasing measuring length, so it was decided that the straight pipes before and after the bend should be as long as practically possible in the available laboratory. This was approximately 18 metres, so satisfying the above requirement for twelve metres minimum. The additional pipeline to carry the solids from the feeder to the test sections and then return to the feeder would result in a loop of length 73m. The test sections would be located as near to the beginning of the loop as possible, to enable tests to be carried out close to the minimum conveying velocity of the product, bearing in mind the increasing air velocity along the line.

The bore of pipeline to be used was considered. The sizes most used in the Thames laboratories were 2in., 3in. and 4in. nominal bore, which are towards the smaller end of the size range used for commercial systems; it appeared that experience had shown (e.g. ref. 1) consistent results to be obtainable from all of these, so the decision was made to use a 2in. n.b. (53mm bore) pipeline initially. Provision would be made for the possible installation of larger pipes (up to 4in.) at a later date.

A feeder for the pipeline was available in the form of a high pressure blow tank having a capacity of $1.5m^3$ and pressure rating of 6 bar (90 psi approx.); the air feed to this was from reciprocating compressors with a combined rating of 600 cfm ($17m^3$ /min free air or .34 kg/s), via receivers, filter/water trap, regulator and choked flow nozzles. The receivers, of volume approx. $1m^3$, damped the delivery pulsations of the multi-cylinder compressors to a very low level, about 10^{-5} of the total pressure. The choked flow nozzles served to control the air flow rates to the blow tank and pipeline, enabling fixed flow rates to be set up. Some experiments were made to assess the suitability of this feeder, circulating a batch of pulverised fuel ash around a 2in. n.b. pipe loop of some 80m length, and it was found that a wide range of conveying conditions could be achieved; air velocities from the lowest at which the product would convey (about 3m/s) to values well in excess of the highest used commercially (over 50 m/s), with solids loading ratios (mass flow of product to mass flow of air) from zero to about 150. This represented a volume ratio of about 7% product to air, approximately the highest known to be used industrially. From these tests, and discussions with colleagues who had experience of the use of blow tanks for a wide range of free-flowing and non-free-flowing products, it was decided that the feeder would be suitable for the purpose, although two shortcomings were noted, namely (1) the inconvenience of disconnecting certain pipework between runs, to alter air flow rate by changing the choked flow nozzle sizes, and (2) some difficulty with manual operation of two large (8in. n.b.) valves when re-loading the feeder after a run, from a recieving hopper mounted above the blow tank.

When it came to choosing the types of bends to test, it became apparent that it might be possible to make a very significant contribution to the technology of conveyor design. It was known that there were many different types used in commercial conveying systems; apart from normal pipe bends of varying radius of curvature, a number of special types, usually claimed (by their manufacturers) to give either lower pressure drop or greater resistance to wear from abrasive products, or both, were known to be in service, and the issue of 'which is best' was seen from the literature to be a contentious one which had never been resolved. If this question could be examined in detail, this might be a very significant and direct contribution to the technology of conveyor design. On this basis, the decision was made to test as wide a range of ordinary bends as are commonly seen, plus some of the more common special types. Details of the bends used are given in Appendix F, but essentially five ordinary bends of different radius and two specials would be used.

The next point to be considered was the means of connection of the bends to the adjacent straight lengths. Normally, screwed 'Crane' or 'GF' unions (as commonly found on gas installations) were used in the Thames laboratories, and these leave a gap of some 10mm or so between the ends of the pipes inside the fittings. In commercial installations, however, flanges are much more commonly used and these usually leave no gap between the ends of the pipes but may give some misalignment depending on the care taken in fitting and the fit of the bolts. It was decided that some investigation could be made by using bends both with unions and without. Those not fitted with unions would be installed using either sleeve joints, or flanges with fitting bolts, in order not to have any discontinuity between the ends of the pipes. After some consideration, it was decided to use "Morris" sleeve couplings which clamped around the outside of the pipe, being much easier than the alternative.

The question of a suitable product to convey was considered. The wide range of behaviour of different products in pneumatic conveyors was known, and this suggested that several products should be used. However, it was clear that time limitations would only allow one product to be used with the comprehensive test programme envisaged. It was decided that it should be one which is commonly conveyed pneumatically in industry, that it should be conveyable over a very wide range of velocities and solids loading ratios, that it should not be unduly abrasive (in order not to alter the profile of pressure tappings in the pipe wall), and if possible it should be cheap. A batch of 600kg of white wheat flour would be becoming available after being used for a consultancy project, and this appeared to fulfil these requirements. It was decided that at least some test work should be undertaken with another product of a different type; some available polyethylene pellets were thought suitable, being nearer the opposite end of the spectrum of products conveyed pneumatically whilst again fulfilling the above requirements.

The properties of these two products were as follows:-

Wheat flour - median particle size 78μm, range 40μm<96%<120μm
particle density 1470kg/m^3
bulk density 510 kg/m^3 poured
angle of repose, poured 37° to horizontal

Product properties, continued:-

Polyethylene pellets - median particle size 4.7mm, practically mono-size
particle density 950 kg/m^3
bulk density 505 kg/m^3 poured
angle of repose, poured 32° to horizontal

It was recognised that because the wheat flour would be used for many conveying runs, over a period of time, it would inevitably change by both particle attrition and biological action. Therefore it was decided that it would be necessary to repeat sets of tests at regular intervals to check the extent to which this would affect the results.

2.8 Development of test rig

The conveying plant (i.e. feeder, air supplies and receiving hopper) was already installed as mentioned above (and detailed in Appendix F). The areas which required development were:-

(a) A pipeline with pressure tappings as described above, plus the necessary selection of bends for testing.

(b) Suitable instrumentation to measure and record the pressures at these tappings, and other measured variables (specifically product flow rate and system operating pressures).

(c) A convenient means of controlling the air flow to the rig, obviating the need to disconnect pipework between runs to change the choked flow nozzles.

(d) If possible, some remote control equipment for the various valves on the conveying plant, to enable it to be turned around more quickly between runs.

The details of the equipment acquired or built are described in Appendix F, but the essential points are outlined below:-

(a) The pipeline. This was built up using 2in. n.b. medium weight steel pipe, assembled using sockets screwed fully home on sections where pressure would be monitored, with screwed unions elsewhere. The ends of the straight lengths adjacent to the test bend were cut clean and square, sufficiently far from the apex that all the bends of varying radius could be installed, with short make-up pieces as appropriate, without moving the straight pipes. These joints would be made using sleeve couplings, ensuring a smooth interior without gaps or misalignment. Pipe roughness was thought to be a possible factor in the pressure drop; pipe of this type is generally taken to have an equivalent sand-grain surface roughness of some .05mm (refs. 102 and 105), giving a relative roughness of .001 and is representative of pipes used for this duty industrially. Fig. 2.4 below shows the layout of the line. Pressure tappings were drilled through the pipe wall, the holes de-burred and special fittings (fig. F-7, Appendix F) welded on the outside; these provided for the fitting of a felt pad filter backed up by a sintered permeable metal disc, a pressure transducer, and a non-return valve through which air could be injected to flush the filter clean of dust between runs.

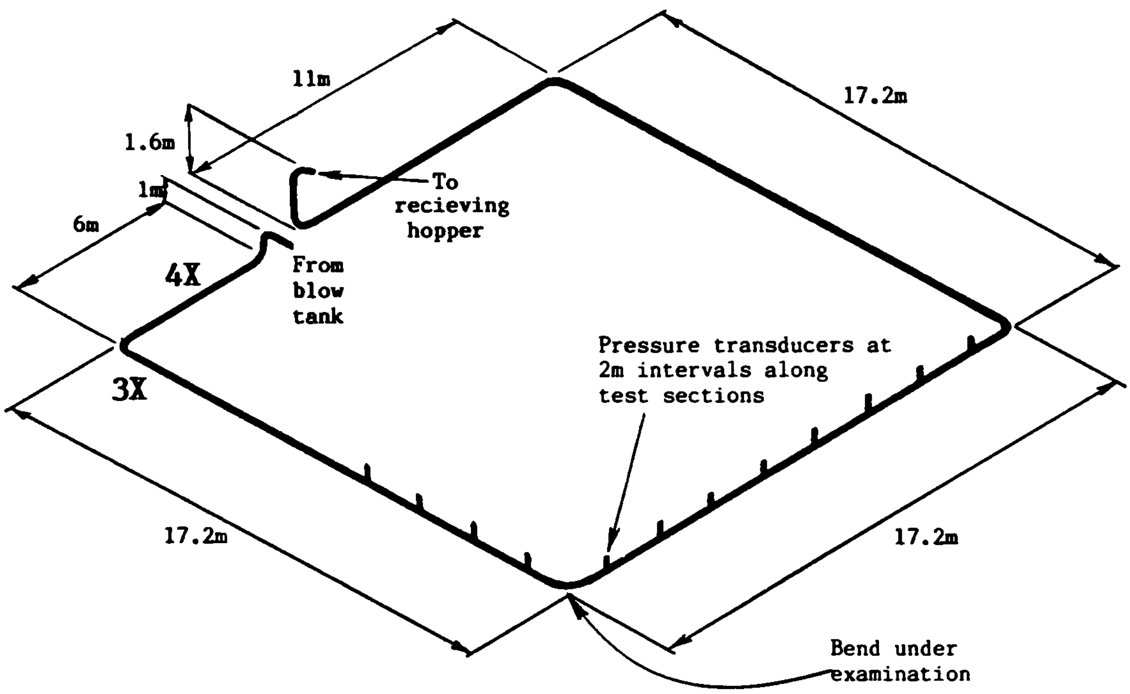


Fig. 2.4
The 2in. n.b. pipeline loop used.
Later 3in. and 4in. loops followed the same layout,
with the expansions to 3in. and 4in. located
at points marked '3X' and '4X' respectively.

The bends of different types and radii (as described in Appendix F), together with the make-up pieces for joining them into the pipeline, were bought in or made up as appropriate.

(b) The instrumentation consisted of electronic pressure transducers fitted to the tappings on the pipeline and tappings adjacent to the choked flow nozzles, monitored by an intelligent data logging unit which in turn communicated with a microcomputer. The data logging unit also monitored the output of load cells on which the receiving hopper was mounted, in order to obtain solids flow rate from gain in weight over a time period. The data logging unit, computer and peripherals were housed in a specially designed cabinet adjacent to the conveying plant, to protect against ingress of dust.

(c) An air flow control system, consisting of two banks of choked flow nozzles with valves in series, was designed and constructed. One bank of nozzles provided air to the blow tank to feed the pipeline with product and the other bank injected 'supplementary' air a little way downstream to dilute the flow. Each bank had eight nozzles in a x2 progression on flow rate, allowing any air flow from nominally 2.3 to 600 cfm f.a. (0.0013 to 0.34 kg/s) to be set up with accuracy, either to blow tank or supplementary air inlet or both.

(d) All valves on the conveying plant and choked flow nozzle bank were fitted with actuators, operated remotely from a mimic panel on the computer cabinet. The starter for the shaker on the filter which cleaned exhaust air from the receiving hopper, and a valve injecting air around the base of the receiving hopper to aid discharge of product when necessary, were also operated from the panel so that the operator was not required to leave his position in front of the computer when using the rig. Positive indications of valve positions, from microswitches, were displayed on the panel and hardware interlocks were fitted where necessary to prevent combinations of valve positions which could lead to dangerous situations. A facility was designed in to allow the rig to be placed under the control of a program in the microcomputer, via the data logging unit and a parallel port on the mimic panel.

2.9 Calibration of Equipment.

Calibration was necessary on the following parts of the test rig and its associated instrumentation:-

(a) The choked flow nozzles, to measure the actual air mass flow rates of each (as distinct from the nominal rates used for design), and also the critical pressure ratio (ratio of absolute pressure at outlet to absolute pressure at inlet above which the flow rate was no longer constant). Measurements were made using an orifice meter manufactured and installed in accordance with BS1042: 1962, with a selection of plates of different bore sizes. Details of the installation and results are given in Appendix H, but essentially only the smallest nozzles departed markedly from design flow rates, with the critical pressure ratio being at least .81 for all nozzles. (Maximum ratio of absolute downstream pressure to absolute upstream pressure at which constant flow rate was maintained). A further subsequent confidence check was carried out by measuring "air only" pressure drop at a particular flow condition and comparing with that predicted by the Darcy equation and Moody diagram, the comparison being within the accuracy of the equipment to measure at such low pressures (.08 bar on a range of 3.5 bar).

(b) The pressure transducers and data acquisition unit, to determine calibration factors for the individual channels. With the conveying line plugged at its end, the plant was pressurised with air to the maximum operating pressure, and a check made for leaks using soap and water solution. Any leaks on the pressure tappings, which could result in false readings, were sealed before calibration began. With the plant pressurised to a predetermined level, as indicated on a certified test gauge, the pressure channels were scanned and the readings recorded. Five pressure values from zero up to the full nominal range of the transducers were used. In order to deal quickly with the number of channels used, a computer program was written to display the calibration data for each transducer on the screen, fit a straight line using the method of least squares, and determine the gain and offset value. The transducers were marked so that they would always be used with the same channel of the data acquisition unit. Details of the procedure and results may be found in Appendix H. The procedure was repeated every time any transducers were disturbed for any reason, but little change was observed over the year or so they were in service.

(c) The load cells and appropriate channel of the data acquisition unit. The procedure for this was very similar to that described in (b) above, but using known weights on top of the hopper to obtain calibration data. Again, detailed results can be found in Appendix H.

2.10 First conveying trials on test rig.

With the control equipment and instrumentation installed, commissioned and calibrated, some initial test runs were undertaken to get the 'feel' of the rig. The first runs were with fairly high conveying air velocities which were expected to yield high pressure drop along the straight pipe and caused by the bend; the bend used was one of short radius without unions, made in house. A test was defined in the data acquisition unit, consisting of 13 scans of all the channels of pressure and weight data at 10 second intervals, giving a test duration of 2 minutes.

Using a blow tank feeder meant that some time elapsed from the moment at which the conveying cycle was started, before a reasonably steady state of operation (in terms of flow rate of product and system pressures) was achieved. Some experiments showed this time to vary between about 30 seconds and two minutes depending upon the proportion of the total air flow directed to the blow tank (and hence the flow rate of product); however, the achievement of such conditions was easily recognised by the operator looking at a continually updated display of measured variables on the computer. Once the operator perceived a steadying of the values, he instructed the data acquisition unit (via the computer) to start the test.

The change in variables was normally found to be relatively small during the duration of the test, typically 5%.

The results from the very first test are shown below; they were averaged manually, and a graph drawn to show the pressure profiles along the pipes near the bend. This gave a very pleasing result (below):-

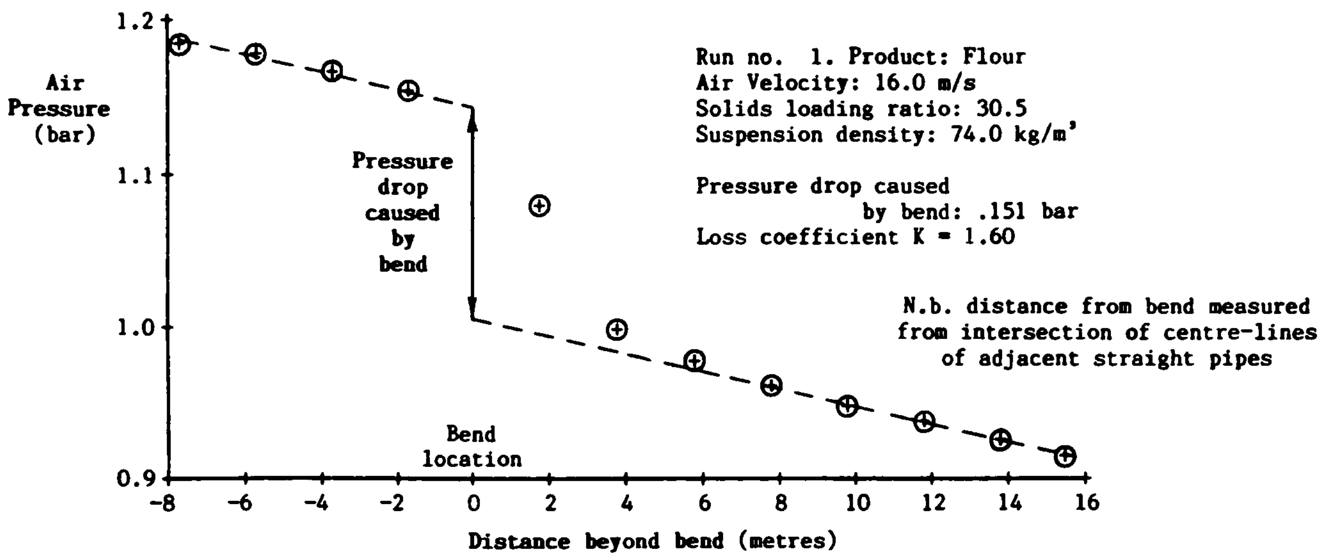


Fig. 2.5

Graph drawn to analyse data from first test run

The same procedure was followed for some 37 runs with a variety of flow rates of product and air, with the graphs drawn manually and the pressure gradient and pressure drop caused by the bend measured off. In all cases, the profiles were very similar to that shown above although of course the values measured off varied.

At this stage, the bend pressure loss data which had been gathered was examined superficially by drawing several graphs; some clear patterns of behaviour began to emerge, which tended to reinforce the opinion that useful data was being obtained. The analysis of this data is examined in Chapter 3 and in greater detail in Appendix K. It was apparent a little later that the bend pressure loss values were in the same order as those expected from considering the energy required to accelerate the particles from rest; the physical modelling had not been brought to a sufficiently advanced state to allow this comparison immediately.

2.11 Development of software to help primary data processing

The manual averaging of data and drawing and measuring of the graphs, to obtain values for pressure drop along the straight pipes and that caused by the bend, had by now become rather tedious, so it was decided to develop a quicker way of doing this. A program was written in BASIC to do this. This analysis of the raw data from a conveying run, referred to here as "primary data processing", took place as soon as the data had been transferred from the data acquisition unit on to floppy disc, following the end of the run.

The program took the raw data (in data bits) and applied the template containing the calibration constants, to produce an array of figures for actual pressures at the various stations, and load cell reading, at the time intervals specified in the test. Then it displayed a graph of line and blow tank pressures versus time during the test, from which the user selected the steady state portion. The program then averaged the pressure values and calculated the flow rate over this period, and plotted a graph of pressure versus position along the test sections on the screen, and asked the user to select the straight-line parts. Parallel straight lines were drawn through the points specified, by means of a least squares analysis, and the user was given the opportunity to review his choice. Finally, the gradients and bend pressure drop values were calculated, displayed and printed out on hard copy together with other data from the run (e.g. mass flow rate of air, air velocity, solids loading ratio, etc), and entered into a data base for later recall. Some of the displays shown to the user during this process are illustrated overleaf:-

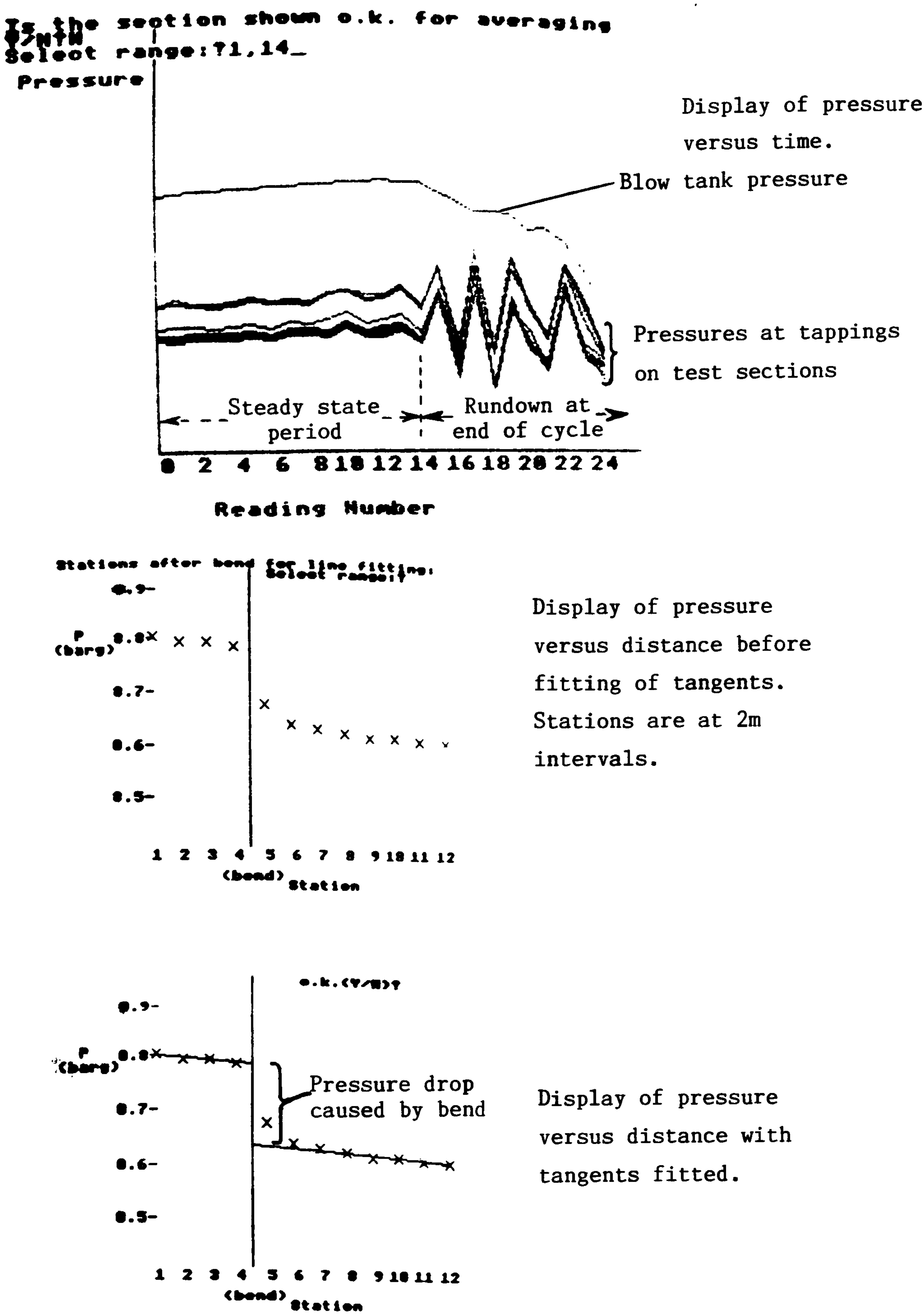


Fig. 2.6

Displays shown to the user during primary data analysis

The evolution of this primary data processing program occurred chiefly during the first few weeks of experimental work, with changes subsequently made to allow for different pipe bores when calculating air velocities etc., and to calculate new variables as desired. A listing of the program can be found in Appendix G, together with the utility program developed to create and handle the data files. All data and programs were stored on 5 1/4 in. floppy discs, each side holding the raw data from 24 runs plus a data file into which the output of the primary processing program was entered, later to be transferred into a single master data file. Again provision was made for space to store new variables calculated from the data, a feature subsequently found to be very useful.

2.12 Execution of the test programme

The way in which the test programme developed, and the decisions made along the way, are described in great detail in Appendix I, so will only be outlined briefly here.

Nine different bends of 2in. n.b. (shown overleaf) were tested with the wheat flour, and one with the polyethylene pellets, covering as wide a range of air velocities and product flow rates as could reasonably be achieved with each. Bends both with and without unions were tested to try to isolate the effects of these. Some sets of tests with the flour were re-run later in order to try to isolate any effects of possible changes in the product as a result of repeated conveying. To assess the effects of air density, one set of tests was re-run several times with the resistance of the conveying line downstream of the bend altered in order to effect a change in absolute pressure, and thus air density, in the test sections.

The pipeline loop was subsequently rebuilt in both 3in. and 4in. nominal bore pipe, with the same layout and arrangement of pressure tappings, and equipment re-calibrated where necessary. Because the blow tank which fed the line had a 2in. discharge pipe, it was necessary to have an expansion at the beginning of the loop; initially this was located at the start of the first test section, which appeared satisfactory when the expansion was only from 2in. to 3in., but led to some peculiar results when using the 4in. loop, and it was subsequently moved right back to the start of the pipeline which proved satisfactory. Again, Appendix I gives a detailed description of this.

Two bends were tested in 3in. n.b. size, and one bend in 4in. n.b. size, all with the wheat flour. (See fig. F-3 for detailed drawings of these.) Again, as wide a range of flow conditions as reasonably possible were covered with each. The upper limit on air velocities was somewhat restricted with the 4in. pipe, because of compressor capacity, and the flow rates of solids were limited by the ability of the blow tank to feed the line. The work with the 4in. line concluded the test programme.

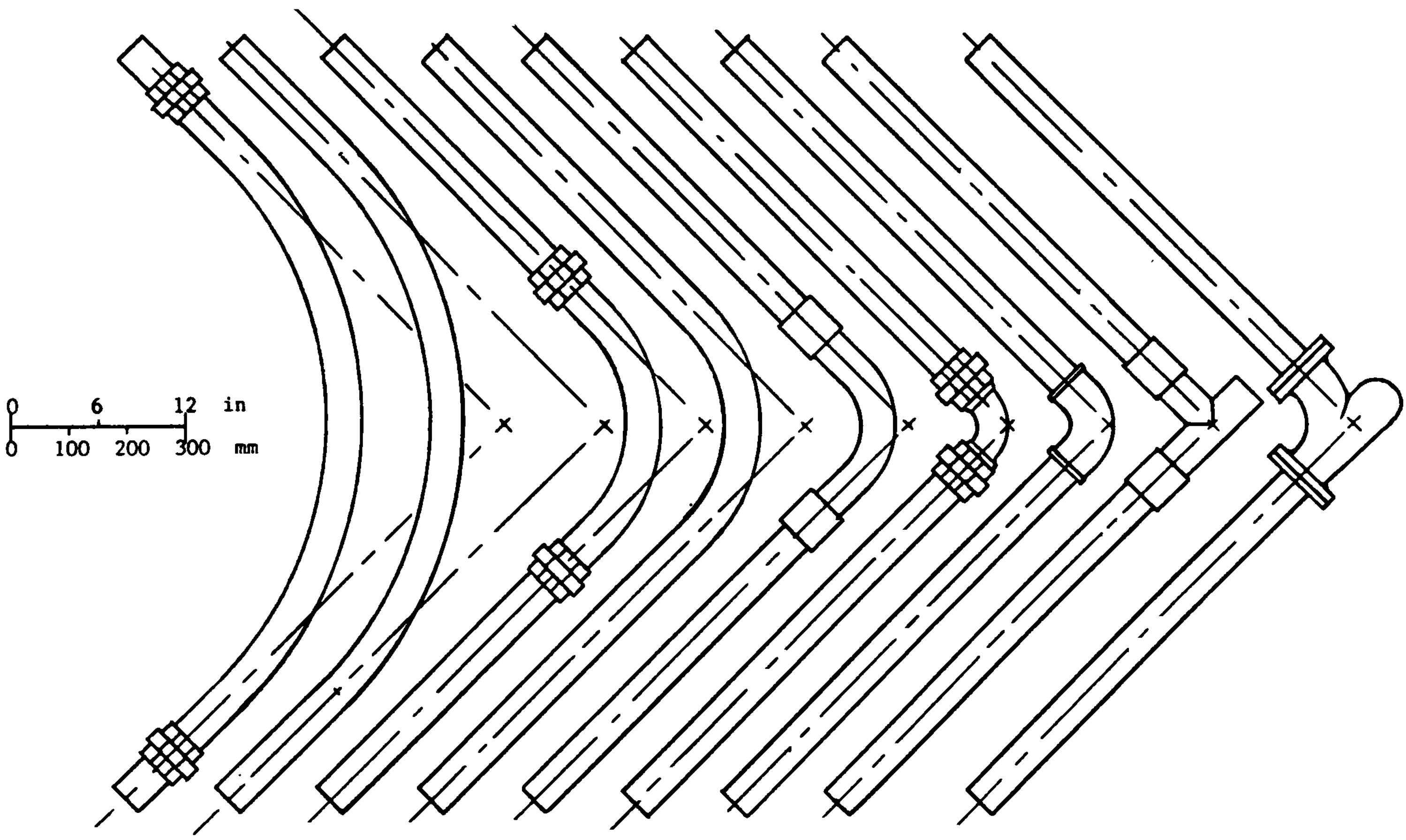


Fig. 2.7
The bends tested in 2in. nominal bore
A detailed description is to be found in Appendix F

A certain amount of data analysis was carried out concurrently with the testing, consisting of no more than plotting measured results immediately on graphs of pressure drop versus solids loading ratio for each chosen value of mass flow rate of air used. This indicated the range covered, gaps to be filled, and any apparently spurious results from tests, which would be re-run to check whether the effect observed was repeatable or not. Some more detailed analysis was undertaken whilst the test programme was proceeding, in order to assess the quality and suitability of the results; where this altered the course of the test programme, it is detailed in Appendix I which deals with the execution of the programme, otherwise all the data analysis is discussed in the next chapter.

CHAPTER 3

DEVELOPMENT OF SYSTEMS FOR STORING THE DATA AND PREDICTING PIPELINE PRESSURE DROP

3.1 Primary data processing

The initial processing of data from conveying runs, referred to in the Chapter 2 as "primary data processing", took place as soon as the raw data had been transferred from the data acquisition unit on to the floppy disc, following the end of the run. A description of primary data processing can be found in section 12 of Chapter 2, enlarged upon in Appendix G.

This chapter deals with the development of the storage and recall systems to handle the data which had been collected.

3.2 The goal and strategy

In order to make the large volume of pressure drop data manageable, it was clear that a system was required which would store it in a form compact enough to be written down easily (e.g. for programming into a computer) and from which it could be extracted conveniently with sufficient accuracy to be useful, for predicting losses in possible pipelines. Two systems would be needed, one for the data on pressure drop caused by the bends and one for the pressure gradients in the straight pipes.

It was decided that the goal should be to develop storage and retrieval systems based on equations which would be dimensionally homogeneous and explicit, involving only measurable physical quantities and coefficients; the coefficients being either constant or capable of being found from a single chart.

The strategy to be employed in developing the systems was not clearly defined at the outset of the work, but developed as the work progressed. Reviewing the work revealed a series of steps:-

a) To search for correlations in the data by drawing graphs of pressure drop versus other measured (or easily derived) flow quantities;

b) To fit simple empirical expressions to the correlations found and find the values of the coefficients necessary to fit these expressions to the data;

c) To look for correlations between the coefficients and the other quantities to see whether their relationships could be represented by simple models, or failing that, represented by a single line on a graph which might in turn be represented by a piecewise linear approximation (i.e. a series of straight lines between limits, for programming into a computer);

d) To check the equations for dimensional homogeneity (i.e. consistency of units on both sides of the equation) and make modifications where possible to improve this, and if possible make the equations easier to understand;

e) To check for the effects of other flow variables on the correlations established, in order to ascertain whether corrections for changes in these would be necessary;

f) To test the system by extracting data from it and using it to predict the pressure drop in a complete conveying line over a wide range of conveying conditions.

Inevitably this did not form a rigid method since it was very often necessary to go back and repeat steps in the light of the outcome of a subsequent step, carrying out iterations and often changing direction; but it is about the clearest summary possible of the strategy behind what constituted some six months of intensive work.

3.3 The data storage systems developed

With the above goal in mind, some considerable time was spent drawing some seventy or so different graphs presenting the data in various ways, looking for correlations between the pressure drop and the other quantities, and trying to find suitable equations to represent them. Much use was made of a microcomputer, at least as much time being spent on continuing software development as on actual examination of data, enabling the desired graphs to be drawn and assessed quickly and easily and hard copies produced as desired.

The development of the systems, the techniques used and decisions taken along the way, are detailed in Appendix K. The outcome is summarised here.

3.3.1 System for bends

The difficulty of dealing with bends had been the inspiration for the greater part of the work described in this thesis; this, coupled with the observation that rather less work had been done on bends by other authors than had been done on straight pipes, seemed to indicate that most effort should be concentrated in this direction.

The system that was eventually adopted, after considering many others, was to use an equation in combination with a graph. The equation was:-

$$\Delta p = K.\frac{1}{2}.\rho_s c^2$$

where Δp = pressure drop caused by the bend, in bar.

ρ_s = notional "suspension density", i.e. kg of product flowing per cubic metre of conveying air (using true volume flow rate of air at the pressure in the pipe, not "free air" conditions)

c = superficial air velocity calculated from pipe cross-sectional area and again true volume flow rate of air, and

K = a coefficient

This expression of the pressure loss as a proportion of a notional dynamic pressure of the flowing suspension is similar to the system used for head losses caused by bends in single phase flow, where the loss is expressed as a proportion of the velocity head of the flow (see section C.6, App. C), the proportion (i.e. the value of the coefficient) being in that case dependent only upon the pipe size and bend geometry.

It will be appropriate at this point to note that this takes no account of product properties, being purely an empirical equation based on other quantities entirely. This should not be taken to imply that the same pressure drop would be expected for different products under the same flow conditions, however; different values of the coefficient K would be obtained for different products. It will become apparent below that not just the value of K but the way in which K varies with other quantities turns out to be different for different products.

It will be apparent that all of the quantities in this dimensionally-homogeneous equation, except the coefficient, are easily derived from measured variables, thereby making it possible to use this equation in a practical application without the need to guess at non-measurable quantities; the coefficient is of course found by the test work and its value for any particular set of flow conditions is in reality where the loss data is stored.

In this case the coefficient could not be made independent of the variables in the equation; however, it was found that for each of the products and all of the bends tested, its value could be represented by a single curve, either against superficial air velocity (for the polyethylene pellets) or against suspension density (for the flour). Examples of the graphs are shown in fig. 3.1 overleaf:-

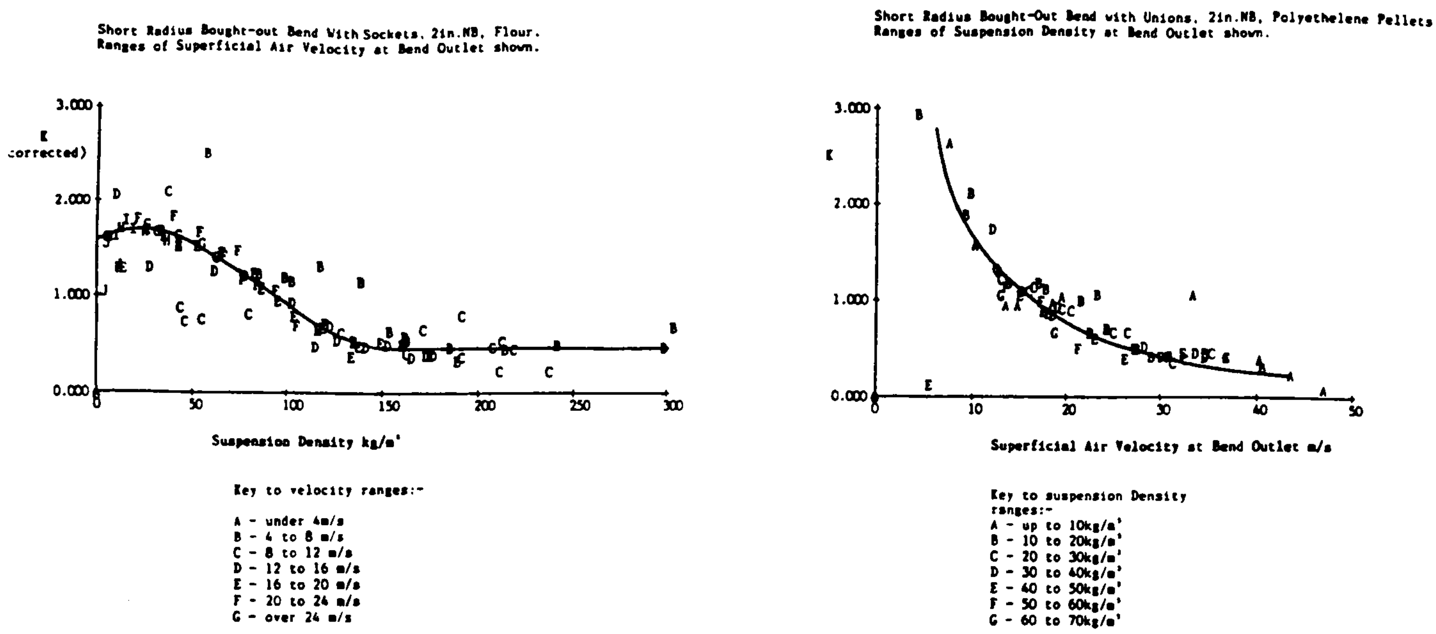


Fig. 3.1

Graphs of bend loss coefficient K versus suspension density and superficial air velocity for the two products in the short radius bought-out bend, from the experimental results obtained with the 2in. n.b. test loop

A major advantage of presenting the data in this way was that neither air density nor pipe bore had any noticeable effect on the K values over the range of 2:1 tested in each case, so simplifying the use of this system. The way in which this was established is described in detail in Appendix K. The effect of product degradation was also examined (the test work involved about 800 test runs with the flour) and it was found that there was a steady reduction in bend losses as the number of test runs increased; the effect (some 20% or so overall) seemed to be quite consistent, so a correction was worked out and applied, to correct experimental K values to the values which would be expected with fresh flour. Again Appendix K details this.

The effect of bend geometry was very significant. It is analysed in detail a little further on in this chapter, but briefly the radiused bends of different radii of curvature exhibited much the same pressure loss coefficient values, varying in the same way with the suspension density. the malleable elbow fittings displaying somewhat higher loss coefficients and the blind tee and Vortice-ell giving the highest of all but varying in a slightly different way with suspension density. The graphs of loss coefficients vs. suspension density for two of these cases are shown in fig. 3.2 below:-

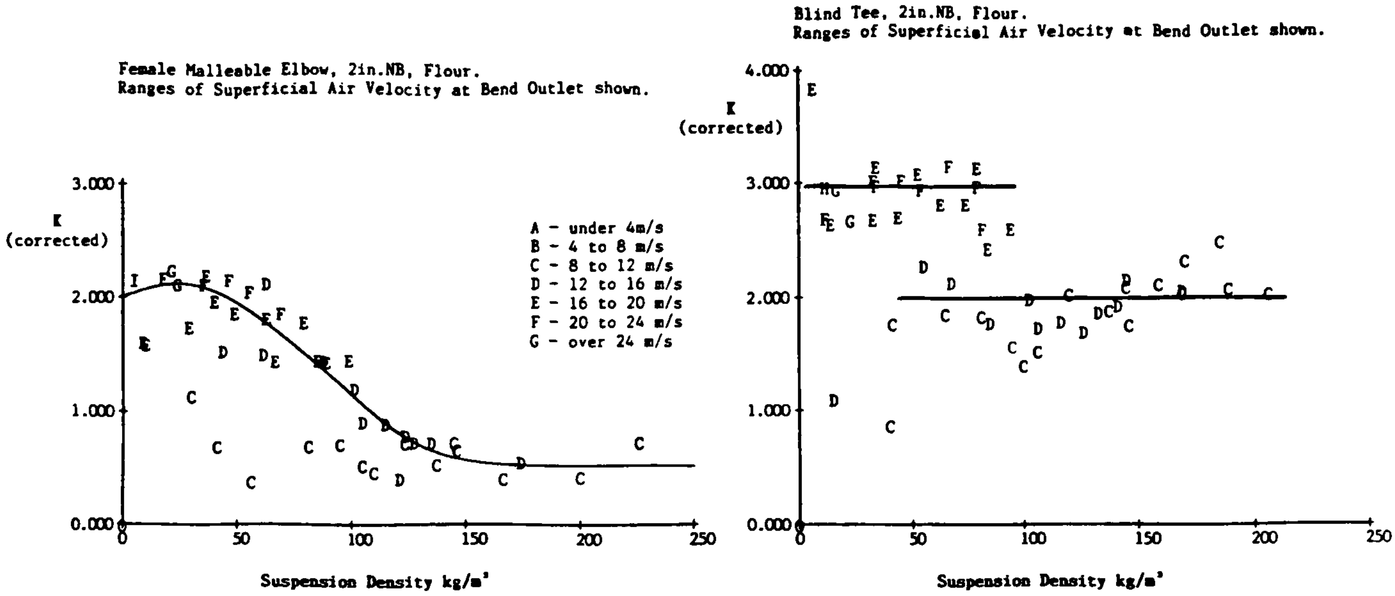


Fig. 3.2

Graphs of loss coefficients versus suspension density for flour, conveyed through the female malleable elbow and the blind tee. These may be compared with the graph for flour in the short radius bought-out bend in fig. 3.1 above.

3.3.2 System for straight pipes

Rather less work was done on straight pipes; a rather imperfect but quite usable system of the following form was developed:-

$$\left(\frac{dp}{dl}\right)_{total} = \left(\frac{dp}{dl}\right)_{air\ only} + \left(\frac{dp}{dl}\right)_{solids}$$

Where $\left(\frac{dp}{dl}\right)_{total}$ = pressure gradient observed in pipe, in bar per metre

$\left(\frac{dp}{dl}\right)_{air\ only}$ = pressure gradient which would be expected with just air flowing in the pipe, predicted using the Darcy equation and Moody diagram, in bar per metre

$\left(\frac{dp}{dl}\right)_{solids}$ = additional pressure gradient, notionally caused by the addition of the solid particles to the air, again in bar per metre.

The additional pressure gradients caused by the addition of the solids could be represented by:-

For the wheat flour,

$$\left(\frac{dp}{dl}\right)_{solids} = 6.5\times10^{-3} \times \left(\frac{\rho_s}{100}\right)^n \qquad \text{where} \quad n = \frac{c}{8}$$

and for the polyethylene pellets,

$$\left(\frac{dp}{dl}\right)_{solids} = 4.4\times10^{-3}\rho_s.c$$

ρ_s and c having the same meanings as above.

These equations are not dimensionally homogeneous and therefore could not be used with other units without reviewing the coefficients; however, the coefficients turned out to be practically constant over the wide range of conveying conditions covered, making the use of charts unnecessary.

At this stage it is worth drawing the attention of the reader to a few points which bear upon the above equations, which are considered in more detail elsewhere. Firstly it is worth bearing in mind that the 'solids contribution' to the pressure gradient in the straight pipe is almost always larger, and usually very much larger, than that caused by air alone, for realistic conveying conditions; with very low suspension densities indeed, combined with moderate velocities, in pipes of small size, the 'air only' contribution can become larger but these conditions are so uneconomic for commercial pneumatic conveying that they are rarely considered. Secondly with regard to the range of applicability of these; the equations were developed to represent the experimental data over the widest range possible, so are applicable with quite good accuracy over this range but not beyond. The exact envelope of superficial air velocity and suspension density will be evident from figs. K-25 and K-30 in Appendix K, but roughly they cover from 4 to 35 m/s and 0 to 220 kg/m^3 for the flour, and from 10 to 45 m/s and 4 to 40 kg/m^3 for the polyethylene pellets. Thirdly it is interesting to compare them with work from other authors; this is difficult because the range of flow conditions mentioned above is far wider than any work previously published, as well as being for different products, however the comparisons which could been made (detailed in Appendix O) show a fair degree of agreement in terms of pattern and order of magnitude of losses. Finally the effect of pipe diameter, not included in the above equations, which may appear strange at first sight; the meaning of this is that for the same air velocity and suspension density, the same 'solids contribution' to pressure gradient is to be expected; conversely for the same pressure drop, solids flow rate is proportional to pipe cross sectional area. This was justified from the results in three pipe diameters measured with the flour, which indicated this to be the case. This of course is quite distinct from the trend with single phase flow, wherein pressure drop reduces as pipe diameter increases, for the same flow per unit area; this could be taken to indicate that the pressure drop in gas-solid flow of reasonable concentration arises mainly as a result of internal mechanisms within the main body of the flow, not much affected by what is going on at the wall of the pipe, whereas in single phase flow the processes going on at the wall are very important.

The approach of splitting the total pressure gradient up into "air only" and "solids" parts was one which other workers in the field had been seen to use, though normally without any justification since invariably little progress was made by these workers towards finding a method for predicting the "solids contribution"; the way in which it was found to be useful is described in Appendix K, but the essential point to note is that it was found entirely incidentally, without (as far as is possible after reading papers by these other authors) any conscious decision to investigate the technique. There is still no apparent justification for using this technique other than the fact that it appears to work satisfactorily, which is considered to be both necessary and sufficient justification in the end.

As an aside, it is interesting to observe that a number of authors of papers suggest the use of a system for calculating the pressure drop in a conveying pipeline (often not recognising the need to distinguish the effects of bends) as a function of the air only pressure drop, typically like so:-

$$\Delta P_t = (1 + C.S).\Delta P_a$$

where ΔP_t = total pressure drop

ΔP_a = pressure drop for air alone

S = mass solids/air loading ratio

C = a coefficient.

This system, which is appealing because it appears delightfully simple at first glance, falls down at determination of the value of the coefficient C; it is at this stage that the reader is generally let down in that no explanation is given as to how to determine a suitable value. However, it may be that this is one of the main reasons why the 'air only' pressure drop has come to be used so much in correlations. The use of a system of this type was considered, but appeared to offer no advantage over simply representing the 'solids contribution' to pressure drop directly as recommended above. It still appears to be sheer coincidence that subtracting an 'air only' pressure drop value from the pressure gradient makes it easier to model empirically.

3.4 Effects of pipeline component options

The effects which changes in various flow quantities had upon pressure drop were examined repeatedly whilst the data storage and recall systems were being developed, in order to try to improve the ability of the systems to handle changes in these variables. Air density has already been mentioned in section 3.3.1 above (and Appendix K), it being found that it was possible to devise a system of data storage which was effectively immune to changes in this quantity, making it unnecessary to take account of this when using recalled data. Pipe bore and bend geometry, mentioned only briefly so far, will be expanded upon here.

3.4.1 Effect of bend radius, unions and ovality of cross-section

The curve drawn through the data for the short radius bought-out bend was scaled to fit the data from the others, and the necessary scaling factor plotted against bend radius; the result is shown in fig. 3.3 overleaf:-

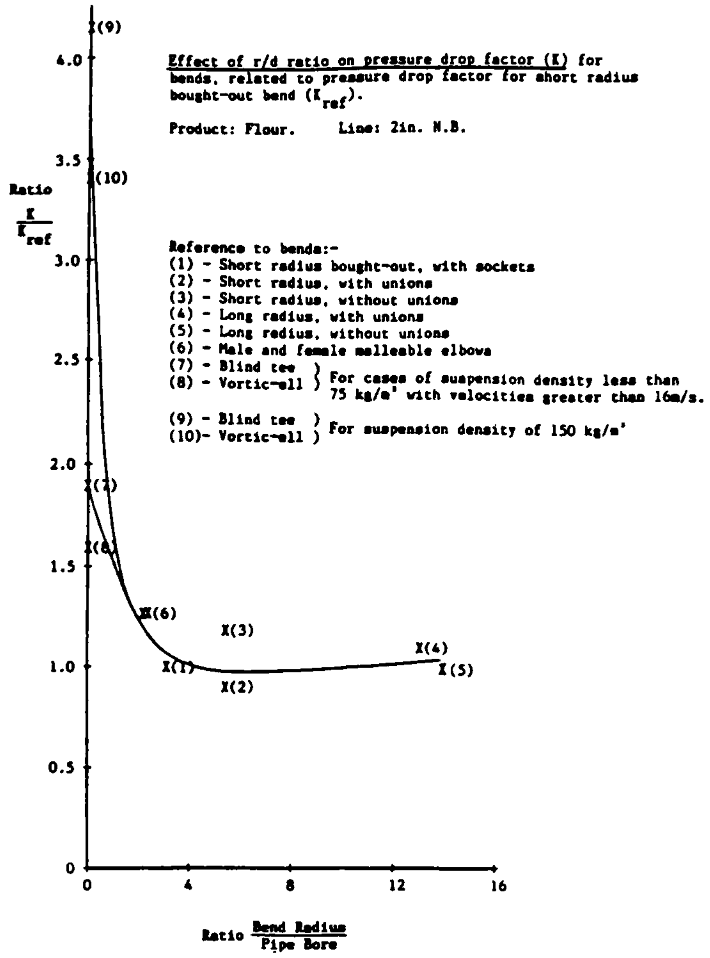


Fig. 3.3

Graph of bend loss coefficients for the flour in various 2in. n.b. bends, compared with those from the short radius bought-out bend, versus ratio of bend radius/pipe bore.

This graph demonstrates the effect of bend radius upon pressure loss. The somewhat different nature of the relationship between K and suspension density, for the blind tee and vortice-ell as against the radiused bends, necessitated a choice of the conditions for which these two were compared with the others; the two chosen conditions being firstly for cases of fairly low suspension density (less than 75 kg/m^3) combined with velocities above 16m/s (i.e. "lean phase", suspension flow conditions) and secondly for a high suspension density (150 kg/m^3), being a "dense phase" condition. As can be seen, the blind tee and vortice-ell displayed noticeably higher loss coefficients than the radiused bends for the "lean phase" conditions (7 & 8 on the diagram), the situation being even worse for the "dense phase" one (9 & 10).

For all the radiused bends, the fact that the variation of K with suspension density followed the same pattern for all means that the values of K/Kref remain constant over the whole range tested. Comparing the radiused bends, the male and female malleable elbow fittings displayed equally the highest losses (6 on the diagram); the short radius bought-out bend (1) was the reference, and displayed the lowest losses except for the short radius made-in-house bend fitted with unions (2) which was slightly lower still; the equivalent bend without unions showed somewhat higher losses (3). The long radius bends (4 & 5) showed losses much the same as the short radius except that the effect of the unions seemed to be reversed, for which no explanation was apparent.

It would appear from this that a very tight radius bend, such as the malleable elbow fittings, or worse still a bend with a pocket on the back such as the blind tee or vortice-ell, results in much more momentum being lost by the particles as they collide with the pipe walls. By contrast the medium and longer radiused bends clearly result in much smaller loss of momentum, presumably because the collisions are at a lower angle as well as spread over a wider area.

The upshot of this is is that the use of long radius bends appears to be unnecessary from the point of view of energy consumption of a conveying system; a bend radius/diameter ratio of 3:1 appears to give virtually as low a pressure drop as any of the longer bends, but these shorter bends are available "off-the-shelf", far cheaper to buy and install than the longer ones which are specially made, heavier and require more space. The very slight saving on cost with the malleable fittings would be more than offset by the greater pressure losses. The blind tee and vortice-ell are particularly costly to run; normally these would only be used to combat a severe bend wear problem which would otherwise lead to a high maintenance overhead, but a radiused bend with a wear-back may be a better proposition in these circumstances depending on the abrasiveness of the conveyed product. In the "dense phase" condition (points 9 & 10) these two are particularly bad, but it would be bad practice to employ these in such an application because wear would be unlikely to be a problem with low velocity flow.

The ovality of cross section of the various radiused bends was noted to be different; therefore it was measured and compared, to see whether this may account for the difference in pressure drop between the radiused bends, but no correlation could be established. The results are shown in Appendix K.

Likewise there seemed to be no clear correlation between unions on the bends and pressure drop, the long radius bend with unions having a higher pressure drop than that without, and vice-versa for the short radius bends.

At this point it is worth noting that although the short radius bends were physically shorter than the longer radius ones, leaving the straight pipes of different lengths, this did not enter the calculations because the pressure gradients before and after the bend were projected to the position of the intersection of the adjacent straight pipes (Fig. 2.5). This approach has the advantage of allowing direct comparison without hindrance, and also allowing measurements direct from plant layout drawings, normally given to the intersection of the straight lengths, to be used in calculation.

bends.

3.4.2 Effect of pipe diameter

When comparing the long radius bends of 2, 3 and 4in. n.b. (which had very similar radii), the loss coefficients appeared to be very much the same although they were somewhat lower for the high suspension densities in the 4in. bend. (See fig. 3.4 below). There was no obvious reason why this may be so, but it does mean (if the effect is repeatable) that using data taken from a 2in. or 3in. test loop will slightly over-predict the pressure drop in a larger pipe, resulting in a conservative design which will at least ensure reliable operation, a more important consideration than absolute lowest possible power consumption.

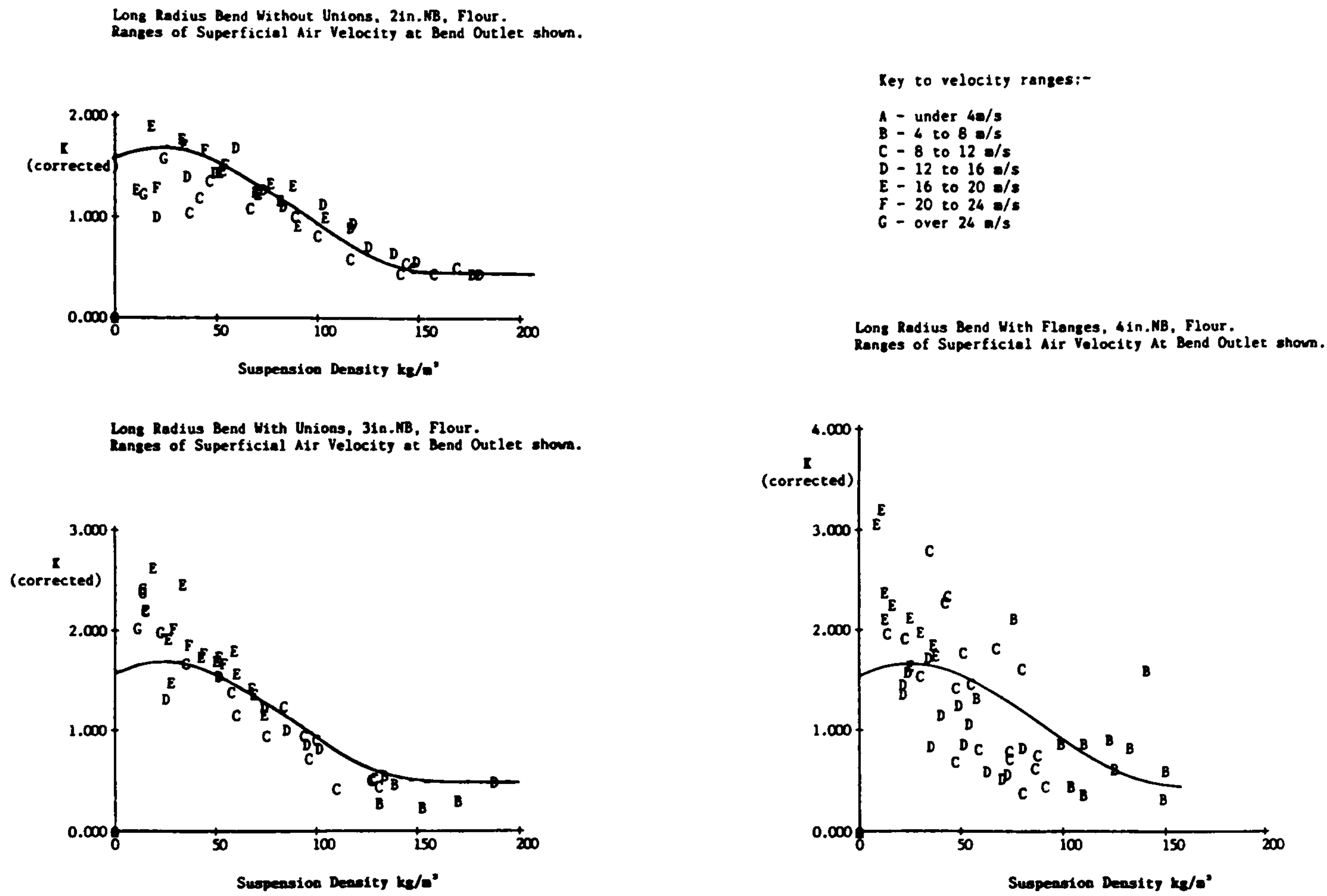


Fig. 3.4

Comparison of loss coefficients for the flour in bends of 2, 3 and 4in. n.b. with similar (long) radii.

In the three graphs above the curve from the 2in. bend has been forced onto the data for the 3in. and 4in. ones purely to show the comparison and demonstrate the level of departure from consistency. It is not clear why the data from the 4in. bend shows so much more scatter, but this was pushing the rig to its limits in terms of air flow availability.

The 3in. female malleable elbow displayed higher pressure drop than the 2in. one, by a factor of some 50% or so over most of the range (see fig. 3.5 below); it is thought that this may be explained by the relatively tighter radius of the 3in. fitting (ratio of bend radius/pipe bore of 1.8 as opposed to 2.3 for the 2in. fitting).

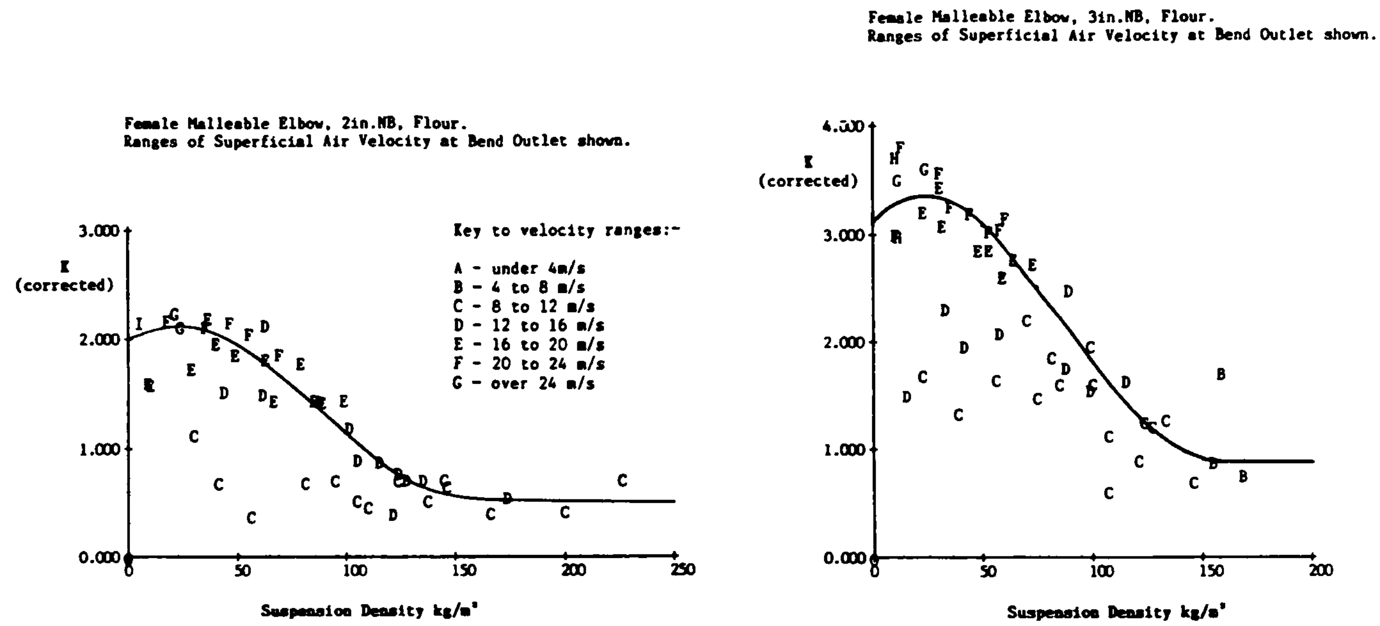


Fig. 3.5

Comparison of loss coefficients for flour in the female malleable elbows of 2 and 3in. n.b.

This would tend to indicate that with different pipe diameters, the ratio of bend radius to pipe diameter is probably more important than the actual radius, as far as pressure loss is concerned. This makes the malleable fittings even less attractive in the larger pipe sizes.

3.5 Recall and use of data - the pipeline synthesis program

Once systems were available for storing the data from test runs, it became necessary to consider methods for using this data. Thought was given to this at an early stage of development of the data storage systems; the process of estimating the pressure drop along a proposed pipeline for chosen values of flow rates of air and product would involve working along the pipeline from the end (atmospheric pressure reference), finding pressure loss in each straight and bend in turn, recalculating air velocity and suspension density each time. A worked example is given in Appendix M. It would be quite possible to do this using a pocket calculator, but it would be very tedious and time consuming to build up a comparison of different possible pipelines for a given duty. Therefore it was decided early on that it would be necessary to use a computer for this, and this was the motive to try to avoid the use of graphs in the data storage systems.

The procedure for predicting pressure loss along a pipeline is as follows:

a) The layout and bore of the pipeline must be known. The mass flow rates of product and air to be used in the prediction are chosen.

b) Calculation normally begins at the outlet end of the pipeline where pressure is atmospheric (for a vacuum system pressure would be atmospheric at the inlet end, so calculation would begin here).

c) The conveying conditions (air velocity and suspension density) at this point are calculated; the actual volume flow rate of air is calculated from the mass flow rate using $pV=mRT$*, then air velocity is found by dividing volume flow rate by pipe cross-sectional area. Suspension density at the same point is found by dividing mass flow rate of product by actual volume flow rate of air.

   * - V = volume of gas mass m at absolute temperature T and absolute pressure p, R being the gas constant for air (the conveying gas).

d) Knowing the conveying conditions at this point, the pressure gradient in the straight section before this point is found by using the correlations for straight pipe pressure drop; the 'Air Only' pressure gradient is calculated using the Darcy equation, and the 'Solids contribution' is found from the correlations given in section 3.3.2. These are added together to obtain the total pressure gradient, then multiplied by the length of the straight to find the total pressure loss along the straight section.

e) Now knowing the pressure at the inlet to the straight section, the conveying conditions at this point are calculated using the same procedure as in (c) above, but with the current pressure rather than atmospheric. The inlet to the straight is of course the outlet of the preceding bend, so these new conveying conditions are used with the correlation for bend pressure loss (section 3.3.1 above) to establish the loss caused by this bend.

f) Knowing the pressure at inlet to the bend, the conveying conditions are again recalculated and the pressure loss in the next straight section back is found using these conveying conditions ((d) above).

g) This procedure is repeated again and again, working back along the pipeline taking each bend and straight length in turn, treating each one by taking the conveying conditions at its outlet and using these to establish the pressure loss, then adding this pressure to the running total and calculating the conveying conditions at its inlet, until the inlet end of the pipe is reached. Thus the air pressure and velocity at the inlet to the pipeline is found.

A glance at the worked example (Appendix N) will show that the process involved in performing the calculations requires not only patience, but also some degree of accuracy because of the relatively small increments in the pressure at each point; also any mistake in calculation at any point will be consequential since it will upset the following calculations. Most real pipelines have rather more bends and straights than the example used, so the problem is usually more acute than the worked example demonstrates.

Also, obtaining the greatest benefit from this method of prediction of pipeline pressure drop requires the procedure to be repeated over and over again with different values of air flow rate and pipeline bore, firstly to find the most economic air flow rate for each pipeline bore and then to compare possible bore sizes.

These facts, taken together, make the use of the personal computer a very great asset with this method of pipeline design.

3.5.1 Structure of the computer program

The program to do this repetitive procedure was extremely simple; the flow diagram is shown below whilst a listing is given in Appendix G.

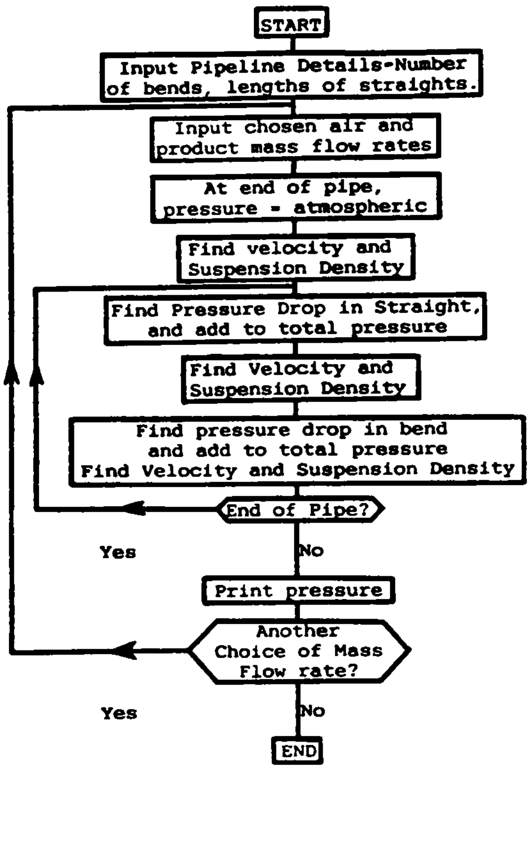


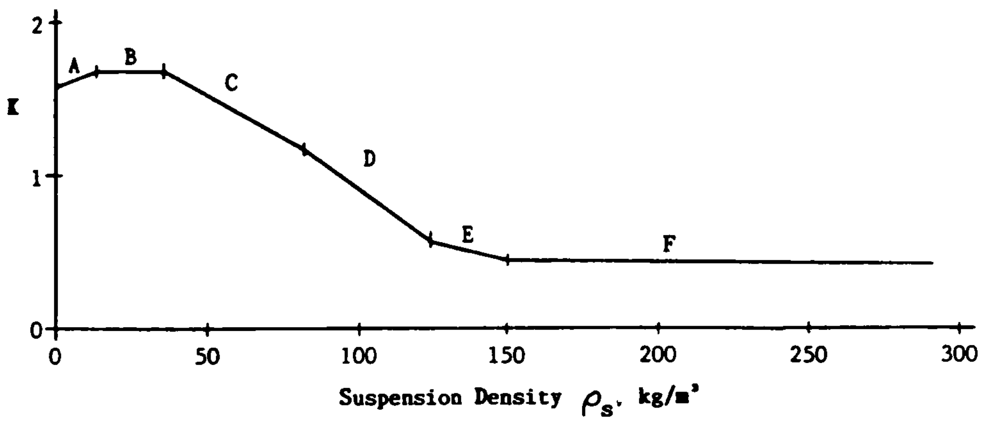
Fig. 3.6

Structure of computer program to synthesise pipeline pressure drop

Using this program, the use of different pipeline bores and flow rates of air for a given flow rate of product can easily be examined so that the user can quickly compare the benefits of different systems; also, the effect of altering pipeline layout, using a different type of bend, or even increasing the bore size of the line along its length (to keep air velocities down) can be evaluated easily.

3.5.2 Input of graphical data

Although (as mentioned in section 3.3) it was possible to store the data on losses in straight pipes using only simple equations, for the bends it was necessary to use a graph to represent the loss coefficients against suspension density (for the flour) or air velocity (for the polyethylene pellets); this information was programmed into the computer using a piecewise linear approximation, i.e. a series of straight lines with gradients and intercepts calculated to allow them to represent the curves between certain limits. The approximation for the flour in the short radius bend is shown in fig. 3.7 below; for the other radiused bends it was only necessary to scale this with a factor, since they displayed the same shape of curve with just a different height.



Section of line	Slope A_1	Intercept A_0	From kg/m³	To kg/m³
A	0.0080	1.57	0	14
B	0	1.69	14	37
C	-0.0112	2.10	37	83
D	-0.0145	2.37	83	125
E	-0.0045	1.11	125	152
F	0	0.45	152 up	

$$K = A_0 + A_1 \rho_s$$

Fig. 3.7
The piecewise linear approximation to the flour data

3.5.3 Non-horizontal pipe sections

No work was done during the project to examine the effect of vertical sections; therefore the program was not adapted to account for these. It was considered that for vertical-up sections it might be possible to use the relationship demonstrated by Mills (ref. 1, from the work of Marjanovic), that these display a pressure drop twice that of horizontal sections, and treat vertical sections as horizontal ones of double the length; alternatively the static head of suspension in a vertical pipe might be added to the pressure drop calculated for a horizontal of the same length, but some work would be necessary to verify these approaches and if they were not found to be satisfactory then an instrumented vertical section would be needed in a test rig to obtain data for design of pipelines with significant vertical lengths. The 2:1 ratio arising from the work of Mills and Marjanovic seems remarkable, but that work did cover a very wide range of conveying conditions as well as product types and the relationship was very consistent.

For vertical-down sections, treating them as horizontals of the same length would result in over-prediction of pressure drop and hence conservative design; fortunately long vertical-down sections are hardly ever used in real pipelines (with some notable exceptions, e.g. ref. 56); if designing a system with a long vertical-down section it would be necessary to have a test section with this orientation.

As far as inclined sections are concerned it would again be necessary to use a test section at the correct angle to obtain data for accurate design, subject to test work demonstrating whether losses in such sections could be predicted from measurements made in horizontal sections.

3.5.4 Hazards of using the program

It must be pointed out that no safeguards were written into this program to check whether the calculated flow conditions are within the range of air velocity and suspension density used in the conveying trials; equations used to represent the data are continuous to infinity beyond the ranges of these quantities for which they were developed. It was left up to the operator to satisfy himself about this. The fact that the flow conditions are recalculated at every bend means that this can be done quite easily by printing out this information, but if the program were to be used generally some safeguard would be considered necessary.

Also, since the program uses the flow conditions at the end of each straight to find the pressure loss in that straight, it would be in error if used for a pipeline containing a very long straight pipe (long enough for there to be a significant expansion of the air in the straight section). It would be necessary to subdivide such sections; this would need to be done automatically if the program were to be used in general circulation.

CHAPTER 4

TRIALS OF THE METHOD DEVISED

4.1 Introduction

Once a usable method for the prediction of pressure drop along a conveying line had been developed, it was clearly necessary to test it by predicting losses in some real conveying pipelines and comparing with measurements from them.

Data on total pressure drop in the various pipe loops which contained the test sections had been recorded whilst the test runs were taking place, at the time simply on the basis of 'it might come in useful'. On reflection, it was apparent that to try to predict this 'overall pipeline' data using the method would be a useful test for the method, which had been developed without any reference to this data whatever.

4.2 Synthesis of characteristics

Using the pipeline synthesis program described in section 3.5, four complete sets of conveying characteristics were synthesised; these predicted pressure loss along the entire line for the flour in the 2in, 3in, and 4in. nominal bore pipelines, and for the polyethylene pellets in the 2in. n.b. line, over a wide range of conveying conditions. Only after synthesis of these, were the corresponding true characteristics plotted from the measurements which had been taken during the test runs.

When entering the pipeline geometry into the synthesis program, account was taken of the fact that the first section of each of the lines was of 2in. n.b. whatever the size of the loop, and the position of the expansion was located carefully. The vertical riser at the end of the pipe was treated as an horizontal section of twice its actual length, for the reasons explained in section 3.5.3. The effect of pipe bore was accounted for by scaling flow rate of product proportional to pipe cross sectional area for otherwise similar conveying conditions, which was felt to be fairly accurate for the reasons discussed in section 3.3.2. The actual layouts of the pipelines can be found in Appendix F, fig. F-1. The pipe diameters used were of course the measured internal diameters of the lines, not the nominal bore sizes - i.e. 53mm for 2in., 81mm for 3in. and 104mm for 4in. (medium weight tube).

The bend loss coefficient data was taken from the 2in. short radius bought-out bend for all cases, even though the actual 3in. and 4in. loops employed longer radius bends; this was justified by the minimal effect of bend geometry for bends of r/d ratio greater than that of this bend, as demonstrated in fig. 3.3 of Chapter 3, and the also minimal effect of pipe diameter described in section 3.4.2. For the same reason, the data on 'solids contribution' to the straight pipe pressure gradients were taken from the 2in. line.

The synthesised and measured conveying characteristics for the various lines are compared on the following pages.

4.3 Comparison of synthesised and measured characteristics

The 2in. line conveying flour was simulated first. The diagram showing measured and predicted pressure drop for a wide range of flow rates of solids and air is reproduced below:-

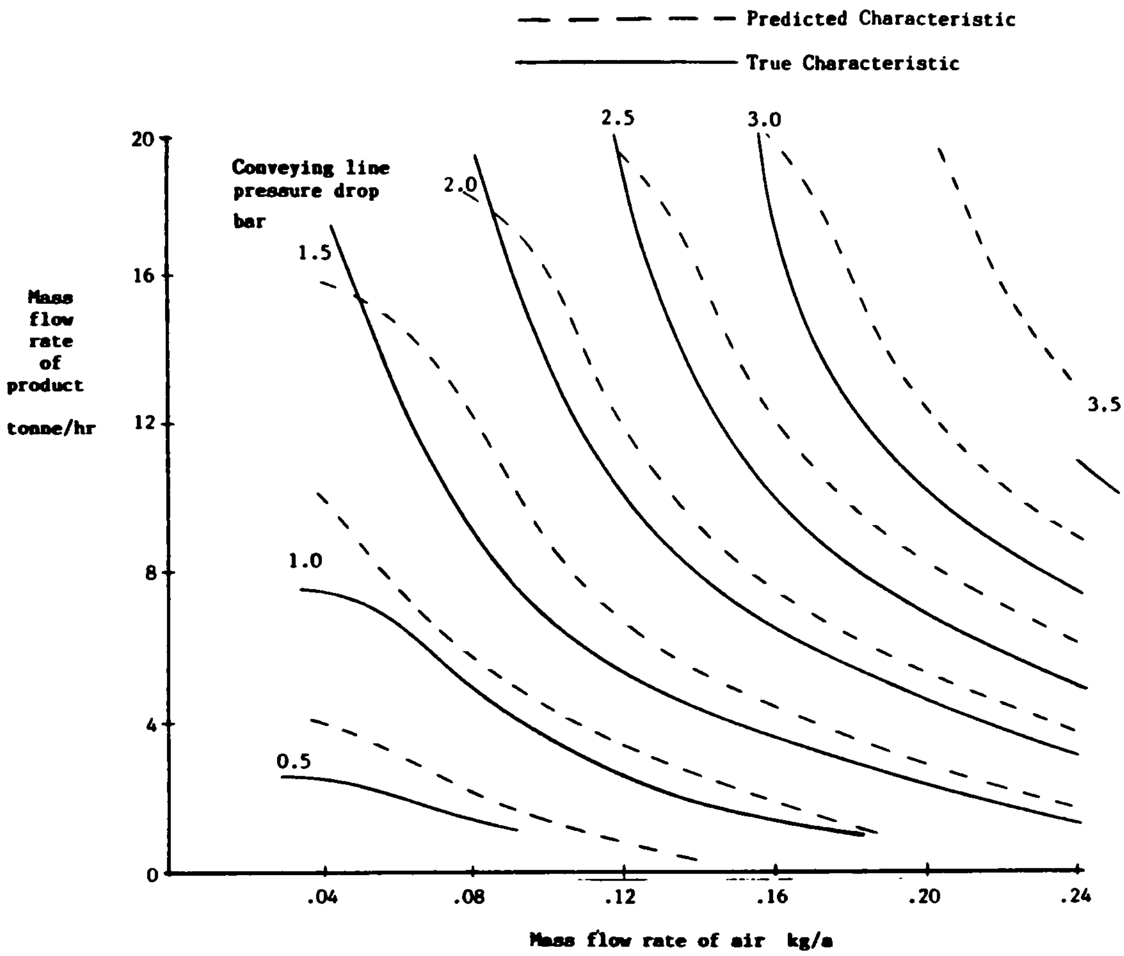


Fig. 4.1
Measured and predicted conveying characteristics
for the 2in. n.b. line conveying flour

At this point it should be observed that the contribution of the air to the pressure drops in these loops was small in comparison with the total pressure drops, typically up to 0.1 bar at the higher air flow rates in the diagram above and much less for the 3in. and 4in. nominal bore lines. This means that these diagrams give a good comparison between actual and predicted solids contribution to the pressure drops in the lines.

From this it is apparent that over most of the range the simulation under-predicts the pressure drop for any given flow rate of product, by a fairly consistent 0.2 bar. This is not terribly serious in the areas where the pressure is 2 bar or more (i.e. "dense phase" conditions), since to leave such a margin for uncertainty in design would not be unreasonable anyway; but where the pressure is quite low, say less than 1 bar, this represents quite a serious underestimate which could result in a system throughput significantly lower than the design objective; e.g. approximately 30% for the 0.5 bar line.

The data fed into the program and the algorithms used were checked for correctness and no errors could be found; therefore the discrepancy was taken as genuine and it was decided to continue with the calculations for the other lines.

The equivalent diagram for the 3in. line is shown below:-

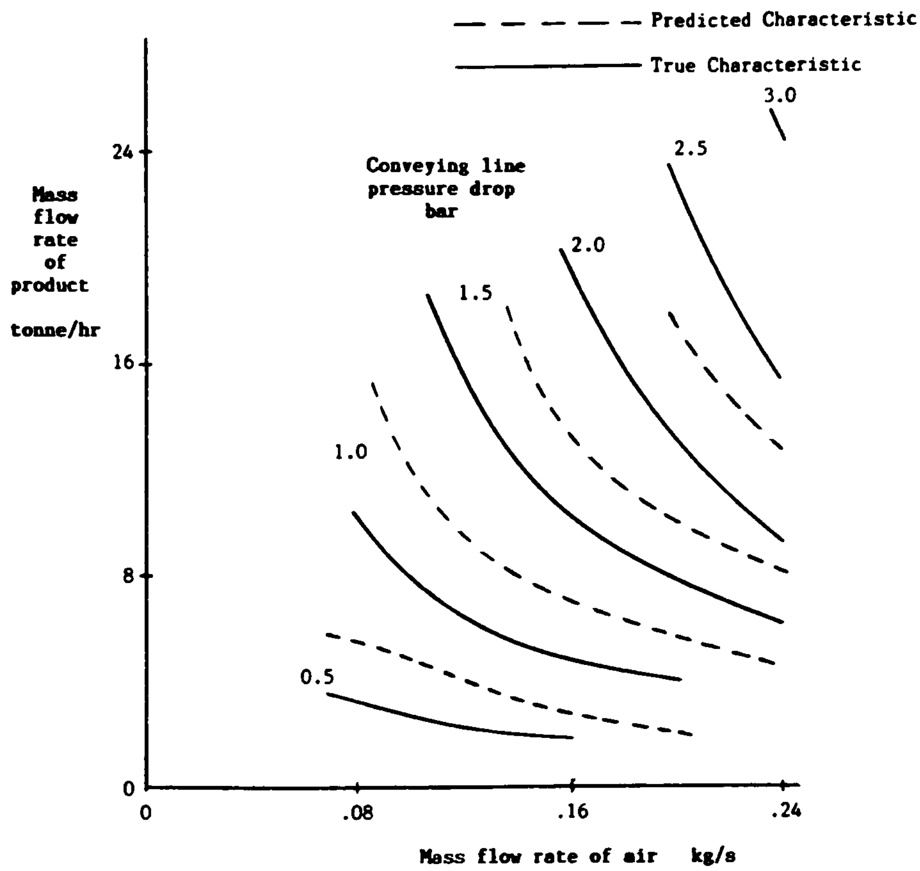


Fig. 4.2
Measured and predicted conveying characteristics for the 3in. n.b. line conveying flour

Note that the range of air flow rates covered was more restricted at the lower end than for the 2in., owing to the larger pipe bore needing approximately 2.3 times as much air to achieve the same velocities. Also the maximum pressure drop achieved was a good deal less for the same reason.

The amount of under-prediction of the pressure drop was very similar to that for the 2in. line, and the same comments apply.

For the 4in. line the range covered was more restricted still, but the magnitude of under-prediction of the pressure drop was again very similar as shown below:-

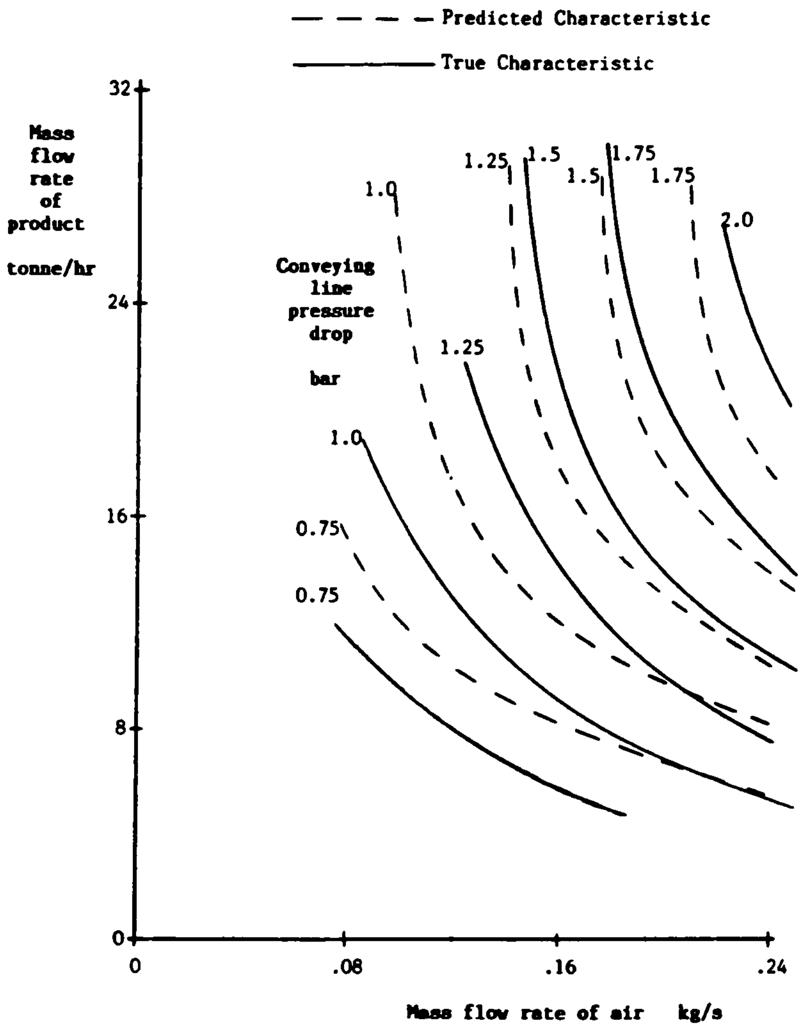


Fig. 4.3
Measured and predicted conveying characteristics for the 4in. n.b. line conveying flour

Once this work had been done for the flour, the same was done for the polyethylene pellets. These were conveyed only in the 2in. line and the result is shown below:-

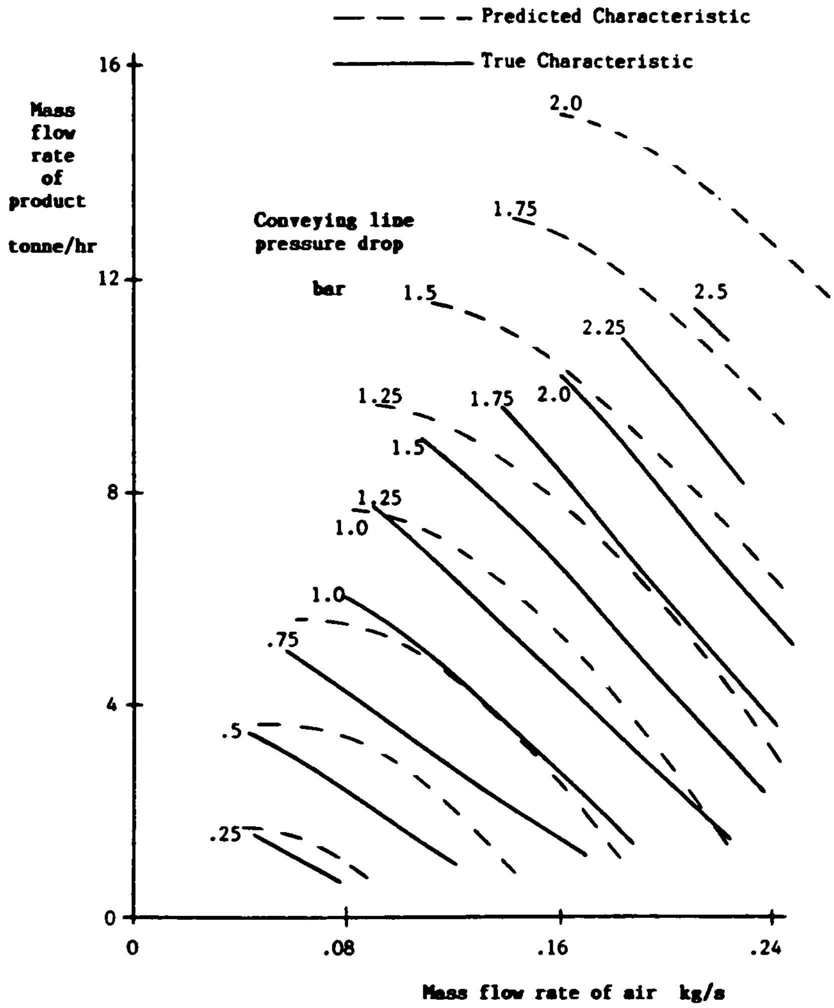


Fig. 4.4
Measured and predicted conveying characteristics
for the 2in. n.b. line conveying polyethylene pellets

Here again the simulation under-predicts the pressure drop over most of the range, this time by a fairly consistent 25% or so rather than a consistent pressure difference. Again this is sufficient to lead to a significant under-sizing of system components unless a fairly generous allowance for uncertainty is made at the design stage.

Comparing the diagram above with Figs. 4.1 to 4.3 for the flour, the direction of curvature of the lines is opposite. This is commonly seen when comparing two very different products, as related in section B.3.1, and although no firm conclusion on this effect can be drawn fron this work, it appears likely that this is caused by the very different variation in bend pressure loss with conveying conditions (as demonstrated in Fig. 3.1) between these two very different products.

4.4 Outcome of the trials

On the face of it, the results obtained from the trials were a little disappointing in that when using this method as it stands, a fairly generous allowance for uncertainty would still need to be made when designing a system. In this respect it seems to offer little improvement over the testing-and-scaling method which went before, at least for systems with a fairly usual distribution of bends.

However, it must be said that a generous allowance for uncertainty is not a very bad thing when designing a pneumatic conveying system, because in spite of testing a sample of the product which it is being designed to convey, there is always a possibility of significant difference between this and the product conveyed in the final system. This is particularly so where (as is often the case) the system will be part of a new plant and the product tested comes from an existing plant, with most likely different size distribution and moisture content. The fact that the resulting system may not have the absolute lowest possible energy consumption is less important than the fact that it works reliably, because even a very short stoppage on a plant producing products of high cost but relatively low added value (as is often the case for foodstuffs) will wipe out many weeks of saving on energy with a more efficient system.

Also it must be borne in mind that there were certain areas used in simulating the loops which have not been investigated properly but just an intelligent guess taken. Firstly the effect of vertical sections, for which Mills' correlation of pressure drop being equal to twice that in horizontal sections was used (see section 3.5.3 for an explanation); it is now known from this work that Mills over-estimated the contribution of horizontal straight lengths to the total pressure loss in a pipeline, which would account for some under-prediction of pressure loss using his correlation in conjunction with the current method. Secondly it has been seen that different bends of ostensibly the same type can display significantly different loss coefficients and that this cannot be simply explained in terms of the means by which they are joined to adjacent pipes, as explained in section 3.4.1. Thirdly no discrimination has been made between bends in different attitudes; the actual loop contained bends between horizontal and vertical sections, which were treated just as though they were between two horizontal sections like the ones from which the data was obtained. Finally, no account was taken of the fact that in the 3in. and 4in. loops, the bend at the top of the riser by the receiving hopper was a short radius elbow or a blind tee respectively, which are known to give higher pressure drops than radiused bends.

A further effect which might be mentioned is that of electrostatic build-up in the conveyed product; it has been the experience of some previous workers at Thames that the conveying characteristics of certain products can change with repeated conveying due to this effect. One product particularly known for this is a pvc powder which is both fine and highly non-conducting; the two products which were used for the work reported here have frequently been conveyed in the Thames laboratories by other workers as well as this author, and no such effect has ever been noticed, presumably because in the case of flour it contains some moisture and so will discharge static electricity whilst in the case of the polyethylene pellets they are large enough that their inertia is sufficient to overcome electrostatic forces.

4.5 Assessment of the trials

Given the factors mentioned above, it seems remarkable that the predictions are as accurate as they are, bearing in mind that the method used has not been 'tuned' in any way as a result of these trials. The fact that the error is fairly consistent would tend to support the fundamental value of the method described, and suggest that with further refinement it may well be capable of much more accurate results. The areas in need of refinement have been mentioned above, i.e. vertical sections, bends in different attitudes and the variation between bends of supposedly the same type.

Two of the lines simulated included expansions in pipe sizes of very significant proportions (factors of 2.3 and 4 on area) yet these were dealt with quite easily and this did not seem to compromise the accuracy of the results. This is important with the increasing trend towards the use of "stepped" pipelines which enable energy savings to be made and longer conveying distances to be achieved, by keeping air velocities down even when there is very significant expansion of the air along the line.

At this stage it was considered that the trials undertaken had served to verify the method as far as could be expected without studying the unknown factors mentioned above. Therefore it was decided to call a halt to further development and to begin to bring together the work for this report.

4.6 A case study

With the results of these tests seeming to confirm the value of the method described, a final stage of work was undertaken consisting of using the method to perform a case study of pipelines proposed for a real application. Using the data on flour, several alternatives of pipe size, bend type and number of steps up in bore size along the length were considered on the basis of fixed throughput (60 tonne/hr), fixed minimum conveying velocity (11.8m/s) and fixed layout (60m long with 15 bends). It was found that the pipeline synthesis program enabled power consumption and maximum air velocities for each alternative to be estimated and compared very quickly, yielding information necessary for choosing the best option. The proposed system has not been built, so no data is available for verifying the accuracy of the predictions. Further details of the case study will be found in Appendix L.

CHAPTER 5

CONCLUSIONS

At the outset of this project, the primary goal was to improve the accuracy of prediction of pressure drop along pneumatic conveying pipelines. It is believed that this goal has been achieved, and that the method which has been developed is not only capable of greater accuracy than that which it supersedes, but is also more convenient for the designer to use when comparing different possible options for pipelines for a given duty. Furthermore it may be used to predict pressure drop along pipelines with successive increases in bore size, which have been shown to be potentially very useful. The limits over which the method can confidently be applied, in terms of flow regimes, air velocities and product rates, are restricted only by the range of conveying conditions which can be achieved in the test plant when undertaking the trials which are a necessary part of the method.

The means by which the goal has been fulfilled is rather different from that which was expected at the inception of the project, in that the methods available then have been superseded rather than improved. This was a result of the limitations inherent in the methods which had been developed up to that time, which is not to say that they were developed wrongly but simply that they had been developed as far as was possible. It was the examination of, and attempts to improve, these methods which showed the need for a fresh start.

It will be appropriate at this point to recapitulate on the thread of the project. Firstly a survey was made of the means then available for making the prediction. These means were assessed, and their weaknesses and strengths identified in relation to improving the accuracy of the predictions made. It quickly became apparent that the practical limitations of the methods available had been reached but that it may be possible to evolve a new method which would overcome these limitations. This method was considered and a decision taken to proceed along this path. The new method would involve using the product to be conveyed, in a special test facility. A suitable facility was developed and a large amount of test work undertaken, both to assess the viability of the method proposed and to assess the effect of a number of variables bearing upon the use of the method in pipeline design. The proposed method appeared viable and gave good results when subjected to trials predicting the pressure loss in some real pipelines. An assessment of the method and the outcome of the trials showed that the new method is a little more demanding of the user at the testing and data analysis stage, but is far more convenient and more powerful in making predictions of pressure loss for pipeline design.

The new method involves:-

(a) conveying the product for which the plant pipeline is to be designed, to obtain data on pressure losses along straight lengths and those caused by bends. A special test facility was developed for this, involving much more instrumentation than was previously used on any known pneumatic conveying test facility.

(b) feeding the data obtained from (a) into a storage and recall system, in which it is compact and easily accessible. It appeared that the system needed is dependent upon the product which is being conveyed, in that the correlations used varied between the products which were tested.

(c) predicting the pressure loss along a proposed plant pipeline by working along the pipeline from one end, where conveying conditions (air velocity and suspension density) are known, finding the pressure drop caused by each bend and straight length in turn by recalling the pressure drop data for the conveying conditions in existence at the point in question, then applying this pressure drop to obtain the pressure at the next point along and recalculating the conveying conditions there. This procedure is applied repeatedly until the user has worked right along the pipeline and thus determined the total pressure drop.

The advantages of this method, over what went before, include:-

(i) the ability to assess, quickly and accurately, the effects of pipeline layout, e.g. positions and number of bends, and bends of different geometry.

(ii) the ability to compare many possible designs for a duty in the space of a few hours, by changing pipe bore, layout, and mass flow rates of air and product, and re-running the computer program which uses the data to make the predictions.

(iii) the ability to deal with stepped pipelines. Previous methods involving the use of data could not be used for the prediction of pressure losses along such pipelines, which may be of great importance as described below. Using the new method, not only is it possible to predict the pressure drop along a stepped pipeline, but also the best positions of the steps, for economy of operation and maintenance, may be located.

It is apparent that the method relies upon the availability of an experimental facility for generation of the data which is central to its operation. Currently there appears to be no easy way around this; all the products tested to date have given data differing not only in value but also in basic form (i.e. the way in which the losses vary with conveying conditions). Further work may well be undertaken to try to understand what properties of products, measurable on a small scale, determine the behaviour of the product in a pipeline, and no doubt this will shed some valuable light on possible ways of classifying products in terms of conveying behaviour; however it is not the belief of this author that this will overcome the need for loss data for the specific product which a plant pipeline is to be designed to convey. It appears more likely that the need for experimental work will gradually decline as a bank of data is built up over a period of years relating to common products. For example, products such as, say, ordinary portland cement, or perhaps wheat grain, do not vary very much from one source to another and in any case any sensible designer would hope to accommodate such natural variations in his plant design anyway; so for these products, once the data has been obtained by one set of tests, and perhaps confirmed by another one or two sets with product from different sources, it would most likely be adequate to rely on this for future designs. For products which have not been met before, however, the need for such experimental work will probably always remain unless some risk is taken in using data from another product.

Major avenues of research within the development of this method have been:-

(a) the effect of product type; the data for the two products tested displays not only different values of pressure drop between the products at similar conveying conditions, but also a different type of relationship between pressure drop and conveying conditions. There is no sign that it will ever be possible to dispense with the need for tests using the actual product for which the design is to be undertaken.

(b) the effect of bend geometry; it would appear that the general use of bends of long radius for conveying pipelines represents an unnecessary expense, bends of much shorter radius appearing to perform just as well; it has, however, been shown that the use of too short a radius leads to excessive pressure drop.

(c) the effect of pipe bore; it would appear that this has little effect on the pressure drop for given conveying conditions, i.e. air velocity and suspension density (implying the same pressure drop for the same flow per unit cross sectional area of pipe).

(d) the effect of air density; this has been shown to be minimal, provided the pressure drop data is stored and used on the basis of air velocity and suspension density and not mass solids loading ratio.

(e) the use of the traditional quantity Solids Loading Ratio (on a mass basis, often wrongly referred to as "phase density"); this has been shown to be a misleading quantity since it cannot take account of the change in conveying conditions which occur along a pipeline as the air expands with fall in pressure. If this is used then pressure must be taken into account, not only because conveying conditions will change according to pressure but also because air density will then have a significant effect upon the pressure drop.

(f) the use of "Equivalent Length" values is a poor way to account for the pressure drop caused by pipeline bends. The relationships between pressure drop and conveying conditions for a bend are totally different from those for a length of straight pipe, so that for one product in one bend, the equivalent length varied from 3m to 60m over a range of conditions. It is the very fact that these relationships are so disparate, which limits the accuracy achievable with the previous testing-and-scaling method.

(g) stepped pipelines; it has been shown, by using the method of prediction which has been developed, that these can give very much lower pressure drop (and even further reduced energy consumption) than lines having a constant bore size from end to end. Also, the effect of keeping the maximum air velocities lower, which gives rise to this advantage, is also expected to have very significant implications in terms of reducing wear and product attrition, both of which are often major problems to the operators of plants with pneumatic conveyors.

Inevitably, arising from a project of this type there are a number of major questions. These are:-

(a) The prediction of pressure drop in vertical pipes, both risers and downcomers; the work which was done previously has been shown to be slightly suspect since this related pressure drop in these to the pressure drop in horizontal pipe obtained from measurements made on total pipeline loops, and it is now certain that the contribution made by the bends in those loops was seriously underestimated. Currently it is recommended that the previous work is used, as described in Chapter 3, in the absence of anything more satisfactory, and that due allowance is made in design.

(b) The effect of inclined/declined pipes; these are rare in practice although occasionally their use is a necessity. Work recently done by this author on pipes declined at an angle of 1 in 4 (Pneumatech 4, UK, 1990), has yielded reduced pressure drop, by more than would be expected from a simple consideration of the gain in static head with loss of height, and shown that this is affected by particle size. Whether the converse is true of inclined pipes is a matter of conjecture, but it is recommended that if significant lengths of incline or decline are to be included in a plant pipeline, then tests should be made using the product in a pipeline of the correct inclination or declination.

(c) The effect of product type; a major test programme is needed to obtain data for other products, so that it can be seen whether there is any pattern in the type of correlations which need to be used to store the data. It is possible that some relationship between these and product bulk properties measurable on a small (e.g. bench-top) scale may emerge. Since the bulk of this thesis was written, two more products have been dealt with using this method, and these have needed different correlations again to store the data, so it would seem that a good deal of work would be necessary in this direction to gain any understanding. In this respect, an extension of the work described in Appendix K using surfaces in three dimensions to represent the loss data may help in the recognition of trends and similarities which might be invisible from normal two-dimensional graphs. It opens up the possibilities of attempting to relate the change in shape of the surface to differences in measurable bulk properties. For example, in the case of the flour the suspension density was the quantity which controlled the loss coefficient, whilst air velocity had no effect; in the case of the pellets the reverse was true. It may be that this could be related to (for example) the obvious difference between a powdered product and a granular product such as the wheat flour and polyethylene pellets used in this project.

(d) The effect of bend attitude; all work done during this project was using horizontal-horizontal bends, and there remains the question of whether bends between horizontal and vertical pipes would exhibit the same pressure drop. There are four possibilities, i.e horizontal to vertical up, horizontal to vertical down, and vice-versa. If the use of inclined pipes is included, then even more possibilities arise, but these are fairly rare in practice.

(e) The effect of bend radius on wear and product attrition; although it has been shown that (at least for the product tested) the radius of a bend has little effect on pressure drop provided that a certain minimum ratio of radius to bore is exceeded, it is not known by this author whether the radius affects the rate of wear or amount of product attrition. This point would also need a major programme of work to deal with conclusively.

(f) The stability of operation of blow tank feeders; it has been found that blow tank feeders appear to operate on a limit cycle, not reaching a true steady state during conveying but suffering from a continual cycling of feed rate between limits. This was discussed at some length in Appendix F, and it is apparent that there is the potential for significant problems to arise if feeding long conveying lines using a blow tank. Some analysis of this problem using control engineering methods may yield an understanding.

Comparison of the data from this project with work of others has been difficult; the only measurements which appear to have been made in any comparable way relate only to very lean phase conveying conditions, i.e. suspension flow of low suspension densities, and with only granular products, for straight pipes only, so there is little overlap onto the largest part of the data taken in this project. However, the areas where there is overlap show fairly good agreement, as outlined in Appendix N. As far as comparing the predictions made using the method against actual pipeline pressure drop are concerned, the trials outlined in Chapter 4 indicate a good agreement.

In closing, it is appropriate to discuss the methods which were used for solving the problem of how to store the data, for the data storage system is the central part of any method for making predictions from an accumulation of data. Firstly, the need for computer literacy is clear; the very size of the database established was essential to the number of questions which were addressed and the progress which was made. To handle and process this data, which in its raw form was equivalent to some seven thousand pages, would have been impossible by hand. Also the speed and ease with which the data could be recalled, re-arranged and re-presented on different graphs made it possible to try different approaches and ideas very quickly and easily.

In analysing the data, many strategies were used, although they may be divided into principally four kinds, namely the fitting of curves to obtain empirical expressions, the mathematical analysis of physical models, dimensional analysis, and mental modelling. The purpose of course was always the same, to see some order in the data which at first appears chaotic; to "introduce order and coherence to facilitate the retention in an available form" of the data, in the words of Rayleigh quoted at the start of this volume. The methods have been used indiscriminately, in that they have all been applied and re-applied many times at different stages, in parallel and in serial, gradually leading to the perception of patterns which could be exploited to impose order. It may be argued that in some cases this has been done rather forcibly, for example the application of the curve for pressure loss in a 2in. bend to the graph of loss data for a 4in. bend in fig. 3.4; however, where this has been done it has been considered very carefully and the criterion which has been applied is whether it enables progress to be made. Different interpretations are always possible, especially where the data is subject to some significant scatter as is always apparent with gas-solid flow, and of course these different interpretations may lead to different conclusions.

In the end, only experience of using the method which has been devised, as an engineering tool, will show whether an acceptable compromise has been reached. In this respect, it is extremely rewarding that this author is now using this approach regularly for the design of pneumatic conveying pipelines for industrial applications; it appears to be extremely powerful and convenient in use, and experience will show how well it works. It would appear, from the work which has been carried out to date, that some inaccuracy in the prediction of pressure loss is of little consequence in that this will not lead to a change in the choice of system components for the duty.

Finally a comment may be made, that it is sincerely hoped that the work which has been described in this volume has been presented in such a form that the reader will be able to extract as much useful information as is contained herein. If so, the greatest benefit will be gained from the work, and the rediscovery of this in the library need not be a more difficult and uncertain process than was the first discovery in the laboratory.

APPENDIX A

EXAMINATION OF THE PIECEWISE MATHEMATICAL APPROACH

A.1 Introduction

It appeared that over the years there had been many attempts at producing mathematical methods for predicting the pressure loss in a gas-solid flow along a pipeline, as evidenced by the results of the literature survey in Appendix M. Most of the papers published dealt only with small parts of the problem, for example the loss along a straight pipe under certain conditions or the loss caused by the acceleration of particles; this is quite understandable in view of the natural supposition that the pressure losses caused by each of these parts result from different processes so should need distinct models. Only one example of a complete approach to prediction of the loss along an entire pipeline was found, and since this is one of the most sophisticated methods as well as embodying many of the principles employed by other authors, it will be described here.

It should be emphasised that all of the mathematical techniques found dealt only with lean phase flow, i.e. the flow regime wherein the particles travel along suspended in the air flow, rather than sliding along the bottom of the pipe or moving in waves or plugs.

It should be pointed out that the current trend in dealing with complex problems in many areas of fluid mechanics, is to employ a method known as Computational Fluid Mechanics, commonly known as CFM. In fact this is simply the application of computer numerical techniques to the solution of the analytical models predicting the behaviour of various parts of a flow, and relies on the same basic models although the means of solution allows many more factors to be taken into account. This technique deserves some mention in its own right.

A.2 The method developed by Mwabe

A.2.1 Historical

P. Mwabe was an undergraduate student at Thames Polytechnic from 1979 to 1983, and for a final year project he undertook the development of a system of analytical models for predicting pressure loss in lean phase flow in pneumatic conveyors. The outcome was a piece of work superior even to many PhD theses, which formed the basis of a commercial software package known variously as PNUECAD or PNEUCON.

A.2.2 Description

Mwabe's method begins by dividing the pipeline up into the straight lengths between bends, further subdividing straight sections if over one metre in length. These step lengths are known as 'fields'. The prediction of pressure loss in each field was achieved through applying balances for mass, momentum and energy. Mwabe's first stage in developing the method was to use it to predict the pressure loss with air alone in the pipeline, by applying accepted models to each field in turn, and once it had been shown to give results close to measured data for this, he then introduced models of the solid particles as sources and sinks of momentum to the fields. In that it uses numerical methods for the solution of each field, it could be said to fall into the category of computational fluid mechanics although most exponents of CFM would regard it as rather crude - the amount of computing power required to run it is minimal, i.e. a BBC model B micro.

A.2.2.1 Mathematical models used

The balance for mass was of course easily achieved, simply taking mass flow rate of solids and air into and out of each section to be equal.

The initial acceleration of particles from rest he took to occur in the first field, and appears to have taken the increase in momentum flux of the air/solids mixture through the field as a momentum source term, equal to the momentum flux of the solids at exit from the field, thus determining a pressure drop using force = rate of change of momentum. He seems to have taken the associated energy source term as being equal to a proportion of the energy flux of the solids at exit from the field, the proportion being found from the test results of other authors.

The losses in straight pipe after the initial acceleration he found by taking the pressure drop caused by air only in a field (from Darcy) and thus finding the increase in velocity caused by expansion of the air, obtaining a momentum source term from this effect; and also considering the drag effect of particles colliding with the pipe wall, using the work of Muschelknautz (ref. 51) to account for this factor, again obtaining a momentum source term based on the number of particles in the field.

He treated vertical sections by taking the drag force on particles as equal to their weight, finding the number of particles in a field as a function of particle terminal velocity in free fall and the air velocity, thus obtaining a momentum source term. This was dependent on terminal velocity of the particles in free fall, which was worked out from particle size and density information making certain decisions about drag coefficient.

The pressure drop caused by bends he treated by constructing a numerical model of the deceleration of a particle sliding around the inside of a curved wall under the influence of centripetal force, and found that the proportion of velocity lost was dependent only on the coefficient of dry sliding friction and independent of particle size or bend radius. He considered a range of coefficients of friction, which appeared to have been obtained from a general engineering data book for such materials as steel, teflon, hemp and babbit metal; he took the extremes to apply to what he termed 'abrasive' and 'elastic' materials and found the percentage of velocity lost for each (10% for an 'elastic' material, 50% for 'abrasive' materials), with an 'average' material taken to be half way between. The momentum source term for reaccelerating particles after losing the appropriate proportion of their velocity in a bend was found in the same way as for the initial acceleration from rest.

A.2.2.2 The program to use the mathematical models

The solution of each field required iteration because the outlet air velocity of each was dependent on the pressure drop, which in turn was dependent on the outlet air velocity; therefore the calculations for each field needed to be repeated a number of times until the values converged. Convergence was considered to have occurred when the change of pressure drop between iterations for one field fell to less than 50 N/m^2, on the basis that it was thought this would give the pressure drop within (100 x 50 n/m^2) = .05 bar on a typical conveying line of 100m divided into 100 fields.

The initial conditions of the first field were set, either at the end or the beginning of the pipeline (the choice of the user) and calculation proceeded along the pipeline to the other end, taking each field in turn and iterating until convergence occurred, taking the conditions at the opposite end of the field as those for the starting end of the next field.

A.2.3 Evaluation

Mwabe finished off by trying to assess the accuracy of pressure loss predictions made by his program, by comparing measured conveying characteristics for a number of products which had been conveyed by other workers at Thames against predictions made with the program.

The models and program which Mwabe developed were based on a fully suspended (i.e. "lean phase") flow of particles and air. Observations of flows in sight glasses showed that this was obtained with most products provided that air velocities were greater than 15m/s. Lower air velocities would not sustain flow with many products, attempts at achieving them leading to pipeline blockage, but products which would flow at lower velocities normally displayed quite different flow regimes when doing so; generally with a bed of material sliding along the bottom of the pipe, sometimes with waves or dunes superimposed which occasionally filled the pipe as they passed, giving the impression of "slugs" crawling along the pipe on top of stationary material.

For this reason it was supposed that a model developed for suspended flow would not give reliable results for these other flow regimes, and so it proved; a few attempts at using the program to predict pressure drop in low velocity flow cases showed there to be little if any correlation between predicted and measured pressure drop. This seemed hardly surprising.

For lean phase flow conditions, however, the situation was somewhat different; provided the correct 'category' was chosen when entering the product details (i.e. abrasive, average or elastic, which controlled the energy loss at the bends), the pressure drop predicted by the program generally fell within about 30% of the measured value; even within 20% over a large part of the conveying characteristics. The difficulty, of course, lay in knowing which category to choose unless the product is tested first and pressure drop measurements made - making the use of the program irrelevant.

Therefore it was concluded that although this approach had a certain range of application, in reality its usefulness was strictly limited.

