SOME TECHNICAL AND COMMERCIAL ASPECTS OF FLUIDISED BED GAS-TO-GAS HEAT EXCHANGERS

by

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SUMMARY

The work reported in this thesis concerns the development of a novel fluidised bed gas-to-gas heat exchanger patented by Stone-Platt Fluidfire Ltd. This joint University/industry project aimed not only to cover some of the technical aspects of the feasibility and design, but also to search for applications for the heat exchanger where it could be marketed successfully.

The novelty of the heat exchanger lies largely with the distributor plate which moves the fluidised bed across its surface because the fluidising gas emerges with a significant transverse component of velocity. A review of the literature established that no similar system had been reported and that previous flowing bed studies were not directly relevant. A small experimental heat exchanger was built at the University to test the feasibility of the ideas contained in the patent and this heat exchanger performed sufficiently well to justify further work.

A theoretical model was also developed to provide an accurate measure from the experimental data of the effectiveness of the unit. The particular nature of the heat exchanger means that there is an inherent leakage path between the two gas streams and the experimental unit was modified to enable this leakage to be measured. Information relating to the present uses of gas-to-gas heat exchangers was collected together with details of existing products. This enabled the possible market openings for the new fluidised bed unit to be identified and the maximum cost which would allow the unit to be sold into those markets was also established.

Overall the development of the heat exchanger was judged to have demonstrated the concept and it is thought to merit the application of further resources to obtain a commercial product, since a market appears to exist for such a system.

The author wishes to thank his team of supervisors; Dr. J.R. Howard, Dr. A.J. Cochran, Mr. M.J. Virr and Mr. S.N. Woodward who all contributed significantly to this investigation.

GAS-TO-GAS HEAT EXCHANGER FLUIDISED BED WASTE HEAT RECOVERY

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NOMENCLATURE

Variables

A	cross sectional area of bed
Bi	Biot number $Bi = h_p d_p / k_p$
C	specific heat at constant pressure
C_D	drag coefficient
D	distributor plate hole diameter
d	diameter
E	efficiency
$E(\vartheta)d\vartheta$	fraction of particles passed between ϑ and $\vartheta + d\vartheta$
$F(\vartheta)$	fraction of particles passed in ϑ
G	surface area of particles per unit area of distributor plate
g	acceleration due to gravity
h	heat transfer coefficient
K	constant in equation (4.8)
k	thermal conductivity
$\dot{m}$	mass flow
n	number of revolutions
Nu	Nusselt number $Nu = h_p d_p / k_g$
P	distributor plate hole pitch
Δp	pressure drop
Re	Reynolds number $Re = \rho_g U_f d_p / \mu_g$
T	temperature
ΔT	temperature difference
U	velocity
Z	bed depth

α	constant in equation (6.1)
δ	temperature difference between gas and particles
ε	voidage
η	effectiveness
ϑ	dimensionless time
μ	viscosity
ρ	density
φ	particle sphericity
ψ	thermal inequilibrium

Subscripts

air	between the two air streams
b	bed
c	cold
d	distributor
f	fluidising
g	gas
h	hot
i	inlet
mf	minimally fluidised
o	outlet
p	particle
s	surface
T	terminal

CHAPTER ONE

INTRODUCTION

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1.1 GENERAL REMARKS

The phenomenon of fluidisation has been known for more than a century with Robinson filing the first patent in the field [1]. The first important use of fluidised beds was in 1926 when a coal gasifier was built in Germany by Winkler (see [2]). Since then the petrochemical industries have made extensive use of the properties of fluidised beds in the development of the catalytic cracker from 1942 onwards [2]. Work on fluidised bed combustion was begun in 1961 by Elliott at the Central Electricity Generating Board and by Wright, Thurlow and Hoy at the British Coal Utilisation Research Association [3].

The major advantages of fluidised beds in chemical engineering relate to the large surface area of the particles and the good contact between fluid and solid. Hence there is the potential for rapid chemical reactions and heat transfer between the fluid and the particles. The heat transfer rate between the bed and a surface immersed in it is also large so heat can be readily added to or withdrawn from the bed [4].

A bed of particles can be fluidised with either a liquid or a gas. Gas fluidisation is distinguished by the formation of bubbles over part of the fluidising velocity range. The bubbling action promotes particle mixing which in turn results in a uniform temperature throughout the bed except very close to the walls and the distributor plate.

The development of fluidised bed combustors in recent years has now led to fluidised bed boilers being offered commercially. The fuel for these can range from gas, oil and coal through to such poor fuels as garbage. Steam is raised in tubes immersed in the bed thus allowing high combustion intensities without the bed temperature exceeding the melting point of the ash.

Practically all of the early work on fluidised beds was conducted with deep beds [5]. Only relatively recently have the characteristics of shallow beds been explored [6], and they have been found to have certain practical advantages over deep beds. There are two key differences between deep and shallow beds. Firstly the pressure drop is lower through a shallow bed so less pumping power is needed for fluidisation. Secondly shallow beds do not contain the large bubbles which are present in deep beds, so the bed is more nearly homogeneous. This leads to improved heat transfer within the bed and also to immersed surfaces.

1.2 WASTE HEAT RECOVERY

The shallow fluidised bed has now been developed for several applications including heat recovery from hot waste gases. With the ever increasing cost and insecurity of supply of fuel, especially oil, it has become necessary to reduce the wastage of energy. One major source of wasted heat is hot exhaust gas from combustion systems of all types. Waste heat recovery systems transfer this heat from the hot exhaust to another fluid.

One necessary characteristic of such systems is that they should present as low a pressure drop to the exhaust as possible to minimise the pumping power needed. Therefore the development of shallow fluidised beds has enabled them to be used in this field whereas deep beds would have had too great a pressure drop. Waste heat recovery units based on shallow beds are now commercially available for transferring heat to steam or pressurised hot water. These are more compact than conventional convective units because the bed-to-immersed surface heat transfer coefficient is up to ten times greater than that for air-to-surface [4].

However it is also desirable to transfer waste heat from an exhaust gas to another gas stream in a gas-to-gas heat exchanger. The existing types of such exchangers all have certain disadvantages, particularly in that they have large fabricated sections which may quickly corrode when exposed to hot exhaust gases. An alternative approach which exploits some of the advantages of a shallow fluidised bed of inert particles was proposed and patented by Elliott and Virr in 1978 [7]. This patent is now owned by Stone-Platt Fluidfire Ltd. and is the subject of this thesis.

1.3 FORMAT OF THE PROJECT

The project was initiated by Stone-Platt Fluidfire Ltd. who wished to see the ideas contained in the patent studied with the aim of developing a new product. The project was conducted under the aegis of the Interdisciplinary Higher Degree Scheme (I.H.D.) and the problem was approached from both technical and commercial standpoints. Since the company market only fluidised bed equipment they were able to provide invaluable expertise, but the detailed studies of the heat exchanger were performed at the University. The I.H.D. Scheme provides an ideal environment for joint industry/university projects of this type.

A successful product cannot be designed solely on technical grounds: it is also vital to have accurate knowledge of the market place. Hence the project moved along two parallel courses intended to establish if the ideas were both feasible and saleable. The coordinating role of I.H.D. enabled assistance to be drawn from engineering and marketing disciplines as well as from the company. This arrangement laid the foundations of the project and its objectives are listed below.

1.4 OBJECTIVES OF THE PRESENT WORK

The objectives of the work reported in this thesis were as follows:-

(a) to study the technical feasibility of the novel system outlined in the patent;

(b) to identify the waste heat recovery market sectors in which this product is likely to be competitive;

and

(c) to develop the criteria necessary for the design of an industrial scale prototype.

CHAPTER TWO

REVIEW OF LITERATURE

CHAPTER TWO

REVIEW OF LITERATURE

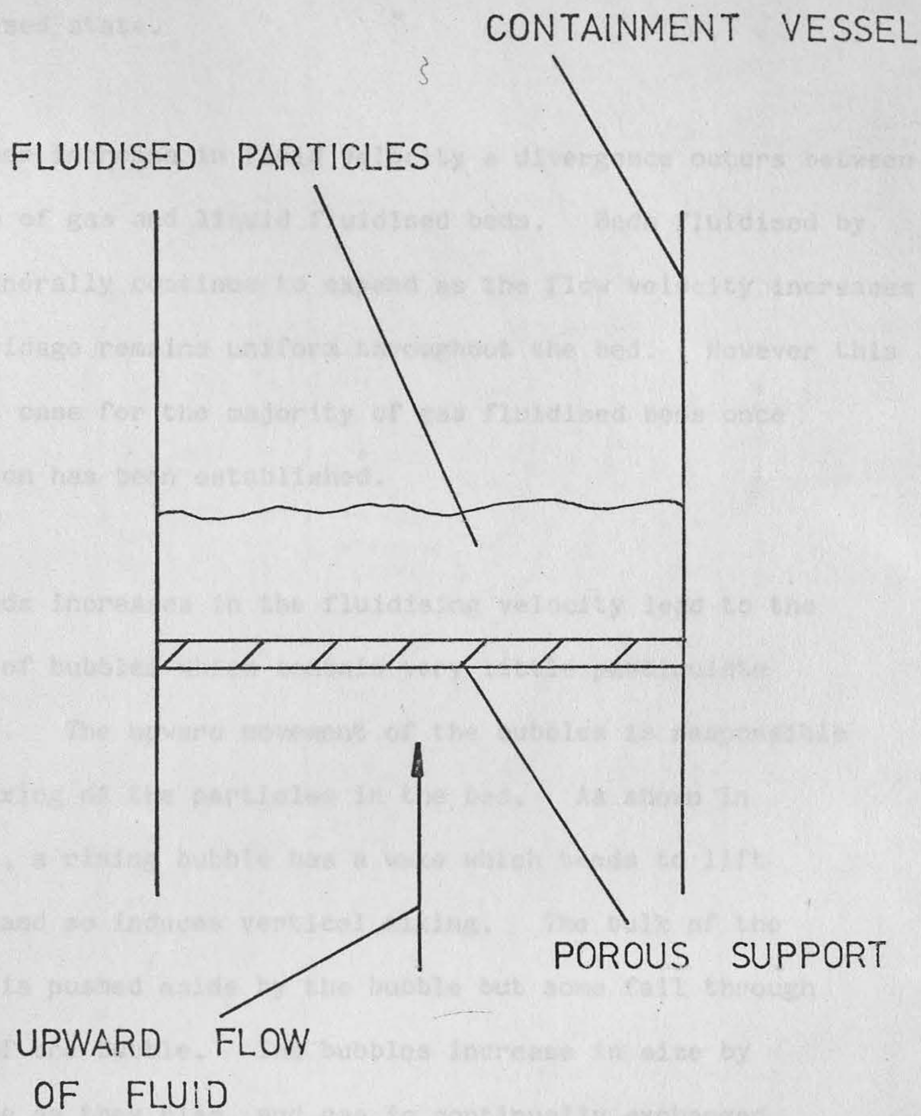
2.1 INTRODUCTION

This review studies the existing work relevant to the development of the gas-to-gas fluidised bed heat exchanger. Firstly there is a description of the general properties of fluidised beds. Then follow four sections dealing with the aspects of fluidised beds most important to the project. These are gas-to-particle heat transfer, the flow of fluidised solids, gas jets in fluidised beds, and electrostatic inter-particle effects.

2.2 GENERAL PROPERTIES OF FLUIDISED BEDS

A fluidised bed can be considered as a mass of particles supported by a moving fluid as shown in Figure 2.1. The bed of particles is contained in a vessel with a porous base through which a fluid flows upward. At low flow rates this fluid permeates the voids between the particles. As the fluid velocity increases, so does the drag force acting on the particles as the fluid rises. At some velocity the drag force on the smallest particles becomes equal to their weight and they become supported on the flow. Slight re-arrangement of the particles can then occur, and as the velocity increases progressively more of the particles become supported. The bed expands with increasing fluid flow because the re-arrangement of the particles causes the inter-

FIGURE 2.1 SCHEMATIC DIAGRAM OF A FLUIDISED BED



particle voids to increase in size. Eventually sufficient of the particles are fully supported by the fluid that they have significant freedom for random movement. This is the onset of the fluidised state.

With further increase in fluid velocity a divergence occurs between the nature of gas and liquid fluidised beds. Beds fluidised by liquids generally continue to expand as the flow velocity increases and the voidage remains uniform throughout the bed. However this is not the case for the majority of gas fluidised beds once fluidisation has been established.

In such beds increases in the fluidising velocity lead to the formation of bubbles which contain very little particulate matter [8]. The upward movement of the bubbles is responsible for the mixing of the particles in the bed. As shown in Figure 2.2, a rising bubble has a wake which tends to lift particles and so induces vertical mixing. The bulk of the particles is pushed aside by the bubble but some fall through the roof of the bubble. The bubbles increase in size by coalescence as they rise, and gas is continually exchanged between the bubble and the surrounding emulsion phase as shown in Figure 2.3. The bubble may be visualised as being surrounded by a cloud, and the interchange of gas occurs in this cloud.

When the bubbles become comparable in size with the containment, the bed is in the slugging régime [9]. There is then less

FIGURE 2.2 PARTICLE MOTION AROUND A RISING BUBBLE

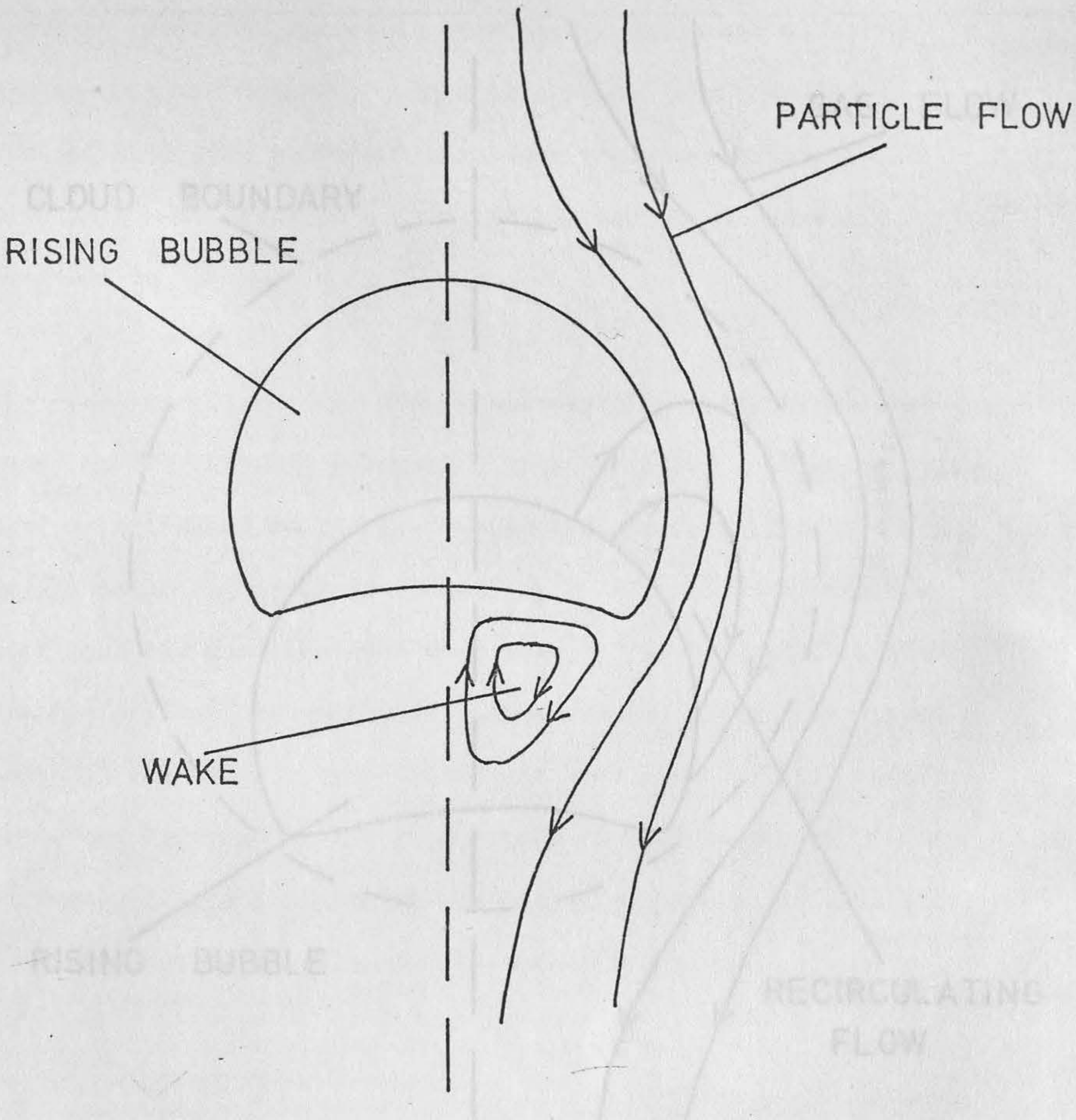
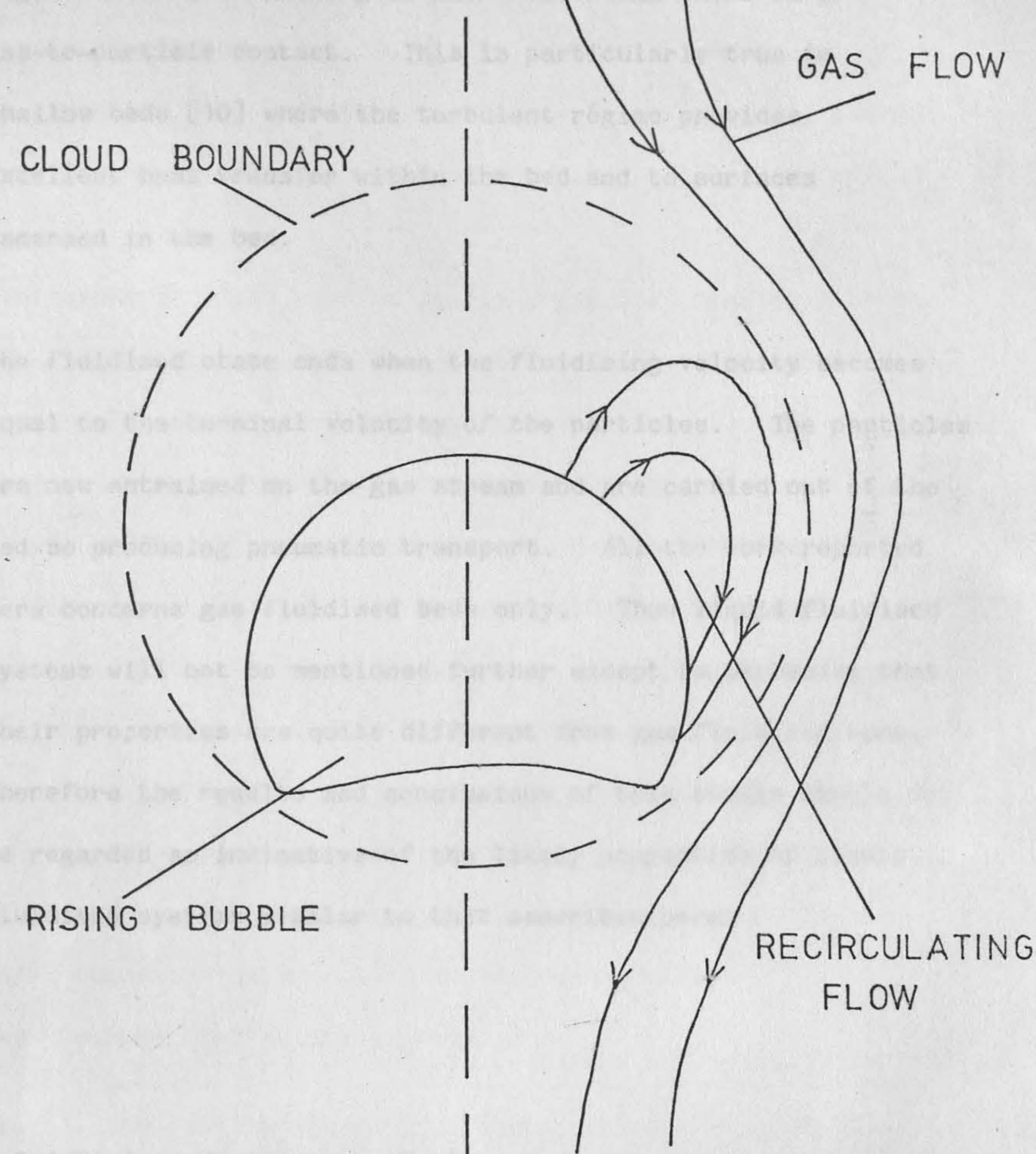


FIGURE 2.3 GAS INTERCHANGE BETWEEN BUBBLE AND EMULSION



contact between gas and particles, but when the fluidising velocity is increased still further we enter the turbulent régime. Here the mixing is much better and there is good gas-to-particle contact. This is particularly true in shallow beds [10] where the turbulent régime provides excellent heat transfer within the bed and to surfaces immersed in the bed.