A.3 Computational fluid mechanics

A.3.1 The basic technique

The essential technique behind CFM is to divide a fluid flow between boundaries at which conditions are known (e.g. pipe walls, planes across the pipe inlet and outlet) into many small fields (more commonly known as "cells"), the behaviour of each of which is modelled analytically using the same basic models of conservation of mass, momentum and energy, then to solve all the equations for all of the cells simultaneously using numerical methods. The essential difference between this and the simple technique used by Mwabe is that very many more fields (cells) are used, generally dividing the pipe cross section up into many parts as well as dividing it along its length. The availability of very powerful mainframe computers has made this possible, so that systems consisting of many hundreds or even thousands of cells may be solved in minutes, or at worst with an overnight run. It appears to have been a spin-off from the development of packages for solving finite element stress analyses numerically.

A.3.2 Difficulties

The central difficulties with CFM are the same as with the basic mathematical techniques. Firstly it is necessary to make decisions about the physical models to use for interactions within each cell - these physical models control the whole analysis. Secondly it is invariably necessary to have values for quantities which cannot be measured or obtained easily from measurable quantities, the usual example being again a value for a coefficient of friction to use when considering collisions between particles and pipe wall.

As far as the physical models to use within the cells are concerned, the more effects that are taken account of, the more time is required to solve the system - it would seem natural to expect that the time would rise approximately exponentially with the number of processes considered. It is also obvious that if too simple a model is used then significant effects may be overlooked.

It seems to be thought amongst the advocates of CFM that if the cells may be made smaller and smaller then the processes taking place in each may come down to a level where they may be represented reliably by physical models simple enough to use, without the need to use non-measurable quantities; that is, a level of what might be called 'fundamental' models, not depending on the results of observations. This would obviate the need to deal with complicated large scale processes, since the supposedly simpler smaller scale processes would predict the larger scale ones. In practice there is a difficulty (apart from the need for ever more powerful computers) in that such fundamental models are not available - all the 'laws of mechanics' rely on observations. Thus as the larger scale processes begin to look after themselves, then the smaller scale processes in turn require the results of observations, which in themselves become more difficult to make and less reliable, and the solution simply becomes more and more involved without achieving greater accuracy.

The philosophical question of whether such things as fundamental laws exist at all arises; even if it was accepted that they do, then it seems clear that they would be on such a small scale that to apply them to a real system of any size would require computing power impossible to achieve. Therefore it seems that the only way forward with CFM is to determine the values of the non-measurable quantities required for calculation by finding values for these which make the predicted pressure drop agree with measurements taken from real pipelines, but as with the more straightforward analytical modelling techniques, it would clearly be necessary to do these measurements for each set of flow conditions for any product under consideration, in which case the use of the CFM technique is unnecessary because it simply becomes a way of storing data which could be more economically stored in a much simpler fashion, e.g. by empirical expressions.

A.3.3 The work of Mason

One piece of work which applied CFM techniques to the prediction of the pressure loss along pneumatic conveying pipelines was that of D.J. Mason (refs. 10 and 60), who attempted to use a commercial CFM package known as PHOENICS for modelling lean and dense phase (i.e. suspension and non-suspension flow) conveying. With some effort he found it was possible to obtain model flow patterns similar to those observed in real pipes, (e.g. 'slugging' or intermittently full-bore flow) but no real progress was ever made towards the prediction of pressure drop with any accuracy.

A.4 The work of other authors

It has already been observed that a great many papers had been published on the subject of mathematical modelling of lean phase flow. Most of the papers were extremely forbidding in appearance, tending to contain many lines of equations with generally little explanation of the development of the method or how it may be used, which led this author quickly to abandon hope of finding anything of real value in them.

A practically universal difficulty lay in the need to know quantities which could not be measured or derived from measured variables - most often the slip velocity between particles and air, and/or a coefficient of friction to use in a model of collisions between particles and pipe wall. Invariably these factors were of primary importance to the results obtained, meaning that in practice the values of these could only be determined by making some pressure drop measurements and working back through some very heavy mathematics to find them. Even having done this, there would be no indication as to whether the value was solely dependent on the product, or would vary in some way with flow conditions; the suspicion would of course be that the latter would be the case, so a comprehensive set of tests would need to be done for the product to determine the variation of the factors with flow conditions, leading to the model becoming nothing more than an overly complicated way of storing measured data - not that there is anything wrong in using a data storage system, but such a system would be better conceived as such from the outset.

It should perhaps be observed at this point that some considerable progress has been made in recent years in applying CFM methods to the flow of single-phase fluids. With the use of enormous computing power, specifically the new and phenomenally powerful Cray mainframe machines, it has proved possible to solve the Navier-Stokes equations for a finite-element analysis of the flow of fluids over some quite complex bodies. Two examples known of are applications to water turbines and flow over aircraft (work being carried out in France and America respectively). The Navier-Stokes equations are three-dimensional partial differential equations of motion for a small element of fluid, involving the shear stresses acting on that element as a result of fluid viscosity, and as such have their roots in laminar flow; nevertheless, it is understood that by 'tuning' the analysis in some unspecified way to give results in agreement with experimental data for some aircraft which have been wind-tunnel tested, the American workers in this field have been able to achieve sufficiently good predictions to enable them to apply the method usefully to the improvement of existing aircraft designs. This application of such an analysis has the benefit of using a fixed solid boundary which is not affected by the flow of the fluid over it; it seems natural to think that to apply a similar method to a fluid-particle system, where the positions and orientations of the solid boundaries are continually changing, affected by and affecting the flow of the fluid, would be at least an order of magnitude more difficult and probably another order more difficult again with most real products where the particles are of non-uniform size and practically indescribable shape.

A.5 Conclusions

The detailed analysis of Mwabe's work, and the examination of the vast volume of literature pertaining to the development of analytical or numerical models for prediction of pressure drop, had shown that the difficulties in developing such models were to all practical intents insurmountable, so this avenue of approach was most unlikely to be rewarding.

However, the great advantage of the mathematical approach to estimation of pressure drop clearly lay in the fact that the effect of individual pipeline features on the performance of the overall pipeline could easily be examined in detail, simply by running the calculations again with (for example) the bends in different positions, or a step change in pipe size at some position along the line; thus with the aid of a computer it would be a simple, fairly quick task to assess the effect of changes of this type on the performance of any proposed pipeline. By contrast, this type of analysis was very difficult to perform using the alternative technique of testing and scaling. It was this observation which led to the decision that any better technique for prediction of pressure loss must deal with straight lengths and bends individually, rather than simply trying to deal with an overall pipeline as with the testing and scaling technique.

APPENDIX B

EXAMINATION OF THE TESTING AND SCALING APPROACH

B.1 Introduction

The method of predicting the pressure drop along a conveying pipeline by taking measurements from another pipeline of different layout but conveying the same product, and scaling these results to account for the differences in layout, was mentioned in section 2.2, where the major limitation of the method was pointed out. This Appendix examines the method in a little more detail.

B.2 History

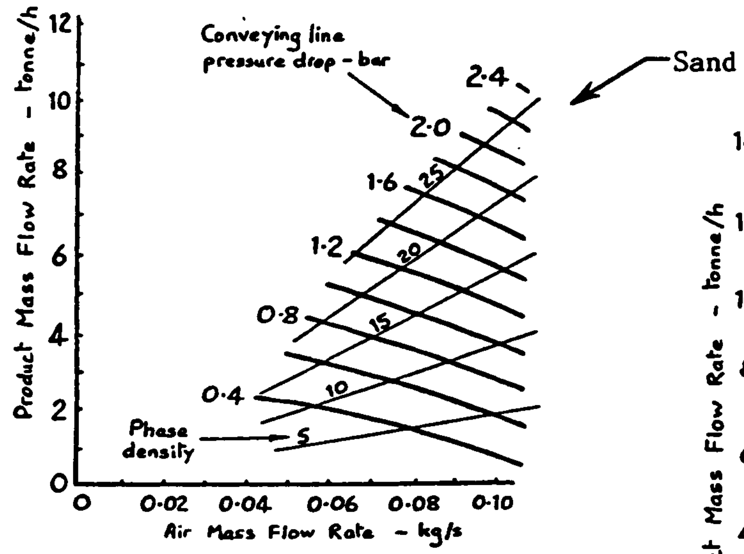
It seems as though much of the development of the particular version of the technique which will be described was carried out by D. Mills working at Thames Polytechnic although it is not known whether he originated the idea; it has been used for a number of years at Thames for design of systems on a consulting basis, with a degree of success. A similar approach has also been used at the University of Wollongong in Australia with some success, and it is believed, from private discussions with engineers working in the industry, that most of the better respected suppliers of pneumatic conveying equipment employ a similar approach to the design of pipelines.

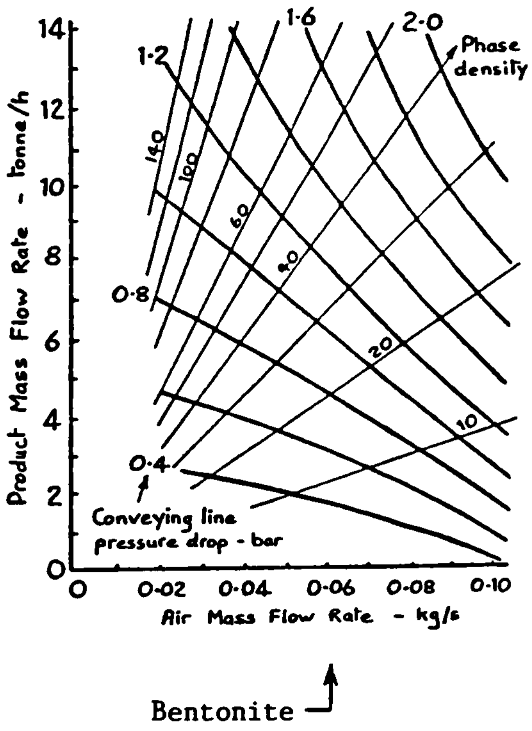
B.3 The method

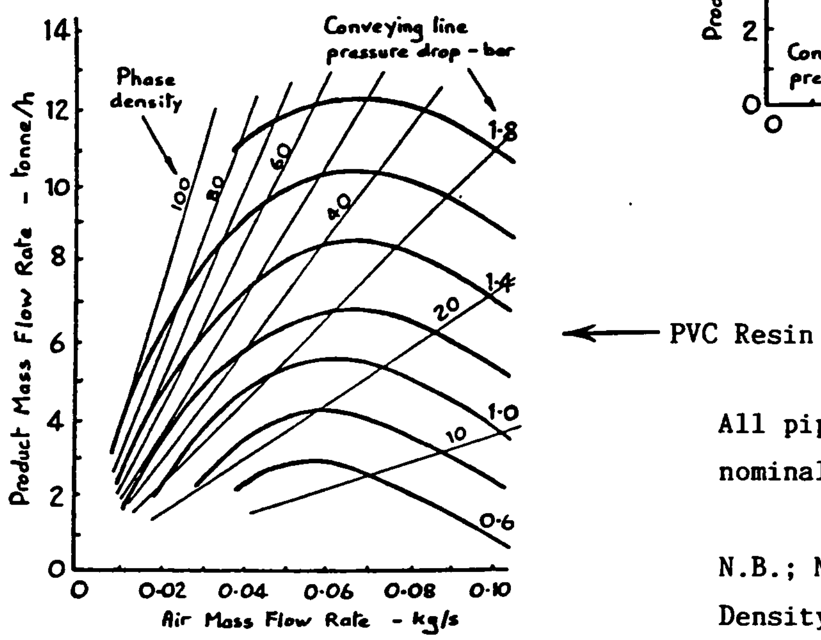
The Mills version of the method is described most clearly in ref. 1, so a good deal of what follows is essentially an analysis of what appears there. Most other versions appear to be broadly similar.

B.3.1 Justification

The essential reason for using the testing and scaling approach is that it uses actual measured data for the product which the final system is to convey, thus overcoming the difficulty caused by the unpredictable differences in conveying behaviour between different products. For example, different products exhibit different minimum conveying air velocities, and different pressure drop values for the same conveying rates; moreover, the way in which pressure drop varies with conveying rates and air velocities is different between different products. Examples of three products with markedly different conveying characteristics are shown in the diagrams below, which indicate the pipeline pressure drop for a range of flow rates of product and air.







All pipelines 60 or 70m, 2in. nominal bore (53mm bore).

N.B.; Mills uses the term 'Phase Density' for Solids Loading Ratio.

Fig. B-1

An example of the marked difference between the conveying characteristics of three different products, from ref. 1

The prediction of the differences between the conveying characteristics of different products is an area which has been the subject of a considerable amount of work, but has proved elusive. Jones (ref. 52), as a result of four years work in this area, managed to devise some bench scale tests whose results showed correlation with the minimum conveying air velocity of products, but no progress has been made towards predicting the pressure loss in a pipeline.

Thus it can be seen that the use of data measured from the product to be conveyed is essential for accurate system design; before the development of the method described in this thesis, the only way of doing this was to use the testing and scaling approach.

B.3.2 The procedure, as employed at Thames

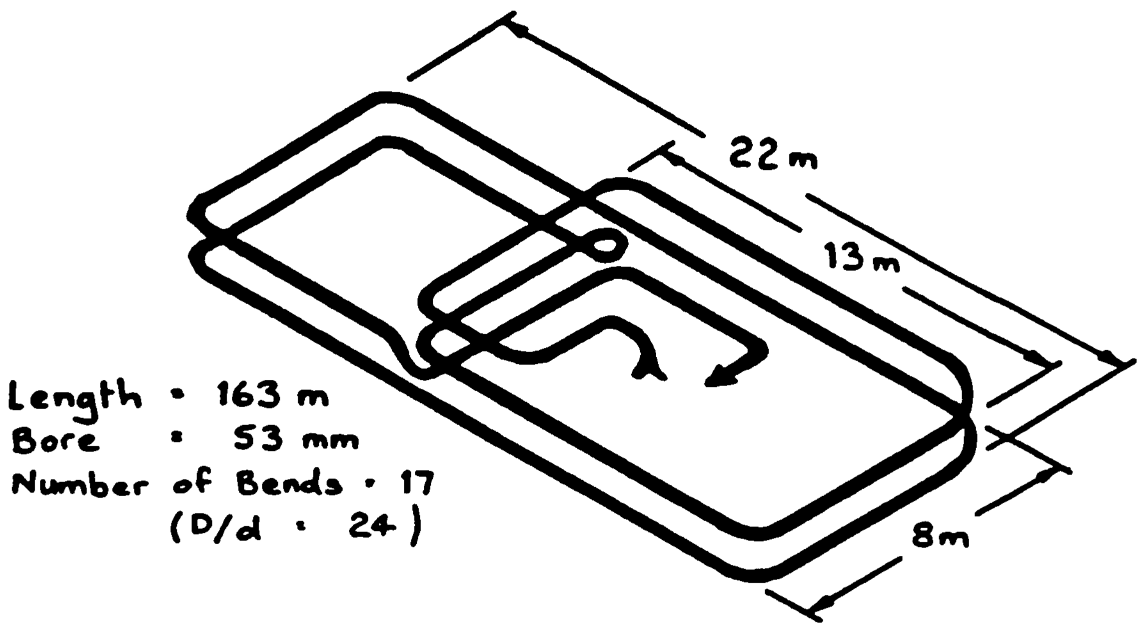
B.3.2.1 The test pipeline

The test pipeline would normally be built up in the laboratory to be as near as reasonably possible in layout to the final pipeline for which a design is required. In diameter, pipes of 2, 3 and 4in. nominal bore were available in the laboratory although air velocities in a 4in. pipe were restricted owing to compressor capacity, so work would normally be done in either 2in. or 3in. pipes unless special circumstances justified the hiring of a large diesel compressor; these sizes are at the smaller end of industrial systems so almost invariably the final pipeline would be of a significantly larger diameter than the test line, typically by 2 to 3 times.

For length, a maximum of some 170m or so of horizontal pipe was available; very often this enabled the test and final pipelines to be of quite similar length.

A major difference normally arose with regard to pipeline geometry. The return of product to the conveying plant called for a complete loop to be used, with a minimum of about 5 bends in pipelines of up to about 50m; the maximum number of bends was about 19 in a pipeline of 100m or more, but of course only certain discrete combinations of length and number of bends was available, with approximately even distributions of bends along the line. This meant that generally the number and distribution of bends would be markedly different between test and final pipelines. Vertical sections also presented a problem, because many commercial systems incorporate relatively long risers, which could not be accommodated and in any case would be balanced by almost equally long downcomers to return the product to the conveying plant.

Bend geometry was a contentious issue, because at that time there was no definitive work available showing the difference in pressure drop caused by different radii of bends. Fortunately most industrial systems use long radius bends (typically 1 to 2 metres radius) so these were normally used in the test pipeline as well.



Sketch of Pipeline used for Determining Conveying Characteristics for Product

Fig. B-2
A typical test pipeline used in the Thames laboratories (ref. 1)

B.3.2.2 The test procedure

With the pipeline installed, the testing usually proceeded by beginning with a high air flow rate at which virtually any product would convey, say 20m/s air velocity at a pressure of 1 bar gauge. Only a small amount of air would be fed to the blow tank to ensure a low product feed rate. In subsequent tests, the product feed rate would be increased and possibly air flow rate reduced as well until pipeline blockage occurred or appeared imminent, to establish the minimum conveying air velocity for the product. Then further tests would cover as wide a range of conveying conditions as possible up to the highest feed rate achievable with the pressure limitation of the feeder (usually 4 bar), to excessively high air velocities (e.g. 45 m/s or so), and down to the minimum conveying air velocity.

The result would be a diagram of the conveying characteristics of the product in the test pipeline, as illustrated in fig. B-1 above.

B.3.2.3 The scaling procedure

The differences between the test and final pipelines, as outlined above. would be:-

- the length of the pipeline
- the bore of the pipeline
- the length and position of vertical sections
- the number and position of bends

The geometry of the final line, i.e. the length, the number and position of bends, and vertical sections, would usually be known. The bore would not be known; only the flow rate of product would be fixed, the object being to determine the bore of pipe necessary to accommodate this economically.

The first step would be to scale the conveying characteristics of the test line to the length and layout of the final line. The scaling for length incorporated allowances for changes in number of bends, and vertical sections; this was by means of expressing the effect of the bends and vertical sections in terms of equivalent length of horizontal pipe. An allowance was also made for changes in the "air-only" pressure drop in the line, i.e. the pressure drop which would be expected with just air alone flowing.

The first stage would be to determine the total lengths of straight horizontal pipe equivalent to the test pipeline and the final pipeline, by adding the equivalent lengths of verticals and bends to the actual horizontal lengths in each case.

For vertical sections, the work in ref. 53 indicated that the pressure drop in a vertical section is approximately twice that in a horizontal section of the same length with the same conveying air velocity and product flow rate, over a wide range of conditions. (This has since been questioned, because although the pressure drop in verticals was actually measured, the pressure drop in horizontals was estimated, for comparison, from the total pipeline pressure drop using the allowances for bends given below). On this basis the equivalent length of verticals was simply taken as twice their actual length.

For bends, the equivalent length would be determined from the graph in fig. B-3 below; this originated from some work done by Mills conveying cement through pipelines of the same length but with different numbers of bends, comparing the results using the scaling for pipeline length outlined below to determine the equivalent lengths of the bends (ref. 1). The graph is the result of a clear correlation between the bend equivalent length and the inlet conveying air velocity for the two pipelines Mills tested; unfortunately the strong dependence of equivalent length on air velocity highlighted the major problem with this approach, that the bends along any one pipeline do not all have equal equivalent lengths because the air velocity increases towards the end of the line, by a factor of at least 2 normally, and often much more. This means that with a different number and distribution of bends along the pipe, the estimates from this graph can be wildly in error which can be most significant when the total bend equivalent lengths in a system are comparable to the horizontal length, which is frequently the case. Also with different products, the position of the line on the graph is known to be different, and it is not economic to test every product to determine this.

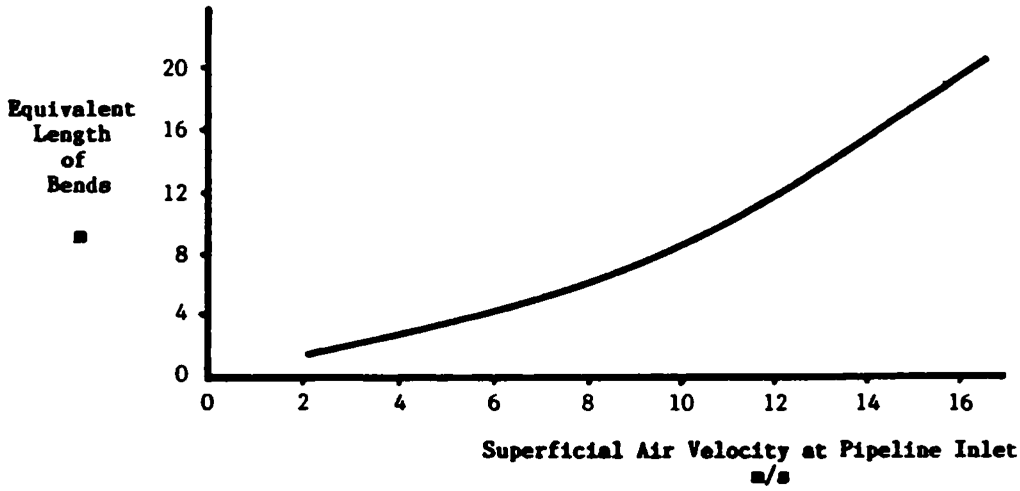


Fig. B-3

The correlation between bend equivalent length and line inlet air velocity for cement, as demonstrated by Mills

Nevertheless, using these methods the total equivalent lengths of test and final pipelines would be determined by adding together the horizontal lengths and all the bend and vertical equivalent lengths for each. These figures would then be used to scale the mass flow rate of product between test and final pipelines of the same bore size, for the same pressure drop and air flow rate, in inverse ratio of the total equivalent lengths; i.e. the model used was:-

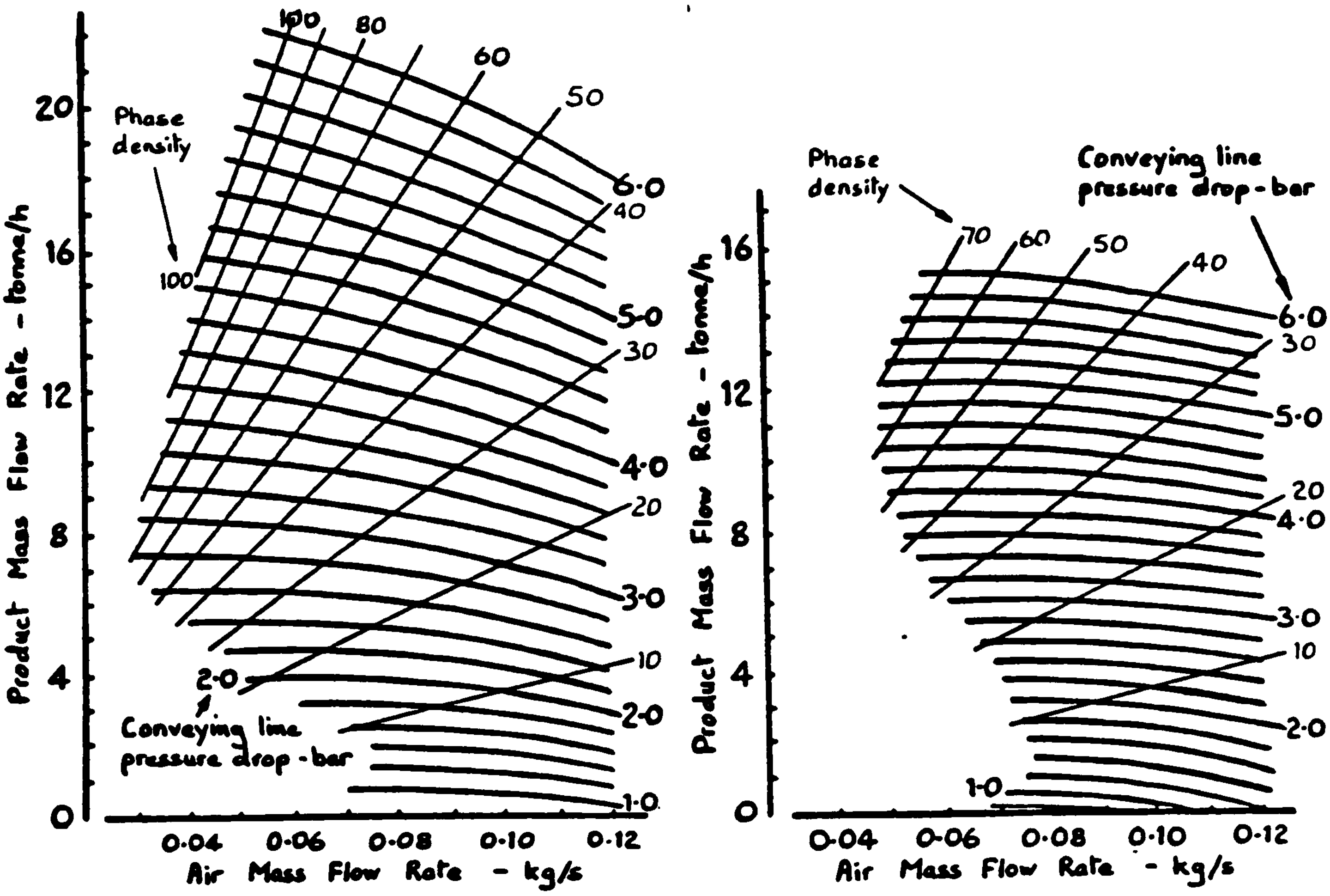
$$\dot{m}_{p2} = \dot{m}_{p1} \cdot \frac{L_{e1}}{L_{e2}}$$

where $\dot{m}_p$ = mass flow rate of product

L_e = equivalent length of pipeline

and subscripts 1 and 2 refer to test and final pipelines respectively, for the same pipe bore, pressure drop and air flow rate.

The diagram representing the conveying characteristic of the product in the test line (e.g. as illustrated in fig. B-1) would be scaled using this rule, each point on the diagram being scaled to an "equivalent" condition for the new pipe equivalent length. This is a very tedious process; it involves using a different scaling factor for every different pipeline inlet air velocity because bend equivalent lengths are affected by this.

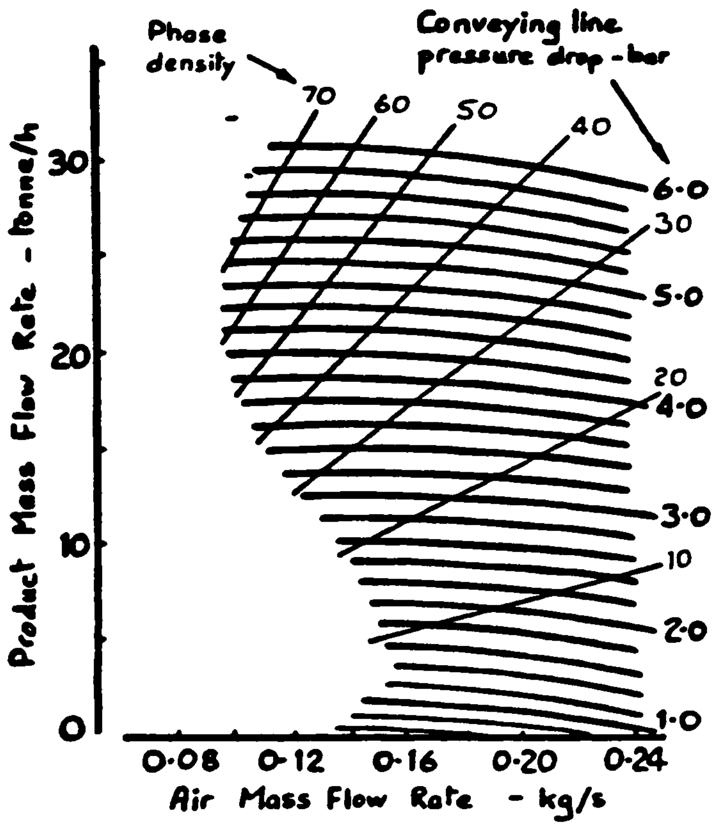


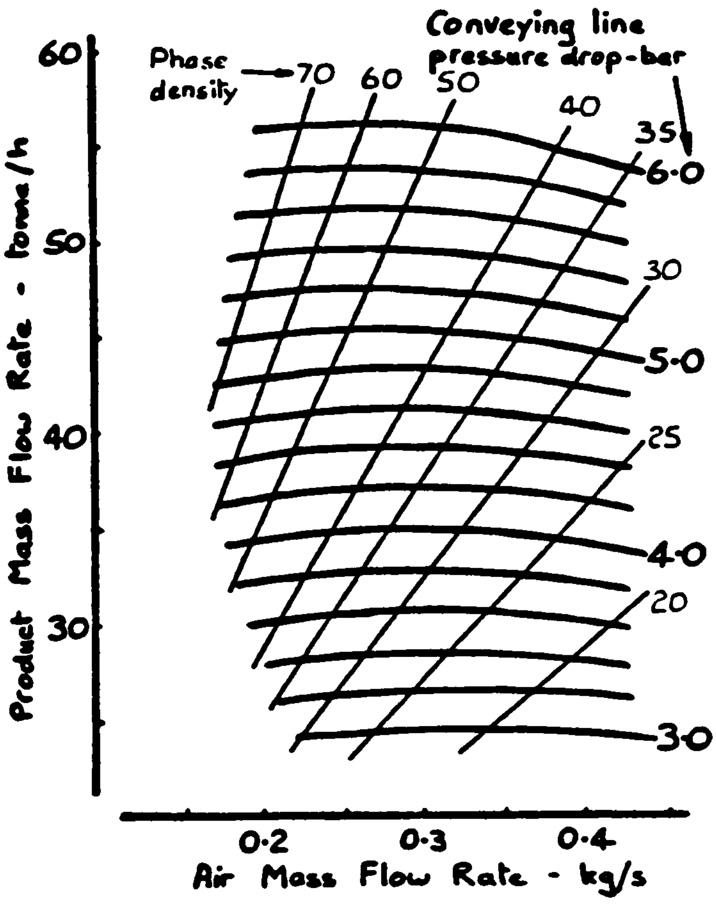
(a) Fig. B-4 (b)

(a) Conveying characteristics of test pipeline (163m, 17 bends), and (b) same scaled to layout of plant pipeline (250m, 5 bends).

Eventually a new set of conveying characteristics for a pipeline of the final layout, but the bore size of the test line, would be arrived at. A further correction would then be applied to allow for the increase in pressure drop with air only in the line, based on the notion that with a longer line and a given pressure drop, more of this pressure drop is used to convey the air and so less would be available to transport the solids; so to enable the same amount of solids to be transported, the increased "air-only" pressure drop would be added for each condition, and new contours of pressure drop would be drawn on the conveying characteristics diagram for the final line.

Next the data would need to be scaled for pipeline bore; almost invariably the use of a bore size the same as the test line would not accommodate the flow rate of product demanded in the final line. The scaling for bore size would be on the simple basis of flow rate of product being proportional to pipe area for the same pressure drop and air velocity, but again the new diagram for each bore size would need to be corrected for reduction in the "air only" pressure drop in larger lines.





(a) Fig. B-5 (b)

Same characteristics, for plant layout, scaled to two more pipe sizes
(a) 75mm bore (2.5in. n.b.), (b) 100mm bore (4in. n.b.)

Once the procedure for pipe bore had been repeated a number of times for different pipe sizes to obtain a set of conveying characteristics for the product in each, a suitable conveying condition with the design flow rate of product and required conveying air velocity could be read off each, and then the designer would be in a position to compare a number of possible systems for the duty, each with a different pipe size, air flow and pressure requirement - determining the type of feeder and air mover used, and power consumption - to decide on what type would be most economical from the points of view of capital and running costs.

B.3.3 Features and difficulties of the method

As has been demonstrated, this procedure of scaling is not by any means straightforward, it being at least a couple of man days of work to obtain the information required for system design from the test data. The hazards resulting from the uncertainty in values for bend equivalent lengths were clearly quite considerable, leading to a need to design very much with conservatism in mind in order to ensure reliability of satisfactory system performance; the risk of very costly plant downtime far outweighing the benefits to be obtained from absolute minimum power consumption in general.

Thus it was possible, in principle at least, by conservative design using this rather time consuming method, to obtain outline designs for pneumatic conveying systems which stood a high chance of working satisfactorily. In practice, this method was used at Thames mostly to assess the suitability of products for pneumatic transport, or to compare the economics of different systems which had been put forward to a client for a plant by different vendors. Even where it had been used for outright system design, it was difficult to assess the accuracy of predictions made using the method because most plant systems once installed do not have the instrumentation necessary to do this.

The result of this was that the degree of error inherent in the method, and thus the amount of conservatism necessary in design using it, was almost totally unknown. This was clearly not a happy situation because it would undoubtedly lead to excessively conservative designs which would consume more power and suffer greater wear and/or product degradation than necessary.

The final difficulty was that there was no way in which the testing and scaling method could be used to deal with pipelines which incorporate an increase in bore size at some point along the pipe, to keep air velocity down as pressure reduces and the air expands. An undergraduate student project under the direction of Mills had shown that by this means, the power requirements of a system working at high inlet pressure could be reduced very significantly; commercial systems incorporating such "stepped" pipelines are increasingly finding favour because they allow much greater conveying distances to be achieved, even in excess of 800m which would be quite impossible with single-bore lines. Examples are to be found in ref. 54. It was apparent that there was no way in which a scaling approach could realistically be used to design such a pipeline because it would not give any indication of the optimum positions of the steps, let alone the problems involved in predicting the pressure drop.

B.4 Conclusions

The work which had been done in learning about and analysing the testing and scaling method for pipeline design was considered most valuable because it had indicated the major areas of difficulty with the approach, and had shown that these would not be easy, or perhaps even possible, to overcome; these problem areas being scaling for bends and dealing with "stepped" pipelines referred to above.

The results of this investigation affected the goals of the project described in this thesis very significantly, in particular making it clear that the testing-and-scaling method had reached the limit of its development and any better method would require a different approach.

APPENDIX C

EXAMINATION OF METHODS USED FOR SINGLE PHASE FLOW

C.1 Introduction

In Chapter 2, some mention was made of the methods used to predict pressure loss in the flow of liquids and gases along pipelines, i.e. single phase flow conditions. This Appendix treats in a little more detail some of the techniques mentioned there, and which have been applied in the work done for this project.

C.2 Laminar flow

The method universally used to predict the pressure drop along a pipe carrying a Newtonian fluid flowing in a laminar regime is the Poiseuille equation. Poiseuille, a French medical doctor, published in 1840 the results of empirical work relating to the pressure drop of water flowing along fine glass capillaries, which he had studied in the hope of being able to predict the pressure drop of blood flowing through capillaries in the human body. Poiseuille showed that pressure drop was inversely proportional to the fourth power of capillary bore and proportional to the flow rate, although he did not actually publish an equation connecting them all, nor did he relate the pressure drop to fluid viscosity. A German engineer, Hagen, also obtained similar results working with water in small brass tubes about the same time.

Subsequently (G.J. Wiedermann, publishing in 1856) it was shown that an expression to predict the pressure drop caused by fluids in small tubes could be derived mathematically, by analysing a physical model of an orderly flow of cylindrical laminae of fluid moving along, exerting forces on one another by means of the shear stress caused when shearing a viscous fluid, as shown overleaf:-

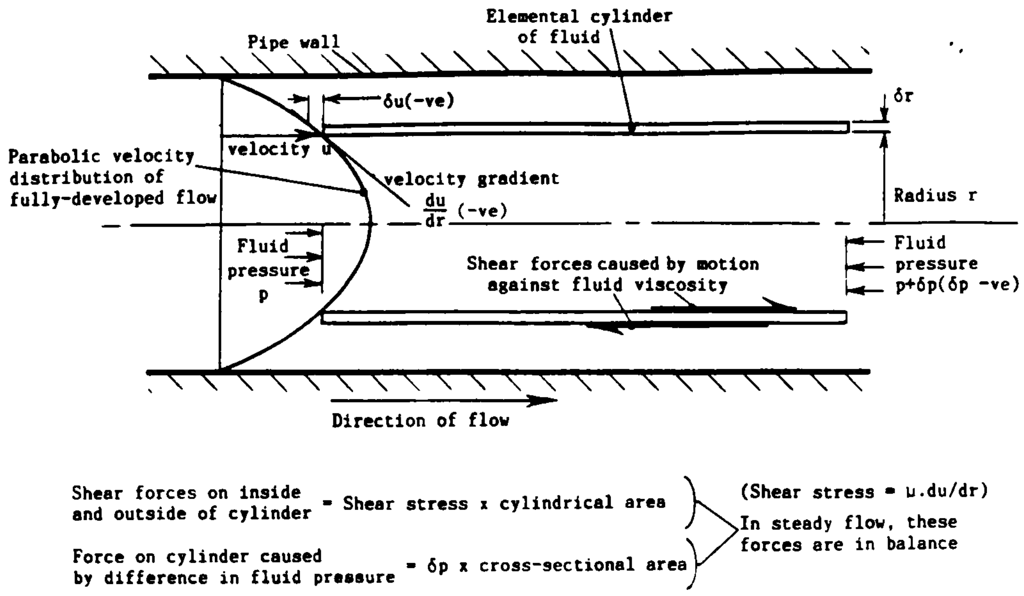


Fig. C-1

The physical model of laminar flow whose analysis yields the Poiseuille equation

Applying Newton's model of fluid viscosity (published much earlier, sometime around 1670) to this it was a relatively simple matter to derive an analytical expression for pressure drop caused by the viscous drag between the cylindrical laminae. The analysis may be found in any basic textbook on fluid mechanics, the resulting expression being

$$\frac{dp}{dx} = \frac{-128\ \mu\ Q}{\pi D^4}$$

where $\frac{dp}{dx}$ = pressure gradient in pipe,

Q = flow rate of fluid,

μ = fluid dynamic viscosity, and

D = pipe diameter.

This expression which has become known as the Poiseuille (occasionally Hagen-Poiseuille) Equation, in spite of the fact that neither Hagen nor Poiseuille actually originated it, was found to be accurate for a certain range of flow conditions; presumably such advanced students of fluid flow would have recognised two distinct regimes of flow, at least for fluids in free fall (e.g. out of a tap); certainly Hagen recognised that above a certain fluid velocity in a pipe, some significant change in the flow occurred, also recognising the effect of fluid viscosity on the transition velocity. It seems unlikely that any clear ideas on this phenomenon would have been about at this time (before Reynolds), and in any case there would certainly have been no means of differentiating between them.

Although the Poiseuille equation is often used in the form given above, an alternative method of using it is to employ the Darcy equation (described below) which was obtained for turbulent flow, but using the 'Friction Factor' coefficient for laminar flow, in turn a function of Reynolds Number ($f = 16/Re$) but plotted on the Moody Diagram along with the friction factors for turbulent flow; substituting this in gives the Poiseuille equation itself. This would appear to be simply the result of an attempt to present all of the means for prediction of pressure drop in a consistent form, to remove the need to use separate equations.

C.3 Turbulent flow

It seems likely that the idea of a flow regime in a pipe, not conforming to the orderly physical model from which the Poiseuille equation was derived, would have been well established before Reynolds performed his well known experiments in the early 1880s. Certainly it had been shown by French engineer Henri Darcy that for higher velocities and larger pipes than used by Poiseuille (Poiseuille's pipes were very small, 0.02 and 0.10mm bore), pressure drop was more nearly proportional to the square of fluid flow rate (and hence velocity) instead of the direct proportionality demonstrated by Poiseuille. Darcy did not actually publish the equation

$$h_f = \frac{4fl}{d}.\frac{c^2}{2g}$$

(h_f = head loss along pipe length l,)
(bore d, with fluid velocity c)

now commonly called after him; the work which he did publish in 1857 included the equation

$$h_f = \frac{c^2 l}{k^2 d}.\ 1+\left(\frac{1}{12d}\right)$$

(k = a coefficient)

within which his major contribution seems to have been the introduction of the term in the bracket to allow for the effect of diameter, as a result of careful measurement and empirical work.

Head loss is of course more convenient than pressure drop as far as civil engineers are concerned, because their large pipes are frequently driven by gravity. Simple empirical methods did not give exactly a square law power of velocity (as evidenced by the second equation above), so it seems likely that the decision was taken by significant workers in the field, to use a square law (both for convenience and also to make the expression dimensionally homogeneous) and accommodate the variation by introducing a coefficient, leading to the Friction Factor in the first equation above. A further attraction to the use of a square law would have been that it would align with the Chezy equation relating velocity to fall in flow along an open channel; Chezy published this in 1796, as a result of empirical work on canals and the river Seine, and it would appear that there was a desire to see open channels and pipes as a single problem with one comprehensive solution.

Two other interesting observations on the form of the Darcy equation are to be made. Firstly the use of the velocity head ($c^2/2g$) term enabled the calculation of losses to fit in with the use of the Bernoulli equation which employs such a term (the work of Bernoulli pre-dates all the work described here, save that of Newton), and also this would have been easily understood by civil engineers, to whom the concept of 'velocity head' is an obvious one. Secondly the use of 4f rather than a coefficient four times as large, which again resulted from the influence of civil engineers whose pipes and ducts are not always circular and frequently run less than full; in order to accommodate this, the expression was originally developed to use 'hydraulic mean depth' instead of diameter, the hydraulic mean depth being simply the ratio of [cross sectional area filled by liquid] divided by [wetted perimeter] (again this quantity came from Chezy). Thus originally the expression was, logically,

$$h_f = \frac{fl}{m}\cdot\frac{c^2}{2g}$$

where m = hydraulic mean depth.

For a circular pipe, however, $m = d/4$, thus fl/m becomes 4fl/d for this case, leading to the form of equation normally used. It is interesting to note that in American texts, the 4f used in European practice is indeed often replaced with a value of f four times as large, and in fact Moody's significant paper (discussed below) embodied this - although where his diagram is reproduced in European texts, his values of f are changed (generally without explanation) to fit.

Returning to the situation existing when Reynolds entered the picture, there was certainly an understanding of two distinctly different forms of flow in pipes, and expressions for predicting pressure loss in each case would have been available.

Nevertheless, there seems little indication that anyone really recognised the essential differences between the flow regimes until Reynolds took the lead by demonstrating conclusively that up to a certain velocity of fluid in a pipe fed from a settled tank, the particles of fluid travelled in virtually straight lines with hardly any mixing, which he called 'sinuous flow'; whereas at higher velocities the fluid mixed as it moved along. His experiments, described in all standard texts on fluid mechanics, demonstrated that this was the case for all of the fluids he tested, and he went on to relate the velocity at which the transition occurred to the fluid viscosity, pipe diameter, and fluid density, embodying these in the Reynolds number criterion to determine whether flow would be sinuous (laminar) or mixing (turbulent).

The situation after Reynolds made his contribution was that it was now evident what the difference between the two forms of flow was, but more significantly it was possible to decide which of the two expressions, Poiseuille or Darcy, should be used to predict pressure loss for any given case.

C.4 Pipe roughness and variation of f

It seems to have been generally recognised that for laminar flow, pipe roughness did not affect pressure drop, but for turbulent flow it did. Many experiments which had been done on turbulent flow used very smooth pipes, often polished brass or even polished glass; for these it had been found that the value of f continued to reduce as Reynolds number increased, whereas for commercial pipes this had not been found always to be the case, especially at high Reynolds numbers. Much of the work on the "smooth pipes" curve had been done by Sir Thomas Stanton working with J.R. Pannell; their pipes were mainly commercial smooth drawn, and although these are of course not totally smooth, the range of Reynolds numbers which they employed did not take them into the region where the roughness of the materials became a factor. This, together with the work done by Darcy, resulted in a well recognised curve of f versus Reynolds number for pipes working in the "smooth" region.

Clearly to evaluate the effect of surface finish, it was necessary to firstly find a means for putting a value to the roughness on commercial pipes, and secondly relate the variation of f with Reynolds number to this roughness value. The German engineer Johann Nikuradse made a breakthrough by artificially roughening the internal surfaces of pipes of various sizes in a measurable way using sand grains of closely screened sizes, bonded onto the inside of the pipes, and found that above a certain range of Reynolds numbers, f was constant and dependent only on the ratio of the size of the sand grains to the bore of the pipe, now generally known as the 'relative roughness' of the pipe. The higher the relative roughness, the lower the value of Reynolds number at which f became constant. This not only demonstrated the effect in a qualitative way, but also allowed the roughness of commercial pipe wall materials to be quantified, by comparing pressure losses measured from these commercial pipes against pressure losses along Nikuradse's artificially roughened pipes. In this way, a size of sand grains on the inside of a smooth pipe which would give the same pressure drop as measured from a commercial pipe of the same diameter, could be found, and this would be the 'equivalent sand grain roughness' of the pipe wall material. Nikuradse's results were published in 1931.

Thus the values of f for pipes operating under what is often termed 'hydraulically rough' conditions could be found, i.e. for conditions of sufficiently high Reynolds number that f was constant; likewise for pipes operating under 'hydraulically smooth' conditions, i.e. with relatively low Reynolds numbers and/or low relative roughness. It was shown by C.F. Colebrook, however, that in the intermediate range of Reynolds numbers, over which occurred the transition between these two cases, the values of f for real pipes did not conform to those found by Nikuradse for equivalent sand grain roughness. This seems hardly surprising given that the roughness of commercial pipes is not uniform as was Nikuradse's artificial roughness. Colebrook did experiments on commercial pipes, measuring head loss and finding values of f in the transition region between 'hydraulically smooth' and 'hydraulically rough' flow, and found a much smoother transition than Nikuradse had; he was able to find some usable mathematical expressions to represent his data, and these have become accepted as the method for predicting f in this region. Colebrook's results were published in 1939.

C.5 The Moody diagram

The final stage of development in the prediction of values of f was taken by an Americam, Lewis Moody, who in 1944 published a paper drawing together the work of Stanton & Pannell, Nikuradse, and Colebrook described above, and presented all of this on a diagram of f versus Reynolds number for ranges of relative roughness; the format of diagram, with logarithmic scales of f and Reynolds number, had been published before by Stanton. To the diagram he added a straight line representing values of f for laminar flow, calculated simply by comparing the Poiseuille equation with the Darcy equation and working out values of f for each Reynolds number to make them agree. Moody's diagram was reproduced in Chapter 2, but bears repeating here (overleaf).

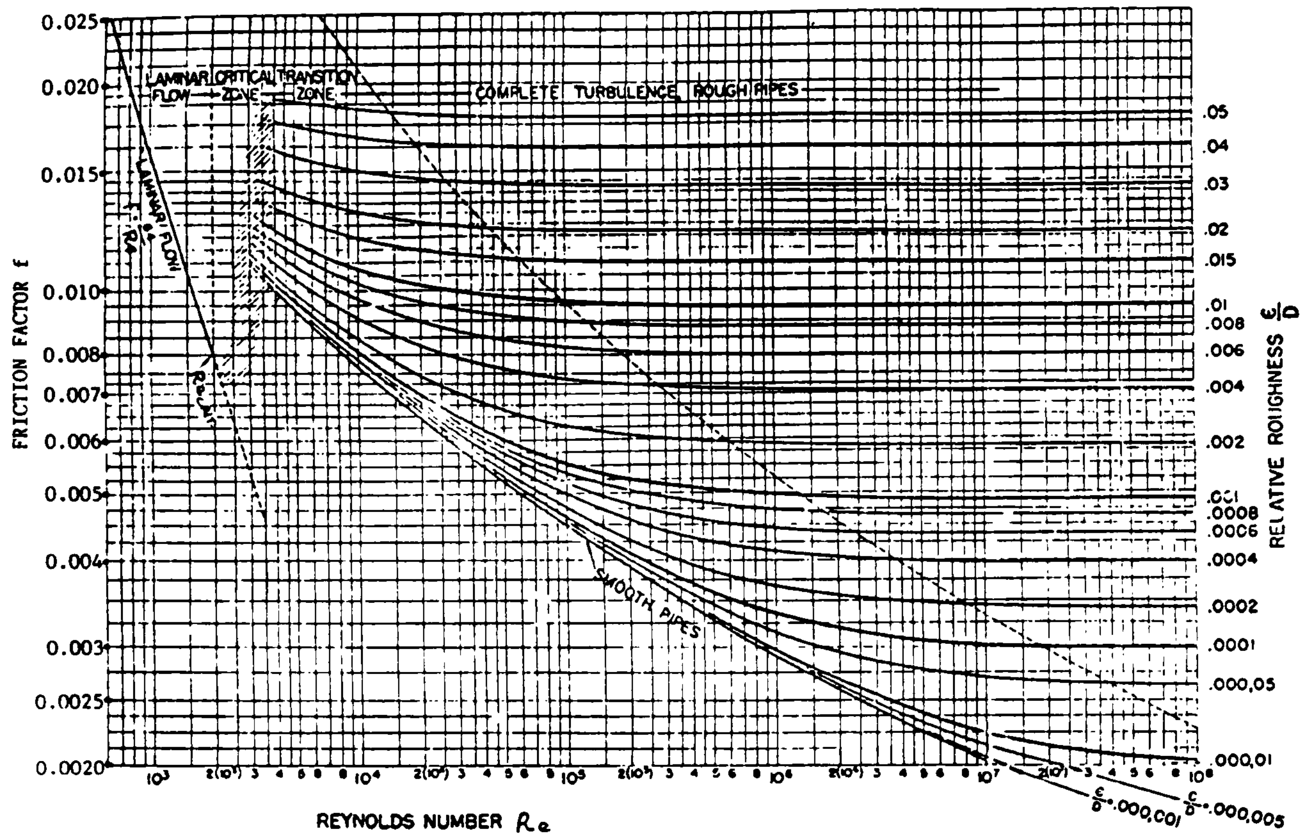


Fig. C-2

The Moody Diagram, from Moody's original paper (ref. 102)

This 'Moody Diagram' is now universally used for the prediction of pressure drop in single phase flow of Newtonian fluids, and has proved so successful that it has survived nearly half a century virtually unaltered.

C.6 Fitting losses

The extra losses caused by bends, valves and other fittings must have been evident from an early stage. It is not known who first developed the method usually used for dealing with these, but it consists of expression of such a loss as a coefficient times the velocity head of the flow. This fits in very well with the form of the Darcy equation so is very convenient; the losses are treated as though they occur as a step change in pressure at the fitting, although workers such as Ito demonstrated that in fact most of the loss develops in the straight pipe downstream of the fitting. It is worthy of note that the value of the relevant coefficient varies with pipe size.

C.7 Analysis of techniques

From the investigation required to discern the development process described above, a number of interesting strategies emerged. First of all was the divergence between strategies used for laminar and turbulent flow, the former proving to be predicted well by the mathematical analysis of a physical model whereas this approach, which would undoubtedly have been attempted for turbulent flow (at least before the reasons for the different behaviour were known), completely failed in the latter case.

Secondly it was interesting to note the reasons for the empirical Darcy equation being presented in the way it is; it was clearly arranged in this way to fit in with its applications, i.e. giving head loss rather than pressure drop, to be more useful to civil engineers, and using 4f as the coefficient because of the need for these users to deal with non-circular and partly-full ducts, as well as the use of a velocity head term instead of simply velocity, for the convenience of fitting in with established practice. Also mentioned was the use of the square law power of velocity although this would not have been strictly accurate, to allow dimensional homogeneity and convenience of use, the discrepancy being accounted for by variation in the coefficient.

Another point was that from starting with a simple empirical approach, trying to develop an expression which would represent some data, the technique grew to eventually embody a comprehensive system of stored data in the form of the Moody diagram. The use of an empirical expression with a coefficient, experimentally-determined values of the coefficient being stored on a graph against other quantities directly calculable from measurable variables, was an identifiable strategy in its own right, one which has found favour in many fields of engineering but particularly in fluids where many problems are not amenable to solution by analysis of a physical model. Other examples of the use of a system employing a dimensionally-homogeneous mathematical expression in combination with the value of an experimentally determined coefficient stored on a graph are found in the prediction of the forces and moments on an aerofoil and the prediction of the power required to drive a centrifugal pump; The prediction of the flow through an orifice meter from measured values of pressures owes something to this strategy as well, although for this the mathematical expression is not wholly empirical but comes from analysis of a simplified physical model, and the same goes for the prediction of life of gears. These techniques were seen to be a particularly important means for handling situations not amenable to physical modelling, and thus of particular relevance to prediction of pressure drop in pneumatic conveyors, for which the literature showed complete failure of such an analytical method.

Next to be mentioned should be the interpretation of Nikuradse's work in accepting that it would be difficult to obtain values for roughness of commercial pipe wall materials, and possibly not very useful anyway because of the different natures of the roughness on different materials, so instead making progress by experimenting with controlled (and hence measurable) artificial roughness and comparing the results of these tests with tests on commercial pipes to find measurable artificial equivalents to the real surfaces. It seems likely that when Nikuradse was performing his work, he was not consciously adopting this as a strategy, but more likely just looking for whatever he could find out using an arificial roughness. However, this is the way in which his results have been used, with a good deal of success. No other examples of this technique come to mind, though it is doubtful whether it is likely to be unique.

The technique used to establish the first correlations between pressure drop and fluid velocity presented by Poiseuille and Darcy, mentioned above, were examples of simple curve-fitting exercises to obtain empirical expressions. Such an approach is very widely used in engineering, an outstanding example being the use of the polytropic model to relate pressure and volume in the thermodynamic analysis of a compression or expansion process - this is purely the finding of a mathematical expression to fit a curve to data; another is found in the latest edition of B.S.1042 which gives the calculations for flow through an orifice meter

- here the graphical presentation of measured coefficients in earlier editions has been supplemented by equations which describe the lines (this is more amenable to calculation by computer). Another example closer to the subject in hand is the Blasius formula,

$$f = 0.079.Re^{-0 \cdot 25}$$

which is simply an empirical expression to represent the 'smooth pipe' curve on the Moody diagram.

The technique used by Colebrook to represent data on friction factor values in the region of transition between 'hydraulically smooth' and 'hydraulically rough' flow was subtly different, and might be seen as a mathematical means of extending the curve fitting technique. He had available an empirical expression relating f to Re for hydraulically smooth flow (the smooth pipe curve), and another for the horizontal lines for hydraulically rough flow. He combined the two equations together to give a curve tangential to each of the existing lines, then plotted experimental data to see how close it fell to his new line; it so happened that the data fell very close, so his new equation was adopted for the transition region.

A more general discussion of curve fitting is in order at this point. Curve fitting to obtain an empirical equation with coefficients is a data storage technique which is extremely economical in terms of paper and ink, and increasingly important with the growth in the use of digital computers, since a computer can deal easily with an equation but not a graph - although to a human, a graph is generally easier and also conveys much more immediate information about trends, computers know nothing of such concepts. It is important to realise that the methods for finding suitable equations to represent data are manyfold, ranging from drawing a straight line and finding its slope and intercept, through the use of logarithmic graphs to persuade the data to yield a straight line, to quite complex statistical methods of which such means as a 'least squares' analysis are but the start. One thing in common is that the user of the technique must consciously decide what type of equation to try to fit;

i.e. a straight line, a simple power law, a power law with an offset, or a higher order equation. The possibilities are infinite, but they are limited by the imagination of the user, and also the complication can become too great if the data proves difficult to model or very high accuracy is required. If all else fails and it proves impossible to obtain a useful and sufficiently accurate equation to describe the data over the whole range, then it can be split up and modelled in sections; in such a case, it becomes more economic to use a series of straight line models between definite limits, often known as a 'piecewise linear representation'. It is of some interest to note that some of the cheaper variety of electronic signal generators use such a technique to simulate a sine wave.

Another technique worthy of mention is the use of dimensionless quantities in empirical equations, which is useful in that such equations can contain fractional or decimal powers without any possibility of objection; for example in the Blasius formula mentioned above, the power of -0.25 on Reynolds number is quite in order because both sides are dimensionless, but if this equation contained quantities which resulted in there being dimensions on either side then the use of such a power could result in a lack of dimensional homogeneity in the equation, which would leave it open to objection. Thus the 'removal' of the dimensions from a quantity by building up a dimensionless group around it opens up the possibilities for the use of methods which might not otherwise be considered acceptable.

The work of Moody deserves a mention, in that he drew together a lot of work which had been done by others, examining their results and techniques to bring it all into a single convenient form. In this context, it should perhaps be pointed out that even there, some 'licence' has been applied, whether knowingly or not. The information on the Moody diagram has been obtained from a limited range of experiments, and it seems quite likely that many real applications for which this data is used lay outside the range which the experiments covered, although the values of Reynolds number and pipe relative roughness are in order. For example, the Moody diagram came principally from experiments with liquids (although Stanton and Pannell did some work using air), yet values from it are frequently used to predict the pressure drop in pipes carrying gas. This must result in some inaccuracy, but in practice this is not a difficulty because of the nature of the problem the data is usually used to solve, namely the choice of a suitable size of pipe to carry a given flow rate of fluid; for any given flow rate, the curve of pressure drop against pipe size has such a pronounced turning point between pipes which are excessively large (thus excessively expensive to buy) and those which give excessive pressure drop (thus being excessively expensive to run), and the choice of available sizes for pipe systems is relatively so limited, that even quite a large inaccuracy in predicting the pressure drop would not result in the wrong pipe size being chosen. For this reason some inaccuracy is acceptable.

Finally the method for dealing with fitting loss shows a couple of interesting points. The idea of treating the loss as though it occurs as a step change at the fitting, although in practice this is not so, is significant; it simplifies matters considerably in comparison with trying to model the true pressure profile near the fitting. Also it seems certain that such a simple coefficient, the variation of which with flow rate is completely ignored, is bound to be inaccurate - but in practice this does not matter because the fitting losses are invariably a small part of the total losses in the system. Another significant point is that the method has clearly been moulded to fit in with the other relevant equations, and not simply developed in isolation.

C.8 Conclusions

The investigation of the techniques discussed here was tremendously rewarding, in that it gave an insight into the many established methods for dealing with the problem of predicting the behaviour of the real world. It was evident that the time which had been required to develop a usable system was considerable, and so the development of such a system for pneumatic conveying pipelines would probably not be a quick task, even starting with the advantage of seeing the many techniques available. It was also apparent that the basic strategy of obtaining data then building a storage system for this looked to be a promisimg one.

Two major strategies were identified, the mathematical analysis of a physical model and the use of the data gathering, storage and retrieval technique. Within the second strategy, which appeared to be the most relevant to the project in hand, some seven distinct methods for aiding a solution had been identified and at various stages in the project, most of these were found to be useful.

Finally it is of some interest to observe that a considerable amount of detective work was necessary to unearth the origins of the method described above. Most of the original papers are now largely forgotten, and the method which has been developed is widely used with little desire to understand how it came into being or why it is in the form which it is. Much of what is written above would no doubt have seemed obvious years ago, but is now lost; this has not jeopardised the success of the method, principally because of the nature of the curve on a graph of head loss versus pipe size which is used to select a pipe for a given duty (the fifth power curve gives a very sharp cut-off between economic and uneconomic pipe sizes). Clearly, though, such a situation leads to the potential for further misinterpretation and misuse of the system.

APPENDIX D

MENTAL AND MATHEMATICAL MODELLING OF GAS-SOLID FLOW

This area was mentioned briefly in Chapter 2 but not enlarged upon. It was a part of the work which occupied a significant amount of time however, and although no clear methods for the mathematical modelling of flow in pneumatic conveyors emerged, it led to some considerable insight into the mechanisms of fluid flow and the effect of the introduction of particles into a flow. Specifically, the mechanisms by which pressure drop develops were considered at length, and some conclusions were obtained about the ways in which the ordered mechanical energy put in by this means is converted into the disordered internal energy in the fluid.

D.1 The aim

It was accepted that to predict accurately the pressure losses by analytical means would be quite impossible because of the complex nature of the processes occurring inside a gas-solid flow system. The original idea was rather that if some sort of analytical model could be developed for pressure drop in straight pipe and that caused by bends, then its predictions could be tried against experimental data, and the comparison may give some clue as to suitable systems for storing this data in a compact and easy to use form. As a first stage in attempting to construct an analytical model, some considerable thought was given to the processes which might occur in the flow of a gas-solid mixture in a pipe.

D.2 Mental models of the mechanisms of pressure drop in gas-solid flow

The mental models which had formed in the mind of the author during the first year or two of the project were outlined in Chapter Two; they deserve repeating and amplifying here.

D.2.1 Flow of air only

Firstly the flow of air alone in a pipe was considered, and possible mechanisms which could cause the pressure drop encountered in such a flow were thought about.

In turbulent flow in a pipe, it is clear that a fairly vigorous mixing of the fluid takes place, which causes a continuous 'randomising' effect on the momenta of the particles of fluid. The reasons why this mixing occurs were considered. It is quite apparent that any velocity difference in a body of fluid will produce the effect provided that the viscosity of the fluid is not sufficient to overcome it. One has only to pour water into a measure of whisky in a glass to observe this motion taking place (the difference in the refractive indices of water and alcohol enable the interfaces to be seen). This is driven by the difference in velocity between the relatively still whisky in the glass and the water entering with a fairly high downwards velocity. One can imagine a small region of fluid on the boundary between the water and the whisky, subjected to a significant velocity difference between its two sides, and as a result of this, experiencing shearing forces (caused by fluid viscosity) which would tend to make the fluid rotate in small eddies. In order for the eddying motion to develop to any extent, the fluid must be sufficiently free to rotate; if the fluid viscosity is high, then the high rates of shear involved with this process would produce high shear stresses, the effect of this being to prevent a large velocity gradient from developing under the relatively small forces involved, and also to damp out any small scale eddying motion quite quickly.

Such eddies appear to us to be quite random. It would seem natural to think that the physical models which we call the laws of motion and fluid viscosity could be applied to such a system, because the eddies are many orders of magnitude larger than the particles of fluid, whether these are considered to be single molecules or blocks of molecules as is more generally thought now. In reality, though, the scale of a glass of water in comparision with the sizes of the eddies is such that the complexity would make it impossible to analyse; the more so because the system operates in three dimensions and any part of it affects all other parts indirectly. Once the eddies become broken down below a certain size then the effects of the motions of the particles of fluid would become a factor in their behaviour, and this again appears to us random (e.g. the Brownian motion demonstration). Therefore although there may be fundamental patterns in the behaviour of the eddying motion, we must consider it as random.

It seems clear that the effect of this eddying motion is to take the ordered momentum of particles moving in a straight line in one direction and convert some of it, continuously, into momentum in other directions, thus turning the ordered kinetic energy of the fluid particles into disordered kinetic energy. This disordered kinetic energy of the particles is what we perceive, on a larger scale, as internal energy of the fluid. In the case of the water mixing with whisky, the initial kinetic energy of the water is converted into a small temperature rise in the mixture - although this temperature rise is not perceptible, such is the physics of the substances involved. In other cases this turbulent eddying motion, if powerful enough, can produce noticeable temperature rises (e.g. in a dynamometer water brake).

Thus, in a turbulent flow, if ordered energy is not continuously fed into the fluid to replace the loss of ordered kinetic energy into disordered kinetic energy, eventually the ordered kinetic energy will disappear and the flow will stop. The input of ordered energy is achieved, in flow through a pipe, by a pressure drop in the direction of flow. This can be thought of as a force x distance (per unit time) effect acting on an elemental length of the flow; we usually refer to this as a reduction in pressure energy.

In flow through a pipe, the very fact that there is a velocity difference across the flow means that provided the fluid is not too viscous, the velocity too low, or the pipe too small, this eddying motion will occur and the flow will be turbulent - the "Mixing Flow" which Reynolds first observed.

The eddying will begin in any region of sufficiently high velocity gradient, i.e. near the wall of the pipe, and will by its random nature spread across the pipe. Very near to the wall, however, the fluid will not be as free to rotate (because of the proximity of the walls), so in this region the viscous forces will damp the eddying and a laminar boundary layer will develop.

The effect of the roughness of the wall of the pipe was considered. If the laminar boundary layer was thick enough to accommodate all protuberances on the wall, then these could have little effect on the intensity of the eddying in the turbulent core of the flow where most of the fluid passes. In such a case, making the wall smoother would not affect the pressure drop at all; this is generally accepted as an explanation for the phenomenon of "Hydraulically smooth flow" in a pipe, i.e. the condition where any reduction in the magnitude of the roughness does not cause the pressure drop to reduce (see Appendix C for further explanation of this in the context of prediction of pressure drop).

If, however, the laminar boundary layer is so thin (owing to a very high velocity gradient or very low fluid viscosity - i.e. high Reynolds number), or the pipe wall is so rough, that the peaks of the roughness protrude through the laminar boundary layer into the turbulent core, then these will create small regions of high velocity gradient which will produce further eddying, beginning in turbulent wakes downstream of the peaks but of course spreading out to add to the intensity of the general turbulence in the flow. Greater turbulence of course leads to a faster breaking down of the ordered kinetic energy of the flow into the disordered kinetic energy which we know as internal energy, thus greater pressure drop. This condition would manifest itself as "Hydraulically rough flow", where any increase in the roughness leads to increased pressure drop.

D.2.2 "Lean phase" pneumatic conveying through a straight pipe

Having obtained a fairly clear mental model of turbulent flow, the complication of adding particles would now have to be considered. "Lean phase" flow, where the particles are well dispersed in an air stream of high enough velocity to prevent settlement on the bottom of a horizontal pipe, was considered first.

The introduction of small stationary particles into a fast moving air stream was thought about. Initially, the relative velocity between the particles and the air would be high, which would have two effects; the drag forces on the particles would be high, leading to rapid acceleration, and also there would be a turbulent wake behind each particle. The eddies formed in the turbulent wakes would add to the general turbulence of the flow in the same way as that behind a protuberence from the pipe wall referred to above, and in the same way this would contribute to increased pressure drop in the flow.

As the particles accelerate in the direction of flow, so the relative velocities between particles and air will reduce. The acceleration of the particles will become less rapid, and the size of the turbulent wake will reduce so the extra contribution to pressure drop in the flow, over and above that for air only, will become smaller.

Continuing on further downstream, the velocities of the particles will begin to approach that of the air stream. However, they will not quite achieve this, for a number of reasons. In a horizontal pipe, gravity will tend to pull the particles towards the bottom of the pipe, and when a particle approaches the pipe wall it will enter the boundary layer where the air moves more slowly than that in the core, and possibly more slowly than the particle itself; some calculations using the accepted "seventh-power" law for a turbulent boundary layer showed this to be a good deal thicker than the diameters of the particles of many powdered products. In this case the particle will be slowed down and will at the same time produce a wake creating further turbulence. The particle will experience a difference of air velocity across it which will make it spin, distorting the flow around it; this will produce a sideways force which will further affect the motion of the particle, tending to push it back into the central core of the flow if it is still travelling faster than the air in the boundary layer. There is no reason to suppose that this process would be stable, pushing the particles to a position where they would flow along just in equilibrium between the boundary layer and the core; the trajectories of the particles will be subject to other disturbances as the turbulence in the flow behind them passes over them, so there seems every likelihood that even in a straight pipe the particles will be continually travelling into and out of the boundary layer, being alternately speeded up and slowed down in a random manner, never reaching the full velocity of the air in the core and always producing turbulence causing increased pressure drop over that with air alone flowing. The particles may also collide with the wall if they are sufficiently massive to penetrate the boundary layer, and this will have a very similar effect to that described above, making the particles spin in the same way. Collisions between particles are another possible factor, although some calculations showed that the mean inter-particle distance for a typical lean phase flow would probably make these fairly uncommon.

Even in a vertical pipe without gravity acting across the pipe, this last process will still occur, but gravity will now have an obvious effect on the velocities of the particles, differing depending on whether the flow is up or down.

The shapes of the particles will undoubtedly have some effect on the magnitude of the pressure drop which they cause, because a rougher shape will produce more turbulence in its wake, particularly if spinning. The distribution of sizes of the particles in a product could also be a factor because if wide, the smaller particles could achieve significantly different velocities from the larger ones, thus making inter-particle collisions more likely.

D.2.3 "Dense phase" pneumatic conveying through a straight pipe

The term "dense phase" is usually taken to mean conveying where a significant amount of product is in some sort of contact state in the pipe, anywhere from a stationary or moving bed on the bottom of a horizontal pipe, possibly with waves or dunes moving along on the top of this, through to full bore plug flow. To obtain any sort of mental model of the processes here would be very difficult, although some time had been spent observing dense phase flows through sight glasses.

Beginning with a condition where particles are settling out on the bottom of a pipe from an air stream, probably because the velocity of the air was insufficient to entrain the particles properly, then the reducion of cross sectional area caused by the occupation of the lower part of the pipe by stationary particles would increase the air velocity in the passage above so that it would carry particles just about in entrainment; there would be a continual trading of particles between the air flow and the surface of the bed, and the particle velocities would be considerably lower than the air velocity, leading to a great deal of turbulence in the air and thus high pressure drop.

The process of interaction between the air flow and the surface of the bed would be unlikely to be stable; any small undulation in the interface would cause further disturbance, resulting in velocity variation and rotating flows, in turn leading to mobile duning in the same way as sand dunes move in the wind. Such a process would become more marked as air velocity reduced.

Under extreme conditions this process could lead to the waves or dunes touching the top of the pipe and filling the cross section. A number of things could happen under such circumstances. If such a dune was small in length and had some momentum then the air could simply push through it and sweep it away. This would cause a pulse in the air flow downstream, which might affect processes going on there. If the dune was of too great a length for the air to simply sweep it away then clearly the passage of air would be restricted, and the pressure would build up behind the dune (now a full-bore slug). At this stage it would seem that the bulk properties of the product would come into play. There seem to be three distinct extremes of behaviour of bulk products in response to the interaction of air with contact beds, from the work of Geldart. Some products are easily permeable to air (granular products); some products which are of low permeability retain air well, remaining in a fluid state for some time after mixing with air; whilst some products are neither easily permeable nor do they retain air well, very quickly losing fluidity unless continually re-mixed with air. Observation of some products shaken in jars showed the last two cases, for example a jar of cement dust remains in an expanded, semi-fluid state for some time after a vigorous shaking whereas a jar of alumina, of similar particle size, returns to a "dead" state immediately.

Should the product be easily permeable by air, e.g. a granular product, then air will flow through the slug; there will be some pressure drop involved with this, as the velocity of the air through the interstices will be high and extremely turbulent. Air flowing through the slug at high local velocities will tend to lift the product from the front of the slug and deposit it just a little further downstream, effectively moving the slug along. This author had observed such slugs of permeable product moving along a pipe one after another, quite stably.

In the case of a product which is not easily air permeable but retains air well (e.g. cement) the slug would effectively block the passage of air but the fluidity of the material in the slug would enable the air to push it along, perhaps like a lubricated piston in a cylinder. The slug would most likely be unstable, leaving material behind and eventually breaking up and new slugs forming elsewhere. A pressure difference between front and back of such a slug would be necessary to push it along, the energy input from this being randomised mechanically by the friction between the slug and the pipe walls, and internally within the slug. The pressure difference needed would depend on internal and wall friction properties of the product of the slug in its semi-fluid state as well as its dimensions, and the instability of such a slug would cause fluctuations in air pressure downstream, affecting the motion of other such slugs.

If the product was neither air permeable nor retentive of air (e.g. alumina), then a slug of any significant length would again block the passage of air but would require such force to move it, because of the quick de-aeration and loss of fluidity of the product, that insufficient air pressure would be available. This would cause a pipeline blockage. Such a situation could arise even with a product with some air retention properties, if the time between the slug forming and the air pressure being sufficient to move it was long enough to allow de-aeration. This could happen with low air flow rates, and indeed this author gained considerable experience of unblocking pipelines of such products when experimenting with low air flow rates.

D.2.4 Bends

A little thought showed the mechanism of pressure drop caused by bends to be fairly straightforward in the light of the foregoing mental models. For air only, the inevitable secondary flows resulting from a change in direction would contribute to the general level of turbulence in the flow, thus leading to a higher pressure gradient immediately downstream of the bend where this extra turbulence persists.

In lean phase flow it was equally clear that a change in direction of the pipe would lead to an increase in the rate of collisions of particles with the pipe wall, or at least with the boundary layer. This would cause a general slowing down of the particles, which would lead to increased turbulence and pressure drop in the air downstream of the bend where the relative velocities between particles and air would be greater; in much the same way as the pressure drop caused by acceleration of the particles from rest, as described above.

In dense phase flow the mechanism might be far more complicated; a flow might change its nature immediately downstream of the bend, because of the reduction in particle velocities and thus reduction in cross-section free for air flow; this might promote slugging. On the other hand, the reduction in the already low particle velocities in dense phase flow might be a good deal less than in lean phase, so the effects of bends may be of different significance in such a flow.

D.2.5 Conclusions from mental modelling

From all the above it was apparent that the processes involved in pressure drop with turbulent flow along a pipe were very complex, even without particles present, the addition of these causing probably an order of magnitude of further complication. Obviously any comprehensive analytical modelling of these would be out of the question, but having perhaps some inkling of what these processes might involve led to a few ideas for exploration in the hope that they might give a little guidance to the general direction of the work.

One specific avenue of approach would be to try to model the acceleration of solids from rest in an air stream. It was expected that a physical model of this process could be built up, which could be analysed on the bases of conservation of mass, momentum and/or energy, (perhaps using the Steady Flow Energy Equation) to obtain a value for the pressure drop of the air in accelerating a mass of particles. This might indicate the pressure loss to be expected at a feeder, and possibly the upper limit of pressure drop to be expected after a bend, since it was thought that only a part of the kinetic energy of the particles would be lost in colliding with the wall at a bend.

Another idea was to construct and analyse a physical model of the acceleration of a single particle in an air stream, using the commonly accepted method for predicting drag on an immersed body. Again this might give some useful information on pressure losses after bends and feeders.

No ideas were readily forthcoming for modelling the pressure drop in fully developed flow along a straight pipe, however.

D.3 Mathematical modelling of acceleration pressure losses

D.3.1 First attempts

Several false starts were made on this. Early attempts were based on conservation of energy, and failed because it could not be understood how the increase in velocity, and hence kinetic energy, both of particles (caused by the effect of the air flowing past them) and air (caused by the expansion of the air with falling pressure) could occur given that temperature appeared to be constant along a conveying line.

The problem was that it appeared that the flow of pressure energy past any point along the pipeline would be the same, because the pressure energy of a gas is a function only of temperature ($pV = mRT$). Taking the internal energy of the gas into account did not help, because this is a function only of temperature ($U = C_v T$) in the same way as pressure energy is. The only other energies available being potential energy (constant in a horizontal pipe), and kinetic energy, which was known to be increasing, it could not be understood how the process operated, let alone how it could be modelled. The only obvious way in which sense could be made of this would be if temperature was actually falling along a conveying line, allowing a reduction in enthalpy (the sum of pressure energy and internal energy) to balance the increase in kinetic energy. Yet experience showed that there was no difference in temperature perceptible (by touch) along a conveying line.

Some calculations were done to estimate the temperature change necessary in a flow to achieve such a balance, and the result indicated a fall of a degree or so, which would be unlikely to be picked up except through careful measurement. Even then, it was thought that there could well be some significant energy interchange by means of heat transfer between the gas and the solids and/or the pipe wall, which would mask the effect. The more important conclusion was that this avenue did not, at this stage, offer much hope as far as analytical modelling was concerned. The problem would be approached again at a later stage, from a slightly different direction as described further on in this Appendix.

D.3.2 Modelling of the acceleration of a single particle

A much more fruitful avenue proved to be the consideration of the process by which a particle is accelerated when an air stream flows over it. This began by looking at the usual model used for such a process, i.e.

$$\text{Drag force } F = C_D . \tfrac{1}{2} . \rho c^2 . A$$

where ρ is the air density, c is the relative velocity between particle and air, and C_D is a coefficient, the "coefficient of drag". A is the frontal area of the body.

The rate of transfer of energy by this process was considered, using the simple relationship

Power = Force x Speed (in same direction)

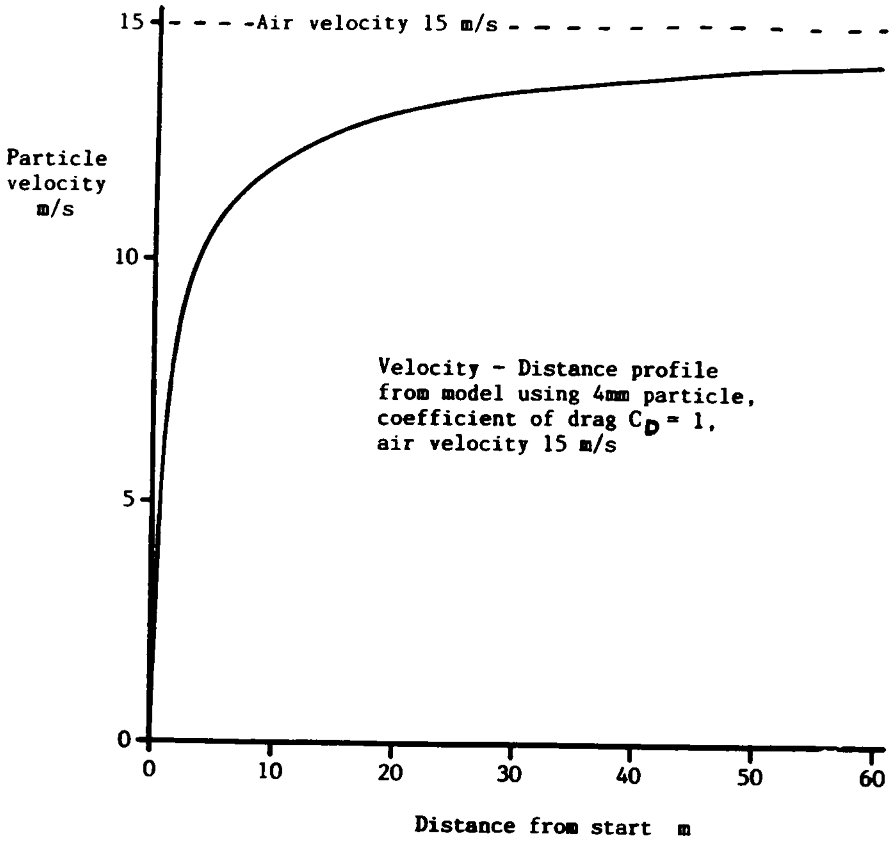
A mechanical power of drag x particle speed acts on the particle; the power taken from the air, however is equal to drag x air speed. Since the speed of the air is greater than that of the particle, the difference, a power of drag x relative velocity between particle and air, is expended in hèat through friction of the air over the particle and in its turbulent wake.

For example, when the particle is travelling very slowly, then the force acting on it is high but the mechanical power acting on it is low because of its low velocity. Thus of the power given up by the air, which is at its highest because the drag force is at its highest (air speed is constant), most of this is simply turned into heat. When particle velocity is higher, but still a good deal less than the air velocity, the mechanical power acting on the particle is high because the drag force is still fairly high but the particle velocity is quite high as well; the power taken from the air is somewhat less than when the particle velocity was low, because the drag force has reduced (air velocity is the same of course); so the power turned into heat is reduced. Once the particle has accelerated to nearly the velocity of the air, the power acting on the particle is low because although its velocity is high, the drag force is very low; the power taken from the air is low because the drag force is low; and the power turned into heat will also be small because the relative velocity is small.

A program was written, which performed repeated calculations to build up a finite difference simulation of this process. Taking a fixed value for the mass and coefficient of drag of a particle, and a fixed air speed, the particle speed was initially set at zero and the drag calculated; the acceleration of the particle was calculated from $F = ma$, and the increase in velocity and distance travelled in a time interval was found. The new particle velocity was used to find the reduced relative velocity, the drag recalculated, acceleration recalculated, increase in velocity and distance travelled in the next time interval found, and so on until the particle velocity converged within 5% of the air velocity. Various values of time interval were experimented with to ensure it was short enough (i.e. when shortening it further resulted in no further change in results).

This program was run many times to explore the effects of particle size, mass, coefficient of drag, air density and air velocity. As was expected, these all affected the rate of convergence between the velocities of particles and air. Taking the convergence to 5% as a yardstick, the distance was reduced in proportion to particle size (e.g. particles half the diameter reached 95% air velocity in half the distance), and reduced in inverse proportion to coefficient of drag and air density (e.g. doubling C_d halved the distance, as did doubling air density). The distance was unaffected by the air velocity. What was perhaps more interesting was that whatever the value of these, the cumulative energy given up by the air (time integral of power) always converged to just twice the cumulative energy acquired by the particle. By this token the energy dissipated as heat was equal to the kinetic energy acquired by the particle by the time the particle reached the velocity of the air.

Typical results from a run of this model are shown in Fig. D-1 overleaf:-



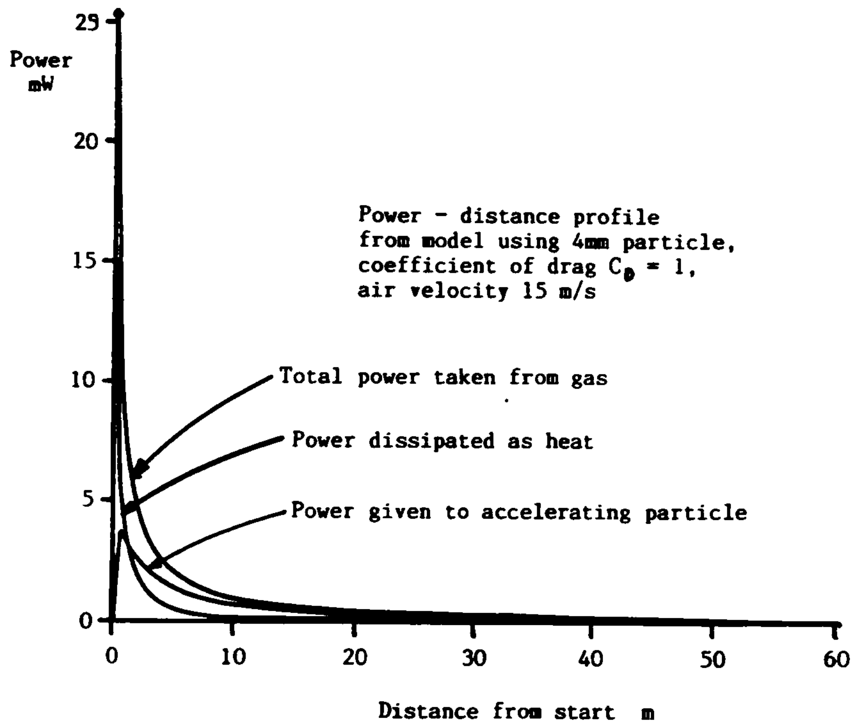


Fig. D-1
Velocity and power profiles from a run of the finite difference single-particle acceleration program

An interesting question arising from this work was as to the effect of the variation in coefficient of drag which would be expected as a result of the changing relative velocity between gas and air during the accelerating process.

It is generally recognised (e.g. ref. 105) that although the coefficient of drag for a smooth axi-symmetric body such as a sphere or spheroid is substantially constant over a range of Reynolds numbers of 1000 to 100 000 approximately, there is some increase as Reynolds number reduces to 100 and considerable increase in the coefficient with further reductions of Reynolds number below 100, approaching Stokes law ($C_d = 24/Re$) below Re = 10. Thus the model should account for the change in C_d with reducing Re during the acceleration process, to be strictly correct. The case of Re = 100 occurs, with a particle of 4mm as used to produce the above curves, at a relative velocity between particle and air of some 0.4m/s, very small in comparison with the air velocity of 15m/s (anything significantly less is not relevant to suspension flow conditions on which this model is based).

Consequently this increase in coefficient of drag is only relevant once the particles have been accelerated close to the air stream velocity anyway, by which time most of the work has been done and energy interchanges completed, so its effect upon values of either acceleration length or pressure drop could only be minimal. Therefore it was decided not to try to model this but to stick with a constant value of coefficient of drag to avoid unnecessary complication.

D.3.3 Use of the SFEE and a thermodynamic process model

Knowing now a little more about the mechanism of transfer of energy from air to particles, it was decided to attempt again the modelling of pressure loss. Since it had been seen that the only way to make sense of the process of energy exchange between air and particles was to allow temperature to vary, it seemed that this might form a basis for a model, whether it be correct or not.

The steady flow energy equation was examined again, as it might be applied to a flow in a horizontal pipe; past any two points in a pipe, provided there was no energy interchange between the flow and its surroundings, the sum of the flows of

Enthalpy + K.E. of solids + K.E. of air = constant,

enthalpy H being of course the sum of pressure energy and internal energy. It was apparent from a few calculations that the K.E. terms were small in comparison with the enthalpy term (taking the conventional datum of H = 0 at T = 0), so calculations would need to be carried out with care.

By considering the K.E. of solids to be zero at the start of acceleration, and calculating the final K.E. of both air and solids on the basis of the initial air velocity, it was possible to find the drop in enthalpy during acceleration of the particles. Knowing that $\Delta H = m.C_p.\Delta T$, the temperature drop could be calculated. However, it was apparent that there were many possible combinations of pressure and volume which would satisfy the new air temperature, as the following p-V diagram shows.

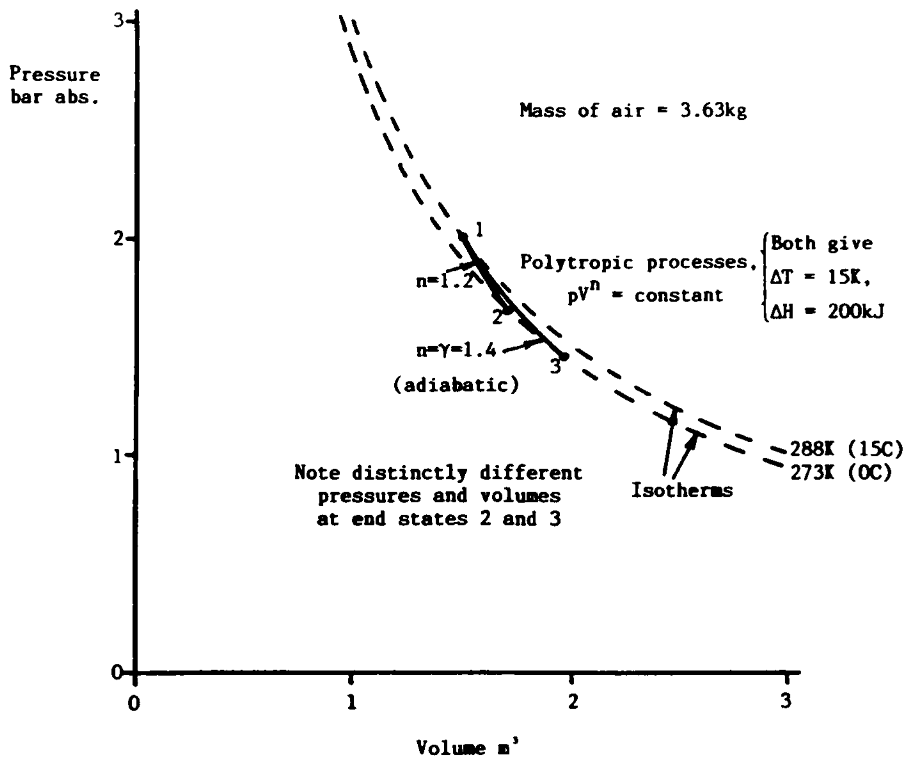


Fig. D-2

p-V diagram for air doing work with enthalpy reducing

In order to determine at what combination of p and V on the line of T the air would end up, it was apparent that a thermodynamic process model would have to be used. It had been an initial decision, to allow the use of the SFEE as written down above, to not allow any transfer of energy from the flow to the surroundings. Therefore the appropriate model must be the adiabatic one, pV^{γ} = constant.

A number of calculations were performed on the basis outlined, using a computer program, and it was apparent that the final velocity of the air after acceleration of the particles, based on the new pressure and temperature, was very little increased from the initial value, avoiding the need for iteration to obtain a value for the final K.E.

It emerged from the calculations that the pressure drop experienced by the air was equal to the dynamic pressure of the mixture of air and solids after acceleration. This seemed useful, because by this time the approach of representing the measured bend pressure loss data as a function of the dynamic pressure of the suspension had formed in the mind of the author, as a result of correlating experimental data. This meant that if the loss was written down as

$$\Delta p = K.\tfrac{1}{2}.\rho_s c^2$$

where Δp is the pressure loss,
ρ_s is the density of the suspension calculated on the basis of the mass of solids flowing, the pipe area and the air velocity (ignoring the mass of air, normally small in relation to the mass of the solids), and
c is the velocity of the air

and the value of the coefficient K worked out, the modelling gave a value for acceleration pressure loss of 1; experimental results of bend pressure loss gave K values of about 0.5 to 1.5.

It was borne in mind whilst using this model, that it was based on the energy given up by the gas being equal to the energy gained by the particles. This was known not to be the case, since the single-particle acceleration model showed that the energy given up by the gas would be equal to twice the energy gained by the particle. An attempt was made to use a polytropic expansion model in place of the adiabatic model in the program mentioned above, which would allow for there to be energy interchange by means of heat transfer as well as work transfer. Unfortunately this proved very difficult to achieve owing to difficulties in deciding where transferred heat should go, accommodating this in the SFEE, and determining a suitable value for the polytropic index to allow the work transfer from the gas to be equal to twice the final energy of the particles with the heat transfer to be equal to the difference (in order to conform to the findings from the single-particle acceleration model).

D.3.4 Modelling of a multi-particle system

At length, the line of attack was abandoned and a new one sought. It was decided that there may be some merit in trying to use the finite-difference approach (as used for the single-particle acceleration model) in combination with the adiabatic process model to calculate the pressure loss profile caused by the acceleration of a number of particles in a pipe.

The first step was to re-work the original finite-difference program to work on a distance interval basis as opposed to the original time interval basis. With this successfully achieved and giving results the same as the original program, it was extended to calculate work transfer to each particle in each element by air drag. To find total rate of work transfer in each element with this now considered as a continuous process, it was necessary to know the number of particles in each element. This of course decreases further away from the point where the particles are injected, because the particles become spaced out as they accelerate. This was easily calculated for given flow rates of solids and air and pipe sizes, so leading to the power being consumed in each element and consequently the pressure drop in each element. This pressure drop became smaller and smaller in successive elements as the particle velocity converged on the air velocity, with the resulting total pressure loss converging on a value equal to twice the dynamic pressure of the final fully-accelerated suspension.

It was hardly surprising, on reflection, that to take account of the work done against friction resulted in a pressure loss twice that resulting from not taking account of it, in the light of the results from the original single-particle acceleration program.

The question of effect of concentration upon coefficient of drag was examined to see whether this would affect the modelling; it is well documented that at high concentrations of solids in a flow space the effective coefficients of drag on the particles increases markedly. For example, Wen & Yu (ref. 58) give an equation which modifies the 'free stream' drag coefficients of particles under conditions of high concentration as follows:-

$$C_d' = C_d . \xi^{-4.7}$$

Where C_d' = coefficient of drag in a free stream
C_d = coefficient of drag corrected for particles in a concentration
ξ = voidage

The conditions in the pipe for the models were examined, and it was found that the concentrations of particles in the air in individual elements were such that the drag coefficients were virtually unmodified using the above equation. This is perhaps hardly surprising since the work of Wen & Yu was in packed and fluidised beds, wherein the particle concentrations are considerably higher than in pneumatic conveying pipelines even in entrainment regions. For these reasons, the effect of particle concentration on drag coefficients was ignored.

D.4 Conclusions

The work which had been done suggested that the pressure losses caused by the acceleration of particles either after injection into the pipe or after being slowed down by collision with the pipe wall in a bend, may well be of the order of one or two times the dynamic pressure of the flowing suspension, calculated with the particles flowing at the same speed as the air; in other words, loss coefficients of the order of 1 or 2 would be expected with the pressure loss expressed as described in D.3.3 above.

It was recognised that a model could be developed to analyse the amount of slowing down experienced by particles in colliding with the wall given values for such things as a coefficient of friction between particles and wall, but the effects of the boundary layer of air causing a difference in relative velocity between particle and air across the particle, meant that there was no way in which an accurate model could be developed so it would not offer any advantages to do so.

Therefore it was apparent that as much as could reasonably be learnt from this exercise had been obtained and there was little point in carrying it further.

APPENDIX E

ATTEMPTS AT APPLYING DIMENSIONAL ANALYSIS TO THE PROBLEM

E.1 Introduction

It was thought from an early stage that the approach of using dimensional analysis may assist in the search for a data storage system. Some time was spent examining the methods of dimensional analysis and the problems to which it was normally applied in textbooks. The most common application appeared to be the dimensional analysis of the quantities involved with the performance of a centrifugal pump. In this exercise the quantities would be listed and the Pi method used to obtain the dimensionless groups usually used to scale the test results taken from a pump to predict the performance of the pump at another speed, or the performance of another similar pump of different size, using the principle of dynamic similarity.

This exercise was tried, and it was noted that if a different set of repeating variables was chosen for the analysis other than those chosen in the textbook analysis, a different set of dimensionless groups would result; yet from experience of testing a pump on a rig at different speeds, it was known that the usually-accepted set of dimensionless groups were the correct ones to use for scaling because they gave quite good results when scaling, at least for speed. No reference was found in any of the books as to the effect of choosing the 'wrong' repeaters, or what the meaning of this was. This seemed to indicate that the correct dimensionless groups for scaling of pump performance had most likely been established by means other than dimensional analysis, and that this was used as a sort of tenuous 'justification' for something which had probably come from empiricism.

It was thought that perhaps the effective use of dimensional analysis may be dependent upon a fundamental understanding of the processes involved in the phenomenon to which it is being applied. As related in Appendix D, such a fundamental understanding had proved elusive to the author, so on this basis the only hope for dimensional analysis would be to resort to trial.

This did not lead to a great deal of confidence in the value of dimensional analysis, but it was apparent that by running analyses using different combinations of repeating variables, it may be possible to obtain some indications of how a set of quantities might be fitted together in equations in ways which would be dimensionally homogeneous, even though no indication could be gained as to the usefulness of these equations other than through empirical methods, which would clearly be necessary in any case.

E.2 The analysis

E.2.1 Quantities

All quantities that could possibly be considered to be related to the pressure drop caused by a bend were listed, which were:-

Quantity	Symbol	Units	Dimensions
Superficial air velocity	c	m/s	LT^{-1}
Air density	ρ_a	kg/m^3	ML^{-3}
Suspension density	ρ_s	kg/m^3	ML^{-3}
Bulk or particle density of product	ρ_b	kg/m^3	ML^{-3}
Pipe bore	d	m	L
Mass flow rate of product	$\dot{m}_p$	kg/s	MT^{-1}
Mass flow rate of air	$\dot{m}_a$	kg/s	MT^{-1}
Bend radius	r	m	L
Pressure drop	Δp	N/m^2	$ML^{-1}T^{-2}$

It was clear that many of these were tied up together, for example the mass flow rate of air being related to the air density, velocity and pipe bore as

$$\dot{m}_a = \frac{\pi d^2 c \rho_a}{4}$$

In the same way the mass flow rate of product would also be related to the velocity and suspension density, and so on. In order to make progress, it was decided to ignore this fact, and the fact that some of the quantities would form obvious dimensionless groups (because they have the same dimensions), and simply use the whole lot in an analysis to see what would emerge.

E.2.2 First attempt at analysis

Using the Pi method, these nine quantities with three dimensions would be expected to yield six groups. The selection of repeating variables for the analysis was based on the usual advice of choosing one property of the fluid (ρ_b was chosen from ρ_a, ρ_s, and ρ_b), one property of the flow (c was chosen from c, $\dot{m}_a$, $\dot{m}_p$, Δp) and one geometric property (d was chosen, the alternative being r).

The outcome of the analysis were the following pi groups which were written down in a function:-

$$\frac{\Delta p}{\rho_s c^2} = \text{function}\left(\frac{\rho_s}{\rho_b}, \frac{\dot{m}_a}{\rho_b c d^2}, \frac{\dot{m}_p}{\rho_b c d^2}, \frac{\rho_a}{\rho_b}, \frac{r}{d}\right)$$

It was noted that since the mass flow rate of air was a function of air density, pipe bore and superficial air velocity, and likewise the mass flow rate of product was a function of suspension density, pipe bore and superficial air velocity (suspension density being calculated from these others), then a substitution could be made to reduce the number of variables.

Since $\dot{m}_a = \dfrac{\rho_a \pi d^2 c}{4}$, so $\dfrac{\dot{m}_a}{\rho_b c d^2} = \dfrac{\rho_a \pi d^2 c}{4 \rho_b c d^2} = \dfrac{\pi}{4}\dfrac{\rho_a}{\rho_b}$

and likewise $\dot{m}_p = \dfrac{\rho_s \pi d^2 c}{4}$, so $\dfrac{\dot{m}_p}{\rho_b c d^2} = \dfrac{\rho_s \pi d^2 c}{4 \rho_b c d^2} = \dfrac{\pi}{4}\dfrac{\rho_s}{\rho_b}$

From this it was evident that the $\frac{\dot{m}}{\rho c d^2}$ groups were the same as the $\frac{\rho}{\rho_b}$ groups so that to include both was unnecessary. The function therefore reduced to

$$\frac{\Delta p}{\rho_s c^2} = \text{function}\left(\frac{\rho_s}{\rho_b}, \frac{\rho_a}{\rho_b}, \frac{r}{d}\right).$$

At this stage, the possibility of representing the pressure drop as a constant (K) times the notional 'dynamic pressure' of the flowing suspension had already been considered, so the function was re-written as:-

$$\Delta p = K.\frac{1}{2}\rho_s c^2 \text{ function}\left(\frac{\rho_s}{\rho_b}, \frac{\rho_a}{\rho_b}, \frac{r}{d}\right)$$

The effect of the air density was already known, from experimental work, to be negligible, so the group $\frac{\rho_a}{\rho_b}$ was discarded, leaving

$$\Delta p = K.\frac{1}{2}\rho_s c^2 \text{ function}\left(\frac{\rho_s}{\rho_b}, \frac{r}{d}\right).$$

It was expected from the outset that the ratio of bend radius to tube diameter would affect the pressure drop to be measured (and hence the value of K), and in fact this had already been confirmed by experimental work. The meaning of the other group, the ratio of the suspension and bulk densities, was equally straightforward - the suspension density had been found to affect the value of K, and dividing it by the bulk density would simply render it dimensionless, in fact a volumetric solids loading ratio or volume fraction.

E.2.3 Second attempt

However, it was known from experiment that the superficial air velocity (c) also affected the value of K, yet it did not turn up in the outcome of the analysis. Therefore it was decided to repeat the analysis with the aim of having c incorporated in the result.

The quantities air density and mass flow rate of air were abandoned, because it had already been found from experiment that they were of no consequence - the value of K was independent of these for given values of air velocity. Likewise the quantity mass flow rate of product was dispensed with since it was accounted for in terms of pipe diameter, suspension density and superficial air velocity (as described above).

This left two fluid properties, ρ_s and ρ_b, from which ρ_s was chosen,
two flow properties, c and Δp, from which Δp was chosen, and
two geometry properties, r and d, from which r was chosen to be
repeating variables. Three dimensionless groups were expected.

The analysis gave three of the same groups as the first analysis, thus resulting in the same

$$\Delta p = K.\frac{1}{2}\rho_s c^2 \text{ function } \left(\frac{\rho_s}{\rho_b}, \frac{r}{d}\right).$$

It was apparent that if d had been chosen instead of r as a repeating variable, then the result would have been no different.

E.2.4 Introduction of 'g' or other extra quantities

It was decided out of interest to try using the acceleration due to gravity, g, in the analysis to see what would happen.

The same three groups as listed above resulted, but with the addition of a fourth, $\frac{\rho_s rg}{\Delta p}$. This did not seem particularly useful, so another attempt was made using d instead of r as a repeater, which yielded $\frac{\rho_s dg}{\Delta p}$. Neither of these appeared very useful, because in combination with the other groups they gave two groups incorporating Δp, the pressure drop which was the object of the whole exercise - which would make it difficult to obtain an explicit function in this quantity.

It was apparent that if Δp was not used as one of the repeating variables, then this problem would not arise. Accordingly, the analysis was undertaken again, using the superficial air velocity c as the flow property instead of p. the resulting four groups were

$$\frac{\rho_s}{\rho_b}, \frac{\Delta p}{\rho_s c^2}, \text{ and } \frac{r}{d} \text{ as before plus } \frac{dg}{c^2}$$

It was apparent that the new group could be written as $\frac{c}{\sqrt{gd}}$ in which form it looked like a Froude number. It had been observed that other authors, in papers dealing with dimensional analyses of this problem, had obtained what they referred to as Froude numbers, but some scepticism had been applied to the significance of these since the original Froude number had been derived and used for ship resistance, where it would appear that quite different mechanisms were at work. It should be said that this had been acknowledged by at least one of these authors (ref. 37) who consequently referred to this quantity as the "dimensionless criterion of the Froude form" to emphasise this.

It was equally apparent that if instead of introducing g, a viscosity had been introduced (whether the viscosity of the air or a 'pseudo-viscosity' for the flow), then instead of this Froude-type group, a Reynolds number would have appeared, also containing velocity of course.

This seemed to beg the question as to whether g or viscosity would be significant variables - in other words, whether the incorporation of them would add anything useful to the final result. It was considered whether alteration of the value of the acceleration due to gravity (as distinct from altering the direction of gravity - i.e. the orientation of the bend in the gravitational field) would affect the pressure drop. The conclusion was that it would not have a very great effect, certainly far less than that of changes in the velocity c. It was observed, though, that g does not alter to any extent in the range of systems normally considered, and if the use of it helped in predicting the pressure loss, then that would surely be justification enough.

The value of the pipe diameter, d, on the other hand was very much a real variable, 2in., 3in., and 4in. pipes being used in the experiments. The value of the loss coefficient K was known to be dependent on velocity c, so it was considered whether it may be better to plot graphs of K versus $c/\sqrt{gd}$ instead of simply against c. This would clearly not affect the relationship between the K curves for different bends of the same pipe diameter, but would affect the relationship between them for different pipe diameters. It was known, though, that for the radiused bends, the graphs of K were practically the same for all pipe diameters, so to start involving the pipe diameter would serve to complicate this useful relationship with no obvious return.

At this stage, the work done was discussed with colleagues, and it was suggested that perhaps another characteristic length apart from the pipe diameter might be found for use in the Froude-type group. The length which sprung to mind was the acceleration length of the particles, say for example from rest to 95% of the air velocity. This would of course be dependent on the particle size, shape, density and no doubt many other factors; approximations could possibly be made using the simple model of particle acceleration which had been developed for prediction of pressure loss by mathematical means, described in Appendix D. However it was apparent that this would be of little use in trying to correlate the pressure drop for one product with varying flow conditions, because the value of the length would be dependent on the product only, and would not vary with flow conditions.

By the same token the characteristic acceleration could be replaced with an acceleration other than g, perhaps the centripetal acceleration of particles travelling at the superficial air velocity around the bend. This was considered; if r was the bend radius, a the acceleration and c the superficial air velocity then $a = \frac{c^2}{r}$; this gives increasing acceleration with reducing bend radius, which might be correlated with increased pressure loss with small values of r (e.g. less than five pipe diameters), but causes a difficulty when r reduces to zero as with the blind tee bend because acceleration increases to infinity. Therefore this did not appear very rewarding either.

E.3 Conclusion

By this time a considerable amount of time and effort had been directed at the use of dimensional analysis, and there appeared to be no rewards forthcoming, so it was decided that the time could better be directed towards more fruitful avenues of work and it was abandoned.

Finally it was noted in passing that this experience appeared to be in line with other workers who had attempted to use dimensional analysis in connection with the prediction of pressure drop along pneumatic conveying pipelines. Examining for example the work of Rose and Duckworth from 1969 (ref. 39), the authors attempted to apply this approach to one of the simplest systems, consisting of the suspension flow of spherical particles along a straight horizontal pipeline. They finished up with a plethora of dimensionless groups whose significance and application appeared hard to understand, and the application of which appeared even harder to grasp, even after extended study; the authors themselves offered no help in this respect and no further publications ever appear to have been made by them even though Duckworth was still known to be pursuing this line of research some twenty years later. This, together with the fact that no other authors are known to have published any furtherance of this work or indeed anything else of significance in this field in the intervening time, tends to reinforce this author's view that there are far more rewarding avenues of approach for this problem, and that dimensional analysis, and the associated technique of dynamic similarity, are best reserved for fields wherein they undoubtedly have a contribution to make, for example the scaling of performance of water turbines, centrifugal pumps, and ships.

APPENDIX F

DEVELOPMENT OF THE TEST RIG

The evolution of the thinking which shaped the test rig, and the essential features of the rig and its associated instrumentation, have been described in sections 2.7 and 2.8 of Chapter 2. This Appendix is concerned with the details of the various items of equipment used and the way in which they operated.

The complete equipment may conveniently be broken down into four parts, namely the air supply, the conveying plant (feeder and separator), the conveying pipeline and bends, and the instrumentation and control systems. Much of the air supply was already installed in the laboratory when the project was started, and there was a conveying plant on an industrial scale available which was thought to be suitable. Barring these, the pipeline was the main issue to be considered because a particular set-up would be needed to create the conditions which it was hoped to measure. The instrumentation would follow on from the quantities to be measured.

F.1 The pipeline and bends

F.1.1 Pipeline loops

In order to determine the loss caused by a bend, and the pressure losses along straight pipe, it was seen to be necessary to measure the pressure profiles in two straight sections of conveying line with a bend in between, as described in section 2.7. For the reasons also described, the two straight pipes would be made as long as possible within the laboratory. The return of the solids to the conveying plant would necessitate the pipeline completing a full circuit, the resulting loop being as shown in fig. F-1 below.

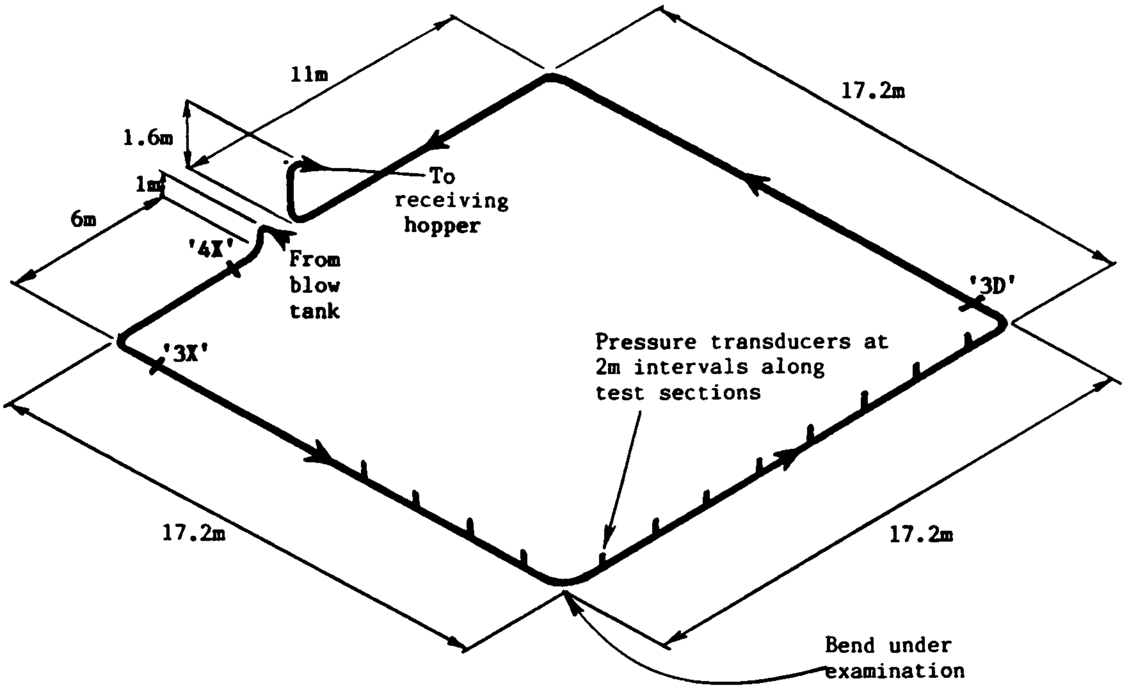


Fig. F-1

The conveying pipeline loop

This pipeline was initially built from 2 inch nominal bore medium weight seam welded steel pipe (bore 53mm diameter), being subsequently rebuilt in 3in. n.b. (81 mm bore) and 4in. n.b. (104mm bore) pipe. The exit pipe from the feeder was fixed at 2in., so the use of the larger bore pipelines necessitated an enlargement; the step from 2 to 3 inch was located at point "3X" on the diagram, and found to be satisfactory. The enlargement from 2 to 4 inch was initially located at the same place, but this was found to affect the measurements (as described in section I.13 of Appendix I) so it was moved forward to point "4X", which was satisfactory.

Two further variations on the layout were used, to achieve some control over the air density in the test sections. Both retained 2in. test sections. The first consisted of the addition of another 2in. loop of 58.5m (including 16m vertically down and the same vertically up, and 9 bends) to the end of the line, in order to increase pressure (and hence air density) in the test sections; the additional loop is shown in fig. F-2 below. The second variation was to enlarge the pipe downstream of the test sections from 2in. to 3in., to reduce pressure and air density in the test sections. The enlargement was located at point "3D" in fig. F-1 above.

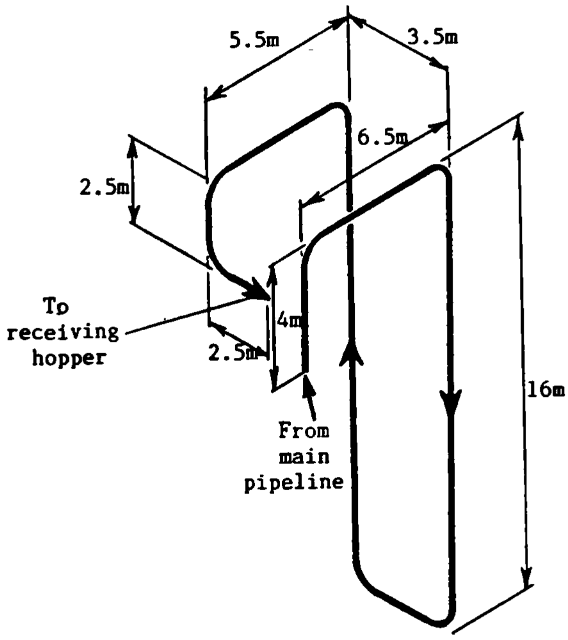


Fig. F-2
The additional 2in. loop used at the end of the standard 2in. pipeline to extend it

F.1.2 Bends

A considerable amount of thought was given to the selection of bends for the test programme, as outlined below.

It became apparent, from looking at a number of commercial pneumatic conveying systems, that there is a very wide range of bend types specified by different manufacturers. The most common type is a radiused bend, made from tube by bending, with a relatively long radius compared to bends used for fluid pipelines, typically of the order of 1.5m for a 4in. pipe. Clearly a bend of this type would have to be tested. Such bends are relatively expensive because they are normally manufactured to order and not part of the usual run of the mill; therefore it was decided that it would be useful to test a bend of similar construction (bent from tube) but shorter radius, nearer to the type available cheaply off the shelf from most pipe fitting stockists, to ascertain whether the extra cost of the special long radius type is justified by lower pressure drop.

The means by which a bend is fitted into a pipeline was perceived to be possibly significant; in most commercial systems, either flanges or sleeve couplings (e.g. "Morris" or "Mucon" couplings) are used, which may or may not result in some gap between the ends of the pipes inside the joint, depending on the care taken during installation. In the Thames Polytechnic laboratories, screwed unions ("Crane" or "GF" type) are more commonly used, which result in a consistent gap of some 10mm between the pipe ends. To measure the significance of this, both the long and short radius bends would be tested with unions and without. In order to allow interchangeability of the bends without moving the straight sections, it was decided that all bends would be made or fitted with integral straight pieces of suitable length (see fig. F-3 below), to be joined to the adjacent straight pipes using "Morris" sleeve couplings which would allow perfect alignment and no gap between the pipe ends if installed carefully. The bends with unions would simply have the unions installed in the integral straight lengths.

The four bends mentioned above, i.e. long radius with and without unions, short radius with and without unions, were all made in house from a single 6m length of 2in. pipe to ensure no variation of bore size. The short radius ones were each bent in one operation on the former of an hydraulic pipe bender, giving a radius of 290mm to centre line; the long radius one was bent in stages moving the pipe bender between marked points on the pipe in turn, which actually gave a remarkably smooth finished bend of radius 711mm to centre line. Unions were subsequently fitted to one of each, cutting the pipe as close to the bend as the threading machine would allow, i.e. about 80mm from the end of the sweep.

A radiused bend of the cheap, commercially available type mentioned above (radius 165mm to centre line) was also obtained for testing, for comparison. This was fitted using screwed sockets, to be representative of the most economical form of industrial practice. To take the investigation of bend radius to its logical extreme, malleable elbow fittings, both male and female, were also obtained for tests. These both had similar radii of about 75mm to centre line, but the female version had an inside diameter through it, between the ends of the pipes screwed into it, about equal to the outside diameter of the pipe (i.e. much larger than the bore of the pipe) whereas the male one had a bore slightly smaller than the bore of the pipe.

The question of wear resistant bends had to be considered; although the investigation of wear of pipeline bends as such was outside the scope of the project, it was recognised that there are many cases of real systems conveying abrasive products, where special bends are used to combat the problem. Sometimes the bends used are simply radiused bends lined with ceramic or mineral materials, which it was thought would not exhibit significantly different pressure losses from standard bends; on other occasions bends of special geometry are used. One example of this case is the blind tee, where the flow enters along the run of a tee, the other end of which is capped, and leaves through the branch. Such bends had been shown by tests in the Thames laboratories to be very good at withstanding wear, having lives many times those of standard bends, but were an unknown quantity as far as pressure loss was concerned. Therefore it was decided that a blind tee must be tested. This was made in house by cutting and welding stock pipe, to reflect usual industrial practice. Another example of a wear resistant bend then recently introduced but finding increasing use is the Hammertek "Vortice-ell" bend, a commercial cast bend whose then distributors were claiming very low pressure loss as well as extended life; an example was obtained for testing.

All of the above mentioned bends were in 2in. nominal bore, decided upon as the smallest size likely to give useful results as explained in section 2.7 of Chapter 2. Time would not allow for every bend to be tested in 3in. and 4in. pipe sizes as well, so only the long radius and the female malleable elbow were tested in 3in., with testing in 4in. restricted to the long radius bend alone. The radii of the long radius bends were 660mm for the 3in. and the 4in. Since the results of testing for the effects of unions with the 2in. bends had not shown the means by which the bends were connected to be of any great significance, the 3in. bends were connected using screwed unions and the 4in. one using stock flanges for convenience. Drawings and a summary of the bends used appear on the next two pages.

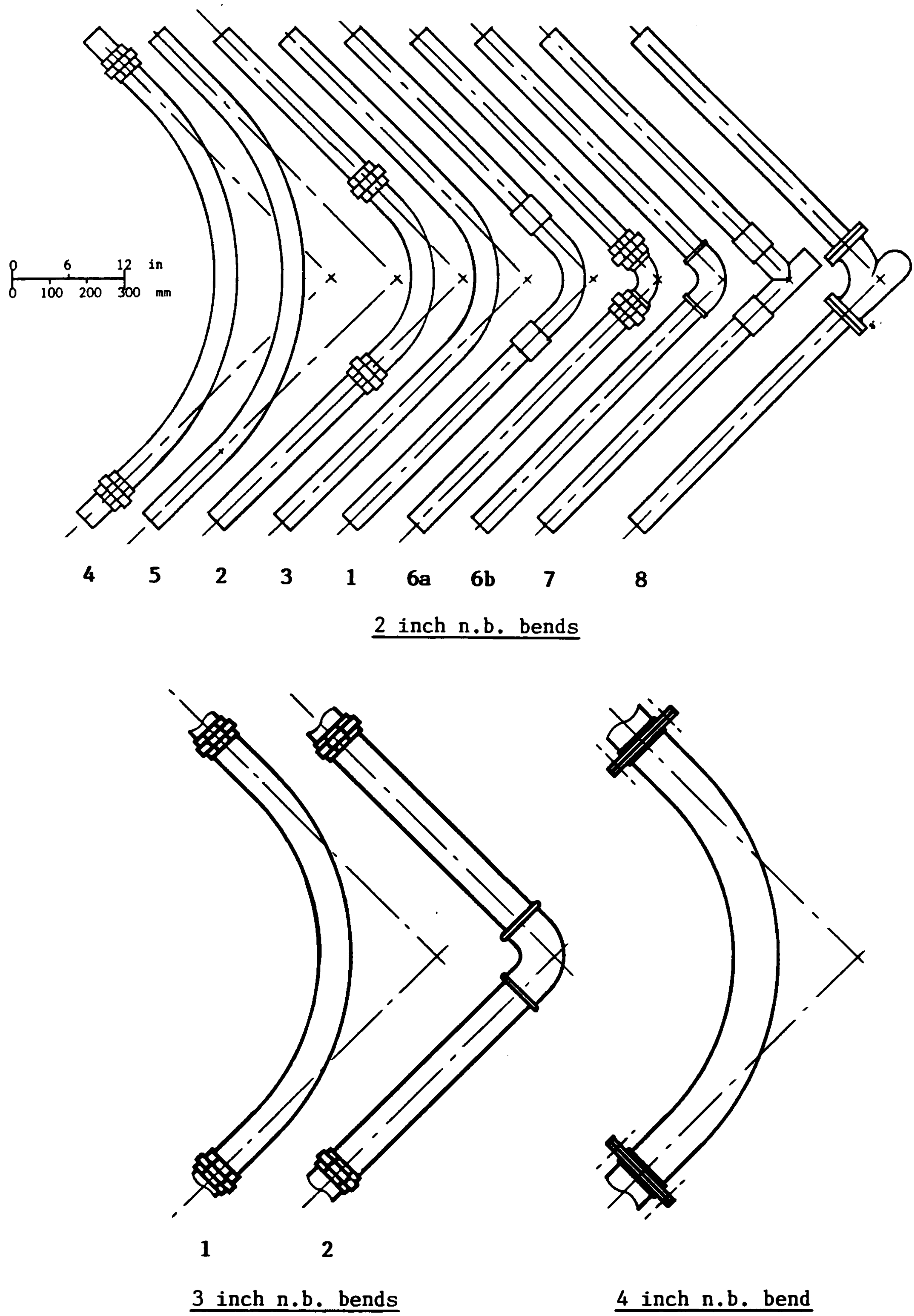


Fig. F-3 The bends tested

The 2 inch nominal bore (53mm bore) test bends:

Numbers tie up with graph of relative loss coeff. vs. r/d ratio, fig. 3.3.

Bend No.	Description	Ratio $\frac{\text{Bend radius}}{\text{pipe bore}}$
1	Short radius bought-out, with sockets	3.1
2	Short radius with unions	5.4
3	Short radius without unions	5.4
4	Long radius with unions	13.1
5	Long radius without unions	13.9
6a	Male malleable elbow with unions	2.2
6b	Female malleable elbow	2.3
7	Blind tee	0
8	Vortice-ell	-

All 2in. bends as shown were connected to adjacent straight lengths using 'Morris' sleeve couplings for internal smoothness of joints.

The 3 inch nominal bore (81mm bore) test bends:

1	Long radius	8.1
2	Female malleable elbow	1.8

All 3in. bends were joined to the straight lengths using screwed unions.

The 4 inch nominal bore (104mm bore) test bend:

Long radius	6.3

The 4in. bend was joined to the straight lengths using bolting flanges.

Fig. F-3 (continued)
The bends tested

F.2 The conveying plant

The ability to feed the pipeline with the required flow rate of air and solids over as wide a range as possible, in order to generate data over the widest range possible, was seen as highly desirable. High flow rates of product are only achievable with high pipeline pressure drop, so it was necessary to use a feeding device which would be capable of operating at high pressure - experience with running a rig showed that a capability to operate at up to at least 3 bar gauge was desirable. Of the common methods of feeding pneumatic conveying pipelines, namely venturi devices (commonly called "eductors"), rotary valves, and blow tanks, only the latter are capable of achieving this.

A blow tank consists of a pressure vessel into which the product to be conveyed is loaded, which is then pressurised in order to drive the product out into the conveying line, where more air is normally injected to achieve the desired ratio of product to air. A blow tank of about 1.5 m capacity, with a pressure rating of 7 bar, discharging through a 2in. pipe was available in the laboratory. Above this was mounted, on load cells, a hopper of similar capacity, with an air filter on top. The blow tank could be charged from the hopper through two 8in. butterfly valves, one of which served to seal the pressure in the blow tank whilst the other served to retain product in the hopper. Between the two was fitted a flexible hose, so that the weight of the hopper was decoupled. Then during conveying from the blow tank around a loop to the hopper, the flow rate of solids could be determined by monitoring the gain in weight in the hopper over a known time period.

The blow tank was also fitted with a valved vent line to enable air to escape into the hopper during charging and depressurising. This also was fitted with a flexible section to ensure weight decoupling, as was the entry point where the conveying line returned to the hopper.

Some trials using the blow tank showed that a fair proportion of the time of the operator was taken up in turning the conveying plant around between runs, which involved turning off the air supply, opening the vent line valve to depressurise the tank, opening the two large butterfly valves and applying a hammer to the hopper to persuade the product to flow (depending on the product), closing the butterfly valves (which required considerable force), then turning the air supply on and closing the vent line valve to initiate conveying. Since the valves controlling the air supply were situated some distance from the valves on the tank, which were accessed from an elevated platform, this was not a convenient procedure. Therefore it was decided to investigate the possibility of controlling these valves remotely and installing a flow promoter, in the shape of a ring of air injectors, on the hopper.

The control systems are described a little further on, but essentially the modifications to the plant involved fitting pneumatic actuators to the two large butterfly valves (actuators of the next size up from the ones usually specified for the 8in. valves were used, in view of the fact that the valves were handling solids); fitting a pneumatically actuated diaphragm valve to the vent line in place of the manual one (a diaphragm valve was thought to be most suitable in view of the fact that this valve could be releasing air contaminated with entrained solids at some pressure; a failsafe open actuator was used in order that under conditions of loss of power, pressure would be vented); and finally the attachment of a ring of four air injection points around the hopper, connected to a solenoid valve. All pneumatic actuators were in turn controlled by solenoid valves.

A diagram and a picture of the plant as finally used appear in figs. F-4 and F-5.

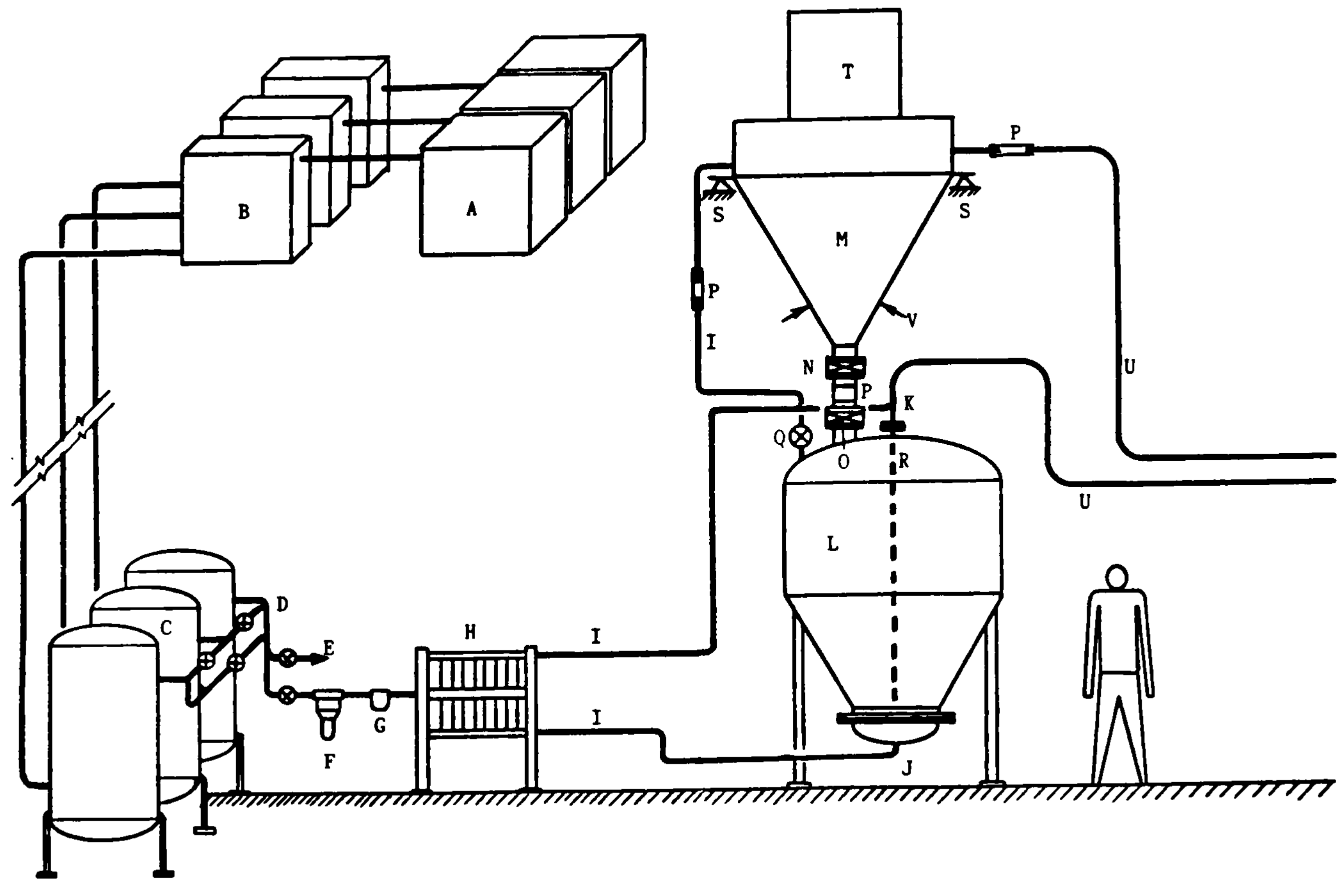


Fig. F-4

Diagram of conveying plant and air supply

Key to code letters

A Compressors } In enclosure
B Aftercoolers } on roof
C Air receivers
D Manifold
E Air takeoff to pneumatic control equipment
F Filter / water trap
G Pressure regulator
H Choked flow nozzle bank (see fig. F-6)
I 2in. nominal bore pipe
J Blow tank air injection point
K Supplementary air injection point
L Blow tank
M Receiving hopper
N Upper charging valve 8in. (actuated butterfly valve)
O Lower charging valve 8in. (as N)
P Rubber sleeve for weight decoupling
Q Blow tank vent valve 2in. (actuated diaphragm valve)
R Blow tank discharge line (2in. nominal bore)
S Load cells
T Exhaust air filter
U Conveying line
V Air injection to aid discharge from hopper

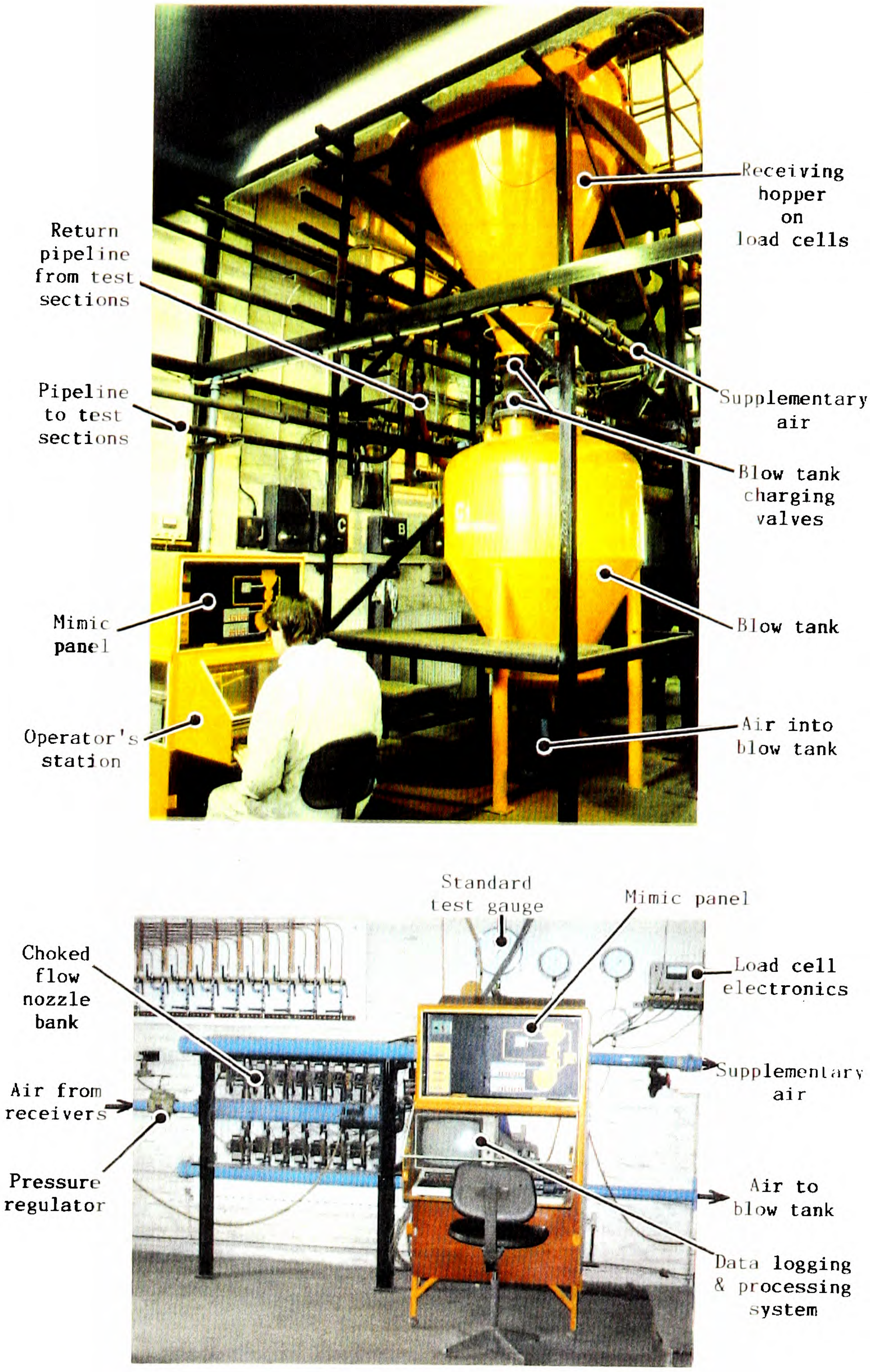


Fig. F-5

Photograph of the conveying plant and operator's station

F.3 The air supply

The essential elements of the air supply in the laboratory were three Broom & Wade V200DA oil free reciprocating air compressors giving a total capacity of 600 cfm free air (0.33 kg/s) at 7 bar, feeding three receivers. From there the air passed through a manifold to two valves which turned the air to the blow tank and supplementary air injection points on or off. The actual air flow rates were determined by the size of a choked flow nozzle after the valves, which for a fixed upstream pressure gave a constant known flow rate irrespective of downstream pressure up to a pressure ratio of about 80%. To obtain different air velocities and solids loadings in the pipeline required the air flow rates to be varied from run to run, which involved disconnecting four unions and two flanges on the 2in. pipework to access the nozzles for changing. This was a most time consuming procedure, so it was decided to re-work the set-up to incorporate a range of nozzle sizes each controlled by a valve.

F.3.1 The choked flow nozzle bank

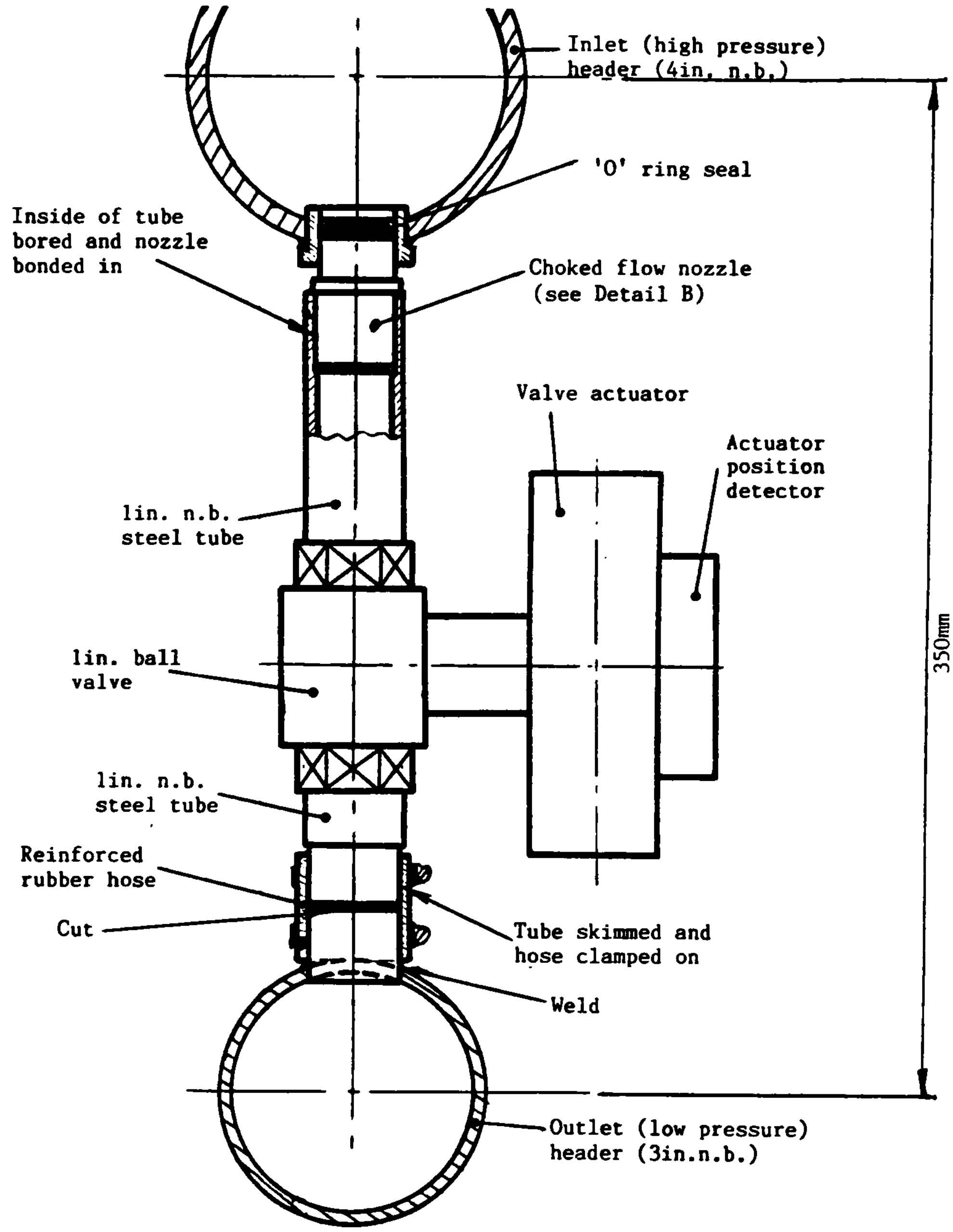
Thus the idea of two banks of choked flow nozzles (one for blow tank air, one for supplementary air) was born. It was apparent that if the nozzles were in a "times two" geometric progression, then they could be used together like bits in a binary number to make up any chosen flow rate up to the maximum with them all open, with a resolution equal to the flow rate of the smallest nozzle. The largest possible flow rate would be the output of all three compressors; if this was passed with all nozzles open, the largest nozzle would have to flow half of this, i.e. .165 kg/s. The smallest air flow rate required was taken to be .0015 kg/s, since this low a flow rate had on occasion been used on another project looking at low velocity conveying. Eight nozzles would be required to give such a range in a binary progression ($2^8 = 256$), the lowest being in fact .0013 kg/s if the highest was to be .33 kg/s. It was decided to design for these flow rates with an upstream pressure of 4 bar gauge, because it had been mooted that two of these air flow control systems should be built, the second to be used on another rig which was rated at only 4 bar.

The nozzles were to be made with a convergent-parallel-divergent section since such a section had given a good critical pressure ratio (typically 80%+) on the original nozzles, as opposed to the 70% or so obtained from a set of parallel-divergent nozzles which had been used on another rig. The throat areas of the nozzles were sized on the equation given in ref. 106. The required throat diameters were from 1.19 to 13.45mm, and bearing in mind the requirement to bore a radiused convergent entry section and general manufacturing tolerances, it was clear that they would need to be calibrated after manufacture to measure actual flow rates as well as critical pressure ratios. Sections of the nozzles are shown in fig. F-6.

The effect of variation in relevant quantities was examined, the most important being throat diameter (flow rate proportional to area - to be calibrated) and upstream pressure (flow directly proportional to absolute pressure); natural variations in atmospheric pressure would cause negligible change in flow rate, the gauge pressure of the air being several times atmospheric; and change in air temperature (small in terms of absolute temperature) would cause very little change in flow rate (e.g. 5K temperature rise leading to just 1.7% reduction in flow rate).

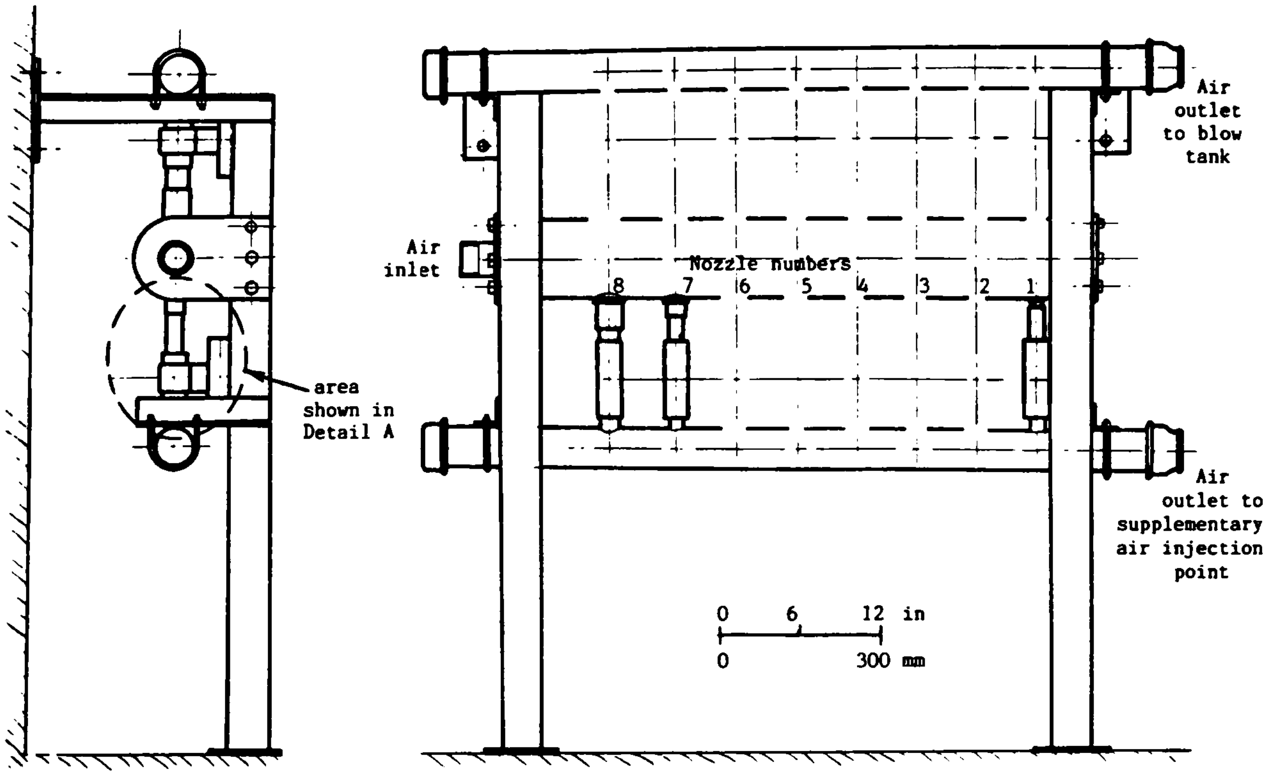
After some consideration, the design shown in fig. F-6 was settled on. This has one central reservoir (the "high pressure header") to which air is fed, the nozzles having their convergent inlets fitting directly into this. The two banks of nozzles are located both above (supplementary air) and below (blow tank air) the high pressure header. From each nozzle the air passes along a short pipe to an on-off valve; having this located after the nozzle means that any slight pressure loss here does not affect the flow rate, although the valves are in any case of the ball type which give a full bore through when open as well as (more importantly) drop-tight shut-off. The tube sizes were chosen to give negligible pressure loss, the no.8 (largest) nozzle using 1 1/4 in. n.b. tube whilst the no.7 used 1 in.; smaller ones were kept at 1 in. also for standardisation, there being no cost saving on actuated ball valves less than 1 in.

After passing through the valves, the air enters the low pressure header, which connects to the pipe to the test rig. The nozzles fit into machined bosses in the high pressure header, sealed by 'O' rings, and the pipes from the valves to the low pressure header incorporate a discontinuity, covered by a length of flexible pressure hose; these measures accommodate any welding distortion as well as thermal movement of parts.



Detail A
(relates to fig. F-6 overleaf)
Layout of valves and nozzles
between inlet and outlet headers

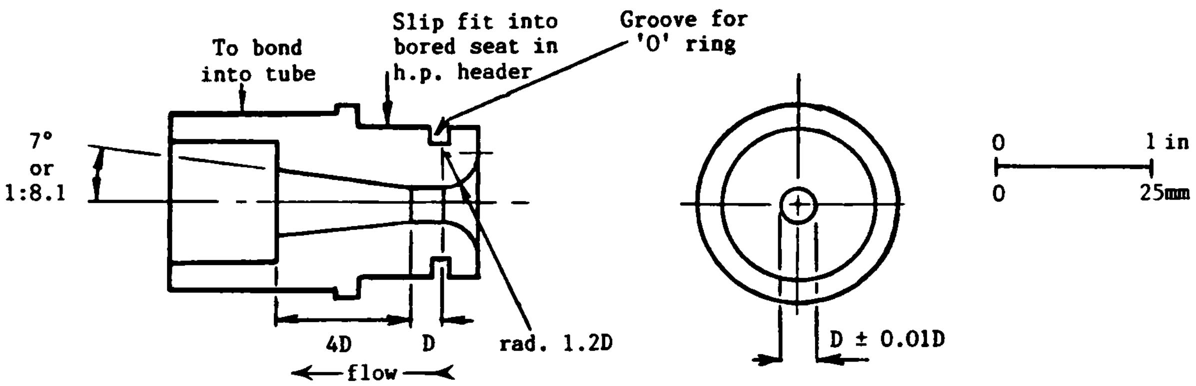
N.b. for no.8, the 1 in.
tube and valve became $1\frac{1}{4}$ in.



General assembly

Fig. F-6
(this and previous page)

The choked flow nozzle bank



Nozzle no.	D mm
1	1.19
2	1.68
3	2.38
4	3.36
5	4.76
6	6.73
7	9.51
8	13.45

Detail B

Functional dimensions of the choked flow nozzles No.5 shown

N.b. nos. 7 & 8 machined from larger stock

F.3.2 Other modifications

Originally the air supply to the choked flow nozzles was directly from the air receivers, which resulted in considerable variation of upstream pressure because of the hysteresis in the compressor governors (cutting in at about 6 bar and out at about 7 bar), thus fluctuation in air flow rates. The only way of overcoming this was to vent air from the receivers to keep the compressors working continuously, which was both wasteful and noisy as well as taking a considerable time for the pressure to settle down. Therefore it was decided to install a regulator in the line between receivers and nozzles. Investigation showed a feedback type to be needed to keep a close control on pressure over the full range of flow rate, accordingly a Norgren type R18-C00 remote regulator with feedback pilot 11-204-004 was obtained; this was found to be remarkably satisfactory, displaying outlet pressure variation of less than 1% over flow rates from zero to 600 cfm. The opportunity was taken to also install a filter/water trap unit in the line, to prevent carry-over of condensate from the receivers into the conveying line.

F.4 The instrumentation and control systems

The quantities which would need to be measured during test runs were primarily the pressure profiles along the two straight pipes either side of the test bend and the weight of product in the receiving hopper which was mounted on load cells; also it would be necessary to monitor the air pressures upstream and downstream of the choked flow nozzles used to control the air flow to the blow tank and supplementary air injection point. The final arrangement of all the control and instrumentation circuits on the rig is shown in fig. F-8 at the end of this section.

F.4.1 Conveying line pressure profiles

The problem of measuring the pressure profiles in the conveying line was thought to be a little more difficult than might at first meet the eye, so some time was spent considering this first. The re-analysis of David Mills' work had shown that pressure gradients in straight pipe of the order of 10 to 30 mbar/metre would be expected, and it was clear that to measure this to any accuracy, in pressures of the order of 2 or 3 bar, would not be easy.

Additionally, experience in running the conveying plant with an existing pipeline had shown that considerable fluctuation of pressure in the line, on top of a tendency for pressure to increase somewhat through the conveying cycle, was normal; typically the line inlet pressure might vary by as much as 0.2 bar in 2 seconds or so, on a pressure of 2 barg, in an apparently random fashion. Consideration suggested that this was to be expected when using a blow tank feeder, because of the transport delay in the pipeline in feeding back changes in pipeline pressure to the blow tank causing the system to operate on a limit cycle. This is explained in the note at the end of this Appendix.

This variation in pressure and product feed rate over short periods would cause disturbances, in the form of waves of increased- and decreased-density flow, to travel down the pipeline at the velocity of the air during conveying, so any single snapshot of pressure along the conveying line would not be truly representative because the flow rate of product would not be the same at all places along the line.

The result of all this was that it would clearly be necessary to measure with accuracy, differences as small as 1% between signals subject to noise of some 10% or so; thus averaging over a time period would be necessary, either by analogue filtering of the signals or by arithmetical averaging of a number of readings. To obtain the resolution required would demand the use of electronic pressure transducers of the highest sensitivity normally available (0.1% accuracy); analogue filtering of the actual pressure at tappings was not thought to be practical, because of a lack of controllability with such methods as using a chamber and a very small hole (especially with powder present in the air), whilst analogue filtering of the signals from the transducer was considered to be very difficult to achieve without compromising the accuracy. The only realistic option would be arithmetical averaging of a number of readings taken from the transducers by a data logging system, but it was obvious that careful choice of the time period and sampling frequency would be necessary in order to achieve satisfactory noise rejection.

The limit cycle operation of a blow tank and conveying line were considered, and it was apparent that for a pipeline 80m long operating with a (very low) inlet air velocity of 4m/s and pressure of 3 barg, thus an outlet velocity of 16m/s, the pipeline residence time of disturbances being about 8 seconds would lead to a lowest frequency of oscillation of some 1/8 Hz or so. It was considered necessary to sample over a minimum of say 10 cycles, i.e. 80 seconds, with a minimum of 10 readings. Then all noise at frequencies equal to or above this (save at harmonics of the sampling frequency - thought unlikely) should be rejected.

If the sampling period was too long, however, then it would not be possible to ensure the quasi-steady-state operation of the blow tank, which could operate only for a limited period of time before depressurising and recharging with product. The word "quasi-" is used as a qualification because even during the steadiest part of the blow tank conveying cycle, feed rate and line pressure generally increased by some 5% or so, which was not thought to be important provided average values were taken. This period could be as short as 1 minute or so at high flow rates of product, meaning that to obtain 10 readings under such conditions would require a minimum scanning frequency of 1/6 Hz. Ideally more than 10 readings should be available, to select the smoothest part, increasing the minimum scanning frequency to say 1/3 Hz to obtain 20 readings.

The rate of scan of the channels was thought to be important, ideally it should be well above (e.g. 10x) the frequency created by a disturbance in flow passing successive pressure measuring stations in order to avoid effects caused by the scan being 'overtaken' by such a pressure disturbance travelling along the line. This meant that all pressure stations in a 40m length of pipe should be scanned in a maximum of 0.1 second with a (high) air velocity of 40 m/s, then the results of the scan could be considered to be a snapshot of the pressure profile along the line.

F.4.2 Other pressure measurements

The pressure of the air in the high pressure header of the choked flow nozzle bank would have to be monitored in order that the air flow rates of the nozzles would be known; also the pressures in the low pressure headers would need monitoring to ensure that the critical pressure ratio of the nozzles was not exceeded, making the flow rates uncertain. It seemed ridiculous not to integrate these measurements into the system for measuring pressure profiles in the conveying line, so the decision was taken to do so.

F.4.3 Load cell

The load cells on which the receiving hopper was mounted were connected to a box which provided power to the cells, and added, amplified and applied offset to the signals to give a readout on a voltmeter, proportional to the weight of product in the hopper. The voltage to the meter was available externally and once certain modifications had been made to the instrument to smooth this output and amplify it to a useable level, the signal was suitable for feeding to a data logging system.

F.4.4 Control

It was apparent that a considerable amount of handling and processing of electronic signals would be needed; not only would analogue signals be gathered, but these would have to be converted to digital form for recording and averaging, requiring the use of a computer system and a considerable amount of interfacing equipment. It appeared that since a computer system would be needed to take readings from the rig, it would be sensible to allow this system to control the rig also, allowing the operator to be free to direct, and analyse results from, the test work. This would also tie in neatly with the aim of making the rig easier to use and quicker to turn around between test runs, mentioned in section F.2 above. If the valves on the choked flow nozzle bank were fitted with actuators as well as the valves on the blow tank, then with suitable interfacing they could all be operated automatically. It would obviously be necessary to have a system to override the computer manually, and feedback circuits from actuators to confirm valve positions so that interlocking to obviate the possibility of dangerous combinations of valve positions could be arranged.

The building of a mimic panel, to be located adjacent to the operator's working position at the computer, was considered. This would carry, on a schematic diagram of the rig, a switch for every valve so that each could be opened or closed manually, or placed under automatic control; also the position feedback circuits would operate lights in appropriate places on the diagram, to indicate the actual status of each valve. The cost of components for this, and the interfacing to link it to the computer, was estimated to be small compared to the cost of the computer system and the transducers and interfaces necessary for logging of the pressures, and although the amount of work involved in building it would be considerable, it was thought that this would be repaid in convenience of operation of the test rig. Ergonomic considerations were examined carefully when designing the operator station incorporating the mimic panel.

Once the mimic panel had been built and commissioned, it was found that the convenience of operation of the rig was so greatly improved over the original arrangement that the extra work involved in developing control software was not thought worthwhile for the project in hand; so although the interfacing was arranged, the rig was actually always driven manually during the test work; the remote control from the mimic panel proved a great asset in helping the work to proceed quickly.

F.4.5 Pressure transducers and tappings

Over a dozen different makes and models of strain gauge pressure transducer were considered for the application in mind, the type finally chosen being a silicon diaphragm transducer from Druck of Germany, chosen for its combined accuracy (linearity + hysteresis + repeatability) specification of 0.1%, with an overpressure capability of x4 without change in calibration; this would give sufficient resolution to read a change of 1% accurately. These devices were relatively new technology at the time, but were very little more expensive than older designs on the market giving no better accuracy than 1%. The output was a differential voltage of 30mV full scale, which it was thought would be reasonably immune to noise over the distance of some 40m or so from the data logging system to the farthest transducer, provided screened cable was used. No problems were experienced in respect of this. A flush diaphragm type of transducer, type PDCR 810, was specified for the conveying line measuring stations so that there would be no cavity to clog should any powder enter the tapping. In retrospect this was a wrong decision, because no problem was ever experienced with powder in the tappings but the transducers needed recalibration whenever they were removed and replaced because the tightening torque affected the offset (the diaphragms being located in the end of the mounting thread); this was not a serious drawback, however.

The pressure tappings on the conveying pipe were considered carefully, from the points of view of number, position and design. It was expected that the re-acceleration length after the bend would be about 4m, from the work in ref. 55, leaving a length of straight pipe of some 13m from the longest straight lengths of 17m which could be accommodated in the lab., in which a steady gradient would be observed. It was thought necessary to have a minimum of four points in this region to obtain a reliable straight line, so a spacing of 2m between stations was decided upon, giving 8 transducers in the straight length after the bend, of which hopefully about 6 would be in a region of steady gradient; only four stations were used before the bend, to obtain the pressure gradient leading up it. In fact the re-acceleration region after the bend, in which the pressure drop caused by the bend is actually developed, turned out to be as much as 7m but this still gave at least four points in the region of steady gradient further downstream.

The position of the tappings around the pipe was considered and it was perceived that if the flow happened to be bouncing off the wall of the pipe near a tapping (not unlikely immediately after a bend) then there may be a pressure difference around the circumference of the pipe. In order to average this, it would be necessary to put in several pressure tappings around the pipe section, connected by small tubes; however, the flow through these tubes caused by the pressure difference, necessary to the averaging, would cause the tubes (or any filters placed in the tappings) to be blocked by powder relatively quickly. Also, it was thought that once the flow had settled down to a steady gradient, no one side of the pipe would be uniquely subject to bouncing of the flow at any place; even the effect of gravity, increasing pressure in the bottom of the pipe and decreasing it at the top, would be consistent from one station to the next. Therefore to tap the pipe wall at just one place at each station would result in an error at stations immediately after the bend, but no error in the region of steady gradient which was the important part from which all measurements would derive (including the pressure loss caused by the bend). Therefore it was decided to tap into the pipe only at the top, for convenience of access.

The tappings would have to be designed to accommodate the M14 x 1.5 threaded mounting of the transducer, also a filter pad of some description to protect the transducer from powder, and an input from a high pressure air supply, via a valve, to blow powder out of the filter after use. The volume of the tappings would be kept small as far as possible, to reduce the amount of air flow into and out of them with fluctuations in pipeline pressure which would tend to take powder into the filters and clog them. The valve would need to be mounted directly on to the tapping for this reason. The arrangement adopted is shown in fig. F-7 below. The two filters are used so that should the passage of solids wear out the nylon pad filter, the sintered bronze disc would prevent the solids entering the tapping; in normal service, the nylon pad does the filtration to prevent the bronze disc from blinding, which they tend to do very easily. Calculations were performed to ascertain the time constant of the tapping, based on the volume of the tapping and the nominal air resistance of the sintered bronze material; it worked out in terms of a few milliseconds, so there was clearly no need to worry about the effect of this on the readings.

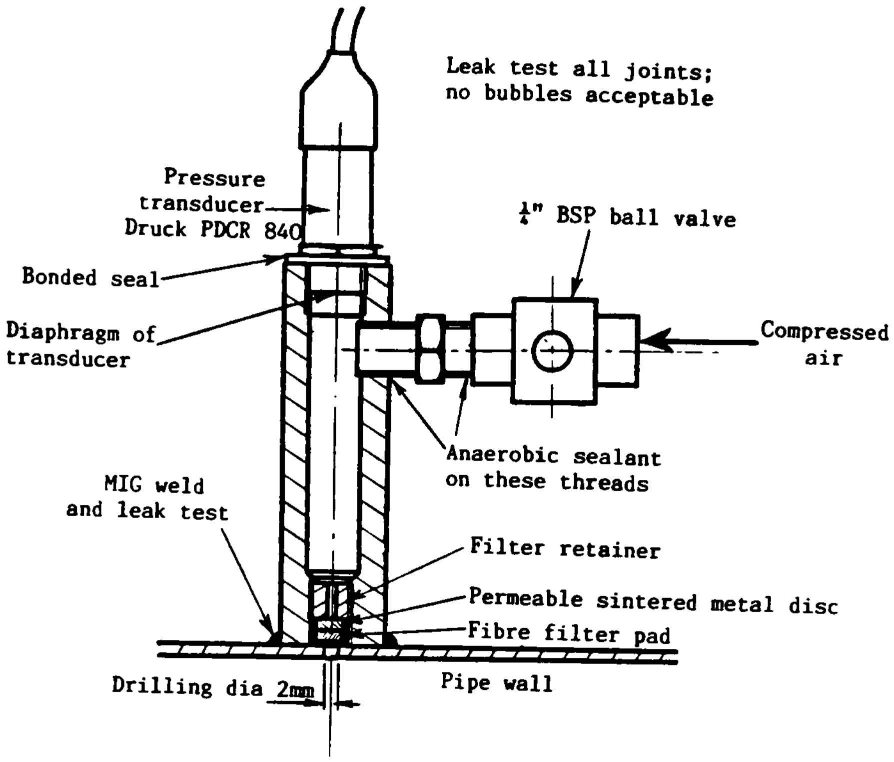


Fig. F-7
The pressure tappings on the pipeline

Once the tappings had been welded onto the pipe and the transducers and valves were in place, the pipeline was blocked and the tappings checked for leaks using soapy water, since any air flow through the filter would result in a pressure drop and thus a false reading. Any leaky welds were caulked with a punch, and leaking joints reworked as necessary to ensure absolute air-tightness. This procedure was repeated whenever the transducers were removed and replaced.

The pressure drop across the filters when blowing them clean between runs was checked occasionally, to monitor their state; with this purging carried out between every test run, no problems with clogging were ever experienced.

F.4.6 The data logging system

A microcomputer system would be linked to the rig for the purposes of taking and processing data, controlling, and performing calculations required by the operator. The BBC Master was chosen as a suitable machine for this purpose, having the advantage that there were a good many of these machines in the Polytechnic, enabling data to be taken away on disc and analysed when convenient.

The question of whether to buy or build the necessary equipment to interface this to the rig was considered very carefully, the requirement being for 18 channels of analogue input (for monitoring of the measured quantities), 22 channels of digital output (for control of the rig) and 44 channels of digital input (for status monitoring of control elements, e.g. valves). Scan rates of the double-ended differential voltage analogue signals, as discussed in section F.4.1, were taken account of. An outline design for a suitable system was worked up and costed, the estimate coming out at about £2000, not including technician and engineer time which it was thought would be very considerable. It appeared that suitable commercial systems were available for around £4000, which had been designed and developed by engineers far more experienced in the field than the current author, and proven in service; so it was clear that a commercial system would offer a much better return.

A great number of systems were examined. The one finally chosen, on the bases of both cost and the amount of work necessary to link it to the rig, was the Mowlem Microsystems ADU, an instrument having on-board intelligence giving it the ability to run measurement and control programs on demand from, but independent of, the host microcomputer, with which it communicated via a serial RS423 link. This was advantageous because it enabled a conveying test run to proceed whilst the operator was using the microcomputer to analyse data from the previous run. Additional useful features of this unit included the gain and configuration of the analogue input channels being set by software from the host machine, these channels being multiplexed into an analogue to digital converter of twelve bit resolution (1 in 4096 or .02%), preserving the resolution of the transducers; also the input channels would accept the transducer signals without any pre-conditioning, and the unit would supply the excitation voltage for the devices. The channel scan rate, of up to 100 channels/second, was well in excess of requirements; the digital outputs, of open-collector configuration, would drive small relays to operate the control functions on the mimic panel and the digital inputs would accept TTL levels (0 and 5 volts) which could easily be derived from the status feedback circuits.

In use this system proved to be totally beyond fault, the only difficulty encountered being the occasional system crash which was put down to mains glitches. A mains filter would probably have solved this, but the effort was not considered worthwhile for the small inconvenience caused.

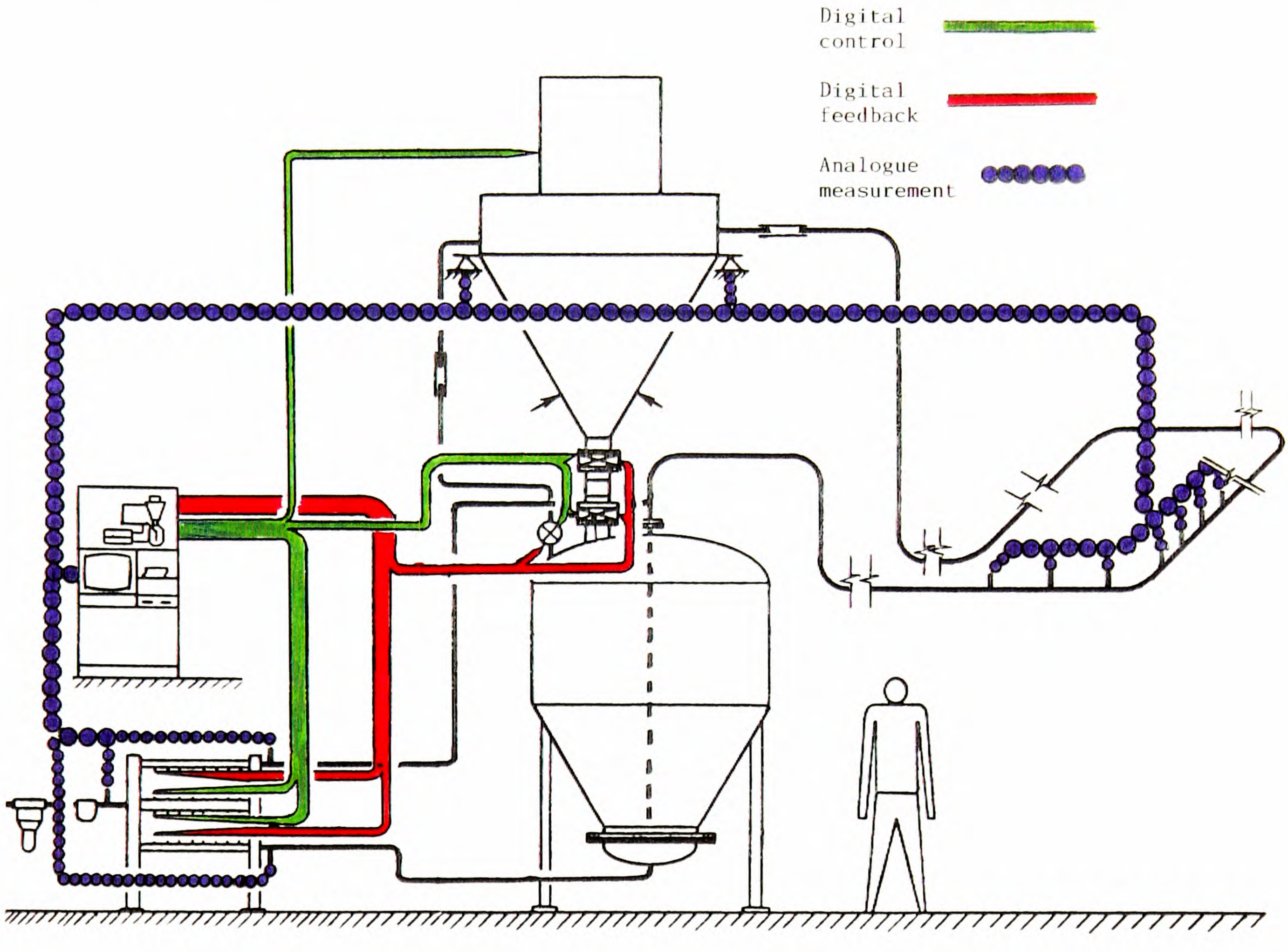


Fig F-8

The control and instrumentation circuits on the rig

For identification of parts see fig. F-4

F.5 Note about stability of blow tank feeder operation

That the operation of a blow tank feeder was expected to be unstable was apparent from the following reasoning:-

A blow tank contains quite a large volume of compressed air whose pressure will not change very quickly; the feed rate is controlled by the back pressure from the conveying line, i.e. if the pressure in the conveying line is low then the blow tank will feed product more quickly (for a given blow tank pressure). The pressure drop in the pipeline is determined by the product feed rate, but if say the feed rate increases, it takes some time for the pipeline pressure to increase because the increased density of solids in the line takes a finite time to travel along the pipeline - typically 4 seconds to travel along an 80m line at 20 m/s.

Therefore if the pipeline pressure is initially low, the blow tank feed rate will be high; so the pipeline pressure will increase, gradually; The blow tank meanwhile is seeing increasing back pressure so its feed rate will be reducing, but the pipeline pressure drop will still be increasing because the section of flow with the increased density of solids is travelling along the line getting faster as the air expands, causing greater pressure drop. The blow tank feed rate therefore continues to decrease until the increased-density section of flow reaches the end of the pipeline and exits, leaving the pipeline relatively empty, causing a drop of pressure at the inlet which in turn causes the blow tank to increase its feed rate and the whole cycle begins again.

Observation of the fluctuations suggested that the time needed for the blow tank to change its feed rate in response to changes in back pressure was very short, compared to the transport delay in the development of changed pipeline inlet pressure in response to changes in feed rate, thus making the whole system operate on a limit cycle. The analysis of this phenomenon would in itself make an interesting and useful project, especially because it would seem natural to suppose that increased pipeline length would make it more severe; increased pipeline length being very much the trend in pneumatic conveying applications in recent times.

APPENDIX G

DESIGN AND DEVELOPMENT OF SOFTWARE

This Appendix explains in detail the software used for gathering and analysis of the data, and the way in which it was developed. The entire suite was based on the BBC Master microcomputer, and except for the DIALOG software used for control of the data acquisition unit, it was entirely written by the author of this thesis.

The software described and listed represents the final version of each piece; inevitably the process of development was a gradual one, the programs evolving according to requirements as they were used. To describe the development processes in detail would take up far too much space, so only a broad outline is given of each.

G.1 Introduction

The software is conveniently divided up into four main areas, namely (i) the software for controlling the data acquisition unit, (ii) the software for performing the primary processing of the data to produce a single data point from the large amount of raw data collected during each conveying run, (iii) the software which enabled the user to examine the processed data and draw graphs, to help in the development of systems for storage of the processed data, and (iv) the software to call on data from the storage systems and use it to predict the performance of conveying pipelines.

There was of course a lot of supporting software for such purposes as helping the user to calculate the necessary combination of choked flow nozzles for obtaining the air flows required, calculating air flow rates from orifice meter manometer readings when calibrating the choked flow nozzles, and many other simple tasks which would otherwise be time-consuming. Many of these are listed purely for completeness, although they are of little moment.

G.2 Descriptions of software items

G.2.1 For controlling the data acquisition unit

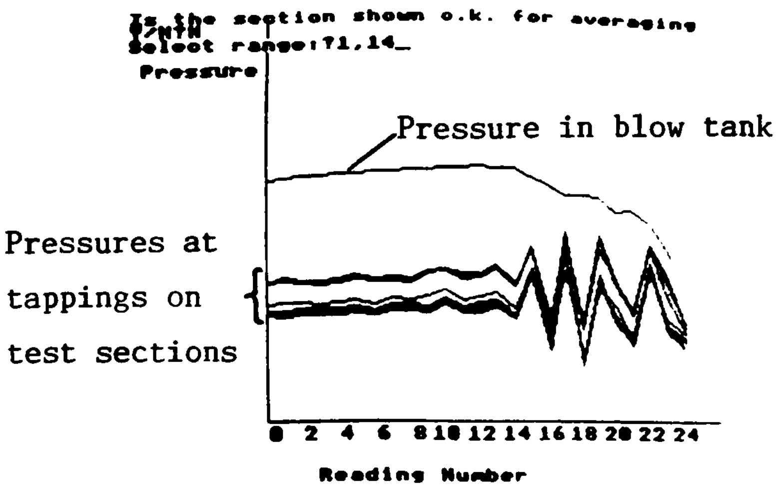
For this purpose the suite of software known as DIALOG, supplied by the manufacturers of the data acquisition unit (Mowlem Microsystems) was used. This was found to be quite easy to use once a little experience had been obtained, the primary functions being the definitions of signal types and analogue gain ranges for the analogue input channels, the definition and initiation of tests in the unit and also templates (i.e. calibration factors to be applied to the raw readings in data bits to obtain values in engineering units), and the transfer of measured data from the unit to disc, via the host computer, for storage.

Each disc of raw data carried 28 files, each file containing the whole of the data collected during one conveying run; this was preserved in case it should need re-examination later.

G.2.2 For primary processing of the data

This program, PROC2B, was written in BASIC. It first read raw data from the data files produced by DIALOG, then read and applied the appropriate template of calibration factors to obtain pressure and weight values in bar and kgf respectively. It then displayed a trace of line and blow tank pressures versus time and asked the user to select a reasonably steady state part of the conveying cycle. The program averaged the pressures over the selected period and, from the increase in load cell reading over the period, calculated the mass flow rate of solids.

Examples of displays from these first two stages are shown in fig. G-1 overleaf:-



Display of pressures vs. time
(Total duration 2min.)

DATA FILE NAME RES218

TITLE	MEAN
P1	0.8115
P2	0.8008
P3	0.8005
P4	0.7927
P5	0.6839
P6	0.6462
P7	0.6381
P8	0.6298
P9	0.6215
P10	0.6217
P11	0.6161
P12	0.6119
P.UPS.	4.199
P.BT.	1.374
P.SUPP.	1.234

MASS FLOW RATE = 84.33 kg IN 55 sec
= 1.533 kg/s or 5.52 tonne/hr
Calcs. carried out over period from
reading 2 to reading 13

Fig. G-1

Displays relating to the selection of a steady state part of the conveying cycle, and determination of pressures and flow rates

The next stage was to display a plot of pressure versus distance along the conveying line, and ask the user to select, firstly, the stations over which a straight line should be fitted downstream of the section where the bend effect developed; a line was fitted as instructed, using the method of least squares, and drawn. The user was given the option to change the selection of points and repeat the process. Secondly the user was asked to select points before the bend for the fitting of a parallel straight line, which was done and displayed in the same way. The displays for this are shown overleaf.

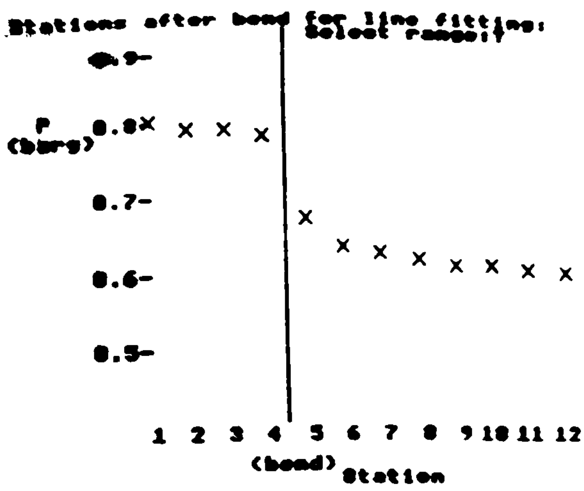


Fig. G-2(a)
Display showing pressure vs. distance along test sections

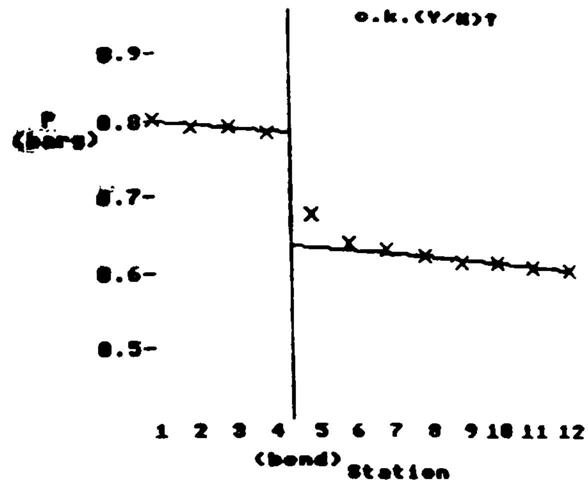


Fig G-2(b)
Fitting of tangents to points in G-2(a)

The user again had an option to change his selection and repeat, or go right back and start again; once the user was satisfied, the program proceeded to calculate the gradients of the straight lines and the pressure drop caused by the bend, and print out these and all other important calculated quantities pertaining to the particular test run on hard copy; the program chained another program which wrote all of this into a summary file of processed data, on the disc which carried the raw data from which it had been obtained. At a later stage it was copied into a master file, containing all the processed data from every conveying run, on another disc. The program which entered the processed data into the summary file, SUMMIN, also obtained from the user some details of the conveying line bore size and layout, the bend employed, and the product being conveyed, and entered these into the summary file as well so that this information would remain linked to the data.

The primary data processing program, PROC2B, and the supplementary program for saving the processed data into the summary file, SUMMIN, are listed, together with flow diagrams, at the end of the Appendix. The program for creating the summary and master files for processed data and moving data between them, MANDATE, is also listed.

G.2.3 For examination of processed data

In order to help in the search for correlations between variables, to assist the development of systems for storage of the data, it would clearly be necessary to draw many graphs. Data from something like 1000 test runs was on hand, divided up into groups of about 60 or so runs for each combination of bend, product, pipe size and pipeline loop tested; each test run in turn resulted in 14 measured or calculated variables, so the possible permutations of graphs to be drawn for each group of runs would be very large, even without allowing for the use of further derived variables which would undoubtedly be required. Additionally, graphs which proved useful for one set of runs would undoubtedly be re-drawn for other sets.

Therefore it was seen that to draw all the necessary graphs by hand would take more time than was available, so it was decided to explore the possibility of making the computer produce graphs of desired combinations of the variables on the screen at the command of the user, and print hard copies of useful graphs as required.

Since the processed data was stored on disc, it was not difficult to get the computer to read this in, and the very user-friendly graphics of the BBC machine meant that it was quite easy to draw graphs on the screen; to obtain hard copies was a little more difficult, but with some development work it was possible to drive an electronic daisy-wheel typewriter as a plotter to plot the graphs and axes on squared graph paper, and label axes etc.

The program, GRAPH, in its final form, is listed together with a flow diagram at the end of this Appendix. It was extended to calculate columns of derived variables to supplement the primary ones read in from the data file, and produced graphs of any combination of three primary or derived variables, by finding maximum values of each then drawing x and y axes, plotting one variable in the x direction, one in the y direction, and representing the third variable by 'coding' the points with letters (for example if the range of the third variable was say 0 to 50, then a point coded 'A' would be in the range 0 to 5, 'B' in the range 5 to 10, 'C' 10 to 15 and so on). An example of a hard copy of a graph is shown below:-

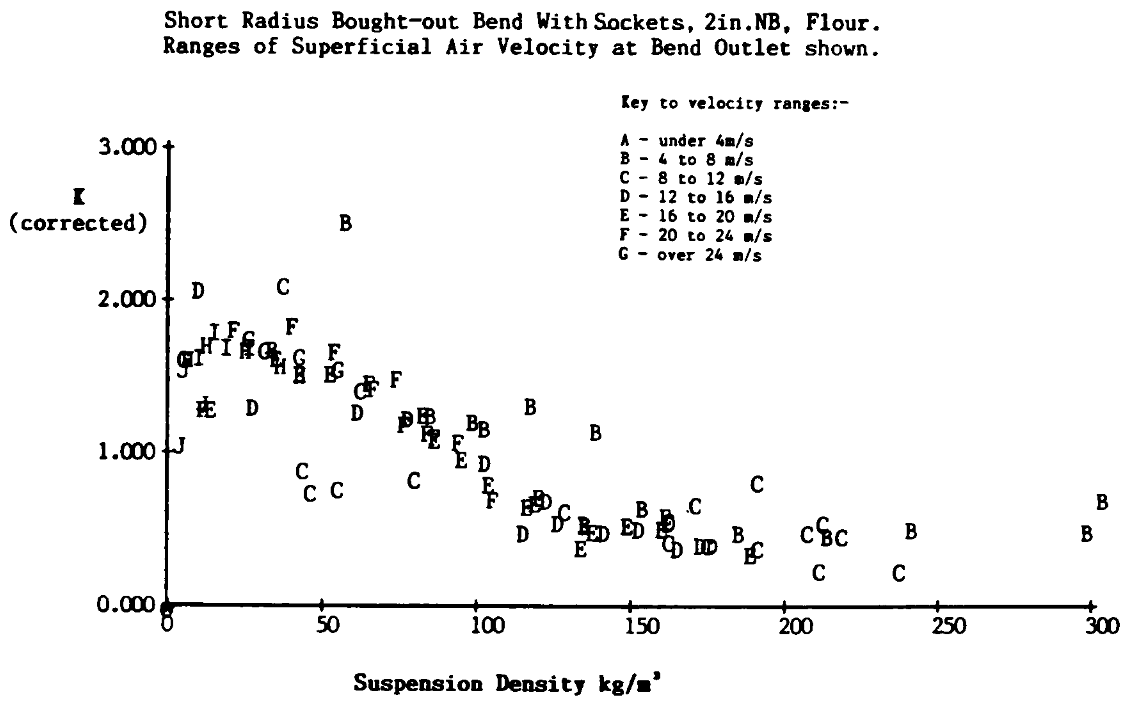


Fig. G-3

Graph of Corrected bend loss coefficient versus Suspension density with ranges of Superficial air velocity shown, produced by program GRAPH

G.2.4 For synthesis of the performance of conveying pipelines

With storage systems for the pressure loss data established, it became necessary to have a means of extracting the data from the systems and using this data to predict the performance of projected conveying pipelines, which was the ultimate goal of the project.

The storage systems which had been developed consisted of separate parts for prediction of losses along a straight horizontal pipe, and prediction of losses caused by bends.

For straight pipe, an equation for the 'solids contribution' to the pressure gradient in a straight pipe, to be added to the 'air only' contribution calculated from the Darcy equation, was used; the equations for the 'solids contribution' were somewhat different for the flour and the polyethylene pellets, but essentially they were similar in that they both gave the 'solids contribution' as a function of conveying air velocity and suspension density, as described in section 3.3.2 of Chapter 3, so would be very easy to enter into a computer program.

The bends were a little more difficult to deal with because the storage system consisted of an equation again containing conveying air velocity and suspension density, but also a coefficient whose value was stored on a graph; to enter this into the computer program required the use of a piecewise linear model, i.e. a series of straight lines between limits. The means used is described in some detail in section 3.5.2 of Chapter 3, so need not be repeated here.

The program worked by first asking the user to provide information about the layout and bore of the pipeline, then the mass flow rates of product and air to be used for the synthesis. Calculation began at the end of the pipeline with pressure equal to atmospheric, the conveying air volume being calculated using pV=mRT then the superficial air velocity found from the continuity equation (volume flow rate = velocity x pipe area). The suspension density was found by dividing the mass flow rate of product by the volume flow rate of air. Using these values the pressure drop in the last straight was found by calculating the 'solids contribution' from the data storage equation and the 'air only' contribution from the Darcy equation and adding the two together.

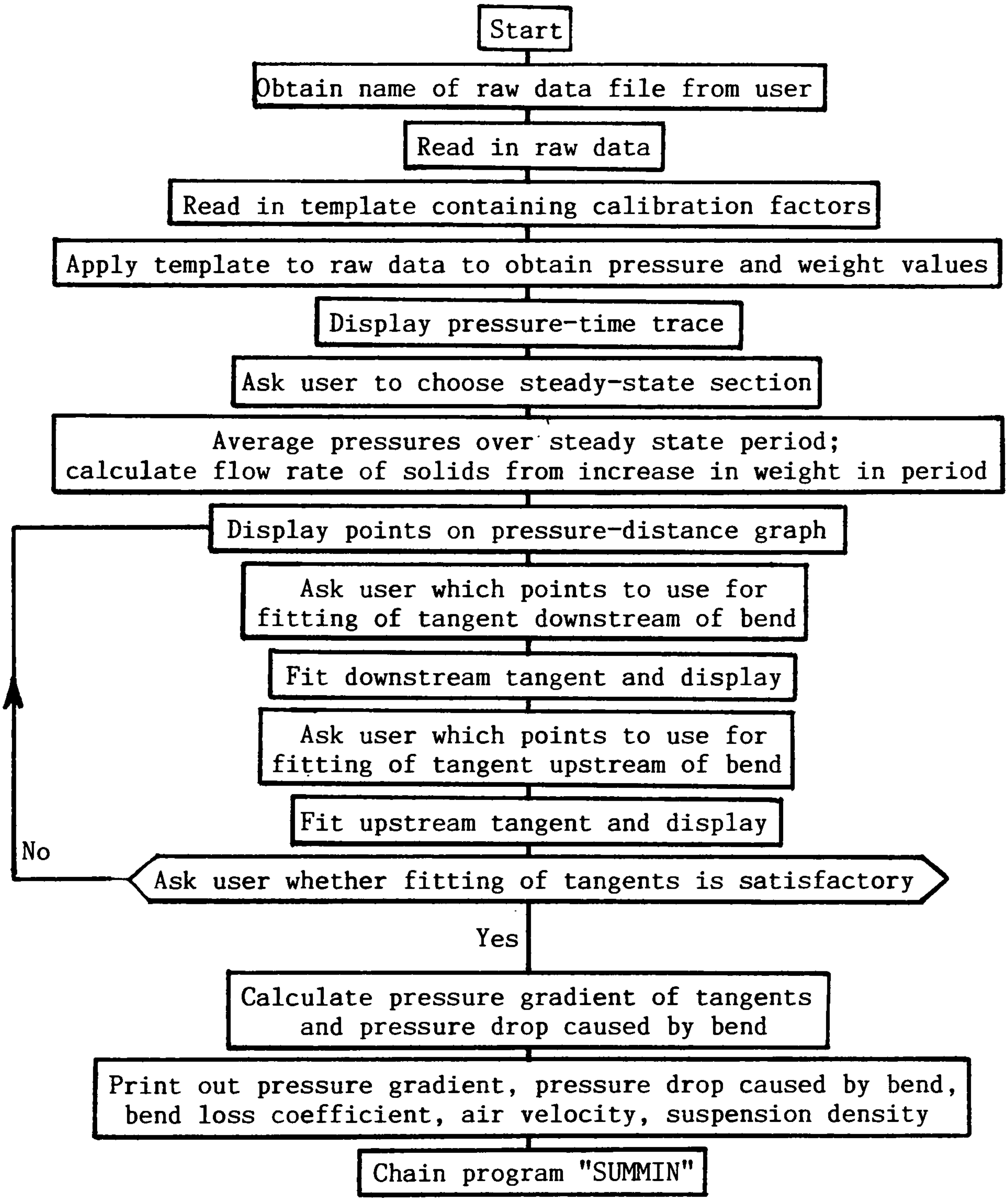
Pressure was incremented by the pressure drop calculated to find the air pressure at the outlet from the bend previous to the straight just dealt with (bearing in mind the model of step losses of pressure caused by the bends). The new volume flow rate of the air was calculated using pV = mRT again, and the new superficial air velocity and suspension density were determined. These values were used in conjunction with the bend pressure loss equation and piecewise linear model of the loss coefficient graph to find the pressure loss caused by the bend, and pressure incremented by this amount to find the air pressure at inlet to the bend.

The procedures for finding losses in bends and straights were repeated alternately back along the pipeline until the inlet was reached, and the resulting values of pressure, suspension density and air velocity at inlet printed out. The user was then given the opportunity to go through the process again with different flow rates of product and air.

Once again a flow diagram and listing of the program, SYSTEM, appear at the end of this Appendix.

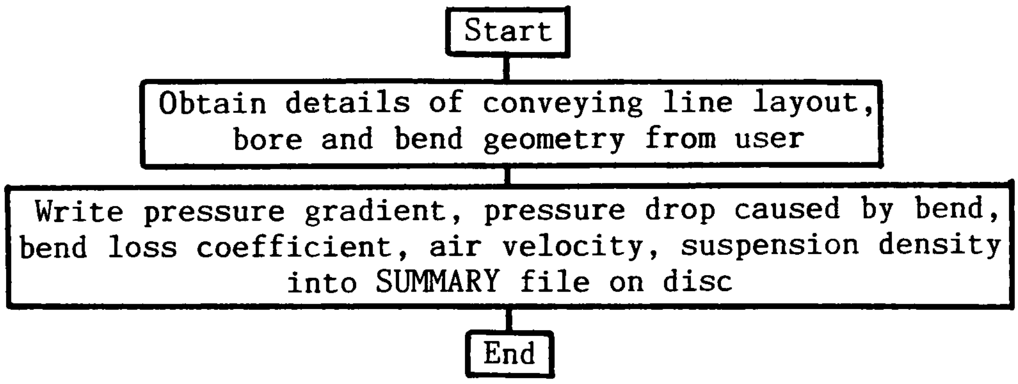
"PROC2B"

Program to process raw data from a conveying run to calculate conveying conditions and pressure loss data.



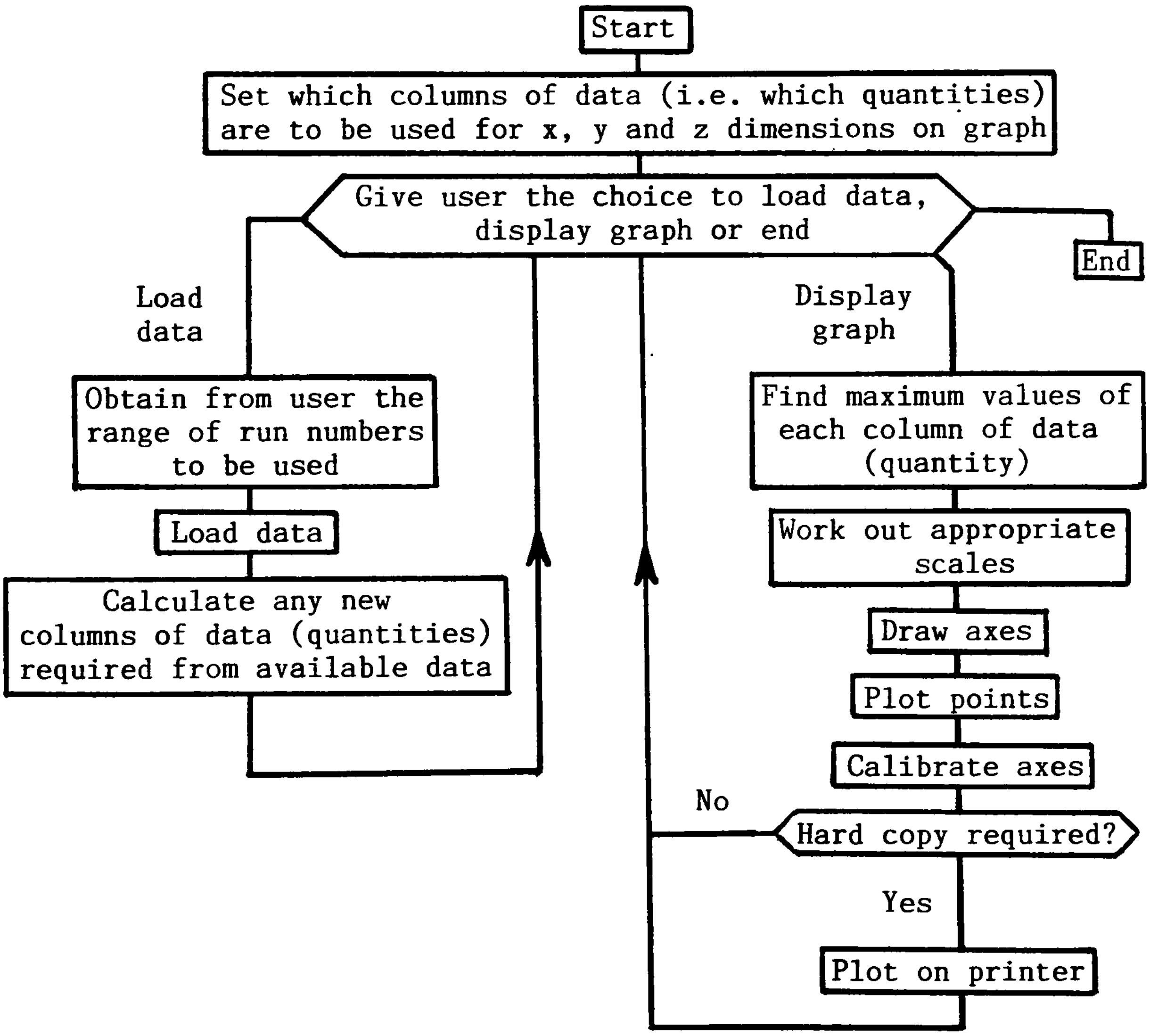
"SUMMIN"

Program to write data resulting from a run of "PROC2B" into summary file of processed data.



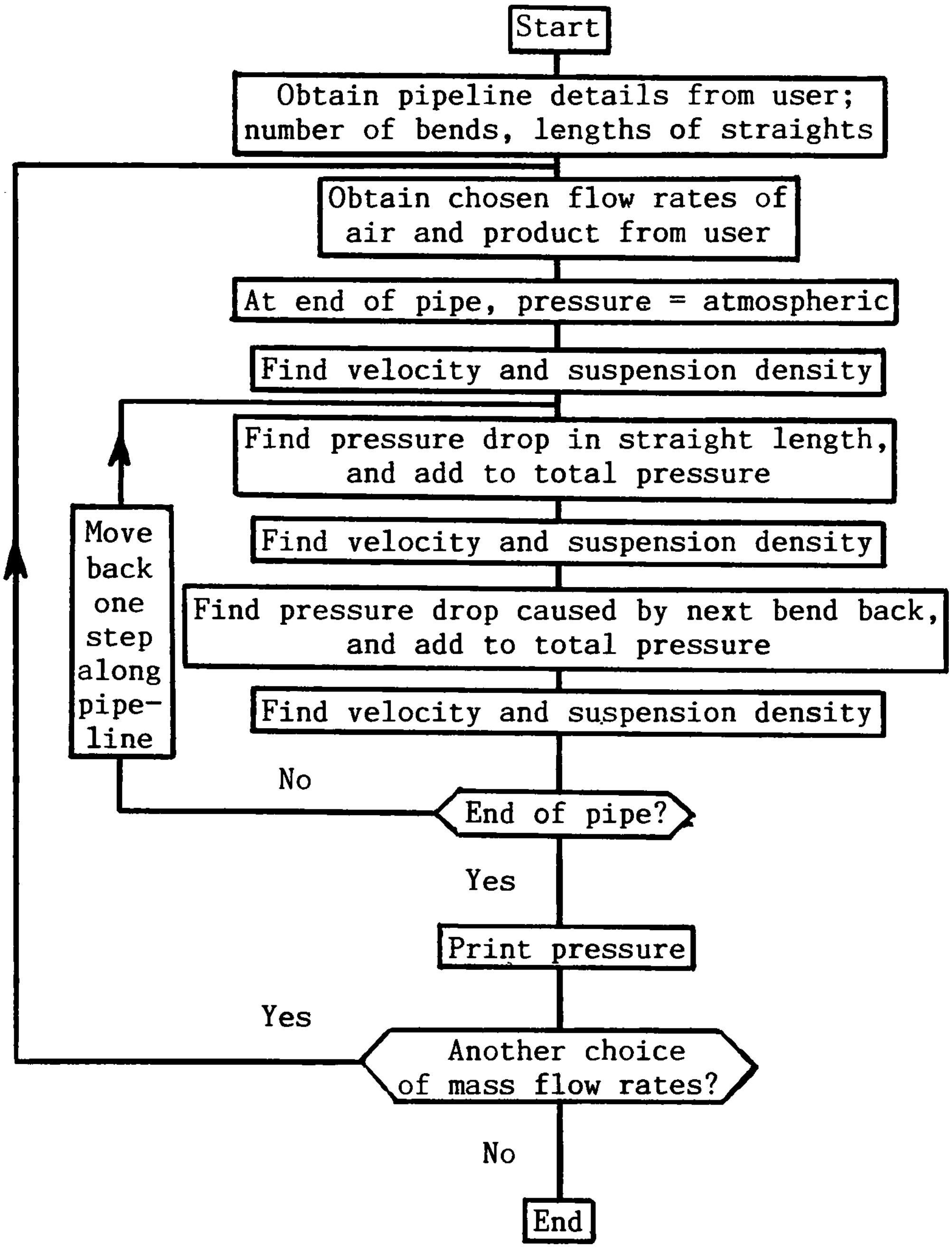
"GRAPH"

Program to plot graphs of measured quantities on the screen of the computer and give hard copies.



"SYSTEM"

Program for synthesising pressure drop along a conveying line from measured data. Written for a positive pressure system (as opposed to a vacuum system, for which calculation would begin at inlet end rather than outlet end, and proceed backwards).



Procedures for finding velocity and suspension density are explained in detail in the worked example, Appendix M, as are those for finding the pressure drop caused by a bend or a straight.

Listing of "PROC2B"

```
10 DIMRAW(30,17),OPROCESSED(30,17),A0(17),A1(17),UNITS$(17),TITLES$(17),PMEAN(17),P(30),DISTANC
E(14)
20 PROCRAWDATA
30 PROCCAL_CONSTS
40 PROCOPROCESS
45 MODE4
47 CHAR_WIDTH=32
50 PROCSTEADY_STATE
55 MODE7
60 PROCMEANS
70 PROCMDOTP
72 MODE4
73 PROCSCALE
75 PROCGRAPH
76 PROCLINES
77 MODE7
80 PROCSLR
90 PROCXSOD
100 PROCSUMMARY
1000 END
2000 DEFPROCRAWDATA
2010 INPUT"What is the name of the raw data file",FILE$
2020 A=OPENIN(FILE$)
2030 INPUT#A,TEST$:PRINT"TEST ";TEST$
2040 INPUT#A,RDGS:PRINT"NO. OF READINGS ";RDGS
2050 INPUT#A,COLS:PRINT"NO. OF COLUMNS ";COLS
2060 INPUT#A,START_DATE$:PRINT"START DATE ";START_DATE$
2070 INPUT#A,START_TIME$:PRINT"START TIME ";START_TIME$
2080 INPUT#A,MYSTERY
2090 FORROW=0TORDGS
2100   FORCOLUMN=1TO17
2110     INPUT#A,RAW$
2120     IFCOLUMN=1THENRAW=VAL(MID$(RAW$,7,2))+60*(VAL(MID$(RAW$,4,2))+60*VAL(MID$(RAW$,1,2)))E
LSERAW=VAL(RAW$)
2130     RAW(ROW,COLUMN)=RAW
2140   NEXTCOLUMN
2150 NEXTROW
2160 CLOSE#A
2170 ENDPROC
3000 DEFPROCCAL_CONSTS
3005 PRINT"TEMPLATE CALLED ON IS 'TEMP.AL'"
3010 A=OPENIN"TEMP.AL"
3020 INPUT#A,TEST
3030 INPUT#A,TEST
3040 INPUT#A,TEST
3050 FORCOLUMN=1TO17
3060   INPUT#A,TITLES$(COLUMN)
3070   INPUT#A,UNITS$(COLUMN)
3080   INPUT#A,A0(COLUMN)
3090   INPUT#A,A1(COLUMN)
3100   INPUT#A,A2,A3,A4
3120 NEXTCOLUMN
3130 CLOSE#A
3150 ENDPROC
4000 DEFPROCOPROCESS
4010 FORROW=0TORDGS
4020   FORCOLUMN=1TO17
4030     IFTITLES$(COLUMN)="TIME"THENOPROCESSED(ROW,COLUMN)=RAW(ROW,COLUMN)ELSEOPROCESSED(ROW,CO
LUMN)=A0(COLUMN)+A1(COLUMN)*RAW(ROW,COLUMN)
4040   NEXTCOLUMN
4050 NEXTROW
4060 ENDPROC
5000 DEFPROCSTEADY_STATE
5005 CLS
5010 SSSTART=0
5020 SSEND=RDGS
5035 FIRSTROW=SSSTART:LASTROW=SSEND
5037 MOVE300,1024:DRAW300,200:DRAW1280,200
5039 XSCALE=900/(LASTROW-FIRSTROW):YSCALE=880/2.5
5040 FORROW=FIRSTROWTOLASTROW
5050   PBT=OPROCESSED(ROW,15)
5060   X=ROW*XSCALE+300-FIRSTROW*XSCALE
5070   Y=PBT*YSCALE+200
5080   IFROW=FIRSTROWTHENMOVEX,YELSEDRAWX,Y
5090 NEXTROW
5095 PROC_PS
5100 VDU5
5102 IF(LASTROW-FIRSTROW)>12THENSTP=(LASTROW-FIRSTROW)DIV12ELSESTP=1
5105 @%=&0303
5110 FORROW=FIRSTROWTOLASTROWSTEPSTP
5120   X=ROW*XSCALE+300-FIRSTROW*XSCALE-2*CHAR_WIDTH
5130   Y=190
5140   MOVEX,Y
5150   PRINTROW
5160 NEXTROW
5170 MOVE400,100
5180 PRINT"Reading Number"
5185 MOVE30,900:PRINT"Pressure"
5190 VDU4
5200 @%=&0406
5210 INPUT"Is the section shown o.k  for averaging Y/N";ANS$
5220 IFANS$<>"Y"THENINPUT"Select range ";SSSTART;SSEND:CLS:GOTO5035
5700 ENDPROC
7000 DEFPROCMEANS
7001 @%=&0406
7003 INPUT"HARD COPY Y/N";HARD$
7005 IFHARD$<>"Y"ANDHARD$<>"N"THENGOTO7003
7007 IFHARD$="Y"THENVDU2
7009 PRINT'"DATA FILE NAME ";FILE$'"____________________"'''"TITLE","MEAN"'"-----","-----"'
7010 FORCOLUMN=2TO16
7020   RUNNING_TOTAL=0
7030   FORROW=SSSTARTTOSSEND
7040     RUNNING_TOTAL=RUNNING_TOTAL+OPROCESSED(ROW,COLUMN)
7050   NEXTROW
7060   PMEAN(COLUMN)=RUNNING_TOTAL/(SSEND-SSSTART+1)
7065   PRINTTITLES$(COLUMN),PMEAN(COLUMN)
7070 NEXTCOLUMN
7080 VDU3
7100 ENDPROC
9000 DEFPROCMDOTP
9010 ELAPSED_TIME=OPROCESSED(SSEND,1)-OPROCESSED(SSSTART,1)
9020 MASS_GAIN=OPROCESSED(SSEND,17)-OPROCESSED(SSSTART,17)
9030 MDOTP=MASS_GAIN/ELAPSED_TIME
9035 IFHARD$="Y"THENVDU2
9040 PRINT"MASS FLOW RATE = ";MASS_GAIN;" kg IN ";ELAPSED_TIME," sec = ";MDOTP," kg/s or ";MDOT
P*3.6;" tonne/hr"
9050 PRINT"Calcs carried out over period from reading ";SSSTART," to reading ";SSEND
9060 VDU3
9070 REPEAT UNTILGET$=CHR$(32)
9090 ENDPROC
11000 DEFPROCVELOCITY(PBAR)
11010 T=288:R=287:DPIPE=.053:PATMBAR=1.01
11020 APIPE=DPIPE*DPIPE*PI/4
11030 PABS=(PBAR+PATMBAR)*1E5
11040 VELOCITY=MDOTA*R*T/(PABS*APIPE)
11042 RHOAIR=PABS/(R*T)
11043 RHOSUSP=MDOTP/(APIPE*VELOCITY)
11045 ENDPROC
11047 DEFPROCSLR
11049 PSUPP=PMEAN(16)
11051 INPUT"What was tot. air mass flow rate ",MDOTA
11055 PROCVELOCITY(PSUPP)
11057 INITIAL_VELOCITY=VELOCITY
11060 PROCVELOCITY(PBENDOUTLET)
11062 BENDOUT_RHOSUSP=RHOSUSP
11063 BENDOUT_VELOCITY=VELOCITY
11064 BENDOUT_RHOAIR=RHOAIR
11065 PROCVELOCITY(PMIDPOINT)
11066 MIDPOINT_VELOCITY=VELOCITY
11067 MIDPOINT_RHOAIR=RHOAIR
11068 MIDPOINT_RHOSUSP=RHOSUSP
11069 SLR=MDOTP/MDOTA
11070 IFHARD$="Y"THENVDU2
11072 PRINT"Straight lines fitted over stations ";STAT1," to ";STAT2," and ";STAT3," to ";STAT4
11075 PRINT"Tot. air mass flow rate ";MDOTA
11080 PRINT"Initial velocity ";INITIAL_VELOCITY;" m/s"
11090 PRINT"Solids loading ratio ";SLR
11100 PRINT"Bend outlet pressure = ";PBENDOUTLET;" bar g."
11102 PRINT"Bend outlet air density = ";BENDOUT_RHOAIR;" kg/cu.m ";" Suspension density = ";BEN
DOUT_RHOSUSP;" kg/cu.m."
11105 PRINT"Bend outlet velocity = ";BENDOUT_VELOCITY;" m/s
11110 PRINT"Pressure gradient after bend = ";B;" bar/m "
11120 PRINT"Pressure at centre of section to which straight line is fitted = ";PMIDPOINT;" bar g
"
11125 PRINT"Air density here = ";MIDPOINT_RHOAIR;" kg/cu.m ";" Suspension density = ";MIDPOINT_
RHOSUSP;" kg/cu.m."
11130 PRINT"Velocity at same point = ";MIDPOINT_VELOCITY;" m/s
11140 PRINT"Pressure drop due to bend = ";PDROP;" bar
11290 PRINT'
11300 VDU3
11310 ENDPROC
13000 DEFPROC_PS
13010 FORSTATION=1TO12
13015   COLUMN=STATION+1
13020   FORROW=FIRSTROWTOLASTROW
13030     P(ROW)=OPROCESSED(ROW,COLUMN)
13040     X=ROW*XSCALE+300-FIRSTROW*XSCALE:Y=P(ROW)*YSCALE+200
13050     IFROW=FIRSTROWTHENMOVEX,YELSEDRAWX,Y
13060   NEXTROW
13070 NEXTSTATION
13090 ENDPROC
14000 DEFPROCSCALE
14010 YRANGE=0.8
14020 FORSTATION=1TO12
14030   READDISTANCE(STATION)
14040 NEXTSTATION
14050 DATA -7.13,-5.13,-3.13,-1.13,1.13,3.13,5.13,7.13,9.13,11.13,13.13,15.13
14060 XRANGE=DISTANCE(12)-DISTANCE(1)
14070 YMIDDLE=(PMEAN(2)+PMEAN(13))/2
14075 YMIN=YMIDDLE-YRANGE/2
14080 YSCALE=880/YRANGE
14090 XSCALE=900/XRANGE
14100 X0=300-DISTANCE(1)*XSCALE
14103 ENDPROC
14107 DEFPROCGRAPH
14110 VDU5:@%=&0303
14115 CLS
14120 FORSTATION=1TO12
14130   COLUMN=STATION+1
14140   X=DISTANCE(STATION)*XSCALE+X0
14150   Y=(PMEAN(COLUMN)-YMIN)*YSCALE+200
14160   IFSTATION=1THENMOVEX,YELSEMOVEX,Y
14165   PROCCROSS
14167   MOVE(X-2*CHAR_WIDTH),190:PRINTSTATION:MOVEX,Y
14170 NEXTSTATION
14180 MOVEX0,200:DRAWX0,1080
14190 @%=&0406
14200 FORP=0TO(YMIN+YRANGE)STEP0.1
14210   IFP>=YMINTHENMOVE90,((P-YMIN)*YSCALE+217):PRINTP;"-"
14220 NEXTP
14240 MOVE64,800:PRINT"P":MOVE0,760:PRINT"(barg)"
14250 MOVE700,100:PRINT"Station":MOVE500,130:PRINT"(band)"
14260 VDU4
14280 ENDPROC
14900 DEFPROCCROSS
14910 MOVE(X-12),(Y-12):DRAW(X+12),(Y+12):MOVE(X-12),(Y+12):DRAW(X+12),(Y-12)
14920 MOVEX,Y
14930 ENDPROC
15000 DEFPROCLINES
15010 PRINT"Stations after bend for line fitting:"
15020 REPEAT
15025   REPEAT
15030     PRINTTAB(20,1);"Select range:";
15040     INPUTFIRSTSTATION,LASTSTATION
15050     PROCSUMS
15060     PROCANALYSE
15070     PROCGRAPH
15080     PBENDOUTLET=A+B*0
15090     PROCAFTER
15120     PRINTTAB(23,0);"o.k.(Y/N)";
15130     INPUTANS$
15140   UNTILANS$="Y"
15145   STAT3=FIRSTSTATION:STAT4=LASTSTATION
15150   MIDPOINT=(DISTANCE(LASTSTATION)+DISTANCE(FIRSTSTATION))/2
15160   PMIDPOINT=A+B*MIDPOINT
```

Listing of "PROC2B" continued

```
15170   PRINTTAB(0,0);"Stations BEFORE bend, for parallel line:"
15180   REPEAT
15190     PRINTTAB(20,1);"Select range. ";
15200     INPUTFIRSTSTATION,LASTSTATION
15210     PROCSUMPLL
15220     PROCANALYSEPLL
15230     PROCGRAPH
15235     PROCAFTER
15240     X=DISTANCE(1)*XSCALE+X0
15250     P=C+B*DISTANCE(1)
15260     Y=(P-YMIN)*YSCALE+200
15270     MOVEX,Y
15280     X=X0
15290     PBENDINLET=C+B*0
15300     Y=(PBENDINLET-YMIN)*YSCALE+200
15310     DRAWX,Y
15320     PRINTTAB(23,0);"o.k.(Y/N)";
15330     INPUTANS$
15340   UNTILANS$="Y"
15342   STAT1=FIRSTSTATION:STAT2=LASTSTATION
15345   PDROP=PBENDINLET-PBENDOUTLET
15350   PRINTTAB(23,0);"Don't want to",TAB(23,1);"alter line after",TAB(23,2);"bend (Y/N)";
15353    INPUTANS$
15355 UNTILANS$="N"
15357 ENDPROC
15360 DEFPROCAFTER
15370 X=X0:Y=(PBENDOUTLET-YMIN)*YSCALE+200
15380 MOVEX,Y
15390 X=DISTANCE(12)*XSCALE+X0:P=A+B*DISTANCE(12):Y=(P-YMIN)*YSCALE+200
15400 DRAWX,Y
15410 ENDPROC
15600 DEFPROCSUMS
15610 SUMXJ=0:SUMYJ=0:SUMXJYJ=0:SUMXJSQD=0
15615 N=LASTSTATION-FIRSTSTATION+1
15620 FORSTATION=FIRSTSTATIONTOLASTSTATION
15630    COLUMN=STATION+1
15640    SUMXJ=SUMXJ+DISTANCE(STATION)
15650    SUMYJ=SUMYJ+PMEAN(COLUMN)
15660    SUMXJYJ=SUMXJYJ+DISTANCE(STATION)*PMEAN(COLUMN)
15670    SUMXJSQD=SUMXJSQD+DISTANCE(STATION)*DISTANCE(STATION)
15680 NEXTSTATION
15690 ENDPROC
15800 DEFPROCANALYSE
15810 B=(SUMXJYJ-SUMYJ*SUMXJ/N)/(SUMXJSQD-SUMXJ*SUMXJ/N)
15820 A=(SUMYJ-B*SUMXJ)/N
15830 ENDPROC
15900 DEFPROCSUMPLL
15910 N=LASTSTATION-FIRSTSTATION+1
15920 SUMXJ=0:SUMYJ=0
15930 FORSTATION=FIRSTSTATIONTOLASTSTATION
15940    COLUMN=STATION+1
15950    SUMXJ=SUMXJ+DISTANCE(STATION)
15960    SUMYJ=SUMYJ+PMEAN(COLUMN)
15970 NEXTSTATION
15980 ENDPROC
16000 DEFPROCANALYSEPLL
16010 C=(SUMYJ-B*SUMXJ)/N
16020 ENDPROC
17000 DEFPROCKSQD
17010 KSQD=PDROP*1E5/(BENDOUT_VELOCITY^2)
17015 IFHARD$="Y"THENVDU2
17020 PRINT"Bend. constant = ";KSQD;" (p in N/m^2)"'''
17025 VDU3
17030 ENDPROC
17200 DEFPROCSUMMARY
17205 PRINT"Save data in summary file ? ";
17210 ANS$=GET$:IFANS$="N"THENGOTO17330
17220 IFANS$<>"Y"ANDANS$<>"N"THENGOTO17210:PRINTANS$
17230 IFANS$="N"THENGOTO17330
17250 NRUNS$=MID$(FILE$,4)
17260 PROCINTEGERS
17270 CHAIN"SUMMIN"
17330 ENDPROC
17400 DEFPROCINTEGERS
17410 FACTOR=1E6
17420 N%=VAL(NRUNS$)
17430 A%=FACTOR*DOOTA
17440 P%=FACTOR*DOOTP
17450 S%=FACTOR*PSUPP
17460 G%=FACTOR*PBENDOUTLET
17470 C%=FACTOR*BENDOUT_VELOCITY
17480 E%=FACTOR*BENDOUT_RHOAIR
17490 F%=FACTOR*BENDOUT_RHOSUSP
17500 D%=FACTOR*PDROP
17510 K%=FACTOR*KSQD
17520 B%=FACTOR*B
17530 M%=FACTOR*PMIDPOINT
17540 V%=FACTOR*MIDPOINT_VELOCITY
17550 R%=FACTOR*MIDPOINT_RHOAIR
17560 H%=FACTOR*MIDPOINT_RHOSUSP
17600 ENDPROC
```

Listing of "SUMMIN" (version 2)

```
10  PROCREAL
20  A = OPENUP "SUMMARY"
30  PROCREAD_HEADING
40  PROCCHECK
50  IF CHECK = 0 THEN GOTO 90
60  ENTRY = VAL(NRUN$)-VAL(FIRSTRUN$)
70  PROCSETUP
80  PROCREVISE
90  CLOSE#A
100 END
110 DEF PROCREVISE
120 PRINT"Please wait"
130 PROCCHECK
140 IF CHECK = 1 THEN PROCCHANGE_DATA
150 ENDPROC
160 DEF PROCMANUAL_IN_SETUP
170 PRINT'"Information on set-up."
180 INPUT"Bend type ";BEND$:IF LEN(BEND$)>25 THEN PRINT "TOO LONG!" GOTO 180
190 INPUT"Product " PRODUCT$.IF LEN(PRODUCT$)>15 THEN PRINT "TOO LONG!" GOTO 190
200 INPUT"Loop ";LOOP$ IF LEN(LOOP$)>15 THEN PRINT "TOO LONG!":GOTO 200
210 ENDPROC
220 DEF PROCREAD_HEADING
230 PTR#A = 0  INPUT#A, VOLUME$
240 PTR#A = 5  INPUT#A, FIRSTRUN$
250 PTR#A =11  INPUT#A, LASTRUN$
260 ENDPROC
270 DEF PROCCHECK
280 IF VAL(NRUN$)>=VAL(FIRSTRUN$) AND VAL(NRUN$)<=VAL(LASTRUN$) THEN CHECK = 1 ELSE CHECK = 0
290 IF CHECK = 0 THEN PRINT"Run doesn't belong in this summary file (Volume ":VOLUME$,")  Out
side range ";FIRSTRUN$," to ",LASTRUN$
300 ENDPROC
310 DEF PROCCHANGE_DATA
320 ENTRY = VAL(NRUN$)-VAL(FIRSTRUN$)
330 PTR#A = (17 + ENTRY*180):PRINT#A, NRUN$
340 PTR#A = (23 + ENTRY*180):PRINT#A, BEND$
350 PTR#A = (50 + ENTRY*180):PRINT#A, PRODUCT$
360 PTR#A = (67 + ENTRY*180):PRINT#A, LOOP$
370 PTR#A = (84 + ENTRY*180)
380 PRINT#A, MDOTA,MDOTP,PSUPP,PBENDOUTLET,BENDOUT_VELOCITY,BENDOUT_RHOAIR,BENDOUT_RHOSUSP,PD
ROP,KSOD,B,PMIDPOINT,MIDPOINT_VELOCITY,MIDPOINT_RHOAIR,MIDPOINT_RHOSUSP
390 ENDPROC
400 DEF PROCSETUP
410 IF ENTRY = 0 THEN GOTO 500
420 PTR#A = (23 + (ENTRY-1)*180):INPUT#A,BEND$
430 PTR#A = (50 + (ENTRY-1)*180):INPUT#A,PRODUCT$
440 PTR#A = (67 + (ENTRY-1)*180):INPUT#A,LOOP$
450 PRINT"Previous setup was :-"'"Bend      ";BEND$'"Product  ";PRODUCT$'"Loop     ",LOOP$'"Sa
me again (Y/N)?";
460 ANS$ = GET$
470 IF ANS$ <> "Y" AND ANS$ <> "N" THEN GOTO 460
480 PRINT ANS$
490 IF ANS$ = "Y" THEN GOTO 510
500 PROCMANUAL_IN_SETUP
510 ENDPROC
520 DEF PROCREAL
530 FACTOR = 1E6
540 NRUN$ = STR$(N%)
550 MDOTA = A%/FACTOR
560 MDOTP = P%/FACTOR
570 PSUPP = S%/FACTOR
580 PBENDOUTLET = O%/FACTOR
590 BENDOUT_VELOCITY = C%/FACTOR
600 BENDOUT_RHOAIR = E%/FACTOR
610 BENDOUT_RHOSUSP = F%/FACTOR
620 PDROP = D%/FACTOR
630 KSOD = K%/FACTOR
640 B = B%/FACTOR
650 PMIDPOINT = M%/FACTOR
660 MIDPOINT_VELOCITY = V%/FACTOR
670 MIDPOINT_RHOAIR = G%/FACTOR
680 MIDPOINT_RHOSUSP = H%/FACTOR
690 ENDPROC
```