The fluidised state ends when the fluidising velocity becomes equal to the terminal velocity of the particles. The particles are now entrained on the gas stream and are carried out of the bed so producing pneumatic transport. All the work reported here concerns gas fluidised beds only. Thus liquid fluidised systems will not be mentioned further except to emphasise that their properties are quite different from gas fluidised beds. Therefore the results and conclusions of this thesis should not be regarded as indicative of the likely properties of liquid fluidised systems similar to that described here.

2.2.1 Minimum fluidising velocity

This parameter is the gas velocity at which the fluidised state is considered to begin. It is a most important characteristic of a fluidised bed, yet it is extremely difficult to predict accurately. The minimum fluidising velocity, U_{mf}, is often defined as that velocity at which the pressure drop developed across the bed by the passage of the gas becomes equal to the weight of the bed per unit area of the distributor plate. U_{mf}, like all the other fluidising velocities mentioned here, is expressed as a superficial velocity, the flow being averaged over the cross-sectional area of the bed. That is $U_f = \dot{m}_g / \rho_g A$.

Real systems do not have an easily definable transition to the fluidised state as shown in Figure 2.4. The transition is gradual for a number of reasons, the most important of which is that all the particles will normally be different sizes and shapes and so the drag force on each of them is different for a given U_f. Hence not all the particles become fully supported simultaneously and there is a range of U_f before the entire bed is in the fluidised state.

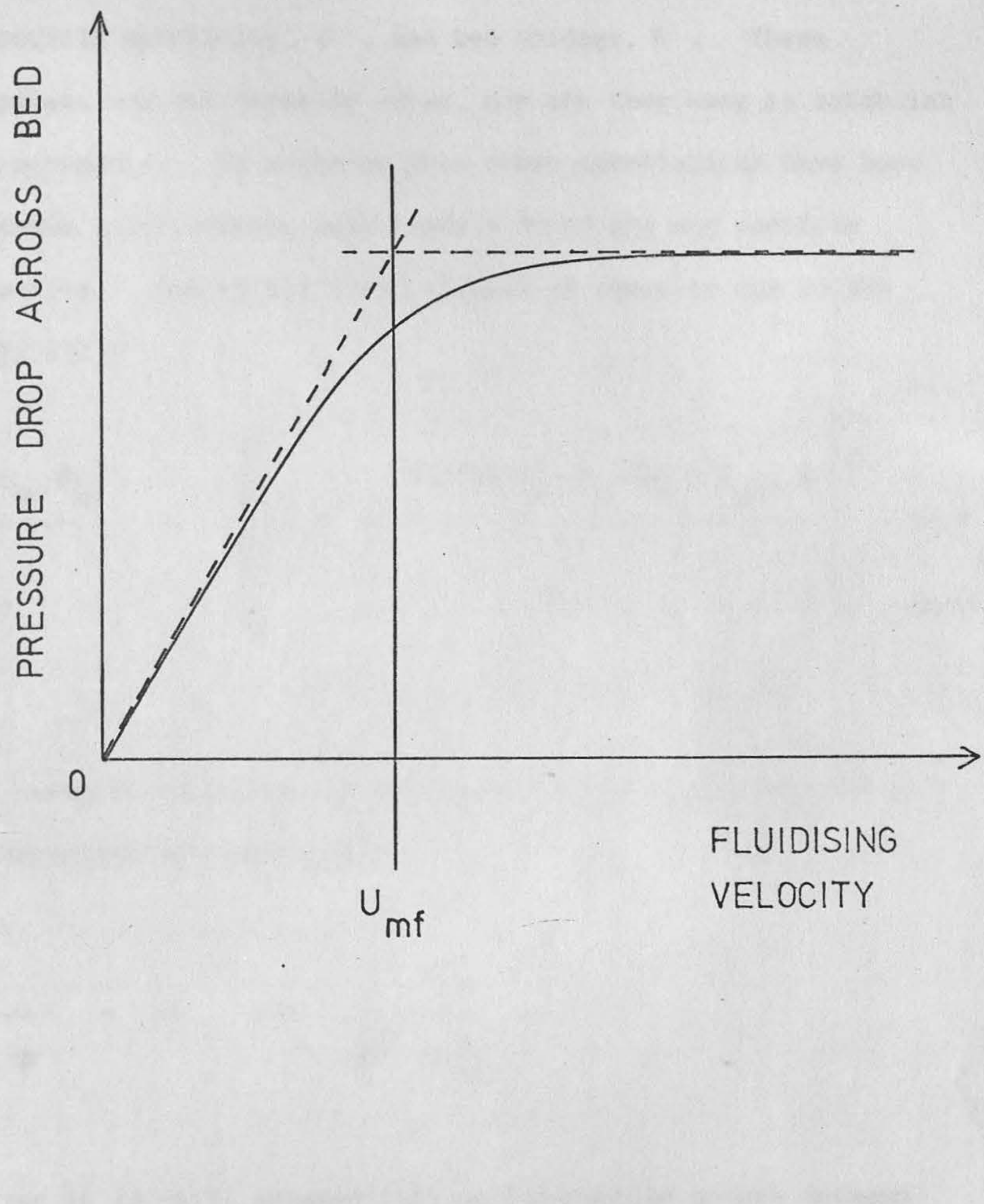
Many attempts have been made to predict U_{mf} from a knowledge of the gas and particle properties. The results of these yield complex equations such as that due to Ergun [11] for the pressure drop across a packed bed:

$$\frac{\Delta p_b}{Z} = \frac{150\,(1-\varepsilon)^2 \mu_g U}{\varepsilon^3\,\varphi^2\,d_p} + \frac{1.75\,(1-\varepsilon)\,\rho_g U^2}{\varepsilon^3\,\varphi\,d_p} \qquad (2.1)$$

This is related to U_{mf} by assuming that the pressure drop across the bed equals the weight of the bed per unit base area, that is

$$\Delta p_b = Z_{mf}\,(1-\varepsilon_{mf})\,(\rho_p - \rho_g)\,g \qquad (2.2)$$

FIGURE 2.4 PRESSURE DROP CHARACTERISTIC OF A FLUIDISED BED



Hence combining equations (2.1) and (2.2) leads to a prediction of U_{mf}.

The difficulty with this approach is that it requires knowledge of particle sphericity, φ, and bed voidage, ε. These parameters are not normally known, nor are they easy to establish by experiments. To overcome this other correlations have been suggested which contain only readily found gas and particle properties. One of the more reliable of these is due to Wen and Yu [12]:

$$\frac{d_p\, U_{mf}\, \rho_g}{\mu_g} = \left[33.7^2 + \frac{0.0408\, d_p^3\, \rho_g\, (\rho_p - \rho_g)\, g}{\mu_g^2} \right]^{\frac{1}{2}} - 33.7 \qquad (2.3)$$

This correlation avoids the problems of that of Ergun by using the experimental results [2]

$$\frac{1}{\varepsilon_{mf}^3\, \varphi} \sim 14 \quad \text{and} \quad \frac{1 - \varepsilon_{mf}}{\varphi^2\, \varepsilon_{mf}^3} \sim 11.$$

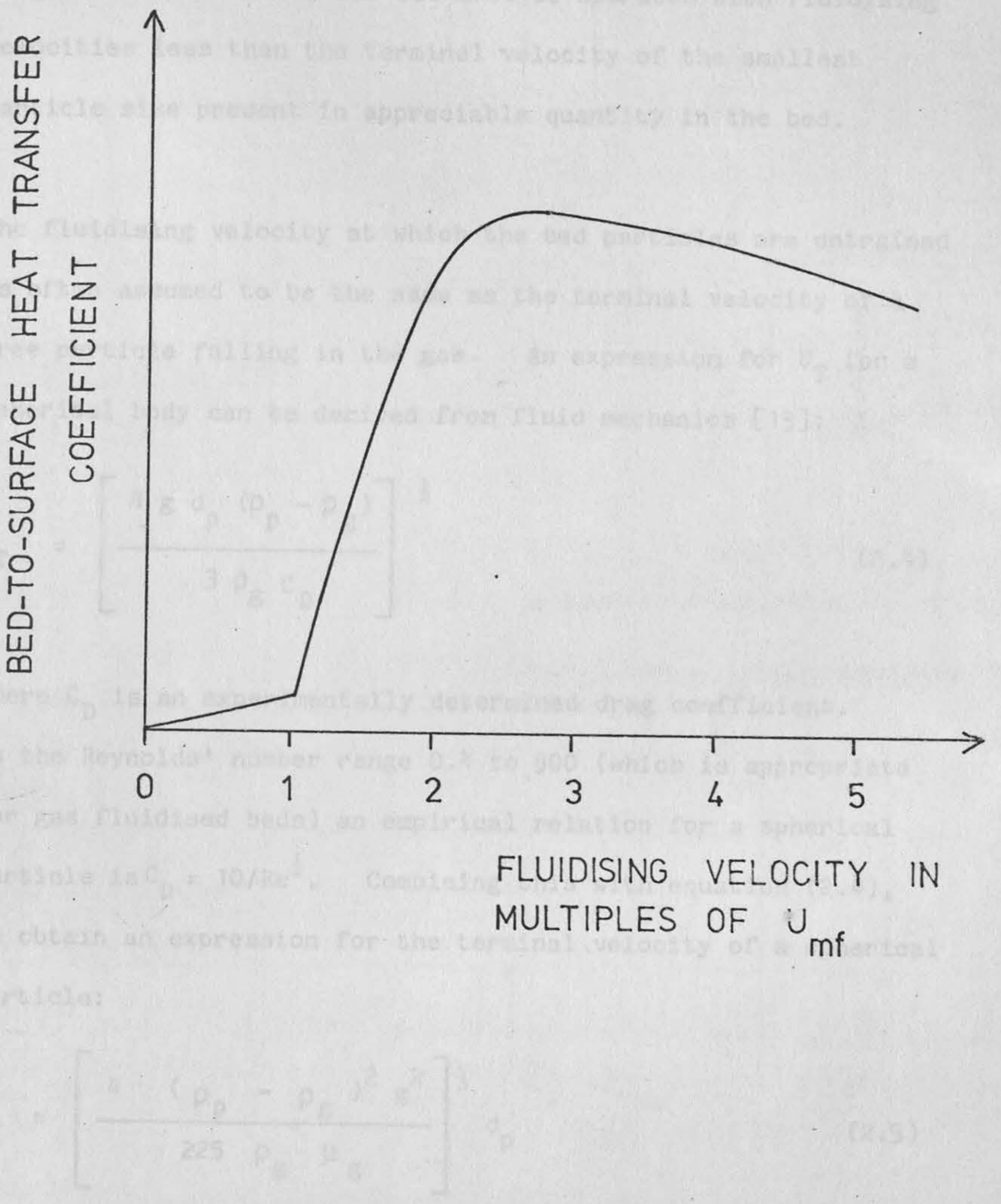
However it is still assumed that no interaction occurs between the particles as the equations are based on results from packed rather than fluidised beds. This assumption is reflected in the inaccuracy of equation (2.3) for Wen and Yu report that their correlation has a standard deviation of 34% in the range Re = 0.001 to 4000. Another important point is that the majority of the proposed correlations, including that of Wen and Yu, are only applicable at room temperature and pressure.

In practice there is no reliable alternative to an experimental measurement of U_{mf} for each particular particle batch. This is a quite simple experiment at atmospheric pressure and will lead to a much more useful result than the application of any of the many correlations. At high temperatures and pressures the experiment is obviously more elaborate and this may rule out an experimental determination of U_{mf}. Fortunately it is rarely necessary to know U_{mf} precisely as most beds operate at fluidising velocities several times the minimum so that heat transfer from the bed to a surface is maximised [4]. The maximum value of the bed-to-surface heat transfer coefficient is usually attained at 2 to 3 U_{mf} as shown in Figure 2.5.

2.2.2 Entrainment velocity

The opposite end of the spectrum of fluidised bed behaviour is defined by the entrainment velocity of the particles. The problems of definition here are similar to those encountered in the discussion of the minimum fluidising velocity. The terminal velocity, U_T, of a particle is dependent on its sphericity. Obviously U_T is least for the smallest particles,

FIGURE 2.5 VARIATION OF HEAT TRANSFER COEFFICIENT WITH FLUIDISING VELOCITY



and in practice beds will normally contain a range of particle sizes and shapes. It is impractical to operate below the U_T of the tiny quantity of fine dust present in most beds as the disengagement space above the bed would have to be very large (Figure 2.6). Rather the bed must be operated with fluidising velocities less than the terminal velocity of the smallest particle size present in appreciable quantity in the bed.

The fluidising velocity at which the bed particles are entrained is often assumed to be the same as the terminal velocity of a free particle falling in the gas. An expression for U_T for a spherical body can be derived from fluid mechanics [13]:

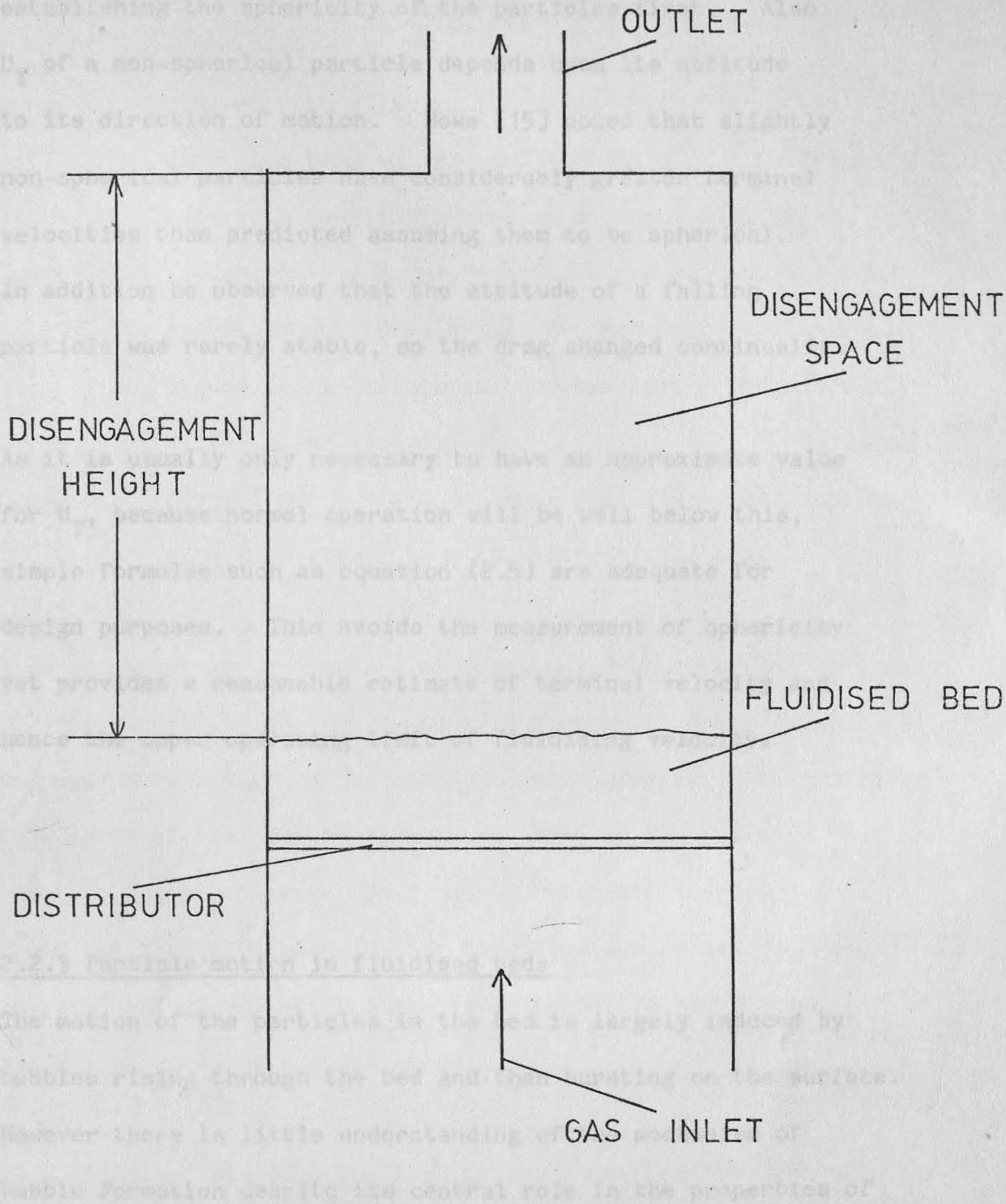
$$U_T = \left[\frac{4 \, g \, d_p \, (\rho_p - \rho_g)}{3 \, \rho_g \, C_D} \right]^{\frac{1}{2}} \qquad (2.4)$$

where C_D is an experimentally determined drag coefficient. In the Reynolds' number range 0.4 to 500 (which is appropriate for gas fluidised beds) an empirical relation for a spherical particle is $C_D = 10/Re^{\frac{1}{2}}$. Combining this with equation (2.4), we obtain an expression for the terminal velocity of a spherical particle:

$$U_T = \left[\frac{4 \quad (\rho_p - \rho_g)^2 \, g^2}{225 \, \rho_g \, \mu_g} \right]^{\frac{1}{3}} d_p \qquad (2.5)$$

However, real particles are not spherical, especially when refractory oxides such as alumina and silica are considered. A graphical method for determining U_T for non-spherical

FIGURE 2.6 PARTICLE DISENGAGEMENT SPACE



particles has been developed by Brown [14]. This requires reference to his chart and is not an analytical solution to the problem. Even with this there is the difficulty of establishing the sphericity of the particles first. Also U_T of a non-spherical particle depends upon its attitude to its direction of motion. Howe [15] noted that slightly non-spherical particles have considerably greater terminal velocities than predicted assuming them to be spherical. In addition he observed that the attitude of a falling particle was rarely stable, so the drag changed continually.

As it is usually only necessary to have an approximate value for U_T, because normal operation will be well below this, simple formulae such as equation (2.5) are adequate for design purposes. This avoids the measurement of sphericity yet provides a reasonable estimate of terminal velocity and hence the upper operating limit of fluidising velocity.

2.2.3 Particle motion in fluidised beds

The motion of the particles in the bed is largely induced by bubbles rising through the bed and then bursting on the surface. However there is little understanding of the mechanism of bubble formation despite its central role in the properties of fluidised beds. One approach has been to classify particles according to their bubbling behaviour. This has been done by Geldart [16] who classified particles into four broad groups as shown in Figure 2.7 [17]:-

Group A: Particles with densities less than about 1400 kg m^{-3} and size in the range 20 to 100 μm. These powders expand considerably beyond U_{mf} before bubbling commences.

Group B: Powders with sizes in the range 40 to 500 μm and densities between 1400 and 4000 kg m^{-3}. For these materials bubbling starts at or just above U_{mf}.

Group C: Powders in which inter-particle forces dominate such as wet or sticky materials or ones where agglomeration occurs due to electrostatic charging. These are very difficult to fluidise.

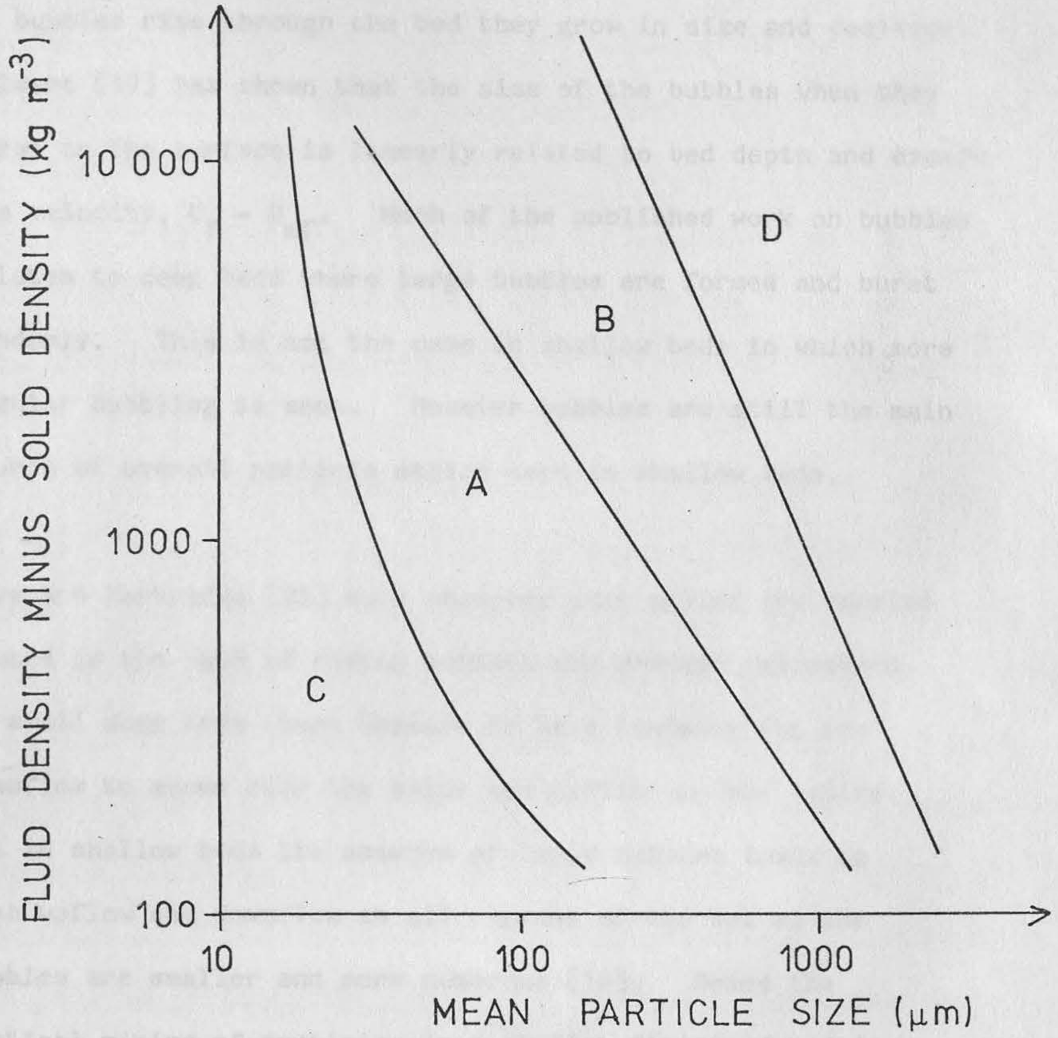
Group D: Large (over 600 μm) or dense particles where the bubbles rise more slowly than the interstitial gas.