Listing of "GRAPH" (version 17)

```
1    *TK 0
2    DIM ARRAY(16,100)
3    *IGNORE
4    f = .005: DPIPE = .053
5    XCOLUMN=14 YCOLUMN=18 ZCOLUMN = 12
6    ZSTEP = 4
7    COL15XPR$ = "2*ARRAY(8,N)*1E5/(ARRAY(7,N)*ARRAY(5,N)^2)"
8    COL16XPR$ = "ARRAY(15,N)/(1.03-0.383E-3*NRUN)"
9    COL17XPR$ = "2*f/DPIPE*ARRAY(13,N)*ARRAY(12,N)^2*1E-5"
10   COL18XPR$ = "-ARRAY(10,N) - ARRAY(17,N)"
12   PROCMENU
15   FILENAME$ = "MASTER"
20   IF CHOICE$ = "1" THEN CLS:PROCLOAD
30   IF CHOICE$ = "2" THEN MODE 4:PROCGRAPH:MODE 7
100  IF CHOICE$ = "9" THEN CLS:END
130  GOTO 10
140  *DR 2
150  END
200  DEF PROCMENU
205  VDU15
210  CLS PRINT"************** MAIN MENU **************"''
220  PRINT "Type          To select"
230  PRINT "----          ---------"''
240  PRINT "  1           Load data from disc"'
250  PRINT "  2           Draw graph"'
280  PRINT "  9           Quit"
290  CHOICE$ = GET$:PRINT''"        OPTION No   ";CHOICE$;" SELECTED"
300  ENDPROC
800  DEF PROCLOAD
810  PROCLOADRANGE
820  A = OPENIN"MASTER":PROCCHECK:CLOSE#A
825  IF CHECK = 0 THEN GOTO 850
830  A = OPENIN"MASTER":PROCREAD_DATA:CLOSE#A
840  PROCSETUP_CHECK
845  PROCNEWCOLUMN(15,COL15XPR$)
847  PROCNEWCOLUMN(16,COL16XPR$)
848  PROCNEWCOLUMN(17,COL17XPR$)
849  PROCNEWCOLUMN(18,COL18XPR$)
850  ENDPROC
1000 DEF PROCNEWCOLUMN(COLNO,XPRESSION$)
1010 FOR NRUN = LORANGE TO HIRANGE
1020      N = NRUN - LORANGE
1030      ARRAY(COLNO,N) = EVAL(XPRESSION$)
1035      IF ARRAY(1,N) = 1.234 AND ARRAY(2,N) = 1.234 THEN ARRAY(COLNO,N) = 0
1040    NEXT NRUN
1050 ENDPROC
3000 DEF PROCLOADRANGE
3010 PRINT"Range of runs:"'"From:  ";:INPUT LORANGE
3030 PRINT"To:    ";:INPUT HIRANGE
3040 IF (HIRANGE-LORANGE)>100 THEN PRINT"100 runs max.!":GOTO 3000
3050 ENDPROC
3100 DEF PROCSETUP_CHECK
3400 ENDPROC
4000 DEF PROCREAD_HEADING
4010 PTR#A = 0: INPUT#A, VOLUME$
4020 PTR#A = 5: INPUT#A, FIRSTRUN$
4030 PTR#A =11  INPUT#A, LASTRUN$
4031 FIRSTRUN = VAL(FIRSTRUN$)  LASTRUN = VAL(LASTRUN$)
4033 ENDPROC
4037 DEF PROCCHECK
4038 PROCREAD_HEADING
4040 IF LORANGE>=FIRSTRUN AND HIRANGE<=LASTRUN THEN CHECK = 1 ELSE CHECK = 0
4050 IF CHECK = 0 THEN PRINT"Range not covered by current master file"
4060 ENDPROC
4100 DEF PROCREAD_DATA
4102 PROCREAD_HEADING
4105 FOR NRUN = LORANGE TO HIRANGE
4107      ENTRY = NRUN-FIRSTRUN
4108      N = NRUN - LORANGE
4110   REM    PTR#A = (17 + ENTRY*180):INPUT#A,ARRAY$(1,N)
4120   REM    PTR#A = (23 + ENTRY*180) INPUT#A,ARRAY$(2,N)
4130   REM    PTR#A = (50 + ENTRY*180) INPUT#A,ARRAY$(3,N)
4140   REM    PTR#A = (67 + ENTRY*180):INPUT#A,ARRAY$(4,N)
4150      PTR#A = (84 + ENTRY*180)
4160      FOR M = 1 TO 14
4170            INPUT#A,ARRAY(M,N)
4180          NEXT M
4190    NEXT NRUN
4200 ENDPROC
4600 DEF PROCGRAPH
4610 @%=&000507
4620 PROCSCALE
4630 PROCDRAW
4635 PROCCAL
4640 REPEAT:UNTIL GET$<>""
4643 PRINT"HARD COPY?"
4645 REPEAT:ANS$=GET$:UNTIL ANS$="Y" OR ANS$="N":PRINT ANS$:IF ANS$="Y" THEN PROCPGRAPH
4650 ENDPROC
4800 DEF PROCSCALE
4810 M=XCOLUMN:PROCMAX:XMAX=MAX
4820 M=YCOLUMN:PROCMAX:YMAX=MAX
4830 VERSCALE=YMAX/600:HORSCALE=XMAX/800
4840 ENDPROC
4900 DEF PROCDRAW
4910 MOVE 240,900:DRAW 240,200:DRAW1140,200
4920 VDU5
4925 REM MOVE 240-13,200+15:PRINT"X"
4930 FOR NRUN=LORANGE TO HIRANGE
4940        N=NRUN-LORANGE
4950        X=ARRAY(XCOLUMN,N):Y=ARRAY(YCOLUMN,N)
4960        MOVE(X/HORSCALE+240-13),(Y/VERSCALE+200+15)
4970   REM      MOVE0,0:PRINT"X":MOVE0,0
4973    D$ = "A": IF ARRAY(ZCOLUMN,N)>ZSTEP THEN D$ = "B":IF ARRAY(ZCOLUMN,N)>2*ZSTEP THEN D$ =
"C":IF ARRAY(ZCOLUMN,N)>3*ZSTEP THEN D$ = "D":IF ARRAY(ZCOLUMN,N)>4*ZSTEP THEN D$ = "E":IF ARRA
(ZCOLUMN,N)>5*ZSTEP THEN D$ = "F"
4974    IF ARRAY(ZCOLUMN,N)>6*ZSTEP THEN D$ = "G":IF ARRAY(ZCOLUMN,N)>7*ZSTEP THEN D$ = "H":IF
RRAY(ZCOLUMN,N)>8*ZSTEP THEN D$ = "I":IF ARRAY(ZCOLUMN,N)>9*ZSTEP THEN D$ = "J"
4975    PRINT D$
4976    REM     PRINT ARRAY(5,N)
4977         VDU4:PRINT TAB(10,30);NRUN:VDU5
4978    REPEAT:UNTIL GET$ <>""
```

Listing of "GRAPH" continued

```
4980        NEXT NRUN
4990 VDU4
5000 ENDPROC
5200 DEF PROCMAX
5210 MAX=0
5220 FOR NRUN= LORANGE TO HIRANGE
5230        N=NRUN-LORANGE
5240        IF ARRAY(M,N)>MAX THEN MAX=ARRAY(M,N)
5250      NEXT NRUN
5260 ENDPROC
5400 DEF PROCCAL
5410 VDU5
5420 XCAL=0
5430 REPEAT
5435        MOVE(XCAL/HORSCALE+(40-15),200 PRINT"!
5440        MOVE(XCAL/HORSCALE+(40-200),150 PRINT XCAL
5450        XCAL=XCAL+50
5460      UNTIL XCAL>=XMAX
5490 YCAL=0
5500 REPEAT
5510        MOVE0,(YCAL/VERSCALE+200+15) PRINT YCAL,"-"
5515        YCAL=YCAL+.01
5520      UNTIL YCAL>=YMAX
5580 VDU4
5590 ENDPROC
6700 DEF PROCUNDEFINED
6800 NOUNDEFINED = 0
6820 FOR NRUN = FIRSTRUN TO LASTRUN
6830        ENTRY = NRUN - FIRSTRUN
6840        PTR#A = (23 + ENTRY*180) INPUT#A, BENDS
6850        IF INSTR(BENDS, "BEND TYPE")<>0 THEN NOUNDEFINED = NOUNDEFINED + 1  UNDEFINED = 1 EL
SE UNDEFINED = 0
6860        IF NOUNDEFINED = 1 THEN PRINT'"The following runs are undefined:-
6870        IF UNDEFINED <> 0 THEN PRINT" ";NRUN;
6880      NEXT NRUN
6882 IF NOUNDEFINED = 0 THEN PRINT "All runs defined":CONT$ = "Y":GOTO 6890
6884 PRINT'"Continue with transfer ?";
6886 CONT$ = GET$:IF CONT$<>"N" AND CONT$<>"Y" THEN GOTO 6886
6888 PRINT CONT$
6890 PRINT
6900 ENDPROC
8000 DEF PROCPGRAPH
8005 VMI = 1  HMI = 2
8010 X$="1"
8020 PROCVMI
8030 PROCHMI
8040 PROCPSCALES
8050 VDU2
8060 XPOS =0:YPOS=0
8063 PROCPPOSITION(0,0):C$="O":PROCPRINT(C$)
8065 LASTY=0
8070 FOR NRUN=LORANGE TO HIRANGE
8080        N=NRUN-LORANGE
8090        X=ARRAY(XCOLUMN,N):Y=ARRAY(YCOLUMN,N)
8092     IF Y<LASTY THEN PROCPPOSITION(0,0)
8095    REM VDU3:PRINT "X ";X    Y";Y:VDU2

8100        PROCPPOSITION(X,Y)
8105      D$ = "A"  IF ARRAY(ZCOLUMN,N)>ZSTEP THEN D$ = "B":IF ARRAY(ZCOLUMN,N)>2*ZSTEP THEN D$ =
"C" IF ARRAY(ZCOLUMN,N)>3*ZSTEP THEN D$ = "D" IF ARRAY(ZCOLUMN,N)>4*ZSTEP THEN D$ = "E":IF ARRA
Y(ZCOLUMN,N)>5*ZSTEP THEN D$ = "F"
8106      IF ARRAY(ZCOLUMN,N)>6*ZSTEP THEN D$ = "G":IF ARRAY(ZCOLUMN,N)>7*ZSTEP THEN D$ = "H":IF
ARRAY(ZCOLUMN,N)>8*ZSTEP THEN D$ = "I":IF ARRAY(ZCOLUMN N)>9*ZSTEP THEN D$ = "J"
8110        C$ = D$
8120        PROCPRINT(C$)
8125    LASTY=Y
8130      NEXT NRUN
8140 PROCPPOSITION(0,0)
8150 C$= "+" PROCPRINT(C$)
8160 PROCAXES
8240 VDU3
8250 ENDPROC
8260 DEF PROCVMI
8270 REM SET VMI
8280 REM ESC RS,VMI
8290 VDU1,&1B VDU1,&1E VDU1 (VMI+1)
8300 ENDPROC
8310 DEF PROCHMI
8320 REM SET HMI
8330 REM ESC,US,HMI
8340 VDU1,&1B:VDU1,&1F VDU1 (HMI+1)
8350 ENDPROC
8360 DEF PROCDOWN(DOWN) LOCAL N
8370 REMVDU3.PRINT"DOWN ";DOWN:VDU2
8380 FOR N=1 TO DOWN
8390        VDU1,&0A
8400      NEXT N
8410 ENDPROC
8420 DEF PROCUP(UP):LOCAL N
8430 IF UP=0 THEN GOTO 8480
8440 REMVDU3:PRINT"UP ";UP VDU2
8450 FOR N=1 TO UP
8460        VDU1,&1B:VDU1,&0A
8470      NEXT N
8480 ENDPROC
8490 DEF PROCRIGHT(RIGHT) LOCAL N
8500 IF RIGHT=0 THEN GOTO 8550
8510 REMVDU3:PRINT"RIGHT  ;RIGHT.VDU2
8520 FOR N=1 TO RIGHT
8530        VDU1,&20
8540      NEXT N
8550 ENDPROC
8560 DEF PROCLEFT(LEFT) LOCAL N
8570 REMVDU3:PRINT"LEFT " LEFT:VDU2
8580 FOR N=1 TO LEFT
8590        VDU1,&08
8600      NEXT N
8610 ENDPROC
8620 DEF PROCPRINT(P$) LOCAL N
8630 FOR N = 1 TO LEN(P$)
8640   PRINT MID$(P$,N,1);:PROCRIGHT(2)
8650 NEXT N
8660 VDU3:PRINT LEN(P$):VDU2
8670 PROCLEFT(LEN(P$)*3)
8680 REMREPEAT UNTIL GET$<>
8690 ENDPROC
8700 DEFPROCPPOSITION(X,Y)
8710 XREL=INT(X/PHORSCALE)-XPOS
8720 YREL=INT(Y/PVERSCALE)-YPOS
8730 REMVDU3 PRINT "XREL ",XREL,    YREL ",YREL:VDU2
8740 IF XREL<0 THEN PROCLEFT(XREL*-1)ELSE PROCRIGHT(XREL)
8750 IF YREL<0 THEN PROCDOWN(YREL*-1)ELSE PROCUP(YREL)
8760 XPOS=INT(X/PHORSCALE)
8770 YPOS=INT(Y/PVERSCALE)
8780 ENDPROC
8790 DEF PROCPSCALES
8800 INPUT"No  of units per inch horizontally";HORUPI
8810 INPUT"No  of units per inch vertically";VERUPI
8820 PHORSCALE=HORUPI/40
8830 PVERSCALE=VERUPI/48
8840 PWIDTH=XMAX/40/PHORSCALE:PHEIGHT=YMAX/48/PVERSCALE
8850 PRINT"Width of graph = ";INT(PWIDTH)+1,"in , Height = ;INT(PHEIGHT)+ [illegible]

8860 REPEAT ANS$=GET$ UNTIL ANS$<>" ":PRINT ANS$
8870 IF ANS$<>"Y" THEN GOTO 8800
8890 ENDPROC
9010 DEF PROCAXES
9015 AT=@%
9020 PROCPPOSITION(0,0)
9030 H=0:C$="+"
9040 REPEAT
9050   PROCPPOSITION(H*HORUPI,0):PROCPRINT(C$)
9060   H=H+1
9070 UNTIL XPOS>XMAX/PHORSCALE
9080 H=0:PROCPPOSITION(0,0)
9090 REPEAT
9100   PROCPPOSITION(H*HORUPI,-6*PVERSCALE):XCAL$=STR$(XPOS*PHORSCALE) [illegible]
9110   H=H+1
9120 UNTILXPOS>XMAX/PHORSCALE
9130 PROCPPOSITION(0,0)
9200 H=0:C$="+"
9210 REPEAT
9220   PROCPPOSITION(0,H*VERUPI):PROCPRINT(C$)
9230   H=H+1
9240 UNTIL YPOS>YMAX/PVERSCALE
9250 H=0 PROCPPOSITION(0,0)
9255 XPOS=XPOS+5
9260 REPEAT
9269   REM YCAL=H*VERUPI.IF YCAL<10 THEN @%=&01020306 ELSE @%=&0102000 [illegible]
OSITION(0,H*VERUPI):PROCLEFT(0*LEN(YCAL$)):PROCPRINT(YCAL$):PROCRIGHT(0*LEN [illegible]
9270   YCAL=H*VERUPI IF YCAL<10 THEN @%=&01020306 ELSE @%=&0102000e
9273   YCAL$=STR$(YCAL):PROCPPOSITION(0,H*VERUPI):PROCPRINT(YCAL$)
9275   VDU3 PRINT"H IN Y-AXIS CAL. = ",H;"YCAL$ = ";YCAL$ VDU2
9280   H=H+1
9290 UNTILYPOS>YMAX/PVERSCALE
9295 XPOS=XPOS-17
9300 PROCPPOSITION(0,0)
9310 @%=AT

9315 REPEAT UNTIL GET$<>
9320 ENDPROC
```

Listing of "SYSTEM"
(version 4)

```
L.
    2REM NEW DP=K*0.5*RHO*C^2 CORRELATION. FLOUR. RADIUSED BENDS. OUTLET PRESSUR
E ZERO GAUGE
    5 @% = &060A
    8 KBEND = 1
   10 DIM LGTH(30),DPST(30),DPB(30)
   20 PROCINPUT
   30 PROCCONSTANTS
   40 PROCMDOT
   50 P = 1E5
   55 PROCHEADINGS
   60 K$ = "END":PROCCONDITIONS
   70 FOR STRT = NOSTRTS TO 1 STEP -1
   80      PROCSTRT
   90      K$ = "STRT":PROCCONDITIONS
  100      PROCBEND
  110      K$ = "BEND":PROCCONDITIONS
  120      NEXT STRT
  130 PROCPRINT
  140 PRINT"Another mdot Y/N "
  150 IF GET$ <> "N" THEN GOTO 40
  160 END
 1000 DEF PROCINPUT
 1005 INPUT"Pipe bore(mm) ";D:D = D/1000
 1010 INPUT"Number of bends ";NOSTRTS
 1020 FOR STRT = 1 TO NOSTRTS
 1030      PRINT"Straight length after bend ";STRT;
 1040      INPUT LGTH(STRT)
 1050      NEXT STRT
 1060 ENDPROC
 1200 DEF PROCCONSTANTS
 1210 G = 9.81: R = 287: T   288: F   .005
 1220 APIPE = D^2 * PI/4
 1230 ENDPROC
 1300 DEF PROCMDOT
 1310 INPUT"Air mass flow rate(kg/s) ";MDOTA
 1320 INPUT"Product mass flow rate(tonne/hr) ";MDOTP:MDOTP - MDOTP/3.6
 1330 ENDPROC
 1400 DEF PROCCONDITIONS
 1410 RHOAIR = P/(R * T)
 1420 CSUPERF = MDOTA/(RHOAIR*APIPE)
 1430 RHONSS = MDOTP/(CSUPERF*APIPE)
 1440 PROCPRINTCONDITIONS
 1450 ENDPROC
 1500 DEF PROCSTRT
 1510 DPDLAIR = 2*F*CSUPERF^2*RHOAIR/D
 1515 N = CSUPERF/8
 1520 DPDLSOL = 6.5E2*(.01*RHONSS)^N
 1530 DPDLTOT = DPDLAIR + DPDLSOL
 1540 DPST(STRT) = DPDLTOT * LGTH(STRT)
 1550 P = P + DPST(STRT)
 1560 ENDPROC
 1700 DEF PROCBEND
 1710 A1 = .0080: A0 = 1.57
 1711 IF RHONSS>14 THEN A1 = 0: A0 = 1.69
 1712 IF RHONSS>37 THEN A1 = -.0112: A0 = 2.1
 1713 IF RHONSS>83 THEN A1 = -.0145: A0 = 2.37
 1714 IF RHONSS>125 THEN A1 = -.0045: A0 = 1.11
 1715 IF RHONSS>152 THEN A1 = 0: A0 = .45
 1717 KREF = A0 + A1*RHONSS
 1718 K = KREF*KBEND
 1720 DPB(STRT) = K*.5*RHONSS*CSUPERF^2
 1730 P = P + DPB(STRT)
 1740 ENDPROC
 1900 DEF PROCPRINT
 1910 PRINT"        No."." dpst(mb)"," dpb(mb)"
 1920 FOR STRT = 1 TO NOSTRTS
 1930      PRINT STRT.DPST(STRT)/100.DPB(STRT)/100
 1940      NEXT STRT
 1950 PRINT "Total pressure drop = ";(P/1E5)-1" bar"
 1955 PB = 0
 1960 FOR STRT = 1 TO NOSTRTS
 1970      PB = PB + DPB(STRT)
 1980      NEXT STRT
 1990 PRINT PB/(P-1E5)*100;"% due to bends"
 2000 PRINT"Supfl. air vel. at inlet = ";CSUPERF;" m/s"
 2010 ENDPROC
 3000 DEF PROCPRINTCONDITIONS
 3010REM****** VDU2
 3030 IF K$ = "BEND" THEN PRINT "BEND ";STRT;" : ";
 3035 IF K$ = "STRT" THEN PRINT "STRAIGHT ";STRT;" : ";
 3037 IF K$ = "END" THEN PRINT "END OF PIPE ":" : ";
 3040 REMPRINT TAB(16),RHOAIR,RHONSS,CSUPERF
 3045 IF K$   "STRT" THEN PRINT TAB(16),DPDLAIR,DPDLSOL." ",RHOAIR.RHONSS.CSUPER
F
 3050 REMIF K$ = "BEND" THEN PRINT TAB(16)," "," ",KSQD,RHOAIR,RHONSS,CSUPERF
 3060 IF K$   "END" THEN PRINT TAB(16)," "," "," ",RHOAIR.RHONSS.CSUPERF
 3070 VDU3
 3100 ENDPROC
 4000 DEF PROCHEADINGS
 4005REM**** VDU2
 4010 PRINT'" "," ","dp/dlair","dp/dlsol.","    k" "Rho air","Rho susp.","Air ve
l."
 4015 VDU3
 4020 ENDPROC
```

Listing of "MANDATE" (version 12)

```
*.L.
    2 DIM ARRAY(14,30) ARRAY$(4,30)
    5 FILETYPE$ = SUMMARY
   10 PROCMENU
   15 FILENAME$ = FILETYPE$
   20 IF CHOICE$ = '1' THEN CLS PROCCREATE
   25 IF CHOICE$ = '2 THEN CLS PROCRECALL
   30 IF CHOICE$ = "3' THEN CLS PROCREVISE
   40 IF CHOICE$ = 4 THEN CLS PROCTRANSFER
   50 IF CHOICE$ = "5 THEN CLS PROCKEY
   80 IF CHOICE$ = '8' THEN CLS PROCFILETYPE
  100 IF CHOICE$ = '9' THEN CLS END
  130 GOTO 10
  150 END
  200 DEF PROCMENU
  205 VDU15
  210 CLS PRINT"************** MAIN MENU **************                    File type ",FILETYPE$
  220 PRINT Type          To select
  230 PRINT ----          ---------
  240 PRINT   1           Create a new file'
  250 PRINT   2           Recall data from old file"'
  260 PRINT " 3           Change data in old file'
  262 PRINT " 4           Transfer data from summary"''                                   to master file"

  265 PRINT " 5           Print key to info column"'"                                 (hard copy only)"
  275 PRINT   8           Change file type"'
  280 PRINT ' 9           Quit"
  290 CHOICE$ = GET$ PRINT "        OPTION No. ";CHOICE$," SELECTED"
  300 ENDPROC
  400 DEF PROCRECALL
  405 PRINT''''"Recalling data ... "
  410 A = OPENIN FILENAME$ PROCREAD_HEADING CLOSE#A
  420 PRINT "Current drive contains runs ";FIRSTRUN$," To ",LASTRUN$''"Is the one you wish to ex
amine in this range ? (Y/N)
  430 ANS$ = GET$
  440 IF ANS$ = "N THEN GOTO 600
  450 IF ANS$ <>'Y' THEN GOTO 430
  460 PRINT ANS$
  470 INPUT"What run no.? ,NRUN$
  480 PROCCHECK
  490 IF CHECK = 0 THEN GOTO 580
  500 ENTRY = VAL(NRUN$)-VAL(FIRSTRUN$)
  510 A = OPENIN FILENAME$
  515 PTR#A = (17 + ENTRY*180) INPUT#A,NRUN$
  520 PTR#A = (23 + ENTRY*180) INPUT#A,BEND$
  525 PTR#A = (50 + ENTRY*180).INPUT#A,PRODUCT$
  530 PTR#A = (67 + ENTRY*180):INPUT#A,LOOP$
  535 PTR#A = (84 + ENTRY*180)
  540 INPUT#A, MDOTA,MDOTP,PSUPP,PBENDOUTLET,BENDOUT_VELOCITY,BENDOUT_RHOAIR,BENDOUT_RHOSUSP,PD
ROP,KSQD,B,PMIDPOINT,MIDPOINT_VELOCITY,MIDPOINT_RHOAIR,MIDPOINT_RHOSUSP
  545 CLOSE#A
  547 PROCPRINT
  550 PRINT"Hard copy (Y/N) ",
  555 HARD$ = GET$
  565 IF HARD$<> "N" AND HARD$<>"Y" THEN GOTO 555

  570 PRINT HARD$
  576 IF HARD$ = Y" THEN PROCCOLUMNHEAD PROCHARD
  580 PRINT'"Recall any more ? (Y/N)
  585 ANS$ = GET$
  590 PRINT ANS$
  595 IF ANS$ = Y' THEN GOTO 410
  600 ENDPROC
  700 DEF PROCREVISE
  710 A = OPENIN FILENAME$ PROCREAD_HEADING CLOSE#A
  720 PRINT '"File contains runs nos ",FIRSTRUN$," To ",LASTRUN$
  730 PRINT'Is it one of these you wish to alter? (Y/N)' IF GET$<>"Y" THEN GOTO 900
  735 PRINT"Please wait "
  740 A = OPENUP FILENAME$
  770 PROCMANUAL_IN_SETUP
  780 PROCMANUAL_IN_RUN
  790 PROCCHECK
  800 IF CHECK = 1 THEN PROCCHANGE_DATA
  810 PRINT "Revise any more entries? (Y/N)"
  815 ANS$ = GET$
  820 IF ANS$='N THEN GOTO 890
  830 IF ANS$    Y THEN GOTO 815
  840 PRINT"Same combination of BEND, PRODUCT & LOOP? (Y/N)'
  850 ANS$ = GET$
  860 IF ANS$ = 'Y THEN GOTO 780
  865 IF ANS$ = 'N' THEN GOTO 770 ELSE GOTO 850
  890 CLOSE#A
  900 ENDPROC
 1000 DEF PROCCREATE
 1002 PRINT''' "Are you certain you wish to create a new ";FILENAME$, file, destroying any cur
rent one? (Y/N)"
 1005 IF GET$<>"Y" THEN GOTO 1180
 1007 PRINT '"Creating new ";FILENAME$;" file"
 1010 A = OPENOUT FILENAME$
 1020 PRINT#A, "VOL"
 1030 PRINT#A, "SRUN"
 1040 PRINT#A, "FRUN'
 1045 IF FILETYPE$ = "MASTER" THEN NENTRIES = 1000 ELSE NENTRIES = 30
 1050 FOR ENTRY = 0 TO NENTRIES
 1060       PTR#A = (17 + ENTRY*180)
 1070       PRINT#A, "RUN '
 1080       PRINT#A, "BEND TYPE0123456789012345"
 1090       PRINT#A, "PRODUCT89012345"
 1100       PRINT#A, "LOOP567890123 45"
 1110       FOR N = 1 TO 14
 1120             PRINT#A, 1.234
 1130           NEXT N
 1140       PRINT#A, "1234567890123456789012345 67"
 1147       PRINT TAB(10,10);ENTRY
 1150     NEXT ENTRY
 1170 PROCHEADING
 1175 PRINT''''"Creation complete; press space bar to continue"
 1178 REPEAT:UNTIL GET$ = " "
 1180 ENDPROC
 1200 DEF PROCHEADING
 1210 INPUT"NO. of first run to be entered ";FIRSTRUN$:IF LEN(FIRSTRUN$)>4 THEN GOTO 1210
 1220 INPUT"NO. of last run to be entered ";LASTRUN$ IF LEN(LASTRUN$)>4 THEN GOTO 1220
 1230 INPUT"Volume no represented ";VOLUME$ IF LEN(VOLUME$)>3 THEN GOTO 1230
 1250 PTR#A = 0: PRINT#A, VOLUME$
 1260 PTR#A = 5: PRINT#A, FIRSTRUN$
 1270 PTR#A =11 PRINT#A, LASTRUN$
 1280 FIRSTRUN = VAL(FIRSTRUN$)
 1290 LASTRUN = VAL(LASTRUN$)
 1300 FOR NRUN = FIRSTRUN TO LASTRUN
 1310       NRUN$ = STR$(NRUN)
 1320       ENTRY = NRUN-FIRSTRUN
 1330       PTR#A = (17 + ENTRY*180) PRINT#A, NRUN$
 1335       PRINT TAB(10,15),NRUN
 1340     NEXT NRUN
 1350 CLOSE#A
 1360 ENDPROC
 2800 DEF PROCMANUAL_IN_SETUP
 2805 PRINT 'Information on set-up'
 2810 INPUT"Bend type ",BEND$.IF LEN(BEND$)>25 THEN PRINT "TOO LONG!" GOTO 2810
 2820 INPUT"Product ";PRODUCT$ IF LEN(PRODUCT$)>15 THEN PRINT "TOO LONG!" GOTO 2820
 2830 INPUT"Loop ";LOOP$:IF LEN(LOOP$)>15 THEN PRINT "TOO LONG!":GOTO 2830
 2840 ENDPROC
 3000 DEF PROCMANUAL_IN_RUN
 3005 PRINT'"Data from run."
 3010 INPUT"Run no ";NRUN$ IF LEN(NRUN$)>4 THEN PRINT "TOO LONG":GOTO 3010
 3015 PRINT
 3020 INPUT"Mass flow of air kg/s ";MDOTA
 3030 INPUT"Mass flow of product kg/s ";MDOTP
 3040 INPUT"Supplementary pressure bar ";PSUPP
 3045 PRINT
 3050 PRINT"Bend outlet:"
 3060 INPUT"      Pressure bar        ";PBENDOUTLET
 3070 INPUT"      Velocity m/s        ";BENDOUT_VELOCITY
 3080 REM INPUT"      Air dens.kg/m3      ";BENDOUT_RHOAIR
 3090 REM INPUT"      Susp. dens. kg/m3 ";BENDOUT_RHOSUSP
 3095 PRINT
 3100 INPUT"Bend pressure drop bar ";PDROP
 3110 REM INPUT"k-squared factor         ";KSQD
 3115 PRINT
 3120 PRINT"Straight section:"
 3130 INPUT"      dp/dx   bar/m       ";B
 3140 INPUT"      Pressure bar        ";PMIDPOINT
 3150 INPUT"      Velocity m/s        ";MIDPOINT_VELOCITY
 3160 REM INPUT'      Air density kg/m3 ";MIDPOINT_RHOAIR
 3165 REM INPUT"      Susp.density kg/m3";MIDPOINT_RHOSUSP
 3170 PRINT
 3180 PRINT "Check data : o.k.(Y/N)?"
 3190 ANS$ = GET$:IF ANS$ = "N" THEN GOTO 3010
 3195 IF ANS$ <> "Y" THEN GOTO 3190
 3200 R=287,T=288
 3210 BENDOUT_RHOAIR = (PBENDOUTLET+1.01)*1E5/(R*T)
 3220 MIDPOINT_RHOAIR = (PMIDPOINT+1.01)*1E5/(R*T)
 3230 BENDOUT_RHOSUSP = BENDOUT_RHOAIR*MDOTP/MDOTA
 3240 MIDPOINT_RHOSUSP = MIDPOINT_RHOAIR*MDOTP/MDOTA
 3250 KSQD = PDROP*1E5/(BENDOUT_VELOCITY^2)

 3400 ENDPROC
 3700 DEF PROCENTER
 3710 A = OPENIN"SUMMARY"
 3720 PROCREAD_HEADING PROCCHECK
 3730 IF CHECK = 0 THEN GOTO 3990
 3750 ENTRY = VAL(NRUN$)-VAL(FIRSTRUN$)
 3760 PROCREAD_DATA
 3860 CLOSE#A
 3910 A = OPENOUT"SUMMARY
 3920 PROCCHANGE_DATA
 3990 CLOSE#A
 3995 ENDPROC
 4000 DEF PROCREAD_HEADING
 4010 PTR#A = 0 INPUT#A, VOLUME$
 4020 PTR#A = 5. INPUT#A, FIRSTRUN$
 4030 PTR#A =11 INPUT#A, LASTRUN$
 4031 FIRSTRUN = VAL(FIRSTRUN$): LASTRUN = VAL(LASTRUN$)
 4033 ENDPROC
 4037 DEF PROCCHECK
 4040 IF VAL(NRUN$)>=VAL(FIRSTRUN$) AND VAL(NRUN$)<=VAL(LASTRUN$) THEN CHECK = 1 ELSE CHECK = 0
 4050 IF CHECK = 0 THEN PRINT"Run doesn't belong in this summary file (Volume ";VOLUME$,") Out
side range ";FIRSTRUN$;" to ";LASTRUN$
 4060 ENDPROC
 4100 DEF PROCREAD_DATA
 4102 PROCREAD_HEADING
 4105 FOR NRUN = FIRSTRUN TO LASTRUN
 4107       ENTRY = NRUN-FIRSTRUN
 4110       PTR#A = (17 + ENTRY*180):INPUT#A,ARRAY$(1,ENTRY)
 4120       PTR#A = (23 + ENTRY*180):INPUT#A,ARRAY$(2,ENTRY)
 4130       PTR#A = (50 + ENTRY*180):INPUT#A,ARRAY$(3,ENTRY)
 4140       PTR#A = (67 + ENTRY*180):INPUT#A,ARRAY$(4,ENTRY)
 4150       PTR#A = (84 + ENTRY*180)
 4160       FOR M = 1 TO 14
 4170             INPUT#A,ARRAY(M,ENTRY)
 4180           NEXT M
 4190     NEXT NRUN
 4200 ENDPROC
 4300 DEF PROCSAVE_DATA
 4303 FOR NRUN = FIRSTRUNSUM TO LASTRUNSUM
 4307       ENTRYMAST = NRUN - FIRSTRUNMAST
 4308       ENTRYSUM = NRUN - FIRSTRUNSUM
 4310       PTR#A = (17 + ENTRYMAST*180):PRINT#A,ARRAY$(1,ENTRYSUM)
 4320       PTR#A = (23 + ENTRYMAST*180):PRINT#A,ARRAY$(2,ENTRYSUM)
 4330       PTR#A = (50 + ENTRYMAST*180):PRINT#A,ARRAY$(3,ENTRYSUM)
 4340       PTR#A = (67 + ENTRYMAST*180):PRINT#A,ARRAY$(4,ENTRYSUM)
 4350       PTR#A = (84 + ENTRYMAST*180)
 4360       FOR M = 1 TO 14
 4370             PRINT#A,ARRAY(M,ENTRYSUM)
 4380           NEXT M
 4390     NEXT NRUN
 4400 ENDPROC
 4500 DEF PROCCHANGE_DATA
 4505 ENTRY = VAL(NRUN$)-VAL(FIRSTRUN$)
 4510 PTR#A = (17 + ENTRY*180):PRINT#A, NRUN$
 4520 PTR#A = (23 + ENTRY*180):PRINT#A, BEND$
 4530 PTR#A = (50 + ENTRY*180):PRINT#A, PRODUCT$
```

Listing of "MANDATE" continued

```
4540 PTR#A = (67 + ENTRY*180):PRINT#A, LOOP$
4550 PTR#A = (84 + ENTRY*180)
4560 PRINT#A, MDOTA,MDOTP,PSUPP,PBENDOUTLET,BENDOUT_VELOCITY,BENDOUT_RHOAIR,BENDOUT_RHOSUSP,PD
ROP,KSQD,B,PMIDPOINT,MIDPOINT_VELOCITY,MIDPOINT_RHOAIR,MIDPOINT_RHOSUSP
4570 ENDPROC
4700 DEF PROCPRINT
4710 PRINT '"Run no ",NRUN$,"   Bend ";BEND$;":  Product ";PRODUCT$,"   Loop ";LOOP$
4715 PRINT
4720 PRINT"Mass flow of air kg/s ";MDOTA
4730 PRINT"Mass flow of product kg/s ";MDOTP
4740 PRINT"Supplementary pressure bar ";PSUPP
4750 PRINT
4760 PRINT"Bend outlet "
4770 PRINT"      Pressure bar         ";PBENDOUTLET
4780 PRINT"      Velocity m/s         ";BENDOUT_VELOCITY
4790 PRINT"      Air dens.kg/m3       ";BENDOUT_RHOAIR
4800 PRINT"      Susp. dens  kg/m3 ";BENDOUT_RHOSUSP
4810 PRINT
4820 PRINT"Bend pressure drop bar ";PDROP
4830 PRINT"k-squared factor        ";KSQD
4840 PRINT
4850 PRINT"Straight section:"
4860 PRINT"      dp/dx   bar/m        ";B
4870 PRINT"      Pressure bar         ";PMIDPOINT
4880 PRINT"      Velocity m/s         ";MIDPOINT_VELOCITY
4890 PRINT"      Air density kg/m3 ";MIDPOINT_RHOAIR
4900 PRINT"      Susp.density kg/m3";MIDPOINT_RHOSUSP
4910 PRINT
4920 ENDPROC
5000 DEF PROCHARD
5010 @% = &020308
5070 PROCSHORTSTRINGS
5085 INFO$ = BEN$+" "+LOO$+" "+PRO$
5090 VDU2:VDU21
5100 PRINT NRUN$,INFO$,MDOTA,MDOTP,PSUPP,PBENDOUTLET,BENDOUT_VELOCITY,BENDOUT_RHOAIR,BENDOUT_R
HOSUSP,PDROP,KSQD,B*(-1000),PMIDPOINT,MIDPOINT_VELOCITY,MIDPOINT_RHOAIR,MIDPOINT_RHOSUSP
5110 VDU6:VDU3
5160 ENDPROC
5300 DEF PROCCOLUMNHEAD
5305 PRINT"Print column headers (Y/N)";
5307 ANS$ = GET$:IF ANS$<>"Y" AND ANS$<>"N" THEN GOTO 5307
5309 PRINT ANS$:IF ANS$ = "N" THEN GOTO 5400
5320 VDU2:VDU21
5325 @% = &020308
5330 PRINT''" RUN"," INFO"," MASS FLOW RATES"," SUPP."," BEND OUTLET CONDITIONS ","DE
LTA-P"," KSQD ","GRAD. IN"," STRAIGHT PIPE CONDITIONS"
5340 PRINT" NO.","B L P"," AIR"," SOLIDS"," PRES."," PRES."," VELOC."," RHOAIR"," RHOSU
S."," BEND"," CONST.","STRAIGHT"," PRES."," VELOC."," RHOAIR"," RHOSUS."
5350 PRINT" "," "," kg/s"," kg/s"," barg"," barg"," m/s"," kg/m3"," kg/m3"," bar"," ","-
mbar/m"," barg"," m/s"," kg/m3"," kg/m3"'
5360 VDU6:VDU3
5400 ENDPROC
5600 DEF PROCSHORTSTRINGS
5605 PRO$ = "*"
5610 IF INSTR(PRODUCT$,"POLY")<>0 THEN PRO$ = "P"
5620 IF INSTR(PRODUCT$,"FLOUR")<>0 THEN PRO$ = "F"
5630 LOO1$ = "*"
5640 IF INSTR(LOOP$,"4")<>0 THEN LOO1$ = "4"
5650 IF INSTR(LOOP$,"3")<>0 THEN LOO1$ = "3"
5660 IF INSTR(LOOP$,"2")<>0 THEN LOO1$ = "2"
5670 LOO2$ = " "
5680 IF INSTR(LOOP$,"EXT")<>0 THEN LOO2$ = "E"
5690 IF INSTR(LOOP$,"EXP")<>0 OR INSTR(LOOP$,"STEP")<>0 THEN LOO2$ = "S"
5700 IF LOO1$ = "2" AND INSTR(LOOP$,"3")<>0 AND LOO2$ <> "S" THEN LOO1$ = "*": LOO2$ = "*"
5710 LOO$ = LOO1$ + LOO2$
5740 BEN1$ = "*"
5750 IF INSTR(BEND$,"SHORT")<>0 THEN BEN1$ = "S"
5760 IF INSTR(BEND$,"LONG")<>0 THEN BEN1$ = "L"
5770 IF INSTR(BEND$,"ELBOW")<>0 THEN BEN1$ = "E"
5780 IF INSTR(BEND$,"BLIND")<>0 THEN BEN1$ = "B"
5790 IF INSTR(BEND$,"VORT")<>0 THEN BEN1$ = "V"
5800 BEN2$ = " "
5810 IF INSTR(BEND$,"BOUGHT")<>0 THEN BEN2$ = "B"
5820 IF INSTR(BEND$,"MALE")<>0 THEN BEN2$ = "M"
5830 IF INSTR(BEND$,"FEMALE")<>0 OR INSTR(BEND$,"FEM")<>0 THEN BEN2$ = "F"
5840 IF INSTR(BEND$,"UNIONS")<>0 THEN BEN2$ = "U"
5850 IF INSTR(BEND$,"NO UNIONS")<>0 OR INSTR(BEND$,"WITHOUT")<>0 THEN BEN2$ = "N"
5870 BEN$ = BEN1$ + BEN2$
5890 ENDPROC
6000 DEF PROCKEY
6010 VDU2:VDU21
6020 PRINT'"Key to INFO column:"'
6030 PRINT" Bend type (B):-   First Digit                                    Second
Digit"
6040 PRINT"                   S - Short rad.      L - Long rad.              M - Ma
le          F - Female"
6050 PRINT"                   E - Elbow           B - Blind tee              U - Wi
th unions     N - Without unions"
6060 PRINT"                   V - Vortic-ell                                 B - Bo
ught-out"'
6070 PRINT" Loop (L):-        First Digit                                    Second
Digit"
6080 PRINT"                   Nominal bore in inches                         E - Ex
tended        S - Stepped"
6090 PRINT" Product (P):-     F - Flour           P - Polyethelene pellets"
6170 VDU6:VDU3
6200 ENDPROC
6300 DEF PROCFILETYPE
6310 IF FILETYPE$ = "SUMMARY" THEN FILETYPE$ = "MASTER" ELSE FILETYPE$ = "SUMMARY"
6320 ENDPROC
6400 DEF PROCTRANSFER
6403 PROCDRIVES
6406 PRINT"Insert disc with SUMMARY file and hit a key"':REPEAT:UNTIL GET$ <>""
6408 OSCLI SUMMDR$
6410 A = OPENIN"SUMMARY"
6420 PROCREAD_HEADING
6430 CLS: PRINT' Summary file (Volume ",VOLUME$," ) contains runs ";FIRSTRUN$," to ",LASTRUN$;
  O.K ?".
6440 ANS$ = GET$
6450 IF ANS$ <> "N" AND ANS$ <>"Y" THEN GOTO 6440
6460 PRINT ANS$ IF ANS$="N" THEN GOTO 6600
6470 PROCUNDEFINED
6480 IF CONT$ = "N" THEN GOTO 6600
6490 PROCREAD_DATA
6500 FIRSTRUNSUM = FIRSTRUN LASTRUNSUM = LASTRUN
6510 CLOSE#A
6520 PRINT"Insert disc with MASTER file and hit a key" REPEAT UNTIL GET$ <>""
6525 OSCLI MASTDR$
6530 A = OPENUP"MASTER"
6540 PROCREAD_HEADING:FIRSTRUNMAST = FIRSTRUN:LASTRUNMAST = LASTRUN
6550 PROCSAVE_DATA
6590 CLOSE#A
6600 ENDPROC
6700 DEF PROCUNDEFINED
6800 NOUNDEFINED = 0
6820 FOR NRUN = FIRSTRUN TO LASTRUN
6830       ENTRY = NRUN - FIRSTRUN
6840       PTR#A = (23 + ENTRY*180):INPUT#A, BEND$
6850       IF INSTR(BEND$,"BEND TYPE")<>0 THEN NOUNDEFINED = NOUNDEFINED + 1  UNDEFINED = 1 EL
SE UNDEFINED = 0
6860       IF NOUNDEFINED = 1 THEN PRINT''"The following runs are undefined -
6870       IF UNDEFINED <> 0 THEN PRINT" ",NRUN,
6880     NEXT NRUN
6882 IF NOUNDEFINED = 0 THEN PRINT "All runs defined".CONT$ = "Y":GOTO 6890
6884 PRINT''"Continue with transfer ?";
6886 CONT$ = GET$:IF CONT$<>"N" AND CONT$<>"Y" THEN GOTO 6886
6888 PRINT CONT$
6890 PRINT
6900 ENDPROC
7000 DEF PROCDRIVES
7010 INPUT"Drive no. for SUMMARY file; C for cat.:-";SDRIVE$
7020 IF SDRIVE$ = "C" THEN PROCCAT. GOTO 7010
7030 INPUT"Drive no. for MASTER file, C for cat.:-";MDRIVE$
7040 IF MDRIVE$ = "C" THEN PROCCAT: GOTO 7030
7050 MASTDR$ = "DR. " + MDRIVE$:SUMMDR$ = "DR. " + SDRIVE$
7060 ENDPROC
7100 DEF PROCCAT
7110 INPUT"Drive no ";DRIVE
7120 DRIVE$ = "DR. " + STR$(DRIVE)
7130 OSCLI DRIVE$
7140 *CAT
7150 ENDPROC
9000 REPEAT UNTIL GET$ <>""
```

APPENDIX H

CALIBRATION OF INSTRUMENTATION AND EQUIPMENT

There were two distinct pieces of equipment which needed calibration. Firstly there was the bank of choked flow nozzles, the actual air flow rates and critical pressure ratios of which needed to be determined, and secondly there was the data gathering equipment consisting of the data acquisition unit with the pressure transducers and load cells connected to it.

H.1 The choked flow nozzles

The difficulties of manufacturing the choked flow nozzles, with throat diameters of between 1.19 and 13.45mm, were expected to lead to small but significant errors on bore sizes, and thus to the actual flow rates being significantly different from the design values. Hence it was necessary to measure the true flow rate of each, as well as determining, for each nozzle, the critical pressure ratio (ratio of downstream/upstream pressures above which the flow rate would begin to drop off).

H.1.1 Equipment

It was decided to use an orifice meter to make the measurements, since this would be cheap, fairly easy to use, and the plates could be made accurately in house if not already available in the required sizes. Standards for manufacture and installation of orifice meters, and published data on their performance, was readily available in the form of British Standards publication 1042. This indicated an accuracy of measurement within less than 1%, which was thought to be quite close enough in the light of experience with pneumatic conveying systems which showed natural variations of operating conditions a good deal greater than this.

The recommendations of BS1042:1962 were followed in the manufacture and installation of the orifice meter, the length of the straight pipe upstream of the meter, and necessary sizes of plates (calculations based on the design flow rates of the nozzles). A valve was fitted at the beginning of the straight section in order to control the pressure downstream of the nozzle, and this assembly was made to fit on either of the outlets from the nozzle bank (i.e. that which would go to the blow tank and that which would go to the supplementary air injection point). The pressures at the tappings on the meter were monitored using one differential and one absolute water U-tube manometers, whilst those upstream and downstream of the nozzles were monitored using bourdon gauges. The set-up is shown in fig. H-1 below.

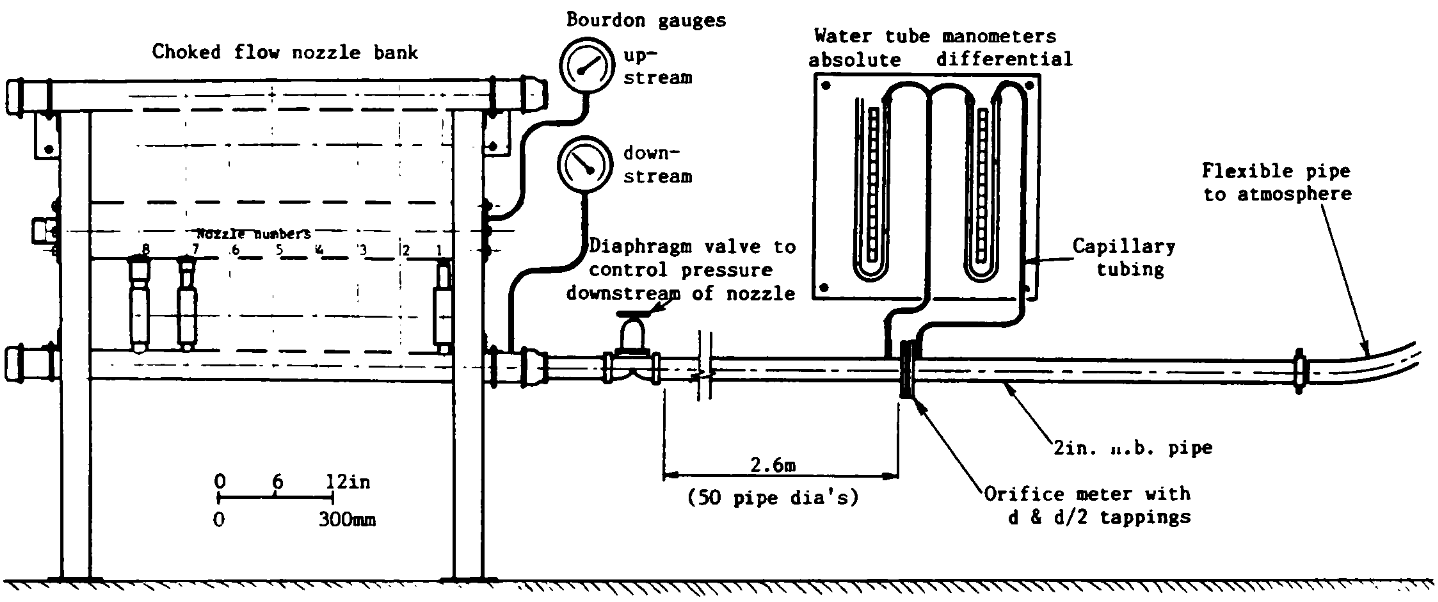


Fig. H-1
The set-up for measuring the air flow rates passed by the choked flow nozzles.

H.1.2 Procedure

The pressure upstream of the nozzles was set to 4.2 bar gauge (the value used for design of the nozzles) using the regulator and the bourdon gauge. This pressure was checked regularly, and the regulator reset if necessary although this was rarely required. Testing began with No. 1 Blow Tank nozzle, with the appropriate size of orifice plate in the meter. Initially the restricting valve was fully open, readings of the U-tubes being recorded. The restricting valve was closed down in stages to increase the pressure downstream of the nozzle, first to 1 barg then 2 and 3 barg, with U-tube readings repeated for each. It was suspected that flow would begin to fall off with much higher pressures, so smaller steps were then taken; at 3.2 bar no marked difference was observed, but the differential pressure across the meter dropped measurably at 3.44 bar and markedly at 3.8 bar.

H.1.3 Results

A graph of orifice meter differential pressure reading versus pressure downstream of the nozzle was drawn, shown below:-

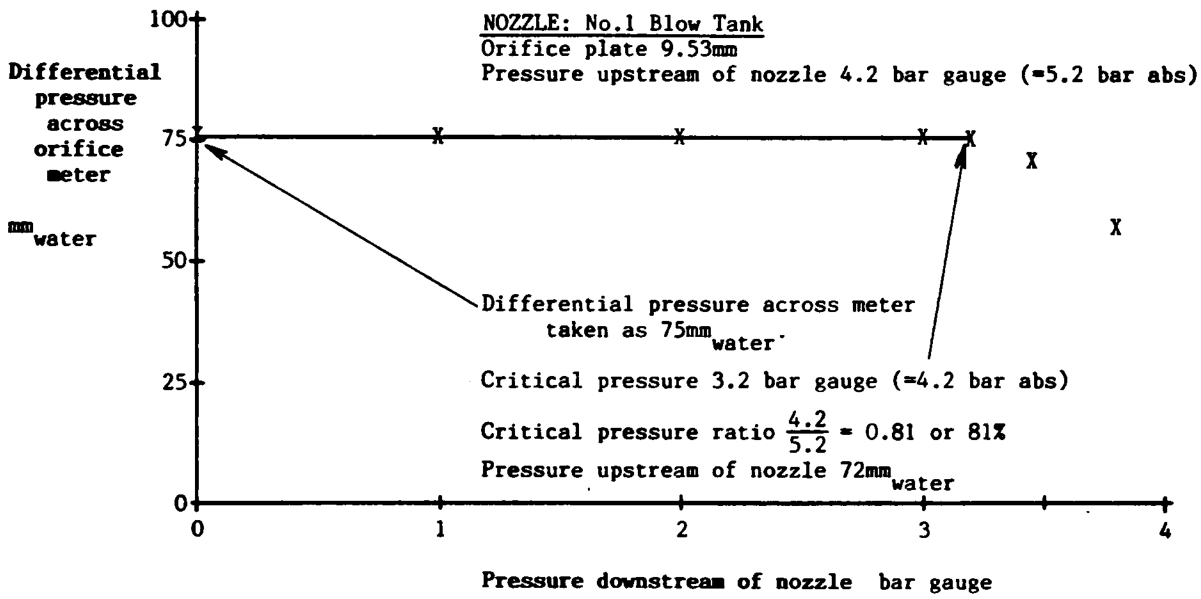


Fig. H-2

Graph of orifice meter differential pressure reading (indicating air flow rate) versus pressure downstream of the nozzle, for the no.1 blow tank nozzle.

From this, the downstream pressure up to which air flow remained constant was clearly seen to be 3.2 barg. Thus the limiting pressure ratio was calculated, using the upstream pressure of 4.2 barg, to be (overleaf)

$$\frac{(3.2 + 1.01) \text{ bar abs.}}{(4.2 + 1.01) \text{ bar abs.}} = 0.808 \text{ or } 81\%.$$

This was considered to be quite good, very much in line with the experience of other workers in the department who had designed and tested choked flow nozzles.

Using the data from the manometers and the equations and graphs of BS1042:1962, the flow rate up to the critical pressure ratio was calculated. The value was .00182 kg/s, some 40% up on the design value of .0013 kg/s at 4.2 barg upstream pressure. Although this appeared to be a considerable difference, reflection (after checking of calculations) showed that this would be accounted for by an increase in throat diameter of the nozzle of 18% (flow rate proportional to area), i.e. just 0.2mm on the design value of 1.19mm, not very much considering the difficulty in boring the convergent section without enlarging the throat.

The whole procedure was repeated with the other nozzles, which actually gave flow rates much closer to the design values (mostly within less than 8%), and critical pressure ratios between 77% and 85%. The results were:-

	Blow Tank		Supplementary	
Nozzle no.	Mass flow rate kg/s	Crit. pres. ratio %	Mass flow rate kg/s	Crit. pres. ratio %
1	.00182	80.8	.00142	76.9
2	.00288	86.5	.00281	80.8
3	.00519	80.8	.00531	76.9
4	.00979	76.9	.00963	76.9
5	.0207	76.7	.0215	78.8
6	.0449	82.7	.0446	80.8
7	.0786	82.7	.0772	80.8
8	.1617	84.7	.1610	78.8

Fig. H-3

Table of flow rates and critical pressure ratios for the choked flow nozzles. Upstream pressure 4.2 barg.

H.1.4 Variable upstream pressure test

In order to check the relationship between mass flow rate of air through the nozzles and upstream pressure, normally quoted to be in direct proportion, a test was carried out on one nozzle measuring the flow rate with a range of upstream pressures. The results are shown on the graph below, which gave confidence in using the accepted relationship.

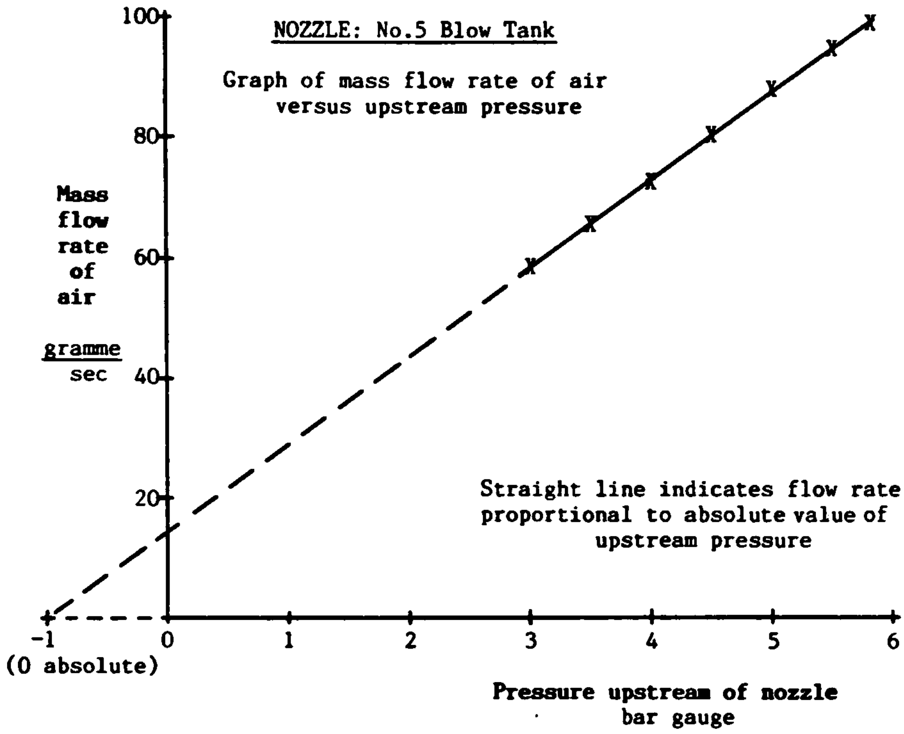


Fig. H-4

Results of variable upstream pressure test on nozzle no. 5 blow tank.

With the information above available, it was felt that enough was known about the performance of the choked flow nozzles to rely on their use for metering the conveying air.

H.2 The data gathering equipment

All necessary data was taken from the rig using the computer data gathering system; this consisted of a BBC Master microcomputer which communicated with a Mowlem Microsystems ADU intelligent data interface unit. This unit was connected to the 17 pressure transducers (as detailed in Appendix F) and the amplifier box for the load cells on which the receiving hopper was mounted. All of these channels needed calibration.

The data interface returned to the computer, on request, a value for each channel in data bits, this being a number between 0 and 2048 representing (in direct proportion) the voltage coming in on that channel. The relationship between voltage and pressure on the transducers, and also between voltage and weight on the load cells, were also known to be linear, with offsets likely. Therefore in order to convert readings in data bits into pressure in bar or weight in kgf, it would be necessary to use an equation of the form

$$\text{Value of quantity} = A0 + A1 \times \text{reading in data bits}$$

Where A0 and A1 are constants representing an offset and a constant of proportionality respectively, determined by calibration.

H.2.1 Pressure transducers

Each point of pressure measurement was allotted a specific transducer and a specific channel of the data interface, not to be interchanged. This was in recognition of the inevitable differences in calibration of the nominally identical transducers, and any possible differences in the characteristics of the analogue multiplexing channels directing the signals to the single analogue to digital converter in the interface.

The first step was to match the sensitivity of the channels on the data acquisition unit to the output of the transducers. The transducers were nominally 0 to 3.5 barg range, 10V input with a differential output of 100mV full scale. A stable 10V supply was available from the data acquisition unit. The gains of the amplifiers ahead of the analogue to digital converter in the unit were software configurable to discrete values from 2 to 2^{10}, and the range of the ADC was -10 to +10V differential, 10V being read as 2048 bits. A gain of 2^6, i.e. 64, was chosen as appropriate since this would give a nominal maximum input of 100mV x 64 = 6.4V, or 1311 bits. Accordingly the gain reference was set to 6 for all the pressure channels.

After taking a few readings it was clear that the calibration of each pressure channel would be different, at least in terms of offset. Therefore a separate set of data and calculation would be necessary for each channel.

H.2.1.1 Pressure standard

It was recognised that any available pressure standard against which calibration could be carried out, except possibly a primary standard (i.e. a dead-weight tester), would have significantly poorer linearity, hysteresis and repeatability than the combination of the transducers and analogue-to-digital converter, both rated to 0.1% or better combined accuracy. Since the transducers required calibration in-situ in the tappings (because the offset was known to be affected by tightening torque for this type of flush-diaphragm transducer), the use of a dead-weight tester was not a practical proposition.

Therefore it was expected that measured errors of linearity, hysteresis and repeatability would be on the part of the standard test gauge which was the only practical instrument to use for the calibration. Nevertheless the calibration would still be useful because the readings of all pressure channels could be taken simultaneously for each test pressure set up, and straight lines fitted in consistent manner. This would ensure as far as possible identical calibration for all channels, more important than absolute accuracy because they would be used to measure small differences in relatively large pressures (e.g. it was hoped to measure about 4 mbar differences in pressures around 2 barg).

The gauge used was Budenburg standard test gauge serial no. 10755060, 0-10 barg.

H.2.1.2 Procedure

With the transducers installed in their tappings in the conveying line and nozzle bank, and the conveying line blanked off at its end, air was admitted to the system to pressurise it. This was done using a hand operated diaphragm valve which was found to result in a more stable pressure than using the regulator. As the desired pressure was approached, the valve was closed down until the pressure just crept above the desired value, then the valve was closed a little further so that pressure would slowly fall. When the reading of the standard test gauge, tapped to overcome 'stiction', indicated the desired value (always on slowly falling pressure), a previously-defined test in the data acquisition unit was initiated. This test scanned all of the transducers instantaneously (i.e. within milliseconds) to obtain readings from them.

This procedure was repeated for pressures of 0, 0.5, 1.0, 1.5, 2.0 and 2.5 barg., the range over which it was intended to operate, to obtain six data points for every pressure information channel. The readings were of course in data bits, so to convert involved the drawing of a calibration graph for each channel, of pressure versus reading in data bits, then the fitting of a straight line to determine offset and gain coefficients for each channel.

H.2.1.3 Results

In order to keep a consistency of interpretation of these graphs, a computer program was written to draw them one at a time on the screen and fit a line using the method of least squares, then calculate the coefficients. A typical display is shown overleaf:-

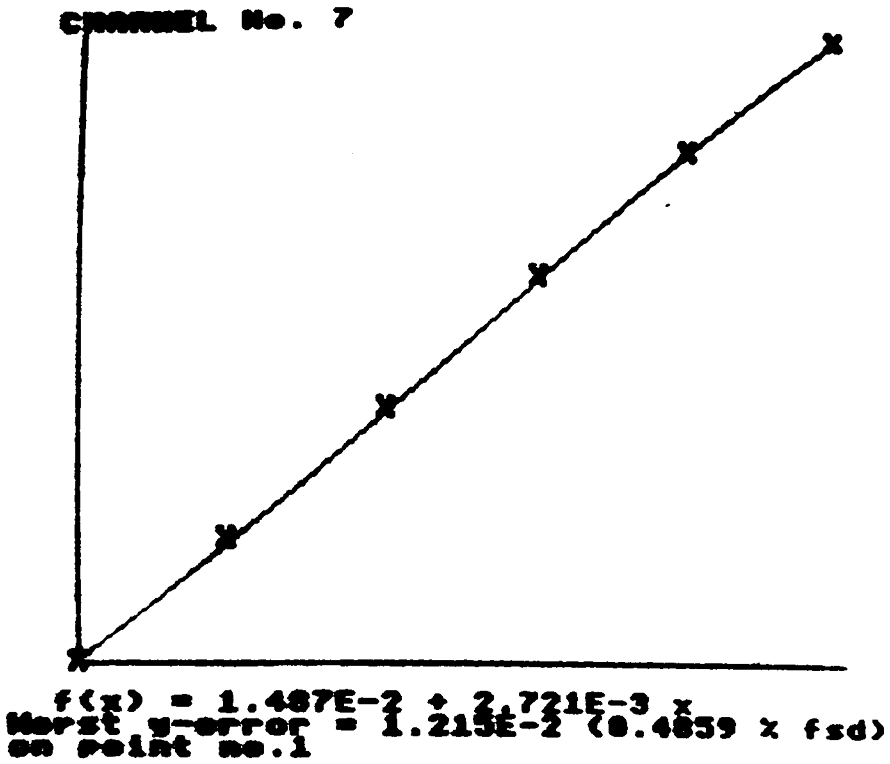


Fig H-5

A pressure channel calibration graph, as displayed by the calibration program.

The two coefficients A0 (the offset) and A1 (the gain) could then be used to determine the actual pressure at any transducer from the channel reading in data bits using

$$\text{Pressure} = A0 + A1 \times \text{Channel reading in bits}$$

This procedure was repeated for all of the channels of pressure measurement.

The coefficients varied slightly from channel to channel, the offset being more subject to variation than the gain. Typical values would be offset (A0) = -3 to $+5 \times 10^{-2}$, gain (A1) = 2.68 to 2.72×10^{-3} bar per bit.

H.2.2 Load cell

The calibration of the load cell channel was carried out in exactly the same way as that of the pressure channels except that the test inputs were known weights placed on the top of the receiving hopper. The known weights were in the form of people, previously weighed on a set of calibrated scales. The range of the calibration was from zero to 171.7 kg, the analogue gain reference of the channel on the data acquisition unit being defined as 0, i.e. a gain of 2 = 1 since the full scale output of the load cell readout box was 10V.

H.2.2.1 Results

The calibration coefficients were determined in exactly the same way as for the pressure channels, the offset being 0.785 kg and the gain 0.611kg per bit.

H.3 Recalibration

The calibration of the data gathering channels were re-checked several times over a period of days after the initial calibration, and not found to vary significantly, i.e. not by more than 1 bit. At intervals during the first several months of service, the calibration was again re-checked, with no significant variation found. However, the calibration of the pressure channels was found to alter once the transducers were removed from the tappings on the first conveying line used and placed in the next line used. This was hardly surprising given that the measuring diaphragm was located flush with the end of the thread so that tightening torque would undoubtedly strain it slightly. Consequently, the calibration exercise was repeated every time the transducers were disturbed for any reason.

APPENDIX I

EXECUTION OF THE TEST PROGRAMME

I.1 Introduction

This appendix details the way in which testing proceeded and the decisions which were taken along the way, prompted by findings from the work as it progressed.

I.2 Test procedure

As the work got under way, a strategy for conducting the tests emerged. For each bend tested, a start was made by selecting a fairly high air flow rate which would be expected to give an air velocity of the order of 15 or 20m/s: a low blow tank air ratio (percentage of total air flow rate going to the blow tank) would be chosen, say 10%, and the necessary combinations of nozzles on the choked flow nozzle bank calculated with the help of the computer and set up. This first test would then result in a data point for a low solids flow rate and low suspension density. Testing would proceed by increasing the blow tank air ratio, giving higher suspension densities and lower velocities (because of air compression), then once the highest possible solids flow rate had been reached (governed by the critical pressure ratio of the choked flow nozzles), a new air flow rate would be chosen. Flow rates both higher and lower than the first would be employed, to cover air velocities from close to the minimum conveying velocity for the product right up to those which would be considered excessive in a properly designed system. For each flow rate of air, the widest range possible of solids flow rates would be covered, again by varying the blow tank air ratio.

I.3 First seven sets of tests - flour in the 2in. line

The first 68 test runs, using the short radius bend without unions, (bend no. 3 in fig. F-3, App. F) covered a range of air velocities from about 8 to 21 m/s, with solids loading ratios of between 1 and 80. The lower limit on air velocity was determined by a desire to avoid blocking the pipeline, whilst the upper limit was thought to be the maximum which might be used in a well designed pneumatic conveying system. The limits on solids loading ratio were set by the blow tank feeder, the lowest rate of reliable operation fixing the bottom limit, the top limit being set by the maximum pressure available without infringing the critical pressure ratio of the choked flow nozzles on the air supply. This was considered to be a sufficiently wide range for one bend so it was decided to move on to another.

The second bend tested was one of identical radius and construction but with unions fitted, in the hope that the effect of these might be isolated (bend no. 2 in fig. F-3, App. F). This gave a very similar pattern of results, no major differences in pressure drop values being apparent at this stage.

Next the long radius bends both with and without unions (bends 4 and 5 in fig. F-3, App. F) were used, the one with unions being tested with flow in both directions to look for any signs of effects caused by some slight misalignment on one of the unions. Again similar patterns of results were obtained. The programme continued with testing of the male and female malleable elbow fittings and the blind tee (bends 6a, 6b, 7 in fig. F-3, App. F), the same ranges of flow conditions being covered as for the first bend in all cases; some differences in the pattern of results emerged with these bends, which will be described later.

I-4 Check for change in product

By this time some 332 test runs had been completed over a time period of five weeks, with the same 600kg batch of wheat flour used for the whole period. Although the product had been conveyed a number of times beforehand in another rig, it was felt that some change in the product was likely under these conditions, through either particle attrition, changing moisture content or biological action. In order to assess the effect on measured data, the conditions of the first set of test runs (nos. 1 to 68) were recreated, with the same bend, and another series of 20 runs covering the same ranges of flow conditions were carried out. No marked differences in results were apparent at this stage although later detailed analysis was to reveal some differences, as discussed in Chapter 3.

I-5 Air density - first attempt

A question which had been considered during planning of the experimental work, but been laid aside at that stage, was that of the effect of air density. With different flow conditions causing different pressure drop along the pipeline, the absolute pressure and hence density of the air at the test bend was not constant, varying by a factor of about three between extremes of conditions. Without a knowledge of the effect on pressure drop which this may have, it was possible that the changing pressure drop could be wrongly attributed to other flow variables.

Careful consideration showed that it would be possible to gain some control over the air density at the test bend by means of increasing or decreasing the resistance of the pipeline further downstream, thus artificially raising or lowering the absolute air pressure at the test bend. Using the same bend as used for the first set of test runs (the short radius without unions, also used again to assess the effect of product change), the first attempt at doing this used a restrictor in the form of a ten foot length of 1½in. n.b. pipe, inserted inside the 2in. n.b. pipeline in the straight immediately after the test sections. Comparing results after 52 test runs (nos. 353 to 405), the increase in air density at the bend, for conditions of similar product flow rate and air velocity, was of the order of just five to ten per cent which was not considered to be a sufficient range on which to base a correlation.

I-6 Vortice-ell

Whilst considering how to proceed with the air density experiments, another bend was tested with the standard loop. This was the Hammertek "Vortice-ell" bend (no. 8 in fig. F-3, App. F), a proprietary type of cast bend specifically designed to resist wear. The pattern of results appeared to be similar to those from the blind tee.

I-7 Air density - second attempt

Consideration of the problem of controlling air density showed that it was unlikely that any simple means of restricting the flow could provide sufficient pressure drop without causing pipeline blockage. An alternative which could give any required increase in resistance was apparent, in terms of increasing pipeline length. To achieve this, the vertical riser at the end of the test loop was diverted into an existing loop of 2in. n.b. pipe which added approximately 58.5m (including 16m vertically up and the same down, plus nine bends) on to the original distance of m from the test bend to the receiving hopper, an increase of %. The resulting increase in resistance gave an air density increase of between 40% and 80% which was considered much more satisfactory although the outcome of this work would not be examined in detail until later. It was apparent that another variation would be possible by reducing the resistance of the pipeline downstream of the test section; although it could not be shortened, a larger diameter pipe would be possible which would achieve this. It was decided that this may be attempted once the standard 2in. n.b. loop was entirely finished with.

I-8 Bought-out bend

One final type of bend was to be tested which had not so far been used; this was a proprietary bend of short radius (short by the standards of pneumatic conveyors; described as long radius by the manufacturers of pipeline fittings, since it is about the longest type used in normal fluid systems - bend no. 1 in fig. F-3, App. F). This was thought to be most important since it was available off the shelf from stockists, much cheaper and more convenient to install than the long radius bends usually specified in pneumatic conveyors. An effort was made to cover the widest range of flow conditions achievable with this, air velocities being extended down to under 5m/s and up to nearly 50m/s, with solids loading ratios of up to 106. Again the data displayed a very similar pattern to that for the other radiused bends.

I-9 One set of tests with the second product

The next step was to employ the second product, polyethylene pellets, in a programme of work with the proprietary short radius bend. Again the widest range of flow conditions achievable were covered, giving air velocities from 4 to 40 m/s and solids loading ratios of from 1 to about 30 in 79 test runs. The pellets were examined carefully before and after conveying and no change in particle shape could be perceived, therefore it was thought unnecessary to investigate any effects of product change. After this, the rig was emptied of the pellets and recharged with the original batch of wheat flour, which would be used for the remainder of the work.

I-10 Review of achievements and consideration of how to proceed

At this stage, the work already done was reviewed. A comprehensive set of data had been obtained for the behaviour of the wheat flour in bends of all common types, under all practicable flow conditions, for a single pipeline bore size. A similarly comprehensive set of data had been generated with the polyethylene pellets but relating to only one bend. Also a set of data for the flour flowing through one of the bends with air densities different from those in the first set, had been obtained. It was quite clear that in order to get the most out of this work, the question of pipeline diameter would have to be addressed; this would entail rebuilding the pipeline loop in at least another one, or preferably two, pipe diameters. A third set of data relating to air density variation would also be very welcome, which could be obtained first by installing the larger pipes only downstream of the existing 2in. n.b. test sections. Therefore the decision was taken to first enlarge the pipe here to 3in. n.b. for this work, then to replace the existing test sections with 3in. sections and test probably two bends, and finally go to 4in. for the test sections if time and resources allowed. Only one product, the wheat flour, would be used for this.

I-11 Air density - completion

The next step therefore was the rebuilding of the 2in. n.b. return line downstream of the test sections, using 3in. pipe. With this done, another full set of tests was undertaken using the short radius bend without unions (which had been used for the first set of tests, the increased air density tests and the tests to measure effect of product change) to measure the effect of reduced air density. Air densities mostly about 25% lower than in the first series of tests were achieved. The pattern of results did not appear markedly different from that observed in the other series of tests, and detailed analysis was deferred until later.

I-12 Changes in pipeline bore - 3in. and 4in. nominal bore

The 2in. n.b. test sections were replaced with 3in. sections, an identical arrangement of pressure tappings to those on the 2in. sections being used. The discharge line from the blow tank being a fixed installation of 2in. n.b., this size of line was used up to and including the bend before the long straight incorporating the first test section, with an expansion immediately after this bend as shown in fig. I-1 overleaf. Re-calibration of the pressure transducers showed only a small change in gain and offset, but nevertheless the values in the data processing software were revised accordingly.

Two bends were tested in this loop, a long (660mm) radius one and a female malleable elbow. They were both installed using unions at the same distance from the apex of the bend, since the work done with the 2in. loop did not appear to show any very significant effect caused by these. Testing over the same ranges of conveying conditions as used for the 2in. bends gave patterns of pressure drop results for both bends which were very similar to those obtained from the corresponding 2in. bends, although the actual values with the elbow were noticeably higher with the 3in. than with the 2in. Detailed examination was again deferred until later.

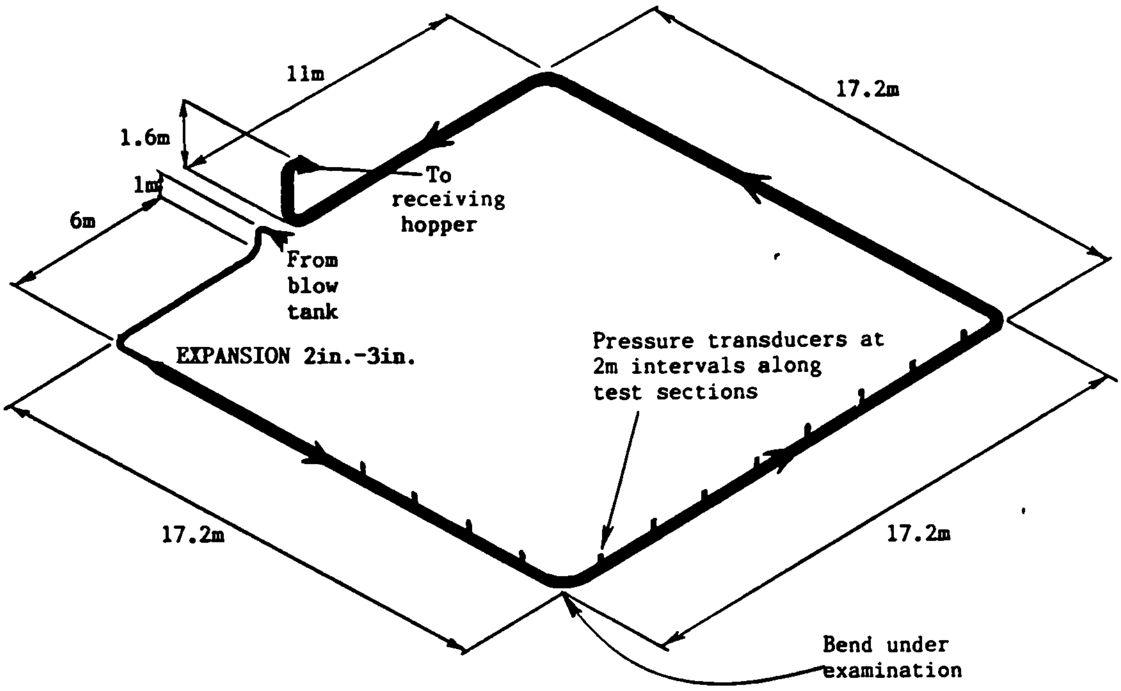


Fig. I-1
The test loop incorporating 3in. n.b. test sections.

The 3in. line was removed and replaced with a 4in. n.b. line, again with exactly the same arrangement of pressure tappings, and with the expansion from 2in. at the same place. Re-calibration was performed as before, with again only a small variation in constants being observed. The 4in. line was assembled using flanges instead of unions, because experience in using 4in. unions on other lines in the laboratory had shown them to be very difficult to get undone and to re-seal after a period of service.

I-13 Problems with pipe expansion effects

Only one type of bend, a long (660mm) radius, was used with the 4in. line. This was fitted with flanges as used on the rest of the loop. The first twenty or so test runs showed up a peculiarity in that no pressure loss in the straight length before the bend could be detected, the measured gradient being zero. The transducers were re-calibrated and the tests re-run but with the same result. Consideration showed that a possible explanation might lie in the expansion from 2in. to 4in. n.b. at the start

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With this arrangement, the pressure gradient in the test section before the bend measured very like that towards the end of the next test section, downstream of the zone where the bend pressure drop developed. Thus it was thought likely that the explanation advanced above was probably justified, and useful measurements could be made.

The upper limit on air velocity was somewhat restricted with the 4in. line, because of compressor capacity, and solids loading ratios were limited by the ability of the blow tank to feed the line through its 2in. discharge; 60 tests covered a ramge from 5 to 20 m/s with s.l.r.'s from 6 to 82. Very similar patterns of results to those observed for the 2in. and 3in. long radiused bends emerged.

I-15 Conclusion of test programme

This last set of tests using the 4in. loop concluded the test work, which finally amounted to some 910 conveying runs with 2 products, 3 pipe sizes, 5 different pipe loops and 12 different bends.

APPENDIX J

PRINTOUT OF TEST DATA

J.1 Introduction

The data listed here has already been subjected to the Primary Data Processing described in Chapter 2 and Appendices K and G. It is in the form of one line for each test run, giving the conveying conditions at outlet of the bend and at the centre of the straight pipe downstream of the bend, the pressure drop caused by the bend (and calculated loss coefficient) and the pressure gradient in the straight section. Although this does not permit examination or re-interpretation of the data from first principles, i.e. the selection of the steady state portion from the conveying cycle and the fitting of the tangents to pressure points upstream and downstream of the bend, it is felt to be more useful in this form. In any case, the raw data occupies some 7 megabytes of disc space; this would cover about 7000 pages of paper even if reduced in size to the same degree as what follows.

Key to interpretation of INFO column on printouts:-

Bend type (B):-	First Digit		Second Digit	
	S - Short rad.	L - Long rad.	M - Male	F - Female
	E - Elbow	B - Blind tee	U - With unions	N - Without unions
	V - Vortic-ell		B - Bought-out	
Loop (L):-	First Digit		Second Digit	
	Nominal bore in inches		E - Extended	S - Stepped
Product (P):-	F - Flour	P - Polyethelene pellets		

J.2 General information

Runs nos.	Bend (x) refers to fig. F-6	Product (Flour/Pellets)	
	2in. nominal bore:-		
1 to 68	Short rad., no unions (3)	F	
69 to 117	Short rad. with unions (2)	F	
118 to 162	Long rad., no unions (5)	F	
163 to 178	Long rad. with unions (4)	F	
179 to 190	as above, reversed (4)	F	
191 to 235	Male malleable elbow (6a)	F	
256 to 281	Female malleable elbow (6b)	F	
282 to 332	Blind tee	F	
333 to 405	Short rad., no unions (3)	F	insert in pipe
406 to 487	Short rad., no unions (3)	F	loop return extended (higher air density)
488 to 581	Short rad. bought-out (1)	F	
582 to 660	Short rad. bought-out (1)	P	
661 to 714	Short rad., no unions (3)	F	Loop return enlarged (lower air density)
	3in. nominal bore:-		
715 to 761	Long rad. with unions (1)	F	
762 to 808	Female malleable elbow (2)	F	
	4in. nominal bore:-		
809 to 841	Long rad. with flanges	F	Data no good; expansion too close to test section
842 to 902	Long rad. with flanges	F	Expansion moved back

RUN NO.	INFO B	L	P	MASS FLOW RATES AIR kg/s	SOLIDS kg/s	SUPP. PRES. barg	BEND OUTLET CONDITIONS PRES. barg	VELOC. m/s	RHOAIR kg/m3	RHOSUS. kg/m3	DELTA-P BEND bar	LOSS COEFF.	GRAD. IN STRAIGHT -mbar/m	STRAIGHT PIPE CONDITIONS PRES. barg	VELOC. m/s	RHOAIR kg/m3	RHOSUS. kg/m3
1	SN	2	F	0.087	1.080	1.111	0.626	19.922	1.979	24.563	0.118	2.421	2.356	0.600	20.247	1.948	24.170
2	SN	2	F	0.085	0.703	0.808	0.438	22.042	1.752	14.451	0.021	0.591	5.742	0.368	23.156	1.668	13.756
3	SN	2	F	0.086	1.385	1.187	0.662	19.229	2.022	32.651	0.122	2.027	2.317	0.634	19.558	1.988	32.102
4	SN	2	F	0.085	1.584	1.297	0.686	18.885	2.052	38.013	0.127	1.880	2.170	0.660	19.183	2.020	37.423
5	SN	2	F	0.087	1.290	1.202	0.662	19.495	2.023	29.992	0.131	2.301	2.556	0.634	19.832	1.988	29.482
6	SN	2	F	0.085	1.971	1.471	0.791	17.745	2.179	50.340	0.162	2.044	2.576	0.760	18.059	2.141	49.466
7	SN	2	F	0.085	2.180	1.563	0.874	16.983	2.279	58.174	0.146	1.745	2.964	0.841	17.285	2.239	57.155
8	SN	2	F	0.085	2.323	1.628	0.891	16.831	2.300	62.566	0.154	1.743	3.157	0.856	17.148	2.257	61.409
9	SN	2	F	0.087	2.574	1.755	0.983	16.317	2.411	71.509	0.156	1.635	3.912	0.939	16.682	2.358	69.947
10	SN	2	F	0.086	2.544	1.737	0.954	16.465	2.376	70.026	0.167	1.757	4.883	0.890	17.021	2.298	67.740
11	SN	2	F	0.086	2.704	1.796	1.022	15.835	2.459	77.407	0.151	1.560	6.244	0.947	16.448	2.367	74.522
12	SN	2	F	0.085	2.666	1.728	0.981	16.071	2.409	75.189	0.140	1.444	5.244	0.922	16.557	2.338	72.985
13	SN	2	F	0.086	3.118	1.974	1.165	14.780	2.631	95.621	0.128	1.225	7.438	1.104	15.203	2.558	92.963
14	SN	2	F	0.085	0.122	0.361	0.216	26.096	1.483	2.123	0.012	1.613	2.264	0.193	26.594	1.456	2.083
15	SN	2	F	0.064	0.830	0.719	0.379	17.179	1.681	21.902	0.058	1.787	1.826	0.359	17.434	1.656	21.582
16	SN	2	F	0.064	0.412	0.490	0.264	18.756	1.542	9.968	0.039	2.223	2.227	0.237	19.162	1.509	9.757
17	SN	2	F	0.064	1.243	0.907	0.478	16.134	1.801	34.909	0.080	1.770	2.543	0.448	16.476	1.763	34.185
18	SN	2	F	0.064	1.421	0.991	0.527	15.530	1.859	41.469	0.088	1.763	2.275	0.501	15.790	1.829	40.786
19	SN	2	F	0.064	1.533	1.044	0.553	15.341	1.891	45.291	0.099	1.854	2.394	0.529	15.582	1.862	44.588
20	SN	2	F	0.064	1.923	1.212	0.654	14.410	2.013	60.502	0.113	1.802	3.209	0.625	14.668	1.978	59.437
21	SN	2	F	0.064	2.131	1.322	0.722	13.799	2.096	70.004	0.107	1.611	4.458	0.681	14.132	2.046	68.359
22	SN	2	F	0.065	2.399	1.437	0.807	13.406	2.198	81.097	0.109	1.498	6.352	0.755	13.799	2.135	78.791
23	SN	2	F	0.061	2.462	1.477	0.815	12.606	2.208	88.530	0.139	1.981	6.593	0.761	12.987	2.143	85.929
24	SN	2	F	0.063	2.790	1.621	0.959	11.950	2.382	105.843	0.085	1.121	10.987	0.859	12.592	2.261	100.450
25	SN	2	F	0.062	3.036	1.698	1.024	11.475	2.461	119.931	0.073	0.924	11.764	0.905	12.189	2.317	112.904
26	SN	2	F	0.063	3.239	1.762	1.088	11.197	2.538	131.109	0.050	0.613	15.170	0.949	11.989	2.371	122.453
27	SN	2	F	0.063	3.554	1.886	1.154	10.978	2.618	146.752	0.098	1.109	13.300	1.046	11.555	2.487	139.419
28	SN	2	F	0.044	0.524	0.485	0.272	12.739	1.551	18.628	0.037	2.445	2.274	0.249	12.972	1.524	18.293
29	SN	2	F	0.045	0.271	0.282	0.155	14.468	1.410	8.479	0.028	3.130	1.172	0.142	14.632	1.394	8.384
30	SN	2	F	0.044	0.672	0.568	0.311	12.477	1.598	24.420	0.041	2.164	2.640	0.284	12.735	1.566	23.925
31	SN	2	F	0.043	0.703	0.578	0.344	11.924	1.638	26.714	0.025	1.317	4.297	0.309	12.240	1.596	26.025
32	SN	2	F	0.043	1.158	0.814	0.473	10.989	1.794	47.771	0.032	1.112	5.881	0.419	11.402	1.729	46.041
33	SN	2	F	0.043	1.412	0.945	0.530	10.583	1.863	60.485	0.063	1.862	5.544	0.485	10.902	1.809	58.715
34	SN	2	F	0.042	1.442	0.949	0.560	10.021	1.900	65.237	0.044	1.336	7.560	0.484	10.534	1.807	62.055
35	SN	2	F	0.044	1.660	1.040	0.614	10.081	1.965	74.622	0.050	1.306	8.386	0.538	10.580	1.872	71.104
36	SN	2	F	0.045	0.735	0.556	0.327	12.578	1.618	26.493	0.049	2.319	3.058	0.303	12.816	1.588	26.001
37	SN	2	F	0.043	0.811	0.623	0.356	11.931	1.653	30.821	0.052	2.387	3.586	0.323	12.224	1.613	30.082
38	SN	2	F	0.063	3.511	1.848	1.140	10.978	2.601	144.983	0.081	0.928	13.552	1.016	11.649	2.451	136.639
39	SN	2	F	0.064	3.960	2.009	1.272	10.558	2.760	169.998	0.049	0.517	17.521	1.112	11.355	2.567	158.079
40	SN	2	F	0.063	3.928	1.966	1.257	10.396	2.743	171.267	0.039	0.417	17.835	1.094	11.200	2.546	158.964
41	SN	2	F	0.062	4.082	2.009	1.301	10.034	2.796	184.411	0.043	0.469	17.970	1.137	10.800	2.598	171.321
42	SN	2	F	0.064	4.127	2.030	1.305	10.373	2.801	180.347	0.047	0.485	17.674	1.144	11.150	2.606	167.776
43	SN	2	F	0.064	4.208	2.036	1.330	10.264	2.831	185.850	0.033	0.341	18.867	1.158	11.080	2.622	172.168
44	SN	2	F	0.064	4.464	2.079	1.342	10.194	2.846	198.470	0.056	0.546	17.248	1.185	10.926	2.655	185.181
45	SN	2	F	0.064	4.608	2.127	1.395	9.970	2.910	209.488	0.062	0.593	18.092	1.230	10.705	2.710	195.100
46	SN	2	F	0.063	4.602	2.104	1.397	9.775	2.912	213.388	0.052	0.508	19.779	1.236	10.474	2.718	199.133
47	ABORTED																
48	SN	2	F	0.063	4.665	2.099	1.381	9.856	2.893	214.518	0.037	0.358	18.618	1.230	10.523	2.709	200.937
49	SN	2	F	0.064	5.033	2.157	1.470	9.684	3.000	235.583	0.038	0.340	18.752	1.317	10.319	2.816	221.100
50	SN	2	F	0.085	3.218	1.888	1.087	15.135	2.537	96.386	0.141	1.276	7.387	1.019	15.638	2.455	93.285
51	SN	2	F	0.085	3.732	2.107	1.273	13.946	2.763	121.296	0.108	0.917	11.998	1.152	14.731	2.616	114.839
52	SN	2	F	0.086	4.068	2.188	1.353	13.571	2.859	135.857	0.108	0.864	14.302	1.237	14.274	2.718	129.172
53	SN	2	F	0.086	4.365	2.272	1.455	13.041	2.982	151.713	0.080	0.619	15.973	1.325	13.767	2.825	143.720
54	SN	2	F	0.084	5.095	2.457	1.591	12.142	3.147	190.212	0.071	0.503	19.039	1.417	13.011	2.937	177.500
55	SN	2	F	0.086	5.119	2.415	1.562	12.485	3.111	185.839	0.076	0.527	18.543	1.392	13.365	2.907	173.605
56	SN	2	F	0.044	1.908	1.134	0.661	9.909	2.022	87.287	0.047	1.101	9.868	0.571	10.473	1.913	82.582
57	SN	2	F	0.042	1.999	1.186	0.705	9.241	2.075	98.041	0.041	0.975	10.047	0.623	9.703	1.976	93.371
58	SN	2	F	0.045	2.253	1.290	0.763	9.426	2.145	108.348	0.059	1.223	10.463	0.657	10.025	2.016	101.870
59	SN	2	F	0.044	2.363	1.336	0.805	8.998	2.196	119.016	0.048	0.993	11.038	0.705	9.527	2.074	112.409
60	SN	2	F	0.042	2.529	1.410	0.877	8.378	2.283	136.810	0.019	0.389	13.905	0.750	8.982	2.130	127.606
61	SN	2	F	0.043	2.696	1.475	0.920	8.268	2.336	147.808	0.030	0.601	13.905	0.793	8.850	2.182	138.087
62	SN	2	F	0.123	1.418	1.558	0.877	24.415	2.284	26.321	0.153	1.951	3.534	0.835	24.982	2.232	25.723
63	SN	2	F	0.123	1.137	1.363	0.770	25.883	2.154	19.905	0.106	1.592	4.524	0.720	26.637	2.093	19.342
64	SN	2	F	0.123	0.811	1.092	0.631	28.081	1.985	13.095	0.059	1.141	4.695	0.579	29.005	1.922	12.677
65	SN	2	F	0.123	1.627	1.700	0.954	23.467	2.376	31.420	0.173	1.996	4.042	0.909	24.018	2.321	30.700
66	SN	2	F	0.124	2.271	2.041	1.154	21.470	2.618	47.951	0.212	1.919	3.905	1.110	21.910	2.565	46.988
67	SN	2	F	0.122	2.541	2.145	1.224	20.457	2.703	56.304	0.217	1.844	4.231	1.177	20.898	2.646	55.117
68	SN	2	F	0.123	3.137	2.430	1.411	19.032	2.929	74.712	0.221	1.631	5.937	1.345	19.566	2.849	72.673

RUN NO.	INFO B	L	P	MASS FLOW RATES AIR kg/s	SOLIDS kg/s	SUPP. PRES. barg	BEND OUTLET CONDITIONS PRES. barg	VELOC. m/s	RHOAIR kg/m3	RHOSUS. kg/m3	DELTA-P BEND bar	LOSS COEFF.	GRAD. IN STRAIGHT -mbar/m	STRAIGHT PIPE CONDITIONS PRES. barg	VELOC. m/s	RHOAIR kg/m3	RHOSUS. kg/m3
69	SU	2	F	0.044	0.453	0.421	0.233	13.145	1.503	15.629	0.024	1.799	2.440	0.203	13.465	1.468	15.257
70	SU	2	F	0.044	0.581	0.497	0.272	12.917	1.551	20.371	0.051	3.014	2.028	0.249	13.149	1.524	20.013
71	SU	2	F	0.043	0.838	0.640	0.365	11.742	1.664	32.338	0.039	1.750	3.918	0.333	12.020	1.625	31.589
72	SU	2	F	0.043	0.991	0.700	0.388	11.662	1.691	38.532	0.039	1.480	4.114	0.350	11.984	1.645	37.496
73	SU	2	F	0.045	1.151	0.764	0.448	11.438	1.763	45.608	0.040	1.335	5.541	0.397	11.849	1.702	44.025
74	SU	2	F	0.044	1.718	0.997	0.586	10.257	1.931	75.908	0.034	0.844	8.077	0.512	10.754	1.842	72.400
75	SU	2	F	0.044	1.923	1.099	0.640	10.039	1.996	86.840	0.042	0.962	8.160	0.549	10.624	1.886	82.058
76	SU	2	F	0.042	1.384	0.877	0.507	10.371	1.836	60.479	0.038	1.171	5.785	0.472	10.619	1.793	59.065
77	SU	2	F	0.042	1.968	1.132	0.666	9.457	2.027	94.315	0.033	0.789	9.704	0.568	10.046	1.909	88.782
78	SU	2	F	0.044	2.425	1.304	0.787	9.090	2.174	120.930	0.036	0.712	11.189	0.685	9.638	2.051	114.055
79	SU	2	F	0.043	2.568	1.351	0.823	8.705	2.218	133.728	0.044	0.867	11.368	0.708	9.288	2.079	125.329
80	SU	2	F	0.064	0.311	0.385	0.221	19.412	1.490	7.253	0.016	1.175	1.666	0.208	19.628	1.473	7.174
81	SU	2	F	0.064	0.642	0.588	0.334	17.762	1.626	16.374	0.019	0.718	2.752	0.311	18.063	1.598	16.101
82	SU	2	F	0.064	1.180	0.845	0.468	16.245	1.789	32.923	0.055	1.263	2.650	0.444	16.515	1.759	32.384
83	SU	2	F	0.064	1.493	0.978	0.531	15.560	1.864	43.499	0.064	1.221	2.790	0.503	15.850	1.830	42.701
84	SU	2	F	0.064	1.813	1.104	0.601	14.881	1.949	55.222	0.082	1.349	3.328	0.568	15.199	1.909	54.066
85	SU	2	F	0.062	2.261	1.275	0.697	13.720	2.065	74.699	0.089	1.268	5.062	0.645	14.145	2.003	72.455
86	SU	2	F	0.063	2.701	1.453	0.852	12.638	2.252	96.875	0.054	0.702	9.304	0.776	13.173	2.161	92.938
87	SU	2	F	0.063	3.074	1.593	0.954	11.964	2.376	116.460	0.060	0.725	11.735	0.858	12.575	2.260	110.801
88	SU	2	F	0.063	3.476	1.719	1.052	11.446	2.495	137.642	0.046	0.506	13.592	0.915	12.265	2.328	128.452
89	SU	2	F	0.063	3.798	1.798	1.127	11.028	2.585	156.092	0.062	0.655	14.054	1.013	11.651	2.447	147.746
90	SU	2	F	0.063	3.334	1.688	1.033	11.625	2.472	129.986	0.053	0.602	13.715	0.908	12.384	2.321	122.020
91	SU	2	F	0.062	4.094	1.896	1.195	10.519	2.667	176.435	0.054	0.551	16.198	1.063	11.187	2.508	165.896
92	SU	2	F	0.064	4.420	1.901	1.241	10.652	2.723	188.075	0.041	0.381	15.605	1.098	11.372	2.551	176.171
93	SU	2	F	0.064	5.042	2.057	1.408	9.931	2.926	230.123	0.065	0.572	16.495	1.307	10.365	2.803	220.500
94	SU	2	F	0.063	3.055	1.573	0.932	12.095	2.350	114.507	0.071	0.848	10.781	0.845	12.667	2.244	109.339
95	SU	2	F	0.061	2.718	1.447	0.828	12.514	2.224	98.445	0.074	0.961	8.139	0.754	13.041	2.134	94.466
96	SU	2	F	0.087	1.553	1.251	0.687	19.209	2.053	36.650	0.114	1.686	2.123	0.663	19.481	2.024	36.140
97	SU	2	F	0.085	0.772	0.796	0.446	21.926	1.761	15.949	0.053	1.377	2.385	0.424	22.259	1.735	15.711
98	SU	2	F	0.085	1.873	1.309	0.705	18.682	2.074	45.432	0.118	1.489	2.740	0.674	19.020	2.038	44.624
99	SU	2	F	0.085	2.047	1.368	0.743	18.230	2.121	50.902	0.119	1.403	2.736	0.713	18.552	2.084	50.018
100	SU	2	F	0.085	2.511	1.539	0.847	17.230	2.247	66.046	0.126	1.283	3.387	0.809	17.587	2.201	64.705
101	SU	2	F	0.085	2.700	1.582	0.873	16.987	2.279	72.056	0.117	1.126	4.267	0.826	17.427	2.221	70.239
102	SU	2	F	0.086	3.022	1.723	0.975	16.194	2.402	84.589	0.105	0.944	5.138	0.918	16.674	2.332	82.152
103	SU	2	F	0.085	3.644	1.936	1.142	14.796	2.604	111.622	0.101	0.827	10.565	1.046	15.490	2.487	106.619
104	SU	2	F	0.086	4.542	2.189	1.373	13.489	2.883	152.625	0.066	0.472	14.712	1.239	14.295	2.721	144.022
105	SU	2	F	0.084	4.591	2.126	1.337	13.454	2.840	154.665	0.055	0.396	16.645	1.169	14.496	2.636	143.555
106	SU	2	F	0.085	3.440	1.852	1.075	15.216	2.523	102.477	0.118	0.991	7.721	1.005	15.749	2.438	99.013
107	SU	2	F	0.086	4.020	2.031	1.249	14.199	2.733	128.318	0.080	0.619	13.937	1.108	15.145	2.562	120.297
108	SU	2	F	0.123	1.534	1.544	0.892	24.235	2.301	28.698	0.111	1.322	3.978	0.847	24.812	2.247	28.030
109	SU	2	F	0.123	1.366	1.456	0.837	24.944	2.235	24.818	0.103	1.340	4.155	0.787	25.644	2.174	24.141
110	SU	2	F	0.123	0.805	1.072	0.610	28.452	1.960	12.818	0.067	1.295	4.195	0.563	29.297	1.903	12.449
111	SU	2	F	0.123	1.365	1.459	0.826	25.104	2.221	24.642	0.103	1.322	4.813	0.772	25.859	2.156	23.923
112	SU	2	F	0.124	2.348	2.003	1.146	21.545	2.609	49.390	0.167	1.460	3.897	1.099	22.028	2.552	48.307
113	SU	2	F	0.122	2.620	2.087	1.201	20.674	2.675	57.445	0.174	1.414	3.735	1.159	21.070	2.625	56.365
114	SU	2	F	0.123	3.096	2.304	1.316	19.809	2.814	70.847	0.193	1.390	4.584	1.265	20.254	2.753	69.293
115	SU	2	F	0.122	3.510	2.409	1.394	19.012	2.909	83.682	0.183	1.207	4.832	1.340	19.447	2.844	81.810
116	SU	2	F	0.124	3.815	2.438	1.418	19.131	2.938	90.390	0.168	1.018	6.126	1.350	19.683	2.855	87.852
117	SU	2	F	0.122	4.168	2.513	1.501	18.206	3.037	103.765	0.156	0.906	7.825	1.414	18.860	2.932	100.166

RUN	INFO			MASS FLOW RATES		SUPP.	BEND OUTLET CONDITIONS				DELTA-P	LOSS	GRAD. IN	STRAIGHT PIPE CONDITIONS			
NO.	B	L	P	AIR	SOLIDS	PRES.	PRES.	VELOC.	RHOAIR	RHOSUS.	BEND	COEFF.	STRAIGHT	PRES.	VELOC.	RHOAIR	RHOSUS.
				kg/s	kg/s	barg	barg	m/s	kg/m3	kg/m3	bar		-mbar/m	barg	m/s	kg/m3	kg/m3
118	LN	2	F	0.124	3.813	2.417	1.411	19.192	2.929	90.060	0.151	0.910	7.062	1.339	19.777	2.842	87.398
119	LN	2	F	0.122	4.234	2.525	1.473	18.408	3.004	104.260	0.174	0.985	7.774	1.387	19.072	2.899	100.627
120	LN	2	F	0.123	0.871	1.127	0.643	27.885	1.999	14.155	0.067	1.213	4.929	0.588	28.842	1.933	13.685
121	LN	2	F	0.123	1.350	1.484	0.837	24.952	2.234	24.515	0.120	1.572	4.305	0.785	25.678	2.171	23.822
122	LN	2	F	0.124	2.159	1.913	1.073	22.302	2.520	43.885	0.179	1.640	4.038	1.024	22.839	2.461	42.853
123	LN	2	F	0.122	2.531	1.985	1.176	20.907	2.645	54.872	0.181	1.509	4.206	1.129	21.365	2.588	53.697
124	LN	2	F	0.123	3.066	2.285	1.345	19.569	2.849	71.030	0.165	1.216	5.437	1.279	20.132	2.769	69.040
125	LN	2	F	0.122	3.440	2.381	1.390	19.045	2.904	81.884	0.172	1.155	5.662	1.327	19.558	2.827	79.734
126	LN	2	F	0.085	0.932	0.897	0.504	21.089	1.831	20.030	0.057	1.276	3.979	0.455	21.784	1.773	19.391
127	LN	2	F	0.087	1.456	1.202	0.648	19.660	2.006	33.580	0.114	1.756	3.175	0.609	20.127	1.959	32.800
128	LN	2	F	0.085	2.097	1.403	0.758	18.074.	2.139	52.601	0.124	1.441	3.342	0.718	18.498	2.090	51.395
129	LN	2	F	0.085	2.852	1.646	0.895	16.799	2.304	76.949	0.143	1.313	4.691	0.838	17.316	2.235	74.650
130	LN	2	F	0.085	3.799	1.936	1.145	14.777	2.607	116.546	0.113	0.890	11.927	1.036	15.563	2.476	110.657
131	LN	2	F	0.087	2.631	1.609	0.883	17.178	2.290	69.432	0.127	1.236	4.586	0.827	17.698	2.223	67.392
132	LN	2	F	0.086	3.165	1.775	0.967	16.259	2.392	88.231	0.152	1.306	4.812	0.914	16.712	2.327	85.840
133	LN	2	F	0.086	4.556	2.136	1.306	13.882	2.802	148.769	0.079	0.553	13.864	1.179	14.684	2.648	140.637
134	LN	2	F	0.086	4.314	2.070	1.248	14.201	2.732	137.707	0.088	0.633	13.501	1.112	15.117	2.567	129.367
135	LN	2	F	0.084	5.242	2.242	1.392	13.148	2.906	180.718	0.067	0.428	15.894	1.231	14.092	2.712	168.605
136	LN	2	F	0.085	5.161	2.195	1.395	13.209	2.910	177.104	0.067	0.431	18.386	1.209	14.318	2.685	163.390
137	ABORTED																
138	LN	2	F	0.064	0.438	0.485	0.270	18.676	1.548	10.630	0.023	1.260	3.083	0.233	19.238	1.503	10.319
139	LN	2	F	0.064	0.728	0.633	0.339	17.697	1.632	18.652	0.054	1.855	2.426	0.314	18.026	1.602	18.312
140	LN	2	F	0.064	1.253	0.849	0.458	16.361	1.776	34.706	0.078	1.668	2.618	0.426	16.723	1.737	33.955
141	LN	2	F	0.064	1.694	1.036	0.542	15.454	1.877	49.693	0.084	1.410	2.536	0.513	15.741	1.843	48.789
142	LN	2	F	0.064	1.937	1.130	0.600	14.896	1.947	58.947	0.109	1.665	3.493	0.561	15.265	1.900	57.523
143	LN	2	F	0.062	2.489	1.314	0.717	13.559	2.089	83.220	0.084	1.093	6.151	0.661	14.015	2.021	80.513
144	LN	2	F	0.063	2.920	1.471	0.832	12.772	2.229	103.622	0.093	1.103	8.738	0.744	13.416	2.122	98.643
145	LN	2	F	0.063	3.210	1.565	0.906	12.280	2.318	118.488	0.083	0.927	11.107	0.805	12.966	2.195	112.217
146	LN	2	F	0.064	2.284	1.242	0.670	14.224	2.033	72.768	0.091	1.242	4.616	0.619	14.673	1.971	70.543
147	LN	2	F	0.063	3.728	1.741	1.054	11.437	2.497	147.735	0.048	0.492	14.843	0.903	12.336	2.315	136.972
148	LN	2	F	0.063	4.125	1.818	1.128	11.024	2.586	169.611	0.051	0.496	14.591	0.995	11.756	2.425	159.042
149	LN	2	F	0.063	3.680	1.702	1.042	11.578	2.482	144.087	0.052	0.535	12.876	0.911	12.364	2.324	134.926
150	LN	2	F	0.062	3.902	1.747	1.067	11.168	2.512	158.387	0.044	0.442	12.916	0.936	11.919	2.354	148.408
151	LN	2	F	0.063	3.348	1.604	0.939	12.112	2.358	125.274	0.064	0.696	10.539	0.832	12.815	2.228	118.411
152	LN	2	.F	0.044	0.574	0.498	0.285	12.614	1.567	20.624	0.016	0.973	3.412	0.247	12.995	1.521	20.019
153	LN	2	F	0.044	0.937	0.654	0.363	12.010	1.661	35.365	0.035	1.369	4.187	0.324	12.354	1.614	34.380
154	LN	2	F	0.043	0.965	0.667	0.373	11.680	1.673	37.446	0.026	1.017	4.365	0.333	12.026	1.624	36.367
155	LN	2	F	0.043	1.073	0.712	0.389	11.652	1.692	41.734	0.033	1.156	4.150	0.351	11.976	1.646	40.604
156	LN	2	F	0.045	1.199	0.752	0.428	11.593	1.740	46.890	0.042	1.333	4.749	0.380	11.994	1.682	45.322
157	LN	2	F	0.042	1.259	0.769	0.434	10.894	1.747	52.376	0.044	1.404	5.056	0.388	11.254	1.692	50.702
158	LN	2	F	0.044	1.581	0.916	0.520	10.702	1.851	66.969	0.040	1.051	5.853	0.466	11.090	1.786	64.629
159	LN	2	F	0.044	2.004	1.067	0.624	10.133	1.977	89.665	0.045	0.974	7.482	0.556	10.575	1.895	85.916
160	LN	2	F	0.042	2.116	1.114	0.660	9.489	2.021	101.092	0.036	0.798	9.684	0.562	10.081	1.902	95.154
161	LN	2	F	0.044	2.419	1.218	0.727	9.406	2.101	116.551	0.029	0.572	11.169	0.613	10.062	1.964	108.957
162	LN	2	F	0.043	2.748	1.324	0.814	8.751.	2.206	142.317	0.024	0.444	12.582	0.686	9.409	2.052	132.371
163	LU	2	F	0.064	0.401	0.438	0.252	18.947	1.526	9.584	0.020	1.150	2.820	0.223	19.386	1.492	9.367
164	LU	2	F	0.064	0.888	0.712	0.368	17.322	1.667	23.229	0.070	2.014	2.220	0.341	17.667	1.634	22.775
165	LU	2	F	0.064	0.883	0.692	0.361	17.514	1.659	22.860	0.062	1.771	2.042	0.336	17.837	1.629	22.447
166	LU	2	F	0.064	1.126	0.801	0.433	16.615	1.746	30.708	0.092	2.170	2.549	0.405	16.948	1.712	30.104
167	LU	2	F	0.064	1.202	0.843	0.437	16.566	1.751	32.883	0.068	1.498	2.178	0.413	16.849	1.722	32.332
168	LU	2	F	0.062	2.264	1.207	0.642	14.177	1.998	72.378	0.101	1.387	4.709	0.589	14.642	1.935	70.081
169	LU	2	F	0.063	2.929	1.463	0.820	12.857	2.214	103.278	0.104	1.214	8.482	0.743	13.425	2.120	98.907
170	LU	2	F	0.063	3.223	1.536	0.897	12.318	2.307	118.603	0.063	0.695	10.087	0.805	12.943	2.196	112.875
171	LU	2	F	0.064	2.250	1.221	0.641	14.481	1.997	70.427	0.114	1.539	4.393	0.592	14.924	1.938	68.341
172	LU	2	F	0.063	3.448	1.611	0.960	11.960	2.384	130.681	0.056	0.604	11.310	0.846	12.699	2.245	123.083
173	LU	2	F	0.063	4.122	1.787	1.092	11.210	2.543	166.677	0.077	0.732	14.289	0.947	12.039	2.368	155.200
174	LU	2	F	0.063	3.387	1.597	0.944	12.155	2.364	126.313	0.052	0.556	11.771	0.825	12.945	2.220	118.605
175	LU	2	F	0.062	3.893	1.719	1.060	11.203	2.505	157.515	0.043	0.430	14.627	0.912	12.067	2.325	146.240
176	LU	2	F	0.064	2.017	1.132	0.580	15.078	1.924	60.623	0.113	1.642	2.857	0.551	15.358	1.889	59.520
177	LU	2	F	0.064	1.737	1.022	0.511	15.768	1.840	49.919	0.118	1.902	2.194	0.484	16.049	1.808	49.045
178	LU	2	F	0.064	1.321	0.883	0.460	16.341	1.778	36.647	0.085	1.740	2.353	0.433	16.637	1.746	35.993