Beds of materials of Groups C or D are unlikely to behave as true fluidised beds and are not considered further. Group A particles are used extensively in the petrochemical industry in fluidised beds of catalysts, but these are too light to be of use in heat recovery systems because their entrainment velocity is so low. Hence the particles used in this investigation behave as Group B powders, though Geldart's classification should not be used rigidly but only as a rough guide.

In a fluidised bed of Group B particles there are two distinct phases. These are the bubble phase which contains little or no solid material, and the emulsion phase with a much higher solids' loading. Gas passes through the bed in both these

FIGURE 2.7 PARTICLE CLASSIFICATION FOR FLUIDISATION

from Baeyens and Geldart [17]



phases and there is an interchange of gas between them. Experiments by Davidson and Harrison [18] suggest that all the gas in excess of that needed just to fluidise the bed passes through as bubbles, while the emulsion remains at minimally fluidised conditions.

As bubbles rise through the bed they grow in size and coalesce. Geldart [19] has shown that the size of the bubbles when they burst on the surface is linearly related to bed depth and excess gas velocity, $U_f - U_{mf}$. Much of the published work on bubbles relates to deep beds where large bubbles are formed and burst randomly. This is not the case in shallow beds in which more regular bubbling is seen. However bubbles are still the main source of overall particle motion even in shallow beds.

Rowe and Partridge [20] have observed that solids are carried upward in the wake of rising bubbles and downward elsewhere. In small deep beds there appears to be a tendency for the downflow to occur near the walls and upflow at the centre. But in shallow beds the absence of large bubbles leads to both upflow and downflow in all regions of the bed as the bubbles are smaller and more numerous [10]. Hence the vertical mixing of particles in a shallow fluidised bed is generally good and the bed is in a turbulent state.

Diffusion models have been used to quantify the rate of mixing in beds. Vertical diffusion coefficients have been measured by Lewis et al. [21] by measuring heat transport in the bed.

They assumed that all the vertical heat transport was due to vertical solids' movement alone and hence could establish a vertical mixing diffusion coefficient. This they found to be linearly related to fluidising velocity but independent of the size of the bed. The measurement of an analogous coefficient for lateral mixing has been carried out by Mori and Nakamura [22]. They observed the mixing of different coloured particles in an initially partitioned bed. Comparison of their results and those of Lewis et al. suggests that vertical mixing is some five to ten times faster than lateral mixing. This result is of importance when we consider the residence time distribution of particles in flowing fluidised beds.

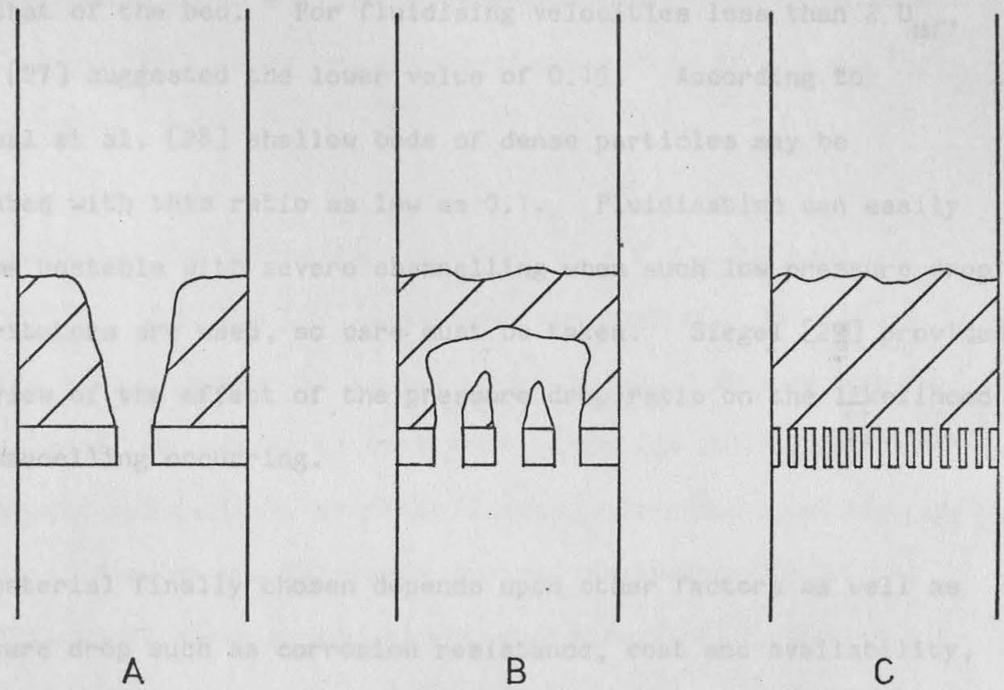
Particle motion is also affected by the size distribution of the particles in the bed. For commercial units it is necessary to use readily available bed materials rather than specially selected narrow size cuts, so the importance of the effects of size range is perhaps greater here than in laboratory investigations. The principal effect is segregation; that is the settling out of the larger particles which tend to form a stagnant layer on top of the distributor. This can affect the performance of the distributor and could be particularly troublesome in the heat exchanger where the distributor is essential to generate the solids' flow.

Some work has been done on binary mixtures of differently sized particles [23, 24], but these are quite different from a broad, continuous size distribution of particles. The most useful work is perhaps that of Boland [25] among whose conclusions are the following points. Firstly bubbling beds give good mixing if all the constituents are fully fluidised. Secondly the degree of segregation depends on the standard deviation of the particle size distribution. Lastly he concluded that segregation was negligible for a standard deviation below 50 μm. Industrially available particles have larger size ranges with deviations nearer 100 μm. The shallow bed in the heat exchanger, however, should be sufficiently turbulent to prevent segregation occurring despite the broader size distribution.

2.2.4 Distributor plate

The purpose of the distributor plate is to provide even fluidisation and to support the bed when it is slumped. In shallow beds the latter is easy to achieve as they are not heavy. To obtain uniform fluidisation the distributor must provide equal gas flows to all areas of the bed and should do this with as low a pressure drop as possible. The quality of fluidisation is dependent on the type of distributor used as shown in Figure 2.8. If it has only a few openings the bed may channel or slug and the overall bed density then fluctuates wildly. This is particularly so in shallow beds where there is insufficient height for lateral gas diffusion to occur. Hence for shallow beds distributors with many holes must be used which restricts the distributor to porous materials and plates with many small holes.

FIGURE 2.8 QUALITY OF FLUIDISATION



A. Single opening allows channelling

B. Few openings produce large bubbles

C. Many small holes allow even fluidisation

To get even fluidisation various "rules of thumb" have been developed which relate the pressure drop through the distributor to that through the bed. In deep beds Whitehead [26] has suggested that the distributor pressure drop should be at least 0.4 that of the bed. For fluidising velocities less than 2 U_{mf}, Hiby [27] suggested the lower value of 0.15. According to Agarwal et al. [28] shallow beds of dense particles may be operated with this ratio as low as 0.1. Fluidisation can easily become unstable with severe channelling when such low pressure drop distributors are used, so care must be taken. Siegel [29] provides a review of the effect of the pressure drop ratio on the likelihood of channelling occurring.

The material finally chosen depends upon other factors as well as pressure drop such as corrosion resistance, cost and availability, particularly when commercial development is anticipated. The guidelines above are only indications of when uneven fluidisation may occur, and practical experiments are the only sure way to test a particular distributor. It should also be noted that quality control is difficult in the manufacture of distributors, and supposedly identical plates may differ substantially in actual performance.

2.2.5 Heat transfer to immersed surfaces in fluidised beds

The heat transfer properties of fluidised beds are those which are exploited the most. Two models of the mechanism of bed-to-surface heat transfer have been developed. The first, due to Mickley and Fairbanks [30], considers small volumes or "packets" of gas and solid which approach the surface. These then stay at the surface and heat transfer occurs until they move away and are replaced at the surface by other packets. In effect this theory assumes that the thermal properties of the packet of gas and solid are uniform.

To account for the heterogeneous nature of a fluidised bed, Botterill and Williams [31] considered the heat transfer between a single sphere surrounded by gas to a surface. This has since been extended to a model involving two particles which takes into account gas and particle properties [32]. This model is superior to that of Mickley and Fairbanks when the residence time of the packet is short which is the more useful situation. However the packet theory has been improved by Kubie and Broughton [33] by the physically justified inclusion of a boundary layer at the surface.

Many correlations have been developed which attempt to predict the heat transfer coefficient between the bed and an immersed surface, h_s. These vary considerably as workers used different distributors, bed depths and particle shapes. As an approximate correlation for Geldart's Group B [16], Botterill [34] recommends the following due to Zabrodsky [35]:

$$h_s = 35.8\ \rho_p^{0.2}\ k_g^{0.6}\ d_p^{-0.36} \qquad (2.6)$$

This relation is adequate for temperatures up to 900K above which radiant heat transfer becomes significant.

Attention has recently been drawn to the very high values of h_s found in shallow fluidised beds [10]. Pillai [6] measured coefficients in beds up to 50 mm deep and found them to be several times greater than in deep beds. This discovery has allowed low pressure drop fluidised bed heat exchangers to be developed as described by Botterill and Virr [36].

2.3 GAS-TO-PARTICLE HEAT TRANSFER

The greatest difficulty in any discussion of gas-to-particle heat transfer is the great variation in the published data. Kunii and Levenspiel [2] state that this variation is over a thousandfold between different observers. The results are usually presented as a Nusselt number, $Nu = h_p d_p / k_g$, which includes h_p, the gas-to-particle heat transfer coefficient.

The simplest case of gas-to-particle heat transfer is that of a single sphere in an infinite medium. In this instance Nu is given by Ranz and Marshall [37] as

$$Nu = 2 + 0.6\, Pr^{\frac{1}{3}}\, Re^{\frac{1}{2}} \qquad (2.7)$$

If we extrapolate equation (2.7) to zero fluid velocity then Re tends to zero. Hence theoretically Nu approaches 2 for the condition of conduction alone in an infinite stagnant medium. The same authors demonstrated this experimentally [38] yet Kunii and Levenspiel [2] present several sets of data with Nu less than 0.001.

The divergence of the data from Nu = 2 and their variation lead to consideration of the experimental conditions. Measurement of temperature in a bed is difficult - does a bare thermocouple measure the solids' temperature, that of the gas, or a combination [39] ?. Also the gas flow pattern assumed in the analysis was plug flow for some and perfect mixing for others.

These factors are discussed by Frantz [40] who concluded that the only true coefficients were measured by Heertjes and McKibbins [41] and Walton et al. [42]. Both groups measured the particle temperature deep in the bed with a bare thermocouple, and the gas temperature with a suction thermocouple. The former may be questioned as a very large probe was used, so Frantz concluded that the results of Walton et al. were the more reliable. Their correlation was

$$Nu = 0.0028\ Re^{1.7} \left[\frac{d_b}{d_p} \right]^{0.2} \tag{2.8}$$

Kunii and Levenspiel [2] considered that the data calculated assuming plug flow were more reliable because they were consistent. They correlated the pattern by

$$Nu = 0.03\ Re^{1.3} \tag{2.9}$$

but Nu still falls below 2 when Re is less than 25. This discrepancy was explained by Kato and Wen [43] who considered that the effective heat transfer area is much less than the total particle area because the regions near the points of contact between the particles cannot be reached by the gas. The thermal boundary layers which surround the particles overlap and the gas cannot easily penetrate these to reach the particles' surfaces. As the gas velocity increases, the movement of the particles breaks down the boundary layers so Nu rises with Re towards the predictions of the single sphere equation.

Despite predicting Nu less than 2, equation (2.9) provides a quick way of estimating heat transfer in a bed and as such is a useful result. However, as with many other aspects of fluidised bed behaviour, the geometry of the containment and the distributor can be very important. Walton's correlation (equation (2.8)) includes the diameter of the bed, d_b, but McGaw [44] pointed out that the design of the distributor was not considered. He developed a correlation for shallow beds of particles a few mm in size up to 33mm in depth and with a distributor having holes on a triangular array:

$$Nu = 0.353\ Re^{0.9} \left[\frac{d_p}{Z}\right]^{0.47} \left[\frac{d_p}{P}\right]^{0.19} \left[\frac{d_p}{D}\right]^{-0.19} \qquad (2.10)$$

McGaw's experiments used a crossflow fluidised bed to obtain steady state gas-to-particle heat transfer. In addition to the correlation (equation (2.10)), he found that the ratio of gas to particle mass flows did not affect Nu. This independence suggests that the heat transfer, and hence the gas flow pattern, is not significantly affected by the flow of the solids. His correlation should be used with caution, though, if the distributor is different from that which he used.

The heat transfer coefficient between gas and particle is always small - somewhere in the range 3 to 50 W m^{-2} K^{-1}. However a bed of small particles exposes a very large surface area to the gas of the order of 10 000 m^2 per unit volume of bed. The large area compensates for the low heat transfer coefficient so heat is exchanged very rapidly between gas and particles [45]. This means that the gas reaches the particle temperature within a short distance of the distributor in view of the low thermal capacity per unit volume of the gas. Indeed Zabrodsky [46] has suggested that very little of the temperature difference between the gas and the particles remains after the gas has passed through a monolayer of particles. Thus the gas follows the particle temperature, not vice versa.

2.4 FLOW OF FLUIDISED SOLIDS

There are three distinct methods by which fluidised beds have been made to flow; (i) a fluidised bed may be fed continuously from a hopper at one end and the depth kept constant by a weir at the other; (ii) the distributor may form the base of an inclined channel; (iii) the distributor can be formed into a continuous loop shaped like a running track, the bed being fluidised in the usual manner but paddle wheels push the material around the circuit.

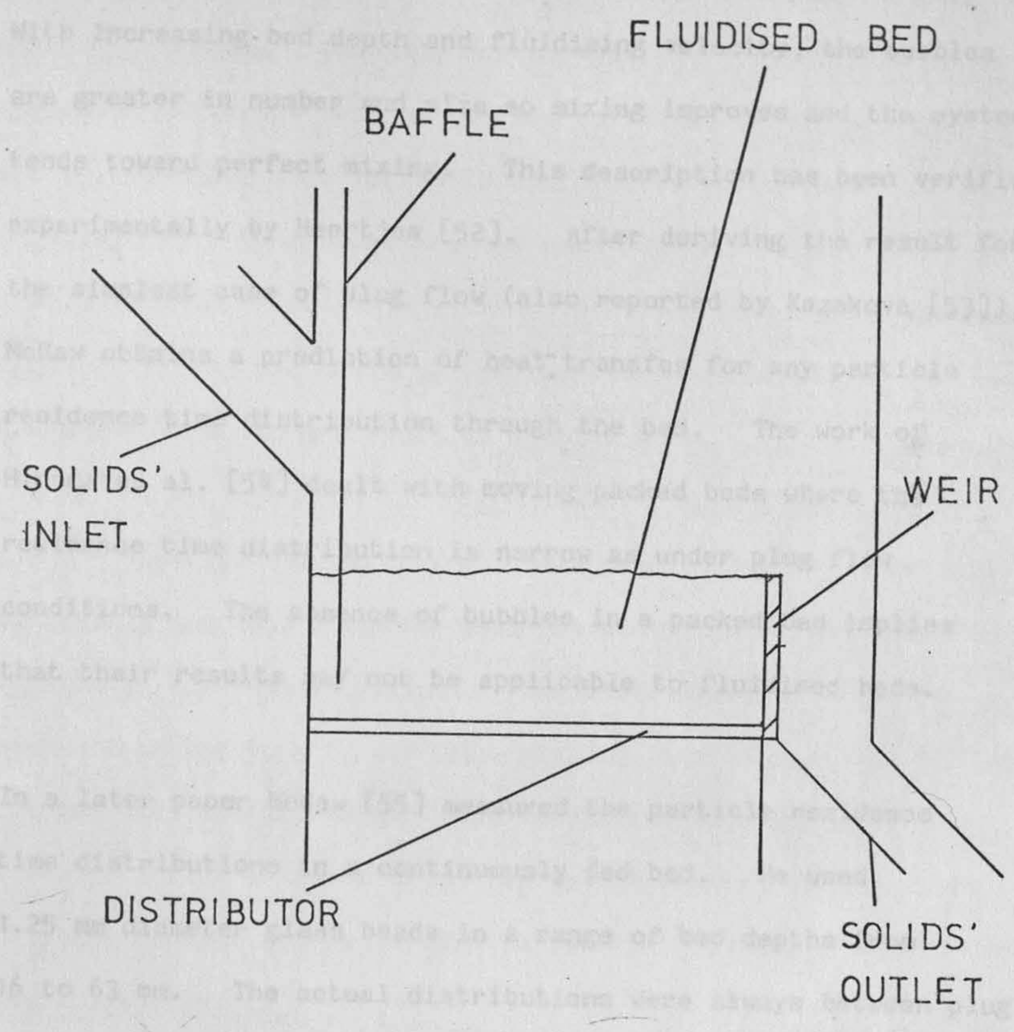
Studies have been made using all three of these methods which have one thing in common. The circulation is achieved by external influences, for example, by gravity in an inclined channel. In the heat exchanger being investigated in this project, solids' circulation is driven internally by the fluidising air by virtue of the design of the distributor plate. Thus the work reported here is fundamentally different from these experiments. Whilst bearing this important difference in mind, we shall consider the results of all three methods in turn.

2.4.1 Continuously fed beds

The study of vertical flow in continuously fed beds has been carried out for a number of years to investigate various models of solids' mixing [47]. However the reported work on horizontal flow produced when a bed is fed at one end and removed over a weir at the other (Figure 2.9) is very limited in extent. The use of this technique for industrial heating and cooling of granular materials has been reported [48, 49], but its analysis has been neglected.

Thermal calculations on such a system have been carried out by Borodulya [50]. His work develops design procedures and includes the heat transfer due to mixing within the bed as well as that due to the bulk motion of the material along the bed. This leads to some very complicated mathematics, and it is not clear how the model can be used for design purposes. Although solids' mixing is considered in the analysis, there is no experimental data on what mixing patterns might be expected in the bed.

FIGURE 2.9 CONTINUOUSLY FED FLOWING FLUIDISED BED



McGaw [51] has since developed a simpler approach and describes the possible mixing patterns. He states that if the bed is operated such that few bubbles are formed, then lateral mixing is minimised and the situation approximates to plug flow. With increasing bed depth and fluidising velocity, the bubbles are greater in number and size so mixing improves and the system tends toward perfect mixing. This description has been verified experimentally by Heertjes [52]. After deriving the result for the simplest case of plug flow (also reported by Kazakova [53]), McGaw obtains a prediction of heat transfer for any particle residence time distribution through the bed. The work of Hirama et al. [54] dealt with moving packed beds where the residence time distribution is narrow as under plug flow conditions. The absence of bubbles in a packed bed implies that their results may not be applicable to fluidised beds.

In a later paper McGaw [55] measured the particle residence time distributions in a continuously fed bed. He used 1.25 mm diameter glass beads in a range of bed depths from 16 to 63 mm. The actual distributions were always between plug flow and perfect mixing: neither was an adequate approximation. As the bed depth decreased the fluidisation approached plug flow, though the fluidising velocity did not appear to affect the mixing greatly. He stated that heat transfer was faster with plug flow and so shallow beds should be better. In contrast to McGaw's work with shallow beds, Hiraki et al. [56] found that fluidising velocity greatly affected the lateral dispersion of gas in beds 300 to 400 mm deep. This suggests that McGaw's conclusions will not apply to beds more than about 100 mm in depth.

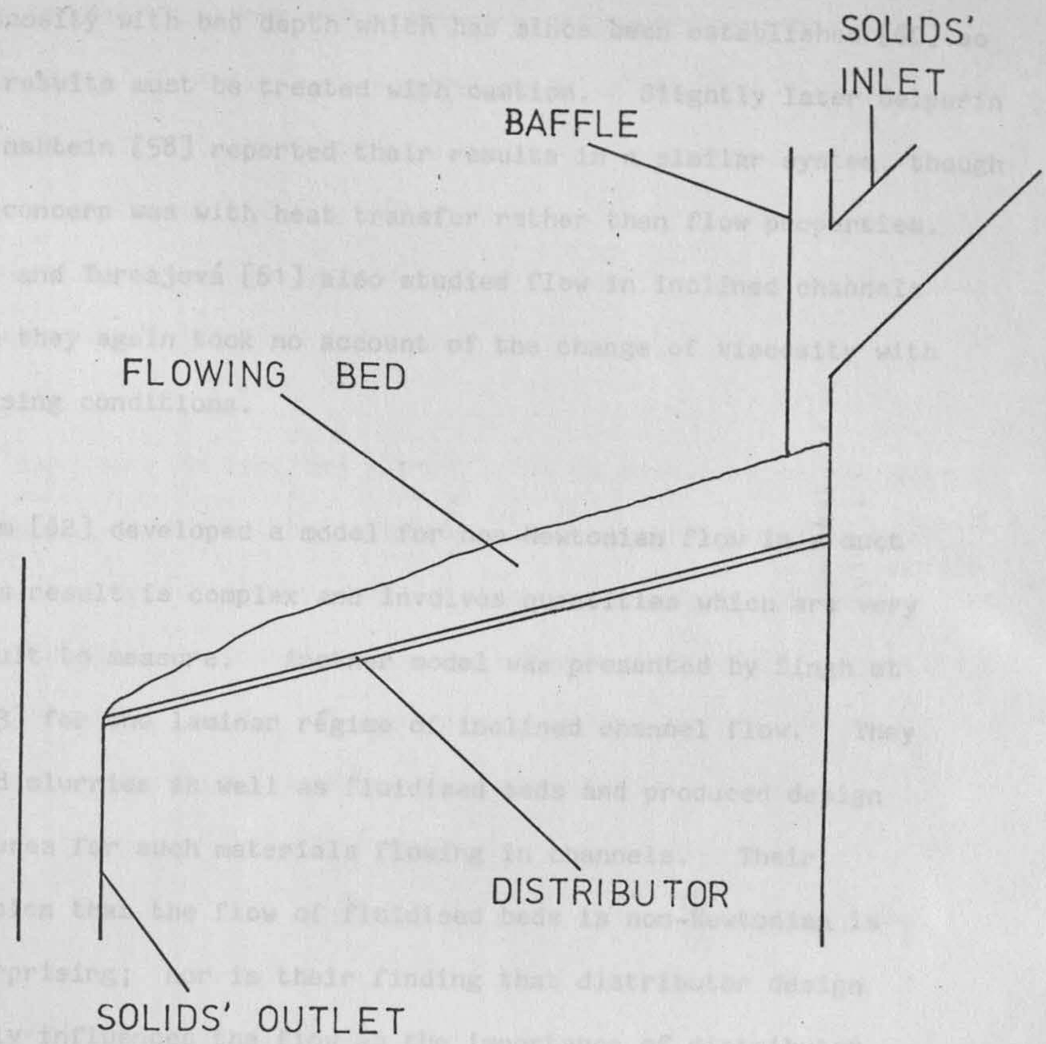
The most useful analysis to date is given by McGaw [57] and he describes its experimental verification elsewhere [44]. In this he extends the work of Gelperin and Ainshtein [58] who first introduced the gas-to-particle heat transfer coefficient into the analysis. McGaw's equation can be applied in situations where thermal equilibrium between gas and particles is not achieved and for any particle size and residence time distribution. Of particular relevance to this project are the equations he derives for the special case of negligible internal particle temperature gradients. These will form the basis of the theoretical analysis of the heat exchanger which is described in Chapter 4.

McGaw's experimental testing of these predictions [44] was most successful and this suggests that his model might reasonably be applied to the heat exchanger. The most important point to emerge from the work on continuously fed beds is that the shallower the bed, the closer the gas flow conditions approach plug flow. Since plug flow is the optimum for heat transfer, this implies that the shallow bed to be used in the heat exchanger should work satisfactorily.

2.4.2 Inclined channel flow

Studies of the flow of fluidised beds down inclined channels (Figure 2.10) have been carried out mainly to understand flow phenomena such as slip and viscosity. The first published work was presented by Siemes and Helmer [59]. They assumed

FIGURE 2.10 INCLINED CHANNEL FLOW



the flow to be fully Newtonian and used analogies with liquid flow to analyse their results. This approach neglects the variation of viscosity with bed depth which has since been established [60] so their results must be treated with caution. Slightly later Gelperin and Ainshtein [58] reported their results in a similar system, though their concern was with heat transfer rather than flow properties. Neužil and Turcajová [61] also studied flow in inclined channels though they again took no account of the change of viscosity with fluidising conditions.

Quassim [62] developed a model for non-Newtonian flow in a duct but his result is complex and involves quantities which are very difficult to measure. Another model was presented by Singh et al. [63] for the laminar régime of inclined channel flow. They studied slurries as well as fluidised beds and produced design procedures for such materials flowing in channels. Their conclusion that the flow of fluidised beds is non-Newtonian is not surprising; nor is their finding that distributor design strongly influences the flow as the importance of distributor effects is well-known in fluidised bed work.

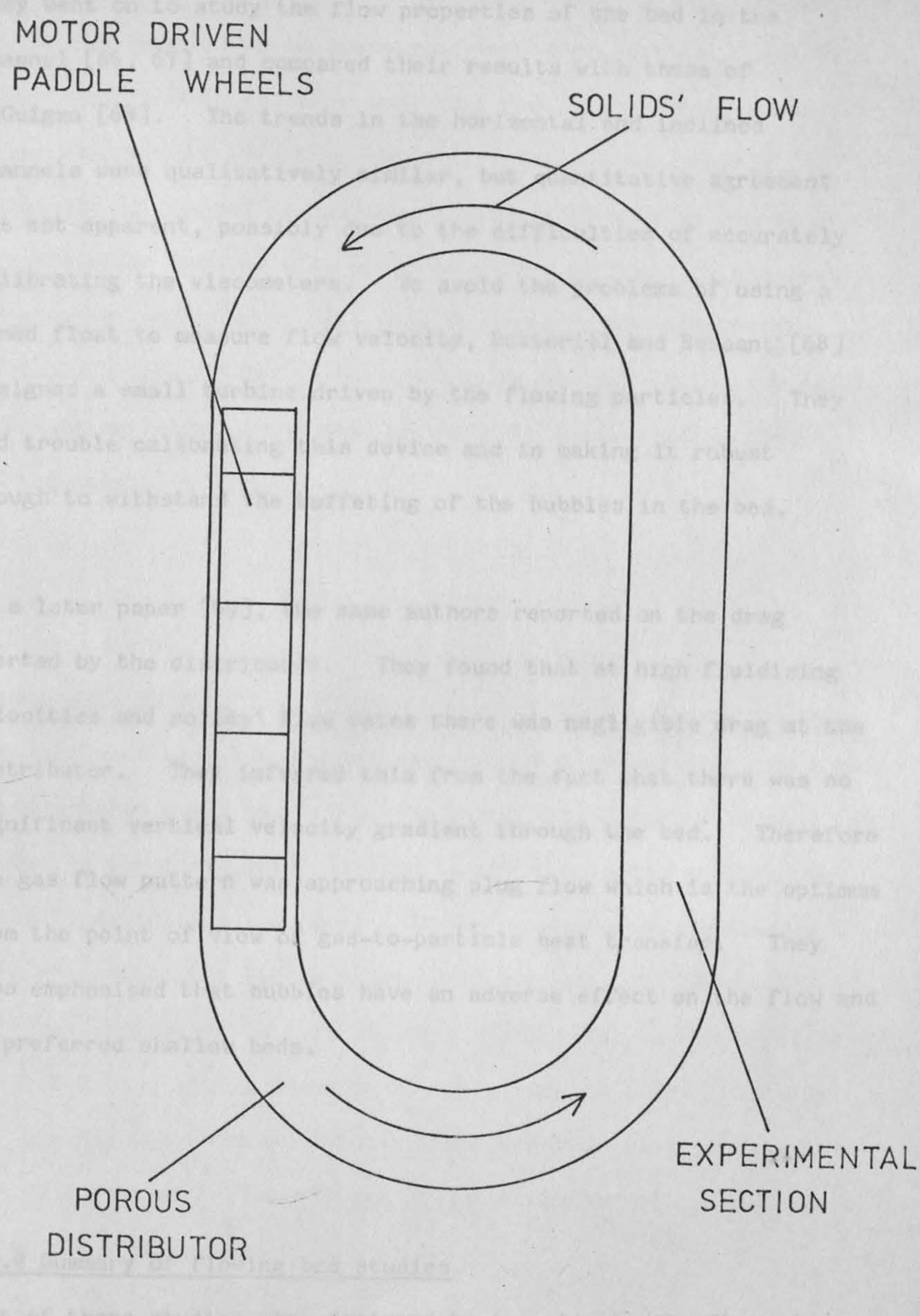
However, without doubt the most valuable work on flow in inclined channels is that of McGuigan [64]. His studies were also intended to elucidate the flow behaviour of the bed and its associated viscosity and shear properties. Since the aim of the current project is to develop the heat exchanger rather than to gain a deep understanding of the phenomena, the work will not be discussed in detail. McGuigan does make several general observations which are of interest though. Firstly at low fluidising velocities he found that the flow was adversely affected by segregation, especially in shallow beds. Secondly the quality of fluidisation appeared to deteriorate with increasing bed depth. Thirdly he found that the use of a timed float moving with the stream could not be expected to give a reliable measurement of flow velocity. Lastly he observed hydraulic jumps in the channel under certain fluidising conditions.

Overall the work on inclined channel flow is mostly concerned with the detailed study of flow phenomena. The general point emerges that shallow beds flow more easily than deep ones, though segregation can be a problem. Industrially available bed materials have quite wide size ranges, so the problem of segregation may be worsened when these rather than laboratory sieved materials are used.

2.4.3 Open horizontal channel flow

This section is devoted to the work of Botterill and his co-workers who constructed a horizontal continuous loop channel around which the particles were moved by a paddle-wheel arrangement as shown in Figure 2.11. The circulation rate was variable and was measured using a timed float, a method that McGuigan [64] considered could give misleading results. Their first work [65] concerned the flow of a fluidised bed past arrays of heat transfer tubes and its effect on heat transfer. One difficulty of this type of work in a horizontal channel is that there is a head loss

FIGURE 2.11 OPEN HORIZONTAL CHANNEL FLOW



around the circuit. Hence the bed depth changes and this may affect the fluidisation and therefore the flow of the bed.

They went on to study the flow properties of the bed in the channel [66, 67] and compared their results with those of McGuigan [64]. The trends in the horizontal and inclined channels were qualitatively similar, but quantitative agreement was not apparent, possibly due to the difficulties of accurately calibrating the viscometers. To avoid the problems of using a timed float to measure flow velocity, Botterill and Bessant [68] designed a small turbine driven by the flowing particles. They had trouble calibrating this device and in making it robust enough to withstand the buffeting of the bubbles in the bed.

In a later paper [69], the same authors reported on the drag exerted by the distributor. They found that at high fluidising velocities and solids' flow rates there was negligible drag at the distributor. They inferred this from the fact that there was no significant vertical velocity gradient through the bed. Therefore the gas flow pattern was approaching plug flow which is the optimum from the point of view of gas-to-particle heat transfer. They also emphasised that bubbles have an adverse effect on the flow and so preferred shallow beds.

2.4.4 Summary of flowing bed studies

Most of these studies were designed to investigate the flow of fluidised beds in terms of their non-Newtonian fluid properties. Since a similar appreciation of the processes occurring in the heat exchanger was not contemplated, only the more general results of this work are useful. The most important is that shallow beds flow more easily due to the absence of large bubbles. The data from continuously fed beds show that for the best heat transfer plug flow should obtain in the bed. This is more nearly achieved if the bed is shallow. However it must be remembered that similar results will not necessarily be found in the heat exchanger where the solids' flow is driven internally by the fluidising gas. The basic question as to what is the optimum fluidising velocity for flow to occur is unanswered in the literature.

2.5 JETS IN FLUIDISED BEDS

In shallow fluidised beds the distributor plate usually has a small open area of about 5%. Thus the actual gas velocity leaving the holes in the distributor is some twenty times greater than the overall fluidising velocity. It may therefore be considered that the gas enters the bed as a jet which expands into the bed. In the heat exchanger this is especially important as the particles move due to momentum transfer to them from the jets. A study of the existing work on jets in fluidised beds will be valuable in this context. Two aspects of jet behaviour will be examined: momentum transfer and heat transfer.

2.5.1 Momentum transfer

One of the first mentions of jets comes from Kunii and Levenspiel [2] who asserted that if the velocity pressure of the jet exceeds the bed pressure drop, the jet will punch right through the bed. This takes no account of the diameter of the jet or its angle of entry into the bed. Merry [70] realised that with the advent of shallow beds it might be necessary to promote lateral solids' mixing in some way. To study this he determined the penetration of a horizontal gas jet into a fluidised bed. Material was entrained into the jet region, especially near to the orifice, and momentum transferred to those entrained particles. All of his measurements were taken with jets very close to the bed containment and the inevitably large wall effects had an unknown influence on the properties of the jet.

The entrained material is mostly at the jet boundary so there are few particles in the bulk of the jet. By assuming that the inter-particle spacing was large enough to be able to treat each particle as isolated, Merry [70] developed a semi-empirical equation for jet penetration. This correlation fitted his experiments for orifice diameters of 2.5 to 14 mm and exit velocities of 40 to 300 m s^{-1}. However in the heat exchanger these values will be only about 0.4 mm and 25 m s^{-1} respectively and so much less than Merry's. Hence it would be unwise to expect his equation to provide accurate predictions for the much smaller jets in the heat exchanger.

Merry has also studied vertical jets [71] in two-dimensional beds so wall effects were again most important. He developed a correlation for this situation, though one must have the same reservations about its applicability to the heat exchanger. Other workers [72, 73] give correlations quite different from that of Merry [71] which indicates the difficulties of interpreting and collecting data from jet experiments. Merry also reported on jets in liquid fluidised beds [74] and suggested that the effects were similar to those seen in gas fluidised beds. These results should be treated with great caution in view of the well-known differences between gas and liquid fluidised beds.

The only study of jets away from the walls is that of Rowe et al. [75] who used X-ray techniques to see voids deep within the bed and so undisturbed by the walls. Their observations are most interesting as it would appear that jets are not stable within the bed. Instead bubbles form above the orifice which elongate and then detach. The bubble size is unchanged by increasing the gas flow rate, but they form more frequently. Bubbles remain in contact with the orifice for only about 70 ms, so very short exposures are needed to differentiate between growing bubbles and stable jets. They did not see any stable jets at exit velocities up to 70 m s^{-1} unless a surface was placed near the orifice which could stabilise the jet. Hence they concluded that the gas enters as a stream of bubbles rather than as a continuous, stable jet.

Minaev and his colleagues [76-78] applied theoretical aerodynamics to the shape and size of jets in beds of large particles. The predictions are compared with the observations of jets, but once again the work was conducted in a two-dimensional bed so is of limited value. In view of Rowe et al. [75] concluding that jets do not exist deep in the bed, theoretical approaches will not be discussed further.

2.5.2 Heat transfer

The heat transfer from jets to fluidised beds has been studied less than momentum transfer. The first reported work is that of Behie et al. [79] as a continuation of their work on momentum dissipation in beds over 1m deep [80, 81]. These studies are clearly very different from those in shallow beds, but some results are of interest. Firstly they found that the radial temperature drop from jet to bed was very sharp implying a clear interface there. Secondly the excess temperature of the jet above that of the bed fell very rapidly. Using the jet quenching model of Baerns and Fetting [82], Behie et al.[79] were able to calculate the heat transfer coefficient between the jet and the bed and found it to be about 5 kW m^{-2} K^{-1}. They also derived equations to describe the axial and radial temperature profiles of the jet.

A study of the mechanisms of the jet-to-bed heat transfer has been made by Shakhova and Lastovtseva [83]. They also concluded that there was a sharp interface between jet and bed, and intense interaction there transferred heat to the bed. They found that the temperature relaxation of the jet occurred almost entirely within the jet, so the bubbles breaking away transferred only a small amount of heat beyond the jet. Donadono and Massimilla [84] reported on heat transfer, but only in two-dimensional beds where wall effects are dominant.

2.5.3 Summary of previous work on jets

In conclusion, momentum and heat transfer from a jet to the surrounding fluidised bed are clearly related. Much of the work has been in two-dimensional beds in which wall effects dominate the properties of the jet. Rowe et al. [75] presented data showing that stable jets do not form away from the walls, only a stream of bubbles. Thus it is doubtful that the correlations for jet penetration and heat transfer can be usefully applied to the heat exchanger.

2.6 ELECTROSTATIC INTER-PARTICLE EFFECTS

If the particles in a fluidised bed are electrostatically charged, the behaviour of the bed can be greatly affected. Generally charge causes the particles to stick together and form agglomerations which restrict particle motion in the bed. This leads to loss of fluidisation and channelling, its effect being most serious in beds of small, light materials. The authors of the works on fluidised bed flow referred to earlier all noted the significant effect charging can have on flow behaviour, hence its importance in this project. Little work appears to have been carried out on the use and effectiveness of possible precautions to prevent the build-up of charge.

An early investigation into charging was reported by Ciborowski and Wlodarski [85]. They found that the electrostatic potential varied directly with increasing fluidising velocity, but was little influenced by the particle size in the range 300 to 750 μm. They demonstrated that even minute amounts of moisture in the fluidising gas could cause the charge to leak away by increasing the conductivity of the bed, and also that charging decreased with increasing temperature. Indeed no difficulties attributable to charge have been reported in high temperature systems which suggests that it will not be a problem in the heat exchanger.

Geldart and Boland [86] found that although conducting bed containment allowed the leakage of charge near the wall, there was little reduction in the bulk of the bed. Shikhov et al.[87] also studied the influence of construction materials on charging. Metal walls reduced the potentials by about 50% compared with beds with insulating walls of glass or PVC. Thus overall the metal walls and high operating temperature of the heat exchanger should prevent the accumulation of charge in the bed.

2.7 SUMMARY OF EXISTING WORK

The properties of fluidised beds are now well-documented but not yet fully understood. In general it is much wiser to make practical measurements than to apply empirical correlations developed elsewhere. Conversely it is difficult to predict the performance of a fluidised bed system on the basis of other people's work. Hence it is necessary to try out ideas in experiments before deciding on the practicability of a novel system such as the gas-to-gas heat exchanger.

The reported work on the flow of fluidised solids all concerns externally driven motion and so is of limited value. The major result from that work is that shallow beds should have the flow and heat transfer characteristics best-suited to the heat exchanger. The aim of the project is to demonstrate the usefulness of a patented system, and nothing in the literature suggests that it will be unsuccessful. The technical question asked in this project is simply; "will the heat exchanger work?" In answering that, information will be gained for the first time about flowing beds driven along by the fluidising gas.

CHAPTER THREE

DESIGN CONSIDERATIONS OF THE HEAT EXCHANGER

CHAPTER THREE

DESIGN CONSIDERATIONS OF THE HEAT EXCHANGER

3.1 INTRODUCTION

The purpose of this chapter is to discuss the design considerations of the heat exchanger. The design is essentially governed by two aspects: firstly the covering patent, and secondly the desired characteristics of the heat exchanger. After describing the claims of the patent, it will be seen how the properties of the heat exchanger are affected by such factors as pressure drop, fluidising velocity and particle type. If the heat exchanger is to be a successful commercial product, then patent protection is essential. This will prevent the exploitation of the features of the unit by other manufacturers. Hence the contents of the patent and their effects on the design will be considered first.

3.2 THE PATENT

The starting point of this discussion is the patent of Elliott and Virr [7] which protects this particular type of heat exchanger. A copy of the patent is presented in Appendix 1, but it is important to extract the salient points from the legal jargon. The patent relates to a gas-to-gas heat exchanger in which heat is transferred from one gas stream to another by the movement of particles in a fluidised bed. The possible applications of the heat exchanger are not fully disclosed in the specification, only outlined. They are discussed in Chapter 7 in relation to the commercial constraints on the implementation of this particular heat exchanger. The overall concept of the system will be described with reference to Figures 3.1 and 3.2.

The distributor is formed in four sectors such that the directions of the fluidising gas flows follow on from each other. The distributor plate design is described in detail in Section 3.3, and it supplies the transverse momentum to the particles to enable them to flow. The two opposing long sectors of the distributor are separated by a central wall to keep the solids' flows apart. The plenum chamber beneath the distributor is split into two sections by the plenum division. These sections need not necessarily have the same area for if the two gas streams have different volumetric flows then the portion of the bed each fluidises may differ in area. The two parts of the plenum (Figure 3.2) accept the hot and cold gas streams between which heat is to be transferred.