RUN	INFO			MASS FLOW RATES		SUPP.	BEND OUTLET CONDITIONS				DELTA-P	LOSS	GRAD. IN	STRAIGHT PIPE CONDITIONS			
NO.	B	L	P	AIR	SOLIDS	PRES.	PRES.	VELOC.	RHOAIR	RHOSUS.	BEND	COEFF.	STRAIGHT	PRES.	VELOC.	RHOAIR	RHOSUS.
				kg/s	kg/s	barg	barg	m/s	kg/m3	kg/m3	bar		-mbar/m	barg	m/s	kg/m3	kg/m3
179	L	2	F	0.062	4.003	1.728	1.060	11.205	2.504	161.915	0.055	0.543	12.840	0.930	11.957	2.347	151.739
180	L	2	F	0.063	3.676	1.683	1.026	11.667	2.463	142.819	0.036	0.375	12.850	0.896	12.464	2.306	133.687
181	L	2	F	0.063	3.297	1.585	0.942	12.036	2.361	124.161	0.060	0.664	12.406	0.804	12.952	2.194	115.377
182	L	2	F	0.063	3.006	1.492	0.835	12.756	2.232	106.815	0.091	1.050	8.694	0.755	13.329	2.136	102.218
183	L	2	F	0.062	2.610	1.337	0.744	13.350	2.122	88.610	0.079	0.995	7.506	0.660	14.017	2.021	84.390
184	L	2	F	0.064	1.763	1.035	0.538	15.492	1.873	51.584	0.092	1.480	2.788	0.504	15.838	1.832	50.456
185	L	2	F	0.064	1.424	0.918	0.466	16.276	1.785	39.643	0.101	1.928	1.691	0.445	16.506	1.760	39.092
186	L	2	F	0.064	0.917	0.703	0.379	17.186	1.680	24.176	0.055	1.548	2.099	0.353	17.507	1.649	23.733
187	L	2	F	0.064	0.433	0.467	0.272	18.643	1.551	10.524	0.013	0.702	3.163	0.240	19.121	1.512	10.261
188	L	2	F	0.064	2.244	1.193	0.634	14.535	1.990	69.990	0.090	1.218	4.252	0.587	14.966	1.932	67.976
189	L	2	F	0.063	3.558	1.629	0.974	11.898	2.400	135.543	0.057	0.598	11.679	0.855	12.653	2.257	127.459
190	L	2	F	0.063	4.119	1.775	1.077	11.291	2.525	165.354	0.059	0.560	14.056	0.935	12.117	2.353	154.074
191	EM	2	F	0.044	0.507	0.457	0.258	12.883	1.534	17.845	0.025	1.714	2.634	0.229	13.188	1.498	17.432
192	EM	2	F	0.044	0.876	0.638	0.370	11.950	1.669	33.225	0.023	0.959	4.643	0.341	12.201	1.635	32.539
193	EM	2	F	0.043	1.054	0.719	0.405	11.411	1.712	41.873	0.024	0.879	5.352	0.362	11.773	1.659	40.586
194	EM	2	F	0.043	1.324	0.832	0.478	10.951	1.801	54.813	0.026	0.790	7.318	0.411	11.465	1.720	52.352
195	EM	2	F	0.045	1.333	0.816	0.469	11.274	1.789	53.607	0.036	1.043	6.687	0.408	11.759	1.715	51.394
196	EM	2	F	0.044	1.567	0.920	0.537	10.586	1.871	67.083	0.032	0.863	7.861	0.465	11.101	1.784	63.971
197	EM	2	F	0.044	1.960	1.062	0.619	10.166	1.971	87.370	0.026	0.579	9.240	0.525	10.786	1.857	82.349
198	EM	2	F	0.042	2.097	1.108	0.638	9.617	1.994	98.839	0.052	1.145	9.386	0.543	10.206	1.879	93.136
199	EM	2	F	0.044	2.435	1.238	0.730	9.388	2.105	117.542	0.040	0.773	11.259	0.627	9.978	1.981	110.598
200	EM	2	F	0.043	2.716	1.328	0.794	8.845	2.183	139.181	0.044	0.809	12.284	0.682	9.431	2.047	130.530
201	EM	2	F	0.045	2.281	1.178	0.690	9.831	2.056	105.186	0.035	0.682	10.526	0.594	10.420	1.940	99.239
202	EM	2	F	0.042	2.472	1.252	0.733	9.073	2.108	123.514	0.045	0.892	11.054	0.643	9.566	2.000	117.144
203	EM	2	F	0.064	0.443	0.463	0.255	18.896	1.530	10.627	0.039	2.070	2.078	0.236	19.184	1.507	10.468
204	EM	2	F	0.064	0.964	0.703	0.370	17.288	1.670	25.275	0.063	1.669	2.222	0.348	17.575	1.643	24.863
205	EM	2	F	0.064	1.142	0.791	0.413	16.882	1.721	30.655	0.077	1.758	2.491	0.385	17.218	1.687	30.057
206	EM	2	F	0.064	1.706	0.993	0.515	15.727	1.845	49.154	0.097	1.601	2.188	0.490	15.983	1.815	48.369
207	EM	2	F	0.064	1.904	1.079	0.573	15.151	1.915	56.965	0.107	1.638	2.754	0.545	15.422	1.881	55.961
208	EM	2	F	0.064	2.160	1.192	0.636	14.526	1.991	67.415	0.105	1.480	3.617	0.603	14.823	1.951	66.062
209	EM	2	F	0.062	2.452	1.283	0.694	13.744	2.061	80.864	0.113	1.479	5.624	0.642	14.171	1.999	78.426
210	EM	2	F	0.063	2.851	1.421	0.789	13.082	2.176	98.768	0.123	1.452	7.123	0.731	13.517	2.106	95.588
211	EM	2	F	0.063	3.340	1.607	0.919	12.179	2.334	124.327	0.082	0.888	12.053	0.797	13.002	2.186	116.457
212	EM	2	F	0.063	3.620	1.684	0.990	11.877	2.419	138.132	0.078	0.797	13.875	0.849	12.775	2.249	128.424
213	EM	2	F	0.063	3.845	1.742	1.034	11.548	2.473	150.934	0.072	0.713	13.711	0.881	12.479	2.288	139.665
214	EM	2	F	0.063	3.959	1.742	1.037	11.512	2.477	155.888	0.096	0.933	13.051	0.905	12.306	2.317	145.820
215	EM	2	F	0.062	4.155	1.785	1.067	11.167	2.513	168.670	0.098	0.931	14.218	0.923	11.999	2.338	156.972
216	EM	2	F	0.063	3.296	1.562	0.914	12.268	2.328	121.778	0.066	0.718	12.295	0.789	13.117	2.177	113.895
217	EM	2	F	0.085	0.458	0.586	0.310	24.182	1.597	8.591	0.045	1.785	2.245	0.290	24.564	1.572	8.457
218	EM	2	F	0.087	1.563	1.234	0.641	19.741	1.998	35.881	0.144	2.065	2.360	0.617	20.031	1.969	35.361
219	EM	2	F	0.085	2.150	1.355	0.701	18.673	2.071	52.181	0.154	1.689	2.212	0.677	18.945	2.041	51.431
220	EM	2	F	0.087	2.700	1.565	0.819	17.782	2.213	68.836	0.171	1.571	2.902	0.789	18.072	2.177	67.729
221	EM	2	F	0.085	2.754	1.560	0.824	17.445	2.219	71.545	0.147	1.348	4.012	0.783	17.841	2.170	69.959
222	EM	2	F	0.086	3.246	1.747	0.937	16.510	2.356	89.128	0.175	1.444	4.871	0.893	16.896	2.302	87.092
223	EM	2	F	0.085	4.201	2.047	1.211	14.338	2.687	132.799	0.106	0.780	14.071	1.054	15.426	2.498	123.435
224	EM	2	F	0.086	4.066	1.962	1.135	14.954	2.595	123.238	0.131	0.949	13.953	0.979	16.122	2.407	114.313
225	EM	2	F	0.086	4.499	2.057	1.222	14.402	2.700	141.584	0.129	0.877	13.831	1.082	15.367	2.531	132.696
226	EM	2	F	0.084	4.627	2.029	1.245	14.008	2.728	149.716	0.083	0.562	16.583	1.060	15.257	2.504	137.460
227	EM	2	F	0.085	4.837	2.076	1.284	13.850	2.775	158.284	0.109	0.715	15.242	1.129	14.850	2.588	147.630
228	EM	2	F	0.123	0.760	1.050	0.571	29.144	1.913	11.818	0.098	1.955	3.201	0.539	29.755	1.874	11.576
229	EM	2	F	0.123	0.951	1.193	0.646	27.821	2.004	15.487	0.121	2.025	3.107	0.615	28.360	1.966	15.193
230	EM	2	F	0.124	2.210	1.925	1.056	22.488	2.499	44.549	0.234	2.076	3.337	1.022	22.862	2.459	43.820
231	EM	2	F	0.122	2.492	2.032	1.113	21.530	2.568	52.469	0.247	2.029	3.355	1.079	21.880	2.527	51.629
232	EM	2	F	0.123	3.149	2.312	1.269	20.219	2.757	70.593	0.277	1.918	3.093	1.235	20.529	2.716	69.526
233	EM	2	F	0.122	3.437	2.374	1.317	19.638	2.816	79.318	0.269	1.758	4.041	1.277	19.990	2.766	77.923
234	EM	2	F	0.124	3.895	2.424	1.349	19.694	2.854	89.642	0.261	1.499	6.334	1.291	20.189	2.784	87.444
235	EM	2	F	0.122	4.182	2.470	1.376	19.157	2.887	98.941	0.258	1.422	7.009	1.305	19.745	2.801	95.996

RUN	INFO			MASS FLOW RATES		SUPP.	BEND OUTLET CONDITIONS				DELTA-P	LOSS	GRAD. IN	STRAIGHT PIPE CONDITIONS			
NO.	B	L	P	AIR kg/s	SOLIDS kg/s	PRES. barg	PRES. barg	VELOC. m/s	RHOAIR kg/m3	RHOSUS. kg/m3	BEND bar	COEFF.	STRAIGHT -mbar/m	PRES. barg	VELOC. m/s	RHOAIR kg/m3	RHOSUS. kg/m3
236	EF	2	F	0.044	0.566	0.481	0.265	12.810	1.543	20.020	0.033	2.009	3.270	0.229	13.187	1.499	19.449
237	EF	2	F	0.044	1.063	0.708	0.405	11.650	1.712	41.348	0.019	0.664	5.931	0.351	12.114	1.646	39.766
238	EF	2	F	0.043	0.817	0.598	0.349	11.885	1.644	31.170	0.024	1.069	4.903	0.304	12.290	1.590	30.143
239	EF	2	F	0.043	1.375	0.839	0.480	10.936	1.803	56.992	0.012	0.353	7.056	0.416	11.430	1.725	54.528
240	EF	2	F	0.044	1.856	1.001	0.579	10.306	1.922	81.610	0.028	0.646	9.117	0.514	10.745	1.843	78.271
241	EF	2	F	0.044	2.133	1.092	0.622	10.149	1.974	95.271	0.033	0.670	9.011	0.539	10.688	1.875	90.467
242	EF	2	F	0.042	2.212	1.120	0.659	9.495	2.019	105.608	0.023	0.491	9.288	0.574	10.003	1.917	100.243
243	EF	2	F	0.045	2.393	1.189	0.701	9.766	2.070	111.091	0.023	0.439	10.771	0.603	10.361	1.951	104.706
244	EF	2	F	0.044	2.585	1.236	0.717	9.459	2.089	123.864	0.037	0.668	10.017	0.625	9.989	1.979	117.303
245	EF	2	F	0.043	2.727	1.265	0.766	8.986	2.149	137.578	0.027	0.491	11.506	0.684	9.421	2.050	131.223
246	EF	2	F	0.064	0.410	0.442	0.240	19.125	1.512	9.716	0.027	1.514	1.946	0.220	19.432	1.488	9.562
247	EF	2	F	0.064	0.453	0.453	0.251	18.925	1.526	10.856	0.029	1.491	1.663	0.238	19.130	1.509	10.739
248	EF	2	F	0.064	1.100	0.776	0.400	17.002	1.706	29.325	0.069	1.631	1.972	0.378	17.271	1.680	28.869
249	EF	2	F	0.064	1.568	0.957	0.497	15.914	1.823	44.676	0.081	1.435	2.558	0.471	16.192	1.792	43.907
250	EF	2	F	0.064	2.046	1.118	0.591	14.975	1.937	61.934	0.098	1.415	2.641	0.567	15.204	1.908	61.001
251	EF	2	F	0.062	2.621	1.315	0.722	13.516	2.096	87.897	0.108	1.343	4.679	0.684	13.819	2.050	85.967
252	EF	2	F	0.063	2.974	1.456	0.829	12.796	2.224	105.346	0.073	0.845	9.094	0.746	13.402	2.124	100.588
253	EF	2	F	0.063	3.292	1.527	0.916	12.197	2.330	122.352	0.034	0.377	12.728	0.787	13.072	2.174	114.162
254	EF	2	F	0.063	3.597	1.633	0.960	12.058	2.383	135.228	0.066	0.672	12.061	0.838	12.855	2.235	126.840
255	EF	2	F	0.063	3.789	1.672	0.998	11.757	2.429	146.076	0.067	0.664	11.930	0.889	12.431	2.297	138.150
256	EF	2	F	0.063	3.793	1.669	1.001	11.720	2.433	146.699	0.062	0.613	13.202	0.867	12.555	2.271	136.941
257	EF	2	F	0.062	4.129	1.710	1.053	11.241	2.496	166.498	0.039	0.373	14.362	0.908	12.094	2.320	154.756
258	EF	2	F	0.064	4.844	1.856	1.185	10.925	2.655	200.992	0.048	0.402	15.621	1.042	11.684	2.483	187.931
259	EF	2	F	0.064	5.405	1.935	1.218	10.780	2.695	227.257	0.087	0.661	17.177	1.061	11.597	2.505	211.258
260	EF	2	F	0.085	0.848	0.844	0.448	21.897	1.764	17.558	0.085	2.022	1.874	0.429	22.186	1.741	17.329
261	EF	2	F	0.087	1.609	1.263	0.664	19.466	2.026	37.455	0.146	2.051	1.797	0.644	19.701	2.002	37.007
262	EF	2	F	0.085	1.757	1.245	0.643	19.375	2.000	41.109	0.141	1.833	2.118	0.622	19.630	1.974	40.575
263	EF	2	F	0.085	2.033	1.326	0.685	18.856	2.051	48.865	0.151	1.736	2.389	0.658	19.156	2.018	48.098
264	EF	2	F	0.087	2.525	1.526	0.793	18.037	2.181	63.449	0.175	1.692	2.653	0.763	18.337	2.146	62.409
265	EF	2	F	0.085	2.614	1.505	0.786	17.810	2.173	66.524	0.142	1.343	3.982	0.742	18.261	2.120	64.883
266	EF	2	F	0.086	3.166	1.700	0.909	16.754	2.321	85.664	0.162	1.351	4.985	0.853	17.252	2.254	83.187
267	EF	2	F	0.085	3.535	1.818	1.014	15.675	2.449	102.219	0.140	1.115	8.465	0.929	16.369	2.345	97.889
268	EF	2	F	0.085	3.861	1.890	1.087	15.185	2.537	115.258	0.108	0.814	11.487	0.959	16.171	2.383	108.232
269	EF	2	F	0.086	4.101	1.955	1.128	14.998	2.587	123.955	0.101	0.726	12.783	0.999	15.965	2.430	116.448
270	EF	2	F	0.086	4.237	1.952	1.144	14.927	2.605	128.659	0.097	0.674	13.766	1.004	15.961	2.437	120.328
271	EF	2	F	0.084	5.186	2.141	1.340	13.440	2.843	174.905	0.080	0.508	16.675	1.171	14.481	2.639	162.332
272	EF	2	F	0.123	0.429	0.797	0.414	32.372	1.722	6.012	0.063	1.995	2.918	0.381	33.128	1.683	5.875
273	EF	2	F	0.123	1.357	1.461	0.792	25.570	2.180	24.048	0.154	1.955	3.327	0.755	26.107	2.136	23.554
274	EF	2	F	0.123	1.255	1.382	0.752	26.158	2.131	21.739	0.154	2.073	2.616	0.720	26.638	2.093	21.347
275	EF	2	F	0.123	1.831	1.740	0.960	23.394	2.383	35.467	0.191	1.964	2.945	0.930	23.754	2.347	34.930
276	EF	2	F	0.124	2.270	1.965	1.081	22.216	2.530	46.319	0.228	1.996	3.064	1.047	22.585	2.489	45.563
277	EF	2	F	0.122	2.612	2.098	1.156	21.101	2.621	56.100	0.236	1.887	3.588	1.120	21.462	2.577	55.159
278	EF	2	F	0.123	3.117	2.290	1.271	20.207	2.759	69.912	0.246	1.726	4.140	1.229	20.585	2.708	68.626
279	EF	2	F	0.122	3.463	2.375	1.318	19.635	2.816	79.942	0.253	1.641	3.852	1.275	20.003	2.765	78.470
280	EF	2	F	0.124	3.896	2.413	1.351	19.677	2.856	89.739	0.230	1.323	6.581	1.291	20.191	2.784	87.455
281	EF	2	F	0.122	4.155	2.469	1.403	18.943	2.919	99.433	0.237	1.329	6.198	1.346	19.398	2.851	97.101

RUN NO.	INFO B	INFO L	INFO P	MASS FLOW RATES AIR kg/s	MASS FLOW RATES SOLIDS kg/s	SUPP. PRES. barg	BEND OUTLET CONDITIONS PRES. barg	BEND OUTLET CONDITIONS VELOC. m/s	BEND OUTLET CONDITIONS RHOAIR kg/m3	BEND OUTLET CONDITIONS RHOSUS. kg/m3	DELTA-P BEND bar	LOSS COEFF.	GRAD. IN STRAIGHT -mbar/m	STRAIGHT PIPE CONDITIONS PRES. barg	STRAIGHT PIPE CONDITIONS VELOC. m/s	STRAIGHT PIPE CONDITIONS RHOAIR kg/m3	STRAIGHT PIPE CONDITIONS RHOSUS. kg/m3
282	B	2	F	0.044	0.461	0.431	0.253	12.936	1.528	16.149	0.014	1.032	3.677	0.219	13.289	1.487	15.720
283	B	2	F	0.044	1.071	0.709	0.397	11.715	1.702	41.438	0.047	1.639	5.949	0.337	12.240	1.629	39.663
284	B	2	F	0.043	1.013	0.701	0.397	11.473	1.703	40.036	0.022	0.818	6.039	0.336	11.995	1.629	38.296
285	B	2	F	0.043	2.186	1.139	0.650	9.816	2.009	100.961	0.063	1.303	9.850	0.531	10.577	1.864	93.695
286	B	2	F	0.044	1.550	0.920	0.517	10.720	1.848	65.551	0.065	1.718	8.320	0.433	11.346	1.746	61.934
287	B	2	F	0.044	1.882	1.048	0.572	10.467	1.914	81.505	0.076	1.700	8.791	0.474	11.157	1.796	76.464
288	B	2	F	0.042	2.049	1.112	0.623	9.704	1.976	95.687	0.065	1.441	9.654	0.525	10.323	1.857	89.957
289	B	2	F	0.045	2.322	1.211	0.678	9.899	2.042	106.329	0.074	1.417	11.823	0.558	10.655	1.897	98.784
290	B	2	F	0.044	2.523	1.266	0.709	9.504	2.079	120.357	0.101	1.858	10.660	0.601	10.141	1.949	112.795
291	B	2	F	0.043	2.884	1.366	0.775	8.939	2.160	146.257	0.094	1.613	11.783	0.656	9.580	2.016	136.478
292	B	2	F	0.064	0.279	0.372	0.193	19.870	1.455	6.354	0.044	3.543	1.166	0.180	20.087	1.440	6.286
293	B	2	F	0.064	0.601	0.564	0.289	18.370	1.572	14.827	0.061	2.439	1.502	0.272	18.609	1.552	14.637
294	B	2	F	0.064	1.238	0.842	0.416	16.847	1.725	33.322	0.117	2.481	1.332	0.403	16.992	1.710	33.038
295	B	2	F	0.064	1.566	0.988	0.483	16.064	1.806	44.187	0.142	2.494	2.247	0.462	16.287	1.781	43.580
296	B	2	F	0.064	1.864	1.112	0.553	15.343	1.891	55.064	0.135	2.090	3.302	0.513	15.746	1.842	53.653
297	B	2	F	0.062	2.552	1.367	0.734	13.426	2.110	86.168	0.126	1.626	8.787	0.628	14.299	1.981	80.902
298	B	2	F	0.063	2.992	1.539	0.838	12.734	2.235	106.497	0.137	1.589	11.232	0.713	13.658	2.084	99.291
299	B	2	F	0.064	2.181	1.226	0.633	14.552	1.987	67.922	0.140	1.950	4.770	0.575	15.084	1.917	65.529
300	B	2	F	0.062	2.922	1.501	0.799	12.906	2.188	102.616	0.156	1.830	9.523	0.693	13.710	2.060	96.602
301	B	2	F	0.063	3.447	1.689	0.925	12.277	2.341	127.250	0.149	1.554	12.999	0.780	13.269	2.166	117.734
302	B	2	F	0.064	3.798	1.770	0.981	12.100	2.409	142.253	0.184	1.767	12.076	0.846	12.977	2.246	132.649
303	B	2	F	0.063	3.799	1.777	0.987	11.802	2.416	145.888	0.194	1.905	13.196	0.853	12.649	2.254	136.121
304	B	2	F	0.062	3.622	1.675	0.946	11.858	2.366	138.447	0.167	1.718	12.000	0.824	12.643	2.219	129.842
305	B	2	F	0.064	4.330	1.893	1.059	11.606	2.504	169.117	0.210	1.847	14.600	0.911	12.499	2.325	157.029
306	B	2	F	0.064	4.105	1.827	1.043	11.698	2.484	159.048	0.210	1.933	14.679	0.894	12.611	2.304	147.528
307	B	2	F	0.064	4.308	1.867	1.081	11.469	2.529	170.268	0.236	2.111	14.867	0.930	12.359	2.347	158.003
308	B	2	F	0.064	4.658	1.910	1.094	11.397	2.545	185.251	0.273	2.273	13.140	0.961	12.167	2.384	173.530
309	B	2	F	0.063	4.646	1.893	1.110	11.100	2.564	189.728	0.222	1.899	15.467	0.953	11.987	2.375	175.703
310	B	2	F	0.063	5.021	1.926	1.131	11.006	2.590	206.767	0.232	1.852	15.903	0.970	11.901	2.396	191.210
311	B	2	F	0.087	1.518	1.250	0.629	19.882	1.983	34.597	0.197	2.876	1.687	0.612	20.092	1.963	34.236
312	B	2	F	0.085	0.581	0.687	0.361	23.276	1.659	11.306	0.075	2.448	2.177	0.339	23.656	1.632	11.124
313	B	2	F	0.085	1.550	1.194	0.568	20.299	1.909	34.619	0.194	2.715	1.994	0.548	20.562	1.885	34.176
314	B	2	F	0.085	2.168	1.470	0.713	18.543	2.085	52.990	0.256	2.815	1.528	0.698	18.711	2.066	52.514
315	B	2	F	0.085	2.483	1.585	0.772	17.953	2.156	62.679	0.260	2.576	2.678	0.745	18.231	2.123	61.724
316	B	2	F	0.086	2.854	1.709	0.834	17.454	2.231	74.112	0.290	2.571	3.519	0.798	17.798	2.188	72.679
317	B	2	F	0.086	3.121	1.786	0.907	16.768	2.319	84.377	0.261	2.202	5.364	0.853	17.257	2.254	81.986
318	B	2	F	0.085	3.865	2.010	1.118	14.964	2.575	117.067	0.211	1.609	13.044	0.947	16.274	2.368	107.645
319	B	2	F	0.086	4.348	2.107	1.157	14.837	2.621	132.822	0.249	1.700	12.303	1.020	15.838	2.456	124.427
320	B	2	F	0.084	4.622	2.138	1.188	14.367	2.660	145.821	0.294	1.954	14.944	1.037	15.430	2.476	135.779
321	B	2	F	0.086	5.228	2.313	1.292	13.949	2.785	169.887	0.308	1.861	15.450	1.135	14.967	2.595	158.335
322	B	2	F	0.123	0.739	1.060	0.553	29.487	1.891	11.366	0.133	2.699	3.108	0.521	30.094	1.853	11.137
323	B	2	F	0.123	1.029	1.272	0.665	27.510	2.027	16.949	0.171	2.663	3.176	0.630	28.103	1.984	16.592
324	B	2	F	0.123	1.289	1.450	0.778	25.776	2.163	22.665	0.183	2.430	2.995	0.744	26.266	2.123	22.242
325	B	2	F	0.123	1.783	1.748	0.913	23.962	2.327	33.735	0.265	2.739	2.177	0.889	24.268	2.297	33.310
326	B	2	F	0.124	2.273	2.009	1.063	22.406	2.509	45.975	0.316	2.739	2.401	1.037	22.699	2.476	45.382
327	B	2	F	0.123	2.605	2.150	1.125	21.580	2.584	54.720	0.341	2.676	2.573	1.099	21.846	2.552	54.052
328	B	2	F	0.123	3.015	2.357	1.236	20.517	2.717	66.604	0.401	2.864	3.606	1.200	20.856	2.673	65.520
329	B	2	F	0.122	3.440	2.499	1.303	19.764	2.798	78.886	0.436	2.831	4.034	1.258	20.155	2.744	77.355
330	B	2	F	0.122	3.517	2.457	1.269	20.055	2.757	79.495	0.429	2.682	4.800	1.221	20.492	2.699	77.798
331	B	2	F	0.123	3.661	2.408	1.254	20.352	2.739	81.533	0.395	2.338	5.485	1.199	20.864	2.672	79.532
332	B	2	F	0.121	4.087	2.517	1.312	19.527	2.809	94.861	0.423	2.340	6.864	1.242	20.130	2.725	92.020

RUN NO.	INFO B	INFO L	INFO P	MASS FLOW RATES AIR kg/s	MASS FLOW RATES SOLIDS kg/s	SUPP. PRES. barg	BEND OUTLET CONDITIONS PRES. barg	BEND OUTLET CONDITIONS VELOC. m/s	BEND OUTLET CONDITIONS RHOAIR kg/m3	BEND OUTLET CONDITIONS RHOSUS. kg/m3	DELTA-P BEND bar	LOSS COEFF.	GRAD. IN STRAIGHT -mbar/m	STRAIGHT PIPE CONDITIONS PRES. barg	STRAIGHT PIPE CONDITIONS VELOC. m/s	STRAIGHT PIPE CONDITIONS RHOAIR kg/m3	STRAIGHT PIPE CONDITIONS RHOSUS. kg/m3
333	SN	2	F	0.044	0.509	0.480	0.260	12.863	1.536	17.945	0.030	2.019	2.780	0.232	13.155	1.502	17.547
334	SN	2	F	0.063	0.402	0.453	0.246	18.796	1.519	9.702	0.027	1.554	2.317	0.218	19.227	1.485	9.484
335	SN	2	F	0.064	0.519	0.518	0.282	18.478	1.563	12.742	0.033	1.513	2.409	0.255	18.870	1.530	12.477
336	SN	2	F	0.064	1.080	0.782	0.415	16.852	1.724	29.039	0.065	1.583	2.545	0.389	17.162	1.693	28.514
337	SN	2	F	0.064	1.758	1.023	0.525	15.617	1.858	51.039	0.103	1.658	2.864	0.494	15.948	1.819	49.979
338	SN	2	F	0.064	2.128	1.139	0.588	15.009	1.933	64.258	0.112	1.548	3.095	0.553	15.340	1.891	62.872
339	SN	2	F	0.064	2.255	1.191	0.620	14.668	1.972	69.675	0.106	1.409	4.280	0.572	15.109	1.914	67.638
340	SN	2	F	0.062	2.605	1.312	0.706	13.647	2.076	86.519	0.100	1.238	7.116	0.641	14.184	1.997	83.243
341	SN	2	F	0.063	2.974	1.432	0.776	13.171	2.161	102.352	0.100	1.129	8.219	0.701	13.748	2.070	98.053
342	SN	2	F	0.062	3.234	1.508	0.847	12.571	2.246	116.615	0.095	1.026	10.714	0.738	13.351	2.115	109.798
343	SN	2	F	0.063	3.518	1.568	0.904	12.412	2.315	128.476	0.070	0.706	10.787	0.784	13.243	2.170	120.415
344	SN	2	F	0.064	4.107	1.720	1.035	11.781	2.474	158.000	0.058	0.527	13.036	0.903	12.594	2.314	147.796
345	SN	2	F	0.063	3.835	1.621	0.972	11.892	2.398	146.162	0.062	0.600	12.600	0.844	12.711	2.243	136.748
346	SN	2	F	0.062	4.090	1.663	1.031	11.362	2.469	163.155	0.051	0.483	15.104	0.878	12.282	2.284	150.925
347	SN	2	F	0.064	4.061	1.676	1.066	11.570	2.511	159.111	0.056	0.521	15.947	0.904	12.547	2.316	146.727
348	SN	2	F	0.064	4.251	1.716	1.070	11.547	2.516	166.878	0.049	0.440	13.989	0.928	12.392	2.345	155.507
349	SN	2	F	0.064	4.596	1.789	1.066	11.551	2.511	180.374	0.077	0.642	12.106	0.943	12.276	2.363	169.718
350	SN	2	F	0.064	4.950	1.848	1.177	10.964	2.646	204.638	0.061	0.500	14.170	1.048	11.653	2.489	192.533
351	SN	2	F	0.063	4.640	1.758	1.095	11.179	2.546	188.118	0.060	0.512	13.653	0.984	11.802	2.412	178.196
352	SN	2	F	0.063	4.626	1.722	1.059	11.388	2.504	184.106	0.071	0.591	12.317	0.959	11.967	2.382	175.196
353	SN	2	F	0.064	0.260	0.379	0.234	19.217	1.505	6.126	0.018	1.586	2.165	0.210	19.596	1.476	6.007
354	SN	2	F	0.064	0.463	0.521	0.321	17.930	1.610	11.715	0.031	1.625	2.389	0.294	18.295	1.578	11.481
355	SN	2	F	0.064	0.705	0.695	0.415	16.850	1.724	18.974	0.042	1.565	1.994	0.395	17.092	1.700	18.705
356	SN	2	F	0.064	1.030	0.850	0.533	15.545	1.866	30.020	0.073	2.022	3.281	0.499	15.887	1.826	29.373
357	SN	2	F	0.064	1.655	1.163	0.713	13.919	2.084	53.891	0.088	1.681	4.526	0.662	14.338	2.023	52.315
358	SN	2	F	0.064	1.833	1.248	0.781	13.347	2.167	62.260	0.085	1.525	4.895	0.736	13.689	2.113	60.706
359	SN	2	F	0.062	2.290	1.440	0.918	12.147	2.332	85.450	0.053	0.838	9.073	0.826	12.756	2.221	81.376
360	SN	2	F	0.063	2.820	1.642	1.065	11.336	2.511	112.772	0.076	1.054	9.379	0.970	11.880	2.396	107.609
361	SN	2	F	0.062	2.837	1.650	1.074	11.199	2.522	114.816	0.078	1.090	8.225	0.991	11.666	2.421	110.225
362	SN	2	F	0.063	3.239	1.806	1.197	10.764	2.670	136.393	0.049	0.617	11.341	1.082	11.355	2.531	129.292
363	SN	2	F	0.064	3.607	1.870	1.254	10.639	2.739	153.687	0.042	0.489	12.316	1.130	11.260	2.588	145.219
364	SN	2	F	0.063	3.538	1.825	1.242	10.462	2.725	153.265	0.043	0.513	12.378	1.142	10.951	2.603	146.418
365	SN	2	F	0.062	3.755	1.889	1.283	10.113	2.775	168.303	0.051	0.597	11.000	1.194	10.523	2.666	161.740
366	SN	2	F	0.064	4.050	1.952	1.350	10.176	2.855	180.415	0.053	0.571	13.705	1.239	10.680	2.721	171.897
367	SN	2	F	0.064	3.915	1.883	1.305	10.373	2.801	171.083	0.054	0.584	13.806	1.193	10.901	2.665	162.788
368	SN	2	F	0.064	4.029	1.902	1.301	10.377	2.796	175.968	0.048	0.503	12.192	1.202	10.842	2.676	168.419
369	SN	2	F	0.064	4.533	2.024	1.421	9.863	2.941	208.325	0.046	0.454	12.860	1.316	10.307	2.815	199.365
370	SN	2	F	0.064	1.395	1.061	0.673	14.181	2.036	44.600	0.060	1.331	4.058	0.628	14.572	1.981	43.403
371	SN	2	F	0.064	2.093	1.405	0.895	12.549	2.305	75.591	0.070	1.178	6.191	0.832	12.976	2.229	73.102
372	SN	2	F	0.062	2.288	1.473	0.954	11.921	2.377	87.009	0.065	1.049	7.626	0.877	12.409	2.283	83.587
373	SN	2	F	0.061	2.567	1.556	1.020	11.334	2.455	102.642	0.037	0.564	9.704	0.921	11.911	2.337	97.670
374	SN	2	F	0.085	0.840	0.957	0.599	19.837	1.947	19.199	0.062	1.645	2.727	0.571	20.184	1.913	18.870
375	SN	2	F	0.087	1.456	1.303	0.817	17.837	2.211	37.011	0.105	1.792	2.564	0.791	18.094	2.179	36.485
376	SN	2	F	0.085	1.691	1.415	0.892	16.842	2.301	45.503	0.109	1.682	2.763	0.864	17.093	2.267	44.833
377	SN	2	F	0.084	1.976	1.561	0.987	15.662	2.416	57.182	0.125	1.788	2.559	0.959	15.889	2.382	56.367
378	SN	2	F	0.085	2.182	1.645	1.052	15.519	2.494	63.720	0.120	1.569	3.389	1.011	15.835	2.445	62.449
379	SN	2	F	0.086	2.591	1.838	1.204	14.539	2.678	80.782	0.109	1.273	5.259	1.145	14.934	2.607	78.645
380	SN	2	F	0.086	3.055	1.963	1.304	13.892	2.799	99.693	0.100	1.039	7.405	1.221	14.405	2.700	96.142
381	SN	2	F	0.085	3.623	2.140	1.459	12.898	2.987	127.338	0.100	0.949	9.201	1.357	13.456	2.863	122.056
382	SN	2	F	0.084	4.313	2.335	1.625	11.985	3.188	163.108	0.051	0.434	13.154	1.505	12.557	3.043	155.674
383	SN	2	F	0.084	4.675	2.351	1.626	11.981	3.189	176.889	0.070	0.552	14.925	1.490	12.634	3.024	167.746
384	SN	2	F	0.086	4.710	2.340	1.637	12.128	3.203	176.023	0.065	0.499	15.471	1.481	12.891	3.013	165.602
385	SN	2	F	0.044	0.441	0.447	0.268	12.781	1.546	15.622	0.027	2.079	2.739	0.238	13.094	1.509	15.249
386	SN	2	F	0.044	0.886	0.712	0.440	11.366	1.755	35.350	0.049	2.132	4.463	0.400	11.694	1.705	34.357
387	SN	2	F	0.043	0.949	0.748	0.450	11.057	1.767	38.893	0.030	1.265	4.290	0.411	11.361	1.719	37.850
388	SN	2	F	0.043	1.206	0.861	0.519	10.659	1.850	51.279	0.038	1.310	4.783	0.475	10.973	1.797	49.814
389	SN	2	F	0.044	1.670	1.057	0.667	9.762	2.029	77.561	0.036	0.971	6.582	0.601	10.166	1.949	74.478
390	SN	2	F	0.045	1.156	0.825	0.520	10.998	1.851	47.645	0.039	1.362	5.599	0.474	11.335	1.796	46.227
391	SN	2	F	0.044	1.883	1.124	0.709	9.635	2.079	88.574	0.043	1.052	6.288	0.651	9.968	2.010	85.615
392	SN	2	F	0.042	1.901	1.135	0.716	9.181	2.088	93.848	0.029	0.733	8.458	0.639	9.611	1.995	89.650
393	SN	2	F	0.044	2.399	1.325	0.853	8.769	2.254	124.022	0.021	0.436	11.415	0.737	9.350	2.114	116.323
394	SN	2	F	0.042	2.506	1.334	0.858	8.463	2.260	134.188	0.025	0.517	12.197	0.759	8.938	2.140	127.065
395	ABORTED																
396	SN	2	F	0.045	2.217	1.263	0.804	9.211	2.195	109.092	0.036	0.770	9.310	0.719	9.663	2.092	103.981
397	SN	2	F	0.123	0.687	1.161	0.752	26.159	2.131	11.912	0.069	1.684	2.722	0.721	26.617	2.095	11.707
398	SN	2	F	0.123	0.947	1.416	0.884	24.330	2.291	17.646	0.106	2.034	2.527	0.856	24.697	2.257	17.384
399	SN	2	F	0.123	0.952	1.406	0.904	24.073	2.316	17.926	0.078	1.511	3.437	0.869	24.519	2.274	17.600
400	SN	2	F	0.123	1.370	1.694	1.098	21.863	2.550	28.407	0.136	2.010	1.915	1.080	22.046	2.529	28.171
401	SN	2	F	0.124	2.058	2.086	1.372	19.507	2.881	47.830	0.158	1.733	2.885	1.342	19.749	2.846	47.243
402	SN	2	F	0.122	2.419	2.262	1.500	18.207	3.037	60.219	0.171	1.717	2.759	1.470	18.433	3.000	59.482
403	SN	2	F	0.123	2.869	2.498	1.664	17.234	3.235	75.455	0.190	1.696	2.604	1.635	17.423	3.200	74.638
404	SN	2	F	0.122	3.043	2.515	1.685	16.959	3.261	81.324	0.183	1.565	2.661	1.656	17.147	3.225	80.430
405	SN	2	F	0.122	3.514	2.696	1.837	16.058	3.444	99.188	0.170	1.333	3.681	1.799	16.271	3.399	97.888

RUN	INFO			MASS FLOW RATES		SUPP.	BEND OUTLET CONDITIONS				DELTA-P	LOSS	GRAD. IN	STRAIGHT PIPE CONDITIONS			
NO.	B	L	P	AIR	SOLIDS	PRES.	PRES.	VELOC.	RHOAIR	RHOSUS.	BEND	COEFF.	STRAIGHT	PRES.	VELOC.	RHOAIR	RHOSUS.
				kg/s	kg/s	barg	barg	m/s	kg/m3	kg/m3	bar		-mbar/m	barg	m/s	kg/m3	kg/m3
406	V	2	F	0.044	0.570	0.483	0.262	12.847	1.538	20.124	0.028	1.667	2.873	0.241	13.057	1.514	19.800
407	V	2	F	0.044	0.884	0.606	0.339	12.216	1.633	32.783	0.039	1.591	3.867	0.308	12.508	1.595	32.019
408	V	2	F	0.043	1.073	0.713	0.383	11.702	1.685	41.581	0.033	1.172	5.487	0.327	12.189	1.618	39.922
409	V	2	F	0.044	1.528	0.878	0.468	11.081	1.788	62.495	0.067	1.735	5.961	0.413	11.504	1.722	60.193
410	V	2	F	0.045	1.207	0.762	0.422	11.645	1.732	46.980	0.048	1.494	5.366	0.367	12.104	1.666	45.196
411	V	2	F	0.044	1.705	0.934	0.534	10.723	1.868	72.070	0.054	1.314	8.156	0.460	11.266	1.778	68.595
412	V	2	F	0.042	1.816	0.988	0.542	10.210	1.878	80.617	0.070	1.658	7.413	0.467	10.729	1.787	76.717
413	V	2	F	0.044	2.509	1.210	0.681	9.659	2.046	117.742	0.086	1.566	9.960	0.590	10.208	1.936	111.411
414	V	2	F	0.043	2.861	1.327	0.757	9.031	2.138	143.622	0.074	1.257	11.406	0.653	9.596	2.012	135.159
415	V	2	F	0.046	2.344	1.179	0.660	10.407	2.021	102.084	0.062	1.129	10.183	0.557	11.091	1.896	95.780
416	V	2	F	0.064	0.412	0.463	0.245	19.047	1.518	9.817	0.049	2.731	2.016	0.235	19.205	1.506	9.736
417	V	2	F	0.064	0.613	0.558	0.293	18.317	1.576	15.160	0.056	2.198	1.279	0.282	18.465	1.564	15.039
418	V	2	F	0.064	0.906	0.696	0.359	17.547	1.656	23.415	0.073	2.025	1.187	0.349	17.672	1.644	23.250
419	V	2	F	0.064	1.364	0.907	0.456	16.354	1.774	37.796	0.114	2.248	2.317	0.433	16.621	1.745	37.191
420	V	2	F	0.062	1.681	1.016	0.525	15.259	1.857	49.920	0.100	1.717	3.067	0.493	15.575	1.819	48.909
421	V	2	F	0.062	2.361	1.269	0.667	13.966	2.029	76.621	0.150	2.010	5.772	0.597	14.574	1.944	73.421
422	V	2	F	0.063	2.848	1.430	0.787	13.094	2.174	98.576	0.109	1.293	10.194	0.663	14.062	2.024	91.792
423	V	2	F	0.063	3.281	1.569	0.891	12.354	2.300	120.371	0.108	1.172	11.942	0.747	13.373	2.125	111.200
424	V	2	F	0.063	3.655	1.655	0.932	12.151	2.350	136.351	0.132	1.306	13.137	0.799	13.045	2.189	127.009
425	V	2	F	0.063	3.911	1.696	0.980	11.844	2.407	149.679	0.158	1.501	13.716	0.841	12.733	2.239	139.226
426	V	2	F	0.062	3.916	1.662	0.962	11.759	2.386	150.965	0.119	1.137	13.326	0.841	12.532	2.239	141.652
427	V	2	F	0.064	4.720	1.846	1.107	11.324	2.562	188.905	0.154	1.275	15.011	0.955	12.201	2.378	175.338
428	V	2	F	0.064	5.194	1.895	1.123	11.256	2.581	209.164	0.191	1.439	16.981	0.951	12.244	2.373	192.299
429	V	2	F	0.085	0.637	0.721	0.377	23.010	1.679	12.542	0.074	2.217	2.150	0.360	23.300	1.657	12.383
430	V	2	F	0.087	1.394	1.156	0.601	20.230	1.949	31.229	0.133	2.073	2.005	0.583	20.470	1.927	30.873
431	V	2	F	0.085	1.859	1.283	0.632	19.460	1.987	43.305	0.185	2.255	1.526	0.619	19.620	1.970	42.938
432	V	2	F	0.087	2.718	1.639	0.828	17.700	2.223	69.616	0.237	2.177	3.959	0.795	18.010	2.184	68.396
433	V	2	F	0.086	3.144	1.746	0.886	16.950	2.294	84.063	0.258	2.136	4.691	0.829	17.480	2.225	81.541
434	V	2	F	0.085	3.232	1.752	0.894	16.670	2.304	87.903	0.255	2.088	6.211	0.819	17.350	2.213	84.427
435	V	2	F	0.086	4.198	2.023	1.121	15.050	2.578	126.438	0.207	1.444	11.620	0.992	16.020	2.422	118.772
436	V	2	F	0.086	4.422	2.041	1.157	14.830	2.622	135.119	0.272	1.831	13.530	1.020	15.840	2.456	126.577
437	V	2	F	0.084	4.543	2.043	1.153	14.600	2.617	141.025	0.281	1.868	13.960	1.153	14.600	2.617	141.025
438	V	2	F	0.085	3.824	1.928	1.056	15.410	2.500	112.449	0.225	1.688	11.070	0.933	16.390	2.350	105.738
439	V	2	F	0.123	0.835	1.130	0.597	28.670	1.944	13.202	0.125	2.300	2.958	0.564	29.270	1.905	12.932
440	V	2	F	0.123	1.237	1.392	0.755	26.120	2.135	21.470	0.157	2.148	2.954	0.725	26.570	2.099	21.106
441	V	2	F	0.123	1.215	1.385	0.761	26.010	2.143	21.170	0.146	2.037	3.313	0.728	26.520	2.103	20.769
442	V	2	F	0.123	1.729	1.690	0.921	23.860	2.336	32.840	0.188	2.015	3.310	0.891	24.240	2.300	32.326
443	V	2	F	0.124	2.278	1.959	1.063	22.410	2.508	46.074	0.248	2.147	2.890	1.031	22.760	2.469	45.363
444	V	2	F	0.122	2.608	2.083	1.129	21.370	2.588	55.320	0.261	2.069	3.577	1.093	21.740	2.544	54.389
445	V	2	F	0.123	3.069	2.295	1.241	20.470	2.723	67.951	0.324	2.276	4.829	1.197	20.880	2.670	66.622
446	V	2	F	0.122	3.468	2.456	1.311	19.700	2.808	79.822	0.380	2.450	5.549	1.260	20.130	2.746	78.068
447	V	2	F	0.124	3.709	2.423	1.253	20.530	2.738	81.893	0.420	2.434	5.021	1.202	21.000	2.676	80.047
448	V	2	F	0.122	4.045	2.472	1.297	19.820	2.791	92.541	0.378	2.082	5.734	1.244	20.280	2.727	90.415
449	SN	2E	F	0.085	0.456	1.064	0.826	17.390	2.221	11.872	0.028	1.584	2.004	0.803	17.600	2.194	11.728
450	SN	2E	F	0.087	0.470	1.093	0.866	17.370	2.270	12.250	0.034	1.817	1.669	0.848	17.550	2.248	12.129
451	SN	2E	F	0.085	0.790	1.382	1.123	15.020	2.581	23.856	0.060	2.215	2.541	1.095	15.020	2.547	23.543
452	SN	2E	F	0.085	1.180	1.781	1.402	13.250	2.918	40.368	0.045	1.281	3.007	1.369	13.440	2.878	39.816
453	SN	2E	F	0.087	1.681	2.217	1.787	11.630	3.384	65.534	0.050	1.133	5.706	1.730	11.870	3.315	64.198
454	SN	2E	F	0.085	1.878	2.280	1.834	11.250	3.441	75.665	0.045	0.942	7.042	1.769	11.510	3.362	73.935
455	SN	2E	F	0.086	2.243	2.583	2.103	10.330	3.766	98.457	0.040	0.762	9.021	2.012	10.640	3.656	95.579
456	SN	2E	F	0.064	0.370	0.723	0.549	15.330	1.886	10.944	0.019	1.444	2.021	0.526	15.560	1.859	10.786
457	SN	2E	F	0.064	0.471	0.855	0.646	14.410	2.003	14.813	0.024	1.566	2.354	0.619	14.650	1.971	14.579
458	SN	2E	F	0.064	0.657	1.025	0.793	13.320	2.182	22.357	0.041	2.082	2.313	0.768	13.510	2.150	22.038
459	SN	2E	F	0.064	1.006	1.376	1.068	11.540	2.514	39.517	0.035	1.318	3.467	1.029	11.760	2.467	38.776
460	SN	2E	F	0.064	1.151	1.519	1.182	10.940	2.652	47.694	0.038	1.349	5.354	1.133	11.190	2.593	46.628
461	SN	2E	F	0.062	1.686	1.874	1.484	9.387	3.017	81.395	0.042	1.178	7.435	1.402	9.710	2.918	78.719
462	SN	2E	F	0.063	1.989	2.026	1.625	8.930	3.188	100.967	0.028	0.695	9.304	1.540	9.227	3.085	97.710
463	SN	2E	F	0.063	2.478	2.324	1.865	8.172	3.478	137.467	0.021	0.453	11.080	1.764	8.469	3.356	132.637
464	SN	2E	F	0.063	2.711	2.437	1.955	7.960	3.587	154.362	0.033	0.682	11.300	1.852	8.247	3.463	148.999
465	SN	2E	F	0.060	2.853	2.426	1.963	7.611	3.597	169.897	0.025	0.517	12.300	1.863	7.876	3.476	164.182
466	SN	2E	F	0.063	3.361	2.638	2.149	7.436	3.822	204.869	0.022	0.391	13.290	2.041	7.699	3.691	197.865
467	SN	2E	F	0.083	2.734	2.762	2.261	9.528	3.957	130.041	0.025	0.431	11.470	2.168	9.808	3.845	126.344
468	SN	2E	F	0.085	2.881	2.869	2.338	9.555	4.051	136.646	0.036	0.580	10.560	2.242	9.839	3.934	132.728
469	SN	2E	F	0.085	2.845	2.832	2.310	9.559	4.017	134.916	0.031	0.501	10.930	2.210	9.855	3.896	130.852
470	SN	2E	F	0.085	2.979	2.828	2.300	9.622	4.005	140.348	0.019	0.295	12.390	2.199	9.924	3.882	136.065
471	SN	2E	F	0.084	3.233	2.894	2.382	9.301	4.104	157.571	0.033	0.484	12.200	2.282	9.581	3.983	152.925
472	SN	2E	F	0.084	3.347	2.936	2.410	9.168	4.138	165.456	0.021	0.307	12.210	2.311	9.442	4.018	160.666
473	SN	2E	F	0.085	3.547	3.050	2.495	9.054	4.240	177.579	0.029	0.403	13.540	2.385	9.348	4.107	172.006
474	SN	2E	F	0.086	4.192	3.249	2.665	8.799	4.446	215.970	0.031	0.370	14.250	2.535	9.121	4.289	208.330
475	SN	2E	F	0.122	0.693	1.716	1.378	19.140	2.889	16.401	0.055	1.815	2.197	1.356	19.320	2.862	16.250

RUN NO.	INFO B	L	P	MASS FLOW RATES AIR kg/s	SOLIDS kg/s	SUPP. PRES. barg	BEND OUTLET CONDITIONS PRES. barg	VELOC. m/s	RHOAIR kg/m3	RHOSUS. kg/m3	DELTA-P BEND bar	LOSS COEFF.	GRAD. IN STRAIGHT -mbar/m	STRAIGHT PIPE CONDITIONS PRES. barg	VELOC. m/s	RHOAIR kg/m3	RHOSUS. kg/m3
476	SN	2E	F	0.121	1.258	2.322	1.882	15.680	3.499	36.376	0.075	1.670	2.533	1.859	15.800	3.471	36.087
477	SN	2E	F	0.120	1.343	2.397	1.929	15.300	3.555	39.800	0.093	2.006	2.439	1.901	15.442	3.522	39.432
478	SN	2E	F	0.121	2.032	2.949	2.420	13.217	4.149	69.681	0.088	1.452	2.742	2.392	13.325	4.116	69.117
479	SN	2E	F	0.121	2.548	3.236	2.645	12.402	4.422	93.118	0.101	1.416	5.487	2.590	12.593	4.355	91.702
480	SN	2E	F	0.120	2.841	3.352	2.776	11.874	4.581	108.460	0.062	0.810	8.677	2.688	12.156	4.475	105.942
481	SN	2E	F	0.174	0.825	2.398	1.947	22.043	3.578	16.964	0.079	1.923	2.763	1.919	22.254	3.544	16.804
482	SN	2E	F	0.174	1.400	3.058	2.503	18.557	4.250	34.208	0.112	1.897	2.509	2.478	18.692	4.219	33.960
483	SN	2E	F	0.172	1.925	3.558	2.939	16.317	4.778	53.474	0.109	1.537	3.392	2.905	16.460	4.736	53.009
484	SN	2E	F	0.246	0.978	3.358	2.773	24.363	4.577	18.191	0.112	2.075	2.949	2.740	24.577	4.537	18.033
485	SN	2E	F	0.245	0.466	2.575	2.115	29.371	3.781	7.191	0.053	1.721	4.124	2.069	29.809	3.725	7.085
486	SN	2E	F	0.246	1.091	3.611	2.979	23.103	4.826	21.411	0.125	2.191	3.132	2.944	23.307	4.784	21.224
487	SN	2E	F	0.121	0.346	1.222	0.962	22.989	2.386	6.828	0.037	2.040	1.920	0.941	23.241	2.360	6.754
488	SB	2	F	0.043	0.312	0.328	0.180	13.690	1.440	10.342	0.017	1.762	2.044	0.158	13.957	1.413	10.144
489	SB	2	F	0.044	0.782	0.529	0.294	12.528	1.577	28.282	0.025	1.118	3.909	0.258	12.881	1.534	27.508
490	SB	2	F	0.043	1.156	0.665	0.369	11.741	1.668	44.629	0.023	0.761	5.112	0.322	12.152	1.611	43.118
491	SB	2	F	0.043	1.239	0.671	0.358	11.834	1.655	47.445	0.021	0.647	3.330	0.324	12.133	1.614	46.275
492	SB	2	F	0.043	1.406	0.725	0.417	11.369	1.726	56.040	0.024	0.655	4.388	0.381	11.660	1.683	54.639
493	SB	2	F	0.042	1.846	0.873	0.485	10.429	1.808	80.242	0.031	0.715	5.961	0.436	10.778	1.749	77.640
494	SB	2	F	0.043	2.721	1.154	0.692	9.530	2.059	129.415	0.031	0.533	10.452	0.586	10.162	1.931	121.365
495	SB	2	F	0.043	3.254	1.277	0.783	9.068	2.169	162.661	0.033	0.487	11.057	0.682	9.609	2.047	153.503
496	SB	2	F	0.044	3.444	1.317	0.797	9.083	2.186	171.835	0.040	0.570	11.089	0.706	9.561	2.077	163.260
497	SB	2	F	0.043	3.885	1.410	0.899	8.458	2.310	208.189	0.030	0.404	12.617	0.784	9.001	2.170	195.627
498	SB	2	F	0.042	3.645	1.289	0.801	8.587	2.191	192.377	0.049	0.693	11.181	0.687	9.160	2.054	180.343
499	SB	2	F	0.045	4.206	1.437	0.907	8.698	2.319	219.201	0.033	0.402	12.698	0.778	9.324	2.163	204.490
500	SB	2	F	0.042	3.963	1.358	0.870	8.429	2.275	213.135	0.035	0.459	12.589	0.755	8.978	2.136	200.106
501	SB	2	F	0.043	4.339	0.102	0.955	8.273	2.378	237.728	0.017	0.205	14.350	0.824	8.864	2.219	221.881
502	SB	2	F	0.031	0.748	0.524	0.293	8.913	1.577	38.020	0.027	1.765	4.702	0.241	9.286	1.513	36.493
503	SB	2	F	0.033	1.211	0.708	0.396	8.658	1.702	63.401	0.028	1.198	5.794	0.332	9.074	1.623	60.494
504	SB	2	F	0.031	1.758	0.888	0.520	7.690	1.851	103.633	0.030	0.977	8.402	0.451	8.050	1.768	99.005
505	SB	2	F	0.030	2.076	0.992	0.610	6.961	1.960	135.206	0.015	0.457	10.455	0.536	7.296	1.870	128.985
506	SB	2	F	0.031	2.330	1.072	0.668	6.834	2.030	154.540	0.020	0.548	10.940	0.579	7.216	1.922	146.347
507	SB	2	F	0.030	2.635	1.159	0.737	6.435	2.113	185.613	0.016	0.418	12.031	0.639	6.817	1.995	175.219
508	SB	2	F	0.032	3.097	1.263	0.803	6.532	2.193	214.932	0.018	0.383	12.911	0.698	6.933	2.066	202.485
509	SB	2	F	0.032	3.451	1.302	0.830	6.455	2.226	242.338	0.021	0.419	12.942	0.725	6.846	2.099	228.479
510	SB	2	F	0.032	4.082	1.427	0.927	6.190	2.343	298.933	0.024	0.411	14.270	0.811	6.584	2.203	281.027
511	SB	2	F	0.031	4.069	0.121	0.940	6.051	2.360	304.809	0.033	0.589	15.022	0.803	6.508	2.194	283.375
512	SB	2	F	0.024	0.544	0.411	0.228	7.204	1.497	34.254	0.013	1.412	4.080	0.199	7.378	1.462	33.449
513	SB	2	F	0.022	0.813	0.515	0.275	6.325	1.555	58.241	0.025	2.116	4.205	0.233	6.542	1.503	56.311
514	SB	2	F	0.024	1.212	0.694	0.388	6.406	1.691	85.760	0.019	1.054	6.564	0.334	6.660	1.627	82.486
515	SB	2	F	0.022	1.249	0.708	0.406	5.690	1.713	99.531	0.016	1.023	6.966	0.328	6.019	1.619	94.081
516	SB	2	F	0.024	1.544	0.844	0.494	5.903	1.820	118.589	0.023	1.098	8.521	0.391	6.339	1.695	110.440
517	SB	2	F	0.023	1.702	0.919	0.561	5.557	1.901	138.855	0.021	0.969	9.770	0.472	5.892	1.793	130.971
518	SB	2	F	0.064	1.309	0.819	0.420	16.763	1.731	35.389	0.068	1.367	1.797	0.400	17.001	1.706	34.894
519	SB	2	F	0.061	1.983	1.054	0.564	14.614	1.904	61.508	0.070	1.064	4.497	0.514	15.093	1.844	59.552
520	SB	2	F	0.064	0.598	0.549	0.305	18.294	1.591	14.823	0.027	1.076	3.167	0.266	18.845	1.544	14.389
521	SB	2	F	0.063	2.443	1.202	0.654	14.096	2.013	78.565	0.080	1.025	6.096	0.598	14.583	1.946	75.937
522	SB	2	F	0.063	3.239	1.441	0.838	12.729	2.236	115.329	0.038	0.407	10.245	0.724	13.566	2.098	108.214
523	SB	2	F	0.063	3.815	1.568	0.925	12.162	2.341	142.187	0.043	0.413	11.688	0.806	12.955	2.197	133.485
524	SB	2	F	0.060	4.103	1.606	0.976	11.397	2.402	163.160	0.037	0.354	12.532	0.849	12.176	2.249	152.728
525	SB	2	F	0.062	4.397	1.665	1.023	11.352	2.460	175.581	0.037	0.331	15.044	0.871	12.272	2.275	162.419
526	SB	2	F	0.061	4.667	1.677	1.060	11.024	2.504	191.879	0.037	0.318	14.054	0.917	11.838	2.332	178.680
527	SB	2	F	0.065	4.397	1.600	1.005	12.009	2.438	165.953	0.038	0.315	14.468	0.859	12.951	2.261	153.884
528	SB	2	F	0.063	5.158	0.134	1.131	11.023	2.591	212.092	0.026	0.200	16.588	0.963	11.961	2.387	195.447
529	SB	2	F	0.084	0.634	0.647	0.355	23.035	1.651	12.476	0.036	1.081	2.823	0.323	23.578	1.613	12.189
530	SB	2	F	0.086	1.034	0.903	0.461	21.923	1.780	21.374	0.077	1.500	2.588	0.433	22.360	1.745	20.956
531	SB	2	F	0.083	2.162	1.311	0.689	18.325	2.055	53.474	0.113	1.261	2.889	0.660	18.646	2.020	52.553
532	SB	2	F	0.086	3.135	1.628	0.879	17.036	2.285	83.421	0.126	1.041	5.731	0.821	17.577	2.215	80.857
533	SB	2	F	0.084	1.801	1.200	0.639	19.175	1.995	42.576	0.099	1.266	2.662	0.612	19.494	1.962	41.879
534	SB	2	F	0.083	3.562	1.726	0.974	15.711	2.400	102.760	0.100	0.789	6.983	0.910	16.233	2.323	99.458
535	SB	2	F	0.084	4.063	1.814	1.098	14.951	2.550	123.192	0.080	0.579	9.939	0.997	15.701	2.428	117.307
536	SB	2	F	0.084	4.182	1.838	1.090	14.894	2.541	127.265	0.065	0.457	12.169	0.967	15.823	2.392	119.796
537	SB	2	F	0.082	4.738	1.904	1.159	14.080	2.624	152.546	0.063	0.416	13.091	1.026	14.997	2.463	143.217
538	SB	2	F	0.086	5.179	2.001	1.211	14.440	2.687	162.571	0.076	0.450	13.271	1.077	15.370	2.524	152.730
539	SB	2	F	0.085	5.322	1.989	1.267	13.919	2.755	173.309	0.055	0.330	16.062	1.104	14.991	2.558	160.925
540	SB	2	F	0.083	5.394	0.097	1.242	13.857	2.725	176.434	0.057	0.338	15.287	1.087	14.881	2.537	164.302
541	SB	2	F	0.123	0.840	1.124	0.617	28.325	1.968	13.436	0.076	1.406	3.600	0.580	28.975	1.924	13.134
542	SB	2	F	0.123	1.465	1.498	0.833	25.007	2.230	26.557	0.120	1.448	3.543	0.797	25.503	2.186	26.039

RUN	INFO			MASS FLOW RATES		SUPP.	BEND OUTLET CONDITIONS				DELTA-P	LOSS	GRAD. IN	STRAIGHT PIPE CONDITIONS			
NO.	B	L	P	AIR	SOLIDS	PRES.	PRES.	VELOC.	RHOAIR	RHOSUS.	BEND	COEFF.	STRAIGHT	PRES.	VELOC.	RHOAIR	RHOSUS.
				kg/s	kg/s	barg	barg	m/s	kg/m3	kg/m3	bar		-mbar/m	barg	m/s	kg/m3	kg/m3
543	SB	2	F	0.122	2.049	1.771	0.975	23.032	2.401	40.325	0.163	1.523	2.915	0.942	23.415	2.362	39.665
544	SB	2	F	0.123	2.586	2.030	1.142	21.413	2.604	54.733	0.174	1.384	3.456	1.104	21.803	2.557	53.755
545	SB	2	F	0.123	3.071	2.071	1.163	21.209	2.629	65.629	0.178	1.204	3.617	1.123	21.609	2.580	64.413
546	SB	2	F	0.123	3.466	2.180	1.232	20.554	2.713	76.430	0.158	0.976	5.306	1.173	21.110	2.641	74.417
547	SB	2	F	0.123	3.798	2.278	1.315	19.816	2.813	86.863	0.154	0.905	6.265	1.246	20.429	2.729	84.259
548	SB	2	F	0.123	3.863	2.270	1.276	20.156	2.766	86.873	0.163	0.922	6.501	1.204	20.815	2.679	84.124
549	SB	2	F	0.122	4.107	2.300	1.330	19.536	2.831	95.303	0.145	0.800	8.765	1.241	20.306	2.723	91.686
550	SB	2	F	0.122	4.411	2.352	1.384	19.094	2.896	104.715	0.125	0.655	9.004	1.293	19.851	2.786	100.725
551	SB	2	F	0.121	4.858	2.345	1.398	18.830	2.913	116.938	0.110	0.532	10.048	1.296	19.661	2.790	111.994
552	SB	2	F	0.123	4.861	2.419	1.477	18.528	3.009	118.917	0.112	0.547	13.037	1.332	19.676	2.833	111.980
553	SB	2	F	0.123	5.005	2.446	1.450	18.733	2.976	121.116	0.124	0.582	12.606	1.310	19.866	2.806	114.208
554	SB	2	F	0.121	5.266	2.447	1.544	17.750	3.090	134.466	0.066	0.310	13.902	1.403	18.786	2.920	127.051
555	SB	2	F	0.121	5.402	2.483	1.495	18.096	3.031	135.312	0.098	0.440	14.852	1.345	19.253	2.849	127.185
556	SB	2	F	0.122	5.340	2.484	1.607	17.464	3.167	138.601	0.086	0.407	16.277	1.442	18.638	2.967	129.869
557	SB	2	F	0.122	5.693	2.588	1.647	17.205	3.214	149.981	0.095	0.427	16.664	1.478	18.373	3.010	140.450
558	SB	2	F	0.123	6.142	2.646	1.642	17.374	3.209	160.243	0.102	0.423	16.553	1.475	18.547	3.006	150.112
559	SB	2	F	0.123	6.376	2.656	1.572	17.845	3.124	161.962	0.125	0.485	17.589	1.394	19.167	2.909	150.787
560	SB	2	F	0.123	6.879	0.127	1.788	16.469	3.385	189.335	0.072	0.279	22.324	1.562	17.917	3.112	174.033
561	SB	2	F	0.174	0.866	1.489	0.848	35.093	2.247	11.182	0.091	1.329	4.767	0.795	36.125	2.183	10.863
562	SB	2	F	0.173	1.141	1.714	0.973	32.684	2.399	15.820	0.124	1.464	4.115	0.923	33.528	2.339	15.422
563	SB	2	F	0.174	1.656	2.046	1.179	29.780	2.648	25.207	0.153	1.371	4.282	1.131	30.443	2.591	24.658
564	SB	2	F	0.173	0.463	1.125	0.622	39.713	1.975	5.289	0.052	1.258	4.997	0.566	41.114	1.907	5.109
565	SB	2	F	0.174	1.989	2.255	1.322	27.952	2.822	32.251	0.171	1.361	4.589	1.271	28.578	2.760	31.545
566	SB	2	F	0.172	2.466	2.501	1.468	26.004	2.998	42.989	0.194	1.331	4.921	1.413	26.592	2.932	42.039
567	SB	2	F	0.173	3.027	2.749	1.611	24.731	3.171	55.472	0.214	1.259	5.134	1.554	25.282	3.102	54.262
568	SB	2	F	0.171	3.501	2.885	1.690	23.729	3.266	66.881	0.220	1.167	6.066	1.622	24.338	3.185	65.208
569	SB	2	F	0.174	3.845	3.000	1.757	23.558	3.348	73.986	0.247	1.202	5.692	1.694	24.110	3.271	72.292
570	SB	2	F	0.173	4.242	3.089	1.838	22.758	3.446	84.487	0.200	0.914	8.036	1.749	23.496	3.337	81.833
571	SB	2	F	0.174	4.543	3.157	1.967	21.896	3.602	94.035	0.196	0.870	8.476	1.873	22.613	3.488	91.055
572	SB	2	F	0.174	5.142	3.176	1.954	21.996	3.586	105.965	0.145	0.565	12.037	1.820	23.037	3.424	101.175
573	ABORTED																
574	SB	2	F	0.246	0.414	1.588	0.952	46.985	2.373	3.989	0.037	0.844	9.567	0.845	49.682	2.244	3.773
575	SB	2	F	0.245	1.018	2.173	1.308	39.593	2.805	11.660	0.098	1.068	8.507	1.214	41.278	2.690	11.184
576	SB	2	F	0.245	1.499	2.614	1.564	35.667	3.114	19.046	0.166	1.374	6.613	1.483	36.815	3.016	18.453
577	SB	2	F	0.244	1.925	2.925	1.774	32.837	3.368	26.572	0.198	1.380	6.485	1.695	33.792	3.273	25.821
578	SB	2	F	0.244	2.445	3.235	1.988	30.493	3.627	36.351	0.216	1.278	6.811	1.912	31.284	3.535	35.431
579	SB	2	F	0.246	2.805	3.470	2.143	29.233	3.814	43.492	0.228	1.227	7.232	2.062	29.999	3.717	42.381
580	SB	2	F	0.124	0.505	0.867	0.481	31.156	1.804	7.349	0.046	1.302	3.482	0.442	31.988	1.757	7.158
581	SB	2	F	0.085	0.300	0.448	0.250	25.283	1.524	5.387	0.023	1.309	2.101	0.226	25.762	1.496	5.287
582	SB	2	P	0.078	0.066	0.217	0.118	25.740	1.365	1.166	0.013	3.345	0.984	0.106	26.016	1.350	1.153
583	SB	2	P	0.078	0.300	0.350	0.193	24.255	1.456	5.615	0.012	0.716	2.073	0.170	24.729	1.428	5.507
584	SB	2	P	0.079	0.545	0.468	0.251	23.476	1.525	10.521	0.031	1.064	2.067	0.226	23.952	1.495	10.312
585	SB	2	P	0.079	0.739	0.551	0.307	22.551	1.594	14.862	0.026	0.679	3.064	0.273	23.151	1.553	14.478
586	SB	2	P	0.079	0.942	0.652	0.356	21.746	1.653	19.637	0.047	1.008	2.827	0.322	22.306	1.611	19.144
587	SB	2	P	0.078	1.122	0.744	0.418	20.576	1.727	24.714	0.047	0.898	3.716	0.372	21.247	1.673	23.934
588	SB	2	P	0.077	1.268	0.812	0.462	19.603	1.780	29.326	0.052	0.924	4.029	0.413	20.277	1.721	28.352
589	SB	2	P	0.077	1.387	0.868	0.506	19.035	1.834	33.032	0.057	0.945	4.510	0.451	19.748	1.767	31.840
590	SB	2	P	0.079	1.528	0.940	0.562	18.708	1.902	37.016	0.056	0.864	5.277	0.498	19.502	1.825	35.509
591	SB	2	P	0.078	1.655	0.997	0.606	18.148	1.956	41.335	0.061	0.896	5.862	0.535	18.984	1.870	39.517
592	SB	2	P	0.078	1.818	1.072	0.659	17.578	2.019	46.880	0.066	0.907	6.341	0.582	18.428	1.926	44.719
593	SB	2	P	0.080	2.017	0.074	0.713	17.349	2.085	52.689	0.080	1.012	6.283	0.631	18.221	1.985	50.167
594	SB	2	P	0.058	0.112	0.169	0.103	19.527	1.346	2.601	0.005	1.047	1.058	0.090	19.755	1.331	2.571
595	SB	2	P	0.058	0.290	0.265	0.151	18.715	1.405	7.030	0.012	0.979	1.577	0.132	19.029	1.382	6.914
596	SB	2	P	0.058	0.560	0.396	0.220	17.760	1.488	14.296	0.026	1.134	1.832	0.198	18.087	1.461	14.038
597	SB	2	P	0.058	0.733	0.457	0.259	17.216	1.535	19.307	0.034	1.202	1.965	0.235	17.546	1.506	18.944
598	SB	2	P	0.058	0.861	0.510	0.295	16.742	1.578	23.301	0.038	1.164	2.336	0.266	17.114	1.544	22.795
599	SB	2	P	0.058	1.034	0.588	0.347	16.098	1.642	29.108	0.042	1.119	2.917	0.311	16.529	1.599	28.349
600	SB	2	P	0.057	1.184	0.649	0.388	15.412	1.691	34.821	0.046	1.114	3.385	0.347	15.879	1.641	33.798
601	SB	2	P	0.059	1.341	0.728	0.440	15.140	1.754	40.159	0.049	1.075	4.016	0.391	15.666	1.695	38.809
602	SB	2	P	0.054	1.507	0.800	0.491	13.555	1.816	50.406	0.053	1.148	4.599	0.435	14.078	1.748	48.533
603	SB	2	P	0.058	1.882	0.136	0.660	13.037	2.020	65.440	0.060	1.074	6.455	0.581	13.678	1.925	62.371
604	SB	2	P	0.043	0.108	0.128	0.079	14.756	1.318	3.316	0.003	0.964	0.672	0.072	14.858	1.309	3.294
605	SB	2	P	0.042	0.290	0.206	0.124	13.715	1.371	9.593	0.009	0.974	1.248	0.108	13.901	1.353	9.465
606	SB	2	P	0.044	0.550	0.332	0.196	13.795	1.459	18.071	0.021	1.194	1.843	0.173	14.056	1.432	17.736
607	SB	2	P	0.043	0.713	0.371	0.220	13.192	1.488	24.497	0.027	1.244	2.000	0.195	13.458	1.458	24.013
608	SB	2	P	0.042	0.746	0.414	0.245	12.571	1.518	26.901	0.029	1.354	2.347	0.216	12.862	1.484	26.291
609	SB	2	P	0.044	0.920	0.484	0.288	12.813	1.571	32.552	0.035	1.314	2.659	0.256	13.140	1.532	31.743
610	SB	2	P	0.043	1.057	0.544	0.315	12.212	1.603	39.227	0.051	1.759	2.442	0.281	12.539	1.562	38.205

RUN NO.	INFO B L P	MASS FLOW RATES AIR kg/s	MASS FLOW RATES SOLIDS kg/s	SUPP. PRES. barg	BEND OUTLET CONDITIONS PRES. barg	BEND OUTLET CONDITIONS VELOC. m/s	BEND OUTLET CONDITIONS RHOAIR kg/m3	BEND OUTLET CONDITIONS RHOSUS. kg/m3	DELTA-P BEND bar	LOSS COEFF.	GRAD. IN STRAIGHT -mbar/m	STRAIGHT PIPE CONDITIONS PRES. barg	STRAIGHT PIPE CONDITIONS VELOC. m/s	STRAIGHT PIPE CONDITIONS RHOAIR kg/m3	STRAIGHT PIPE CONDITIONS RHOSUS. kg/m3
611	ABORTED														
612	ABORTED														
613	SB 2 P	0.030	0.117	0.108	0.071	10.437	1.307	5.087	0.004	1.595	0.508	0.064	10.497	1.300	5.058
614	SB 2 P	0.030	0.266	0.219	0.134	9.859	1.384	12.245	0.013	2.133	1.251	0.121	9.970	1.369	12.109
615	SB 2 P	0.031	0.418	0.330	0.198	9.491	1.461	19.943	0.017	1.912	2.254	0.175	9.674	1.434	19.566
616	ABORTED														
617	ABORTED														
618	ABORTED														
619	SB 2 P	0.031	0.609	0.176	1.060	5.703	2.504	48.412	0.001	0.132	21.576	0.863	6.302	2.265	43.804
620	SB 2 P	0.023	0.142	0.200	0.126	7.651	1.374	8.437	0.007	2.660	1.752	0.108	7.773	1.353	8.305
621	ABORTED														
622	ABORTED														
623	ABORTED														
624	ABORTED														
625	ABORTED														
626	SB 2 P	0.015	0.134	0.380	0.283	4.347	1.564	13.947	0.004	2.939	4.220	0.253	4.451	1.528	13.622
627	ABORTED														
628	ABORTED														
629	ABORTED														
630	ABORTED														
631	ABORTED														
632	SB 2 P	0.112	0.176	0.445	0.248	33.344	1.522	2.388	0.014	1.063	2.906	0.216	34.224	1.483	2.327
633	SB 2 P	0.113	0.581	0.637	0.349	31.162	1.644	8.444	0.018	0.440	4.067	0.307	32.137	1.594	8.188
634	SB 2 P	0.112	0.881	0.776	0.428	29.178	1.740	13.686	0.025	0.434	4.808	0.370	30.412	1.669	13.131
635	SB 2 P	0.112	1.163	0.913	0.510	27.609	1.839	19.087	0.038	0.520	5.045	0.454	28.668	1.771	18.382
636	SB 2 P	0.112	1.432	1.026	0.572	26.531	1.913	24.460	0.058	0.676	5.007	0.516	27.500	1.846	23.598
637	SB 2 P	0.112	1.657	1.150	0.655	25.205	2.014	29.801	0.064	0.679	5.927	0.583	26.343	1.927	28.514
638	SB 2 P	0.111	1.876	1.225	0.703	24.280	2.072	35.020	0.074	0.719	5.932	0.631	25.350	1.985	33.548
639	SB 2 P	0.111	2.129	1.338	0.790	23.100	2.178	41.780	0.068	0.609	7.406	0.701	24.310	2.070	39.696
640	SB 2 P	0.111	2.436	1.518	0.930	21.440	2.347	51.509	0.060	0.508	9.238	0.818	22.750	2.212	48.535
641	SB 2 P	0.111	2.848	0.116	1.182	18.970	2.652	68.043	0.084	0.686	10.680	1.053	20.160	2.496	64.039
642	SB 2 P	0.157	0.327	0.792	0.449	40.330	1.765	3.680	0.012	0.401	5.825	0.378	42.380	1.679	3.502
643	SB 2 P	0.157	0.786	1.050	0.581	36.960	1.925	9.635	0.028	0.429	6.694	0.500	38.950	1.827	9.144
644	SB 2 P	0.157	1.171	1.244	0.690	34.600	2.057	15.342	0.044	0.477	7.262	0.602	36.490	1.950	14.547
645	SB 2 P	0.156	1.478	1.396	0.785	32.560	2.172	20.578	0.044	0.407	8.179	0.686	34.460	2.052	19.441
646	SB 2 P	0.156	1.791	1.488	0.854	31.360	2.255	25.891	0.048	0.373	8.311	0.753	33.150	2.133	24.491
647	SB 2 P	0.156	2.045	1.589	0.923	30.230	2.339	30.662	0.060	0.429	8.535	0.820	31.940	2.214	29.018
648	SB 2 P	0.156	2.399	1.793	1.055	28.300	2.498	38.419	0.080	0.521	8.969	0.947	29.870	2.367	36.405
649	SB 2 P	0.156	2.624	1.886	1.130	27.320	2.589	43.549	0.082	0.503	9.719	1.012	28.910	2.446	41.148
650	SB 2 P	0.157	2.844	1.987	1.223	26.340	2.702	48.938	0.069	0.405	11.520	1.083	28.100	2.532	45.870
651	ABORTED														
652	SB 2 P	0.219	0.396	1.235	0.728	47.210	2.103	3.804	0.004	0.100	9.918	0.608	50.720	1.957	3.541
653	SB 2 P	0.220	0.947	1.525	0.883	43.540	2.290	9.861	0.022	0.240	10.570	0.755	46.700	2.135	9.193
654	SB 2 P	0.219	1.351	1.742	1.011	40.600	2.445	15.084	0.041	0.330	10.980	0.878	43.460	2.284	14.090
655	SB 2 P	0.212	1.761	1.982	1.148	36.810	2.611	21.687	0.062	0.421	11.300	1.011	39.300	2.445	20.310
656	SB 2 P	0.219	2.209	2.242	1.307	35.420	2.803	28.275	0.082	0.464	11.650	1.165	37.720	2.631	26.542
657	SB 2 P	0.218	2.536	2.299	1.358	34.500	2.865	33.327	0.086	0.433	11.710	1.216	36.700	2.693	31.329
658	SB 2 P	0.220	2.913	2.435	1.451	33.500	2.977	39.423	0.103	0.466	11.810	1.307	35.570	2.803	37.117
659	SB 2 P	0.219	3.148	2.518	1.517	32.470	3.057	43.946	0.109	0.473	12.190	1.381	34.310	2.893	41.581
660	SB 2 P	0.218	3.486	2.663	1.638	30.840	3.204	51.229	0.104	0.427	13.420	1.476	32.860	3.008	48.095
661	SN 2S F	0.079	1.422	0.809	0.266	23.200	1.544	27.785	0.124	1.660	2.900	0.234	23.800	1.505	27.084
662	SN 2S F	0.080	2.563	1.142	0.376	21.510	1.677	54.007	0.165	1.321	4.846	0.323	22.380	1.612	51.907
663	SN 2S F	0.080	3.320	1.288	0.427	20.860	1.739	72.154	0.184	1.170	5.117	0.370	21.720	1.670	69.292
664	SN 2S F	0.079	3.596	1.338	0.480	19.930	1.803	81.772	0.204	1.255	7.079	0.402	21.050	1.708	77.449
665	SN 2S F	0.080	4.470	1.541	0.618	18.410	1.970	110.072	0.150	0.804	13.090	0.499	19.860	1.825	101.994
666	SN 2S F	0.078	5.038	1.568	0.690	17.240	2.056	132.487	0.151	0.768	13.390	0.541	18.890	1.876	120.866
667	SN 2S F	0.080	4.488	1.480	0.619	18.400	1.971	110.563	0.146	0.778	12.660	0.478	20.140	1.800	100.993
668	SN 2S F	0.080	5.752	1.639	0.791	16.560	2.179	157.451	0.090	0.418	17.920	0.592	16.560	1.938	140.019
669	SN 2S F	0.079	5.919	1.626	0.867	15.830	2.271	169.507	0.052	0.244	21.290	0.630	18.110	1.984	148.114
670	SN 2S F	0.081	0.626	0.520	0.194	25.230	1.457	11.254	0.050	1.387	3.093	0.160	25.970	1.415	10.931
671	SN 2S F	0.081	2.233	1.110	0.359	22.250	1.656	45.491	0.167	1.486	4.365	0.306	23.140	1.592	43.733
672	SN 2S F	0.081	1.670	0.915	0.293	23.280	1.577	32.509	0.149	1.688	3.226	0.257	23.280	1.533	31.613
673	SN 2S F	0.080	1.136	0.731	0.243	23.920	1.516	21.528	0.105	1.711	2.217	0.218	24.400	1.486	21.103
674	SN 2S F	0.058	0.443	0.355	0.142	18.920	1.394	10.611	0.031	1.638	2.291	0.114	19.390	1.360	10.355
675	SN 2S F	0.055	1.334	0.672	0.265	16.220	1.542	37.269	0.068	1.385	3.160	0.226	16.730	1.496	36.149
676	SN 2S F	0.058	1.777	0.803	0.309	16.480	1.595	48.873	0.097	1.457	3.174	0.273	16.930	1.552	47.564
677	SN 2S F	0.058	0.758	0.487	0.209	17.760	1.475	19.354	0.046	1.495	3.549	0.170	18.360	1.427	18.727
678	SN 2S F	0.058	1.906	0.841	0.336	16.030	1.628	53.881	0.083	1.203	4.111	0.290	16.600	1.573	52.052
679	SN 2S F	0.056	2.261	0.940	0.374	15.240	1.674	67.239	0.099	1.262	5.189	0.316	15.910	1.604	64.431

RUN NO.	INFO B	INFO L	INFO P	MASS FLOW RATES AIR kg/s	MASS FLOW RATES SOLIDS kg/s	SUPP. PRES. barg	BEND OUTLET CONDITIONS PRES. barg	BEND OUTLET CONDITIONS VELOC. m/s	BEND OUTLET CONDITIONS RHOAIR kg/m3	BEND OUTLET CONDITIONS RHOSUS. kg/m3	DELTA-P BEND bar	LOSS COEFF.	GRAD. IN STRAIGHT -mbar/m	STRAIGHT PIPE CONDITIONS PRES. barg	STRAIGHT PIPE CONDITIONS VELOC. m/s	STRAIGHT PIPE CONDITIONS RHOAIR kg/m3	STRAIGHT PIPE CONDITIONS RHOSUS. kg/m3
680	SN	2S	F	0.058	2.597	1.030	0.399	15.550	1.705	75.686	0.122	1.333	5.443	0.344	16.190	1.638	72.726
681	SN	2S	F	0.058	2.881	1.096	0.471	14.620	1.792	89.297	0.105	1.102	7.508	0.387	15.500	1.690	84.256
682	SN	2S	F	0.058	3.028	1.131	0.478	14.680	1.800	93.469	0.111	1.100	8.485	0.400	15.490	1.706	88.606
683	SN	2S	F	0.058	3.210	1.154	0.508	14.390	1.836	101.106	0.097	0.929	8.629	0.446	14.390	1.762	97.009
684	SN	2S	F	0.057	3.697	1.256	0.579	13.440	1.922	124.680	0.072	0.643	12.220	0.467	14.460	1.787	115.923
685	SN	2S	F	0.058	4.107	1.325	0.638	13.120	1.993	141.882	0.055	0.453	14.360	0.478	14.530	1.800	128.121
686	SN	2S	F	0.058	4.297	1.331	0.655	13.120	2.014	148.434	0.046	0.358	14.180	0.511	14.360	1.840	135.629
687	SN	2S	F	0.057	4.631	1.343	0.701	12.480	2.070	168.190	0.056	0.431	14.980	0.549	13.700	1.887	153.279
688	SN	2S	F	0.057	5.194	1.421	0.703	12.570	2.073	187.227	0.078	0.530	14.480	0.556	13.750	1.895	171.184
689	SN	2S	F	0.040	0.912	0.503	0.236	11.906	1.508	34.704	0.013	0.533	5.149	0.179	12.480	1.438	33.108
690	SN	2S	F	0.039	1.319	0.670	0.314	11.011	1.601	54.296	0.035	1.067	6.152	0.251	11.555	1.526	51.740
691	SN	2S	F	0.041	1.971	0.857	0.405	10.882	1.712	82.093	0.046	0.939	7.894	0.325	11.533	1.615	77.454
692	SN	2S	F	0.040	2.397	0.937	0.461	10.135	1.780	107.220	0.042	0.759	9.781	0.362	10.867	1.660	99.999
693	SN	2S	F	0.040	2.628	0.996	0.517	9.891	1.847	120.423	0.029	0.489	11.268	0.402	10.690	1.709	111.418
694	SN	2S	F	0.041	2.847	1.061	0.554	9.822	1.892	131.391	0.020	0.312	13.125	0.421	10.734	1.731	120.222
695	SN	2S	F	0.040	3.221	1.113	0.588	9.470	1.934	154.186	0.036	0.515	12.785	0.472	10.216	1.792	142.925
696	SN	2S	F	0.041	3.819	1.208	0.636	9.309	1.992	185.979	0.033	0.409	13.032	0.504	10.120	1.832	171.064
697	SN	2S	F	0.041	4.241	1.280	0.683	9.053	2.048	212.354	0.031	0.357	15.169	0.529	9.956	1.862	193.076
698	SN	2S	F	0.041	4.105	1.224	0.676	9.066	2.040	205.205	0.028	0.337	14.263	0.546	9.825	1.882	189.355
699	SN	2S	F	0.034	0.446	0.332	0.162	10.711	1.418	18.864	0.015	1.376	3.320	0.135	10.963	1.385	18.429
700	SN	2S	F	0.060	1.184	0.644	0.270	17.568	1.548	30.560	0.065	1.381	3.547	0.230	18.127	1.500	29.617
701	SN	2S	F	0.070	2.272	0.987	0.345	19.388	1.639	53.122	0.123	1.232	3.582	0.305	19.976	1.591	51.559
702	SN	2S	F	0.071	0.699	0.511	0.202	21.789	1.467	14.538	0.037	1.084	3.445	0.167	22.435	1.424	14.120
703	SN	2S	F	0.063	2.987	1.164	0.410	16.570	1.718	81.701	0.152	1.351	4.973	0.355	17.242	1.651	78.516
704	SN	2S	F	0.048	0.784	0.491	0.211	14.694	1.478	24.174	0.039	1.480	3.796	0.173	15.171	1.431	23.413
705	SN	2S	F	0.069	1.705	0.838	0.301	19.776	1.586	39.078	0.113	1.480	3.745	0.256	20.486	1.531	37.723
706	SN	2S	F	0.049	1.827	0.805	0.337	13.739	1.630	60.281	0.090	1.584	5.167	0.290	14.238	1.573	58.170
707	SN	2S	F	0.049	2.473	0.976	0.451	12.441	1.767	90.104	0.062	0.891	8.888	0.352	13.345	1.647	84.001
708	SN	2S	F	0.099	0.489	0.542	0.193	30.899	1.455	7.172	0.040	1.159	3.860	0.154	31.937	1.408	6.939
709	SN	2S	F	0.101	0.759	0.660	0.221	30.748	1.489	11.186	0.055	1.045	3.787	0.175	31.940	1.433	10.768
710	SN	2S	F	0.101	1.025	0.794	0.263	29.723	1.540	15.636	0.126	1.822	4.266	0.216	30.874	1.483	15.052
711	SN	2S	F	0.101	2.267	1.256	0.370	27.414	1.670	37.486	0.223	1.586	4.924	0.316	28.548	1.604	35.997
712	SN	2S	F	0.101	2.134	1.229	0.358	27.663	1.655	34.973	0.215	1.605	4.869	0.299	28.911	1.583	33.463
713	SN	2S	F	0.101	1.967	1.160	0.332	28.187	1.624	31.626	0.213	1.699	3.833	0.290	29.112	1.573	30.621
714	SN	2S	F	0.072	3.485	1.265	0.481	18.022	1.803	87.644	0.139	0.974	8.079	0.407	18.960	1.714	83.307
715	LW	3	F	0.159	1.251	0.956	0.255	19.912	1.531	12.043	0.045	1.874	2.056	0.230	20.312	1.500	11.806
716	LW	3	F	0.162	1.619	1.115	0.288	19.770	1.571	15.701	0.052	1.679	1.534	0.274	19.986	1.554	15.532
717	LW	3	F	0.161	1.992	1.245	0.318	19.206	1.607	19.884	0.073	1.983	1.793	0.300	19.472	1.585	19.612
718	LW	3	F	0.159	2.546	1.401	0.382	18.092	1.685	26.978	0.065	1.461	2.225	0.360	18.390	1.657	26.542
719	LW	3	F	0.160	2.629	1.336	0.385	18.171	1.688	27.732	0.052	1.127	2.808	0.357	18.549	1.653	27.167
720	LW	3	F	0.160	6.050	2.024	0.609	15.658	1.959	74.062	0.085	0.935	2.095	0.586	15.887	1.930	72.995
721	LW	3	F	0.160	6.798	2.118	0.658	15.201	2.018	85.732	0.076	0.766	6.047	0.590	15.840	1.936	82.272
722	LW	3	F	0.161	7.327	2.169	0.724	14.715	2.097	95.449	0.067	0.651	6.266	0.660	15.274	2.021	91.954
723	LW	3	F	0.161	7.596	2.108	0.770	14.330	2.154	101.615	0.065	0.625	6.077	0.709	14.843	2.079	98.100
724	ABORTED																
725	LW	3	F	0.110	1.744	0.940	0.309	13.211	1.596	25.312	0.022	0.993	3.436	0.292	13.390	1.575	24.973
726	LW	3	F	0.110	3.356	1.272	0.409	12.282	1.717	52.384	0.046	1.164	3.754	0.375	12.586	1.675	51.118
727	LW	3	F	0.109	3.770	1.330	0.448	11.847	1.764	60.996	0.037	0.873	4.415	0.407	12.184	1.715	59.309
728	LW	3	F	0.109	4.524	1.433	0.496	11.471	1.821	75.605	0.035	0.706	5.051	0.444	11.875	1.760	73.036
729	LW	3	F	0.109	5.351	1.524	0.575	10.893	1.918	94.159	0.040	0.711	7.291	0.502	11.425	1.829	89.772
730	LW	3	F	0.111	5.747	1.552	0.597	10.942	1.945	100.681	0.041	0.684	7.045	0.526	11.451	1.858	96.210
731	LW	3	F	0.109	6.142	1.550	0.607	10.678	1.957	110.263	0.020	0.323	8.407	0.522	11.272	1.854	104.457
732	LW	3	F	0.110	6.853	1.614	0.685	10.284	2.050	127.743	0.026	0.380	9.163	0.592	10.879	1.938	120.746
733	LW	3	F	0.109	6.974	1.597	0.654	10.379	2.013	128.801	0.027	0.391	6.726	0.586	10.822	1.931	123.526
734	LW	3	F	0.110	7.201	0.077	0.662	10.422	2.023	132.460	0.024	0.333	6.196	0.600	10.828	1.947	127.489
735	LW	3	F	0.240	2.028	1.777	0.430	26.412	1.742	14.716	0.093	1.816	2.122	0.404	26.893	1.711	14.453
736	LW	3	F	0.239	3.566	2.298	0.618	23.259	1.970	29.388	0.121	1.516	1.945	0.594	23.602	1.941	28.962
737	LW	3	F	0.240	4.975	2.686	0.739	21.741	2.116	43.866	0.139	1.337	2.745	0.706	22.163	2.076	43.031
738	LW	3	F	0.239	5.867	2.829	0.807	20.839	2.198	53.962	0.146	1.246	2.949	0.771	21.258	2.155	52.900
739	LW	3	F	0.240	7.129	3.037	0.891	20.006	2.300	68.312	0.147	1.073	2.340	0.862	20.309	2.265	67.292
740	LW	3	F	0.238	7.724	2.964	0.878	19.975	2.284	74.120	0.128	0.869	3.537	0.842	20.361	2.241	72.713
741	LW	3	F	0.240	1.729	1.531	0.371	27.541	1.670	12.034	0.069	1.513	1.819	0.345	28.064	1.639	11.810
742	LW	3	F	0.239	1.942	1.685	0.421	26.466	1.731	14.065	0.087	1.776	1.835	0.399	26.885	1.704	13.846
743	LW	3	F	0.239	2.876	2.110	0.566	24.031	1.906	22.939	0.098	1.485	3.353	0.532	24.560	1.865	22.445
744	LW	3	F	0.238	4.297	2.510	0.676	22.368	2.040	36.820	0.128	1.390	2.369	0.647	22.756	2.005	36.192
745	LW	3	F	0.081	1.821	0.784	0.278	9.960	1.559	35.047	0.022	1.254	4.658	0.236	10.300	1.507	33.890
746	LW	3	F	0.080	2.769	0.951	0.358	9.217	1.655	57.587	0.025	1.031	5.297	0.310	9.555	1.597	55.551
747	LW	3	F	0.076	3.645	1.085	0.442	8.293	1.757	84.244	0.027	0.920	6.817	0.380	8.664	1.681	80.633
748	LW	3	F	0.078	4.131	1.172	0.486	8.219	1.810	96.330	0.017	0.537	7.967	0.413	8.640	1.722	91.646
749	LW	3	F	0.077	5.433	1.280	0.572	7.729	1.915	134.724	0.016	0.409	9.113	0.489	8.158	1.814	127.641
750	LW	3	F	0.075	5.255	1.241	0.550	7.635	1.888	131.942	0.008	0.208	9.573	0.463	8.088	1.782	124.552

RUN NO.	INFO B	INFO L	INFO P	MASS FLOW RATES AIR kg/s	MASS FLOW RATES SOLIDS kg/s	SUPP. PRES. barg	BEND OUTLET CONDITIONS PRES. barg	BEND OUTLET CONDITIONS VELOC. m/s	BEND OUTLET CONDITIONS RHOAIR kg/m3	BEND OUTLET CONDITIONS RHOSUS. kg/m3	DELTA-P BEND bar	LOSS COEFF.	GRAD. IN STRAIGHT -mbar/m	STRAIGHT PIPE CONDITIONS PRES. barg	STRAIGHT PIPE CONDITIONS VELOC. m/s	STRAIGHT PIPE CONDITIONS RHOAIR kg/m3	STRAIGHT PIPE CONDITIONS RHOSUS. kg/m3
751	LU	3	F	.0.078	5.627	1.257	0.583	7.756	1.928	139.077	0.015	0.355	9.266	0.499	8.191	1.825	131.693
752	LU	3	F	0.076	5.936	1.265	0.608	7.442	1.958	152.903	0.007	0.171	9.424	0.522	7.860	1.854	144.772
753	LU	3	F	0.076	6.549	1.273	0.632	7.384	1.986	170.018	0.010	0.223	9.377	0.555	7.743	1.894	162.121
754	LU	3	F	0.076	7.196	0.058	0.643	7.322	2.000	188.391	0.019	0.370	9.430	0.567	7.678	1.907	179.655
755	LU	3	F	0.079	0.862	0.474	0.167	10.687	1.424	15.464	0.014	1.625	2.240	0.144	10.897	1.397	15.166
756	LU	3	F	0.161	5.930	2.027	0.556	16.287	1.895	69.797	0.094	1.012	2.263	0.529	16.578	1.862	68.573
757	LU	3	F	0.161	5.191	1.991	0.511	16.769	1.840	59.335	0.111	1.329	1.526	0.494	16.959	1.820	58.673
758	LU	3	F	0.159	5.183	1.994	0.520	16.464	1.851	60.344	0.095	1.162	2.014	0.496	16.732	1.822	59.380
759	LU	3	F	0.158	4.598	1.873	0.471	16.902	1.792	52.143	0.096	1.292	1.280	0.457	17.067	1.775	51.641
760	LU	3	F	0.159	4.481	1.886	0.469	17.029	1.790	50.446	0.092	1.254	1.309	0.455	17.198	1.772	49.949
761	LU	3	F	0.159	3.927	1.756	0.442	17.351	1.757	43.388	0.083	1.278	1.531	0.425	17.557	1.736	42.878
762	EF	3	F	0.079	1.210	0.584	0.226	10.104	1.495	22.955	0.015	1.251	3.737	0.192	10.391	1.454	22.321
763	EF	3	F	0.081	2.751	0.953	0.363	9.350	1.661	56.408	0.030	1.224	5.391	0.319	9.658	1.608	54.607
764	EF	3	F	0.080	3.457	1.095	0.440	8.741	1.754	75.810	0.032	1.099	7.072	0.383	9.102	1.685	72.803
765	EF	3	F	0.079	3.810	1.129	0.455	8.579	1.772	85.137	0.037	1.196	6.545	0.401	8.903	1.707	82.044
766	EF	3	F	0.080	4.617	1.234	0.537	8.195	1.871	108.003	0.016	0.452	9.589	0.449	8.686	1.765	101.890
767	EF	3	F	0.082	5.205	1.303	0.570	8.205	1.911	121.591	0.028	0.675	9.163	0.486	8.664	1.810	115.151
768	EF	3	F	0.080	5.597	1.310	0.577	8.007	1.920	133.996	0.041	0.950	9.256	0.492	8.458	1.818	126.860
769	EF	3	F	0.080	6.319	1.292	0.611	7.779	1.962	155.714	0.030	0.647	9.658	0.523	8.226	1.855	147.245
770	EF	3	F	0.081	6.718	1.426	0.676	7.619	2.040	169.004	0.027	0.553	11.225	0.574	8.112	1.916	158.733
771	EF	3	F	0.080	6.518	0.714	0.598	7.863	1.945	158.902	0.062	1.262	6.571	0.538	8.168	1.873	152.973
772	EF	3	F	0.079	1.949	0.795	0.300	9.580	1.585	38.993	0.018	0.987	5.641	0.248	9.973	1.522	37.460
773	EF	3	F	0.110	1.178	0.737	0.239	13.949	1.512	16.185	0.017	1.108	2.627	0.221	14.162	1.489	15.942
774	EF	3	F	0.109	2.255	1.056	0.321	12.976	1.610	33.311	0.048	1.704	1.252	0.311	13.076	1.598	33.056
775	EF	3	F	0.109	2.721	1.124	0.365	12.564	1.663	41.518	0.047	1.439	2.851	0.341	12.779	1.635	40.818
776	EF	3	F	0.111	3.702	1.330	0.422	12.282	1.732	57.770	0.067	1.532	2.283	0.403	12.443	1.710	57.021
777	EF	3	F	0.109	4.308	1.411	0.470	11.670	1.790	70.763	0.078	1.620	2.601	0.451	11.819	1.768	69.876
778	EF	3	F	0.112	4.962	1.516	0.514	11.647	1.843	81.661	0.076	1.366	2.419	0.496	11.780	1.823	80.737
779	EF	3	F	0.109	5.656	1.557	0.564	10.974	1.904	98.804	0.085	1.435	4.294	0.533	11.192	1.867	96.881
780	EF	3	F	0.111	5.819	1.565	0.582	11.049	1.926	100.953	0.072	1.174	5.945	0.533	11.395	1.867	97.888
781	EF	3	F	0.109	6.054	1.509	0.597	10.750	1.944	107.949	0.052	0.826	6.644	0.543	11.124	1.878	104.320
782	EF	3	F	0.110	7.022	1.585	0.648	10.513	2.006	128.028	0.063	0.890	8.495	0.570	11.029	1.912	122.038
783	EF	3	F	0.109	6.722	1.599	0.662	10.330	2.023	124.741	0.062	0.925	8.059	0.588	10.805	1.934	119.250
784	EF	3	F	0.111	7.815	0.063	0.717	10.183	2.089	147.110	0.039	0.510	9.848	0.627	10.742	1.981	139.451
785	EF	3	F	0.165	1.446	1.041	0.250	20.755	1.524	13.358	0.080	2.795	0.833	0.242	20.881	1.515	13.277
786	EF	3	F	0.164	1.174	0.949	0.241	20.775	1.513	10.831	0.051	2.196	1.496	0.224	21.055	1.493	10.686
787	EF	3	F	0.164	3.030	1.635	0.389	18.570	1.693	31.274	0.122	2.260	1.383	0.374	18.777	1.674	30.930
788	EF	3	F	0.162	2.294	1.401	0.341	18.996	1.635	23.152	0.098	2.336	1.553	0.322	19.265	1.612	22.830
789	EF	3	F	0.163	2.907	1.591	0.381	18.571	1.682	30.002	0.130	2.508	1.270	0.367	18.761	1.665	29.697
790	EF	3	F	0.166	4.402	1.969	0.481	17.645	1.803	47.824	0.155	2.080	1.716	0.460	17.895	1.778	47.156
791	EF	3	F	0.164	4.821	2.015	0.495	17.268	1.821	53.516	0.167	2.089	1.745	0.474	17.514	1.795	52.763
792	EF	3	F	0.164	5.215	2.025	0.518	17.003	1.849	58.788	0.162	1.903	1.528	0.504	17.160	1.832	58.251
793	EF	3	F	0.164	5.556	2.013	0.553	16.626	1.891	64.056	0.179	2.027	3.335	0.512	17.068	1.842	62.398
794	EF	3	F	0.165	6.147	2.132	0.622	16.022	1.974	73.545	0.187	1.981	2.927	0.598	16.260	1.945	72.472
795	EF	3	F	0.166	7.234	2.174	0.678	15.578	2.043	89.021	0.195	1.801	3.140	0.647	15.877	2.004	87.343
796	EF	3	F	0.165	6.983	2.049	0.710	15.196	2.081	88.081	0.130	1.278	5.639	0.648	15.772	2.005	84.867
797	EF	3	F	0.160	8.535	2.158	0.777	14.185	2.162	115.339	0.138	1.189	7.531	0.701	14.817	2.070	110.415
798	EF	3	F	0.166	7.759	0.068	0.738	15.050	2.114	98.827	0.125	1.121	7.756	0.651	15.832	2.010	93.946
799	EF	3	F	0.247	1.657	1.565	0.382	28.115	1.684	11.294	0.096	2.157	3.822	0.343	28.920	1.637	10.980
800	EF	3	F	0.249	1.518	1.578	0.374	28.513	1.674	10.202	0.112	2.694	1.784	0.356	28.891	1.652	10.069
801	EF	3	F	0.247	1.706	1.700	0.410	27.566	1.718	11.864	0.114	2.532	2.536	0.384	28.074	1.686	11.650
802	EF	3	F	0.246	3.042	2.239	0.580	24.512	1.924	23.789	0.186	2.607	1.726	0.563	24.785	1.903	23.528
803	EF	3	F	0.247	3.813	2.521	0.647	23.612	2.005	30.957	0.223	2.588	1.384	0.633	23.814	1.988	30.695
804	EF	3	F	0.247	4.318	2.632	0.694	22.973	2.061	36.026	0.224	2.359	2.037	0.673	23.254	2.036	35.590
805	EF	3	F	0.247	5.146	2.810	0.765	22.054	2.147	44.732	0.251	2.308	2.077	0.743	22.318	2.121	44.202
806	EF	3	F	0.247	5.886	2.995	0.844	21.104	2.243	53.460	0.260	2.186	2.549	0.819	21.402	2.212	52.716
807	EF	3	F	0.246	6.552	3.130	0.881	20.609	2.288	60.941	0.294	2.275	2.605	0.855	20.900	2.256	60.091
808	EF	3	F	0.248	6.400	3.008	0.837	21.276	2.234	57.666	0.287	2.200	2.554	0.811	21.578	2.203	56.858

RUN NO.	INFO B L P	MASS FLOW RATES AIR kg/s	MASS FLOW RATES SOLIDS kg/s	SUPP. PRES. barg	BEND OUTLET CONDITIONS PRES. barg	BEND OUTLET CONDITIONS VELOC. m/s	BEND OUTLET CONDITIONS RHOAIR kg/m3	BEND OUTLET CONDITIONS RHOSUS. kg/m3	DELTA-P BEND bar	LOSS COEFF.	GRAD. IN STRAIGHT -mbar/m	STRAIGHT PIPE CONDITIONS PRES. barg	STRAIGHT PIPE CONDITIONS VELOC. m/s	STRAIGHT PIPE CONDITIONS RHOAIR kg/m3	STRAIGHT PIPE CONDITIONS RHOSUS. kg/m3
809	ABORTED														
810 to 841 Data uninterpretable															
842	L 4 F	0.179	8.128	1.657	0.568	10.829	1.909	86.674	0.023	0.455	6.571	0.488	11.406	1.812	82.295
843	L 4 F	0.179	2.781	1.155	0.286	13.180	1.568	24.364	0.024	1.134	3.229	0.260	13.452	1.537	23.870
844	L 4 F	0.178	3.779	1.344	0.349	12.507	1.644	34.892	0.034	1.236	4.349	0.313	12.841	1.601	33.984
845	L 4 F	0.174	4.847	1.506	0.417	11.641	1.726	48.086	0.016	0.499	5.357	0.363	12.101	1.661	46.257
846	L 4 F	0.177	5.923	1.622	0.447	11.598	1.763	58.979	0.023	0.575	5.065	0.390	12.065	1.694	56.697
847	L 4 F	0.178	7.207	1.646	0.519	11.110	1.850	74.914	0.026	0.565	5.603	0.440	11.717	1.754	71.036
848	L 4 F	0.179	8.203	1.728	0.569	10.820	1.910	87.549	0.027	0.536	6.774	0.487	11.414	1.811	82.993
849	ABORTED														
850	ABORTED														
851	ABORTED														
852	ABORTED														
853	L 4 F	0.177	2.566	1.040	0.268	13.222	1.546	22.410	0.020	1.036	3.385	0.240	13.513	1.513	21.928
854	L 4 F	0.248	2.109	1.251	0.263	18.602	1.540	13.092	0.038	1.674	1.798	0.241	18.927	1.513	12.868
855	L 4 F	0.249	3.962	1.768	0.344	17.559	1.638	26.058	0.060	1.506	0.714	0.336	17.663	1.628	25.905
856	L 4 F	0.249	5.384	1.994	0.409	16.752	1.717	37.114	0.068	1.302	0.751	0.400	16.860	1.706	36.876
857	L 4 F	0.248	5.544	1.979	0.418	16.576	1.728	38.629	0.066	1.236	1.163	0.404	16.741	1.711	38.247
858	L 4 F	0.249	6.719	2.116	0.494	15.803	1.820	49.100	0.055	0.890	2.441	0.464	16.120	1.784	48.133
859	L 4 F	0.248	7.156	2.166	0.543	15.245	1.879	54.212	0.047	0.751	3.043	0.512	15.554	1.841	53.136
860	L 4 F	0.249	8.854	2.086	0.623	14.558	1.975	70.242	0.028	0.379	4.582	0.581	14.941	1.925	68.442
861	L 4 F	0.248	8.998	1.822	0.642	14.328	1.999	72.529	0.030	0.401	4.869	0.593	14.769	1.939	70.364
862	ABORTED														
863	L 4 F	0.247	3.770	1.733	0.346	17.381	1.641	25.051	0.044	1.153	1.353	0.331	17.577	1.623	24.773
864	L 4 F	0.249	4.570	1.917	0.389	16.996	1.692	31.053	0.062	1.388	1.153	0.376	17.153	1.676	30.768
865	L 4 F	0.250	2.609	1.464	0.286	18.409	1.568	16.370	0.044	1.579	1.451	0.269	18.663	1.547	16.147
866	L 4 F	0.249	2.004	1.276	0.240	19.012	1.513	12.176	0.049	2.243	0.924	0.229	19.184	1.499	12.067
867	L 4 F	0.249	1.589	1.132	0.214	19.412	1.481	9.453	0.038	2.148	1.334	0.198	19.672	1.462	9.328
868	L 4 F	0.124	1.957	0.730	0.187	9.889	1.448	22.857	0.015	1.341	2.424	0.160	10.117	1.416	22.341
869	L 4 F	0.123	2.482	0.827	0.224	9.515	1.493	30.125	0.015	1.087	2.278	0.199	9.714	1.462	29.506
870	L 4 F	0.124	3.463	1.035	0.288	9.120	1.570	43.850	0.030	1.628	3.536	0.245	9.432	1.518	42.401
871	L 4 F	0.123	2.854	0.921	0.246	9.347	1.520	35.260	0.030	1.952	3.099	0.212	9.611	1.478	34.292
872	L 4 F	0.123	4.173	1.108	0.345	8.664	1.640	55.624	0.021	1.023	3.471	0.303	8.942	1.589	53.896
873	L 4 F	0.124	3.991	1.075	0.326	8.858	1.617	52.031	0.025	1.232	3.849	0.283	9.151	1.565	50.363
874	L 4 F	0.123	3.651	1.079	0.314	8.867	1.602	47.555	0.019	0.990	3.611	0.274	9.145	1.553	46.111
875	L 4 F	0.124	3.426	1.006	0.287	9.124	1.570	43.364	0.029	1.582	3.117	0.253	9.374	1.528	42.205
876	L 4 F	0.123	4.986	1.209	0.379	8.450	1.681	68.144	0.031	1.273	4.412	0.326	8.789	1.616	65.519
877	L 4 F	0.123	5.314	1.211	0.419	8.215	1.729	74.699	0.013	0.502	5.384	0.359	8.574	1.657	71.566
878	L 4 F	0.123	5.790	1.191	0.404	8.305	1.710	80.520	0.031	1.127	4.501	0.354	8.610	1.650	77.667
879	L 4 F	0.123	6.406	1.192	0.451	8.038	1.767	92.039	0.009	0.314	5.541	0.428	8.165	1.740	90.597
880	L 4 F	0.121	9.203	1.215	0.528	7.512	1.860	141.478	0.044	1.104	5.995	0.455	7.885	1.772	134.786
881	L 4 F	0.124	8.320	1.154	0.500	7.840	1.827	122.558	0.023	0.624	5.582	0.427	8.240	1.738	116.608
882	ABORTED														
883	L 4 F	0.122	7.333	1.285	0.508	7.672	1.836	110.381	0.008	0.249	7.012	0.416	8.168	1.725	103.685
884	L 4 F	0.122	7.052	1.204	0.481	7.810	1.804	104.282	0.010	0.314	6.798	0.412	8.188	1.721	99.466
885	L 4 F	0.123	5.658	1.201	0.438	8.108	1.752	80.592	0.007	0.263	6.542	0.359	8.578	1.656	76.175
886	ABORTED														
887	L 4 F	0.090	3.283	0.813	0.294	6.586	1.578	57.572	0.011	0.906	4.727	0.256	6.786	1.532	55.876
888	L 4 F	0.088	4.051	0.925	0.360	6.130	1.658	76.321	-0.021	1.453	5.297	0.296	6.431	1.580	72.742
889	L 4 F	0.089	5.211	0.994	0.401	6.021	1.707	99.959	0.011	0.589	6.281	0.331	6.335	1.623	95.006
890	L 4 F	0.088	5.588	1.019	0.433	5.821	1.746	110.870	0.011	0.596	6.504	0.380	6.042	1.682	106.807
891	L 4 F	0.085	6.245	1.048	0.493	5.399	1.818	133.577	0.011	0.564	6.673	0.465	5.500	1.785	131.127
892	L 4 F	0.089	6.238	0.987	0.465	5.761	1.784	125.039	0.009	0.420	7.223	0.406	6.000	1.713	120.060
893	L 4 F	0.090	7.345	0.954	0.509	5.655	1.838	150.001	0.010	0.413	7.408	0.449	5.888	1.765	144.054
894	L 4 F	0.090	7.360	0.052	0.500	5.689	1.827	149.390	0.006	0.229	7.312	0.470	5.806	1.790	146.402
895	L 4 F	0.219	3.039	1.433	0.311	15.830	1.598	22.176	0.026	0.944	2.336	0.287	16.110	1.569	21.778
896	L 4 F	0.209	1.798	1.044	0.235	16.020	1.507	12.961	0.024	1.454	2.603	0.204	16.440	1.468	12.632
897	L 4 F	0.206	4.919	1.704	0.404	13.907	1.711	40.851	0.032	0.799	2.771	0.365	14.303	1.663	39.720
898	L 4 F	0.206	5.966	1.751	0.456	13.417	1.773	51.354	0.027	0.592	3.295	0.419	13.762	1.729	50.068
899	L 4 F	0.205	8.534	1.905	0.589	12.236	1.935	80.543	0.034	0.568	5.248	0.526	12.743	1.858	77.337
900	L 4 F	0.205	7.045	1.908	0.519	12.795	1.850	63.588	0.021	0.404	4.736	0.467	13.252	1.787	61.396
901	L 4 F	0.205	4.329	1.614	0.403	13.852	1.709	36.087	0.020	0.578	4.328	0.372	14.162	1.672	35.299
902	L 4 F	0.150	1.475	0.731	0.189	11.942	1.451	14.263	0.014	1.347	2.819	0.155	12.292	1.409	13.856

APPENDIX K

ANALYSIS OF TEST DATA

K.1 Primary data processing

The first step in analysing data from conveying runs, which has been referred to in chapter 2 as "primary data processing", took place as soon as the raw data had been transferred from the data acquisition unit on to the floppy disc, following the end of the run. Processing this in the way previously described resulted in a file containing values of the measured and calculated flow quantities from consecutive runs. Thus a large data base of this information was gradually built up as the test programme progressed.