Above the distributor the heated and cooled gas streams are separated by the partition. There is, however, a small gap between the bottom of the partition and the distributor which allows the passage of the bed particles between the two parts of the bed. The solids are heated by the hot gas (thereby cooling it) in one part of the bed and then the solids flow around until

FIGURE 3.1 PLAN VIEW OF THE HEAT EXCHANGER

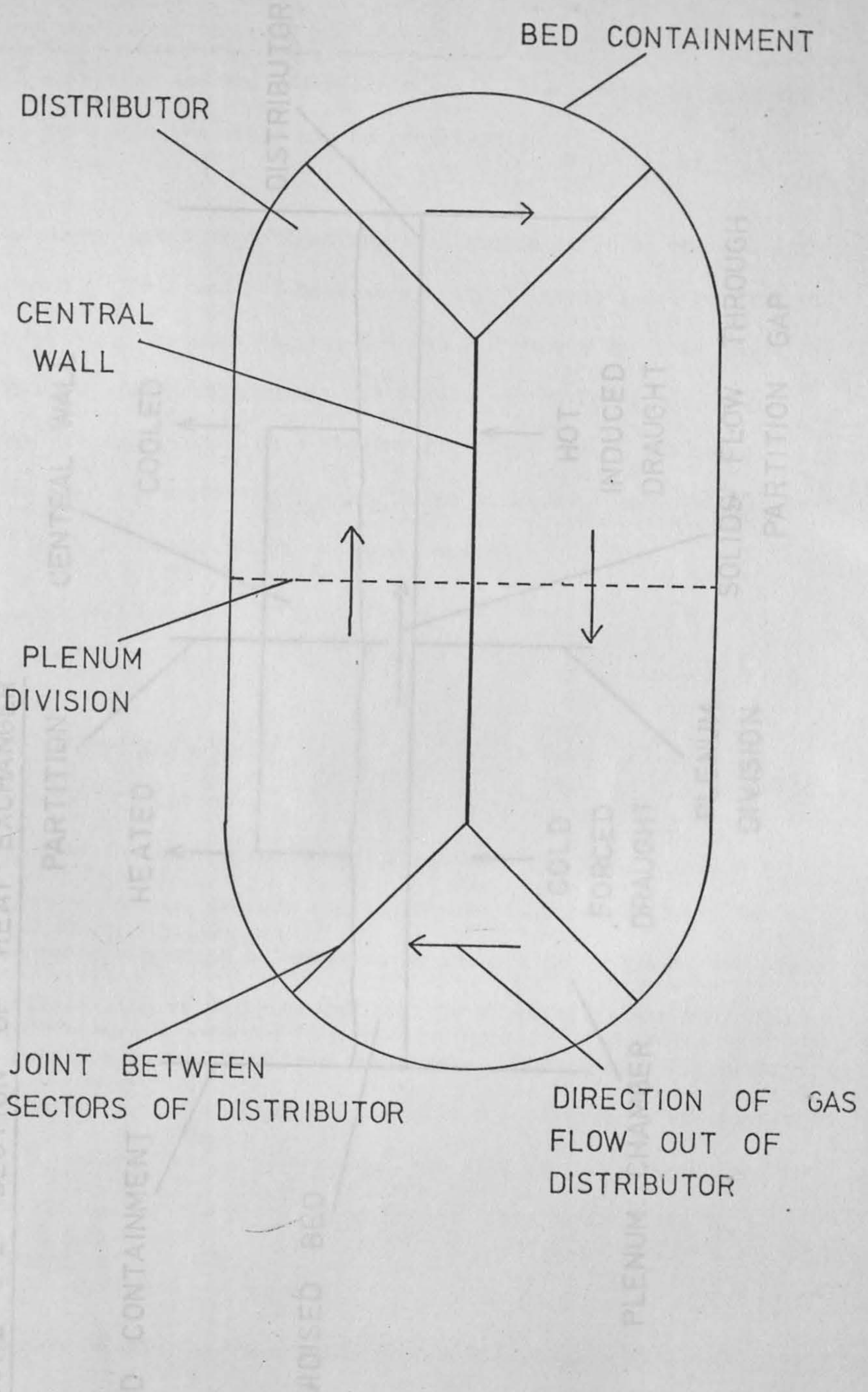
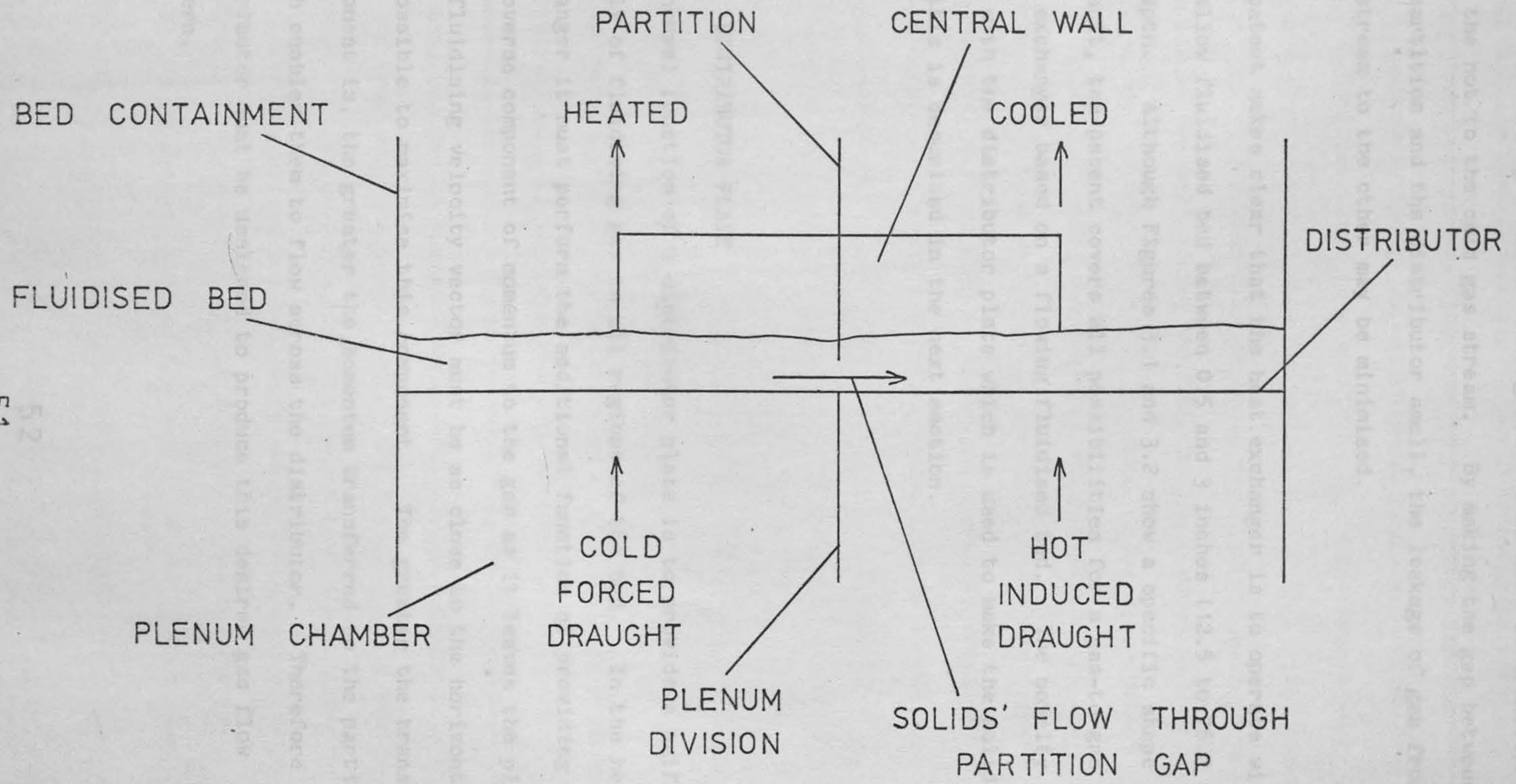


FIGURE 3.2 SECTION OF HEAT EXCHANGER



they are fluidised by cold gas. The particles then give up their heat to the cold gas hence heating it. Thus heat is transferred from the hot to the cold gas stream. By making the gap between the partition and the distributor small, the leakage of gas from one stream to the other may be minimised.

The patent makes clear that the heat exchanger is to operate with a shallow fluidised bed between 0.5 and 3 inches (12.5 to 76.2 mm) in depth. Although Figures 3.1 and 3.2 show a specific shape for the unit, the patent covers all possibilities for a gas-to-gas heat exchanger based on a flowing fluidised bed. The novelty largely lies with the distributor plate which is used to make the solids flow and this is described in the next section.

3.3 DISTRIBUTOR PLATE

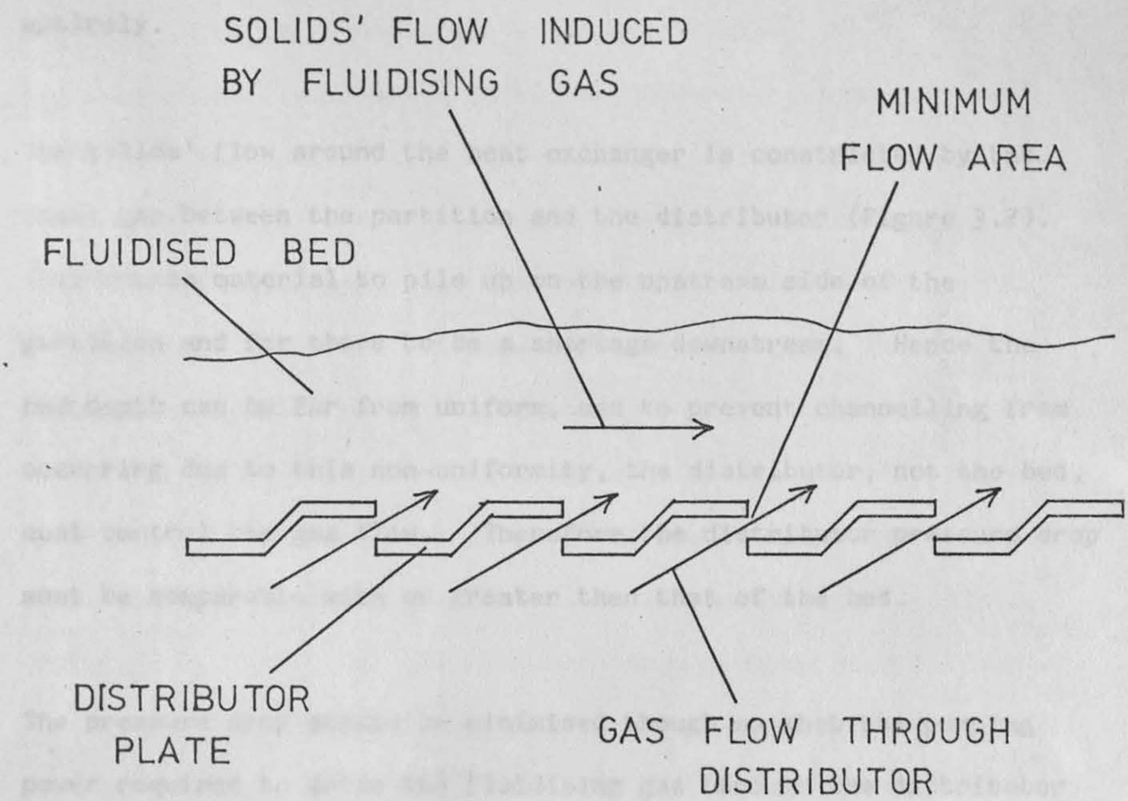
The normal function of a distributor plate is to provide a uniform supply of fluidising gas to all regions of the bed. In the heat exchanger it must perform the additional function of providing a transverse component of momentum to the gas as it leaves the plate. The fluidising velocity vector must be as close to the horizontal as possible to maximise this component. The greater the transverse component is, the greater the momentum transferred to the particles which enables them to flow across the distributor. Therefore the distributor must be designed to produce this desired gas flow pattern.

Figure 3.3 shows a suitable design for a distributor which has directional air flow properties. The fluidising gas emerges as a jet from the minimum flow area in a direction similar to that indicated, though it will diverge rapidly. Under certain conditions there is a sufficiently large transverse component of momentum to make the solids flow with a distributor of this type. A differently designed distributor could probably produce a better defined jet into the bed, but would be more difficult, and so more expensive, to make. The distributor must be capable of being manufactured cheaply because of the low value of the heat recovered by the unit. The distributor plate material must be able to withstand corrosive attack by the fluidising gas at the high operating temperatures, so stainless steel is preferred.

3.4 PRESSURE DROP ACROSS THE DISTRIBUTOR

One important aspect of any distributor is its pressure drop and the ratio of that to the pressure drop across the bed. As discussed in Section 2.2.4, it is considered that a shallow fluidised bed can be operated successfully with a distributor pressure drop only one tenth that of the bed [28]. In this situation the bed controls the flow rather than the distributor. Thus if for any reason the bed depth ceases to be uniform over the distributor, then some of the fluidising gas may channel through the shallowest part of the bed. If this occurs,

FIGURE 3.3 IDEALISED SECTION OF DISTRIBUTOR PLATE



significant areas of the bed can become de-fluidised and remain so unless the particles are redistributed mechanically. This is clearly undesirable in any fluidised bed and is particularly so in a solids' flow unit where the flow can be impeded or stopped entirely.

The solids' flow around the heat exchanger is constricted by the small gap between the partition and the distributor (Figure 3.2). This causes material to pile up on the upstream side of the partition and for there to be a shortage downstream. Hence the bed depth can be far from uniform, and to prevent channelling from occurring due to this non-uniformity, the distributor, not the bed, must control the gas flow. Therefore the distributor pressure drop must be comparable with or greater than that of the bed.

The pressure drop should be minimised though so that the pumping power required to drive the fluidising gas through the distributor is reduced. The distributor should be chosen such that its pressure drop is comparable with that of the bed at the operating fluidising velocity. However in many cases, including this project, the distributor is chosen from what is available because it is expensive to make a distributor specifically for one instance. In the heat exchanger the distributor was of a type already successfully used by the company in other products. This had a higher pressure drop than needed for the beds used in the heat exchanger, but its availability made it the obvious choice.

3.5 FLUIDISING VELOCITY

In the case of this particular heat exchanger, the fluidising velocity must be large enough to ensure that the particles circulate around the unit. However, there are reasons for having U_f both large and small above this limit. If U_f is large then the fluidisation is violent and material is carried high above the bed. This requires a large freeboard zone so that the particles have the necessary height to disengage from the upward flowing gas. By reducing U_f to decrease the height of the unit, the area of the bed must increase for a given volume flow rate of gas. Hence a compromise is needed depending on whether height or base area is more important in a particular application. The pressure drop of the distributor, and therefore the necessary pumping power, increases with fluidising velocity so a low U_f is desirable on these grounds also. However the distributor should be designed to have an appropriate pressure drop at a designated U_f, not the other way round.

3.6 PARTICLE SIZE AND TYPE

The height of a fluidised bed system is largely determined by the disengagement space which is fixed by the particle size and type for a given fluidising velocity. If large particles are used to reduce the required freeboard, then they may be fluidised less uniformly and flow less easily. Also large particles have a smaller surface to volume ratio and so are heated or cooled more slowly by the fluidising gas. According to Geldart [16], in order to get good fluidisation particles less than 500 μm in diameter and density less than 4000 kg m^{-3} must be used. Particles approaching these limits will be most suitable as they will need the smallest freeboard for a given U_f.

Materials of an appropriate density are legion, but that which is to form the particles in the heat exchanger must have other properties also. Firstly, it must be stable at high temperatures and secondly be able to resist possible corrosive attack from the hot fluidising gas. Thirdly it must be unaffected by the thermal shocks that the particles will encounter during their circulation between the hot and cold gas streams. Lastly it should be cheap and readily available as large quantities will be needed if the heat exchanger is to be successfully exploited commercially.

These considerations narrow down the possible substances enormously so that only the refractory oxides, particularly silica (SiO_2) and alumina (Al_2O_3), remain. The latter has the higher density and so is preferred. More importantly alumina has been used most effectively by the company in other fluidised bed applications such as combustion, heat treatment and waste heat recovery [88]. It should also be noted that another reason for preferring alumina is that a health hazard is presented by very fine silica dust. In any fluidised bed it is inevitable that there will be some attrition of the particles due to mechanical and thermal stresses. Very fine silica can induce silicosis, whereas alumina appears to have a lower health risk associated with it [89].

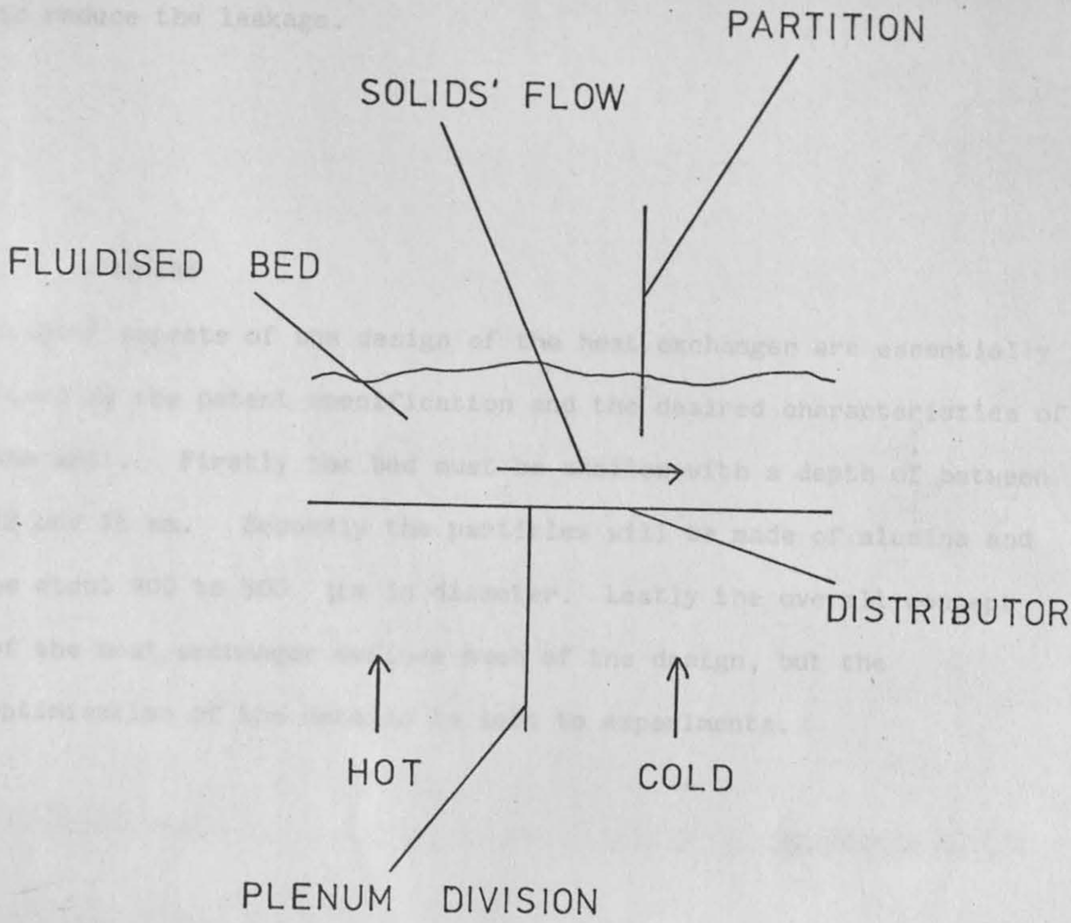
3.7 LEAKAGE BETWEEN THE GAS STREAMS

It is apparent that the gap between the partition and the distributor to allow the circulation of the particles also permits communication between the two gas streams. Therefore there can be leakage between the gas flows which not only reduces the efficiency of the heat exchanger but can also be hazardous. In many instances the hot gas stream will be from a combustion process and so will contain carbon dioxide and possibly some carbon monoxide. If the heated gas stream is to be used for space heating, then such toxic constituents must not be allowed to enter that stream. The leakage can never be eliminated entirely because the circulating particles will carry some gas with them in their boundary layers. However there are several possible ways to reduce this leakage to a tolerable level, and two of these are described in the patent specification.

The first proposal is to adjust the pressures of the gas streams such that the pressure in the hot bed is lower than that in the cold. This means that whatever leakage does occur will be from cold to hot rather than vice versa so the heated gas will remain uncontaminated. This can be achieved by using an induced draught fan on the hot flow and a forced draught fan on the cold to produce the required pressure differential. This arrangement is also helpful because it avoids passing the hottest gases through a fan (Figure 3.2).

The second proposal to reduce leakage is to modify the position of the partition with respect to the plenum division. As shown in Figure 3.4, the idea is to stagger the partition away from the plenum division, the result being that leakage is far more likely

FIGURE 3.4 POSITION OF PARTITION TO REDUCE LEAKAGE



to be from the cold to the hot flow. The leakage may well be dependent on several parameters, particularly the fluidising velocity, and experiments will be needed to establish how best to reduce the leakage.

3.8 SUMMARY

Several aspects of the design of the heat exchanger are essentially fixed by the patent specification and the desired characteristics of the unit. Firstly the bed must be shallow with a depth of between 12 and 76 mm. Secondly the particles will be made of alumina and be about 400 to 500 μm in diameter. Lastly the overall concept of the heat exchanger defines much of the design, but the optimisation of the details is left to experiments.

CHAPTER FOUR

POSSIBLE THEORETICAL MODELS OF THE HEAT EXCHANGER

CHAPTER FOUR
POSSIBLE THEORETICAL MODELS OF THE HEAT EXCHANGER

4.1 INTRODUCTION

The need for a theoretical model of the heat exchanger arises because it is necessary to be able to predict the performance of larger units. The rules for scaling-up fluidised bed systems are not well understood, so any guidance which a theoretical model can provide will be useful. In this chapter it will become apparent that the development of an adequate model is very difficult because of a fundamental characteristic of the heat exchanger. That is the unit operates with a combination of co-current and crossflow heat exchange. This implies that the mathematical formulation of the model is not soluble by analytic methods. Hence numerical solution of the equations is necessary if a complete theoretical model is to be produced.

4.2 AVAILABLE MODELS

It must be remembered that the heat exchanger operates by the crossflow motion of a heat carrier and so has no static walls through which heat is conducted. Hence the very complex but accurate model of crossflow heat exchangers due to Ishimaru et al. [90] is inapplicable. Similar models have been suggested by Spencer [91] and Stevens et al. [92] where simpler iterative techniques are used to solve the equations. Again these do not account for the movement of the heat carrier because the equations are independent of the presence of such a carrier.

Baclic [93] and Krasnoshchekov [94] have presented simplified formulae for crossflow heat exchanger performances, but these cannot be used where there is also co-current flow. Thus it is concluded that a model based on the input and output conditions of the gas flows alone is not practicable. Instead the problem must be approached by considering the heat transfer to and from the heat carrier; this is discussed in the next section.

4.3 MODELS BASED ON THE ROLE OF THE HEAT CARRIER

Models of the above type are based on systems in which both of the heat transfer media are passed through the heat exchanger at right angles to each other. Borodulya [50] has analysed this for the case of the pneumatic conveyor. He introduced the concept of a particle mixing coefficient but only defined it for the two extreme cases of plug flow and perfect mixing. However this has been extended by McGaw [51] and the development of the model due to him will now be considered.

In his first paper on the subject, McGaw [51] described the design of a heat exchanger for heating or cooling particulate solids in a shallow fluidised bed. This has already been referred to in Section 2.4.1 and is shown in Figure 2.9.

He pointed out that earlier investigators had neglected the effect of gas and solid flow conditions on the process of heat transfer. The flow conditions in the bed are affected by the formation of bubbles which causes lateral mixing. This mixing is reflected in the particle residence time distribution in the bed. It should also be noted that this distribution can be introduced by factors other than mixing such as non-uniform solids' inlet conditions. Thus the particle residence time distribution may be related to the geometry and configuration of the system.

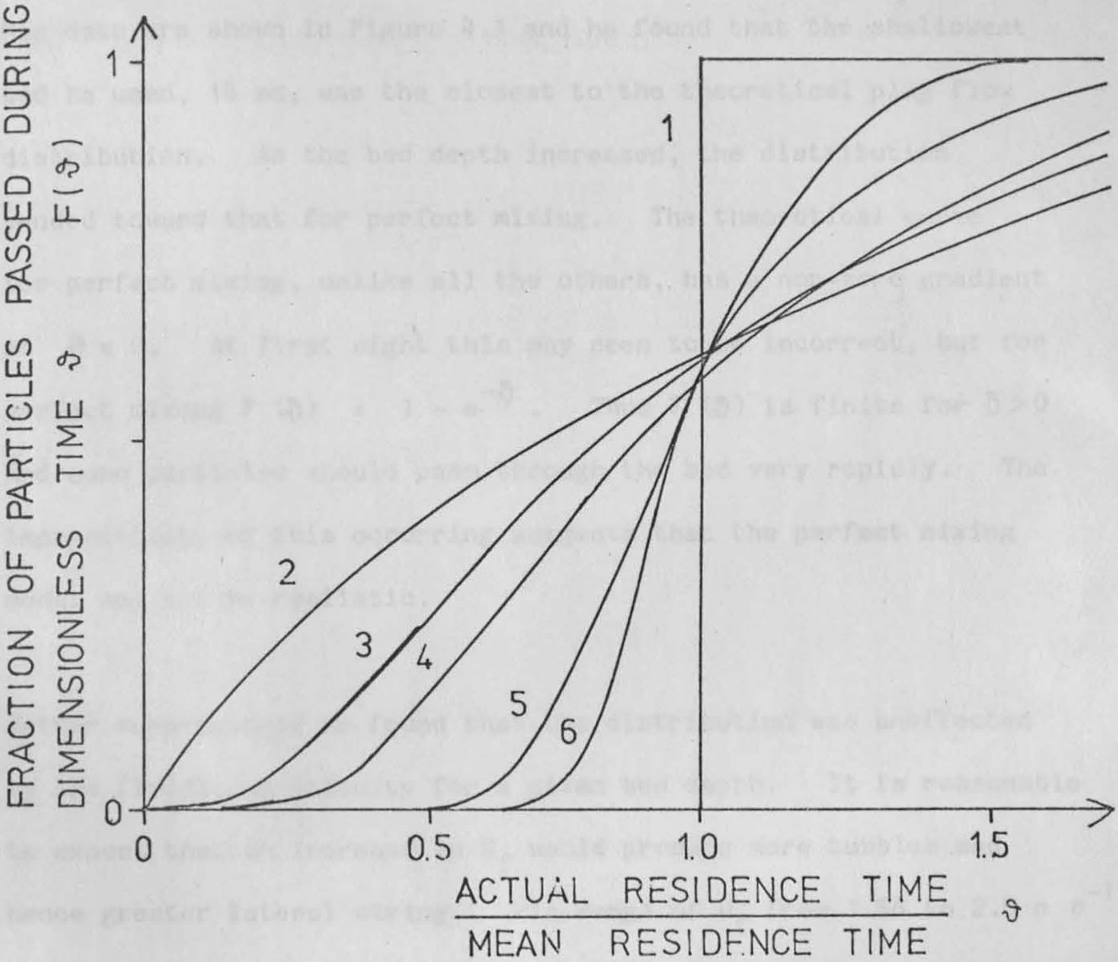
The two extremes of particle mixing are given by (i) plug flow and (ii) perfect mixing. Any real bed must clearly fall between these two extremes and one possibility is to describe the bed as a finite series of perfect mixers [95]. The use of this model, however, requires knowledge of the number of perfect mixers which is unknown. In shallow beds large numbers of small bubbles are formed and lateral mixing is minimised so plug flow obtains. In deeper beds the small bubbles coalesce and the wakes of these larger bubbles and their bursting on the surface disrupt the bed and the system tends to perfect mixing.

In a later paper McGaw [55] carried out experiments to determine the residence time distribution and compared his model with experimental heat transfer results. One major assumption made was that the particles and gas were in thermal equilibrium at the top of the bed. The work of Zabrodsky [46] demonstrated

FIGURE 4.1 MEASUREMENT OF PARTICLE RESIDENCE TIME DISTRIBUTION

adapted from McGaw [55]

Bed 294 mm long, 51 mm wide. Particles 1.25 mm diameter.



KEY:- 1. Theoretical - plug flow

2. ,, - perfect mixing

3. Observed - bed depth of 63 mm

4. ,, - ,, 50 mm

5. ,, - ,, 32 mm

6. ,, - ,, 16 mm

that the gas and particles were in equilibrium above the first few layers of particles for fluidising velocities close to U_{mf}. Thus McGaw's assumption was reasonable for the low velocities of his experiments.

His data are shown in Figure 4.1 and he found that the shallowest bed he used, 16 mm, was the closest to the theoretical plug flow distribution. As the bed depth increased, the distribution tended toward that for perfect mixing. The theoretical curve for perfect mixing, unlike all the others, has a non-zero gradient at $\vartheta = 0$. At first sight this may seem to be incorrect, but for perfect mixing $F(\vartheta) = 1 - e^{-\vartheta}$. Thus $F(\vartheta)$ is finite for $\vartheta > 0$ and some particles should pass through the bed very rapidly. The impossibility of this occurring suggests that the perfect mixing model may not be realistic.

Rather surprisingly he found that the distribution was unaffected by the fluidising velocity for a given bed depth. It is reasonable to expect that an increase in U_f would produce more bubbles and hence greater lateral mixing. His range of U_f from 1.56 to 2.8 m s^{-1} would probably have demonstrated this effect had it been present.

In general McGaw stated that the solids' outlet temperature was given by equation 4.1 overleaf:-

$$T_{po} = T_{gi} + \int_{o}^{\infty} (T_{pi} - T_{gi}) \exp \left[\frac{-\dot{m}_g C_g \vartheta}{\dot{m}_p C_p} \right] E(\vartheta)\, d\vartheta \qquad (4.1)$$

This can be simplified for the two extreme cases shown in Figure 4.1. Firstly for plug flow conditions

$$T_{po} = T_{gi} + (T_{pi} - T_{gi}) \exp \left[\frac{-\dot{m}_g C_g}{\dot{m}_p C_p} \right] \qquad (4.2)$$

Secondly for perfect mixing

$$T_{po} = T_{gi} + (T_{pi} - T_{gi}) \left[\frac{\dot{m}_g C_g}{\dot{m}_p C_p} + 1 \right]^{-1} \qquad (4.3)$$

It should be noted that if plug flow obtains then the solids' outlet temperature, T_{po}, will be closer to the gas inlet temperature, T_{gi}, than if the bed is perfectly mixed. Thus in any fluidised bed system of this type it is desirable for the gas flow conditions to approximate as closely as possible to plug flow.

Returning to McGaw's work, it is worthwhile looking at his data and comparing them with the predictions of the three equations above as done in Table 4.1. It can be seen that the use of the general result, equation (4.1), with the residence time distributions shown in Figure 4.1 is an accurate prediction of the experimental data. Even with the deepest bed the actual result is closer to the prediction of the plug flow model, equation (4.2), than to the perfect mixing case of equation (4.3). Since the deepest beds used in the work described in this thesis were 50 mm in depth, the assumption that plug flow obtains is not unreasonable. These predictions assume that gas and solid are in thermal equilibrium, so the result of any of the equations is independent of bed depth.

To develop an accurate model of the heat exchanger would require measurement of the particle residence time distribution. This is difficult, particularly in the case of the closed loop used in the heat exchanger. The development of an accurate theoretical model was not essential to the success of the project since it was not intended to study the flow of the fluidised bed in detail. Hence it was decided to adopt a simpler approach which would at least identify the important trends. Knowledge of these trends could aid the design of the experimental unit and the interpretation of the data. Thus the major simplification made in the model was to assume plug flow under all conditions. As seen from McGaw's work described above, this still enables useful conclusions to be drawn from the model.

Table 4.1 Heat transfer data

(Adapted from McGaw, D.R., Pow. Tech., 11, 33, (1975))

Bed depth (mm)	Solids' outlet temperature, T_{po} (°C)			
	Expt.	Equation (4.1)	(4.2)	(4.3)
16	52.4	52.4	51.5	63.8
32	54.0	54.0	52.6	64.7
50	52.6	52.2	47.9	51.7
63	54.8	55.1	50.2	62.0

Bed conditions: particles 1.25 mm diameter glass ballotini

$U_{mf} = 0.8 \ \mathrm{m\ s^{-1}}$

$U_f = 1.56 \ \mathrm{m\ s^{-1}}$

T_{pi} in range 104 to 108 °C

T_{gi} in range 35 to 39 °C

4.4 MATHEMATICAL DEVELOPMENT OF THE MODEL

McGaw [57] has developed a generalised theory for heat transfer to and from a shallow crossflow fluidised bed heat exchanger. The general solution takes the internal temperature gradients of the particles into account as well as particle size and residence time distributions. McGaw's full analytic solution is therefore complicated and requires the evaluation of an infinite series. The series is generated by the solution of the diffusion equation inside the particles which is necessary if significant internal temperature gradients exist. Hence it is worthwhile establishing if these can be neglected.

4.4.1 Internal thermal resistance of the particles

The dimensionless group called the Biot number can be used to determine when internal particle thermal resistance effects become important. It is defined as

$$Bi = \frac{h_p d_p}{k_p} \tag{4.4}$$

where h_p is the gas-to-particle heat transfer coefficient, d_p the particle diameter and k_p its thermal conductivity. This project is concerned solely with the use of alumina as the particulate material, and this has a thermal conductivity of about 36 W m^{-1} K^{-1} at room temperature decreasing to 7 W m^{-1} K^{-1} at 900 ^oC [96]. The larger size of particles has a diameter of 520 μm and if a high value of h_p, say 30 W m^{-2} K^{-1}, is used then the largest expected Bi can be calculated. This is the worst case and gives Bi = 0.001. It has been suggested [97] that if Bi < 0.25 then the internal temperature gradients can be neglected. This will clearly be the case in the heat exchanger so this simplifying assumption can be used in the analysis.

Making this assumption McGaw's equation for this special case becomes

$$T_{po} = T_{gi} + (T_{pi} - T_{gi}) \exp\left[\frac{-\dot{m}_g C_g}{\dot{m}_p C_p}\left[1-\exp\left[\frac{-h_p G}{\rho_g U_f C_g}\right]\right]\right] \quad (4.5)$$

where G is the surface area of the particles per unit area of distributor. There is no residence time distribution included in this equation because it is assumed that there are plug flow conditions in the bed. The particles are considered to be the same size, although a particle size distribution can be included if necessary.

4.4.2 Gas/particle thermal equilibrium

Equation (4.5) is the same as (4.2) with an additional term in the exponential. This extra factor allows account to be taken of the possibility that thermal equilibrium between the particles and the gas is not attained. It is often assumed that equilibrium is attained, but Zabrodsky [46] suggests that this will only be true for U_f up to about 2 U_{mf}. The heat exchanger will operate at velocities up to six times the minimum, so it will probably not be valid to assume thermal equilibrium. If thermal equilibrium is attained then $\exp(-h_p G/\rho_g U_f C_g)$ must approach zero. In fact in the heat exchanger described here this factor can be up to 0.2, so it is important to be able to account for this.

Since the gas and particles may not be at the same temperature at the top of the bed, there will be some difficulty in interpreting temperature measurements in that region. McGaw's theory provides an equation which predicts how far from thermal equilibrium the gas and particles will be:

$$T_{go} - T_{gi} = (T_p - T_{gi}) \left[1 - \exp \left[\frac{-h_p G}{\rho_g U_f C_g} \right] \right] \qquad (4.6)$$

This equation is true for any small element of bed which has a uniform temperature, T_p.

4.4.3 Parameters needed for the model

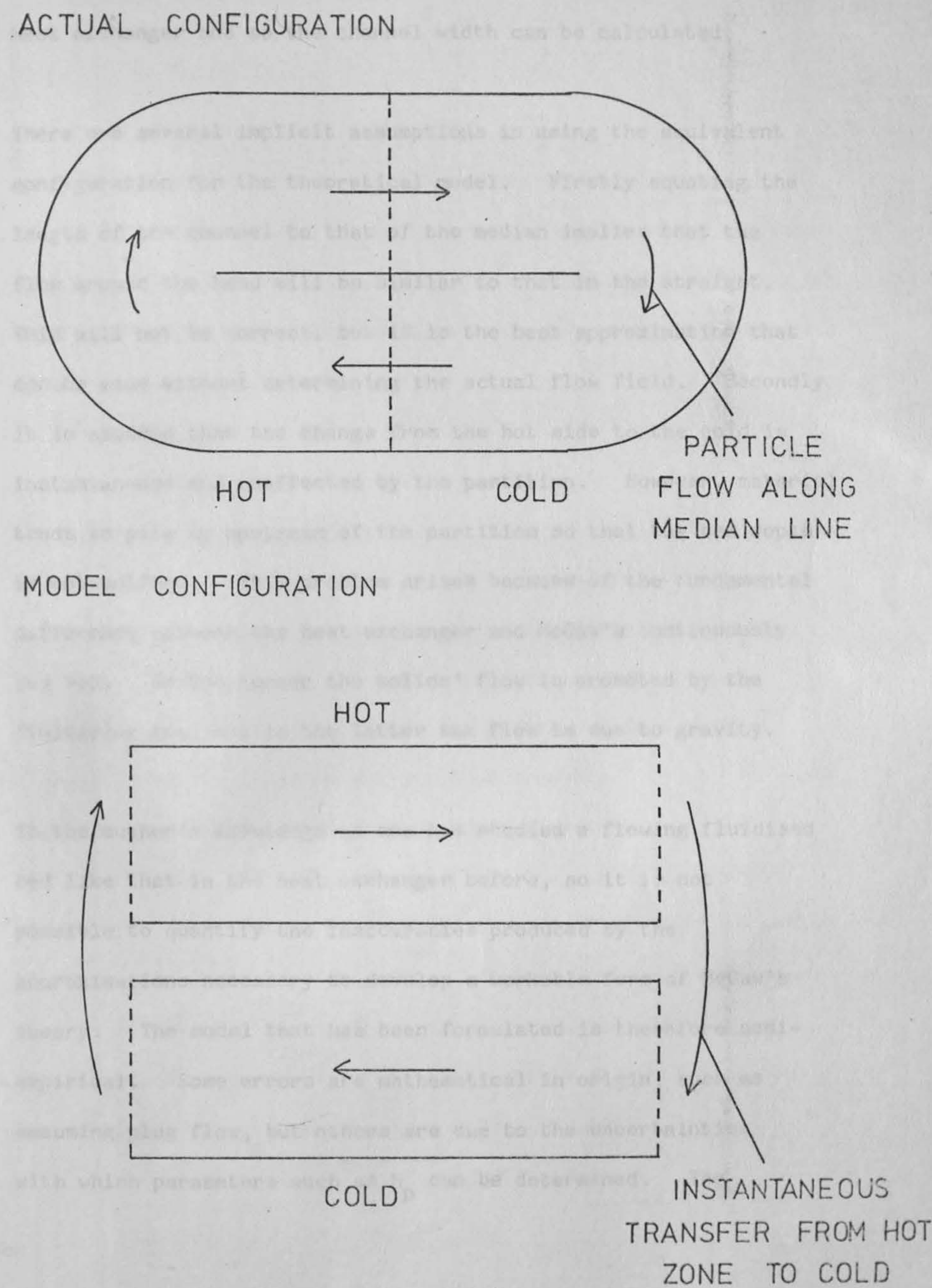
All the parameters in equation (4.5) can be easily found except the gas-to-particle heat transfer coefficient, h_p, and the solids' mass flow, $\dot{m}_p$. As discussed in Section 2.3, h_p is very difficult to measure reliably. McGaw [44] has measured h_p in shallow flowing fluidised beds and found it to lie between 10 and 30 W m^{-2} K^{-1}. Fortunately h_p only appears in the thermal equilibrium part of equation (4.5) so T_{po} is not particularly sensitive to h_p. Hence a typical value of h_p = 20 W m^{-2} K^{-1} can be used.

The second unknown is $\dot{m}_p$ which is difficult to measure in the closed loop which forms the heat exchanger. If the bed is assumed to have a constant depth, then a measurement of the solids' flow velocity, U_p, is needed. This can be estimated by timing the passage of a float around the bed, but this was found to be unsatisfactory and has been criticized by McGuigan [64]. The best estimate for U_p is about 0.2 m s^{-1} once the fluidising velocity exceeds about 3 U_{mf}.

4.5 APPLICATION OF THE MODEL TO THE HEAT EXCHANGER

To apply the model to the heat exchanger the closed loop is replaced by two straight channels as shown in Figure 4.2. The particles are considered to flow instantaneously from the end of the hot channel into the start of the cold and vice versa.

FIGURE 4.2 CONFIGURATION OF HEAT EXCHANGER FOR MODEL



The areas of the two channels are the same as those of the hot and cold sides of the actual unit. The lengths of the channels are set equal to the respective lengths of the median lines in the heat exchanger and so the channel width can be calculated.

There are several implicit assumptions in using the equivalent configuration for the theoretical model. Firstly equating the length of the channel to that of the median implies that the flow around the bend will be similar to that in the straight. This will not be correct, but it is the best approximation that can be made without determining the actual flow field. Secondly it is assumed that the change from the hot side to the cold is instantaneous and unaffected by the partition. However, material tends to pile up upstream of the partition so that the bed depth is not uniform. This problem arises because of the fundamental difference between the heat exchanger and McGaw's continuously fed bed. In the former the solids' flow is promoted by the fluidising gas, but in the latter the flow is due to gravity.

To the author's knowledge no one has studied a flowing fluidised bed like that in the heat exchanger before, so it is not possible to quantify the inaccuracies produced by the approximations necessary to develop a workable form of McGaw's theory. The model that has been formulated is therefore semi-empirical. Some errors are mathematical in origin, such as assuming plug flow, but others are due to the uncertainties with which parameters such as h_p can be determined. The semi-empirical model is, however, adequate for general predictions of the performance of the heat exchanger to be made.

4.6 GENERAL PREDICTIONS OF THE MODEL

The theory has been used to make two types of prediction which are both important to the operation of the heat exchanger. These predictions concern the temperature differences in the bed when it reaches a steady-state, and the time taken to attain that steady-state.