The way in which the primary data processing was carried out is decribed broadly in section 12 of Chapter 2, whilst a more elaborate description of the software used can be found in section 2.2 of Appendix G.

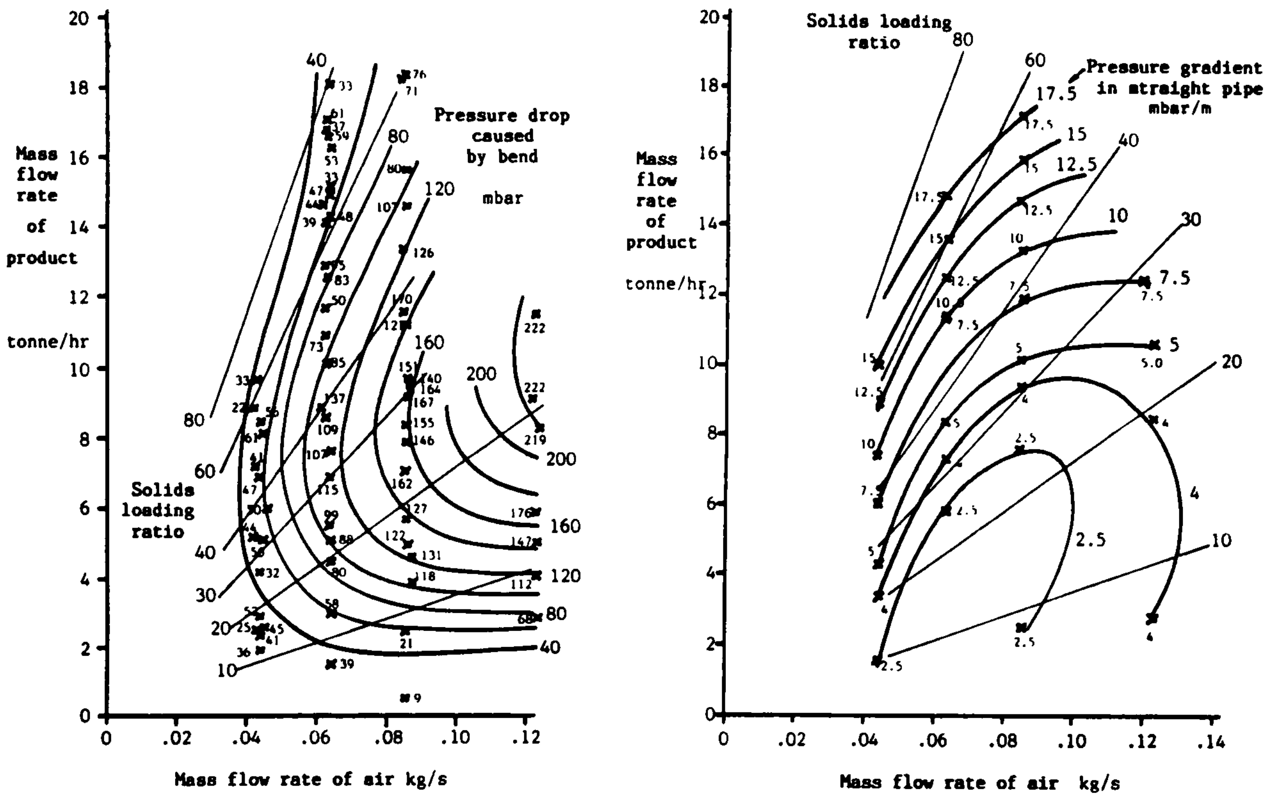
K.2 Secondary data analysis

The aim of this Appendix is to describe the next stage of analysis, comprising the search for suitable correlations for use as part of a system for storing the data in a compact way, convenient for subsequent recall and use.

K.2.1 First attempts

Some attempts were made to look in detail at the data as soon as the first bend had been tested over a wide range of conveying conditions. The first stage of this involved the use of graphs of the same type as the pipeline conveying characteristic graphs described in Appendix B which describes the testing-and-scaling approach to pipeline design, for which these graphs are normally used and with which the author was familiar at this stage. Normally, they are used to represent the conveying performance of the material to which they relate, in a total pipeline system. However, it seemed an obvious starting point, to try to use the same type of graph to present bend losses and straight pipe gradients.

Accordingly, these 'maps' of mass flow rate of product versus mass flow rate of air were drawn, with contours of pressure drop caused by the bend and steady pressure gradient in the straight pipe (measured well downstream of the bend, as described in section 11 of Chapter 2). These related to the wheat flour in the 2in. n.b. pipe, and the short radius bend without unions. The lines of constant pressure drop/pressure gradient were obtained by visually interpolating between the values of these quantities marked adjacent to the points. The results are shown in fig. K-1 below. Superimposed are lines of constant solids loading ratio, which are the straight lines through the origin.



Pressure drop caused by bend

Pressure gradient in straight pipe

Fig. K-1
Data on pressure losses presented in "conveying characteristic" form.
These relate to the wheat flour in 2in. n.b. pipe,
and the short radius bend without unions.

The most striking thing about these graphs was how different in shape the lines of constant pressure were. This clearly explained why the search for reliable and consistent bend equivalent length values, which had been the subject of work by many authors in the past, had met with only limited success. In order to examine this in a little more detail, a similar graph was drawn showing the equivalent length values, obtained by dividing the bend pressure loss by the straight pipe pressure gradient at each point on the graphs. The result is shown below.

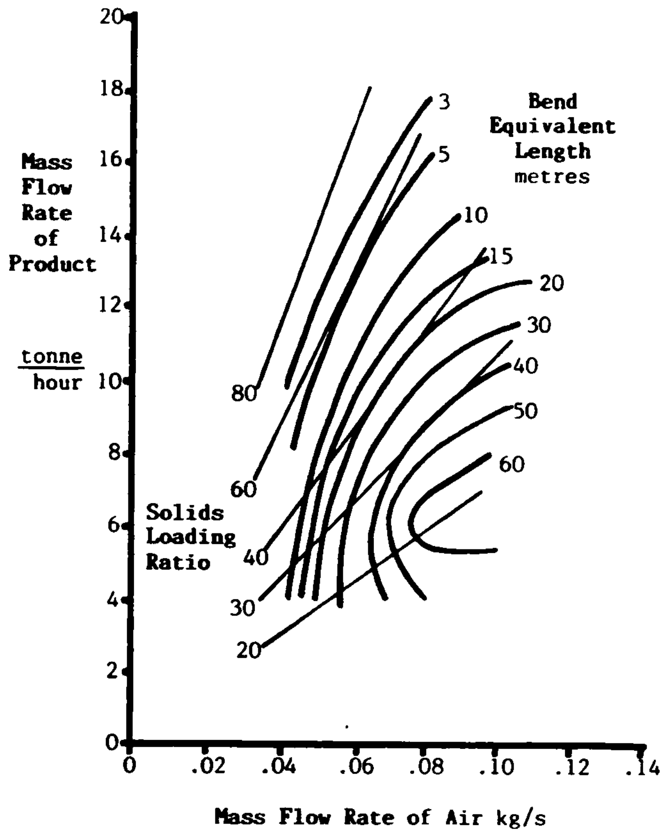


Fig. K-2

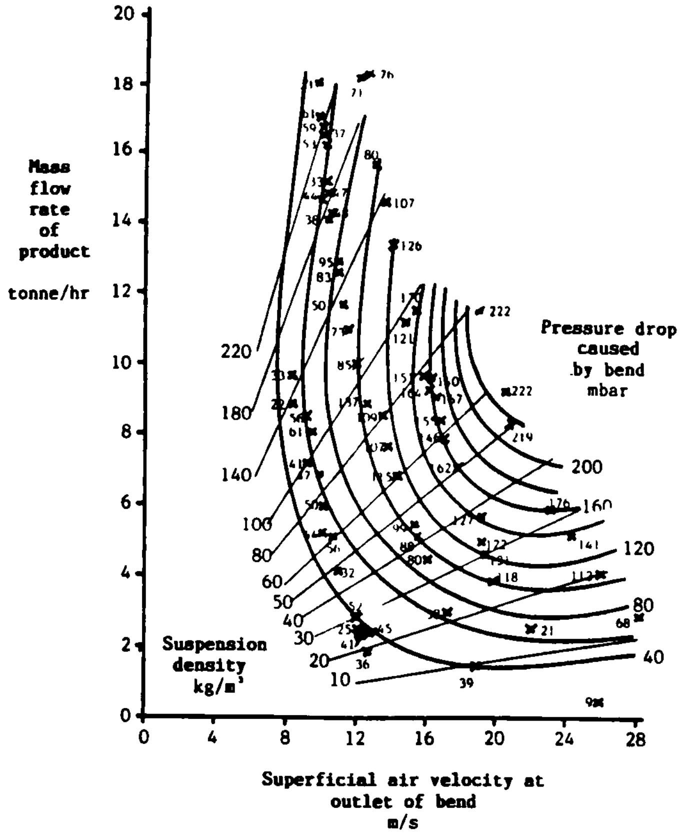
Bend equivalent lengths calculated from the data presented above

This showed equivalent length values between 3 and 60 metres for just a single product, pipe size and bend type, depending on conveying conditions. The conclusion drawn fron this was that the "equivalent length" approach to dealing with bends was even more troublesome than at first suspected and proving beyond all doubt that a better method was needed.

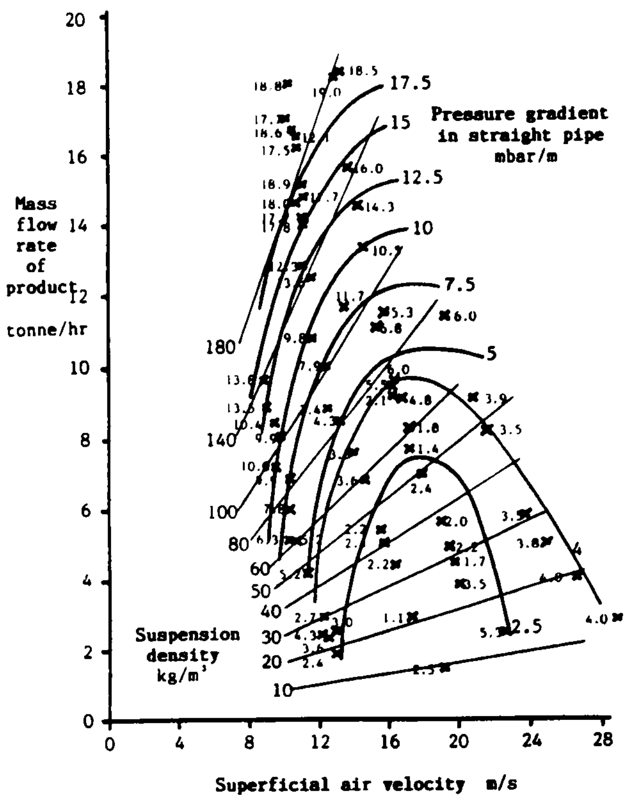
K.2.2 Improving the graphical presentation of test data

Consideration showed that the type of diagram illustrated in fig. K-1 above gave no indication of air velocity, meaning that any data taken off such a graph would be dependent on a specific air density value because a given mass flow rate of air would correspond to a range of different air velocities depending on air density and hence pressure in the line. For example, if predicting pressure drop in a pipeline by working along the line taking pressure drop values from these graphs, the user would obtain the same pressure drop and gradient values for every bend and straight in the line irrespective of the effect of air expansion, since mass flow rates of air and product are constant along the line. This was considered most unsatisfactory since Mills' work (ref. 1, described in section 2.2 of Chapter 2) had shown the pressure losses to be strongly dependent on velocity.

Therefore it was decided to re-draw the graphs using air velocity ("superficial air velocity", calculated from volume flow rate of air divided by pipe area, ignoring volume of product) in place of mass flow rate of air. The result was two graphs of very similar shape to those above, as shown overleaf. For the bend, air velocity at exit (calculated on the "equivalent step change" model of bend pressure loss, illustrated in section 2.11 of Chapter 2) was used since it was expected that such data would in most cases be used in working along a pipeline backwards from the outlet (atmospheric pressure reference) towards the inlet to predict line inlet pressure.



Pressure drop caused by bend



Pressure gradient in straight pipe

Fig.K-3

Data on pressure losses presented relative to mass flow rates of product and superficial air velocities. Again for the wheat flour in the 2in. n.b. pipe, and the long radius bend without unions.

This presentation of the data was considered to be better, since it depended on the velocity instead of the mass flow rate of the air, thus taking account of the expansion of the air along the line. However, it was evident that lines of constant solids loading ratio could not be plotted on diagrams drawn in this way, since any point would correspond to a range of s.l.r.'s depending on air density. This was considered a disadvantage since s.l.r. is frequently used to describe the conditions in a pipeline.

It was noted that the diagram showing pressure gradient in the straight pipe indicated a less steep gradient with increasing velocity, at least up to about 18 m/s (for constant mass flow rate of product). This was the reverse of what had been expected, but checking the measured values showed that this was undoubtedly so. No explanation for this was obvious, so it was simply accepted; subsequent data reinforced this observation.

K.2.3 The development of the quantity "Suspension Density"

The diagram of bend pressure drop displayed two distinct regions of dependence on the other two variables. At the bottom right, the lines were nearly horizontal, indicating that the pressure loss was dependent only on mass flow rate of product and practically independent of air velocity; whereas towards the top the lines were instead approximately vertical, indicating the converse. In between, the lines turned a corner. It was apparent that drawing a line diagonally through the origin split the two areas. The significance of this line was pondered upon, and it was noted that its slope was 0.57 tonne/hour per m/s, or 0.159 kg/s of product per m/s air velocity. It was apparent from the units that this quantity was a sort of "flow concentration" of some kind, and dividing by the area of the pipeline ($0.00221m^2$ for the 53mm bore pipe) gave a flow per unit area of 72 $(kg/s)/m^2$ per m/s. Rationalising the units of this gave 72 kg/m^3.

It was fairly clear from this that this quantity, with units of kg/m^3, would be the density of the suspension of solids in the pipeline if there was no slip velocity between the product and the conveying air. It was also clear that this quantity could be calculated using the continuity equation,

$$\dot{m} = \rho_s . a . c \qquad \text{thus} \qquad \rho_s = \frac{\dot{m}}{a.c}$$

taking $\dot{m}$ as the mass flow rate of solids, a as the cross sectional area of the pipe and c as the superficial air velocity, and giving ρ_s as the density of the suspension as described above. This density quantity could therefore be thought of as the mass of solids conveyed per unit volume of air flowing, or a representation of the extent to which the pipeline is filled with product if the slip velocity were to be ignored.

It was apparent that this 'suspension density' might be a better alternative to the conventional mass solids loading ratio as a basis on which to characterise a flow, because it seemed logical to think that the controlling influences on the regime of flow of solids in a pipe should be air velocity and the extent to which the pipe is filled, rather than a simple mass ratio which would give varying degrees of pipe filling depending on air density. It was noted that the suspension density would decrease along a pipeline, because of air expansion, which would reflect the true increase in distance between particles as the air expands. It was also noted that it would not truly be the actual density of solids in the pipe, because of the slip between solids and air; however it was thought that the difference would be consistent with flow conditions, and thus of no consequence when searching for correlations - but the simplicity of it, in that it is easily calculated from measurable quantities, would be an advantage.

K.2.4 First steps in the search for correlations

It was apparent that the two graphs showing bend and straight pipe pressure loss for ranges of air velocities and mass flow rates of solids could be used directly for prediction of pressure drop along any pipeline of the same diameter and conveying the same product as used in the tests. This would be done by choosing values for flow rates of air and product then working backwards along the pipeline (for a positive-pressure system, where pressure at the outlet is fixed at atmospheric), calculating air velocity and reading off pressure gradient in the last straight, then re-calculating the air velocity to obtain pressure drop caused by the bend before this, re-calculating again and repeating the procedure all the way back along the line to the inlet to obtain the pressure here. Such a repetitive procedure would be time-consuming if done by hand on a calculator, but would be suited to the use of a computer. However, this would not be a convenient way of working because of the need to repeatedly stop the program and ask the operator to read a value from the graph for each bend and straight. Also it would give no indication of the effect of using different pipe diameters. Therefore it was decided that an attempt should be made to find a more compact way of storing the data, which would be easily accessible to a computer program and would allow for the use of different pipe diameters.

Efforts were therefore directed towards finding a suitable system of equations which could be fitted to the data. This would be done by plotting the data on various graphs and looking for correlations which might suggest suitable equations.

K.2.4.1 Bends

The bend was examined first, using the data from the short radius bend without unions (runs nos. 1 to 68). Drawing a graph of pressure drop versus superficial air velocity at bend outlet for a variety of suspension densities suggested the use of a power law relationship between pressure drop and air velocity for each suspension density. Drawing a logarithmic graph of pressure drop versus velocity for each density established a range of powers between 1.8 and 2.4, which further suggested that a square law relationship might give a sufficiently good representation. The graph of pressure drop versus air velocity for various densities, with a family of square law curves superimposed, is reproduced overleaf along with a plot of the constant terms necessary against suspension density.

It was observed that data stored in this way, i.e. a square law equation plus a graph of two straight lines, each easily represented by a simple equation, would be very easy to program into a computer. Also it was noted that it is not dependent on pipe diameter, i.e. for any given suspension density pressure drop would be constant, with flow rates of air and solids proportional to pipe cross-sectional area; this was known to be in accordance with observations of overall pipeline pressure drop made by Mills (ref. 1) for pipelines of identical layout but different diameter. The effect of pipeline diameter would be examined in more detail later.

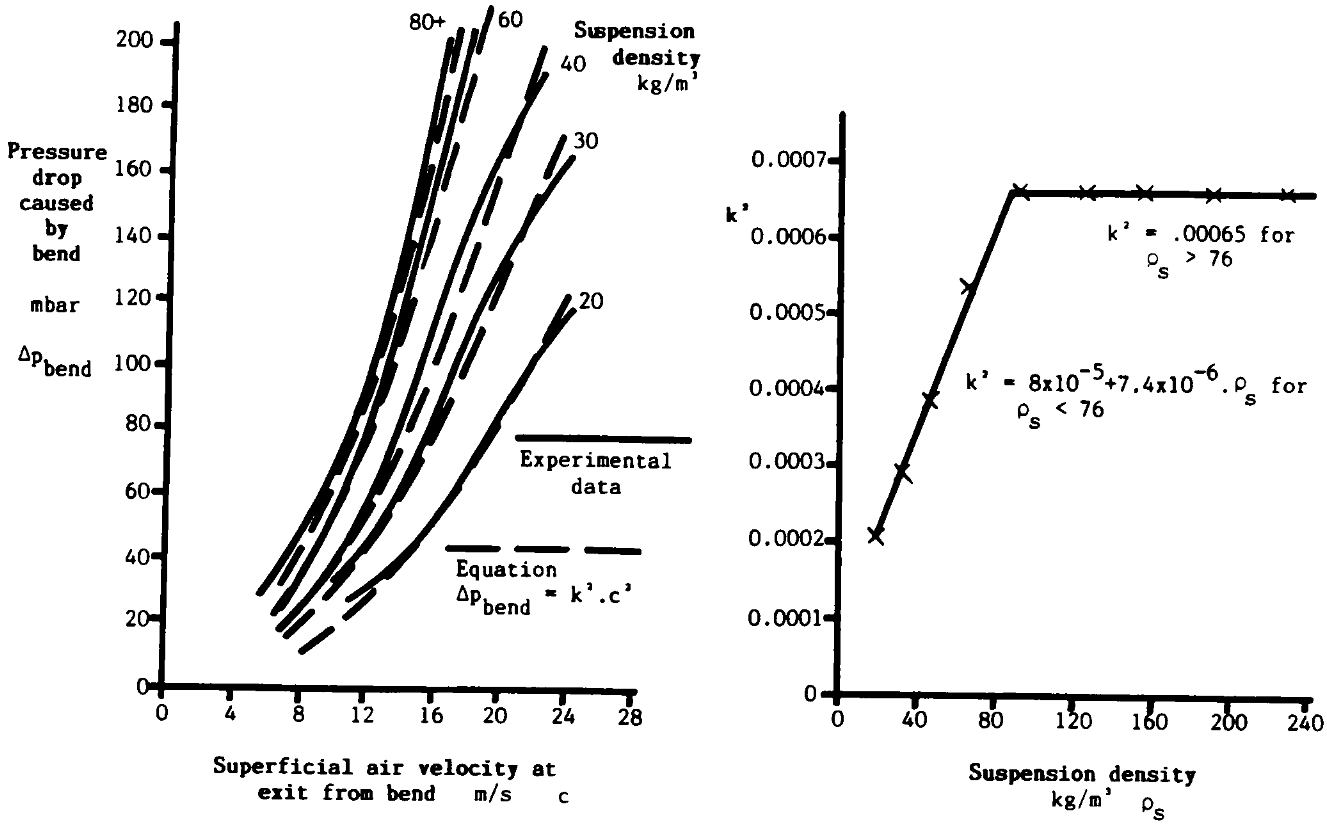


Fig. K-4

Graph of bend pressure drop versus air velocity for various suspension densities, with a family of square laws superimposed, and the correlation between the constant term in the square law and the suspension density.

K.2.4.2 Straight pipes

With some progress made towards developing correlations for bends, it was clearly appropriate to spend some time examining the data on pressure gradients in straight pipes. It was decided to try to base this work on superficial air velocity and suspension density as for the bends, and again the data from runs nos. 1 to 68 would be used.

Several graphs involving the relevant quantities were drawn and each rejected as of no use; at some length a graph of pressure gradient versus suspension density for various velocities was drawn, derived from the graph shown on the right of fig. K-3. This looked promising in that it showed all the data falling close to a single line, suggesting that a single empirical expression might be used to describe it. A power law curve through the origin was fitted, resulting in a power of 1.49. The use of non-integer powers clearly being undesirable, it was wondered whether a square-law with an offset from the origin might not be fitted; the result is shown below:-

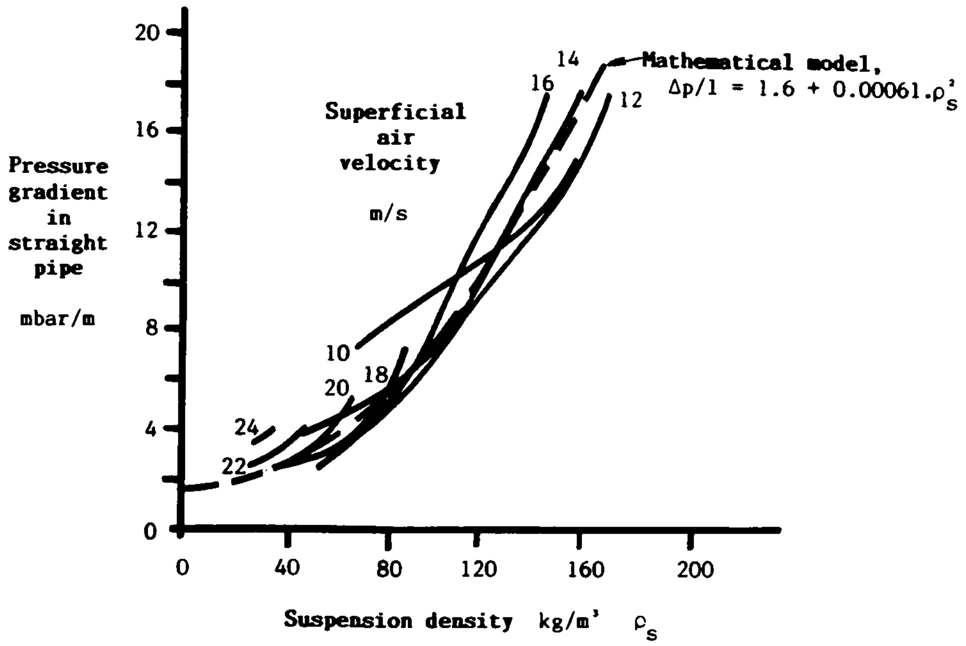


Fig. K-5

Graph of pressure gradient in straight pipe vs. suspension density, for a range of velocities, with a square-law with an offset fitted.

At this stage a calculation was done to estimate the pressure gradient to be expected in a straight pipe with air only flowing; a velocity of 20 m/s was taken as typical, and the Darcy equation yielded a value of 1.5 mbar/m. It was noticed that this was very similar to the offset from the origin of the square law fitted to the data in the graph above, suggesting that it should be re-plotted with the "air only" pressure gradient for each velocity subtracted. The result is shown below, with a square law through the origin fitted which seems a remarkably good fit.

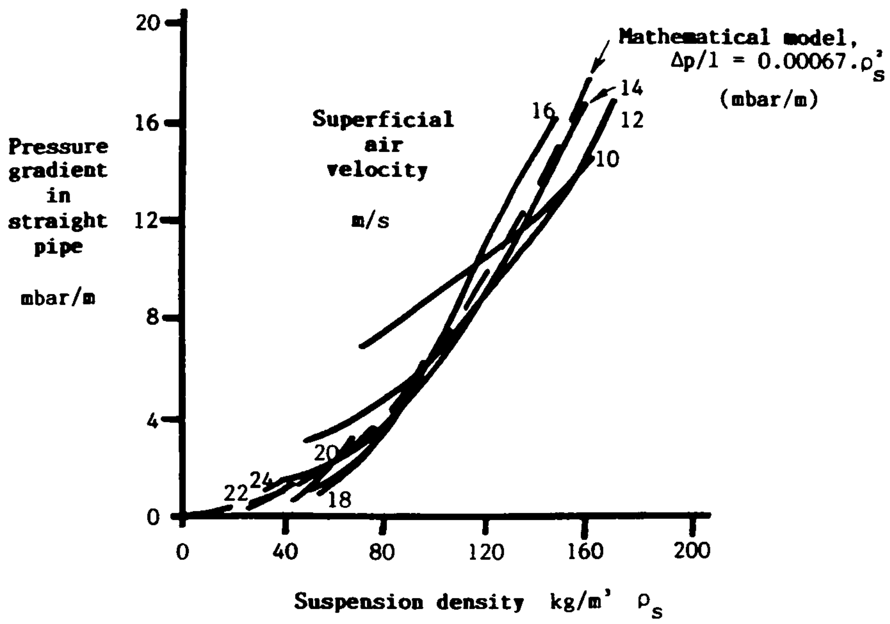


Fig. K-6

Graph fig. K-5 redrawn with the "air-only" pressure gradients subtracted.

It was not understood why the subtraction of the "air-only" pressure gradient should make the remaining part easier to model, nor was it understood why the gradient should be much more dependent on suspension density than velocity; however it was felt that important progress had been made.

K.2.5 Testing the correlation established

A little time was spent analysing what had been achieved, and it seemed that an appropriate time had arrived to try and use the correlations which had been found; the choice was whether to use them to try to predict the bend and straight pipe pressure loss characteristics shown in fig. K-3, or to go straight to attempting to predict the pressure loss characteristics of the overall pipeline used for the test work. It was decided to move in small steps, the first one therefore being to predict the individual bend and straight pipe data.

The empirical expressions described above were easily programmed into a computer, and the results of using them to try to predict the measured pressure loss data for bend and straight pipe is shown below:-

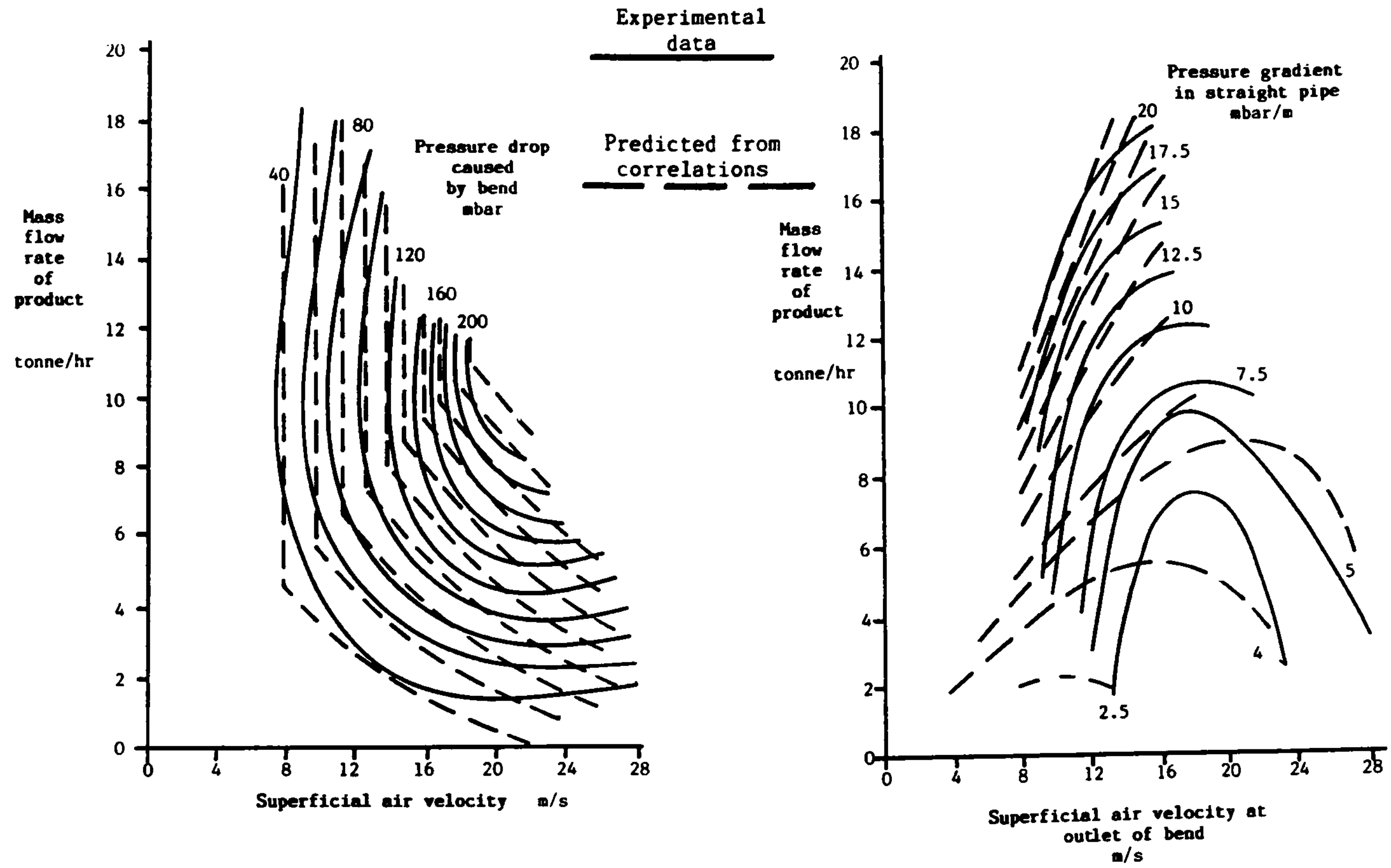


Fig. K-7

Comparison between measured pressure loss data for bend and straight pipe, and that predicted using the correlations

It will be seen from this that the predicted bend pressure drop characteristic was quite close over much of the range, at least as close as could be expected with the variability usually experienced in taking measurements from pneumatic conveying systems. However, it somewhat over-predicted the pressure loss at the higher velocities, particularly where suspension densities were low. For the straight pipes, the predicted gradients were somewhat higher than the measured ones for suspension densities above about 160 kg/m^3, and rather low at low suspension densities and velocities. However, the overall agreement was most encouraging so it was decided to attempt to predict the pressure drop characteristics of the overall pipeline.

The layout of the pipeline is shown in fig. F-1 of Appendix F. A computer program was written which used the equations developed to represent the test data. This simply took the pipeline layout and mass flow rates of air and product from the user, then worked along the pipeline calculating flow conditions and using the correlations to predict pressure loss for each bend and straight in turn, finishing up with the inlet pressure; it was the first step of the work which ultimately led to the program "SYSTEM" listed in Appendix G.

The layout of the loop was fed into the program, with the vertical riser at the end of the line being treated as a horizontal section of double the length, following the correlation found by Marjanovic (ref. 53, mentioned in Appendix B). Conveying characteristics of the line were predicted by repeatedly running the program with different values of flow rates of air and product. The true conveying characteristics of the line were plotted from data taken during the test runs, and the two compared. The result, shown overleaf, was extremely encouraging:-

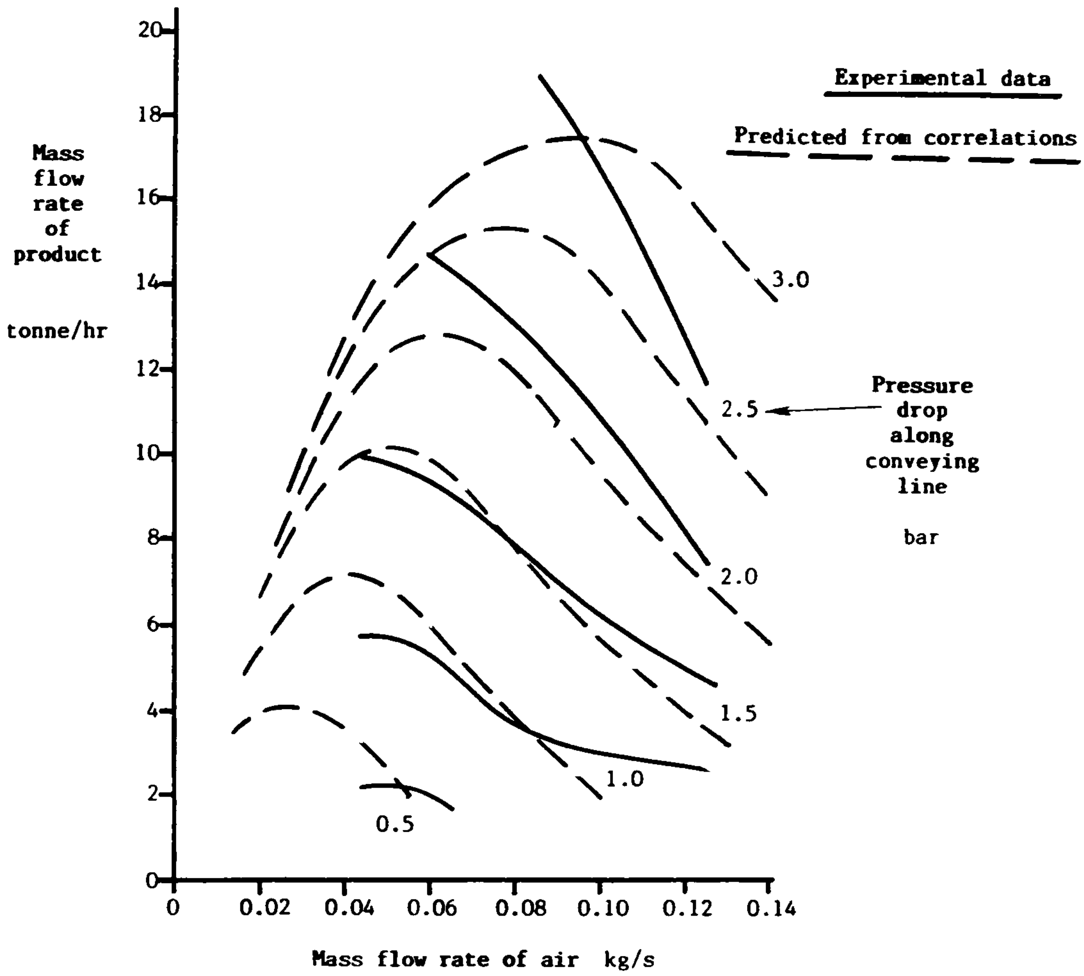


Fig. K-8

Comparison between true characteristics of the conveying line in which the test sections were located, and those predicted from the correlations.

At this stage, the work done so far was written up for presentation as a conference paper. This gave time to consider the way to proceed.

K.3 Attempts at improving the first correlations

K.3.1 Bends

By far the greater amount of work was directed at bends, because it was in this aspect of prediction of pressure losses that least progress seemed to have been made by previous workers and the need for improvement was particularly evident.

The first attempt at improving these was directed at trying to obtain a better model to represent the bend loss data shown in fig. K-4. The effect of taking account of an "air only" pressure loss at the bend, as found useful for the straight pipe, was considered but some calculations showed this to be insignificant compared with the total pressure loss caused by the bend (typically a fraction of a millibar as compared with several tens or hundreds of millibars). The use of a line through the origin for the straight section up to a suspension density of 68 kg/m^3 was tried, especially because the pressure drop would be expected to be very near to zero with no product flowing (zero suspension density, i.e. air only, for which data is commonly available), but the effect on the conveying characteristics was not very great.

It was apparent that the values of the factor, k^2, in the expression

$$\text{Bend pressure drop } \Delta p = k^2.c^2 \qquad \text{(section K.2.4.1)}$$

where c = superficial air velocity at outlet of bend

could be determined directly from the data for each test run, the other two quantities being available from the results. This would obviate the need to go through the drawing of logarithmic graphs. The values of k^2 for each test run with the first set of data, runs 1-68, were determined and a graph plotted as shown overleaf:-

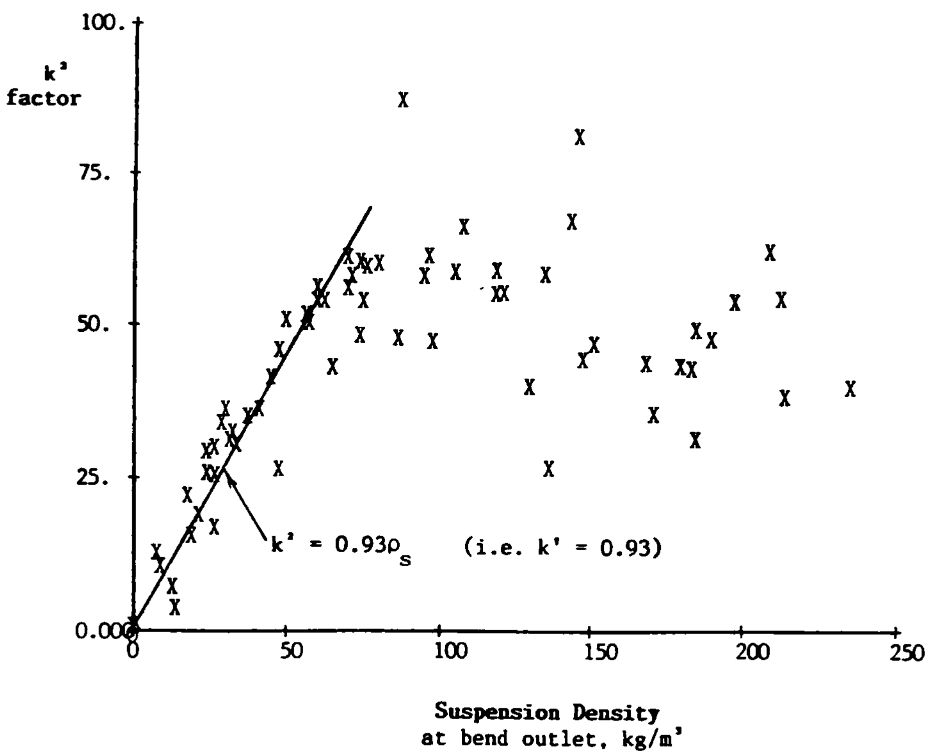


Fig. K-9

Graph of k² factor in $\Delta p = k^2.c^2$ expression,
plotted versus suspension density for runs 1-68

There was a clear correlation up to a suspension density of about 80kg/m³, but above this no correlation was apparent.

By this time, data had already been taken for a number of other bends apart from the one on which all the foregoing work was based; an external influence now came to bear, in the shape of a colleague asking for some compatative data between pressure drop caused by blind tees and that caused by radiused bends. The data on the blind tees had been taken, so the exercise resulting in graph K-9 above for the short radius bend was repeated on this data, to obtain the graph overleaf:-

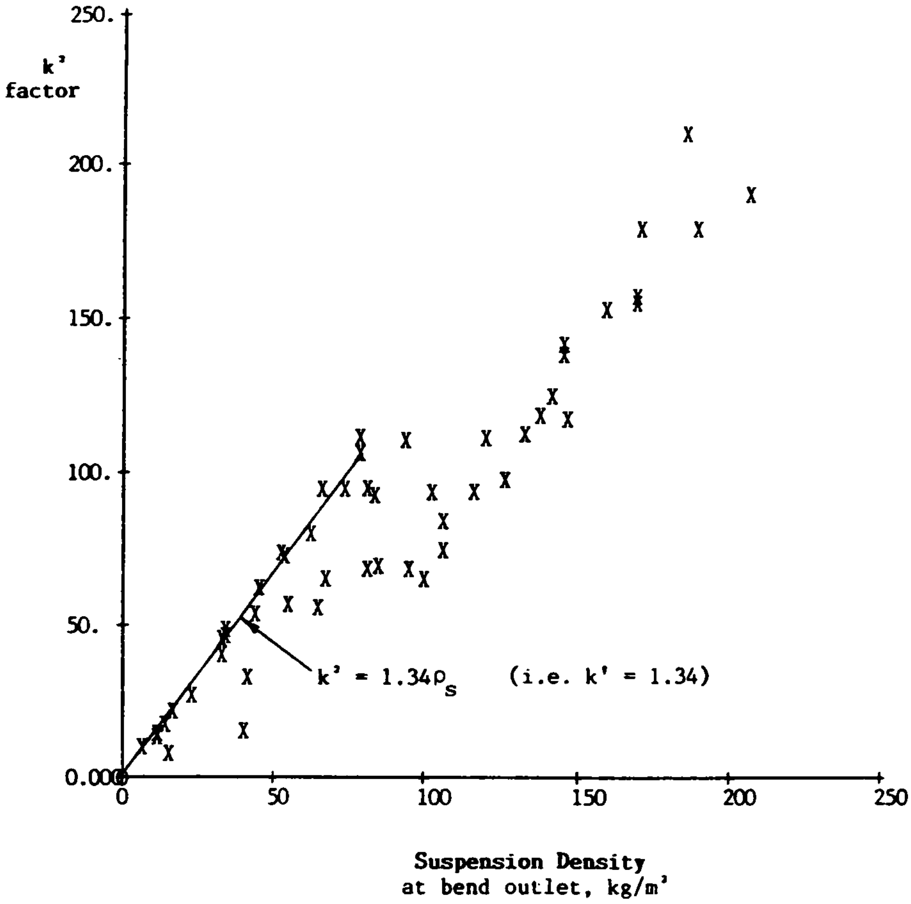


Fig. K-10

Graph of k^2 factor versus suspension density for the blind tee; compare with K-9 for the short radius bend, above

This clearly suggested the use of a straight line through the origin to represent the data up to a suspension density of about 80, the line $k^2 = 1.34$ being drawn as shown above. The same line-fitting exercise was used for the short radius bend (fig. K-9 above) with a line of $k^2 = 0.93$ being found appropriate up to a suspension density of again 80. Next the data for the long radius bend (without unions) was analysed in the same way, and a line of $k^2 = 0.71$ being found appropriate (fig. K-11 overleaf).

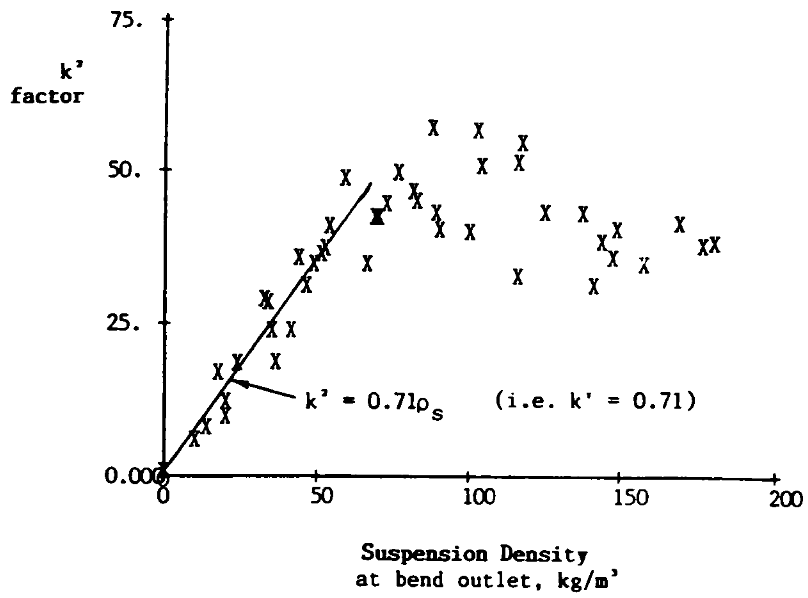


Fig. K-11

Graph of k^2 factor versus suspension density for the long radius bend without unions; compare with K-9 and K-10 above.

It was apparent that a very useful correlation had been established here, at least for a limited range of the data. This was effectively to introduce another factor, k', in an expression $k^2 = k'.\rho_s$,to represent values of the k^2 factor from the expression $\Delta p = k^2.c.\rho_s$

Substituting the former expression into the latter yielded

$$\Delta p = k'.\rho_s.c^2$$

with values of k' of 0.71 for the long radius bend, 0.93 for the short radius bend (both without unions) and 1.34 for the blind tee. It should be noted that this correlation only held up to a suspension density of 80, about half the total range of conveying conditions achieved. However, some clear progress had been made.

Some time was spent pondering this outcome, and looking at different ways to write the equation (e.g. $\Delta p = k'.\text{mass flux}.c^2$) but the way in which it was written above seemed clearest. It became apparent after a while that there was a certain similarity to several expressions used for single phase flow, where a dynamic pressure of an air stream is calculated using the term $\frac{1}{2}.\rho.c^2$; in fact after some thought, it was apparent that if the usual expression for bend head losses normally used for the flow of liquids ($\Delta h = k.c^2/2g$) was rearranged to give a pressure drop, this gave $\Delta p = k.\frac{1}{2}\rho c^2$ which was remarkably similar to the expression found through the empirical correlation process described above.

It was but a small step now to decide to unify the correlations which had been established for bend pressure loss, with the conventional practice for liquid flow, by writing the expression as

$$\Delta p = 2k'.\tfrac{1}{2}\rho_s c^2$$

then replacing the factor 2k' with a symbol K to give

$$\Delta p = K.\tfrac{1}{2}\rho_s c^2.$$

The advantage of doing this was obvious, in that many engineers used to handling flow of fluids would already be familiar with the practice of using a dynamic pressure. It was fortuitous that the expression, which had been established initially by purely empirical work, should have come out to be dimensionally homogeneous.

The data from more bends was by this time available, and this was plotted in the form shown in figs. K-9, 10 and 11 above; the patterns were broadly similar and nothing particular emerged from this exercise.

It was decided that the best way to proceed would be to analyse all the bend loss data so far taken (most of the 2in. bends) to obtain values for the loss factor K directly, and begin drawing some graphs to see how this varied with suspension density and air velocity, and from bend to bend.

It had already become apparent that there were two major controlling variables on the bend loss coefficient K, i.e. air velocity and suspension density. A graph of loss coefficient K versus suspension density was plotted for the 2in. short radius bend conveying flour; this showed a clear pattern. Information on the air velocity was added by coding the points in an attempt to identify any pattern in the scatter which was apparent. The resulting graph for this bend is shown in fig. K-12 below.

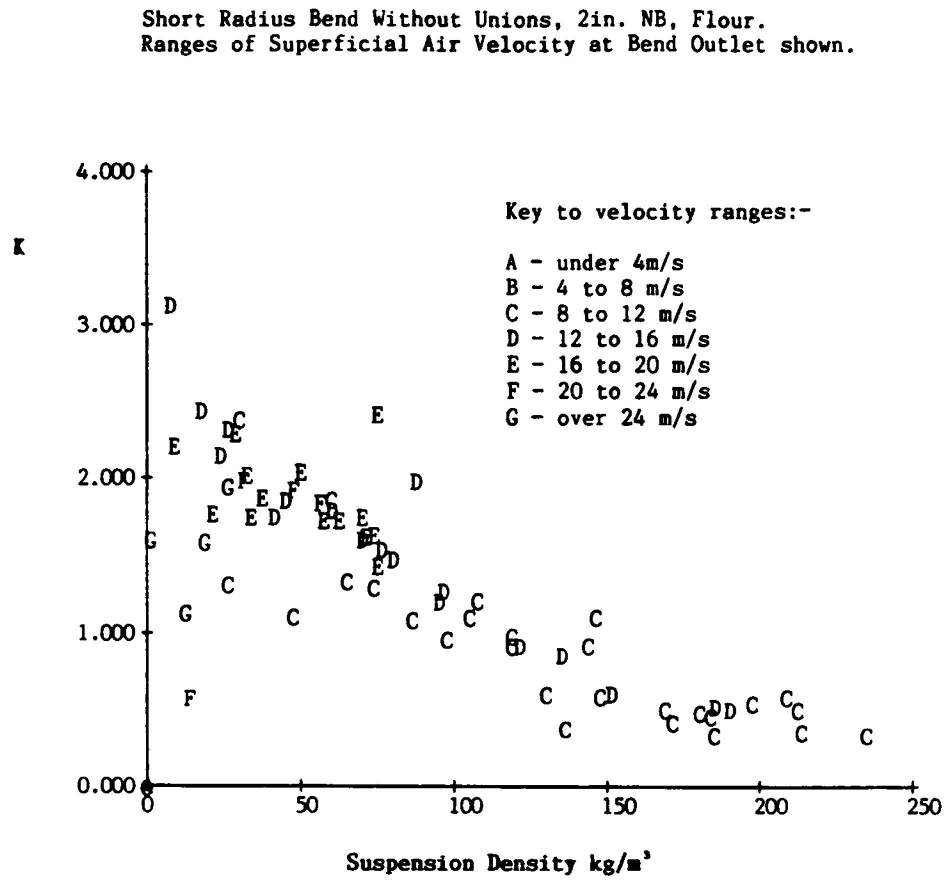
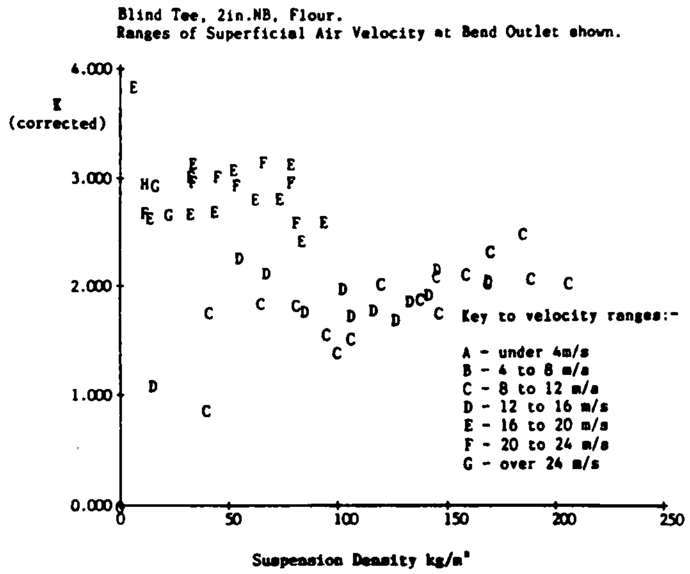


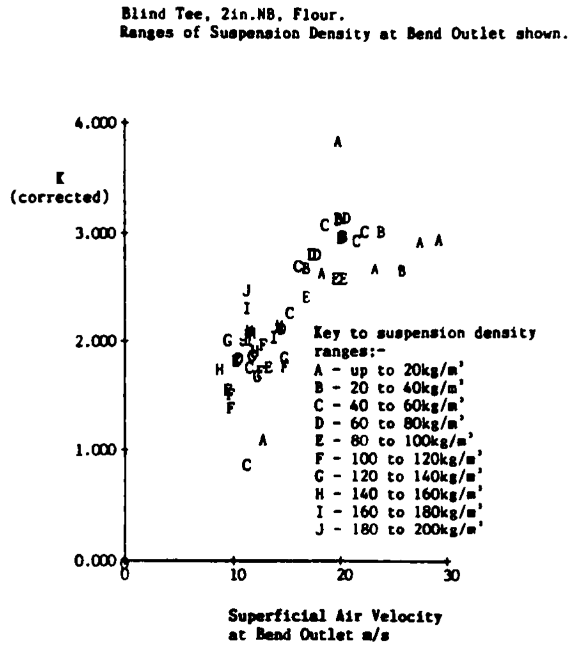
Fig. K-12
Graph of bend loss coefficient versus suspension density
for the short radius bend without unions

When similar graphs for the other radiused bends conveying flour were plotted, they all showed a very similar relationship, albeit scaled slightly in the 'y' direction.

When a similar graph was plotted for the blind tee and vortice-ell bends, the pattern was less clear; for example the graph for the blind tee is shown in (a) overleaf; when a graph of K versus velocity was drawn, however, a clearer pattern was apparent, (b) overleaf:-



(a)



(b)

Fig. K-13

Graphs of loss coefficient versus (a) suspension density and (b) air velocity for the blind tee bend

Graphs for the malleable elbows, both male and female, fell somewhere between those for the radiused bends and the blind tee bend.

K.3.2 Air density, product degradation, and standard curves

The next logical step seemed to be to examine the effect of air density on the loss coefficients, to eliminate this from further consideration if possible.

Four full sets of tests had been run on one bend, the short radius without unions;

(1) using the usual pipeline loop;

(2) using the same loop with an extra pipe loop added to the end to increase air pressure, and hence air density, at the test bend;

(3) using an expanded (3in. nominal bore) return downstream of the test section to reduce air pressure, and thus density, at the bend;

(4) using the same set-up as in (1) to assess any product degradation effect.

The layout of the pipelines are shown in detail in figs. F-1 and F-2, Appendix F.

Drawing the graphs of loss coefficient K versus suspension density for cases (1) to (3) gave results of very similar form, as shown in fig. K-14 below; air density values were coded onto the points rather than air velocity:-

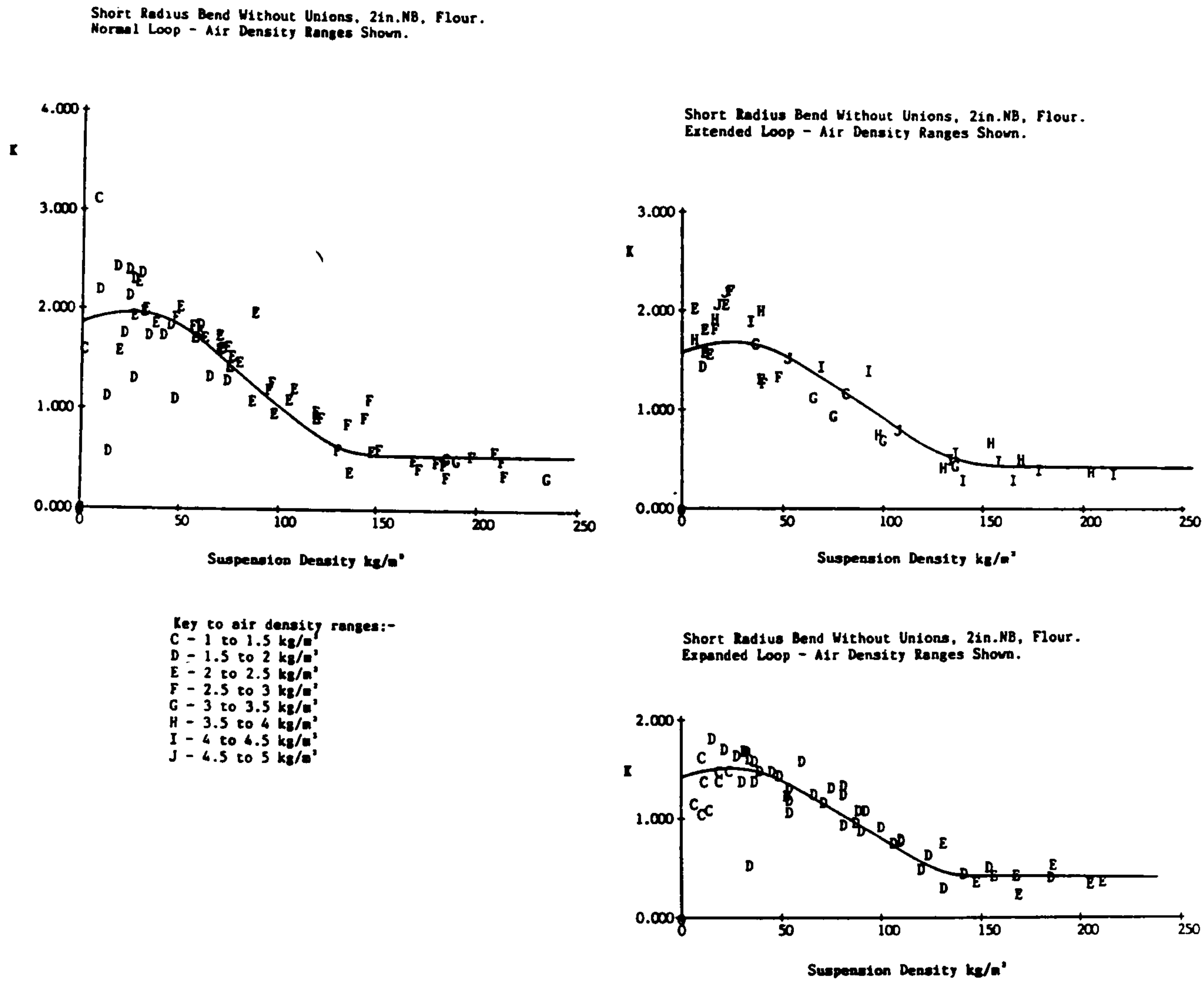


Fig. K-14

Graphs of loss coefficient versus suspension density used to identify air density effects; fitting of the curves is described below

There seemed to be some effect but it did not appear consistent; for both the lower and higher air density cases, the loss coefficients appeared generally lower than for the first set of tests.

It was rapidly becoming obvious that in order to allow comparison between the various sets of results which had been obtained, some sort of standard would need to be established so that the differences between the sets of data could be reduced to a single figure if possible. The strong similarity between the graphs suggested that it might be possible to find a standard curve which could be scaled to obtain the best fit to any of the sets. The possibility of describing such a curve using a continuous mathematical function seemed unlikely so it was decided to draw the best curve through the first set of data and scale it graphically to the other sets.

The reference set of data was taken as the first one, i.e. the one using the normal pipeline loop, and a curve drawn through it; the shape of the curve took qualitative account of all the other sets of data for radiused bends. The curve is shown in fig. K-14 above.

The standard curve thus established was scaled to obtain the best fit (by eye) to the data for increased and reduced air density, the necessary scaling factors being 0.82 and 0.75 respectively. Again the curves are shown in fig. K-14 above. To enable these factors to be related to the change in air density, a single value characterising the change in air density was required. This was found by reading off, from the graphs above for increased and decreased air density data, the value of air density at each suspension density in steps of 25, and dividing by corresponding values for the "normal loop" data, to obtain a factor; the mean of the range of factors than taken for each set. In this way it was established that the air density when using the extended loop was on average 1.58 times the value for corresponding conditions in the normal loop, the factor being 0.79 for the reduced air density in the loop with the expanded return. It was now possible to plot the mean change in loss coefficient versus mean change in air density. The resulting graph is shown in fig. K-15 below, with a fourth point added which was obtained by performing the same process on the data obtained using the normal loop (achieving the same air densities) after many more conveying runs.

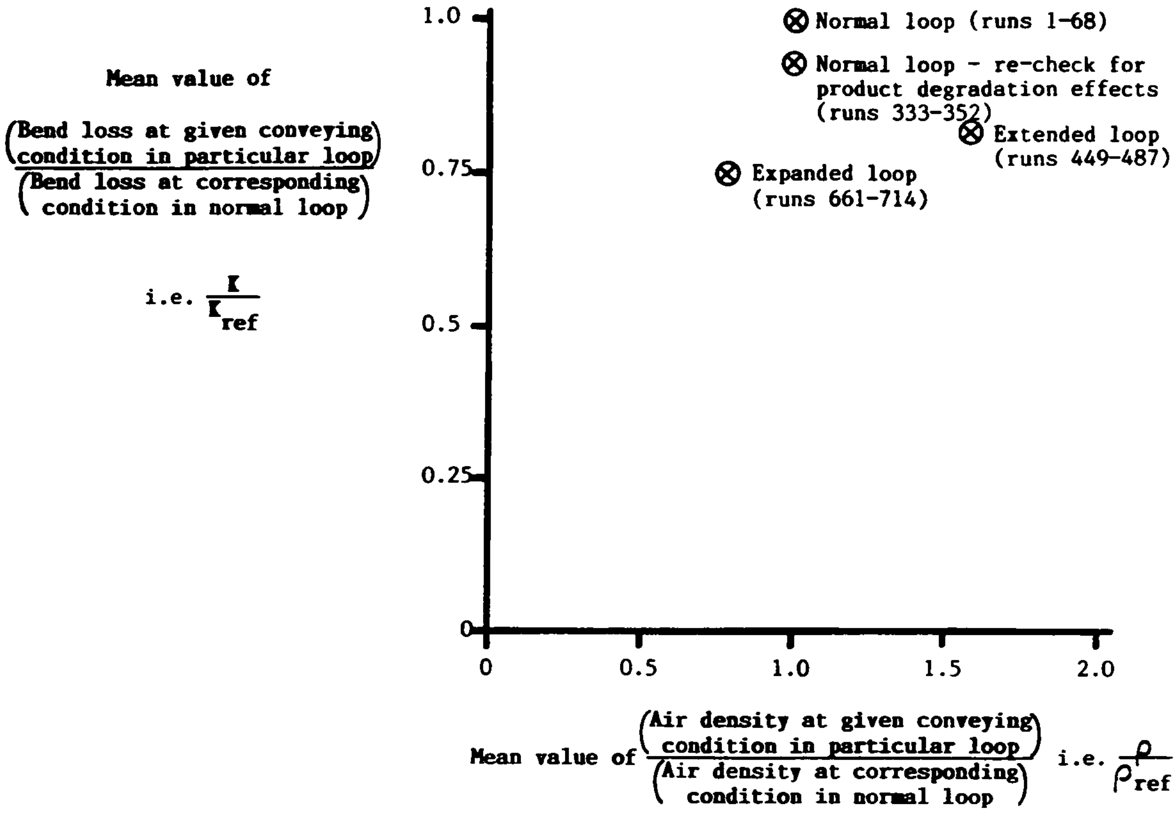


Fig. K-15

Graph of mean change in loss coefficient versus mean change in air density

The effect of air density on loss coefficient seemed inconsistent, the losses apparently reducing with both increasing and decreasing air density. However, it was noticed that the reducing losses corresponded to higher run numbers, so a graph of the mean change in loss coefficient versus number of runs was plotted, and the result was remarkable (overleaf):-

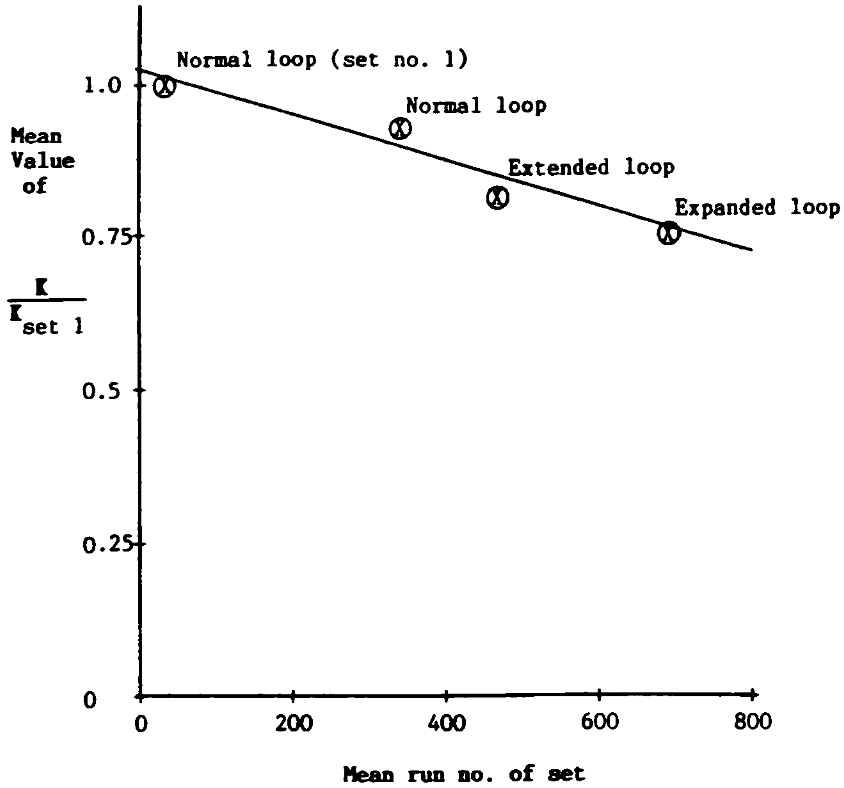


Fig. K-16
Graph of mean change in loss coefficient
versus number of conveying runs

The steady reduction of losses with more conveying runs was quite clear from this, suggesting that the cause of the change was most likely to have been changes in the product as it was conveyed rather than the changes in air density. To have rejected this idea would have been to accept a most peculiar relationship between losses and air density. The trend suggested the use of a straight line model; this was fitted and its equation deduced, which could be used as a mathematical "correction" to enable the results of all loss measurements with the flour to be referred to the values which would have been expected if a fresh batch of product had been used for every run. The change amounted to about 38% reduction in loss per thousand conveying runs.

Applying this correction for product change to the data for increased and decreased air density air tests, the graph of the mean change in loss coefficients versus mean change in air density looked like this:-

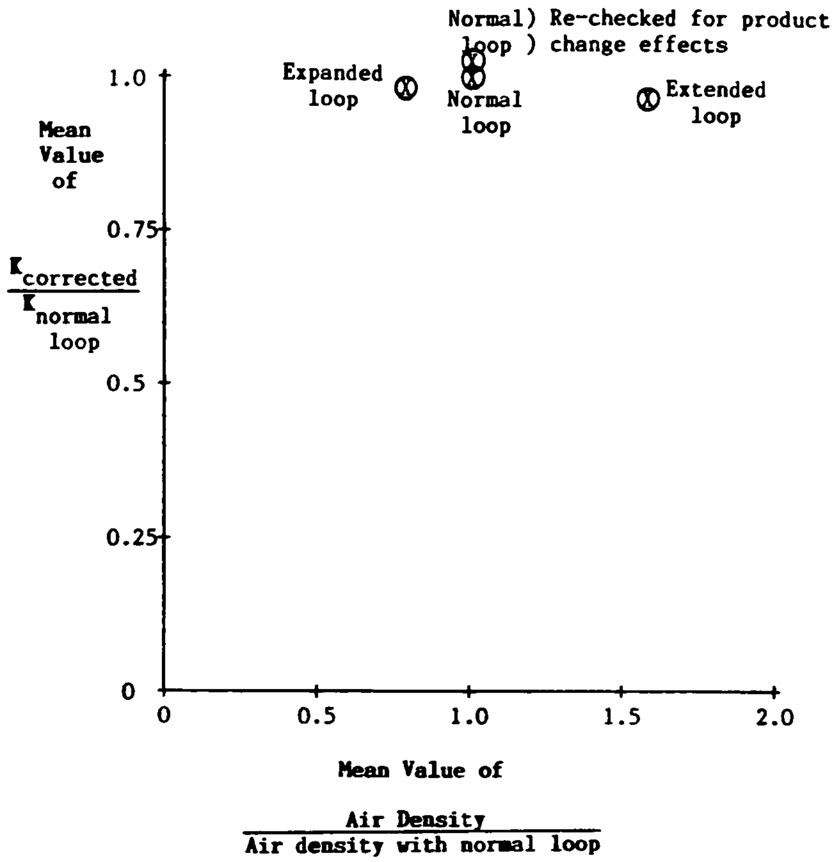


Fig. K-17

Graph of change in loss coefficient versus change in air density, with correction for product change applied

This confirmed that the loss coefficient was not dependent on air density to any detectable extent, and that the differences observed between the sets of data taken with one bend could be attributed to change in the product over the large number of conveying runs used.

The exact nature of the change in the product which could cause the reduction in pressure losses was a matter for conjecture. Flour was known to be biologically active, capable of changing its conveying characteristics with loss of moisture during storage and handling, as well as being subject to becoming 'stale' and also to degradation through normal particle breakage; however it was felt that the most important thing was that a suitable correction for this effect had been established.

K.3.3 More work on corrected K values with other bends

The correction described above was applied to the data for flour in the other bends for further analysis. Graphs of corrected loss coefficients versus suspension density were drawn, with curves of the standard shape (as per fig. K-14) scaled to fit the data for the radiused bends and elbows.

The resulting graphs for all the 2in. bends handling flour are shown overleaf for comparison.

It will be seen from examining these that the standard shape of curve represented the data well for all the radiused bends. The data for the malleable elbow fittings conformed reasonably well over much of the range but a standard curve over-estimated the losses under low-velocity flow. As can be seen, the data from the blind tee and vortice-ell bends did not conform to the same pattern as that for the radiused bends at all well; for these two bends a different model was used, simply two straight lines giving constant loss coefficients between limits of air velocity values.

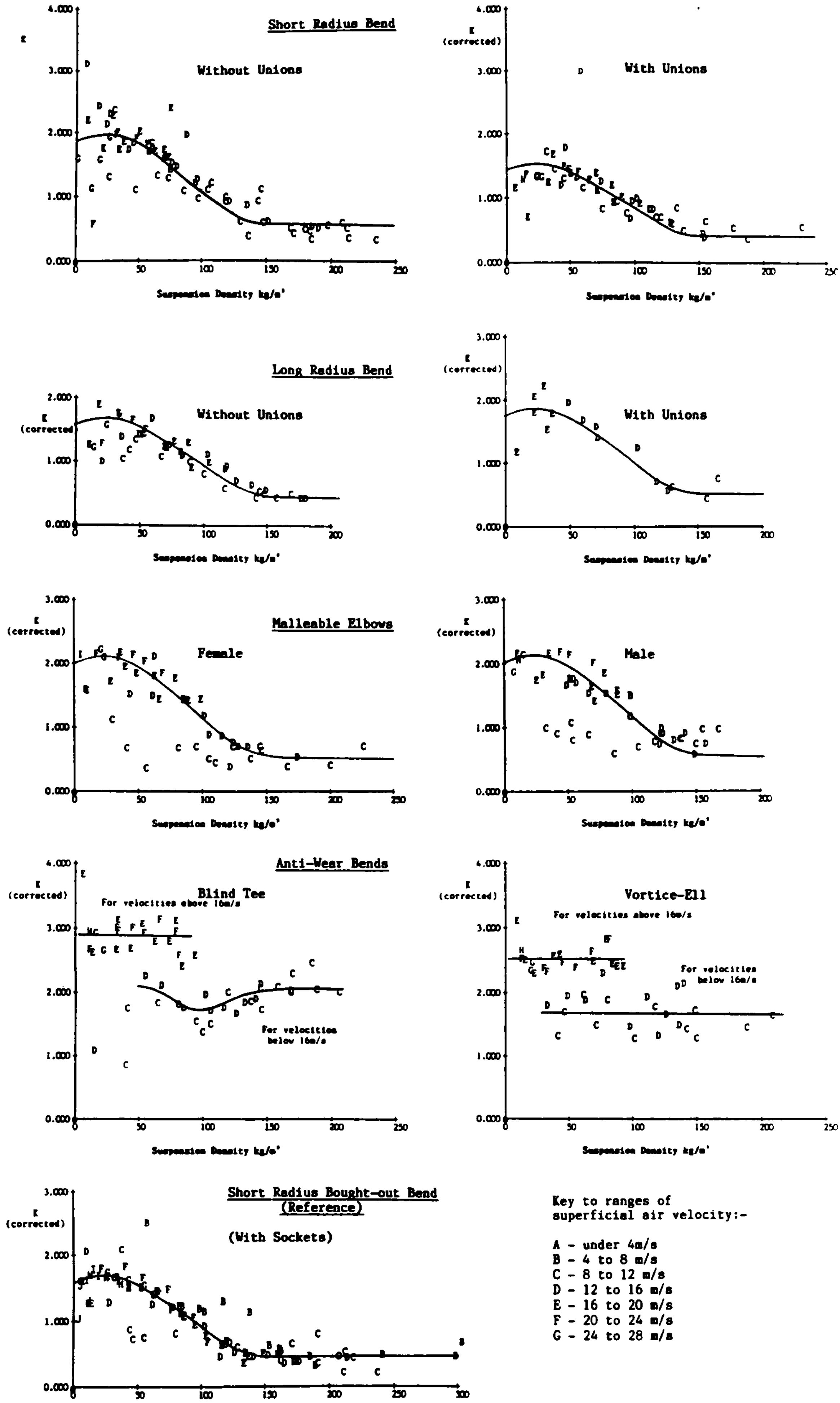


Fig. K-18
Graphs of loss coefficient (corrected for product change) versus suspension density for all the 2in. bends handling flour

K.3.4 Effect of bend geometry

Once it had been established that the standard curve could be scaled to represent the data from any of the radiused bends quite accurately, this raised the opportunity to compare the losses caused by the bends of different radius.

Some consideration showed that the short radius bought-out bend would be the proper one to use as a reference for comparison, since firstly it was the one which was used for the largest number of test runs (nearly a hundred), and secondly the results suggested that it should be the best choice for a conveying pipeline; this will become apparent from the analysis below.

With the short radius bought-out bend used as a reference, the factor scaling the curve through this data to the data of each other bend in turn was found. The values of the factors then became a convenient comparison between the radiused bends. The blind tee and vortice-ell bends displaying a characteristic of different shape, it was necessary to pick certain discrete conveying conditions at which to compare these against the others.

The result of this exercise, a graph of relative magnitude of pressure losses versus the ratio of bend radius to pipe bore, is shown in fig. K-19 overleaf.

The conveying conditions chosen for comparison of the blind tee and vortice-ell were taken broadly as (a) the usual spectrum of 'lean phase' (suspension flow) systems, with air velocities over 16m/s and suspension densities below 75 kg/m^3 , and (b) for a mid-range 'dense phase' (non-suspension-flow) system (suspension density 150 kg/m^3). Apart from these two, the comparison was valid for the wide range of conditions tested, save for the malleable elbows for which the standard curve somewhat over-estimates the pressure drop for a narrow range of low-velocity conditions.

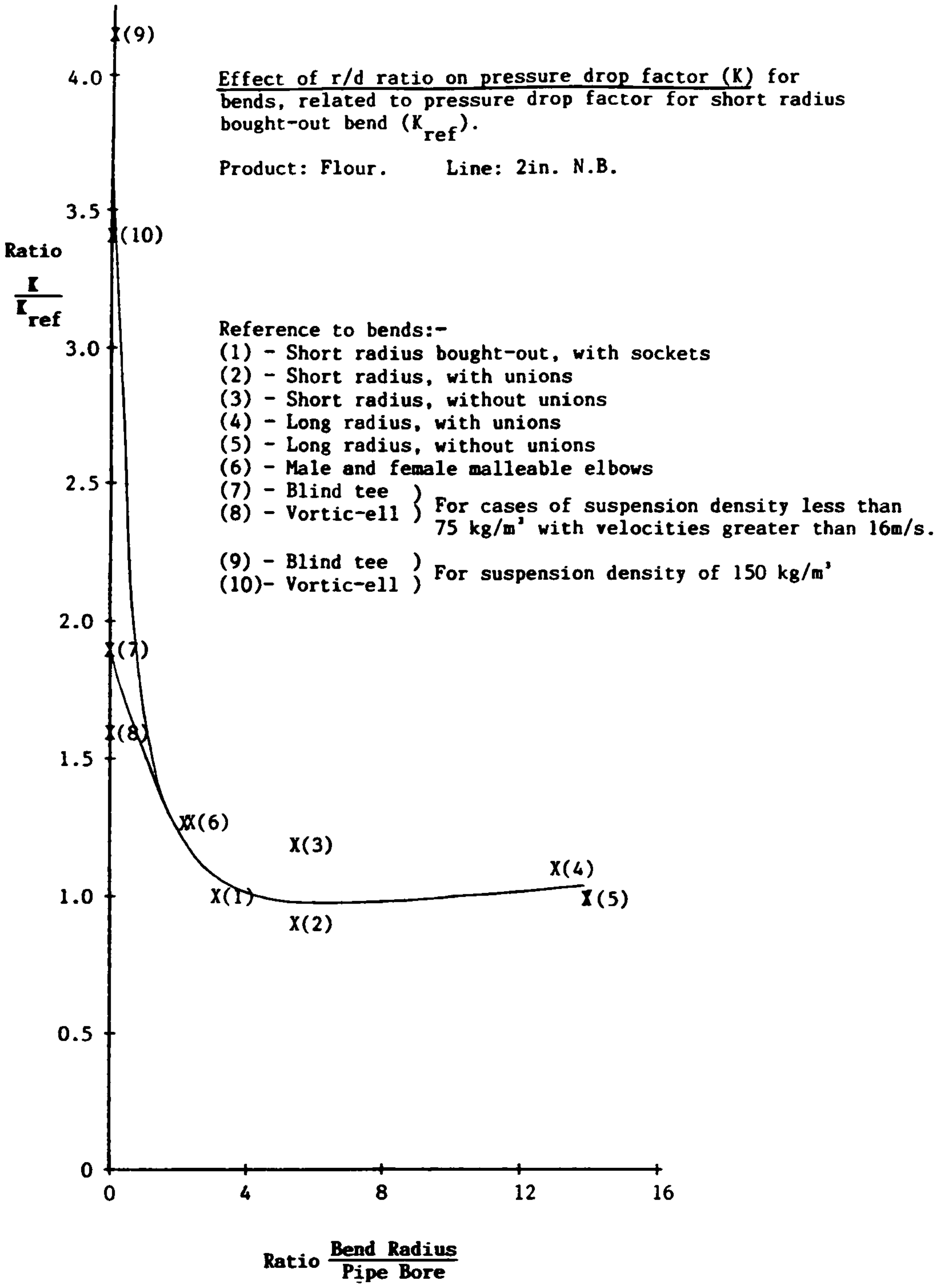


Fig. K-19

Comparison between relative losses between the various 2in. bends conveying flour

From this graph, it was apparent that the radius of the bends was of little consequence in terms of the pressure drop caused, provided an r/d ratio of about 3 was exceeded. The short radius bought-out bend looked to be the best since it is of course much cheaper to buy, and takes up much less space, than the long radius bends as usually specified for pneumatic conveying pipelines. The intermediate radius bends appeared to offer little advantage in comparison.

The malleable fittings did not seem to offer any advantages, because they gave significantly higher pressure drop over most of the range without being significantly cheaper to buy or more convenient to install than the short radius bought-out bend. That part of the range in which they appeared advantageous, i.e. where the standard shape of curve over-stated the pressure drop for these bends, was rather artificial, i.e. conditions of coincident low velocity and low suspension density; normally in a dense-phase system where low air velocities are used, the suspension density is deliberately kept high to achieve good throughput for economical operation.

The blind tee showed the highest loss in all cases, nearly double that of the radiused bends under lean phase conditions and up to four times as high in dense phase. This bend should clearly only be used where it is the only economic way of overcoming serious bend wear problems, and greatly increased pressure drop should be expected; particularly since the effect will be cumulative with a number of bends in a line, greater pressure drop at the first bend resulting in higher air velocity, in turn resulting in a disproportionately higher loss at the next bend, and so on. The poor performance in dense phase suggests that this bend must not be used under such conditions, particularly since far less bend wear is experienced with low velocity, dense phase flow. This may mean specifying blind tee bends only towards the end of a high pressure system where large increases in air velocity are expected.

The vortice-ell bend again showed a much higher loss than the radiused bends, though significantly less than the blind tee, suggesting that this would be a better alternative than the blind tee in cases where the latter could reasonably be used. The same comments with regard to dense phase systems apply.

K.3.5 Effect of fittings and ovality of cross section

The effect of unions on the losses was far from clear. The short radius bends displayed lower losses when fitted with unions than when joined to the adjacent straights with a smooth continuous bore, the reverse being the case for the long radius bends. This did not seem to make sense, so an alternative explanation for the significant differences between the bends of the same radii was sought.

It was apparent from looking at the various bends which had been made in house, that the natural distortion of the cross section occuring during bending was not entirely consistent from one bend to another. It was suspected that this may result in some difference between the losses caused by the bends, so an attempt was made to quantify the ovality of the cross sections for comparison. This was done by measuring the outside diameter of the tube at 11 stations from 0 to 90° around the bend, first in a radial direction and then in a direction at right-angles to this.

The difference between these two dimensions was calculated and plotted for each bend, the result being shown in fig. K-20 overleaf (top).

The obvious next step from here was to plot the relative pressure drop versus mean ovality, as shown in fig. K-21 overleaf (bottom).

As can be seen, no correlation was apparent. The bend with the greatest ovality of cross section displayed the lowest losses, yet the bend with the least ovality did not display the highest losses.

There were no other apparent differences between the bends of each radius, the bore being the same (they were all made up out of one length of pipe) and the radii being identical as far as could be detected, so no reason could be identified for the difference in relative losses between them. One possible explanation could be some non-linearity in the effect of product change with number of conveying runs, though the magnitude of this effect (as described in K.3.2 above) makes it seem unlikely that this could be responsible. No other possible explanations have become apparent.

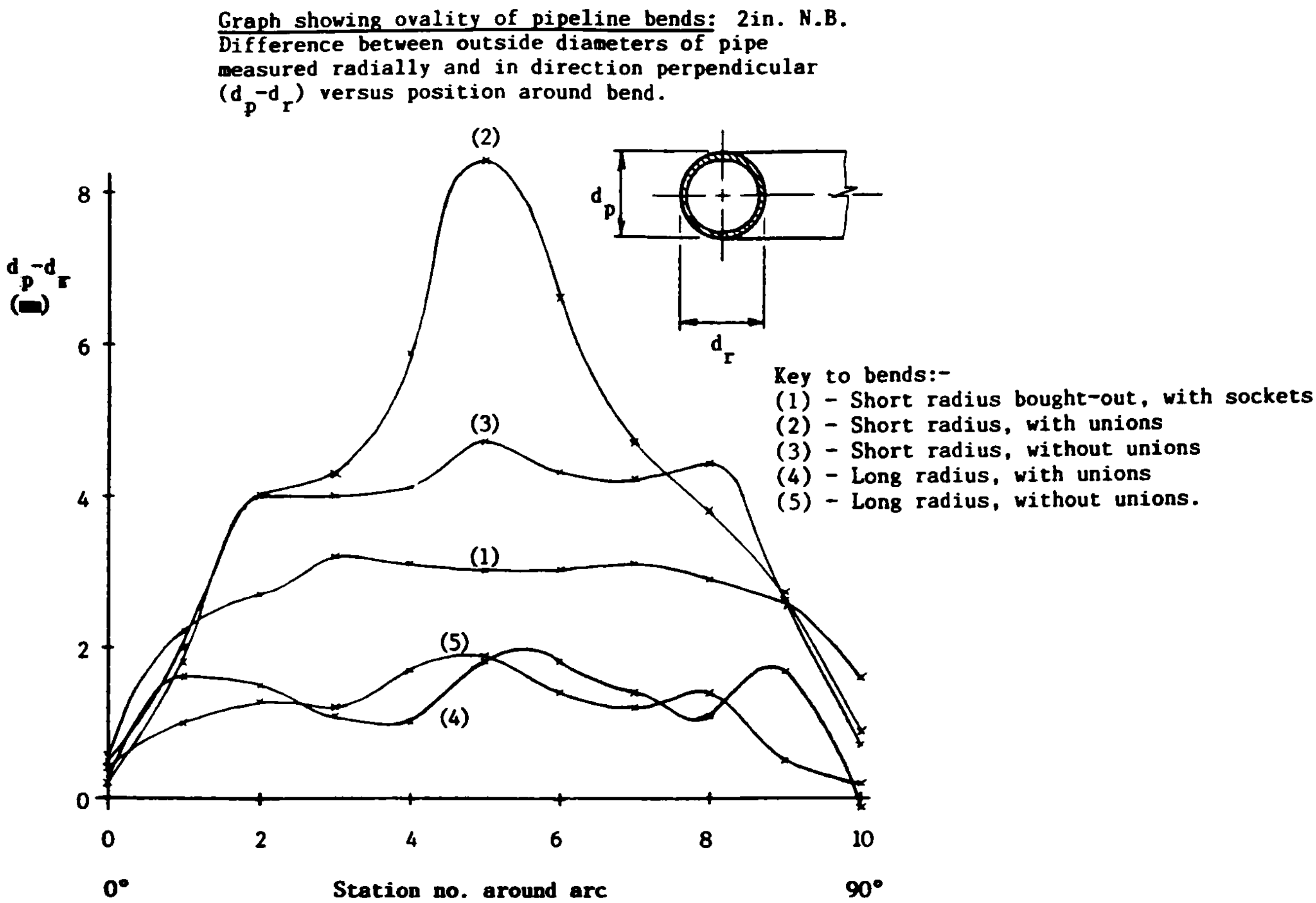


Fig. K-20

The ovality of cross section of the radiused bends

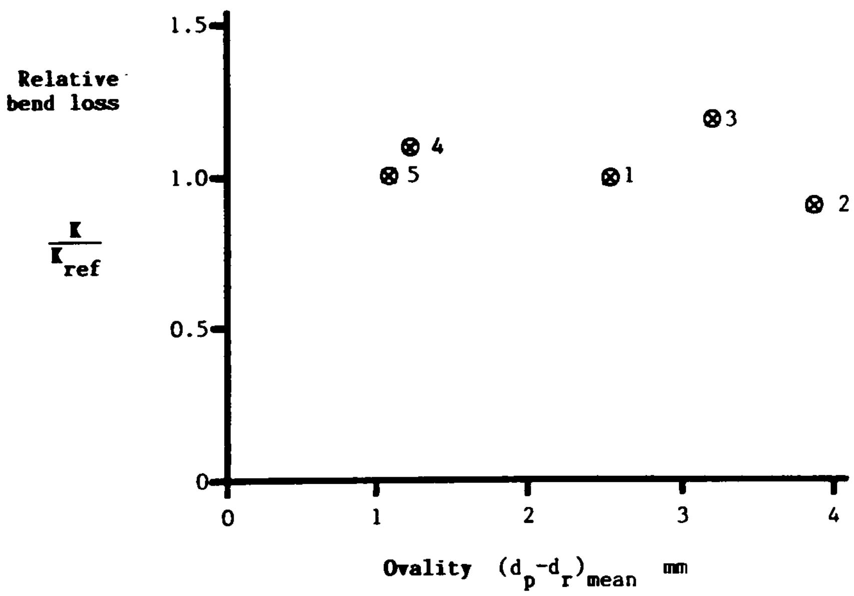


Fig. K-21

Graph of relative pressure drop versus mean degree of ovality of cross section

K.3.6 Effect of pipe bore

The data for the 3in. and 4in. nominal bore radiused bends was compared with the correlation established for the 2in. n.b. short radius bought-out bend, by drawing the correlation for the 2in. data on top of the 3in. and 4in. data. The results are as shown below:-

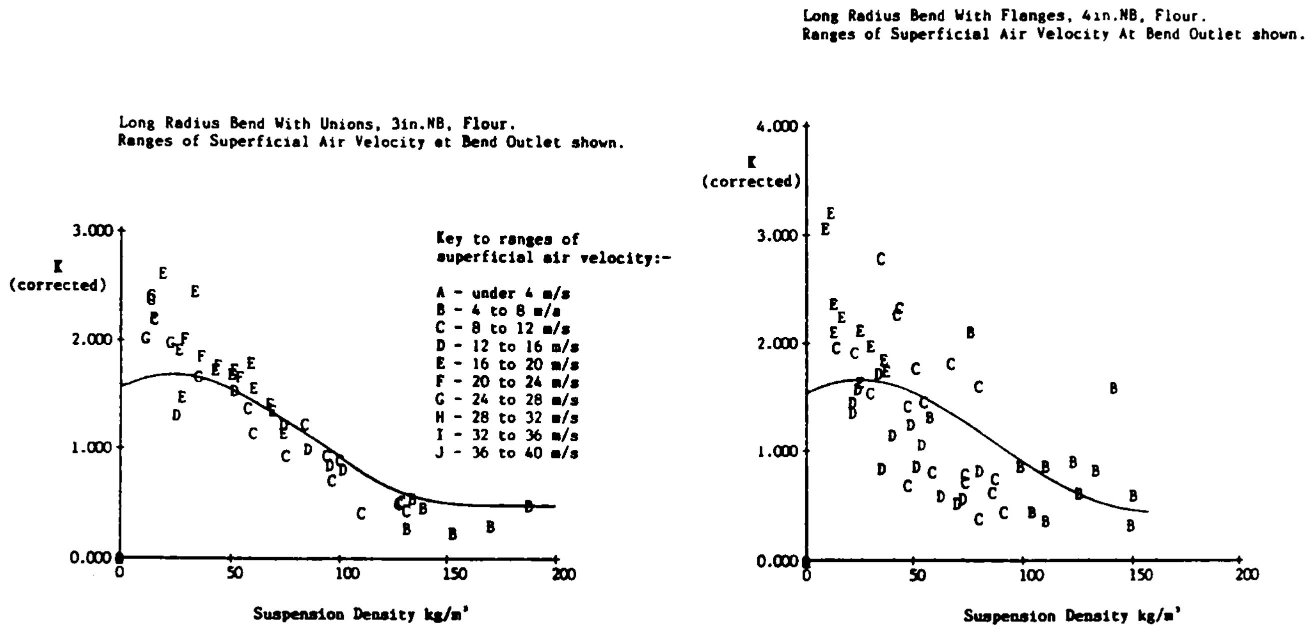


Fig. K-22

bend loss data for the 3in. and 4in. radiused bends, with the correlation for the 2in. short radius bought-out bend superimposed

Taking the 3in. bend first, the correlation from the 2in. data predicted the losses in mid-range operating conditions quite well, and appeared to slightly over-predict the losses at high suspension densities, which would lead to conservative design when scaling up in pipe bore. More serious was the under-prediction of losses at low suspension densities, which could result in over-prediction of system performance when scaling up for lean phase systems (say suspension densities less than 50). It was wondered whether this could be the result of wall effects being more marked in the smaller pipe, where the wetted perimeter is greater in relation to the cross sectional area. However, consideration showed that this ought to lead to lower pressure drop in the larger pipe, as otherwise this would imply that the wall effects are favourable; thus giving rise to smaller pipes being more efficient for the same flow per unit area which seemed unlikely. In general, though, the broad agreement was quite pleasing. It was observed that the magnitude of the deviation was less than the differences measured between apparently identical bends of the same bore as explained in section K.3.4 above.

Regarding the 4in. n.b. radiused bend, the amount of scatter was somewhat disappointing. The general trends were not dissimilar to those apparent from the data for the 3in. n.b. bend, with the 2in. data under-predicting losses at low suspension densities and over-predicting them at higher ones, but the general quality of the correlation was much poorer. The reasons for this are not clear, but it should be observed that the range of conveying conditions achieved with the 4in. bend was rather limited in comparison with the others, owing to limitations in flow rates of air and product imposed by the compressors and blow tank feeder.

In the 3in. nominal bore, a female malleable elbow had also been tested. Comparing data from this with that from the 2in. elbow revealed a significantly higher pressure drop, albeit with the pattern of data remarkably similar. This indicated that the poor performance of this type of bend was likely to be exacerbated with larger pipe bore, and although this only served to back the conclusion that this was not a good choice of bend, it did raise the question of whether the same effect would be observed with blind tees; this question could not be answered without performing further test work.

Overall the result gave some confidence in scaling up in pipe bore on the basis of equal pressure drop for equal flow of air and product per unit cross sectional area. The discrepancies noted were not significant in comparison with those between apparently identical bends of the same bore, indicating that the level of uncertainty in such scaling is less than the level of natural variation between similar bends, perhaps with a tendency to conservatism at intermediate and higher suspension densities.

K.3.7 Effect of product type

Although most of the work had been done using flour as the product conveyed, a relatively much smaller range of work had been done using a very different product, namely polyethylene pellets. This was a product with a single size (3.2 mm) of large, smooth particles, which was thought to be about as different from the wheat flour as it would be possible to get.

The extent of work done covered a single comprehensive set of conveying conditions using a single bend (the short radius, bought-out) with one pipe bore. When the usual graph of bend loss coefficient versus suspension density was drawn for this data, there seemed to be no clear correlation between these two, but a clear grouping with respect to air velocity. A graph of loss coefficient versus air velocity showed this clearly to be the principal relationship for this product:-

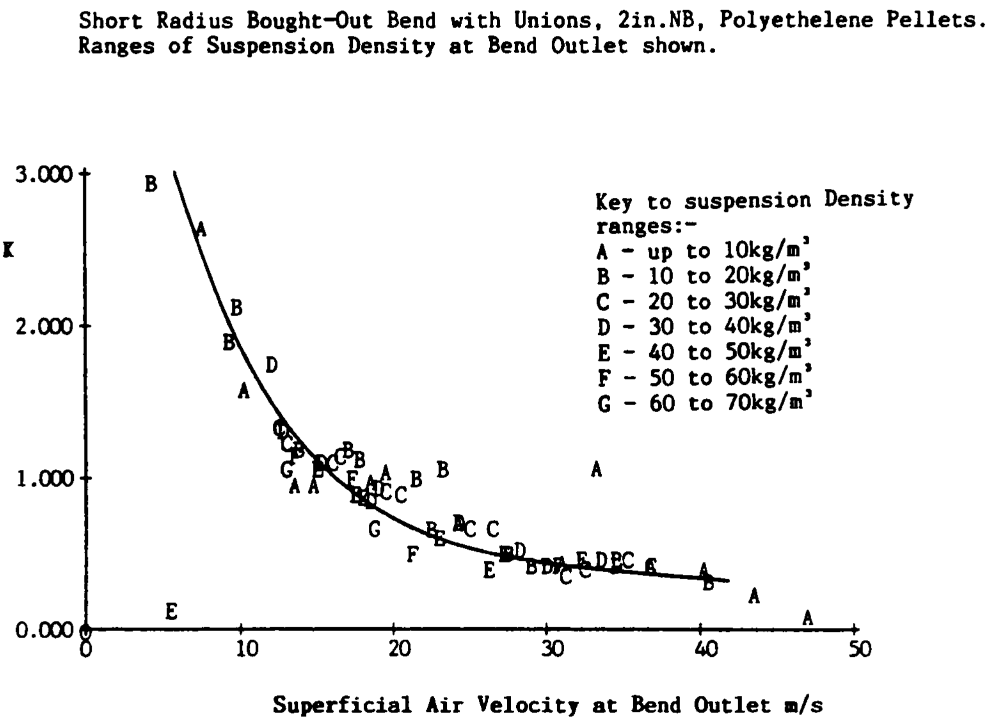


Fig. K-23

Graph of bend loss coefficient versus air velocity
for the short radius bought-out bend handling polyethylene pellets

This was quite unexpected given the consistent relationship between loss coefficient and suspension density, irrespective of velocity in most cases, for the bends handling flour. In other words, whilst the suspension density had the controlling influence with the flour, the air velocity had the controlling influence with the polyethylene pellets. No explanation could realistically be offered, other than to attribute it rather vaguely to the very great differences in character between the products; without having data on some range of products between the two, it would be foolish to try to offer any explanation of why this may be.

K.3.8 The use of surfaces to present the bend loss data

It was apparent that the relationship between the three variables of bend loss coefficient, superficial air velocity, and suspension density, could be visualised as a surface in three-dimensional space. An attempt was made to draw such surfaces from the data, taking three sets of data with distinctly different characters, namely that for the short radius bought-out bend handling both flour and pellets, and the blind tee handling flour. The results (overleaf) were quite revealing.

These showed up quite clearly the different nature of the relationships for the different bends and products. The similarity between the relationships for flour in both the blind tee and the radiused bend was more noticeable than had been the case from the 'flat' graphs, it becoming more apparent that they both displayed high loss coefficients under high velocity, low suspension density conditions, with lower coefficients at higher suspension density, lower velocity conditions. The different shape of the surface for the pellets seemed somehow less surprising when presented in this way, just displaying curvature in a different direction from that for the flour.

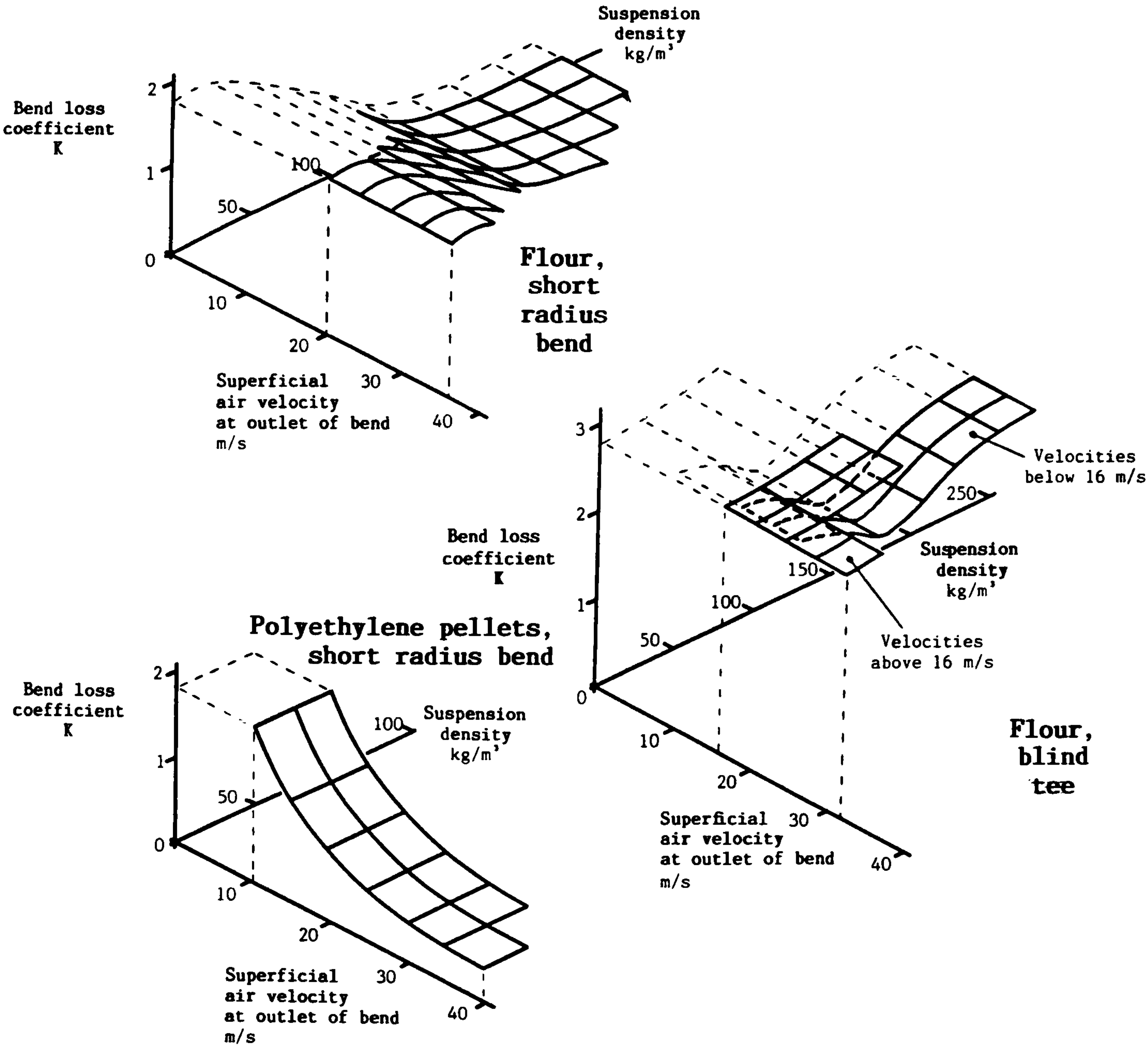


Fig. K-24

Isometric drawings of surfaces of bend loss coefficient versus air velocity and suspension density

It was also apparent that when viewed orthogonally (i.e. looking at the two flat graphs relating to each surface), what one is really viewing in each is a family of curves, with the curves just happening to fall one behind the other in one of the two views, leaving apparently a single curve. It seemed that this had come about purely by chance with these two products, and that in general one might expect the surfaces to be curved in both directions, so that a single correlation between loss coefficient and either suspension density or air velocity would not be apparent.

K.3.9 Summary of bend loss data analysis

At this point it was decided to draw the work on analysis of the bend pressure drop data to a close by taking stock of what had been accomplished. A usable system for storing measured pressure loss data, consisting of a very simple dimensionally-homogeneous equation in combination with a coefficient dependent on two variables, had been developed. The equation included the same two variables, both easily calculated from measurable quantities. The data was stored as the relationship between the coefficient and the two variables, on either of two graphs.

The effect of product type was known to have significant effect, not only in terms of the values of the loss coefficient, but also in terms of the shape of the relationship between the coefficient and the other two variables. The effect of bend geometry had been demonstrated for one product, and the effect of air density had been shown to be insignificant with the data stored in the way described. The effect of change in the product over the large number of test runs had been evaluated and its effect corrected for.

It was therefore felt that considerable progress had been made in this direction and although a number of questions had been raised, the method was sufficiently well developed to be useful as part of the equipment used for design of real pneumatic conveying systems for industrial applications.

K.3.10 Straight pipe pressure drop

Much less work was done on this than had been done on bend pressure drop. This was not because it was felt to be unimportant, but because it was decided that the best policy would be to concentrate efforts on one area to at least make significant progress somewhere. Consequently the progress made with finding a suitable storage system for the straight pipe data was rather less than for the bend loss data.

So far a graph of 'solids contribution' to pressure drop in the straight pipe (i.e. total pressure gradient minus that expected for air only) versus suspension density had been established as giving a good correlation, using a square law curve, irrespective of air velocity, for the flour.

i.e. $$\left(\frac{dp}{dl}\right)_{total} = \left(\frac{dp}{dl}\right)_{air\ only} + \left(\frac{dp}{dl}\right)_{solids}$$

with $$\left(\frac{dp}{dl}\right)_{solids} = (\text{constant}) \times (\text{suspension density})^2$$

This very simple correlation had been established by manually calculating the 'air only' pressure drop corresponding to the air velocity for each conveying run, and then subtracting it from the measured gradient. It should be pointed out that this correlation was not dimensionally homogeneous, so the constant had units.

The next step was to modify the data processing software to calculate the solids contribution to pressure drop directly, on the same basis. With this convenient facility it became quite easy to try some different presentations of the data. A graph of solids contribution versus suspension density was drawn for a large set of data from the work on flour, namely the runs done using the short radius bend without unions. The result is shown overleaf.

It was interesting to note that in some cases, the solids contribution appeared to be negative, i.e. the total pressure gradient measured was less than that which would be expected with air only in the pipe. This had been reported by certain other authors, although no great significance was attached to it; the conditions under which it occurred, very low suspension densities combined with air velocities excessively high for a well designed conveying system, were not of much interest within the current study.