4.6.1 Steady-state conditions

When the heat exchanger is in a steady-state there will be a temperature difference between the solids entering and leaving the channels shown in Figure 4.2. Also the solids and gas may not be in thermal equilibrium at the free surface of the bed even though a steady-state has been reached. It is important to know the extent of the deviation from equilibrium so that thermocouple measurements in the bed can be interpreted correctly. This can be done by plotting equation (4.6) as shown in Figures 4.3 to 4.5. The curves are true for a narrow element of the bed which is perfectly mixed in a vertical direction and has a uniform temperature, T_p. The quantity ψ plotted on the vertical axis is given by equation 4.7 overleaf:-

$$\Psi = \frac{T_{go} - T_{gi}}{T_p - T_{gi}} \qquad (4.7)$$

This is equal to unity if thermal equilibrium is achieved, or less than one if the gas has not attained the temperature of the solids during its passage up through the bed.

Figure 4.3 shows the variation of Ψ with fluidising velocity for a given d_p and h_p and three different bed depths. Clearly the shallower the bed, the less likely it is for thermal equilibrium to be achieved since the gas residence time is shorter. Figure 4.4 is the same except that the particle diameter is larger, namely 520 μm. Comparison of the two shows the curves in Figure 4.4 are always positioned below those on Figure 4.3 so that thermal equilibrium is more difficult to achieve with larger particles.

Figure 4.5 shows that the temperature ratio Ψ is quite sensitive to changes in h_p: as expected the smaller the heat transfer coefficient, the greater is the departure from equilibrium. Since h_p is difficult to measure reliably, Figure 4.5 shows one of the problems of attempting to use the model, namely that h_p is unknown. The possible range of h_p is quite large so the accuracy of the predictions related to thermal equilibrium is poor. Overall it can be seen that for an initial temperature difference between the solids and the

FIGURE 4.3 DEVIATION FROM THERMAL EQUILIBRIUM AT TOP OF BED

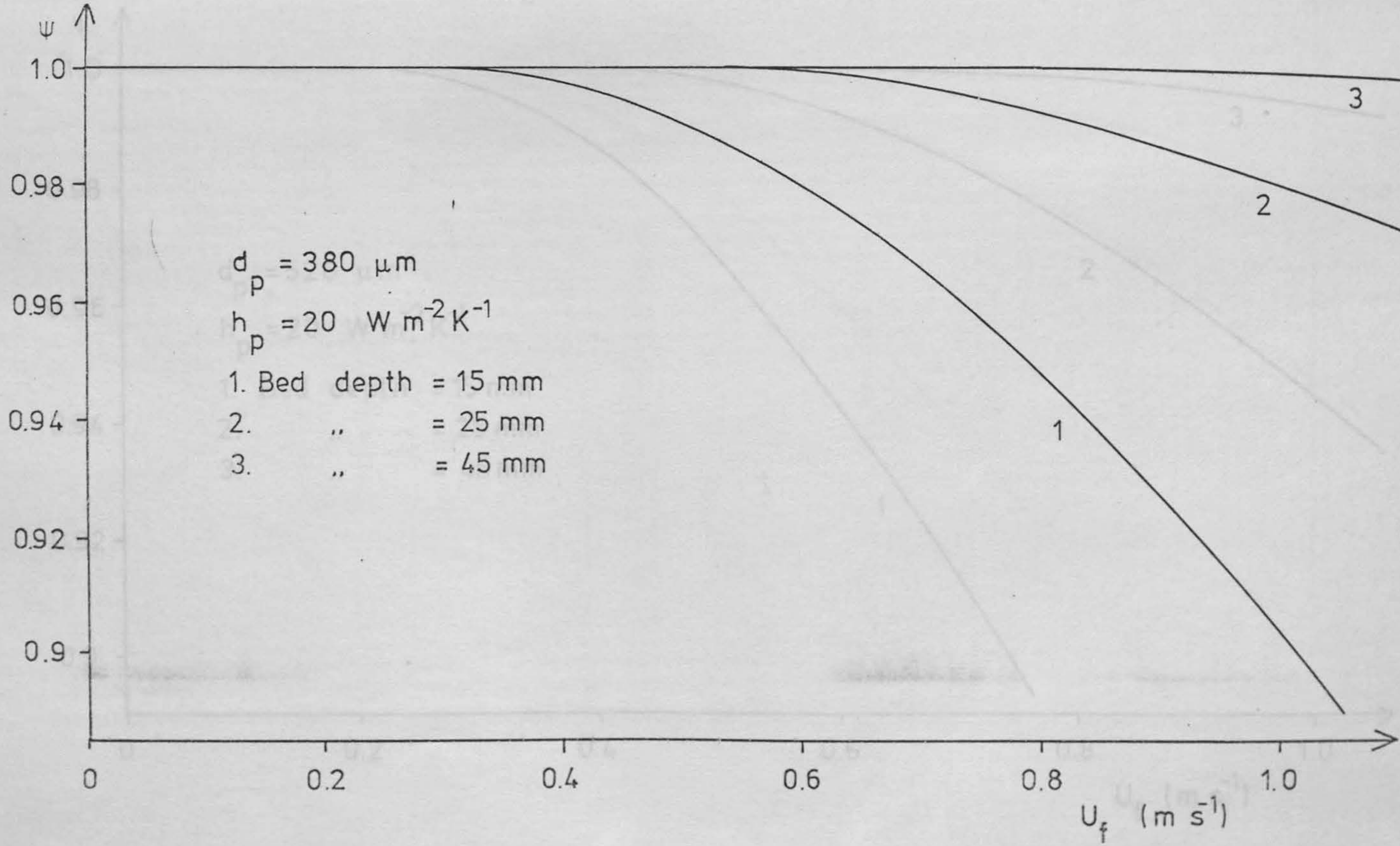


FIGURE 4.4 DEVIATION FROM THERMAL EQUILIBRIUM AT TOP OF BED

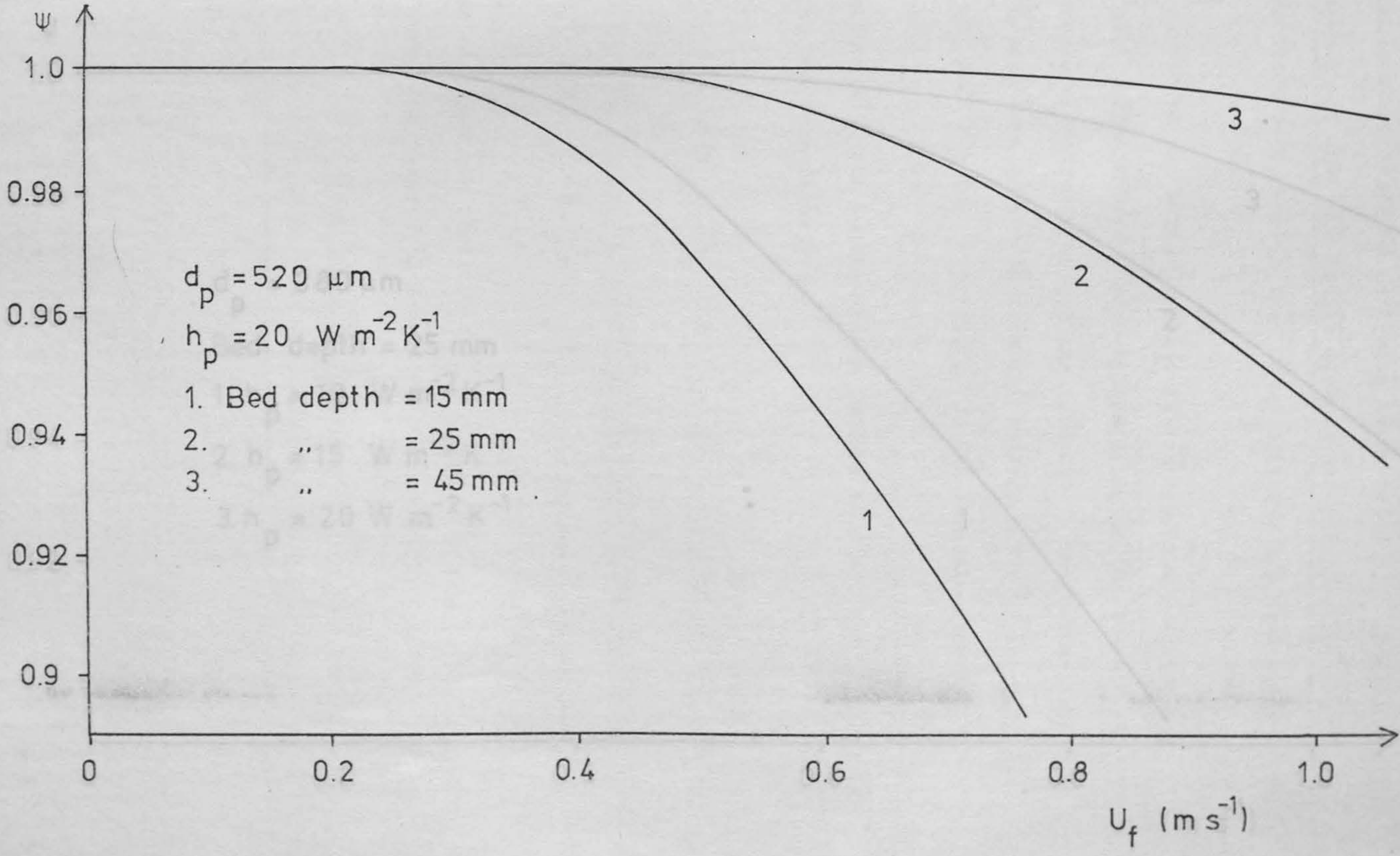
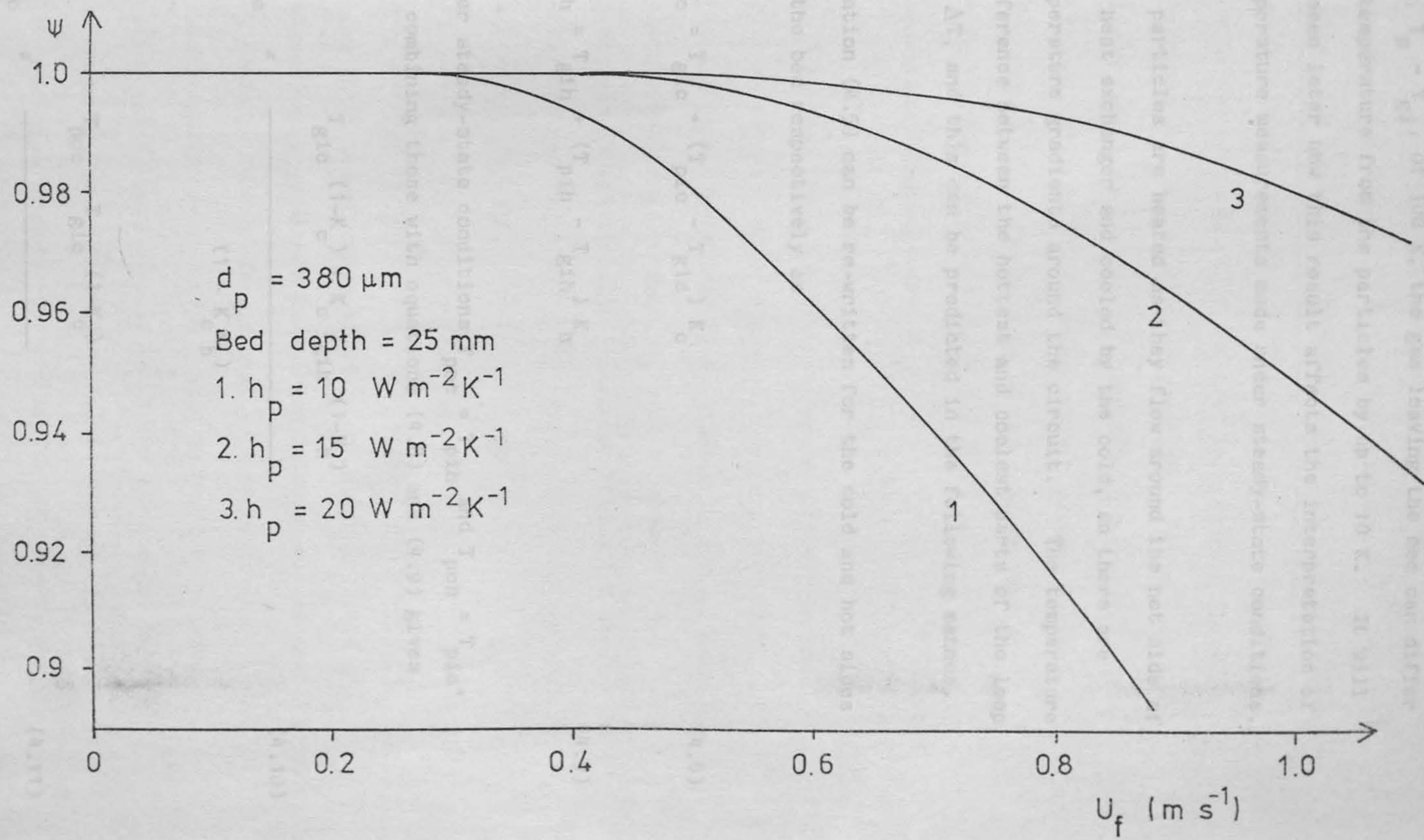


FIGURE 4.5 DEVIATION FROM THERMAL EQUILIBRIUM AT TOP OF BED



gas, $T_p - T_{gi}$, of 100 K, the gas leaving the bed can differ in temperature from the particles by up to 10 K. It will be seen later how this result affects the interpretation of temperature measurements made under steady-state conditions.

The particles are heated as they flow around the hot side of the heat exchanger and cooled by the cold, so there are temperature gradients around the circuit. The temperature difference between the hottest and coolest parts of the loop is ΔT, and this can be predicted in the following manner.

Equation (4.5) can be re-written for the cold and hot sides of the bed respectively as

$$T_{poc} = T_{gic} + (T_{pic} - T_{gic})\, K_c \qquad (4.8)$$

$$T_{poh} = T_{gih} + (T_{pih} - T_{gih})\, K_h \qquad (4.9)$$

Under steady-state conditions $T_{poc} = T_{pih}$ and $T_{poh} = T_{pic}$, and combining these with equations (4.8) and (4.9) gives

$$T_{poc} = \frac{T_{gic}\,(1-K_c) + K_c\, T_{gih}\,(1-K_h)}{(1 - K_c K_h)} \qquad (4.10)$$

and

$$T_{pic} = \frac{T_{poc} - T_{gic}\,(1-K_c)}{K_c} \qquad (4.11)$$

Hence ΔT, which is $T_{pic} - T_{poc}$, can be calculated. If ΔT is large then this will affect the calculation of the average outlet gas temperatures which must be found in order to determine the performance of the heat exchanger. These temperatures are only weakly dependent on h_p and d_p which have been chosen as 20 W m^{-2} K^{-1} and 380 μm respectively. The hot and cold gas inlet temperatures have been taken to be 250 and 40 ^oC. The results of the calculations are shown in Figures 4.6 to 4.8 which are plots of ΔT against hot side fluidising velocity, U_{fh}.

Figure 4.6 shows that ΔT does not depend strongly on the fluidising velocity, while Figure 4.7 demonstrates that ΔT is greatest when the solids' flow velocity is low. Under operating conditions, however, U_p will be about 0.2 m s^{-1} which reduces ΔT considerably. Lastly, Figure 4.8 shows the effect of bed depth on ΔT: shallow beds produce the largest values for ΔT.

In most cases it is found that ΔT is less than about 6 K with gas inlet temperatures of 40 and 250 ^oC. This difference is large enough to be readily measured, so the theory could be tested experimentally if the problems involving the measurement of U_p could be overcome. Also the difference is sufficiently small for a numerical average of the temperatures at only three points in each half of the bed to provide an adequate measure of the gas outlet temperatures. This is important because it avoids having