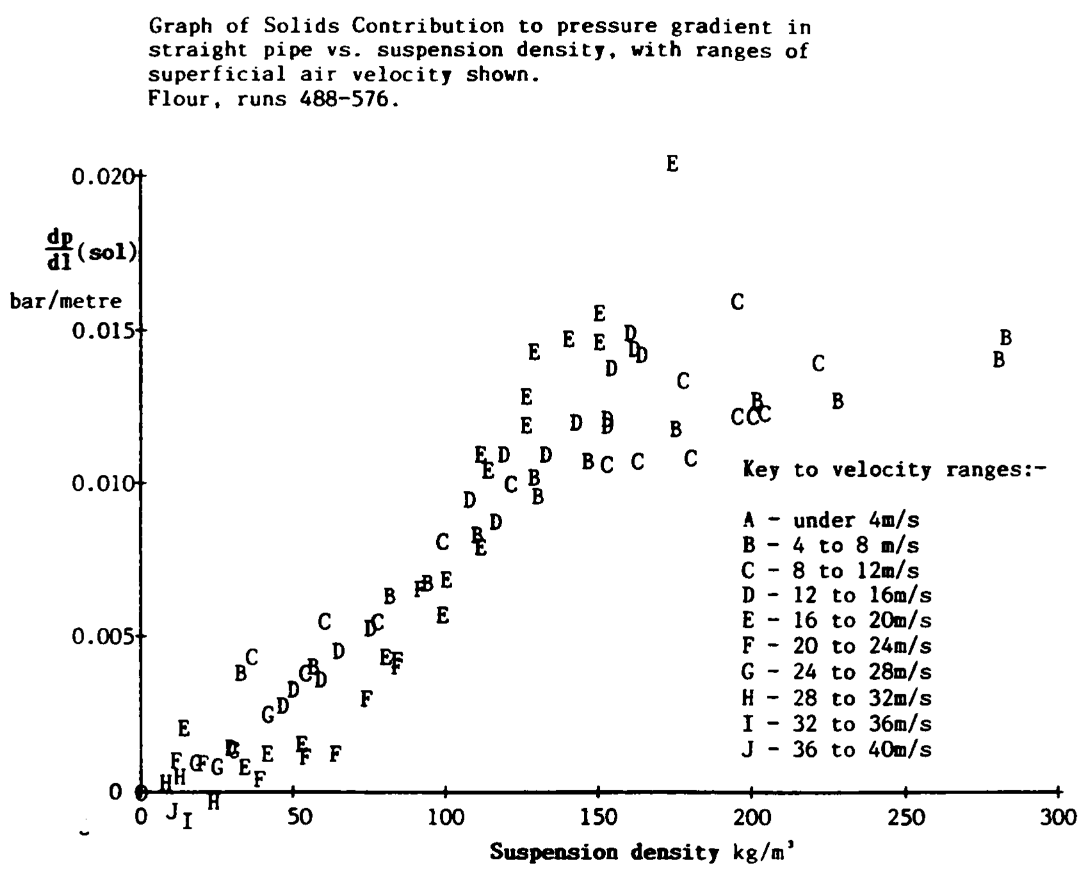


Fig. K-25

Graph of solids contribution to straight pipe pressure drop versus suspension density for a range of the data on flour

Overall, there was again quite a good correlation apparent. It seemed fairly clear that there was a tendency for the shape of the correlation to change with air velocity, being a fairly straight line (or even a slightly convex curve) at lower velocities, turning into a concave curve at higher velocities. In other words, the index of a suitable power law through the data would increase with velocity.

Thus if the equation

$$\left(\frac{dp}{dl}\right)_{solids} = (\text{constant}) \times c^n$$

(Where c = air velocity)

were to be used, the value of the index n would be approximately 1 for the lower range of velocities, giving a straight line, rising to 2 in the higher range of velocities, giving a square law. The lower range of velocities seemed, from the graph, to be centred around 8m/s with the higher range centred around 16m/s. A simple model relating the value of the index n to velocity was used to represent this:-

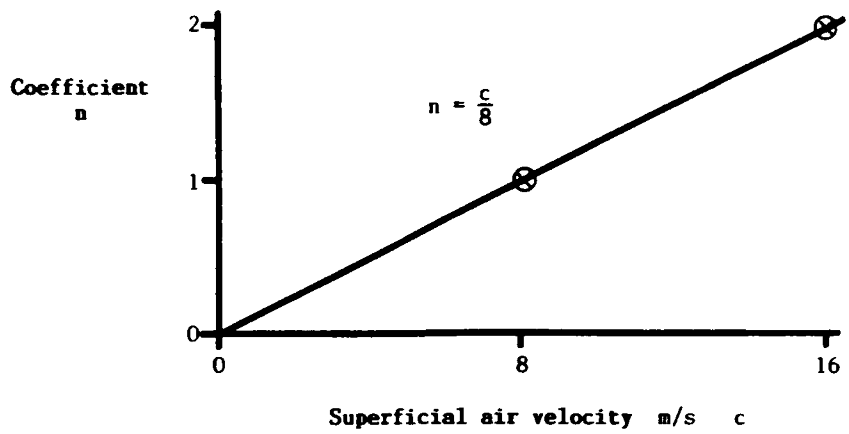


Fig. K-26

model used to represent the relationship
between n and air velocity for flour

To obtain a suitable value for the coefficient of proportionality, the point at which the two lines crossed (the square law for 16m/s velocity and the straight line for 8m/s velocity) was examined and a suitable value calculated to be 6.5×10^{-3}.

In order to test this correlation, a graph of measured values of solids contribution versus values predicted from this model, for the conveying conditions of each of the data points, was drawn (overleaf):-

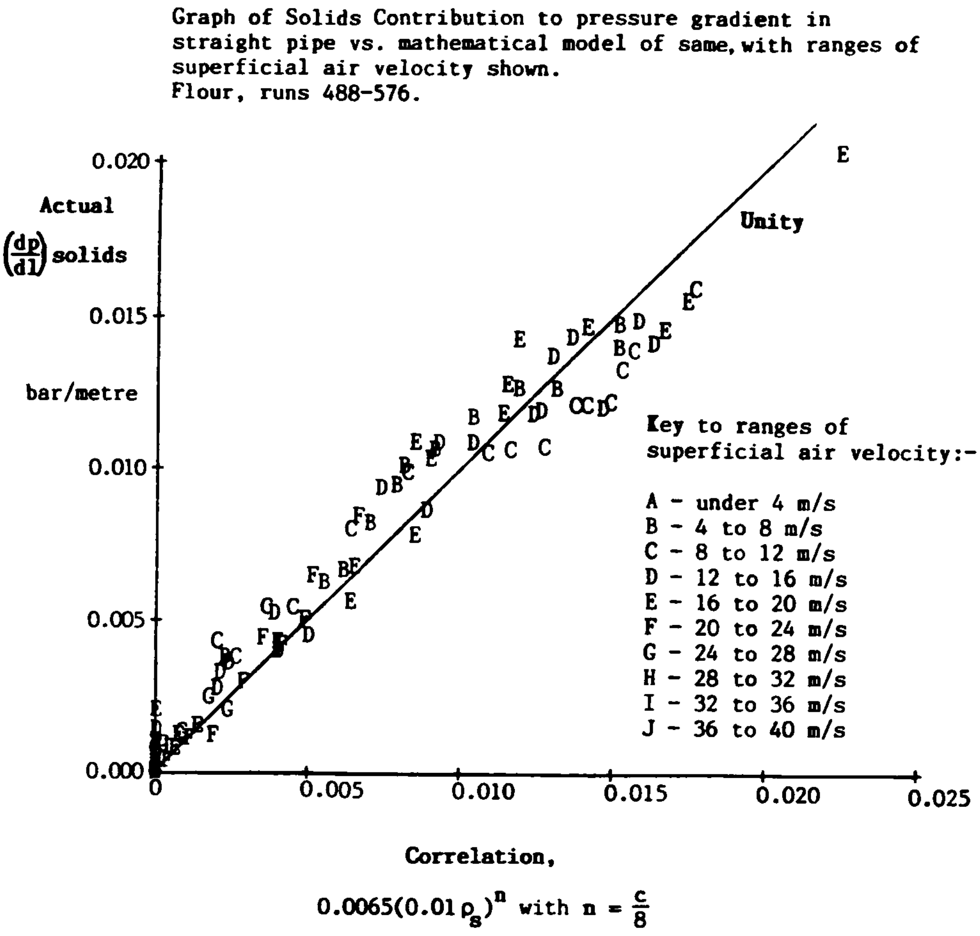


Fig. K-27

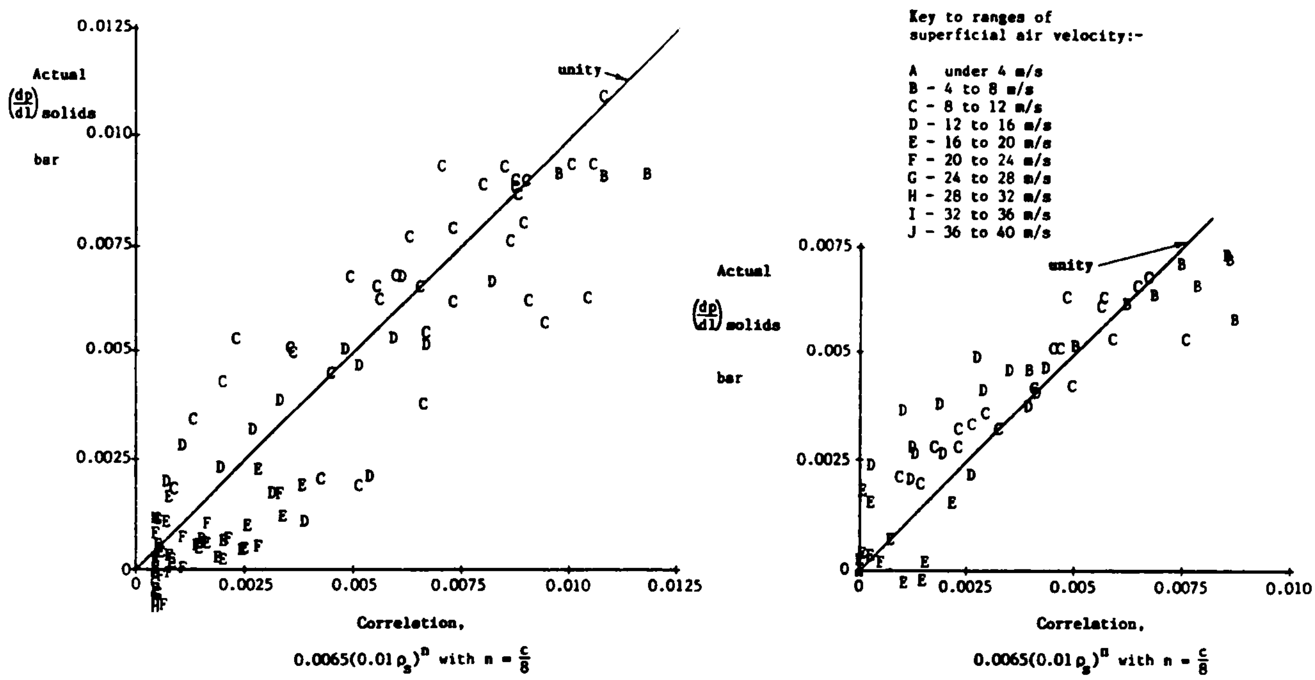
Graph of measured solids contribution versus value predicted from correlation, for flour in 2in. line

In order to do this, the points which showed a negative solids contribution were eliminated since the calculation method involved taking a root of this quantity, it being impossible to obtain a fractional root of a negative number.

The outcome of this exercise was quite pleasing, showing the correlation to be quite accurate over the range tested. Further thought showed that perhaps the idea of using a continuous relationship between the value of the index n and the velocity may not have been appropriate. An alternative would have been to consider n to have two distinct values, i.e. 1 below 16m/s and 2 above 16m/s, which would fit in well with the usual distinction between fully-suspended flow and non-fully-suspended flow; normally it is taken that a velocity of about 15m/s separates these two regimes. Although this was not tried, it was apparent from fig. K-25 above that the data for above 16m/s could be represented well by a square law and that for below 16m/s could be equally well represented by a straight line.

K.3.11 Effect of pipe bore on straight pipe pressure drop

Although no comprehensive work was directed at assessing the effect of pipe bore, an attempt was made to compare the correlation established for the 2in. pipe against the data for the 3in. and 4in. pipes. The result was as shown:-



3in. line 4in. line

Fig. K-28

Comparison of correlation from 2in. n.b. pipe against data for 3in. and 4in. pipes, for flour

It was apparent from this that to use the same correlation would not be unreasonable; it tended to slightly under-predict the pressure gradient with increasing bore, particularly at high velocities and low suspension densities (i.e. for lean phase systems); this would lead to slightly conservative design given that the plant pipeline is usually larger in bore then the test line.

Turning to the solids contribution for the polyethylene pellets, drawing a graph of solids contribution versus suspension density showed some correlation, suggesting the use of a straight line through the origin, of slope increasing with velocity:-

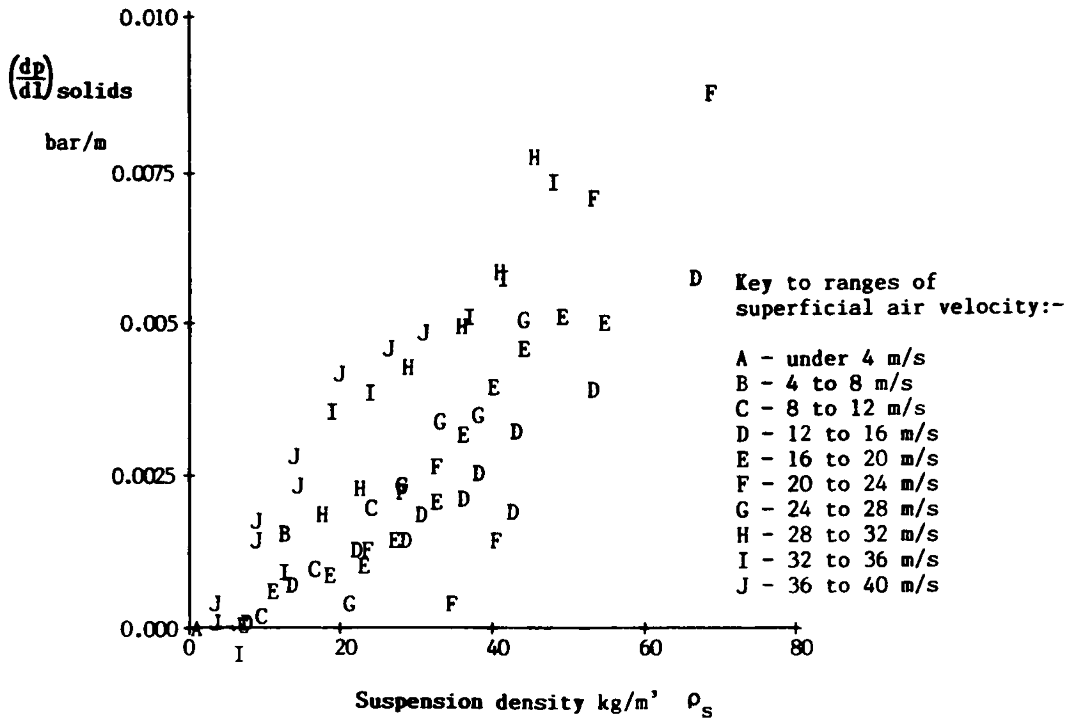


Fig. K-29

Graph of solids contribution to pressure gradient versus suspension density for the polyethylene pellets

The equation of such lines would be

$$\left(\frac{dp}{dl}\right)_{solids} = M \times \text{suspension density}$$

where M would be the slope of the line, expected to increase with velocity. Values of M versus velocity were plotted by solving the above equation for M with the appropriate values of suspension density and solids contribution to pressure gradient for each data point. The resulting graph is shown overleaf:-

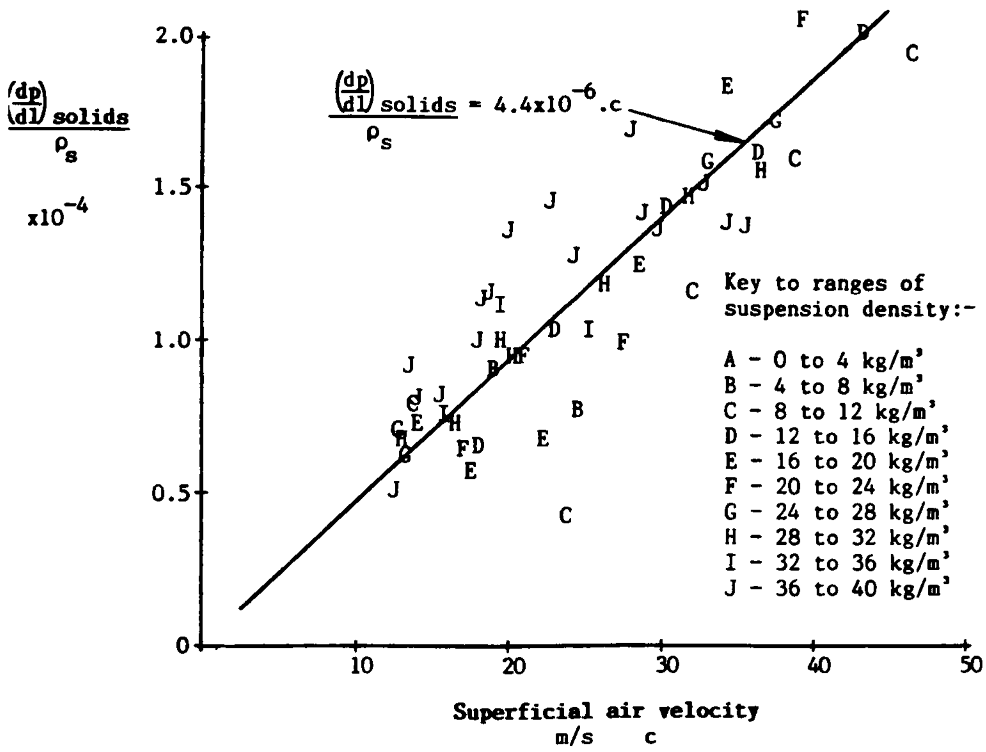


Fig K-30

Graph of value of coefficient M versus velocity
for the data on straight pipe pressure drop for polyethylene pellets

This graph showed a fair amount of scatter, but the trend of M to increase with velocity was clear. The straight line through the origin as shown above was about the best representation achievable, giving

$$M = \frac{\left(\frac{dp}{dl}\right)_{solids}}{\text{suspension density}} = \frac{2.2x10^{-4}}{50} \text{ x air velocity c}$$

Rearranged for the solids contribution this gave the final correlation,

$$\left(\frac{dp}{dl}\right)_{solids} = 4.4 \text{ x } 10^{-6} \text{ . c } .\rho_s$$

where c is the superficial air velocity and ρ_s is the suspension density.

Some further effort was put into trying to find more uniform ways of expressing the correlations established for the solids contribution to pressure gradient in the straight pipe, and perhaps making them more dimensionally homogeneous, but this was not successful and was soon curtailed owing to lack of time. It was apparent, from the very different nature of the correlations necessary to represent the data from flour and polyethylene pellets, that other products would probably require equally different forms of mathematical models, and that to make any real progress on identifying patterns would require a large amount of data from many products. With such data at hand it may be possible to relate the type of model necessary to the quality of the product itself, but without such data it seemed pointless to try to proceed further.

K.4 Summary of data analysis

A number of things came out of the extensive analyses performed on this data. Firstly it was apparent that the use of the microcomputer was a great boon; it would have been very time consuming to have handled and processed the vast volume of data by hand, and practically impossible to have achieved manually the presentation and re-presentation of the data in so many ways, on so many graphs, in the time available. It was very easy to just try out different approaches and discard them easily if not useful; the hundred or so graphs which found their way into hard copy represented only a fraction of the total number which had been looked at.

The prime objective of the data analysis had been to find suitable storage systems for the data, i.e. methods of representing the data in ways which would be easily written down or drawn out, for use in predicting pressure losses when designing pneumatic conveying pipelines for these particular products. It was felt that this had been fulfilled, and that it was likely that equally useful systems could be established for other products through similar test work.

The volume of data obtained was beneficial. With so much data covering such a wide range of flow conditions it was much less easy to become misled into forcing an inappropriate correlation onto a few data points.

That is not to say that the correlations established are thought to be universal for the products tested, after all they were only used as a means for storing the data in a compact and easily accessible form. It would be wrong to apply them outside the range of conditions which were actually tested, but the important point is that this range of conditions is comprehensive in terms of what is required to design an economic conveying system, and it proved possible to find suitable, simple, storage systems for the purpose required.

It had been shown that the types of data storage systems necessary for bend pressure losses and straight pipe pressure gradients were quite different. This explained the general disagreement which had been observed amongst users of the traditional method of expressing bend losses in terms of equivalent lengths of straight pipes, and showed this to be an inappropriate means of accounting for bend losses.

The type of data storage systems needed for bend losses had been shown to be very similar for the two products. This had been shown not to be the case for the pressure gradients in the straight pipes. It was felt that the systems had been developed about as far as possible using the available data; the system for bend losses was felt to be very good, but the system for straight pipe gradients was not entirely satisfactory because of (a) lack of dimensional homogeneity and (b) lack of consistency between the products. No further progress could realistically be made on improving this without more data from other products.

The effect of air density had been comprehensively evaluated for bends, and the effect of pipe bore determined. The effect of bend geometry had been examined in great detail for one product and some useful results obtained.

It was interesting to note that the findings indicated that the 'air only' loss was a significant part of the straight pipe pressure gradients whilst being totally insignificant to the bend losses. This was in line with the much greater proportion of the total pressure loss being caused by the bends in a pneumatic conveyor, compared with the flow of single phase fluids where bend losses are normally only a tiny part of the overall pipeline loss.

Overall, the success of the data analysis was found to be most rewarding, and encouraging in terms of supporting the use of the method of pipeline design which had been proposed in the earlier stages of the work.

APPENDIX L

CASE STUDIES USING THE METHOD DEVISED

L.1 Introduction

A case arose whereby this author was asked to comment upon certain options for a pneumatic conveying system being engineered for a client, to convey 60 tonne/hour of copper concentrate into a smelting process along a route 60m long with probably 15 bends, approximately equally distributed along the line. Being an abrasive product, there was considerable interest in keeping conveying air velocities low, possibly by the use of a line with increases in bore size along its length.

It seemed that this presented an ideal opportunity to try out the method which had been devised for prediction of pipeline performance, since to make such comparisons using the testing and scaling method would be very time consuming and in any case not able to deal with the stepped lines. Initially the data taken with flour was used, in the hope that by showing the potential power savings through careful design, it may be possible to persuade the client to come forward with a contract to test the actual product and carry out the design work. In the event no such contract was forthcoming, but nevertheless the work was extremely useful in demonstrating the ease with which such analyses may be carried out using this method.

L.2 The study

Since the system being considered was to handle an abrasive product, there was every incentive to keep air velocities low by the use of a stepped pipeline; at the same time the extra cost involved with stocking pipeline parts of more than one size, for maintenance, provided a conflicting incentive to use a single bore line. Therefore the analysis would need to begin by looking at single bore lines before examining the possibilities of stepped ones.

The question of bend type was also seen to be important, because some sort of abrasion resistant bends would be necessary. The simplest alternatives were to use either blind tees which are both cheap to buy and very resistant to abrasion, but (from experience) give higher pressure drop; or to use radiused bends with a ceramic lining whose resistance to abrasion is also quite satisfactory and whose pressure drop is much more modest, but which are many times more expensive to buy.

The pipeline would feed into a reactor vessel at 20psig, but this did not complicate the analysis particularly; the pipeline synthesis program was simply altered to begin with this pressure at the end of the pipe instead of atmospheric.

Primary design criterion, apart from the conveying duty and the line layout, was the minimum conveying air velocity of the product (taken to be 11.8 m/s, which was calculated to be the air velocity at inlet for the system which the vendor had specified). This was taken to be the inlet air velocity for all the lines designed, and the velocity aimed for after steps in the line.

L.2.1 Single bore lines

The first line considered was the one specified by the vendor, 4in. nominal bore using 0.57 kg/s (1000 scfm) of air. The data measured with flour was used, for straight pipes and radiused bends. Running the computer program gave a value for the air pressure at inlet to the line of 3.36 barg (i.e. 2 bar pressure drop along the line, to the reactor at 1.36 barg) and a superficial air velocity of 11.8 m/s at inlet, which was taken as the design minimum conveying velocity of the product, as related above. Air velocity at the outlet of the pipe would be 22.4 m/s.

The use of blind tees was next considered. By using the data for flour flowing through such a bend and running the computer program repeatedly with different air flow rates, it was found that an air flow rate of 1.24 kg/s would be necessary to achieve the design minimum conveying air velocity, and this would result in a much higher pressure at inlet, of

8.5 barg, with correspondingly higher outlet air velocity of 49 m/s, clearly very high for an abrasive product. Thus it was seen that to use blind tees would result in considerably greater running cost.

Observing the large increase in velocity along the line, it was apparent that it may be possible to use radiused bends towards the beginning of the line where velocities are lower, then to change to blind tees further along, say half way. Accordingly the program was changed again to use the data on radiused bends from bends 1 to 7 and then blind tees from there on. Trying different air flow rates to obtain the design air velocity at inlet showed that .924 kg/s of air would be needed, giving an inlet pressure of 6.1 barg and outlet air velocity of 36m/s, still high but rather more reasonable. The air velocity in each of the radiused bends was less than 13 m/s, which was considered quite reasonable.

To illustrate the air velocities and pressures in the different systems, diagrams of pressure versus distance and velocity versus distance were drawn. These are shown in fig. L-1 below.

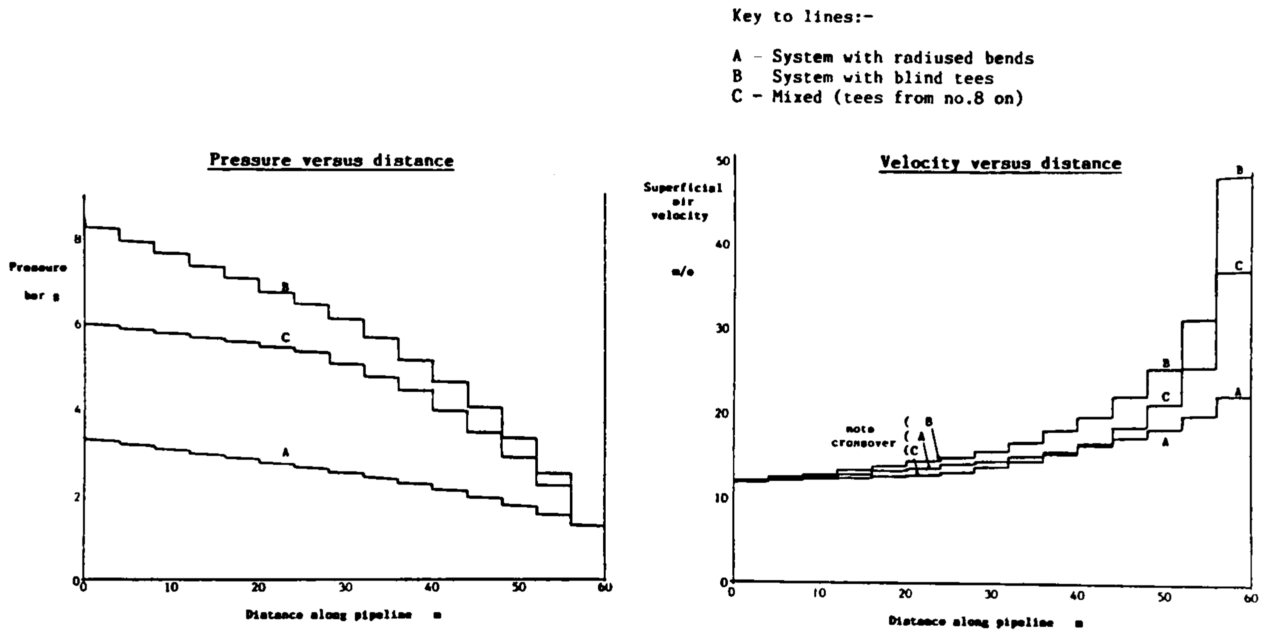


Fig. L-1

Diagrams of pressure and air velocity versus distance for the single bore lines with different bend configurations

It is quite noticeable from these diagrams that the reduced pressure drop in the section with radiused bends, as compared with the same section with blind tees, leads to less expansion of the air and thus a lower pressure drop in the following section with tees for the "mixed" line; in addition to which, the air consumption and pressure drop are significantly lower although the bends which operate at the higher velocities are blind tees.

L.2.2 Stepped bore lines

The next stage was obviously to consider the use of a stepped-bore pipeline. First a step from 4in. to 6in. nominal bore was considered. A couple of runs of the program with the step at different locations showed that this was too much of an increase in bore size in one step, with the air velocities in the initial 4in part needing to be excessively high in order to prevent th air velocity in the 6in line after the step from falling too low - even with the step being located almost at the end of the line.

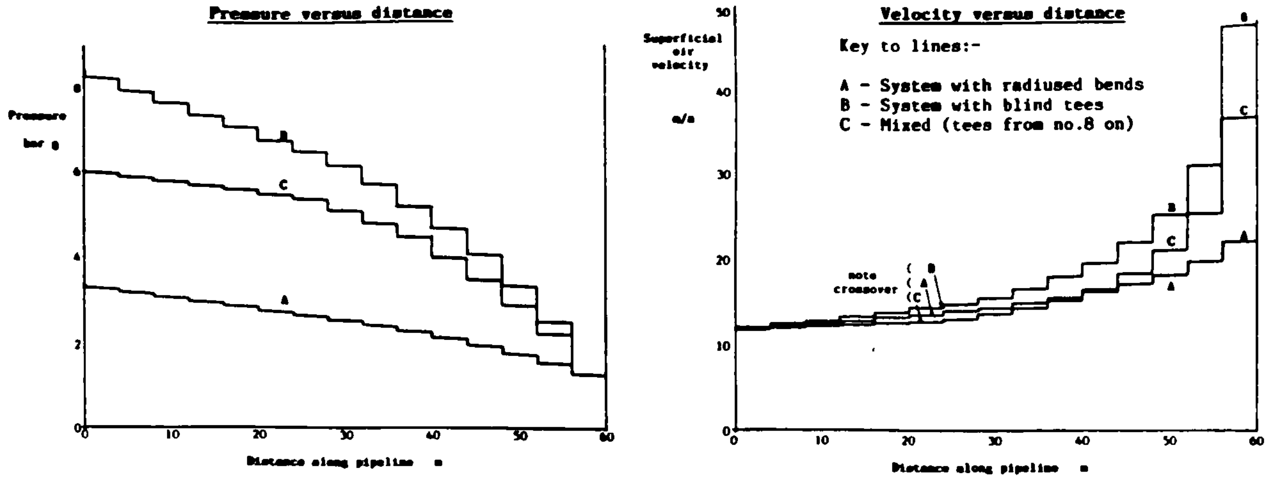
Therefore it was decided to consider a step from 4in. to 5in, using radiused bends. The step was first located at the last bend and various air flow rates tried, to obtain the one which would give the design minimum conveying air velocity either at the beginning of the pipe or just after the step; for this case the lower velocity occurred at the beginning of the pipe. The step was moved back to the previous bend and the procedure repeated, with the same result, and this whole process repeated until the velocities at the inlet and just after the step were approximately equal, which was the best position for the step. This position was at number 13 out of the 15 bends. The pressure drop and air requirement were slightly lower than for the single-bore 4in. line, but more significantly the maximum air velocity was reduced to 18m/s (compared with 22m/s), a worthwhile reduction bearing in mind the strong dependence of wear rate on velocity (wear being proportional to between fourth and fifth powers of velocity is the relationship frequently quoted, e.g ref. 1).

The next stage was to repeat the analysis with a 4in. to 5in. pipeline but using blind tees. The best position for the bend was located in the same way, by finding the location which gave the air velocity just after the step and the air velocity at inlet both equal to the design minimum conveying air velocity, for an appropriate air flow rate. The position turned out to be bend no.8 out of 15, i.e. much nearer the beginning of the pipeline, and compared to the single-bore line with blind tees the pressure drop and air requirement were virtually halved, the improvement in maximum velocity being even more substial; down to 22 m/s from the original 49 m/s with the single-bore line. The reduction in wear caused would clearly be very marked, even taking a conservative view of the fourth-or-fifth-power relationship mentioned above.

It seemed as though there may even be a case of stepping twice, from 4in. to 5in., then to 6in; with the best locations of the steps determined to be bends 7 and 13, by the usual criterion, the pressure drop and air consumption were slightly reduced again and the maximum velocity down to 18m/s, again a worthwhile saving.

Pressure-distance and velocity-distance diagrams for all of the stepped-pipe systems were drawn for comparison; the results are shown in fig. L-2 overleaf, those for the single-bore lines being presented again for comparison.

Single Bore Lines:-



Stepped Bore Lines:-

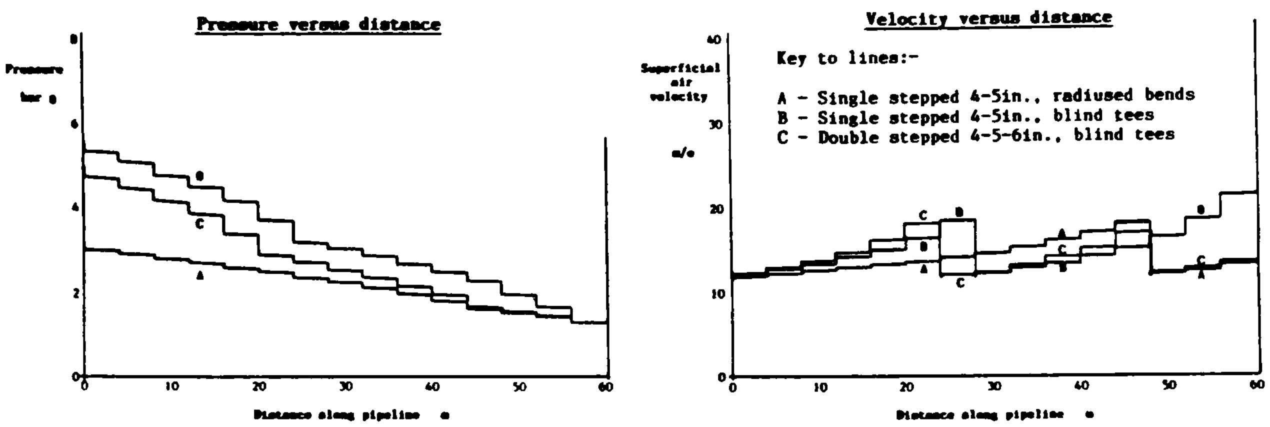


Fig. L-2
Pressure-distance and velocity-distance diagrams
for all of the systems, for comparison.

L.2.3 Comparison of power consumption and velocities

Two major factors to be considered in any design for a system for this duty were clearly the cost of power and the cost of maintenance; the first could be calculated easily from the power requirement, which could in turn be obtained from the air pressure and flow rate, and the second would be dependent on wear of pipes and bends which although not easily calculable would clearly be dependent on air velocities. Therefore it seemed that to bring this study to a conclusion, the various pipeline options should be compared on these bases.

Compressor power was calculated using a fixed value of 46.9kW/(bar.kg/s), obtained simply by examining the motor power and output of a number of compressors of different pressures and throughputs around the laboratory.

The comparison between the systems is shown below:-

Nominal bore in.	No. of steps	Bend type	Line inlet pressure bar g.	Air mass flow rate kg/s	Power kW	Maximum velocity m/s
4	-	rad.	3.4	0.57	90	22
4	-	b.t.	8.5	1.24	494	49
4	-	b.t.	6.1	0.92	264	36
4-5	1	rad.	3.1	0.53	76	18
4-5	1	b.t.	5.6	0.86	225	22
4-5-6	2	b.t.	5.0	0.79	183	18

(b.t. = blind tees: rad. = radiused bends)

From this it is quite obvious that the reduction in power consumption with a line using all blind tees, of the order of 54% with a single-stepped line and 63% with a double-stepped line, are very significant indeed; the reductions in air velocity, of 55% and 63% respectively, are probably far more important considering the strong influence this is likely to have in reducing pipeline wear, and also product degradation in cases where that is a problem.

For the line using all radiused bends, where inlet pressure is much lower, the power saving of 15% is not terribly significant, but again the 18% reduction in air velocity is likely to be more important.

L.3 Conclusions

L.3.1 Stepped pipelines

Although the object of the study was to evaluate the design method, the lessons learned about the advantages of stepped pipelines were obviously very important. The savings in power and reductions in air velocities, particularly for the systems using high operating pressures (i.e. above about 4 bar g.), showed that this is a very useful way of enabling long conveying distances to be covered, or high throughputs to be obtained, where single-bore lines would be prohibitively expensive to run and maintain.

L.3.2 The method

The ease and speed with which pipeline options could be compared using the method developed in this project showed part of its value; the work described in this chapter took about half a man day, compared with the technique of scaling from the results of a test line which would have required considerably longer just to deal with the single-bore lines. This leads on to the other and possibly more significant advantage, that stepped lines could be examined, the best position for the step(s) located and the performance evaluated very quickly, something quite impossible to achieve using the scaling method but very useful as outlined in 3.2 above.

APPENDIX M

WORKED EXAMPLE

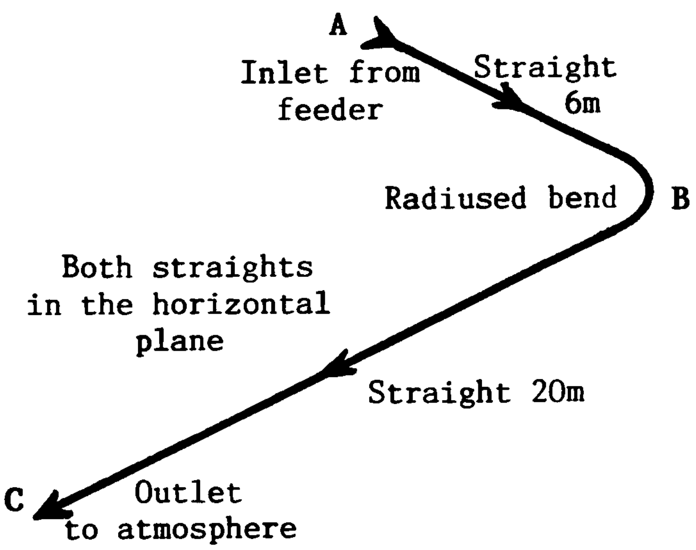
PREDICTING PRESSURE DROP USING THE METHOD RECOMMENDED

M.1 Introduction

This Appendix gives a simple, short example of how the method recommended in the Thesis operates, predicting the pressure loss along a short pipeline conveying one of the products tested.

M.2 The requirement

Say a pipeline is to be designed to convey flour at a rate of 3 tonne/hr over the short route shown (right). To decide upon a suitable size of pipe, it is necessary to consider a range and work out the pressure drop and air requirement for each so that they can be compared on a rational basis.



So let us begin by taking a 2in. nominal bore pipe (53mm bore). An air mass flow rate of say 0.05kg/s might be as good a starting point as any.

M.3 Calculations

If this is to be a positive pressure system ending in a hopper which is at atmospheric pressure, then calculation would begin at the outlet end of the line where the conveying conditions (air velocity and suspension density) can be found;

CONVEYING CONDITIONS AT END OF SYSTEM, POINT C;

The volume flow rate of air, $\dot{V}$, can be established from the mass flow rate of air $\dot{m}$ and air density; air density in turn can be found from pV = mRT;

$$\frac{pV}{m} = \frac{p}{\rho_a} = RT; \qquad \text{thus} \qquad \rho_a = \frac{p}{RT}$$

Taking atmospheric conditions as 1 bar abs., 288K, $\rho_a = \dfrac{1\times10^5}{287\times288} = 1.21\text{kg/m}^3$

so volume flow rate of air, $\dot{V} = \dfrac{\dot{m}}{\rho_a} = \dfrac{0.05}{1.21} = 0.0413\text{m}^3/\text{s}$

Cross sectional area of pipe, A, = $d^2/4 = 3.14\times0.053^2/4 = 0.002206\text{m}^2$.

Thus air velocity at end of pipe, c, = $\dot{V}/A = 0.0413/0.002206 = 18.7\text{m/s}$.

Now the suspension density is $\dfrac{\text{Mass Flow Rate of Product}}{\text{Actual Volume flow rate of air}}$;

Mass flow rate of product = 3 tonne/hr = 0.833kg/s.

Thus suspension density, ρ_s, = $\dfrac{0.833}{0.0413} = 20.2\text{kg/m}^3$.

PRESSURE LOSS IN STRAIGHT B-C;

Knowing these conveying conditions it is now possible to calculate the pressure loss in the last straight length (B-C in the diagram above, 6m long) using the equation given in section 3.3.2 (using the data storage system for pressure loss along straight pipes with flour);

$$\left(\frac{dp}{dl}\right)_{total} = \left(\frac{dp}{dl}\right)_{air\ only} + \left(\frac{dp}{dl}\right)_{solids}$$

the "air only" pressure gradient can be found using the Darcy equation, taking say f = 0.005, on a per metre basis;

$$H = \frac{4fl}{d}\frac{c^2}{2g} = \frac{4x0.005x1x18.7^2}{0.053 \quad 2x9.8} = 6.67\text{m of air per metre;}$$

$$\Delta p = \rho gH = 1.21x9.8x40 = 474N/m^2 = 0.00474 = 0.00079 \text{ bar/m}$$

The "solids contribution" can be found using $\left(\frac{dp}{dl}\right)_{solids} = 6.5x10^{-3}\left(\frac{\rho_s}{100}\right)^n$

with n = c/8 (again section 3.3.2).

$$n = 18.7/8 = 2.33. \left(\frac{dp}{dl}\right)_{solids} = 6.5x10^{-3}x\left(\frac{20.2}{100}\right)^{2.3} = 0.000156 \text{ bar per metre}$$

Thus total pressure gradient = 0.00079 + 0.000156 = 0.000946 bar/m.

Length of this straight = 6m so total pressure loss is 0.000946 x 6 = 0.00568 bar. Thus pressure at inlet to this straight will be outlet pressure plus 0.00568 = 1 + 0.000568 = 1.00568 bar abs.

CONVEYING CONDITIONS AT INLET TO STRAIGHT B-C;

Knowing the new air pressure it is possible to calculate the new air density, thus the new air velocity and suspension density; using the same process as outlined above these come out at 18.6m/s and 20.3 kg/m^3.

LOSS AT BEND B

The conveying conditions at the inlet to straight B-C are those at the outlet of bend B, so knowing these it is possible to calculate the loss at bend B using the data storage system for bend losses with flour; from section 3.3.1 the equation is

$$\Delta p = K.\tfrac{1}{2}\rho_s c^2$$

where the value of K for flour in the short radiused bend was 1.7 for a suspension density of 20.3 (fig. 3.1, Chapter 3).

Thus loss at bend B, $\Delta p = 1.7 \times 0.5 \times 20.3 \times 18.6^2 = 5970 N/m^2$, i.e. 0.0597 bar.

CONVEYING CONDITIONS AT INLET TO BEND B;

The pressure at the outlet of bend B (inlet to straight B-C) was 1.00568 bar abs., and the pressure drop caused by bend B was 0.0597 bar so the pressure at inlet to the bend will be 1.00568 + 0.0597 = 1.0654 bar abs. Recalculating the air velocity and suspension density gives 17.6m/s and $21.5 kg/m^3$ respectively.

PRESSURE LOSS IN STRAIGHT A-B;

The conveying conditions at the end of this straight are those at the inlet to bend B, i.e. 17.6m/s, $21.5 kg/m^3$.

Using these conditions to calculate the pressure gradient in length A-B, in the same way as shown above for length B-C, gives an "air only" pressure gradient of 0.00070 bar/m, and a "solids contribution" of 0.00022 bar/m, total 0.00092 bar/m. Thus the pressure drop will be, over the length of 20m, 20x0.00092 = 0.0184 bar.

PRESSURE AT INLET TO LINE, POINT A

Knowing the pressure at outlet of straight A-B is 1.0654 bar abs., and that the pressure drop in A-B is 0.0184 bar, the pressure at inlet to this straight (i.e. inlet to the line) will be 1.0654 + 0.0184 = 1.0838 bar abs., in other words 0.084 bar gauge.

CONVEYING CONDITIONS AT INLET TO LINE, POINT A

Now using the pressure at point A it is possible to calculate the conveying conditions at this point; using the same procedure as above yields an air velocity of 17.3m/s and suspension density of $21.9 kg/m^3$.

This completes the calculation for this example.

M.4 Comments

Having found the inlet air pressure and velocity for the chosen mass flow rate of air in the chosen size of pipe, and finding in this case that the air velocity at inlet is considerably above the minimum conveying velocity established for the product during the testing (and thus might be considered excessive) the next step would be to try using a lower air flow rate, to establish the most economic operating condition for this size of pipe. The exercise would then be repeated for other sizes of pipes so that all the various options can be compared.

From the above it will be seen that if the pipeline is of any real length or complexity of layout, a great deal of repetitive calculation is called for to carry out a full comparison of the many options usually available. This of course suggests the use of a computer to assist the work; some comment is made on this matter in Chapter 3.

APPENDIX N

LITERATURE SURVEY

N.1 Introduction

When first investigating the literature pertaining to the design of pneumatic conveyors, it quickly became evident that the volume of papers and other publications was enormous; however, closer inspections revealed that the vast majority of these dealt with only small details, very often novel developments or specific solutions for rather specialised problems. The number dealing with the general philosophy of design was in fact quite restricted, and even of those, relatively few gave full procedures which could be followed by practicing engineers. These are reviewed first here.

The particular problem of predicting pressure drop in the pipeline, which as indicated in the opening of Chapter 2 is the necessary basis for any good design, had however received rather more attention over the years; there appeared to be literally hundreds of papers published, mainly in English, German and Japanese, which had some relevance to this problem, but again the majority of them dealt only with very specific areas, generally without reference to the context in which the proposed theory or method might be useful. To catalogue and criticise every one would be a massive, and undoubtedly unrewarding, task, given this fact; so what follows on this subject is a criticism of those which offered to shed some significant light on the problem.

N.2 Design of systems and prediction of pipeline pressure drop

Nine publications dealing with these questions were seen to have something significant to offer, most of them being text books; two gave detailed design strategies (i.e. procedures for deciding on the 'best' system) whilst the others gave more or less deatiled procedures for estimating pressure drop.

N.2.1 The EEUA Handbook

The oldest, and one of the most comprehensive, was ref. 35, the handbook entitled "Pneumatic Handling of Powdered Materials" published by the Engineering Equipment Users' Association. This Association was active in the early Sixties, being made up of a number of large processing and chemical companies in the U.K., with the express purpose of promoting a better understanding of the equipment related to all areas of process plant. The book was number 15 of a series of handbooks, which between them covered a very wide range of processing and handling equipment. With such a pedigree, much might have been expected of this book and indeed it came up to a very good standard, being easy to read and follow and very specific in its recommendations.

Although it does not overtly state a philosophy for choosing the best system, it gives the user a method for calculating the air volume and pressure drop, and thus the power requirement, for systems conveying specific products. It does this by first giving minimum safe conveying air velocities and maximum safe suspension densities for a range of 25 products (these two quantities being calculated in the same way as recommended in Chapter 4 of this thesis), then goes on to present a method for predicting pressure drop based on adding together three components, namely the pressure drops caused by (i) acceleration of the product, (ii) pipeline friction, and (iii) changes of direction. Each component is predicted from a rational, dimensionally-homogeneous expression containing a notional 'dynamic pressure' of the flowing suspension and a 'Friction Factor', experimental values of which for each component are given.

Values of the factor for straight pipe friction are shown to vary very greatly with air velocity and also between the 6 products for which they are given (calculated from the published data of 5 workers), but the factors for acceleration and bend losses are not said to be dependent on either air velocity or product type. It is pointed out that "these are difficult to measure..." but "From analysis of test figures it seems reasonably safe to take the values given". The effect of bend radius is shown to have an effect on the coefficient. The bends are not, however, considered separately, but lumped together as though the losses caused by them occurred in one place. A single velocity figure is used to calculate the notional 'dynamic pressure' of the flowing suspension, which does not take account of the variation in velocity along the section being considered; consequently the recommendation is made that if more than 10 lbf/in^2 pressure drop is expected, then the line should be divided up into sections treated separately; the use of 'stepped' pipelines is mentioned to overcome excessive increase of velocity.

The objections to this method are several. Firstly no account was taken of the variation of minimum conveying velocity with suspension density which since the work of Mills (ref. 1) is now widely accepted.

Secondly the treatment of the whole pipeline using one air velocity, even given the dividing up into sections if the pressure drop exceeds 10 lbf/in^2 (which in many systems it does) - this alone could lead to as much as a 1.6:1 change in air velocity, which according to the graph of straight pipe friction factors vs. velocity given, would result in a very much greater error in the values of friction factors for straight pipe.

Thirdly, for bends no allowance is given for the loss factors to change with velocity, although the work described in this thesis shows that the values are very much dependent on air velocity - perhaps this is hardly surprising given that the values quoted were not from direct experimental data on bend pressure losses but deduced from measurements of whole pipelines, and no doubt this also accounts for the fact that the bend loss factors given are very low compared with those measured in the current work. This results in considerable under-estimation of the losses caused by the bends, which hopefully is balanced by over-estimation of the amount of the losses attributatble to the straight pipes; such a situation is only acceptable for the design of systems having the same number and distribution of bends as those from which the data was taken.

Finally it is not explained how the factors were arrived at from the experimental data referenced, which makes it difficult for the user to fit his own data (from operating experience or a test rig) into the same framework; on this basis the user is restricted to taking a guess at which of the six products quoted may give a similar pressure loss to his own, and using the data from that for design - the possible inaccuracy is not to be underestimated, with a 10:1 range in values of the friction factors over the 6 products for one velocity. Although the text does mention the need for pilot scale testing for reliable design, it gives no indication as to how the data from such tests may be either scaled or used to determine appropriate values for the friction factors.

From the above comprehensive criticism, it may be thought that this reference was not very useful; in fact the opposite was true, it was by far one of the best. It presented an approach which seemed to be clearly in the right general direction, with the possibility of improvement apparent; it was one of only three of the many books examined whose methods were not restricted to lean phase flow only. Its method was not unrelated to the method recommended in the conclusion of this thesis, and it is for this reason that it has been criticised so comprehensively, to illustrate the difficulties involved even if a useful approach is adopted.

N.2.2 Thames Polytechnic notes for 'Short Course in Pneumatic Handling of Bulk Materials'

This (ref. 1) also appeared to be a very comprehensive document, dealing with all aspects of the technology of pneumatic conveying as well as pipeline design; it recommends a rational design method based on minimising the power consumption of a system, to minimise operating costs (considerations of product degradation/equipment wear notwithstanding). The prediction of pressure drop in each possible pipeline size for a given duty is done by using the results of tests on the actual product for which the plant is to be designed. The product would be conveyed in a pilot scale test system, to obtain data on pressure drop with a range of flow rates of solids and air, as well as the minimum conveying velocity. This data would be scaled, using a set of empirical rules, to predict the pressure drop in the possible plant pipelines; bends being dealt with using an 'equivalent length' approach to find a total equivalent length of pipeline and scaling for length. The exact procedure is described and illustrated in Appendix B of this thesis, and the drawbacks, namely the impossibility of taking proper account of changes in the number and distribution of bends and also the impossibility of designing stepped pipelines, are explained.

The advantage of this method is that real data can be employed easily, the techniques are not restricted to lean phase flow only, and it is readily understood; on the other hand, it had clearly reached the limit of its development as far as improving it to take account of these factors was concerned. A more complete analysis of the technique and its limitations is given in App.B.

N.2.3 Other books dealing with determination of pipeline pressure drop

The next book worthy of mention was 'Pneumatic and Hydraulic Conveying of Solids', ref. 36. This offers no strategy for choosing the 'best' system, but at first glance it at least seems to offer a comprehensive method for prediction of pressure drop; a closer inspection shows that most of the material actually relates to pipelines carrying air only. Again a total equivalent length is used to allow for bends, but based on single phase flow data which the work in this thesis, and many other publications, has shown to be very much wide of the mark for gas-solid flow. Some equations relating total equivalent length to throughput for given pressure drop, for lean phase systems only, are given but these include a factor whose value is vaguely said to be dependent on particle shape, a typical value "for powdered and granular material" being given. No indication is given as to how this value may vary or how its value may be established for any particular product.

Ref. 37, "Gravity Flow of Bulk Solids and Transport of Solids in Suspension", deals with hydraulic and pneumatic transport, with the emphasis on pressure losses in straight pipes; it does give an interesting insight into how the 'Froude number' or 'Dimensionless pipe flow parameter of the Froude form', $c/\sqrt{gd}$, which is used by many authors particularly in continental Europe, may have come about. The Darcy equation for head loss in single phase flow, (overleaf)

$$h_f = \frac{4fl}{d} \cdot \frac{c^2}{2g},$$

is rearranged to give a head gradient, J:-

$$J = \frac{h_f}{l} = \frac{4f}{2} \cdot \frac{c^2}{gd}$$

It is pointed out that if $c/\sqrt{gd}$ and f are both constant, then J will be constant as well - although it is added that this will only be so if Reynolds number is fixed. It does not really explain the significance of this, but the attraction of being able to put the relevant quantities into dimensionless form will be obvious given the success this approach has had in dealing with single phase flow phenomena. Many authors simply call this the Froude number, although it is clearly quite a different sort of quantity from the usual Froude number relating to ship resistance, which resulted from the work of W and RE Froude (father and son) about 1830-1900.

An empirical equation for working out pressure gradient in air -solid and liquid-solid flow is given; this is attributed to Durand of the Societe Grenobloise d'Etudes et d'Applications Hydrauliques, Grenoble, France, and is said to be the result of work on pipes from 1.62 to 28 in. diameter carrying grain sizes of from .08 to 4 in, over a wide range of mixture concentrations. It relates the dimensionless ratio of the 'additional' gradient due to the presence of solids against that due to the fluid only, to two other dimensionless quantities including the 'Froude number' described above. It is pointed out that this is accurate only close to a specific value of the 'Froude number' of 31.2, which ties the permissable value of air velocity to pipe diameter. It is not stated whether the equation is suitable only for lean phase flow, but this seems to be implied in the rest of the chapter.

The published experimental results of several authors, mainly concerned with wheat, are introduced and these plotted on the bases of the dimensionless pressure gradient, 'Froude number' and solids loading ratio. An empirical equation relating the dimensionless pressure drop to 'Froude number' is obtained but said to be restricted to grain sizes over 2mm. Some other test results are cited but no clear conclusion drawn. The question of minimum conveying air velocity is mentioned and some data from various authors plotted.

Bends are mentioned only in passing, using the equivalent length approach - some 'typical values' of 5.5 to 18m for grain in lean phase systems are given, based on only the grain throughout and taking no notice of pipe diameter or air speed. An alternative approach is mentioned, to take the loss as equal to the notional 'dynamic pressure' of the flowing suspension, worked out presumably in a similar way to that outlined in Chapter 3 of this thesis.

Ref. 38, "Gas-Solid Transport", like most others, deals only with lean phase flow. The correlations and models suggested by a number of authors working on losses caused by pipe bends and acceleration regions (where particles are introduced into the pipe) are given, the origins of these range from equivalent length values obtained from single phase flow, through relationships obtained from dimensional analysis, to empirical equations. A small amount of experimental data from one worker are compared with loss values predicted by several of these correlations and models; it seems hardly surprising that the correlation which fits best is the one suggested by the worker who was responsible for the data set. The range of flow conditions for which these may be useful is not addressed, and neither is the question of selecting a suitable pipe size.

"Gas Fluidisation Technology", ref. 41, deals mainly with lean phase flow although dense phase flow gets a short mention. Again some review of the literature is presented although the question of predicting pressure drop is hardly addressed, the main emphasis being on minimum conveying velocity and even this is treated vaguely. Few correlations or models and no experimental data are presented. The question of selecting a suitable system is not addressed.

"Flowing Gas-Solid Suspensions" (ref. 42) is notable for the large number of references cited (over 100 in the chapter relating to pressure drop) compared with the small amount of useful guidance given. A small amount of lean phase data on straight pipe and bend pressure losses from one source (bend losses found by projecting gradients in the adjacent straight pipes to obtain an equivalent step change) is presented, in the form of a plot of a dimensionless friction factor versus "Froude number" again, and some rough correlations proposed but these are not universal. The bend friction factors are based on the loss caused by the bend divided by the length of the bend to obtain a pressure gradient - with the differences in lengths of bends of varying radius, this is not a very useful presentation.

Ref. 43, "Conveyors and Related Equipment", is unusual for being a Russian publication in English. It gives an equation for pressure drop, which appears to be for lean phase flow only although this is not stated, based on air velocity, pipe diameter, solids concentration and pipeline equivalent length (the latter taking into account bends, for which some equivalent length values are given - dependent on subjective descriptions of the material, e.g. "lumpy", "granular", etc.). A coefficient is also introduced, and a graph with one curve is presented to give values of this, from "operational and test data".

A book entitled "Pneumatic Grain Conveying" (ref. 45) came out in 1951 and dealt specifically with lean phase conveying of grain in agricultural applications. This gives an equation for pressure loss incorporating a coefficient, a graph of which (against throughput for a range of air velocities) is given for wheat in a $4\frac{1}{2}$in. pipe. Bends and acceleration lengths are treated using an equivalent length, graphs of which versus throughput are given for $6\frac{1}{2}$, 9 and 12in. pipes.

Finally two purely mathematical approaches will be mentioned. The work of Mwabe (ref. 50) is examined in detail in Appendix A; this used a computer to solve numerically physical models for finite pipeline elements comprising acceleration regions, straight pipe sections and bends (including their associated re-acceleration lengths). This seemed quite satisfactory for lean phase flow, in terms of giving the right order of losses and the right response to pipeline changes, although to obtain accurate predictions would involve introducing experimentally-determined factors. Tsuji (ref. 20) used similar models to Mwabe's for bend effects but empirical expressions for straight pipe losses, strictly for lean phase flow again, and comes up with some rather frightening equations involving all sorts of unknown variables. The bend work is described in more detail in section N.3 below.

N.2.4 Summary of texts

The majority of the books, with the exception of the first two, really gave the designer no guidance as to how to decide on the most appropriate system for his particular application. The prediction of the pressure drop in any given system, which was addressed in all, is only the first stage in the selection of equipment. The approach outlined in ref. 1 (N.2.2 above), of comparing operating costs for various options, was clearly the most appropriate and would be understood by a plant engineer. This was on of the most important contributions of ref. 1.

The issue of how to predict the pressure drop was clearly seen to be a contentious issue, with the wide variety of presentations and correlations, most of which related to the data for only one or two products and a narrow range of conveying conditions. The conclusion appeared to be that different products behaved so differently, that accurate prediction of pressure drop would only be achieved by testing the product for which the plant system is to be designed, to obtain a suitable correlation for use.

The omission of an accurate method of dealing with bend pressure losses from most of the books was clearly very important, given the light which was shed on the relative losses of bends and straight pipes, particularly in lean phase systems, by workers such as Mills (ref. 1) and Westaway (ref. 49). Their work showed that the bends contribute such a large proportion of the total pressure drop in a pipeline, that they must be treated with at least as much accuracy as that used for straight pipes if useful predictions are to result.

Finally it was quite obvious from the lack of agreement on bend equivalent length values, that this area would require a great deal of attention - and that to design for any product would probably mean measuring bend pressure losses directly for that product, as well as straight pipe losses.

N.3 General papers

As mentioned above, a great many references pertaining to specific details of pressure losses in pneumatic conveyors were investigated. The matter of bends having been seen to be a most important one, most effort was concentrated on examining this.

N.3.1 Losses caused by bends, and effect of bend geometry

N.3.1.1 Direct measurement

The direct measurement of pressure losses caused by bends, through the use of pressure tappings on the pipework, was covered in some six papers:-

Mason and Smith in 1973 (ref. 9) presented work which had been carried out by Mason, who later founded the bulk solids handling operation at Thames Polytechnic, for his PhD thesis and is also contained therein. Mason used pipes of 1, 2 and 3in., conveying alumina in very lean phase (solids loading ratios less than 4), and instrumented the pipe with pressure tappings around and downstream of a vertical-up to horizontal bend. Unfortunately his pressure tappings extended only over a length of 32in., and from the plots of pressure versus distance it is clear that flow was by no means fully accelerated in this length. Nevertheless he goes on to find values for the bend losses, simply as the drop in pressure as the fluids flows around the bend, not recognising the major part of the loss downstream of the bend; the data is presented as values of a coefficient times the dynamic pressure of the air only.

First on the scene with any work of good quality were Morikawa et al, ref. 32, (1978) who were interested in the pressure losses in quarter-elliptic bends; they instrumented a 40mm test pipe with pressure tappings for some

8m or so before and after the test bend, and conveyed 1.1mm dia. polyethylene pellets at solids loading ratios of up to 8 with air velocities from 18 to 29m/s. They projected their pressure profiles back not to the position of the apex of the bend, but to inlet and outlet of the bend, for no apparent reason. They expressed the bend loss as the air only loss expected from established models (Ito, ref. 103) plus a coefficient times the dynamic pressure of the air flow, showing a graph of this solids loass coefficient versus solids loading ratio. It is interesting to note that if they had used the dynamic pressure of the flowing suspension instead of the air flow then the loss coefficients would have come out constant instead of proportional to solids loading ratio. Incidentally they found the losses were slightly lower using the quarter-elliptic bend (longer radius at inlet) than for the circular bend.

Next came Park and Zenz, ref. 28, (1980) who used a 102mm bore pipe with a horizontal-to-vertical bend configuration, with pressure tappings at 1.5m intervals, three upstream of the bend and three starting 6m downstream of the bend. They conveyed glass beads through three different types of bend at solids loading ratios of up to 5, obtaining the equivalent step loss caused by the bend. They expressed the losses as a coefficient times the dynamic pressure of the air flow (for the 'air-only' loss), plus a coefficient times the dynamic pressure of the flowing suspension.

Yang et al in ref. 5 (1987) conveyed acrylic powder through a 10cm bore pipe with a long straight section fitted with pressure tappings, a bend being fitted at the end of this straight section; unfortunately the straight section downstream of the bend was similarly instrumented for only 5 metres, and although some interesting pressure profiles were obtained, albeit only for a very narrow range of lean phase conveying conditions (solids loading ratios up to 8), it was quite clear that a steady gradient had not been reached at the end of the instrumented section.

Michaelides and Lai, ref. 33, (also 1987) worked with 180° return bends in a 40mm bore pipe, instrumented with pressure transducers for some 9m upstream and downstream of the bend. They claim that full re-acceleration was achieved some 3m or so downstream of the bend, but the "Typical pressure profile" which they illustrate seems very hard to interpret. The flow used was very lean phase indeed. They expressed their bend losses as a coefficient times the dynamic pressure of the air flow, then broke the coefficient into two parts, one for the air flow and one for the addition of the solids, presenting a correlation for these based on dimensionless groups including the "Froude number", and Reynolds number of the air flow. They compared their results with the 90° bend results of Morikawa et al (ref. 32) and Marcus et al (ref. 29), who did not measure bend losses directly, and found that the measured losses for the 180° bends were less than twice those for the 90° bends, but of the same order, at least for low solids loading ratios (up to about 8). They used five fairly coarse solids (median sizes 0.5 to 5mm) but recorded little difference in results.

Westman, Michaelides and Thomson, ref. 31, (again 1987) used a 4in. pipe with several pressure transducers along 6m straight lengths upstream and downstream of the bend, conveying four mono-sized plastic products (all median diameter approx. 3.5mm) at low solids loading ratios (up to 8). They found considerable agreement with results of the other work mentioned here.

In ref. 30, Klinzing and Mathur made some measurements of pressure losses across a long radius bend, conveying pulverised coal in lean phase. They measured the pressure loss by simply tapping into the pipe at entry to and exit from the bend, thus missing the major portion of the bend pressure drop which occurs downstream of the bend, making this paper useless.

From the above it was notable that all of this work only related to very lean phase conditions with mono-sized products, and in most of the cases the all-important re-acceleration length downstream of the bend was either ignored totally or insufficient length allowed for fully-developed flow to become re-established. However, there was apparently some agreement in the expression of bend losses in terms of a coefficient times a dynamic pressure, but disagreement as to whether to use the density of the air or a notional density of the suspension in the dynamic pressure term.

N.3.1.2 Indirect measurement

Most of the publications dealing with indirect measurement of bend losses were concerned with the effect of bend geometry (i.e. type and radius) except where the work was in the specific context of predicting the total pressure loss along a pipeline; those falling into such a category are examined in detail in section N.2 above, so are not generally included here.

The work of Mills in refs. 1, 11 and 12 has been examined in detail elsewhere in this thesis (App. B). Mills set about trying to determine equivalent length values for bends in pneumatic conveyors by using pipelines of the same length but different numbers of bends, comparing performance characteristics between the two pipelines and applying an accepted correlation between throughput and pipeline length to determine equivalent length values. These equivalent length values did not correlate well with pipeline outlet or mean air velocities but correlated quite well with pipeline inlet air velocities, with equivalent length increasing sharply with air velocity. (Fig. B-3 of App. B).

This strong dependence on air velocity is of course the downfall of using such a correlation because it means that bends towards the end of a pipeline will have a much more significant effect than those towards the beginning, making scaling between test and plant pipelines risky unless they share the same number and distribution of bends and also operate at the same pressure drop as the lines from which the correlations were obtained (in order that the velocity would increase in the same ratio). App. B gives a deeper explanation of this problem.

In the more recent ref. 23, Mills wrote up the results of an undergraduate project which examined the effect of bend type and radius by running an identical pipeline fitted with seven long, then short, radius bends, elbows and blind tees in turn, comparing the pressure loss along the whole pipeline for each bend configuration. Tests covered a wide range of conveying conditions (air velocities 4-45m/s, s.l.r. 30-120), with one product (pulverised fuel ash). He found that the long and short radius bends did not display much difference in pressure drop except at the lowest air velocities, when the short radius (R/d of 3) were slightly better; the elbows (R/d = 1) displayed significantly higher pressure drop (e.g. +10 to +20%) and the blind tees were worse still (+40 to +60%). This was one of the best papers examining bend geometry effects.

Hilbert, in ref. 15 compares pressure drop and service life between bends of different types and geometries. The paper is very sketchy with little hard data presented; although it is not stated, it appears to refer to lean phase flow only. The conclusion is that the short radius bends offer less flow resistance than long radius bends, as expected from other work, but that the blind tee falls somewhere in between, which has been shown by the present study not to be so, at least for the product tested.

Marcus, Hilbert and Klinzing in refs. 16 and 29, present a small amount of data (15 test runs) on the subject of bend pressure losses measured from a short (15m straight pipe and one bend) lean phase system conveying cement (maximum solids loading ratio 23). Although they used an 80mm diameter line instrumented over a 15m length with pressure tappings, and they stated the need to take account of the re-acceleration length downstream of the bend, there are several fundamental flaws in these papers; for example the test bend is after the instrumented section, leading straight into the receiving hopper without any length for the acceleration effect to develop, and in one paper the pressure gradient with just air in the pipeline upstream of the feeder is drawn (on a plot of pressure versus distance) as greater than the gradient downstream of the feeder where the solids are being accelerated. The pressure loss caused by the bend is deduced from the overall pressure drop along the straight pipe and bend together. In these papers the suggestion is made that the short radius elbows (R/d of 2) showed the lowest pressure loss with the blind tees in second place and the long radius bends worst; however, the above-mentioned problem with the test rig and the other obvious flaws make the results highly suspect.

In an Indian paper by Low et al (ref. 17), flow through the branches of tee junctions is examined. A 54mm pipe was used but only very lean phase flow was employed (maximum solids loading ratio 2.4) and only granular products used; pressure tappings were placed very close to the branches and no account taken of the acceleration lengths downstream of the branches.

Bodner (ref. 22) measured bend wear and overall system pressure drop conveying sand and calcine granules in very lean phase (s.l.r. 1.3 to 3) through a 60m line of 35mm i.d. with 7 bends. The bends he used were radiused (R/d of 8, 12, 16 and 24), mitred and 'blinded bends' consisting of a radiused bend with a blinded straight-on section welded on the back. He found the line gave lowest pressure drop with the shorter radius bends, with the blind tees and 'blinded bends' somewhere in the middle and the longer radius bends highest; however, the total line pressure drop when conveying was only about twice that with clean air flowing, so low were the solids loadings.

Solt, in ref. 45, conducted similar tests; his primary interest was clearly bend wear. After first establishing the longer life of blind tees using aluminium bends of several radii, plus blind tees, in a 2in. pipeline he then used a 4in. line, 180m long, conveying cement, using all long radius bends (r/d of 20), all short radius bends (r/d of 1.6), then all blind tees; he repeated this with a 3in pipeline, 72m long, conveying plastic pellets. He presents only average values of his data in the first case and no detailed data at all in the second case, and there is no indication of how many test runs were made. He asserts that both lean and dense phase conveying were covered, yet his line pressure drops and throughputs do not substantiate this. He concludes that short radius bends (r/d of 1.6) and blind tees give similar pressure drop, slightly less than that given by the long radius bends at low air velocities, with the situation reversed at higher velocities.

N.3.2 Straight pipe pressure drop - correlations from experimental work

Several of the references relating to mathematical modelling of pressure drops included experimental data and correlations, but since these are described in section N.3.3 below, these are not detailed again here. Specifically refs. 19,20,21,39,40.

Clark et al (ref. 24, 1952) were amongst the earliest to tackle this in any sort of logical way. They used pressure tappings along straight lengths of 1in. pipe conveying several granular products. They recognised the effect of acceleration regions, although their straight pipe lengths seem hardly long enough to have expected fully-accelerated flow. They measured particle velocities indirectly, but rather elegantly, by shutting off a section of pipe during conveying and measuring the quantity of product trapped. They found a good correlation between pressure drop and the slip velocity between air and particles, but this is of little use practically.

Another early group were Mehta et al (ref. 26, 1957) who carried out a very similar exercise but with even smaller pipes ($\frac{1}{2}$in.) with glass beads in very lean phase and ridiculously short test sections (e.g 1.5m or less). Similar results to those referenced above were obtained, and the same comments apply.

Mendies et al, (ref. 18), made some measurements of pressure profiles along a 53mm bore pipe 11m long conveying three granular products in very lean phase; this was really as a by-product of attempting to measure the velocity of the particles using electrostatic sensors. The one profile shown indicates that they barely achieved fully-accelerated flow, and no other data is given.

In conclusion it was apparent that the only attempts directly to measure pressure drop along a straight pipe section had been made with very lean phase flow, mostly with granular products only and generally with small pipes (38mm or less), usually with insufficient length allowed for fully developed flow to become established. Little in the way of correlations had been established.

N.3.3 Mathematical models and dimensional analysis for bend and straight pipe losses

The work of Mwabe, ref. 50, for lean phase flow only, seems to be by far the most comprehensive work in this field, and has been analysed in detail in Appendix A. Other attempts are mentioned here.

Rizk, (ref. 21) divides the line up into acceleration, bend and straight pipe regions and gives empirical expressions for the losses in each; it is not stated whether these are suitable for lean phase only, but since the only products referred to are plastics granules the inference is that this is the case. A number of factors are involved, some of which can be calculated from particle size etc., but key ones can only be found experimentally.

Tsuji (ref. 20) presented mathematical models, based on the analysis of physical models, for calculating pressure losses in lean-phase suspension flow. The models used are somewhat similar to those used by Mwabe, examined in Appendix A. The bend pressure loss was based on the re-acceleration of particles slowed down by contact with the bend wall; to use this model, the velocity of the solids at exit from the bend must be known, and he suggests the calculation of this based on the work of Wiedner (published in German in 1955) which simply uses a coefficient of friction between particle and pipe wall under the effect of centrifugal force. This coefficient of friction could of course only be determined by measuring pressure drop caused by a real bend and working back; it is not stated whether it would be expected to be the same for the same product under all conveying conditions, but of course the likelihood would seem not, making it necessary to test a range of conditions to find the variation, in which case the actual pressure drop could simply be stored for use in design.

Haag, in ref. 25, also uses a similar method to Mwabe to predict the reduction in velocity of particles flowing through a pipe bend, in relation to the increasing likelihood of blockage as particle velocities reduce. He takes friction coefficient values similar to normal dry sliding friction, e.g. 0.36 between grain and steel, and obtains velocity losses of the order of 40%. His equations indicate little effect of bend radius with bends in the horizontal plane, with horizontal to vertically-up bends suffering lower losses with smaller radii and vertically-up to horizontal bends giving lower losses altogether. No experimental data is presented.

In ref. 34, Ikemori and Munakata also carry out a similar analysis and make it more complex to account for the uncertainty of values for the coefficient of friction; this paper is extremely complex and difficult to follow, the eventual relationship obtained being one between an additional friction factor due to solids in the bend, and ratio of solids velocity to air velocity in the straight pipes before and after the bend. Some experiments were done with granular products, presumably in lean phase although this is not stated, which appeared to give good agreement. It is difficult to see how this work might be applied practically.

Kovacs, ref. 46, also uses a similar analysis. His presentation is easier to follow and use, but is not backed by any experimental data.

Molerus, in ref. 19, presents some mathematical models involving many dimensionless groups for conveying in a state where there are strands of product flowing along the bottom of a straight pipe. The resulting equations for pressure drop are of considerable complexity; some quite good correlation with measured results is apparent for a limited range of granular products.

Rose, working first with Barnacle in 1957 (ref. 40) then with Duckworth in 1969 (ref. 39), used dimensional analysis to assist in obtaining correlations to predict pressure losses in very lean phase flow of granular solids; experimental data was obtained from a short 32mm bore straight pipe conveying several products. With pressure tappings along the pipe, the acceleration pressure drop showed up clearly and this was recognised as distinct from the steady-state pressure loss with fully-accelerated flow. A great mass of dimensionless groups emerged from the analyses and several correlations were established, but the scale of rig and limited range of products and conveying conditions would not give confidence in using these even if they could be put into a convenient form.

Mason, in refs. 10 and 60, presents some work using a proprietary software computational fluid mechanics package; this work is described in Appendix A. He found it possible to get the mathematical model to exhibit similar behaviour to the 'slugging' observed in actual dense phase conveying, but this work appears to be of little direct use to practicing engineers in the field. He seems seriously to misunderstand some of the methods used for single phase flow, from which he obtains values for certain quantities which he misuses in his analysis.

In ref. 48, Edwards describes a computer model based on applying conventional equations of motion to individual particles flowing in a pipeline. Up to 1000 particles are used, with drag forces on particles and collisions of the particles with each other and the pipe wall modelled, the trajectories of individual particles being calculated between collisions. The one thousand particles employed make up less than a gramme of product, and even for this considerable computing power is necessary. It seems doubtful whether this work is of any value in predicting pressure drop in a real system, especially given the difficulty in obtaining some of the quantities necessary to the calculations - let alone the computing power necessary to bring this up to a realistic scale, even for a very lean phase system. No calculated results are presented.

Another computer method is presented by Latincsics (ref. 14) who predicts pressure drop in lean phase flow along straight pipe simply by multiplying the 'air only' pressure drop (from Darcy) by an empirical coefficient proportional to solids loading ratio, values for which he obtained from the data of other authors. He uses an equation of unexplained origin for acceleration pressure drops and ignores bends altogether.

N.4 Conclusions

The most obvious conclusion from the above survey was the confusion and lack of cohesion in the strategies used for prediction of pressure drop along pneumatic conveying pipelines. Also it was obvious that few authors had learnt the lessons of history which were available to them through the literature; this may perhaps be due to the fact that few workers seem to recognise the value of understanding how the science in which they work came into being, and the reasons for its having developed in the way which it has.

This indicated that the problem would not easily be solved; some of the approaches examined clearly had merit, but it seemed that in many cases the authors were just floating ideas in the hope that perhaps someone may pick them up and develop them, which in most cases never happened of course. Even those approaches which appeared to have been developed to some degree of sophistication seemed to have been ignored by later workers who usually decided to go their own way.

It was however possible to discern the emergence of two distinct strategies, firstly the use of mathematical (or numerical) analysis, possibly involving the use of empirical coefficients, and secondly the use of a pilot scale test pipeline with results being scaled to the plant pipeline using empirical correlations. The former was undoubtedly the source of most papers, and indeed PhD's, but seemed to be mostly only related to outline research with scant regard for the solution of the problems of practicing engineers; the latter approach seemed to offer more likelihood of success in designing real systems, not only because it was easy to use for this purpose but also because of the fairly direct use of data obtained from the actual product to be conveyed.

Direct measurement of pressure drop along straight pipes and that caused by bends seemed not to have been seriously attempted, other than for very lean phase flow of granular products, a very restricted range compared with that employed in real plant systems.

APPENDIX O

NUMERICAL COMPARISON WITH THE DATA OF OTHER AUTHORS

O.1 Introduction

In order to provide a complete picture of where the data and correlations mentioned in this thesis fit in when compared with the work of other authors, the work contained in several of the references has been re-presented in the forms used in this thesis, to enable a direct comparison on a numerical basis. This has been possible only with a limited number of the references, where the data given was sufficiently detailed and in a suitable form to enable this to be done; however, it gives a good indication of some of the similarities and differences. This work falls naturally into two categories, namely (i) bend pressure drop and (ii) straight pipe pressure gradient.

O.2 Comparison for Straight Pipe Pressure Gradient

There were found to be three sources of data and correlations which were directly comparable with the work in this thesis, namely the EEUA Handbook, that by Richardson and McLeman, and that of Rose and Duckworth.

O.2.1 The EEUA Handbook

Since the EEUA Handbook, ref. 35, gave the most comprehensive set of data available for pressure drops along straight pipes, this was approached first.

This gives an equation for pressure drop along straight pipe,

$$\Delta p = \frac{F_2 L . V^2 . \rho}{D \quad 2g \quad 144}$$

where L is pipe length, D pipe diameter, g acceleration due to gravity, V is air velocity and ρ is suspension density. The 144 in the denominator arises because whilst pipe length L is in ft and suspension density is in lb/ft^3, pipe diameter D is in inches; thus the 144 is only to overcome this inconsistency of units and would not be needed if pipe diameter were put in in ft. The suspension density is calculated in the same way as recommended in this thesis, i.e. mass flow rate of product divided by volume flow rate of air.

F_2 is a coefficient, and it is not hard to recognise that this equation is simply the usual Darcy equation as used for single-phase flow, with the usual fluid friction factor 'f' removed and replaced with the solids loss coefficient F_2. A graph of values of F_2 against superficial air velocity is given for a number of conveyed products, as reproduced below:-

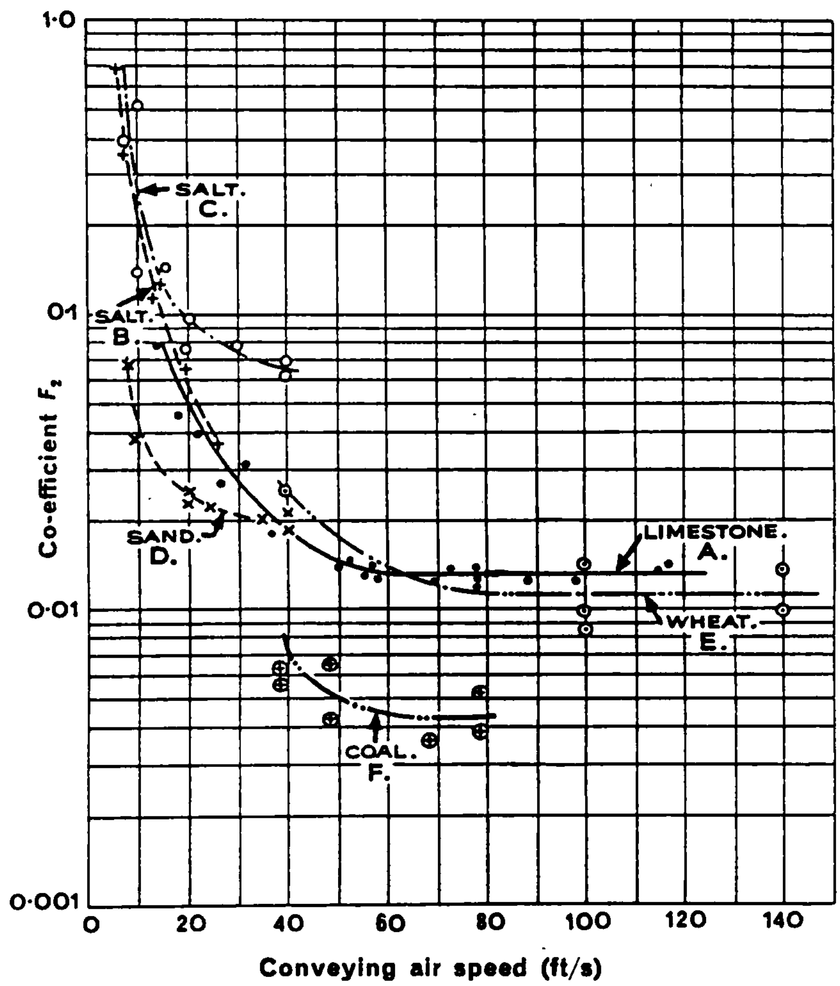


FIG. 49. Values of the Co-efficient F_2 Plotted Against Speed of Conveying Air.

Fig. O-1

Graph of solids friction coefficient F_2 against superficial air velocity, from EEUA Handbook

It is interesting to note that above about 12 to 18 m/s the value of F_2 is practically constant, whereas below this it increases significantly; this is to be expected as this is the usual velocity range which separates suspension from non-suspension flow.

In this graph, the curve for each product comes only from one pipe size; in fact they are all from either 1.75 or 2in. pipes except for the curve marked F, for coal from the data of Richardson and McLeman (which will be considered separately later in this Appendix); that was from a 1 in. pipe.

Although on the face of it this equation and graph appears to be a useful system of data storage, there is a fundamental flaw; all of the data save for one product comes from such a narrow range of pipe sizes, that it was not possible for the authors to assess the effect of changing pipe diameter. The D in the denominator of the equation, which has simply been allowed to remain from Darcy on which this equation is based, may or may not be appropriate. It implies that pressure drop is inversely proportional to pipe diameter for conditions of constant suspension density and air velocity; that is in conflict with the results recorded in this thesis, which indicate little effect of pipe diamater on pressure drop, for constant suspension density and air velocity.

It was felt that this misleading inclusion of pipe diameter in the equation might be responsible for the considerable discrepancy between the data of Richardson & McLeman on the diagram, curve F, and the other curves; therefore the data on the diagram was re-analysed with new values of solids loss coefficient calculated for an equation with D deleted from the denominator (n.b. making the equation non-homogeneous, thus giving units to the constant) to see whether the data would fall closer with the spurious effect of 1/D eliminated.

Taking an example along the horizontal part of the curves, the loss coefficient F_2 is 0.013 for limestone in the 2in. pipe as against 0.004 for the coal of Richardson and McLeman in the 1in. pipe; i.e. a factor of 3.25 different. Now re-calculating with the D struck out of the denominator of the equation, the coefficients become 0.079 and 0.048 respectively, a difference of just a factor of 1.6 which seems much more likely; clearly it would be unreasonable to expect there to be no difference, bearing in mind that different products are being conveyed.

Thus it is clear that the use of the pipe diameter in the denominator of the equation is thoroughly misleading, left over as it is from the Darcy equation and contradicting all evidence for the effect of pipe diameter in gas-solid flow of any practical suspension density. The effect of this would be quite dramatic if using the curves to predict pressure drop in pipes significantly larger that those on which the curves were based; e.g. if applying the data to a 6in. pipe then the actual pressure drop would be under-predicted by a factor of 3, and for a 10in. pipe by a factor of 5.

Going on to make a comparison of the data from the handbook against the data in this thesis, the raw data from which the curves were drawn were back-calculated, taking the friction coefficient and the equation and taking account of the actual pipe diameter used. This was done for three products, namely limestone of top size 3mm, wheat of particle size 4mm, and salt of range 120 to 420 micron. A graph of pressure gradient versus suspension density, for ranges of superficial air velocity, was drawn up for direct comparison with fig. K-25 of this thesis which was the same graph for wheat flour. The comparison is shown overleaf:-

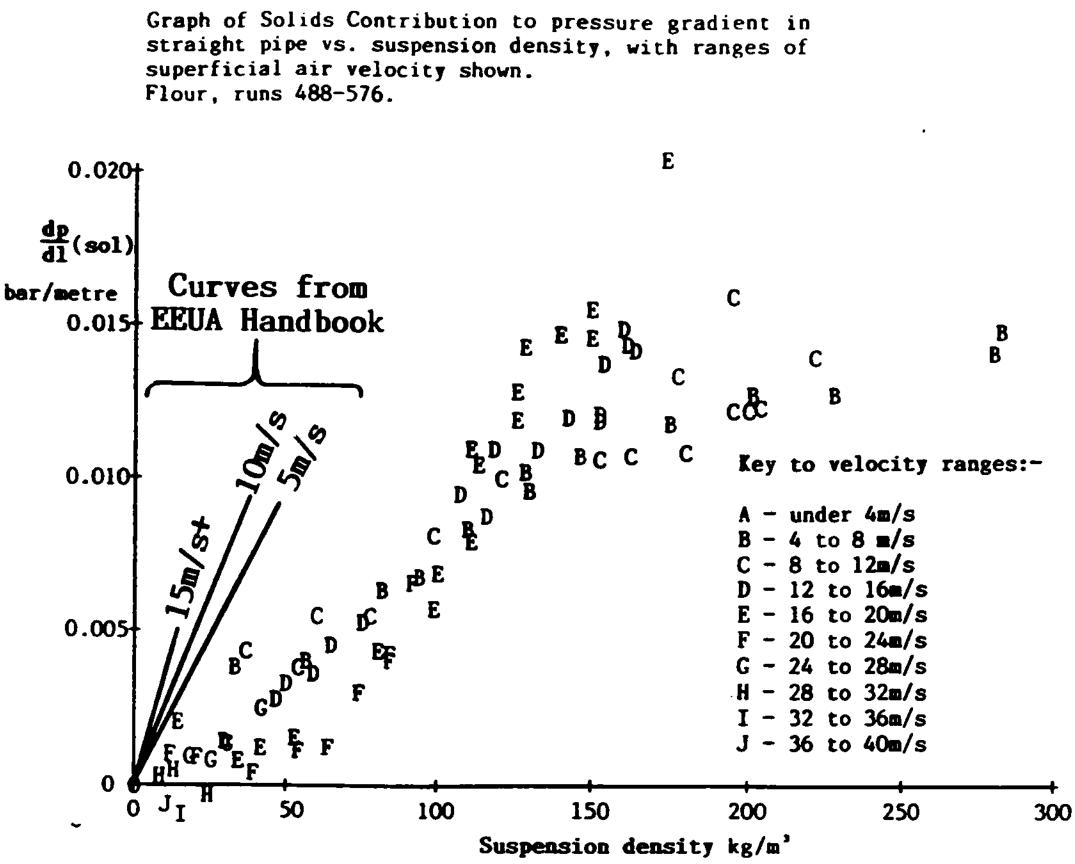


Fig. O-2

Comparison of data from EEUA Handbook for a range of products, against data from this thesis for wheat flour

It is interesting to note that the pattern of data is somewhat similar, although the range of data in the EEUA handbook is rather more limited. The data from the EEUA handbook gives pressure gradient values a little higher, although some discrepancy between different products should be expected; also as will be mentioned later, that source tended to under-predict the pressure losses caused by bends, so these effects would tend to offset one another.

Whereas the above comparison was against the data for flour in this thesis, the same comparison could have been made against the data for polyethylene pellets; this can be found in fig. K-29, and it will be seen that the pressure gradients there were lower still.

Generally speaking the comparisons above were found to be very encouraging, probably as close as could be expected bearing in mind the different products being conveyed.

O.2.2 Richardson and McLeman

The correlations developed by Richardson and McLeman (ref. 59) were fairly complex, involving the terminal falling velocity of the particles, so first the data was plotted on the same scales as used for the data in this thesis.

The results when plotted on the format of graph used for the flour (fig. K-25) fall quite close in value, although the data of R & M tend to show a trend for the pressure gradient to increase with increasing air velocity (at constant suspension density) whereas the opposite is true of the flour:-

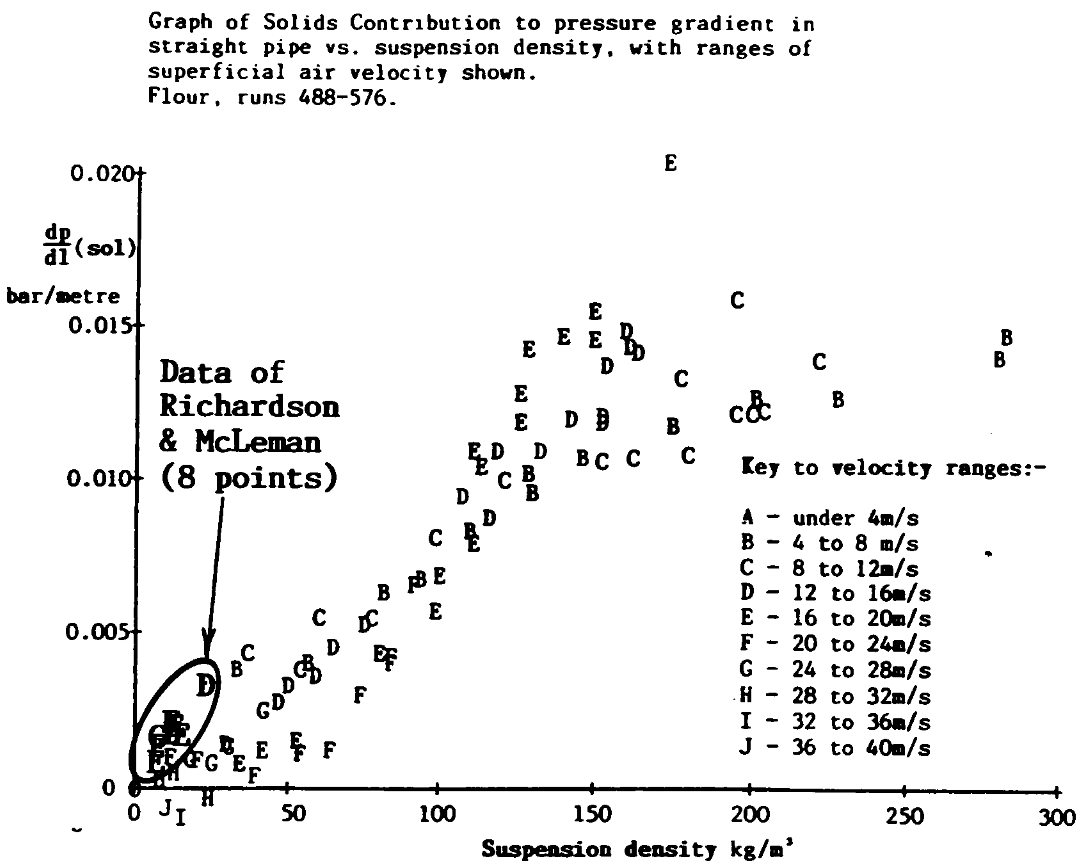


Fig. O-3

Data of Richardson and McLeman plotted on graph used in this thesis for flour (fig. K-25)

The relatively restricted range of the R & M data is apparent, but the similarity is evident none the less.

When plotted on the format of graph used for polyethylene pellets the pattern of the data is more similar, but the values are considerably higher, by a factor of about 2:1, as shown below:-

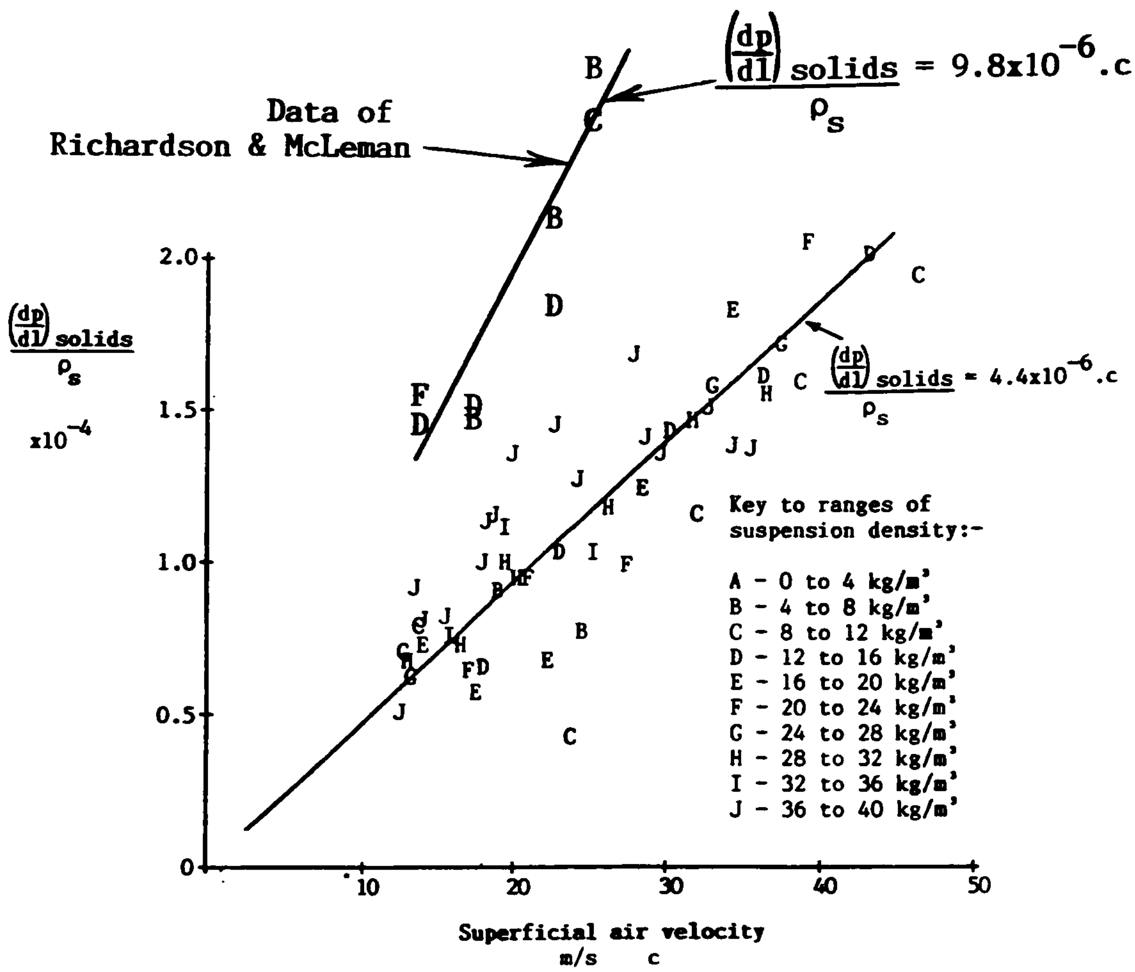


Fig. O-4

Data of Richardson and McLeman plotted on graph used in this thesis for polyethylene pellets (fig. K-30)

It is probably not surprising that the pattern of the data from the coal of Richardson and McLeman bears more resemblance to that from the polyethylene pellets than the flour, in view of the fact that the coal was graded to a narrow size range around 500 micron, so was more like a granular product as against the powdered nature of the flour. The difference in values, however, could be explained by any number of factors such as the difference in friction between coal and metal as compared with polyethylene and metal, or the use of a rather small pipeline of just 1 in. as against 2, 3 and 4in. Since the values are not dissimilar to those from the flour, however, the evidence would suggest that it is largely caused by differences in the products.

Comparing correlations, those of Richardson and McLeman involved the terminal falling velocity in free air in the equations. Whilst this may have been relatively easy to measure for the granular coals of narrow size ranges which they tested, such a measurement is not easily undertaken for a product such as the flour, whose particle sizes are much smaller and, more significantly, have a wide distribution, as do most products which are handled commercially. The difficulty with such a product is that the particles of differing sizes will clearly have differing terminal free falling velocities, so even if they could be measured separately then there would be many possibilities as to what value to take, whether based on a mass-median, a mean value, or a weighted value of some kind. On the other hand, however, being fine the terminal free falling velocities of the flour particles would be much lower than for the granular coal; so the effect of this velocity being small, compared with the conveying air velocity, was examined in the context of the correlations given.

Where the terminal free falling velocity is small compared with the conveying air velocity, the correlation of Richardson and McLeman reduces to:-

Solids contribution to pressure gradient = constant x suspension density

Comparing with the correlations derived in Appendix K of this thesis, the same form of equation was found to be appropriate for the flour at 8 m/s. At higher velocities, though, the suspension density was subject to an increasing power law, e.g. at 16 m/s it was squared in the above equation. The range of the Richardson and McLeman work was much more restricted, however, as discussed below.

Overall therefore, the comparison between data and correlations from Richardson and McLeman and that from this thesis was quite encouraging. Certainly the numerical values of the data were in very much the same range where they overlapped, and there was some similarity in the correlations evolved to represent them. It should be pointed out that the overlap of ranges between the two data sets is relatively restricted, in that the range of suspension densities used by Richardson and McLeman was limited to 23 kg/m^3 and the range of air velocities limited to 25 m/s, whereas the data and correlations in this thesis covered up to 300 kg/m^3 and 50 m/s respectively. In view of this, the comparison was particularly encouraging.

O.2.3 Rose and Duckworth

The data of Rose and Duckworth has been frequently quoted in papers dealing with mathematical modelling of suspension conveying, so is worth comparing with the current data.

This work was undertaken using mustard seed which was a mono-sized product of spherical particles, 2mm diameter; the pipe used was 1.25in. nominal bore. The range of their data was very limited by the apparatus used, so they only achieved very lean phase conveying. Their data is plotted below for comparison on the graph used for flour in this thesis:-

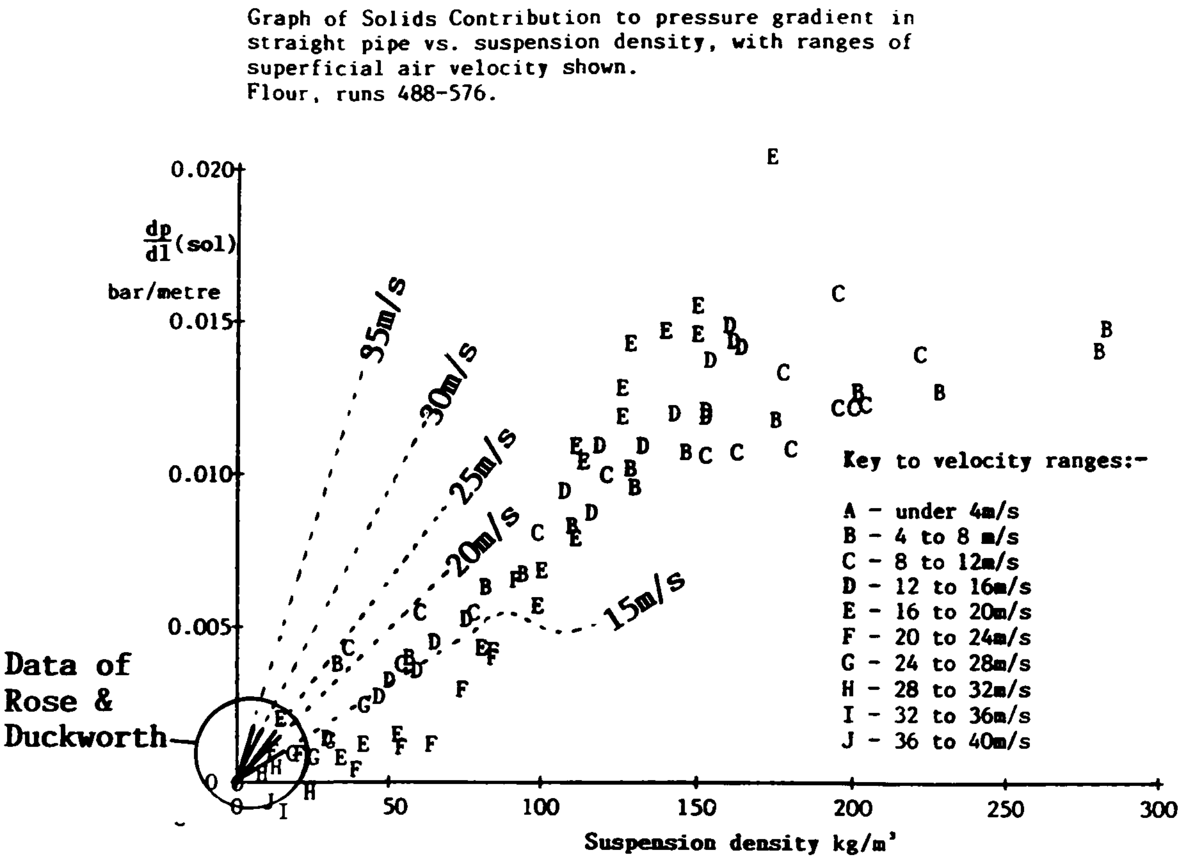


Fig. O-5

The data of Rose and Duckworth plotted onto the graph used in this thesis for flour

The very limited range is the most noticeable feature of this. Because of this, and the fact that the data for the flour is subject to a good deal of scatter at such lean phase conditions (mainly, it is felt, because these were such low pressure drops that they were on the limit of the instrumentation, and almost lost in the air-only pressure drop), it is hard to draw any useful comparison.

When compared against the data for polyethylene pellets, there is more to see:-

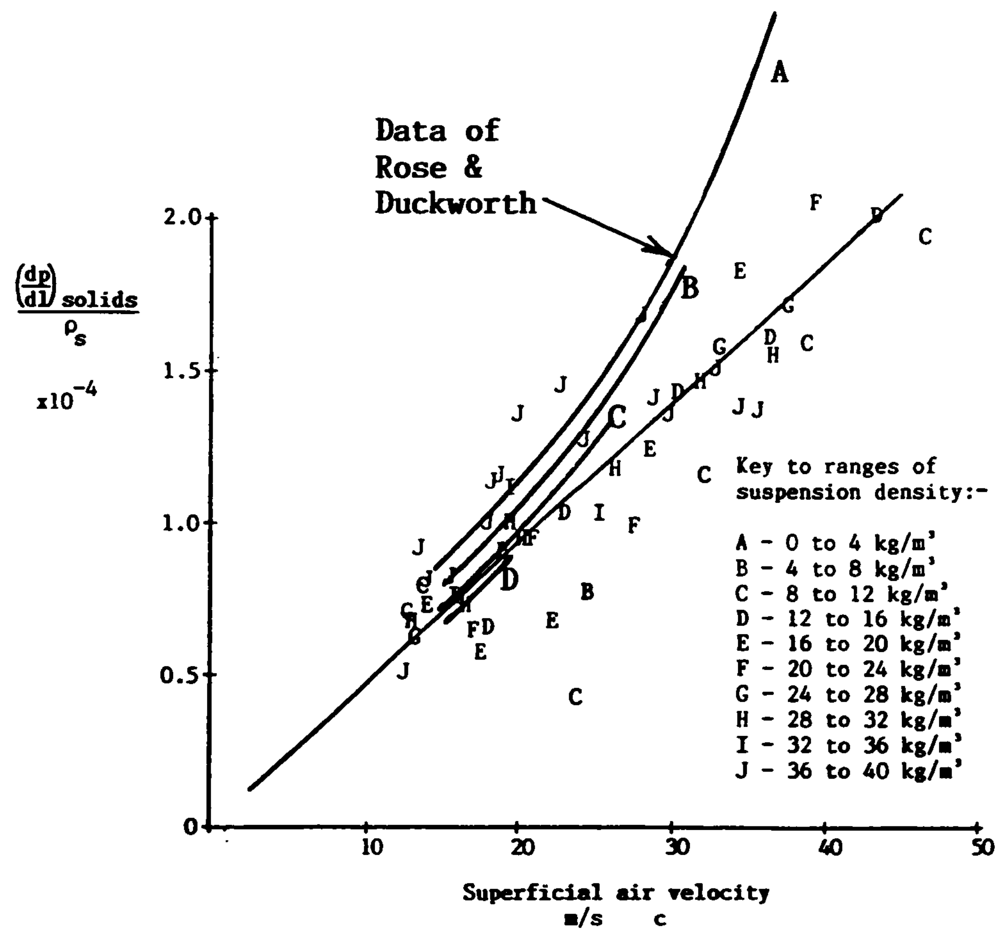


Fig. O-6

The data of Rose and Duckworth plotted onto the graph used in this thesis for polyethylene pellets

The shape of the data is very much the same, but a law slightly higher then the direct proportion used for the polyethylene pellets (and for the granular coal, above) would appear appropriate; approximately a 1.5 power law would be nearer. Again the actual values are significantly higher, although this may not be surprising again in the light of the difference in products.

It is worth pointing out that although the range of air velocities used by Rose and Duckworth was comparable to the present study, their suspension densities were generally very much lower so that there is in fact virtually no overlap of conditions. This may well be expected to lead to some deviation between the data sets. Also, at such lean phase conditions the 'solids contribution' to the pressure drop was fairly small in comparison with that caused by air only, which was of the order of 1.5 to 4 times as great. Consequently under such conditions it is prediction of the 'air-only' portion which is more critical. At the same time, it is also worth noting that conveying under such lean phase conditions is only of academic interest to most commercial applications because it is a most uneconomical mode of flow to employ.

O.3 Comparison for bend losses

Two pieces of work were found to be directly comparable with the work in this thesis on bend losses, namely the EEUA handbook (again) and some work by Mills and Mason.

O.3.1 The EEUA Handbook

This (ref.35) gave as much information as any source, and it was presented in the form of bend loss coefficients, entirely interchangeable in form with the coefficients used in this thesis. The table below was given:-

Ratio of Bend Radius to Pipe Diameter r/d	Loss Coefficient K
2	1.5
4	0.75
6 or more	0.5

The aspect of this data which was most striking was the apparent steadying-out of loss coefficient above an r/d ratio of 6; the comparative work between losses caused by bends of differing radius in this thesis gave similar results. For example, fig. K-19 compares relative bend losses for bends of differing r/d ratio and this demonstrates clearly that a value in the range 3 to 6 is the break-point beyond which longer bends give no reduction in loss. Relatively speaking, though, the increase in losses with bends of smaller r/d ratio was more marked in the above table than in the work of this thesis (again fig. K-19).

As to the actual values, the work in this thesis demonstrates (figs. K-18, K-23) that the coefficients should not be taken as constant because they vary with flow conditions, in different ways with different products, and also that the above values are rather optimistic. A glance at fig. K-18 will show loss coefficients of around 2 at low suspension densities (say less than 60 kg/m^3) decreasing towards 0.5 at very high suspension densities (above 150 kg/m^3); this is for the bends of r/d greater than 6. This is for flour, whereas for the polyethylene pellets the values start around 3 at low velocities (5 m/s) decreasing to about 0.5 at high velocities (above 30 m/s).

The above bend loss coefficients, therefore, seem rather low. However, against this must be offset the fact that the Handbook tended to over-predict the pressure gradient in straight pipe, as compared with the data for the products used in the current project. Thus for prediction of overall pressure drop along a pipeline with bends and straights, the EEUA handbook may not be as much in error as may be expected from the above comparisons, with the notable exception of the effect of pipe bore; as mentioned in section O.2.1 above, the equation given is fundamentally in error in respect of this so would give disastrous results if used to predict losses along pipes of any diameter other than 1.75 to 2in. bore. It is a great shame that the misleading 1/D term has become incorporated in the equation given, as in all other respects it is probably the best and most comprehensive approach previously published.

O.2.2 Mills and Mason

In ref. 23, "The Influence of Bend Geometry on Pressure Drop in Pneumatic Conveying System Pipelines", Mills and Mason reported using bends of four different geometries, namely blind tees, short radius malleable elbow fittings (r/d of 1), short radius bends (r/d of 3) and long radius bends (r/d of 12), i.e. exactly the same geometries of bends as used in the current project. They used a pipeline of 2in. nominal bore, 50m long with 11 bends; 4 of these bends remained always long radius, whereas the other 7 were interchanged in matching sets, between the four geometries mentioned above. A full set of conveying characteristics was obtained for the pipeline with each set of bends, over a range from high velocity, low suspension density, lean phase flow down to low velocity, high suspension density dense phase flow. One product was conveyed, pulverised fuel ash which was a very fine, powdered product.

The effect of changing the pipeline bends was examined from two viewpoints, namely (i) the effect on the throughput capacity of the line for the same total pressure drop, and (ii) the effect on the total pressure drop for the same throughput. Clearly the second comparison was more useful in the context of the current work.

It should of course be understood that Mills and Mason were measuring the total line pressure drop, including that caused by the straight lengths of pipe and that caused by the 4 bends out of 11 which were not changed; therefore the results which they obtained would be expected to show very much less difference than if the bends were examined in isolation.

Mills and Mason plotted the relative pressure drop values for the entire line using the values from one set of bends as a reference. Not surprisingly, given that they covered a very wide range of conveying conditions indeed, there was some variation across differing conditions, but taking that into account their data has been re-plotted into the same format of presentation used to compare bends in this thesis, i.e. that of fig. K-19. The results, for comparison, are shown overleaf:-

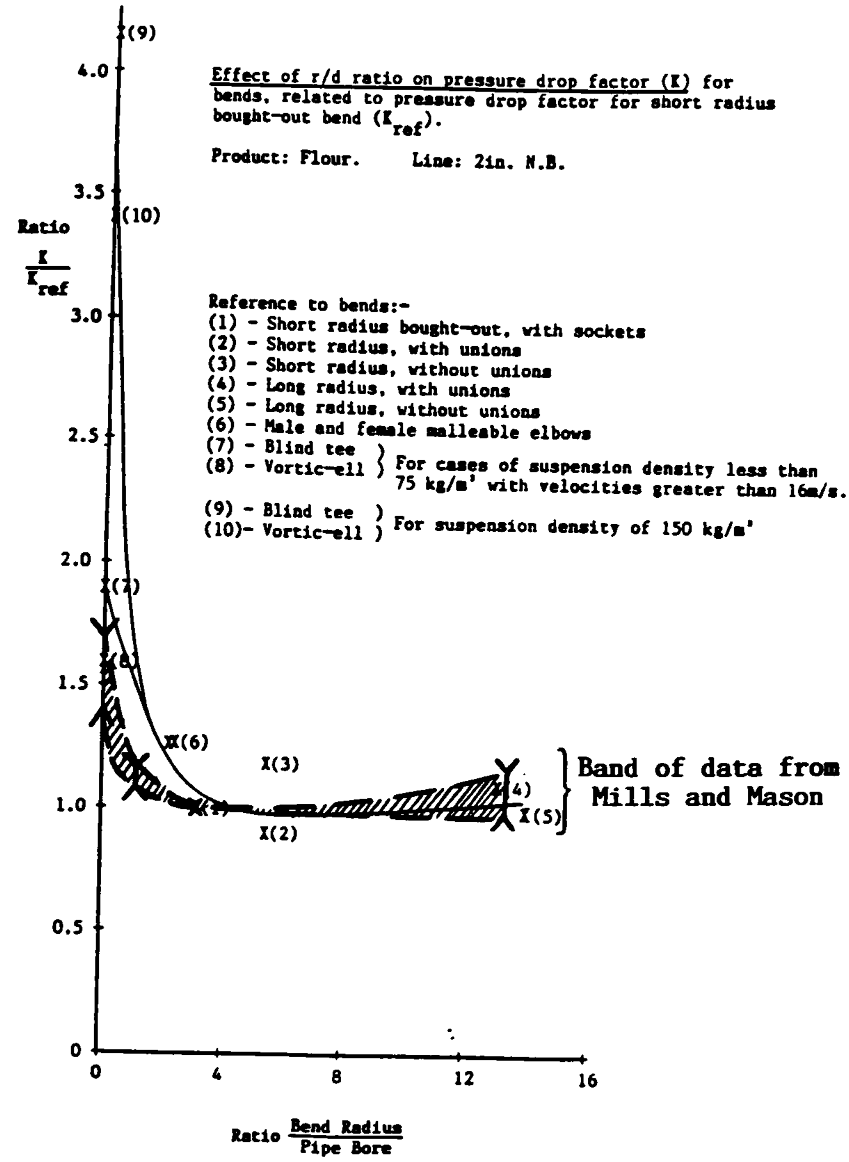


Fig. O-7

The relative bend loss data of Mills and Mason, superimposed on fig. K-19 of this thesis for comparison

The graphs are strikingly similar, bearing in mind that the Mills and Mason data is from the entire pipeline so would be expected to show a less marked difference in loss with changing r/d ratio.

This comparison was most encouraging, showing that the results obtained for the effect of bend r/d ratio are not unique to flour but hold for at least one other powdered product as well. It is unfortunate that no comparable data is available for granular products.

O.3.3 Westman, Michaelides and Thomson

These three authors, publishing in 1987, did some potentially interesting work using a 4in. nominal bore pipe, conveying four types of mono-sized polymeric pellets of marginally different characteristics in lean phase only. They measured pressures at tappings along straight pipes of 7m length before and after a test bend (interchangeable to give bend r/d ratios of 1.5, 5 or 12) and fitted tangents to these to measure bend pressure drop.

Their approach was in principle very similar to that used in the current project. There were, however, some very significant detail differences. Firstly they only used very lean phase flow, with suspension densities below 8. Secondly their acceleration lengths were rather short so that it is questionable whether they fitted their tangents over regions of truly fully-developed flow. Thirdly they used differential transducers connected between successive tappings along the pipe rather than having all the transducers measuring against a common reference, thus incorporating cumulative errors into the pressure gradients.

Fourthly, and probably most significantly, they dealt with their tangents on the pressure-distance graphs in a rather odd way; rather than projecting the tangents to the position of the intersection of the centre-lines of the straight pipes, to give a resulting drop in pressure additional to that which would have occurred if the pipe had continued in a straight line over the same distance, they projected them only to the bend inlet and outlet positions, thus incorporating some pressure drop which is not caused purely by the introduction of the bend but would have been there anyway even if the bend had been absent. The result of this is that the effect of shortening the lengths of the adjacent straight pipes with the longer bends is effectively not accounted for and so the true effect of the bends on the overall system pressure drop cannot be isolated. This was most significant given that in the sample data they quote, only about one third of the pressure drop which they attribute to the bend should really be attributed to this cause, which gives a very misleading impression not only of the actual bend loss, but also when comparing the results of tests with different bend radii.

They quote their bend losses in terms of bend loss coefficients, similar to those used in this thesis, and which appear to be worked out in much the same way although there is not sufficient detail in the paper to be certain of this. Unfortunately, because of the rather odd way of determining the pressure drop which they hold to be attributable to the bend (as explained above), and because they do not give sufficient data to isolate properly either the true pressure drop caused by the bend or the true bend loss coefficients, it has not been possible to make a direct numerical comparison of like quantities between their work and this.

The results which they do give are very sketchy; they draw graphs of loss coefficient versus solids loading for the very short elbow (r/d = 1.5) and long radius (r/d = 12) bends, without reference to the effect of air velocity, and these show only slightly lower coefficients for the long bend than for the very short elbow; the true comparison between the effects of the two bends is distorted by their method of determining bend pressure loss, i.e. because it does not take account of the shortening of the straight pipes when using the longer bend, and if this were taken account of then the loss coefficients for the long bend would almost certainly be significantly lower, more in line with the results in this thesis which showed significantly lower coefficients for long radius bends than for very short elbows. It is also unfortunate that they make no reference to any results from the medium radius bend which they claim to have tested, yet they go on to draw the conclusion that "the bend loss coefficient is inversely proportional to the bend curvature", which statement is most definitely not supported by their graphical results.

O.3.4 Rose and Duckworth

It is worthwhile finally to return to the work of Rose and Duckworth (ref. 39) for a brief examination of their comments on bend losses. They did no measurements in this respect, but suggested that a bend pressure loss may be approximated in a conservative way by considering all of the particles to be brought to rest at the bend, and the resulting pressure drop caused by re-acceleration calculated according to their method for calculating an acceleration pressure drop. In the example they give using their methods of calculation to find pressure loss along a particular pipeline, this yields a bend loss coefficient of 0.8 which seems rather on the low side in comparison with the measurements taken in this project for lean phase flow (see figs. K-18 and K-23 for a comparison) although not altogether unreasonable.

O.4 Conclusions

Comparing the two aspects of pressure drop, it is clear that the work available on straight pipe losses is much more detailed than that on bend losses.

Taking the work in the EEUA handbook first, this covers the widest range of both conveying conditions and products, and aside from the fact that the equation used has a fundamental flaw in including the pipe diameter in the denominator, there is reasonable agreement of the pattern of data with that in the present project. The actual values there are, however, a good deal higher, by as much as a factor of two. This is offset, however, by the fact that the bend losses given there are rather low in comparison, by a factor of again about two. Hence calculating overall line losses using the EEUA handbook would probably not be as much in error as might otherwise be expected provided that the line has a reasonable combination of bends and straights. The pipe diameter effect, however, would result in very serious under-prediction of pressure drop if this was used to design pipelines of any diameter larger than 2in. bore. Although the range of data was the widest found in any publication, it was still quite restricted compared with the work in this thesis.

Taking the work of Richardson and McLeman on straight pipe gradients next, their data for granular coal bore reasonable resemblance to that from polyethylene pellets in this project, although again the values were high by a factor of about two. This could easily have been caused by the different products, or by their pipe being rather small at just 1in. bore.

As far as the work of Rose and Duckworth is concerned, any useful comparison is very limited by the restricted range over which they conveyed their mustard seed. Again the results are more akin to those for the pellets from this project. Their suggestion of how to deal with bends is not substantiated in any way.

Taking the work of Mills and Mason on the effect of bend geometry, it is interesting to see that their data is very much comparable in shape with that from the present project even though their results were obtained using very different methods of measurement. The magnitude of the differences in losses between their overall pipelines containing bends of different geometries was much smaller, as might be expected from the fact that their total losses always included those for the straight pipes and several bends which were not changed.

The work of Westman, Michaelides and Thomson on bend losses seems so badly flawed that it is hard to set any significant store by their results or conclusions, although it could be interpreted as supplying some circumstantial evidence to back up the relative losses between long radius bends and very short elbows as found in this project.

Overall, a number of points are worthy of consideration:-

(i) most of the work published has been related to a very narrow range of conveying conditions compared with the wide range used in this project. It is felt that this may partly be the result of a lack of suitable facilites for obtaining such wide ranging conditions, on the parts of most workers in the field; to obtain such conditions requires equipment of full industrial scale which can mostly be afforded only by organisations having manufacturing interests, in which the drive to undertake research of this type is usually secondary to other requirements. However, there is some suspicion that it may also be due in part to a natural reluctance to tackle the difficult area of predicting losses in dense phase, non-suspension flow at a time when the current level of understanding of, and availability of adequate physical models for, the mechanisms involved in even the much simpler case of suspension flow, is very poor - i.e. perhaps a fear of 'trying to run before one can walk'.

(ii) As a result of this, it has proved impossible to make any comprehensive comparisons between the work in this thesis and any other, rather limiting the scope of what could be achieved. The comparisons which have been drawn indicate quite good general agreement as to both the pattern of the data and the order of magnitude of the losses, as far as the data goes, but the actual values of solids contribution to pressure gradients in straight pipes tend to be rather lower in this thesis than in other published data. No particular explanation springs to mind to account for this, other than the differences in the products conveyed; it seems not implausible that polyethylene pellets may indeed give significantly lower pressure drop than, say, crushed limestone, coal, or mustard seed for example.

(iii) It is further apparent that most of the published work deals with granular solids rather than powders, so it is not surprising that better agreement has been noted against the data from polyethylene pellets than that from flour in this work.

(iv) The comparison for bend losses has shown discrepancies in the reverse direction, with the previously published work showing losses very much on the low side compared with the current work. It is perhaps more important that the general approaches used in the past have been shown to be very inadequate in taking a constant bend loss coefficient for all materials and conditions; the current work has shown that this is very far from being accurate and it is here, in particular, that it is felt significant progress has been made.

(v) As has been mentioned above, the fact that other sources appear to over-predict straight pipe pressure gradients in combination with under-predicting bend losses, suggests that using these sources would not necessarily result in such a serious error as might otherwise be the case, provided the system under consideration has a reasonable balance between bends and straight sections. What constitutes 'reasonable' in this context cannot be quantified.

(vi) The effect of bend geometry on pressure drop has been shown to be in agreement, qualitatively, with other work; unfortunately no work allowing closer quantitative comparison has been found.

Altogether the above comparisons appear quite favourable as far as it has been possible to draw them.

APPENDIX P

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APPENDIX Q

LIST OF SYMBOLS

Throughout this thesis a conscious effort has been made to avoid the use of symbols and abbreviations where not absolutely necessary, and wherever they have been used, to explain each on the spot, in order to aid the reader in obtaining the meaning of the work with as little confusion as possible.

Nevertheless, there are a few places in which symbols have been used in order to make manageable what would be otherwise unweildy expressions. The following may not be an exhaustive list but should include all symbols which are not explained immediately at their point of use:-

a - pipe cross sectional area, m^2

c - superficial air velocity, m/s, $= \dfrac{\text{actual volume flow rate of air, } m^3/s}{\text{pipe cross sectional area, } m^2}$

d - pipe inside diameter, m

f - pipe friction loss coefficient or 'friction factor' in the Darcy equation,

$$H_f = \frac{fl.c^2}{m\ 2g}$$

(Wherein H_f = head loss along pipe, in metres of fluid, and l = pipe length, metres)

K - Bend loss coefficient in expression for pressure loss caused by bend,

$$\Delta p = K.\frac{1}{2}.\rho_s.c^2$$

$\dot{m}$ - mass flow rate of solids, kg/s

m - mean hydraulic depth, i.e. $\frac{\text{filled cross sectional area of pipe}}{\text{wetted perimiter}}$

(for pipes partly filled with liquid)

Δp - Pressure loss caused by bend, N/m^2

Re - Reynolds number

r - bend radius, m

s.l.r. - solids loading ratio, $= \frac{\text{mass flow rate of solids}}{\text{mass flow rate of air}}$

$\left(\frac{dp}{dl}\right)_{solids}$ - 'solids contribution' to pressure gradient in straight pipe, i.e. total pressure gradient minus that for air alone.

ρ_s - suspension density, kg/m^3;

$$= \frac{\text{mass flow rate of solids, kg/s}}{\text{actual volume flow rate of air, } m^3/s}$$

μ - fluid coefficient of dynamic viscosity, Ns/m^2

APPENDIX R

COPIES OF PAPERS PUBLISHED DURING PROJECT

Whilst this project was still in progress, some of the findings included in this report, as well as some related work, were published in various journals and conference proceedings.

Included in this Appendix is a selection consisting of three of the most important papers, as follows:-

(a) "An Improved Method of Predicting Pressure Drop Along Pneumatic Conveying Pipelines", as published in the September 1990 issue of "Powder Handling and Processing" journal, Trans Tech Publications, Clausthal-Zellerfeld, Germany. (First published at the Institution of Engineers, Australia, 3rd International Conference on Bulk Materials Storage, Handling and Transportation, Newcastle, NSW, June 1989).

(b) "Pressure Losses Caused by Bends in Pneumatic Conveying Pipelines - Effects of Bends Geometry and Fittings", as published in the November 1990 issue of "Powder Handling and Processing" journal, Trans Tech Publications, Clausthal-Zellerfeld, Germany. (First published at the 14th International Powder and Bulk Handling and Processing Conference, Chicago, Il., USA, May 1989).

(c) "Pressure Drop in Pneumatic Conveying Through a Pipe Inclined Downwards", published at the 4th International Conference on Pneumatic Conveying Technology, Glasgow, Scotland, June 1990.

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powder handling & processing Volume 2, Number 3, September 1990

An Improved Method of Predicting Pressure Drop Along Pneumatic Conveying Pipelines

M.S.A. Bradley and A.R. Reed, U.K.

Summary

In order to design an efficient pneumatic conveying system, it is essential to be able to predict with accuracy the pressure drop to be expected along the pipeline. In this paper, the approaches currently used for making this prediction are examined and their accuracy assessed.

The limitations of these methods, namely the testing-and-scaling method and the analytical approach, are pointed out, and a new method is proposed which avoids these limitations. The new method involves the conveying of the product for which the final system is to be designed, in a special test plant to obtain data on the separate effects of bends and straight sections. Data obtained in this way are fed into a storage-and-recall system which has been especially designed for the purpose, and then extracted for use in system design.

The means for obtaining such data, the storage-and-recall system, and the method of using the data for system design, are presented; also some of the data around which the system was developed are presented.

1. Introduction

The ability to make an accurate prediction of the pressure drop to be expected in pneumatic conveying pipelines is a major factor in the design of systems which work efficiently from the points of view of power consumption, maintenance, and kindness to the product. Currently there are two methods in common use, each of which has its own advantages and drawbacks. However, it has proved possible to evolve a method which draws on both those in current use, and which may be applied widely to cope with dense phase conveying, stepped pipelines and other troublesome cases.

Paper first presented at Institution of Engineers, Australia 3rd International Conference on Bulk Materials Storage, Handling and Transportation, Newcastle, Australia, 27-29 June 1989

For information on authors - see page 290

2. Methods Currently in Use

2.1 The Global Testing-and-Scaling Approach

One method which is used fairly widely for predicting pressure drop involves building a pilot scale test rig and operating it with a sample of the product which is to be conveyed in the final system, measuring flow rates of air and product and pressure drop. The data obtained are then scaled to predict the pressure drop in the projected system using procedures which have been determined by trial. Such procedures are described in detail in [1 – 3].

This procedure has the advantage that real test data for the conveyed product are used for design work.

The scaling procedures for pipeline length and diameter are generally recognised as reliable. A problem arises, however, when the final pipeline has a different number and/or distribution of bends from the test line. Originally it was hoped that an equivalent length for pipe bends could be used in conjunction with the scaling procedure for pipeline length; however, attempts to determine the necessary values of equivalent length have shown that they are very dependent on conveying conditions (air velocity and solids loading) as well as being affected by product types and bend type. For example, *Westaway* [4] found values varying between 8 m and 20 m for one product and pipe bore in lean phase systems; *Mills* [5] found values from 2 m to 20 m, again for a single product and pipe bore, with a strong correlation with air velocity at line inlet (see Fig. 1).

The result of this, coupled with the fact that falling air pressure along a conveying line leads to increasing velocity through expansion of the air, means that the true equivalent length of a bend will be dependent upon its position in the conveying line as well as other factors. This precludes accurate prediction of pipeline pressure drop using the scaling method when a significant change in number or position of bends is involved, particularly

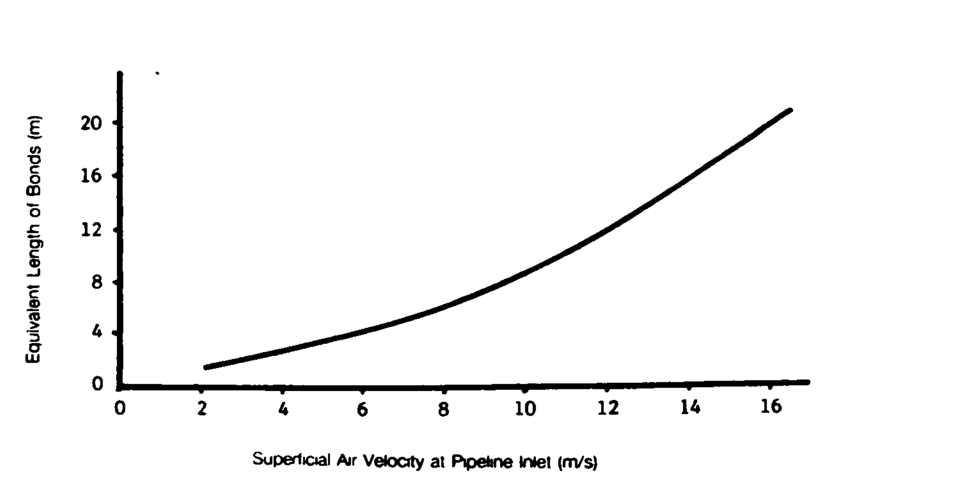


Fig. 1: The relationship between bend equivalent length and air velocity demonstrated by *Mills*

since it is known that for lean phase systems the bends can account for as much as 80% of the pressure drop.

2.2 The Piecewise Analytical Approach

The alternative to dealing with the pipeline as a whole is to treat each of its features separately, starting from known flow conditions at one end of the pipe and estimating the pressure loss and change in flow conditions caused by each bend and straight length in turn, progressing right along the pipe and thus finishing up with a value for the total pressure drop. Such a piecewise method is normally employed where the mathematical modelling of pressure drop is attempted.

By working in this way, the effect of bends in the line can be analysed using the true conveying conditions prevailing at the point where they are located.

However, the difficulty in analysing and modelling the complex processes in two-phase flow makes it unlikely that it will ever be possible to predict pressure loss reliably by analytical means. A brief survey of the literature will show the vast amount of work which has been done in this direction, with very little agreement even for lean phase conditions, let alone for the much more complicated cases of non-suspension flow. Experience shows that even apparently similar products can behave quite differently when being conveyed, but the difference cannot be accounted for by measurable product properties (e.g. size analysis, bulk density, permeability, etc.)

3. A New Method

From the foregoing it is apparent that both have some advantages; the Testing and Scaling approach because it uses real data for the conveyed product thus giving a high certainty level about the effects of product type, and the Piecewise Analytical approach because the effects of pipeline features (especially bends) can be examined in detail. From this it will be seen that if a method could be evolved wherein pressure drop predictions could be made using test data from the actual product, but using a piecewise approach rather than a global one, then it would share the advantages of both the methods mentioned above without suffering from the drawbacks. Such a method is given below.

This method involves:

1. Testing the product to be conveyed in a rig designed to obtain data on the effects of individual pipeline features (straight lengths and bends).
2. Entry of the data into a specially-developed storage system designed to be quick and easy to use.
3. Recall of the data from the storage system and synthesizing the performance characteristics of the proposed pipeline.

The requirements for this method are therefore quite clearly a suitable design of test rig, useable systems for storage and recall of the data, and a means of using the data for the synthesis of pipeline conveying characteristics.

4. Development of the Method

In order to develop such a method, it was necessary to evolve a means for measuring the pressure drop caused by bends and the pressure gradient in straight pipes, then to use this to procure a sufficiently large volume of data using realistic pipelines and products. After analysis, it was hoped to ascertain which variables have a significant effect, and to evolve the necessary data storage system. Finally, a means for using the data to predict the pressure drop in other pipeline systems would be needed.

4.1 Measuring Pressure Drop Caused by Bends and Straights

To measure the steady pressure gradient along a straight pipe represents no great difficulty, requiring simply pressure tappings at intervals along the pipe. Measurement of the pressure drop caused by a bend is a little more difficult though; it has been demonstrated for both single- and multi-phase flow (e.g. [6,7]), that most of the pressure drop occurs not within the bend itself, but in the straight pipe downstream where the disturbed flow is sorting itself out. However, by obtaining pressure profiles along the straight pipes adjacent to a bend it is possible to establish a value for a step change in pressure equivalent to the loss caused by the bend, as illustrated in Fig. 2.

4.2 Experimental Work

4.2.1 The test rig and programme

The rig used for this work consisted of a 1.5 m^3 high pressure blow tank feeding pipeline loops of 2, 3 and 4 inch nominal bore (53, 81 and 108 mm inside diameter) laid out as shown in Fig. 3. A full description of the blow tank plant may be found in [8].

Pressure measurement was by electronic pressure transducers connected via a suitable interface to a computerized data-logging system. Readings were taken at regular intervals over a time span of typically two minutes or so during the steady-state part of the conveying cycle, and averaged to remove the effect of pressure fluctuations.

Bends of seven different designs with different radii were used in the test programme. Products used for the test work were white wheat flour and polyethylene pellets.

As wide a range of conveying conditions as possible were covered, typically from 4 to 45 m/s superficial air velocity, and mass solids loading ratios of from zero to 130 for the flour and zero to 59 for the polyethylene pellets.

4.2.2 Results obtained

The pressure traces observed displayed a steady gradient approaching the bend and a curve after leading into a steady gradient further downstream; the shape

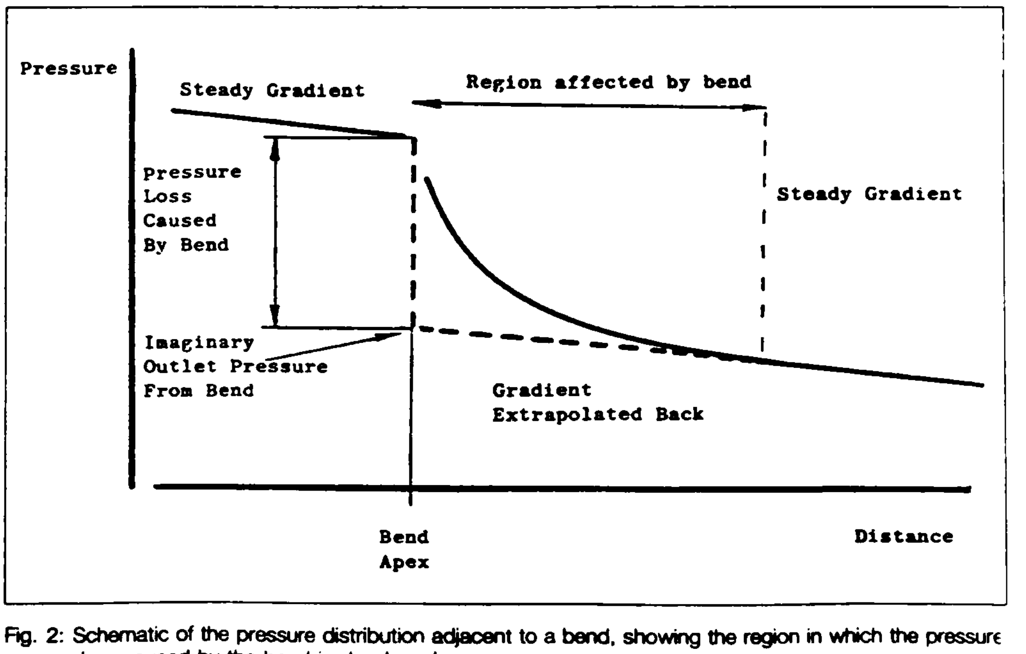


Fig. 2: Schematic of the pressure distribution adjacent to a bend, showing the region in which the pressure drop caused by the bend is developed

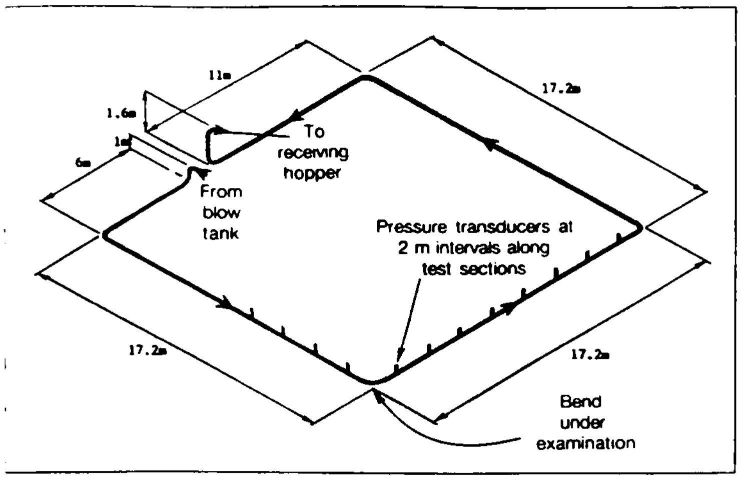


Fig. 3: The test pipeline used, and positions of transducers

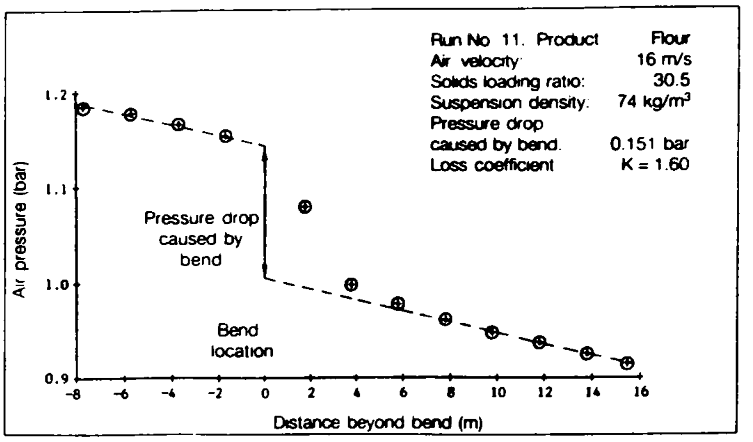


Fig. 4: Example of pressure distribution measured adjacent to bend. N.B. distance from bend measured from intersection of centre-lines of adjacent straight pipes

and length of the curved section varied widely with solids loading but an example of a profile is shown in Fig. 4.

Analysis of such data from more than 900 test runs with the different bends, products and pipe sizes resulted in a large volume of data on bend and straight pipe pressure drops. This formed the basis for developing some storage systems.

4.3 Development of the Data Storage Systems

In order to put the large volume of data into a manageable form, attempts were made to develop systems for storing it in a more compact way. Clearly two separate systems were required, one for bend pressure drop and one for straight pipe. The aim was to determine how the data might be represented in the most compact way possible with sufficient accuracy for design purposes. Ideally, suitable systems would display the following features:

a) A dimensionally homogeneous equation containing only measurable variables and coefficients.

b) These coefficients preferably to be independent of the variables in the equation, or else easily found from a single chart.

4.3.1 System for bends

A great deal of effort was put into trying to find a suitable system for bends, since this was felt to have been particularly neglected by previous workers.

Using a micro-computer it was possible to re-process and re-present the data in many different forms and on different graphs, looking for correlations between the pressure drop and other variables; by this means it was found that the following system was suitable

$$\Delta p = K\frac{1}{2}\rho_s c^2$$

where

Δp = pressure drop caused by bend, in bar

ρ_s = notational *suspension density*, i.e. kg of product flowing per m^3 of conveying air (using true volume flow rate of air at pressure in the pipe, not "free air" conditions).

c = superficial air velocity, calculated from true volume flow rate of air and pipe cross-sectional area

K = coefficient.

This is similar to the system used for bend losses in single-phase flow, where the loss is taken to be proportional to the dynamic pressure or "velocity head" of the flow, the proportion (i.e. the value of the coefficient) depending only on bend geometry and, to some extent, pipe bore.

In this case it proved impossible to make the coefficient independent of the variables in the equation; however, it was found that it could be represented on a single graph for each bend and product type, against either air velocity or suspension density. Examples are shown in Fig. 5.

It was found that with the data expressed in this way, neither air density nor pipe bore had any significant effect on the K values. An analysis of the effect of bend geometry may be found in [9], although it is unnecessary to address the question here since when using the strategy recommended, one would normally be testing a bend of the actual geometry to be used in the final system.

4.3.2 Systems for straight pipes

Rather less work was done on straight pipes; a somewhat imperfect but quite useable system of the following form was developed:

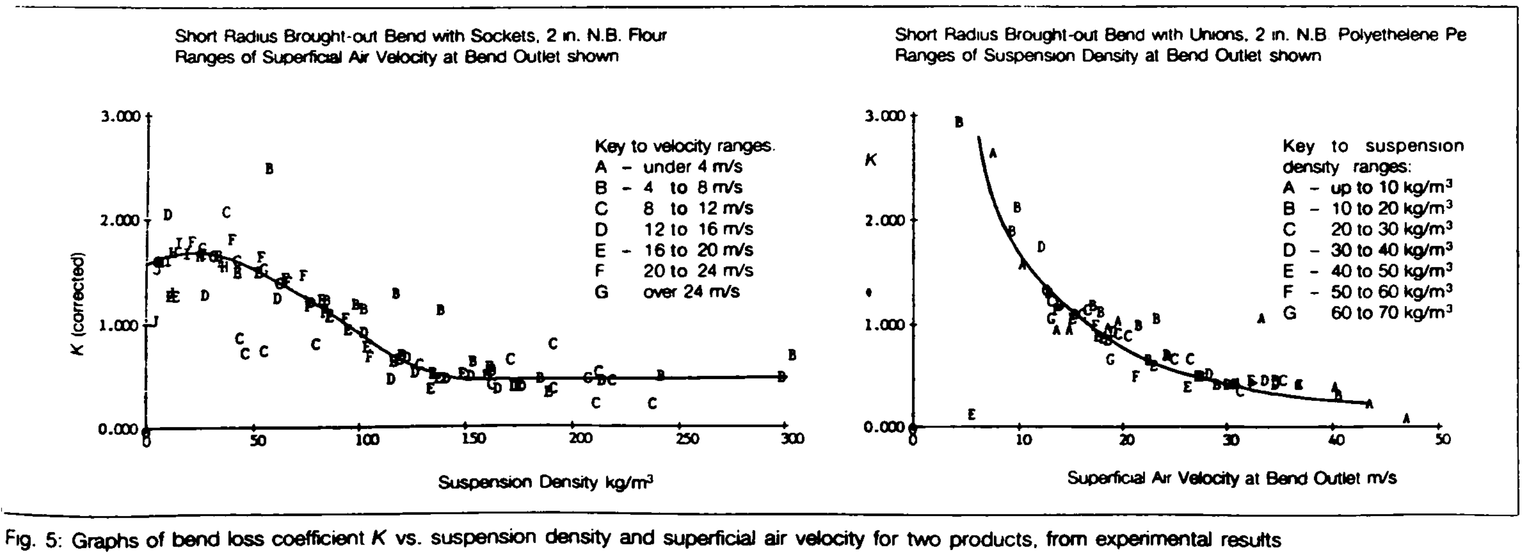


Fig. 5: Graphs of bend loss coefficient K vs. suspension density and superficial air velocity for two products, from experimental results

(*G*)total = (*G*)air only +(*G*)solids

where

(*G*)total = pressure gradient observed in pipe, in bar/m

(*G*)air only = pressure gradient which would be expected with air only in the pipe, calculated from the *Darcy* equation

(*G*)solids = additional pressure gradient, notionally caused by the addition of the solid particles to the air, again in bar per metre.

The additional pressure gradients caused by the addition of the solids could be represented by:

For wheat flour

$$(G)\text{solids} = 6.5 \times 10^{-3}\left(\frac{\rho_s}{100}\right)^n$$

where $n = \frac{c}{8}$

and for polyethylene pellets

$$(G)\text{solids} = 4.4 \times 10^{-3}\,\rho_s \cdot c$$

ρ_s and c as defined above.

It should be borne in mind that these equations are not dimensionally homogeneous and therefore must not be used with other units; however, the coefficients turned out to be constant over the wide range of conveying conditions covered, obviating the need for any charts.

4.3.3 Other products

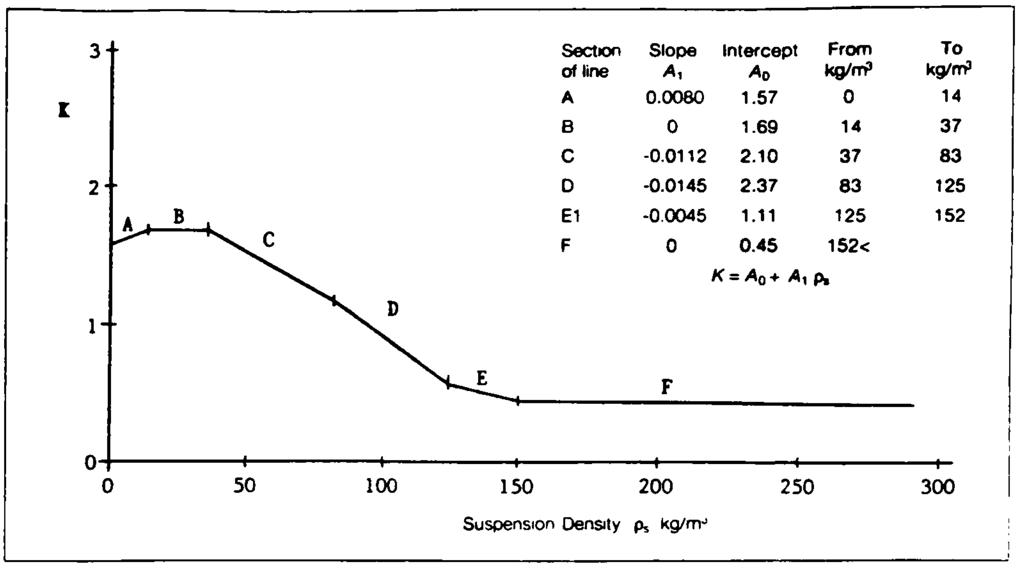
The systems outlined above were developed using data for only two products; therefore it must be expected that when testing other products, some difference will be observed.

For the bend pressure loss relationships the dependence of the loss coefficient *K* on the superficial air velocity and the suspension density will be different, which the test work will establish quite easily.

For the pressure gradient in the straight pipe, the principle of adding a "solids contribution" to the "air only" pressure drop should still prove useful. However, the somewhat different nature of the two equations for the solids contribution for flour and pellets would suggest that some variation in the equations necessary to represent this adequately should be expected. It may be that further work with a range of product types will show up a more general form of equation, or a combination of equation and chart.

4.4 Pipeline Characteristic Synthesis

Having obtained data on pressure loss caused by the bends and straight pipe



Section of line	Slope A_1	Intercept A_0	From kg/m³	To kg/m³
A	0.0080	1.57	0	14
B	0	1.69	14	37
C	-0.0112	2.10	37	83
D	-0.0145	2.37	83	125
E1	-0.0045	1.11	125	152
F	0	0.45	152<	

$K = A_0 + A_1\,\rho_s$

Fig. 6: Piecewise linear approximation of curve of loss coefficient vs. suspension density for flour in the short radius bought-out bend

for the product tested, and shaped it into a manageable form by means of a data storage system, this data can be used to synthesize the conveying characteristics (relationship between flow rates of product and air, and pressure drop) for pipelines of virtually any layout.

4.4.1 Procedure

The first requirement is a datum for pressure, which for a positive pressure system will be at the end of the conveying line where the product is discharged usually to a hopper at atmospheric pressure or occasionally to a vessel at a known pressure.

The mass flow rate of product will be a primary design parameter, so will be known. Values for mass flow rate of air and pipeline bore are chosen, by an educated guess. From this information it is possible to calculate the superficial air velocity and suspension density at the end of the pipe, and use these values in the equation for straight pipe pressure drop to estimate the pressure loss in the final straight section. This gives the pressure at inlet to this straight section, from which new values for velocity and suspension density are calculated; these values are used to estimate the pressure drop caused by the bend at this point, from which the pressure and thus velocity and suspension density at inlet to the bend can be calculated. The procedure is simply repeated for each straight and bend in turn, right back to the start of the pipeline, to obtain the total pressure drop along the pipe and the velocity at inlet.

In the case of a vacuum system, the datum for pressure is at the beginning of the pipe, so, of course, the procedure would start here and progress forwards along the pipe.

This is a very simple procedure, the repetitive nature of which suggests the use of a computer; a suitable program is outlined in the Appendix. Using this program the whole procedure can be repeated quickly and easily with new values for pipe bore and mass flow rate of air to build up a picture of possible systems for the required duty, from which a choice can be made. The only slight inconvenience with the use of the computer is where the data storage system involves the use of a chart such as in Fig. 5. A piecewise linear approximation is recommended, i.e. representing the graph by a series of straight-line relationships as shown in Fig. 6.

4.4.2 Vertical sections

These have not been mentioned so far; ideally an instrumented vertical section would be incorporated into the test rig, but no work has been done on this by the author. In the absence of any other information, the relationship demonstrated by *Mills* [10], that vertical-up sections display pressure drop twice that of horizontal sections, should be used and such sections treated as horizontal ones of double the length.

For vertical-down sections, treating them as horizontal sections of the same length will result in some over-estimation of pressure drop and thus conservative design. Fortunately, long vertical-down sections are not common in real systems.

4.4.3 Stepped Pipelines

The method recommended can cope very easily with pipelines which have changes in bore size along their length, by a simple instruction in the computer program. Such stepped lines are essential to long distance conveying and can result in very substantial power savings in any system which has a high input pressure, say greater than 1 bar g. It may be here that this method has its greatest advantages, especially since comparisons between different systems can be made so quickly.

5. Example

To demonstrate the method, a set of conveying characteristics for the flour in the 2 inch NB pipeline loop containing the test sections was synthesised, and compared with the true characteristics of the loop as subsequently measured. These characteristics cover the whole range of mass flow rates of air and product which it was possible to sustain in the system. The result is quite favourable, as shown in Fig. 7.

6. Conclusions

The method outlined consists of:

a) Testing the actual product to be conveyed to obtain data on pressure drop caused by pipeline bends and straights.

b) Feeding these data into suitable storage systems, such as those demonstrated.

c) Recalling these data and using them to predict the pressure loss to be expected in any system conveying that product.

The advantages of this strategy are:

1. Real test data on the product are used for design, which will always be essential for accurate work; thus removing any element of doubt arising because of the unpredictable effects of different products.
2. The effect of pipeline features, particularly number and location of bends, can be examined in detail very easily, and different systems compared quickly.
3. Stepped pipelines can be considered with equal ease, and systems with different numbers and positions of steps can be compared quickly.

The main requirement for the test rig, beyond what is normally used in laboratories where conveying trials are carried out, is a pipe loop with two adjacent straight sections about 16 m or more long, instrumented with pressure transducers, a data logging system and a microcomputer for data processing. The cost of this equipment is only a small part of the cost of a test rig, and once set up it is easily operated.

Finally, it should be emphasised that, as with the other methods for predicting pressure drop, this method is not in itself proof against designing a system which will not work – it is still up to the designer to satisfy himself that, most importantly, air velocities are within sensible limits, i.e. high enough to prevent blockage but not so high as to cause undue wear or product degradation. The fact that velocity is recalculated at every bend in the system means that such information is readily available.

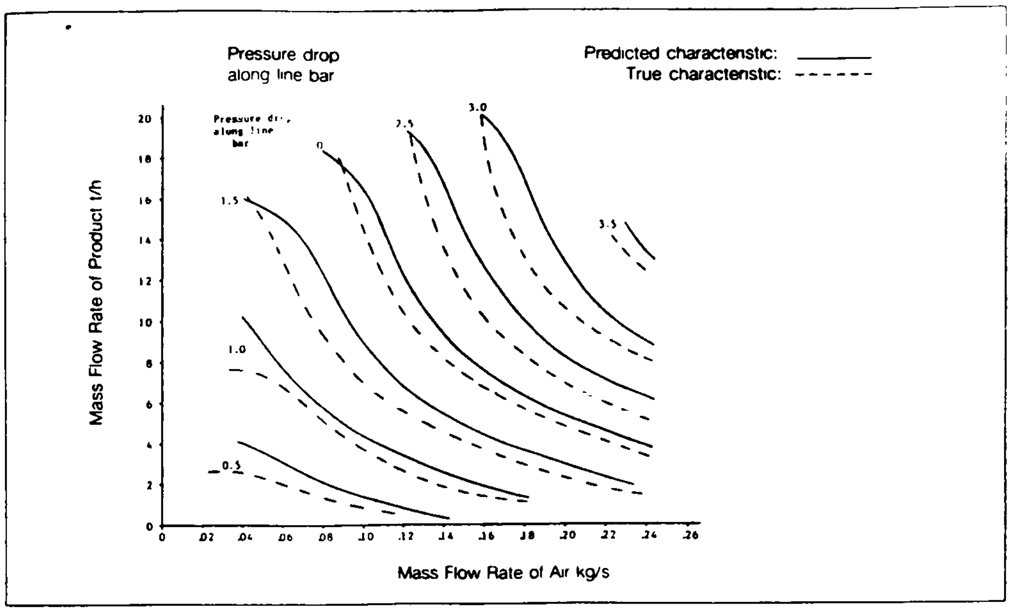


Fig. 7: Comparison between characteristics of loop in which test sections were located, as predicted using the method described, and as subsequently measured

Acknowledgements

The authors would like to thank Mr. *I.R. Bittle* of Thames Polytechnic, for his help and encouragement in this work; and the Science and Engineering Research Council of the U.K. and HG Engineering Ltd. of Canada for supporting this work.

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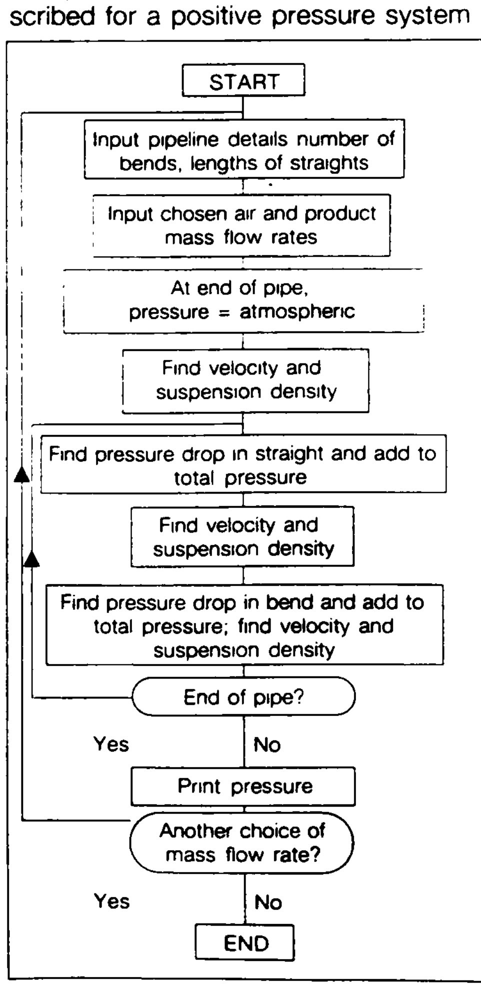
Appendix

Computer program for assessing pipeline pressure drop using method described for a positive pressure system



Pressure Losses Caused by Bends in Pneumatic Conveying Pipelines

Effects of Bend Geometry and Fittings

M.S.A. Bradley, U.K.

Summary

This paper presents the outcome of test work covering bends of seven different common types of varying radius and design, fitted by two different means into a pneumatic conveying pipeline of a single nominal size of bore. The work was undertaken chiefly with a single product, whilst a second product of a very different nature was also used with a single bend type. A wide range of conveying conditions was achieved for each case.

A suitable system for the storage and recall of the pressure loss of information is demonstrated using the resulting data, and the performance of the various bend types is compared across the range of conveying conditions. Conclusions are drawn with regard to the choice of bend types in system design, taking into account not only pressure loss but other factors as well.

1. Introduction

The question of suitable geometries for bends in pneumatic conveying systems is one which has been the subject of a number of papers over the years, but nevertheless no general consensus on how to choose bends has emerged. The effects of a bend in a pneumatic conveying pipeline are twofold; it causes a loss of energy which results in an additional pressure drop, and it can either cause product attrition or suffer from wear, depending on the relative hardnesses of the product and pipe materials. It is widely recognised that the magnitudes of both of these effects are dependent on the type of bend chosen, e.g. the radius of curvature.

Based on a paper first presented at the 14th International Powder and Bulk Handling and Processing Conference, Chicago, IL, USA, 15–18 May, 1989

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Thus there are two criteria which a pipeline designer must consider when choosing suitable bends; these are (firstly) to obtain the least possible energy consumption by using bends which cause a low pressure drop, and (secondly) to keep wear of the bends or attrition of the product to an acceptable level. In this paper it is the question of pressure drop which is mainly addressed, although certain types of wear-resistant bends are considered from this point of view.

1.1 Previous Work

The recommendations of previous authors in the field appear to be conflicting; for example *Marcus*, *Hilbert* and *Klinzing* [1] stated that short radius bends caused the least pressure drop, whilst *Mills* and *Mason* [2] found short radius bends better in some circumstances whilst long radius bends were better in others; they found blind tees particularly bad, yet *Bodner* [4] stated that blind tees caused pressure drops "not significantly different" from radiused bends. Some of these studies dealt only with a very narrow range of conveying conditions which may account in part for the discrepancies.

The work reported here follows on from a paper presented at the 13th Powder and Bulk Solids Conference in Rosemont, IL, USA, in 1988 [3] in which the authors examined how pressure-drop effects of bends in a pipeline might be accounted for in predicting overall pipeline pressure drop for design purposes. The main conclusions presented therein were that the contribution to overall system pressure drop made by the bends may be very significant and that the pressure drop caused by each pipeline feature, for example, each straight and each bend, must be considered separately in order to achieve accurate predictions. Accordingly, the same approach of looking at the effect of an individual bend of each type has been used in this work.

1.2 Approach Used for This Work

The strategy here has been to generate pressure drop data for a single product (white wheat flour) flowing through a wide variety of bend types, and for another product of a very different nature (polyethylene pellets) in a single bend type, and then to look for correlations in this data in order to establish a basis for finding a coefficient for each bend so that these coefficients may be compared.

Results from only one nominal bore size of pipe (i.e. 2 inch) have been examined in this report although the effect of pipe diameter is mentioned briefly later on. Other effects which have been considered are that of air density which has been shown not to be a factor with the results presented as they are here, and that of change of the product with repeated conveying, which has been isolated and corrected for. A comprehensive range of conveying conditions has been covered. Also the effects of the method by which the bend is connected to the adjacent pipes has been examined although the results are not conclusive.

2. Experimental Work

2.1 Test Rig

It has been clearly demonstrated [3] that the pressure loss caused by a bend occurs mainly in the straight section of pipe downstream of the bend, and not in the bend itself (Fig. 1). Accordingly, the method of measuring the pressure drop and the test rig used which were outlined in [3] were used for this work.

Essentially, the bend under examination was installed in a test loop between two

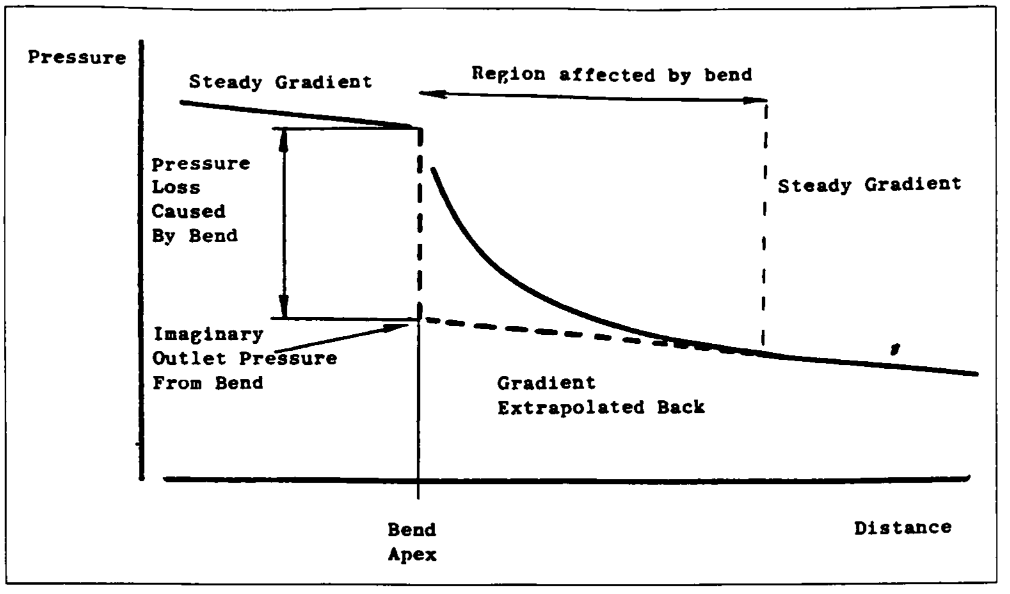


Fig. 1: Showing how the pressure in a pipeline is affected near a bend, and how it may be represented by an equivalent step change

long straight sections of pipe (in excess of 17 m (57 ft)), these straight sections being instrumented with pressure transducers at 2 m (approx. 6.5 ft) intervals (see Fig. 2). The outputs from the transducers were monitored by a computerised data-logging system and in this way the pressure profile along the two straight sections was obtained. These data, being the result of averaging many readings from each transducer over a selected steady-state period of operation (typically 1 - 2 minutes) were analysed to extract a value for the pressure drop caused by the bend.

The test loop was fed by a high pressure blow tank of 1.5 m^3 (54 ft^3) capacity, and ended in a receiving hopper on load cells which were also monitored by the data logging system. A full description of the test rig and method of data analysis may be found in [3].

2.2 Bends Used

Seven bends were used for this work, as illustrated in Fig. 3 and listed below:

1. Short (6.5 in) radius bought out bend, fitted with screwed sockets
2. Short (11.4 in) radius
3. As (2) but fitted with screwed unions
4. Long (28 in) radius
5. As (4) but fitted with screwed unions

Bends 2 to 5 were manufactured in-house from identical pipe, for consistency.

6a. Female malleable elbow

6b. Male malleable elbow fitted with screwed unions

7. Blind tee
8. Hammertek "Vortice-ell" bend.

The significance of the screwed unions or sockets was to introduce a gap of about 3/8 in between the ends of the pipes joined by them. The bends whose method of connection is not listed were formed in one with the straight sections on either side, by bending.

2.3 Experimental Programme

Each bend was tested in turn with the same product (wheat flour) over as wide a range of conveying conditions as possible, which covered velocities from 4 to 45 m/s (800 to 9,000 ft/min) and solids loading ratios of from zero to 130 kg of product per kg of air.

This resulted in approximately 800 test runs and it was suspected that there may be some changes in the characteristics of the product caused either by attrition or changing moisture content or possibly other effects. In order to isolate any such effect, the bend which had been used first was replaced and a full set of tests re-run approximately half-way through the programme.

To look at the effect of air density on the pressure drop caused by a bend, the same bend (short radius) was subjected to two further full sets of tests with the resistance of the return conveying line to the receiving hopper (i.e. the line downstream of the test section) altered. Firstly, by lengthening the conveying line to increase its resistance, the absolute air pressures at the bend were increased, and secondly, by enlarging the return line to reduce its resistance, the absolute air pressures at the bend were reduced. In this way three full sets of data with otherwise identical conveying conditions (i.e. air velocities and solids flow rates), but with differing air densities, were pro duced.

Finally, the polyethylene pellets were loaded into the rig and tested with just the short radius bought-out bend.

3. Results

The pressure-distance profiles obtained displayed a straight line gradient approaching the bend and then a curve after, leading into a steady gradient further downstream. A typical pressure distribution is illustrated in Fig. 4.

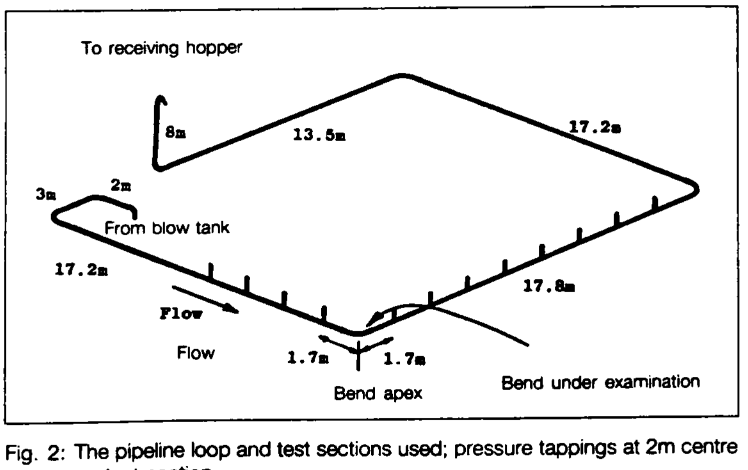


Fig. 2: The pipeline loop and test sections used; pressure tappings at 2m centre on test section

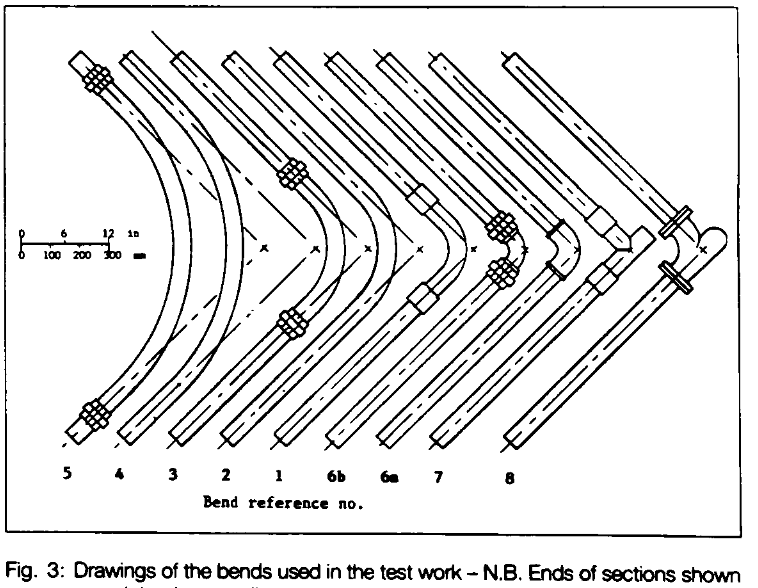


Fig. 3: Drawings of the bends used in the test work – N.B. Ends of sections shown were joined onto adjacent straight lengths using *Morris* couplings (giving joint with smooth interior)

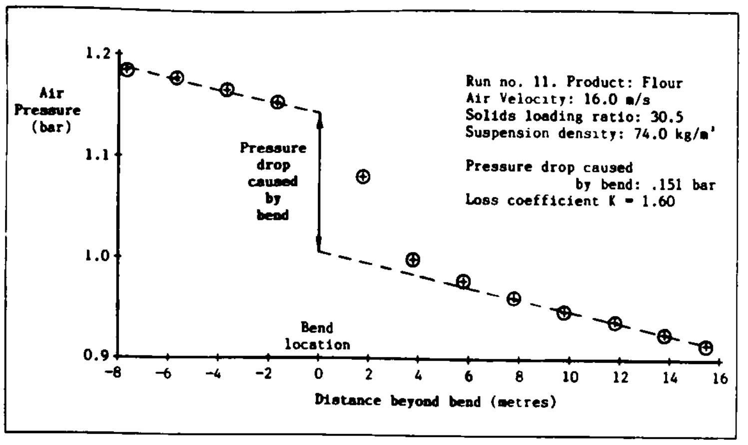


Fig. 4: Example of pressure distribution measured adjacent to bend
N.B. Distance from bend measured from intersection of centre-lines of adjacent straight pipes

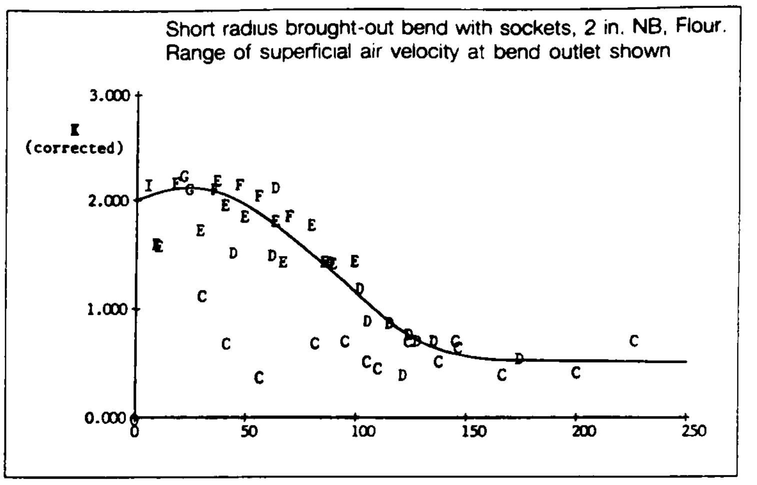


Fig. 5a: Graphs of loss coefficient vs. suspension density for radiused bends; ranges of air velocity shown
(see overleaf for further examples)

3.1 Analysis of Experimental Data

With the number of test runs performed, the resulting volume of data was very large. In order to bring this data into a manageable form so that bend geometry and other effects could be identified, correlations were sought with a view to constructing a data storage system. The aim was to develop a system which ideally would display the following features:

a) A dimensionally correct equation for pressure drop containing only measurable variables and one or more coefficients

b) These coefficients to be dependent hopefully on only

   i) the product type, and

   ii) the bend type

   and ideally independent of the variables in the equation, or if not, then found from a single graph.

After a considerable amount of work, using a computer to enable the quick analysis and re-presentation of the data in many ways and on many different types of graph, some useful correlations emerged which suggested representation of the data in this way:

$$\Delta P = K\frac{1}{2}\rho_{sus}c^2$$

where ΔP is the pressure drop caused by the bend in bar; ρ_{sus} is the "suspension density", an imaginary (but easily calculated) value for the mean density of the gas-solid mixture in the pipeline, simply the kg/s of solids flowing divided by the m^3 per second of air flowing (calculated at the pressure in the pipeline, not the "free air" value); c is the "superficial air velocity", based on the pipe cross-sectional area and again the true volume flow rate of air (m/s); and K is a coefficient.

This equation happens to be very similar to that used for predicting pressure loss at bends in single phase (e.g. pure air or water) flow, where it is found that the loss is proportional to the dynamic pressure or "velocity head" of the flow, the proportion (i.e. the coefficient) being dependent only on the bend type and to some extent pipe size.

Unfortunately it proved impossible in this case to make the coefficient independent of the variables in the equation. However, it was found that with flour, the variation of the K value with superficial air velocity c was very similar in shape for all of the radiused bends, and could be represented by a single curve on a graph in each case (see Fig. 5a).

For the other types of bend, the variation of loss coefficient values was slightly more complex. The malleable elbows (both male and female) displayed characteristics very much like those of the radiused bends except that they gave somewhat lower loss coefficients under conditions of velocities less than 12 m/s (2,400 ft/min). Representing their characteristics by the same shape of curve as used for radiused bends, e.g. Fig. 5b, gives mostly a good approximation but with a slight over-estimate of pressure losses to be expected from a bend of this type at low velocities; such an inaccuracy would result in conservative design, so is thought to be acceptable.

The characteristics of the blind tee and vortice-ell bend were significantly different from the radiused bends and elbows, since they each seemed to display two distinct regions of operation, with one value of loss coefficient above a velocity of 16 m/s (3,200 ft/min) and another, lower value below this velocity, irrespective of suspension density (Fig. 5c). These characteristics could not be properly represented by a curve of the same shape as used for all the other bends.

The above comments all refer to the results obtained from using flour. With the polyethylene pellets, the observed loss coefficients varied in a different way, simply reducing with increasing velocity and independent of suspension density (Fig. 5d).

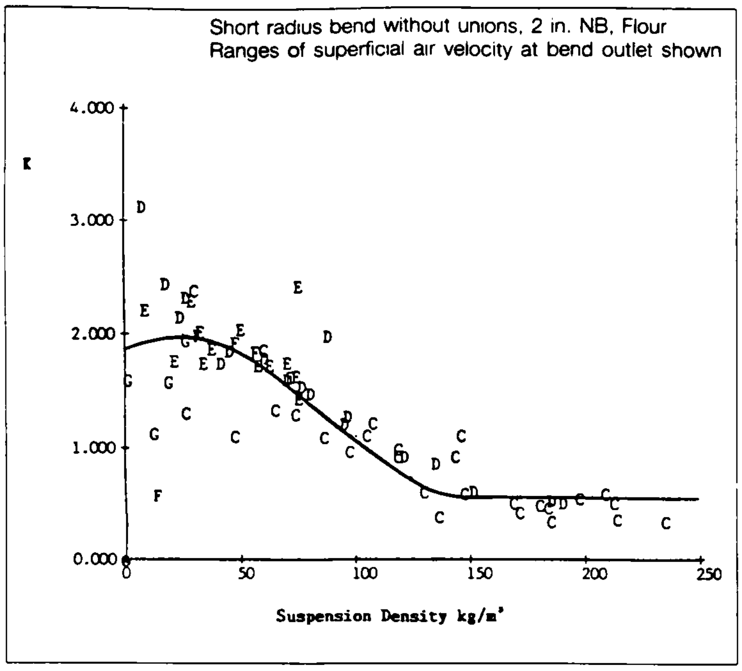
3.2 Effect of Product Change and Air Density

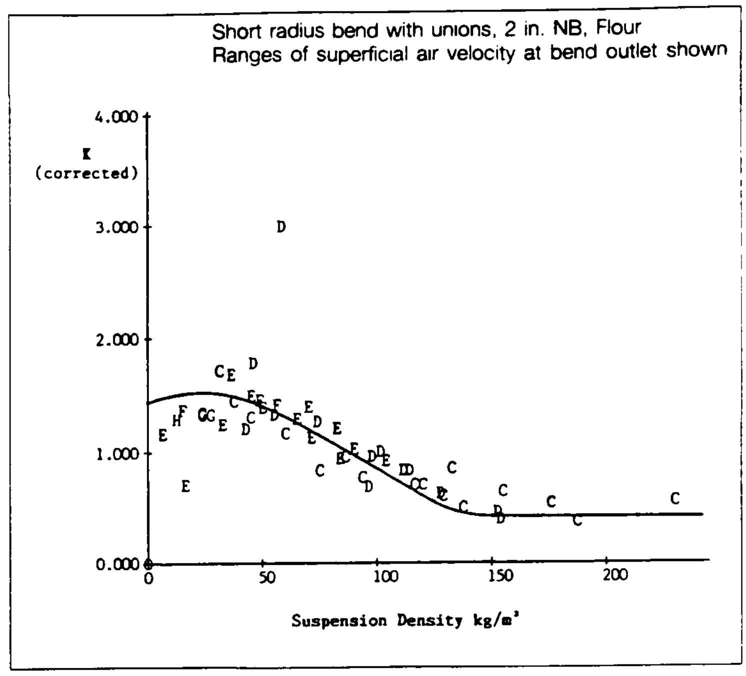
Comparison of the results from the four cases where the short radius bought-out bend was used at various times during the test programme showed a slight but steady reduction in the pressure loss coefficient K, which appeared to be close to linear with respect to the number of test runs (see Fig. 6). With a straight line put through the points, a correction for the effect of product change was deduced such that all pressure loss coefficient figures could be corrected to their "new product" values, i.e. the values which they could reasonably have been expected to have had if every test run had been done with a fresh batch of flour. It is these corrected K values which are shown in Fig. 5.

With the correction applied, it became apparent that air density had very little effect on the loss coefficient values obtained; as Fig. 7 shows, their values did not alter very much between the cases where different return pipes were used to control air density. It should be noted, however, that this would not have been so if the data had been presented on a mass solids loading ratio basis instead of the suspension density as used here. (Refer to Appendix A for an explanation).

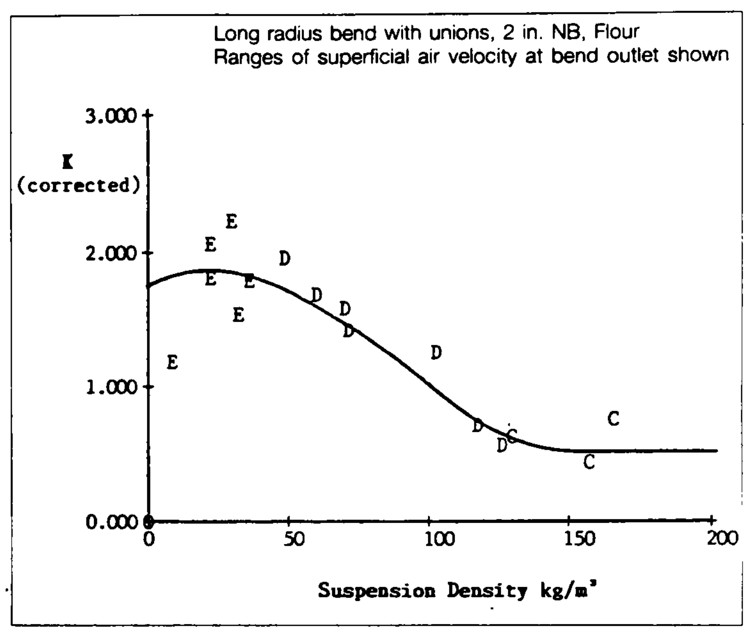
4. Comparison of Bend Types

In order to obtain a comparison of the losses caused by the different types of bend, the short radius bought-out bend was taken as a reference and the curve which best fitted its $K - \rho_{sus}$ graph was scaled to fit the graphs for the other radiused bends and elbows, and cases of different air density, etc. The necessary factor required to scale the "reference" curve to fit the other graphs, i.e. K/K_{ref}, thus became a basis for comparison of the bends. The comparison between the values for the different bends is shown in





Key to velocity ranges:
A –under 4 m/s
B – 4 to 8 m/s
C – 8 to 12 m/s
D – 12 to 16 m/s
E – 16 to 20 m/s
F – 20 to 24 m/s
G –over 24 m/s



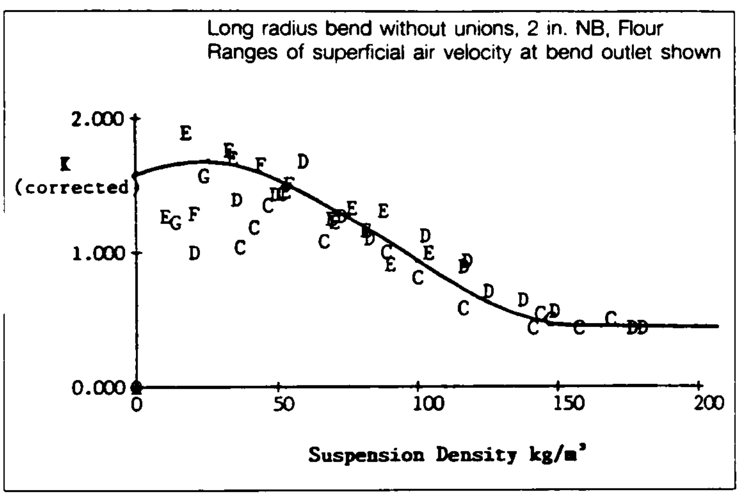
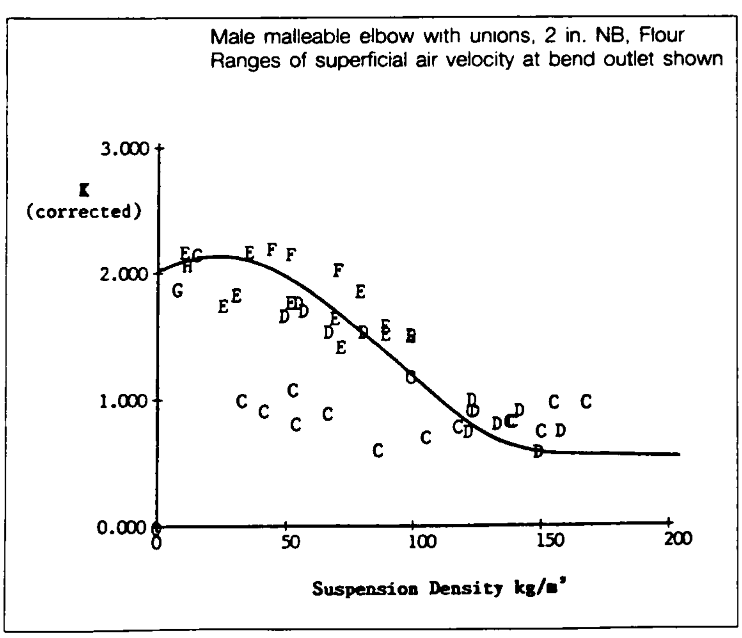


Fig. 5a: Graphs of loss coefficient vs. suspension density for radiused bends; ranges of air velocity shown



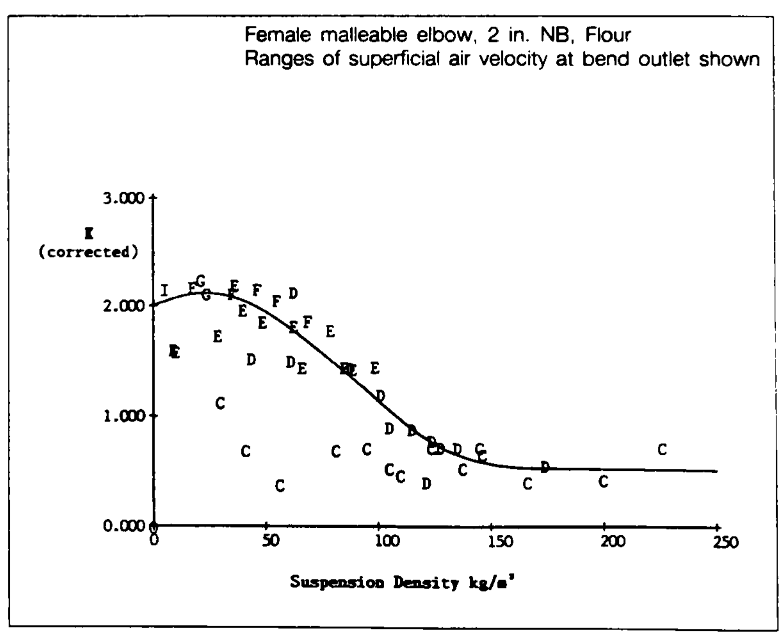
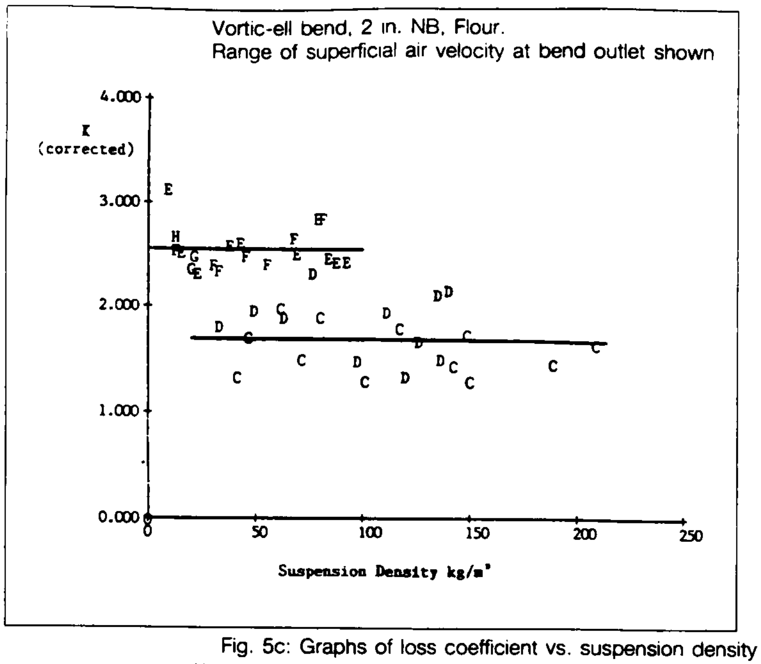


Fig. 5b: Graphs of loss coefficient vs. suspension density for elbows; ranges of air velocity shown



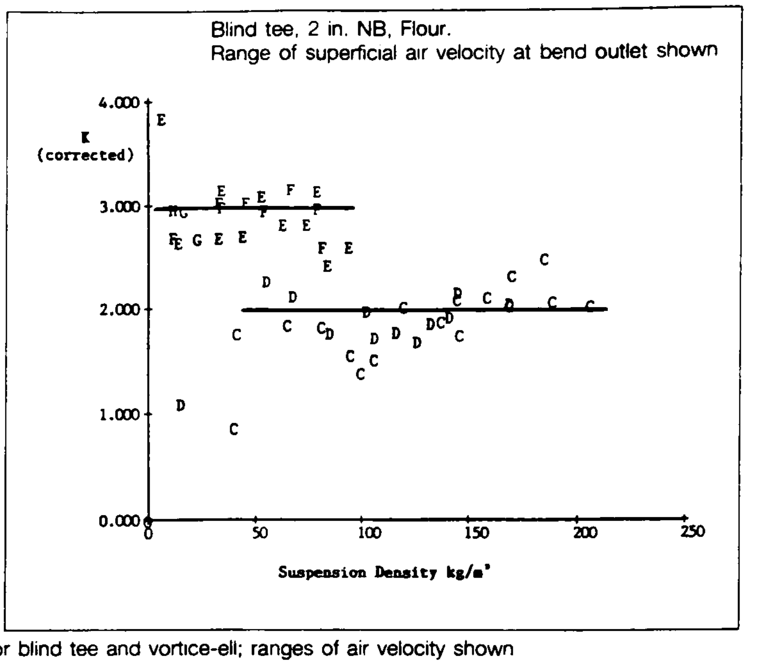


Fig. 5c: Graphs of loss coefficient vs. suspension density for blind tee and vortice-ell; ranges of air velocity shown
Key to velocity ranges: A – under 4 m/s; B – 4 to 8 m/s; C – 8 to 12 m/s; D – 12 to 16 m/s; E – 16 to 20 m/s; F – 20 to 24 m/s; G – over 24 m/s

Fig. 8 against the basis of the ratio Bend Radius/Pipe Bore.

Because the shape of the characteristic displayed by the blind tee and vortice-ell bend was distinctly different from that displayed by all other bends, a common comparison between them across all conveying conditions was not possible; hence the comparison is made for selected conditions, i.e. points 7 and 8 on Fig. 8 which represent "lean phase" systems and points 9 and 10 which represent a "dense phase" system (see footnote).

Footnote: In this context, "lean phase" is taken to mean a system where the air velocity is sufficient to pick up the product from the floor of a horizontal pipe, generally meaning greater than about 15 m/s (3,000 ft/min); "dense phase" is taken to mean lower velocities where a significant portion of the product is either static or slow-moving in the pipe.

4.1 Differences Observed

Referring to the comparative graph, Fig. 8, it is apparent that there is little to choose between the various radiused bends on the basis of pressure drop; curiously the lowest pressure drops are caused by bend no. 2, the r/d = 5.5 bend with screwed unions, whereas the identical bend without the screwed unions (No. 3) caused noticeably greater pressure drop. This was unexpected, and with the long radius bends the situation was reversed, the bend with unions causing the greater pressure drop. Thus the effect of the joints is somewhat uncertain.

The short radius bought-out bend, with an r/d ratio of 3.2 would appear to be the best choice for situations where wear or attrition is not a severe problem, because it is cheaper to buy and install and is less bulky than longer-radius bends whilst offering practically as low a pressure drop.

The malleable elbows, Nos. 6a and 6b, seem to produce greater pressure loss than the short radius bought-out bend, but they offer little advantage in terms of cost, convenience and (probably) wear-resistance so may be discarded for use in pneumatic conveying systems.

Finally to be considered are the wear-resistant bends. The blind tee is commonly used in lean phase systems where abrasive materials are being handled, and is widely recognised as giving very long service life compared with other bends under such conditions. It can be seen that in these tests, for "lean phase" conditions (point No. 7 on Fig. 8) the blind tee gave a pressure drop nearly double that

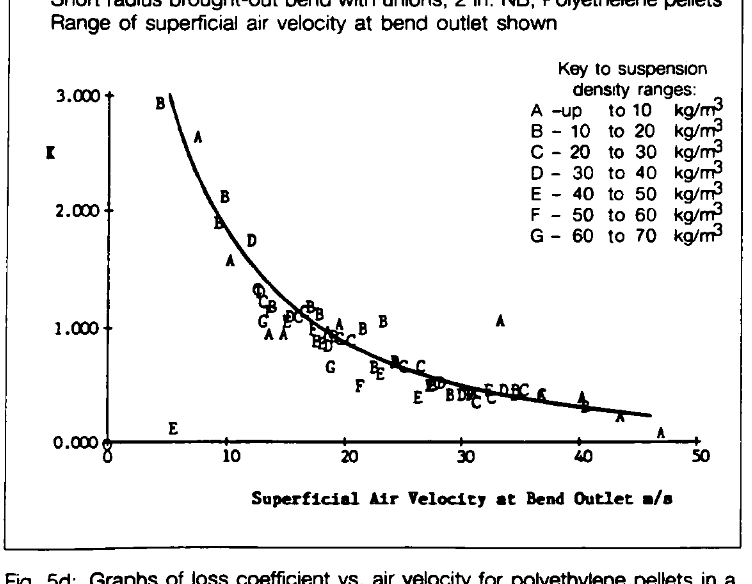


Fig. 5d: Graphs of loss coefficient vs. air velocity for polyethylene pellets in a radiused bend; ranges of suspension density shown

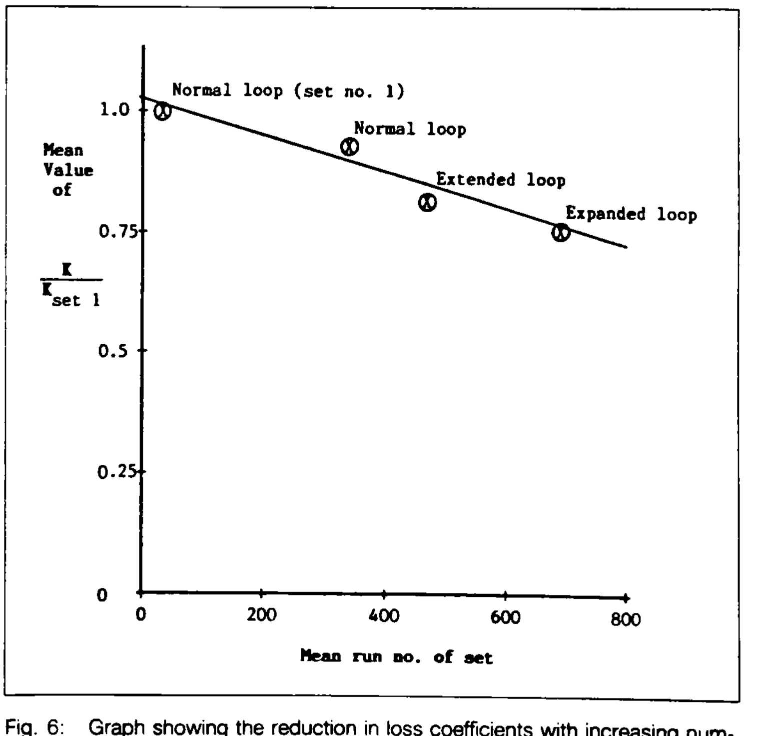


Fig. 6: Graph showing the reduction in loss coefficients with increasing number of runs. Bend: short radius without unions. Product: flour

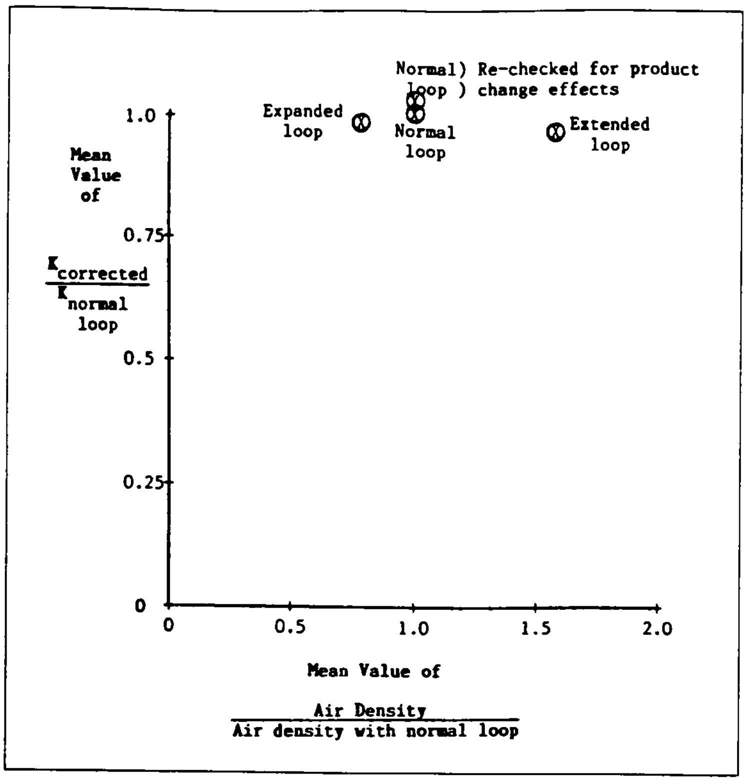


Fig. 7: Graph showing variation in loss coefficients with variation in air density
Bend: short radius without unions. Product: flour

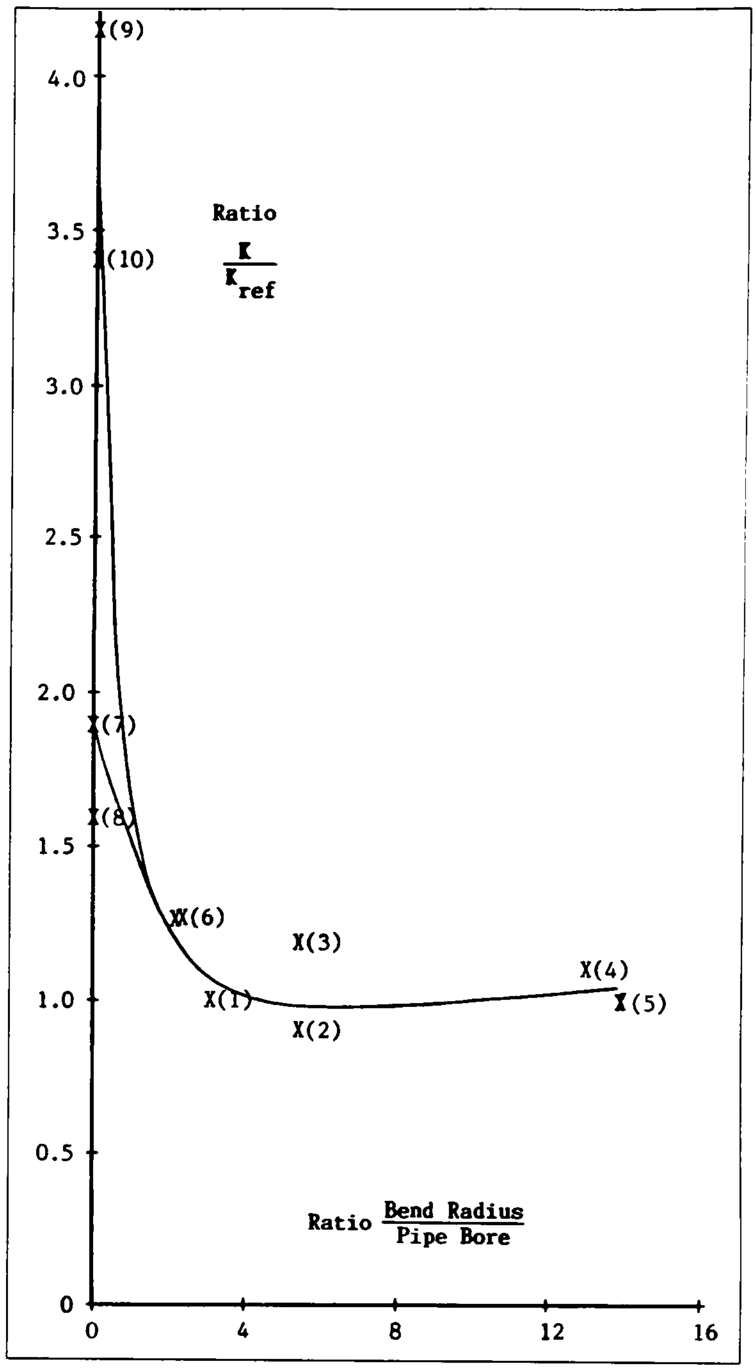


Fig. 8: Effect of r/d ratio on pressure drop factor (K) for bends, related to pressure drop factor for short radius bought-out bend (K_{ref})
Product: flour Line: 2 in N.B.

Reference to bends:

(1) Short radius bought-out, with sockets
(2) Short radius, with unions
(3) Short radius, without unions
(4) Long radius, with unions
(5) Long radius, without unions
(6) Male and female malleable elbows
(7) Blind tee { For cases of suspension density less
(8) Vortice-ell than 75 kg/m³ with velocities greater than 16 m/s
(9) Blind tee { For suspension
(10) Vortice-ell density of 150 kg/m³

of the radiused bends. With the "dense phase" conditions of a suspension density of 150 kg/m^3, point No. 9 on the graph, the loss caused by the blind tee is over four times that of the radiused bends. The "Vortice-ell" bend gave a significant improvement over the blind tee, by about 15%, though it still caused a good deal more pressure loss than the radiused bends.

It should be noted that these comparisons are all based on the data obtained using flour, since only one bend was tested with polyethylene pellets.

5. Conclusions

The conclusions to be drawn from this work are clear:

a) All radiused bends incur much the same loss, irrespective of their radius of curvature; this makes the short radius bought-out bend the best choice where wear is not a serious problem, since it is the cheapest to buy and install.

b) Malleable elbows are not a good choice because they cause more pressure drop but are little cheaper.

c) The use of blind tees should be restricted to situations where wear is a serious problem, (i.e. high velocities with abrasive products) and much higher pressure drops must be expected. They should not be used where suspension densities are high and velocities low (in which case wear is unlikely to be a problem) because that is where their performance is worst of all. This may entail fitting them only towards the end of a pipeline, where the expansion of the air causes reducing suspension densities and increasing velocities.

d) The Hammertek "Vortice-ell" bend gives a useful reduction in pressure compared with the blind tee, though is still a good deal worse than a radiused bend. For this reason they should also only be used if wear (or product attrition) is likely to be a problem.

e) The effect of the means by which the bends are connected to the straight pipes is not quite certain, but does not appear to be very significant.

f) Air density does not affect the pressure loss caused by a bend, provided the comparison is made on the basis of equal suspension density (which means equal volumetric solids loading ratio but *not* equal mass solids loading ratio).

g) Although it has not been examined in this paper, some other work which has been done indicates that pipe bore does not have a major effect on the loss coefficient between bends of similar type.

h) Further work is necessary to explore the effect of product type more fully.

Appendix A

Air density effects when using mass solids loading ratio:

It should be noted that the observation that air density has a negligible effect on bend pressure loss holds true only when the data is presented on the basis of suspension density outlined above; should the more common basis of a mass solids loading ratio be used, then air density will have a very significant effect. This is so because

$$\text{mass solids loading ratio (SLR)} = \frac{\text{mass flow rate of solids}}{\text{mass flow rate of air}}$$

$$= \frac{\text{density of gas - solid suspension}}{\text{density of air in pipe}}$$

$$= \frac{\rho_{sus}}{\rho_{air}}$$

Therefore $\rho_{sus} = SLR\rho_{air}$

Thus the equation $\Delta P = K\frac{1}{2}\rho_{sus}c^2$

becomes, with substitution of the above

$$\Delta P = K\ \frac{1}{2}SLR\ \rho_{air}\ c^2$$

So the effect of air density can be seen for the situation where the mass solids loading ratio is used to describe the flow rather then the suspension density.

Acknowledgement

The author would like to acknowledge the help and encouragement of Professor *A.R. Reed* and Mr. *I.R. Bittle* of Thames Polytechnic, who all contributed in one way or another to the work on which this paper is based; The Science and Engineering Research Council of Great Britain, and H.G. Engineering of Canada who supported the work; and finally Mitchell Cotts Mining Equipment Ltd. who were kind enough to lend a "Vortice-ell" bend for testing.

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PRESSURE DROP IN PNEUMATIC CONVEYING THROUGH A PIPE INCLINED DOWNWARDS

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Synopsis

In the past, most designers of pneumatic conveying pipelines have tended to avoid using pipes inclined at an angle because of the difficulty in predicting the pressure drop in such sections. However, there are occasions when inclined pipes are unavoidable, and this paper reports an investigation undertaken at The Wolfson Centre as a result of a need to design a long pipeline on a continuous falling gradient, to convey a mineral product with a size range of 2-25mm.

The methods for obtaining the pressure gradients in the pipe, and the development of a system for storage and recall of the resulting data, are described. A comparison is made with pressure drop data measured in a horizontal pipe conveying the same product, and conclusions drawn about the general suitability of pipes inclined downwards for pneumatic conveying, and the prediction of pressure drop in such pipes.

1. INTRODUCTION

This work arose as the result of a need to design a pipeline to convey a mineral product with a particle density of approximately 800kg/m^3 and a size range of 2 to 25mm, at a rate of 30 tonnes/hour along a route consisting of a straight line 800m long on a steadily falling gradient of 1 in 4. The design method normally used at The Wolfson Centre (described below) was employed for this project, and the analysis of the data gathered during the test work revealed some interesting, and possibly useful, phenomena to be occurring in the inclined test section.

2. SYSTEM DESIGN

The method used at The Wolfson Centre for design of conveying systems consists of testing the product which the system is to handle, to obtain data on the pressure drop experienced with the product under various flow conditions, then using this data to predict the performance of alternative types of pneumatic conveying systems for the duty, to select suitable system components (i.e. air mover, feeder, pipeline and air filter).

3. EXPERIMENTAL WORK

3.1 Equipment and technique

The data gathering consisted of conveying the product in a test loop of 80mm (3in.) nominal bore, which was instrumented to measure the pressure drop caused when the product flows along straight pipe sections, both horizontal and inclined; the test loop is shown in fig. 1. Pressure gradients in the straight sections were measured by means of pressure transducers and a computerised data logging system. The mass flow rate of product was obtained by monitoring the output of load cells on which the receiving hopper was mounted. Flow rate of air was controlled by a bank of choked flow nozzles; any combination from 8 sizes of nozzle in a x2 progression could be used, giving close control of air flow irrespective of line pressure.

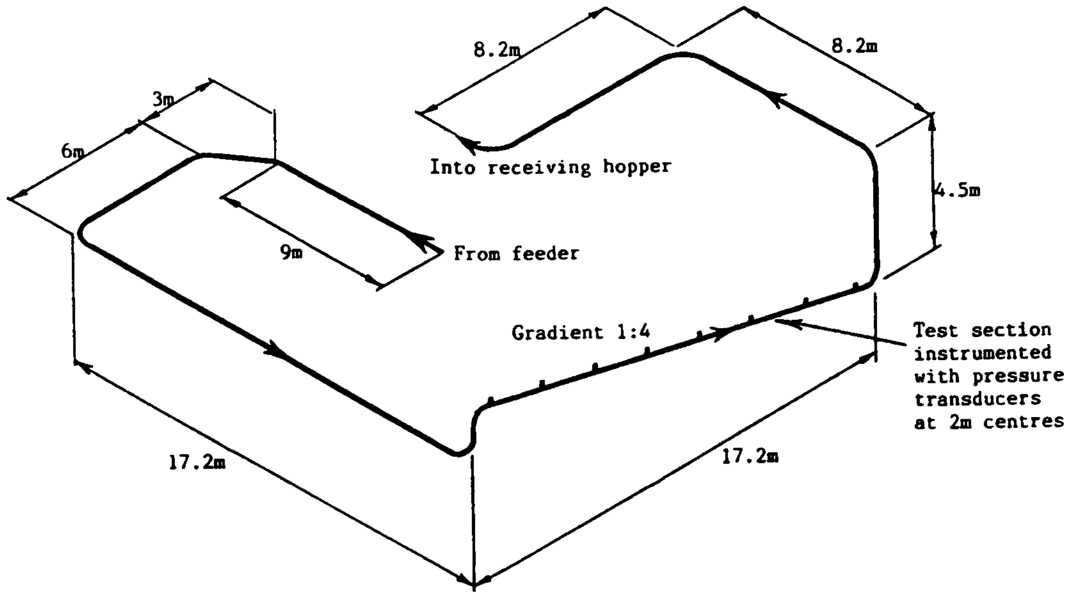

Fig. 1
The 80mm (3in.) n.b. test loop
and pressure tappings

The test pipeline was fed by means of a bottom discharge blow tank of 1.5m^3 capacity; batches of approximately 150 kg of product were used, which gave a running time of approximately three minutes depending on flow rate of product. The reason for the use of such small batch sizes was because of worries about the effects of product degradation; each batch could only be conveyed a limited number of times, and 150kg was the smallest batch size which gave a reasonable period of steady state operation, thus keeping the quantity of product required for testing within reasonable limits.

From the logged data taken at 2 second intervals over the duration of a test run, graphs of pressure versus time were plotted on the screen of the computer attached to the data logging system, and a steady state portion selected. Average pressures at the tappings and flow rate of product were calculated for this period, and plots of pressure versus distance produced, such as shown in fig.2. On these plots, the regions of fully-developed flow were identified (i.e. parts displaying a steady gradient, away from the curved pressure profiles caused by the re-acceleration of solids downstream of bends). To these portions, straight lines were fitted by means of a least-squares algorithm; this approach was used purely for convenience in dealing with the data on the computer and not for any stastical reason.

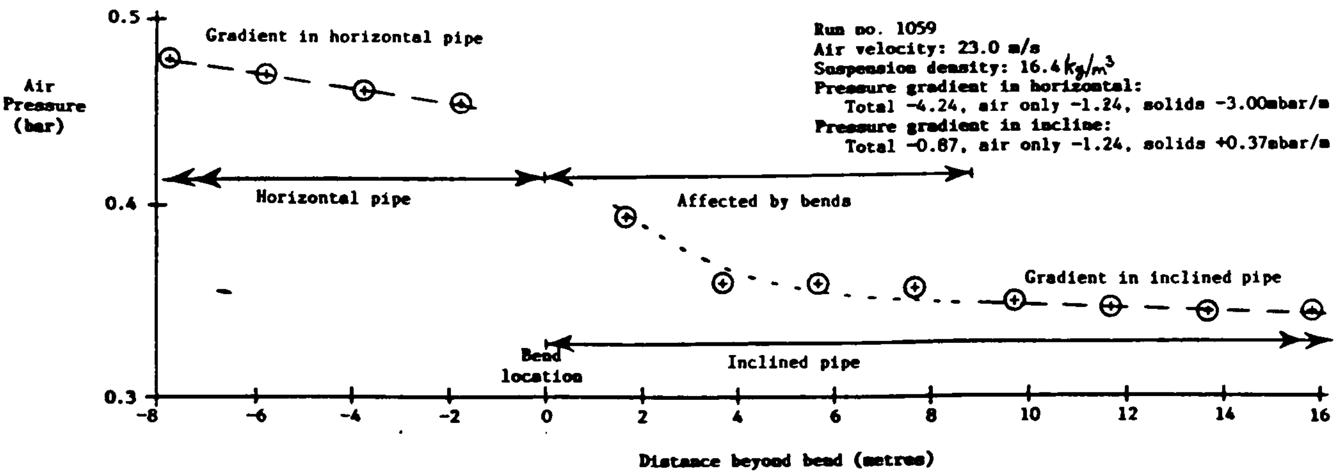

Fig. 2
Plot of pressure versus distance

The gradients of the straight lines fitted in the way described were calculated, these being the actual pressure gradients in the horizontal and inclined straight pipe sections.

3.2 Test programme

One batch of product was conveyed several times to ascertain the effect of degradation on the pressure drop measurements; from the results obtained with this, it was apparent that the pressure drop measured reduced markedly with every conveying run with one batch of product, especially during the first two or three runs. For this reason, subsequent batches were conveyed no more than three times and the data analysed to take account of the effect of degradation.

4. DATA ANALYSIS

4.1 For design purposes

For the purposes of system design, a data storage system was developed around the data resulting from the experimental work; this storage system consisted of a system of an equation and a graph relating the 'solids contribution' to the pressure gradient, the superficial air velocity and the suspension density of the flow; these terms are discussed below. The data from this system was recalled and used by a computer program which predicted the pressure drop along different proposed pipelines for the duty by working along taking step lengths, finding pressure gradient and hence pressure drop in the step length from suspension density and superficial air velocity, re-calculating these at the end of each step.

The 'solids contribution' to pressure gradient was calculated as the actual measured gradient less an 'air only' portion calculated using the normal Darcy expression for pressure drop in a pipe carrying air only; the superficial air velocity was calculated by taking the actual volume flow rate of air (from the known mass flow rate and the measured pressure in the section) and dividing it by the pipe cross sectional area; and the suspension density was calculated as the mass flow rate of product divided by the actual volume flow rate of air, which turns out to be far more useful in obtaining correlations than the alternative criterion of mass solids loading ratio (often wrongly referred to as 'phase density') often used. All these quantities are easily obtainable by calculation from measurable variables.

4.2 For this paper

Whilst the technique described above (discussed in more detail in refs. 1 and 2) sufficed for design of the pipeline for the duty required, it was apparent that a more detailed examination of the data may reveal some information of general interest regarding conveying down inclined pipes. To this end, the pressure gradients in the inclined and horizontal sections were compared for each test run.

4.2.1 Pressure recovery

It was apparent that the solids contribution to the pressure gradient in the inclined pipe was in some cases in opposition to the 'air only' gradient, i.e. the total gradient measured was less than if air alone had been flowing. This seems hardly surprising in itself, and also it seems hardly surprising that this 'pressure recovery' effect was more marked as the product became more degraded.

4.2.2 Comparison with horizontal pipe

The next step was to compare the solids contribution to the pressure gradients in the horizontal and inclined pipes directly; the reduction in solids contribution in the inclined pipe, compared with the solids contribution in the horizontal pipe for the same test run, was calculated and this was compared with the reduction which might be expected from simply taking account of the static head gain due to the density of the flowing suspension acting over a change in height (fig.3).

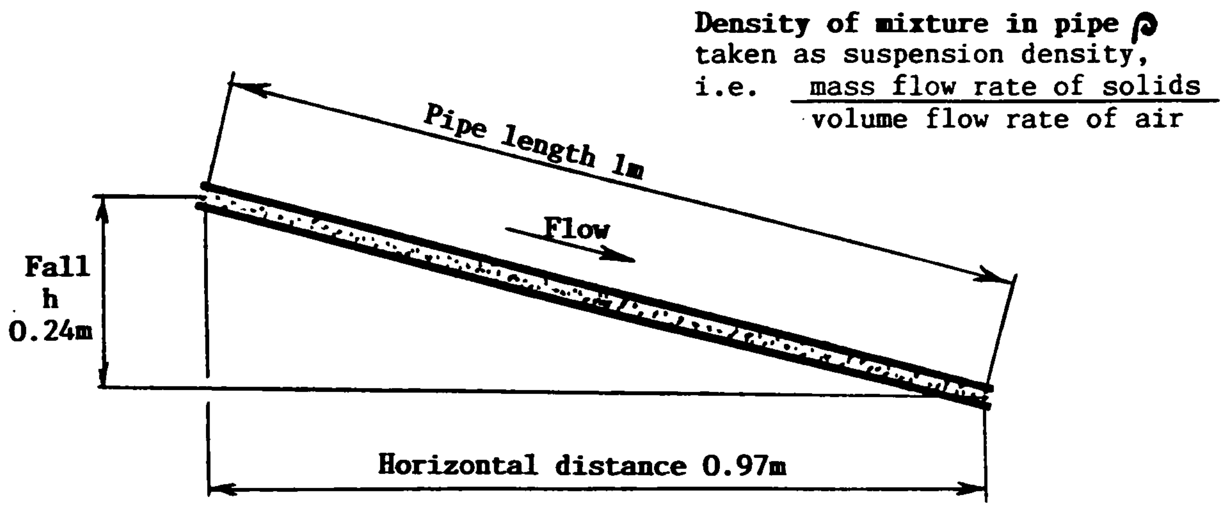

Fig. 3
The model used for calculating the static head effect

The ratio $\dfrac{\text{[reduction in solids contribution to pressure gradient]}}{\text{[reduction expected because of static head effect]}}$,

(referred to here as the Pressure Recovery Ratio), was calculated, and found to be in all cases greater than 1; it increased with increasing product degradation and also with increasing air velocity. Perhaps it should be emphasised at this point that an increase in this ratio means a greater reduction in pressure loss than predicted from considering the static head effect, i.e. a greater saving in conveying cost than would be expected from this simple consideration.

4.2.3 Degradation of product

It was no surprise to find that the pressure gradients were reducing with the number of times a batch of product had been conveyed, given that fairly serious degradation had occurred after some seven runs, to the point where the originally granular product was reduced to a fine powder. The trend is shown in fig. 4. This meant that to obtain a true comparison would involve comparing only the first runs with each batch with each other, then comparing only the second runs, and finally comparing only the third runs (later batches were conveyed only 2 or 3 times).

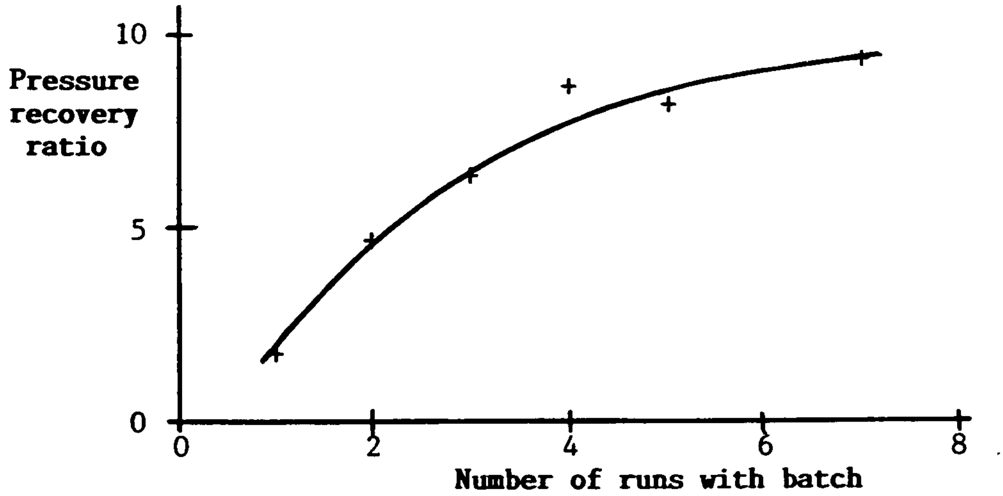

Fig. 4
Effect of product degradation on pressure recovery ratio

4.2.4 Effect of air velocity

A graph of the pressure recovery ratio versus superficial air velocity is shown in fig. 5 below. This clearly shows the effect of product degradation, and comparing all the first runs, the second runs and the third runs, demonstrates the increase of the ratio with velocity for runs done using equal products.

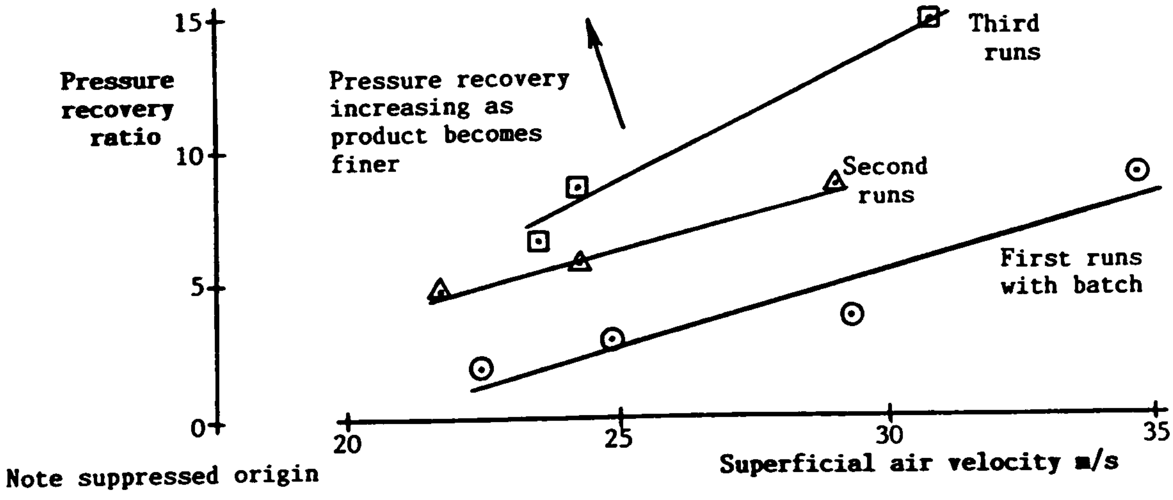

Fig. 5
Relationship between pressure recovery ratio and air velocity

There seems no obvious reason why the pressure recovery ratio should increase with increasing velocities, although a number of possible mechanisms could be postulated. However, the result is that when designing a pipeline on a long falling gradient, the pressure drop will actually be lower with higher velocities, at least in the range tested, provided that there are few bends (the pressure drop caused by bends always increases with higher velocities).

5. CONCLUSIONS

5.1 Pressure recovery and its effect

It is fairly clear from the above that a significant amount of pressure recovery can be achieved in conveying through a pipe inclined downwards; in the application for which the test work described here was undertaken, it resulted in a considerable reduction of conveying cost in comparison with a horizontal pipeline of the same length.

5.2 Prediction of the pressure recovery

It has been seen that the amount of pressure recovery may be expected to be well in excess of that indicated by a consideration of the static head gain along the incline, and that this is particularly so with a fine product; but it appears that it may not be predicted easily from a simple model, and that it is extremely dependent on the exact quality of the product conveyed; these facts make it essential to carry out test work to determine the extent of the effect if designing an inclined section into a pipeline. This test work can only be performed in the type of rig described, where the pressure gradient in the inclined section can be measured directly; as has been demonstrated, these measurements can be carried out concurrently with measurements of the pressure gradients in the horizontal pipes using the set-up described, minimising the quantity of test work required.

6. REFERENCES

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