FIGURE 4.6 STEADY-STATE TEMPERATURE DIFFERENCES

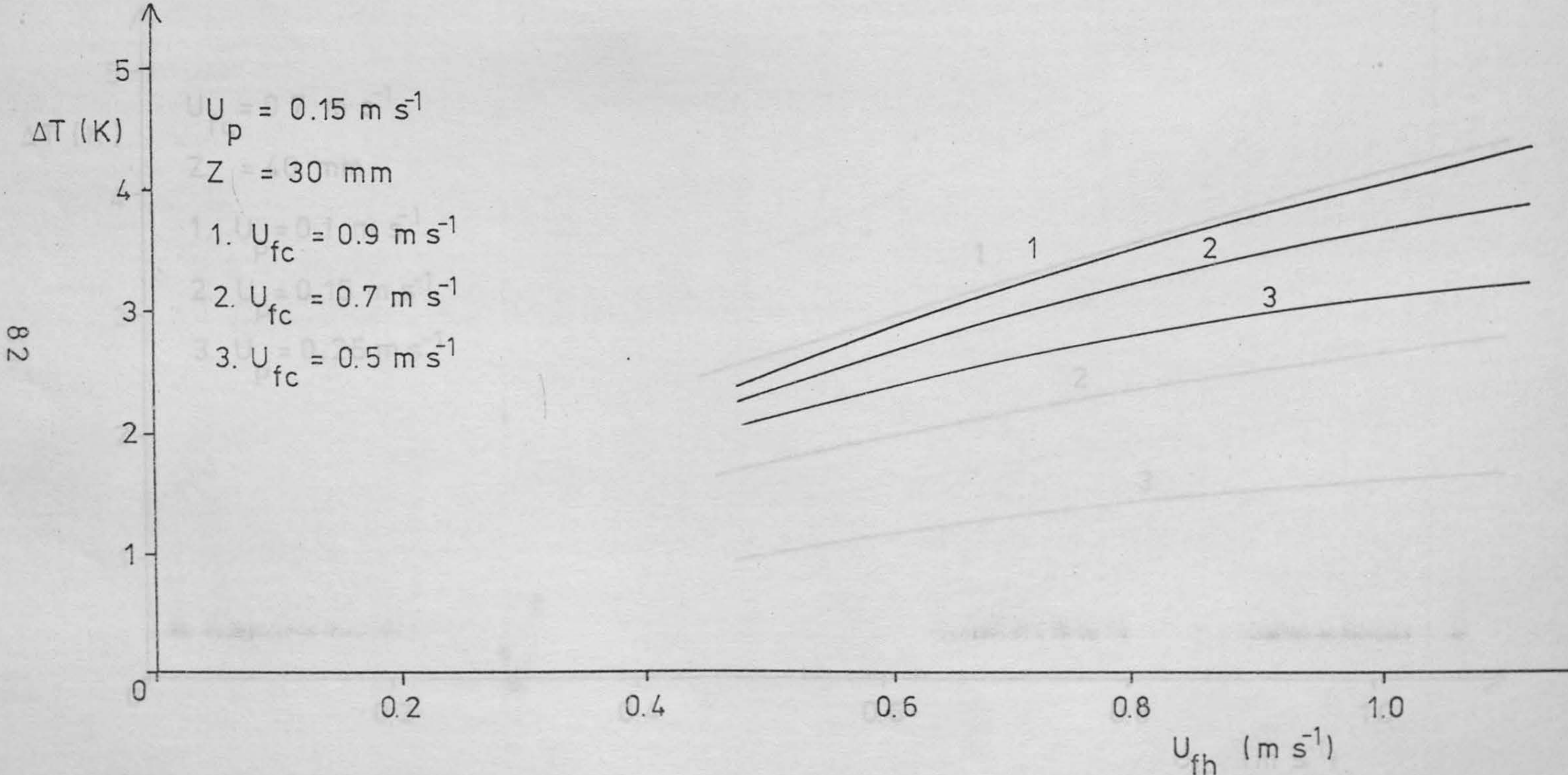


FIGURE 4.7 STEADY-STATE TEMPERATURE DIFFERENCES

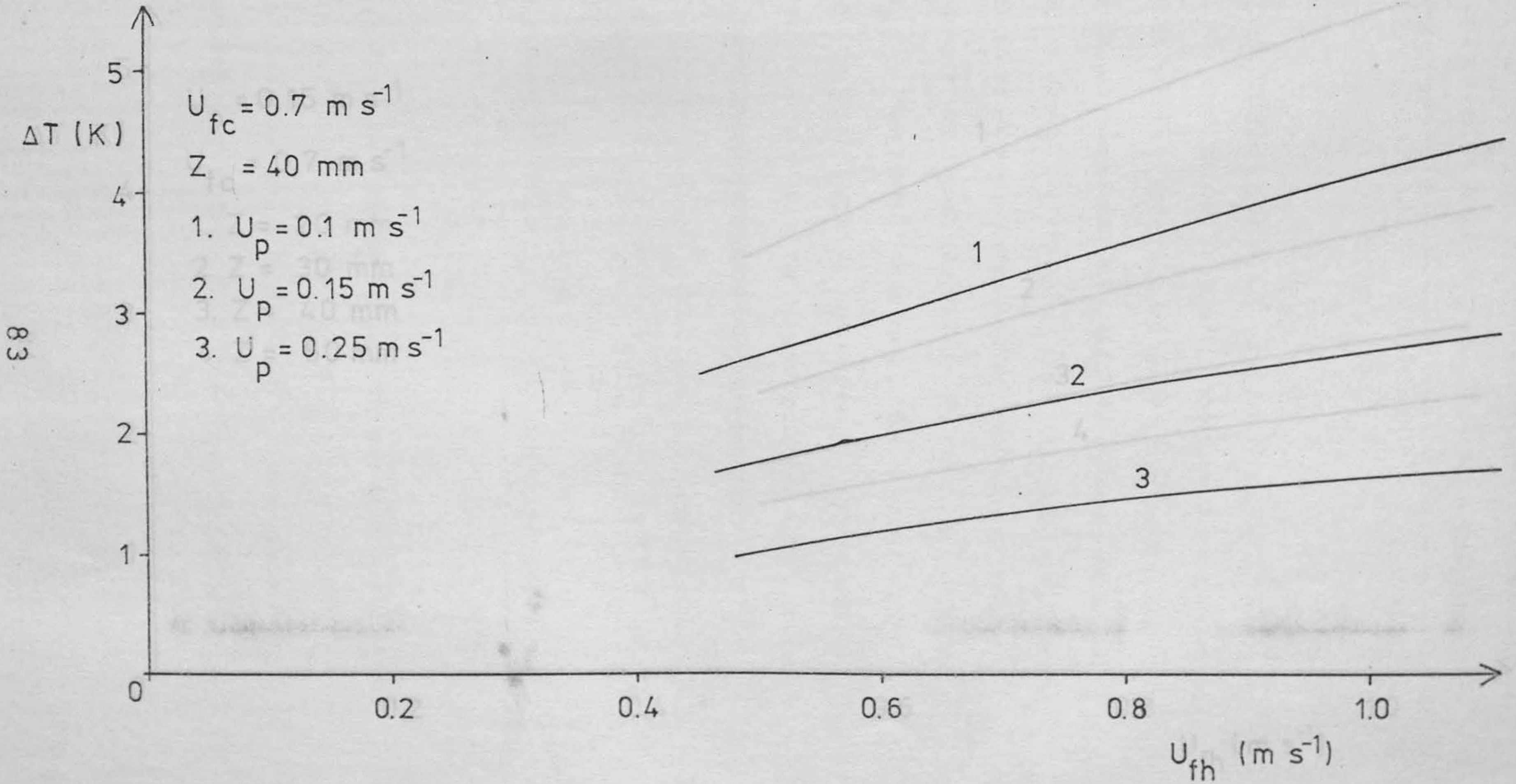
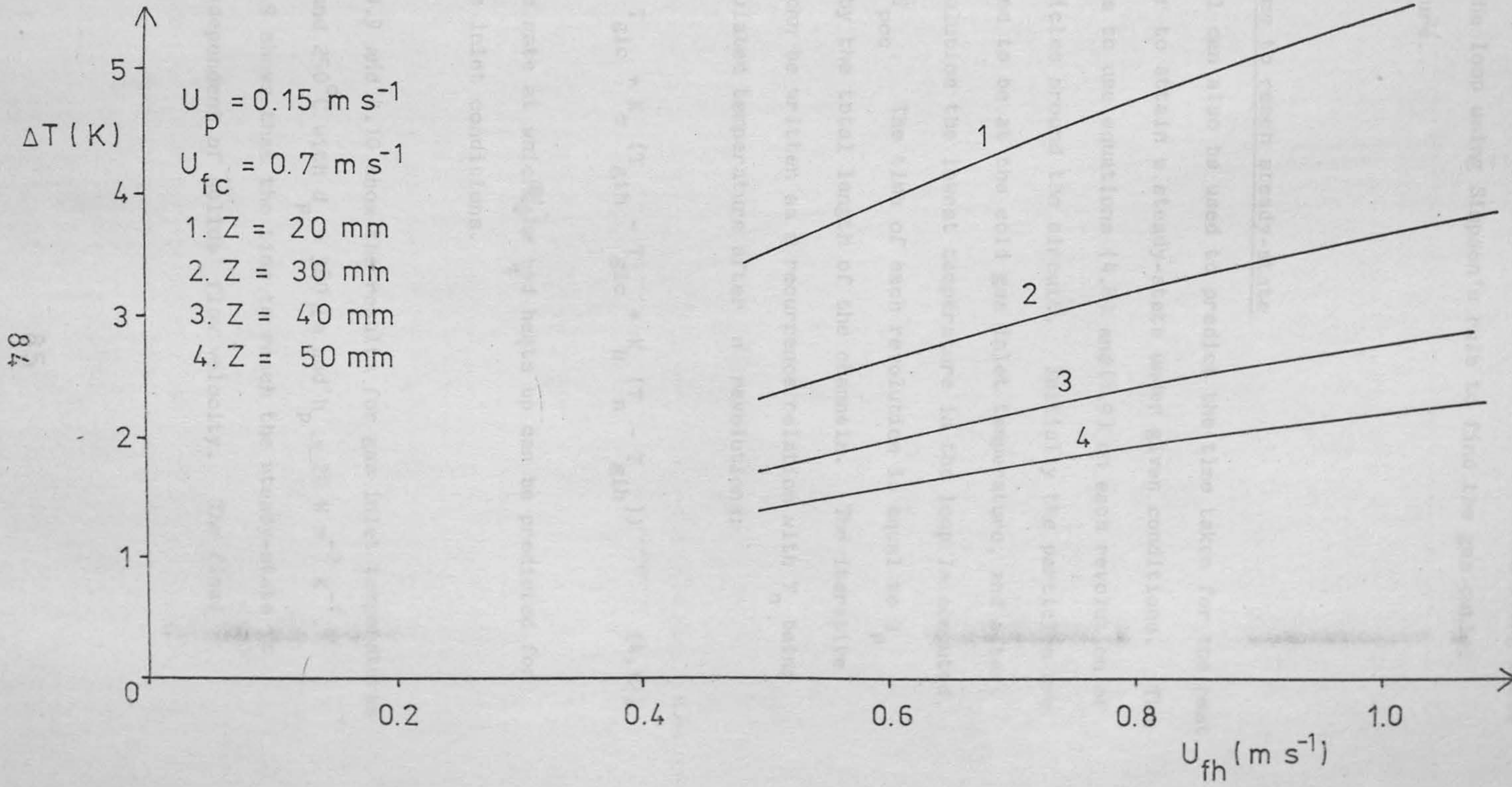


FIGURE 4.8 STEADY-STATE TEMPERATURE DIFFERENCES



to make many temperature measurements and then integrating these around the loop using Simpson's rule to find the gas outlet temperature.

4.6.2 Time to reach steady-state

The model can also be used to predict the time taken for the heat exchanger to attain a steady-state under given conditions. The method is to use equations (4.8) and (4.9) on each revolution of the particles around the circuit. Initially the particles are considered to be at the cold gas inlet temperature, and after each revolution the lowest temperature in the loop is computed, that is T_{poc}. The time of each revolution is equal to U_p divided by the total length of the channels. The iterative process can be written as a recurrence relation with T_n being the calculated temperature after n revolutions:

$$T_{n+1} = T_{gic} + K_c (T_{gih} - T_{gic} + K_h (T_n - T_{gih})) \qquad (4.12)$$

Hence the rate at which the bed heats up can be predicted for given gas inlet conditions.

Figures 4.9 and 4.10 show the results for gas inlet temperatures of 40 and 250°C with d_p = 380 μm and h_p = 20 W m^{-2} K^{-1}. Figure 4.9 shows that the time to reach the steady-state is almost independent of solids' flow velocity. The final

FIGURE 4.9 TIME TO REACH STEADY-STATE

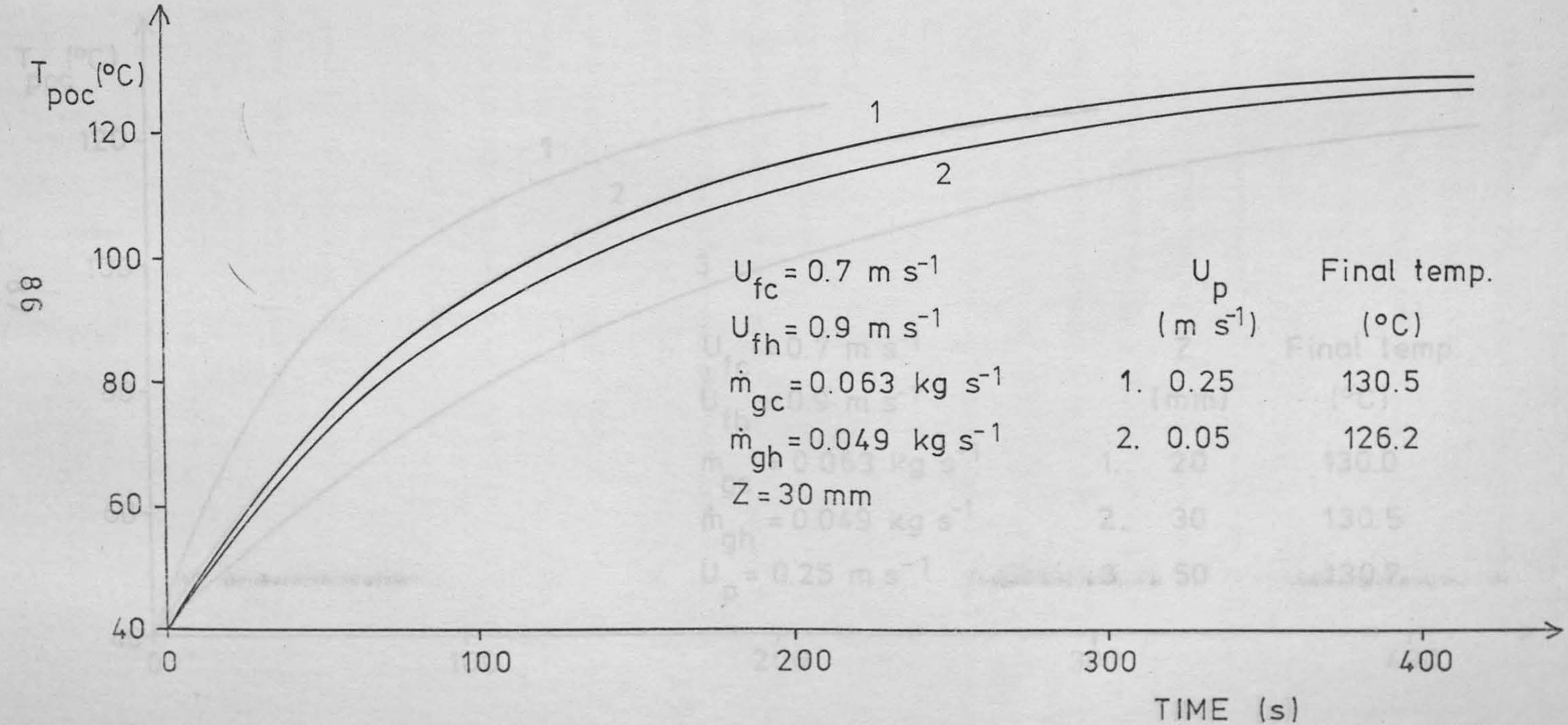
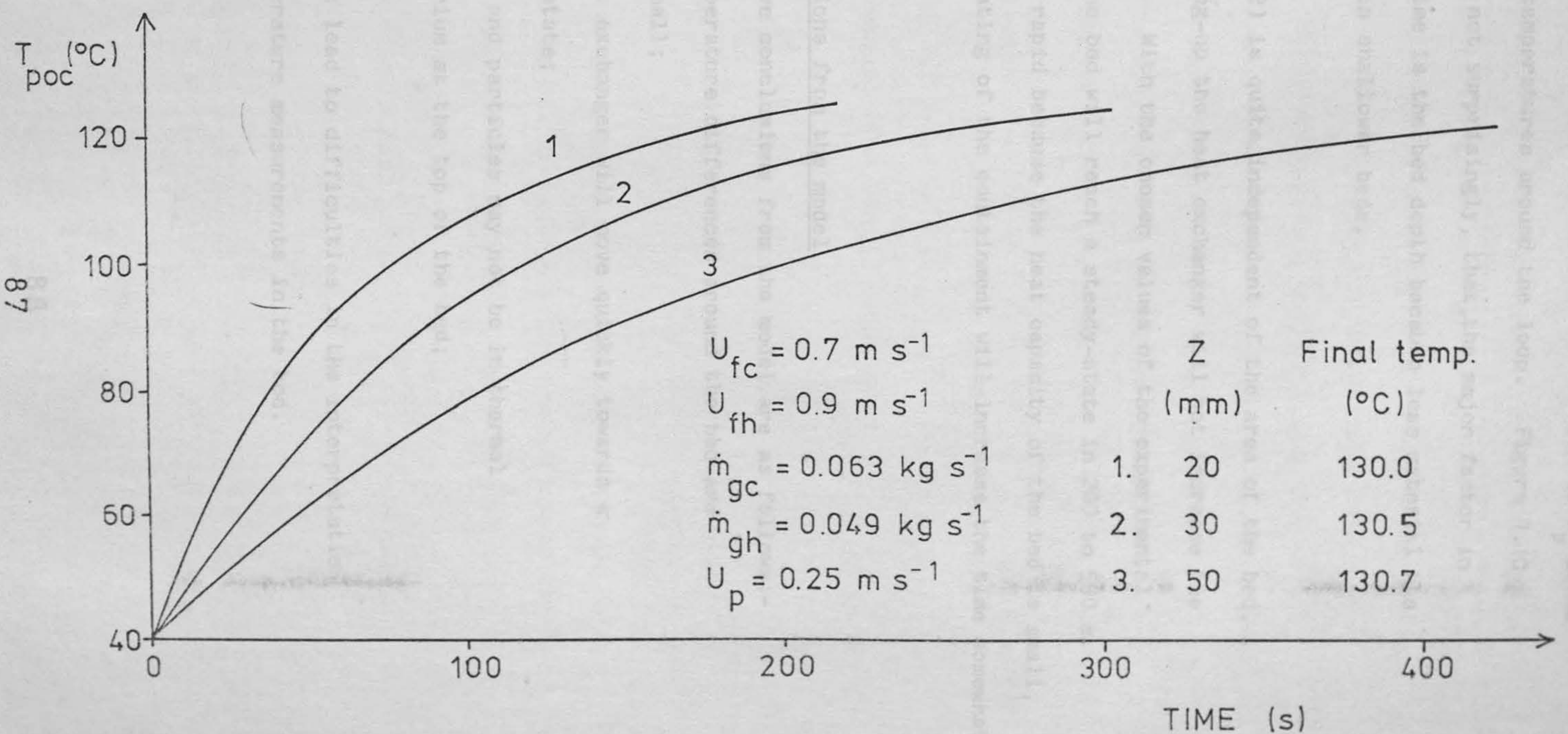


FIGURE 4.10 TIME TO REACH STEADY-STATE



temperatures are different because the variation of U_p leads to differing temperatures around the loop. Figure 4.10 demonstrates, not surprisingly, that the major factor in the heating time is the bed depth because less material has to be heated in shallower beds.

Equation (4.12) is quite independent of the area of the bed, so that scaling-up the heat exchanger will not increase the heating time. With the chosen values of the experimental parameters, the bed will reach a steady-state in 200 to 500 s. This is quite rapid because the heat capacity of the bed is small, though the heating of the containment will increase the time somewhat.

4.6.3 Conclusions from the model

The qualitative conclusions from the model are as follows:-

(i) the temperature differences around the bed are quite small;

(ii) the heat exchanger will move quickly towards a steady-state;

(iii) the gas and particles may not be in thermal equilibrium at the top of the bed;

and

(iv) this can lead to difficulties in the interpretation of temperature measurements in the bed.

4.7 EQUATIONS FOR THE EFFECTIVENESS OF THE HEAT EXCHANGER

For the purposes of completeness the basis for the evaluation of the performance of the heat exchanger will be presented here. There are several criteria which have been developed for rating the performance of heat exchangers. These range from simple ratios of heat transferred to heat available, through to complicated concepts in which entropy increase is considered [98]. As the heat exchanger operates over a range of both hot and cold side mass flows and temperatures, a measure of effectiveness, η, is required which takes these into account.

The heat exchanger is considered to be a purely gas-to-gas co-current (parallel flow) heat exchanger. The efficiency of a perfect parallel flow unit is given by

$$E = \frac{\dot{m}_{gc} C_{gc}}{\dot{m}_{gc} C_{gc} + \dot{m}_{gh} C_{gh}} \tag{4.13}$$

The efficiency of any real heat exchanger will be defined here as the ratio of the heat transferred from the hot gas to the cold, to the total heat available for transfer. That is

$$E = \frac{\dot{m}_{gc} C_{gc} (T_{goc} - T_{gic})}{\dot{m}_{gh} C_{gh} (T_{gih} - T_{gic})} \tag{4.14}$$

The effectiveness of the heat exchanger is its actual efficiency from equation (4.14) divided by the best that can be achieved, namely equation (4.13). Therefore η is given by

$$\eta = \frac{(\dot{m}_{gc} C_{gc} + \dot{m}_{gh} C_{gh})(T_{goc} - T_{gic})}{\dot{m}_{gh} C_{gh} (T_{gih} - T_{gic})} \qquad (4.15)$$

This is the measure of heat exchanger performance that is used in all the analyses presented in this thesis. As a consequence of the above definition of η, in some circumstances the crossflow contribution to the heat exchange can be sufficiently large to render η slightly greater than unity.

CHAPTER FIVE

EXPERIMENTAL HEAT EXCHANGER

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EXPERIMENTAL HEAT EXCHANGER

5.1 INTRODUCTION

This chapter describes the design and construction of the experimental rig which was built to study the ideas contained in the patent [7]. Firstly the aims of the experiments are stated followed by the various constraints placed upon the size of the rig. Then the rig itself is described in detail together with its instrumentation and lastly the experimental procedure is outlined.

5.2 AIMS OF THE EXPERIMENTS

Overall the intention was to establish whether a heat exchanger based on the patent would actually work. Within this very broad scope, several well-defined aims can be specified as follows:-

(i) to study the flow of the solids;

(ii) to measure the heat recovery characteristics of the heat exchanger;

(iii) to measure the leakage between the two gas streams;

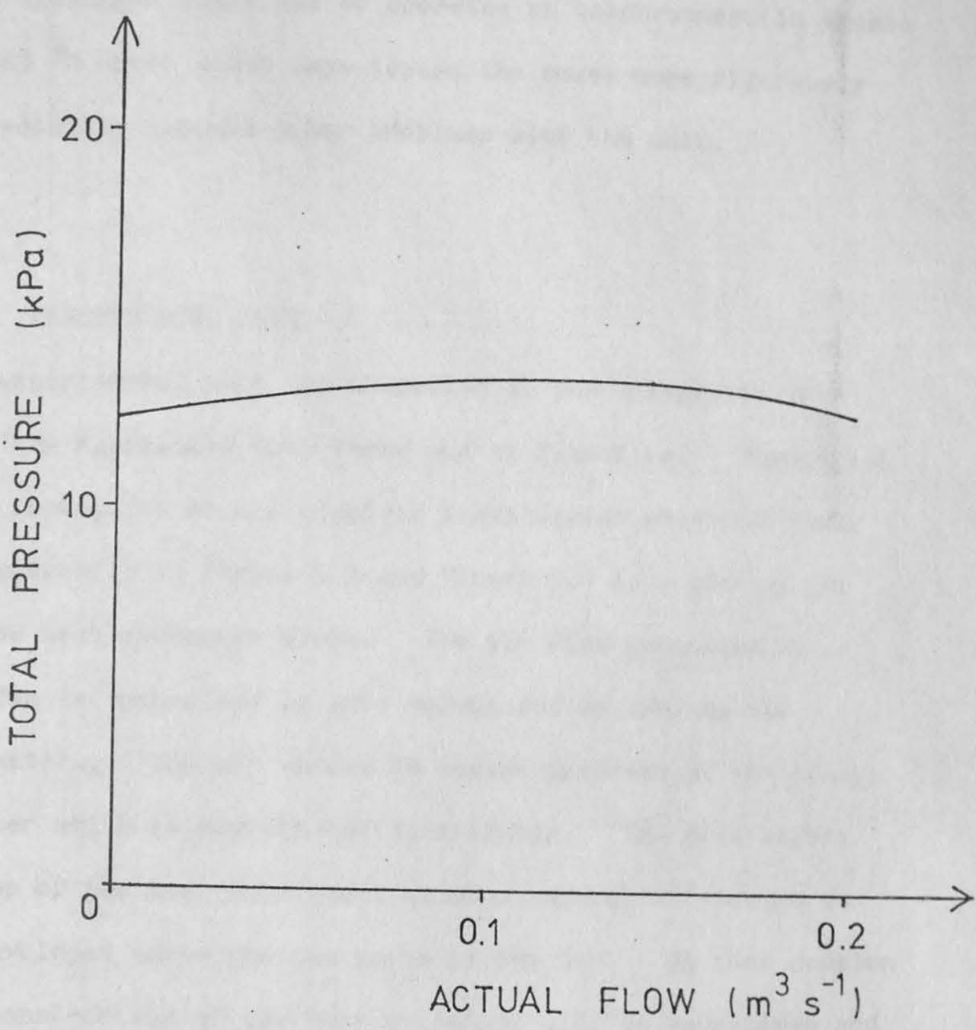
and (iv) to obtain sufficient understanding of the operation of the heat exchanger for a much larger unit to be designed with confidence.

5.3 SIZING OF THE EXPERIMENTAL UNIT

The size of the experimental unit was constrained by the amount of fluidising air available. It had been decided that an existing fan would be used to provide the air to fluidise the bed, and its performance therefore limited the area of the bed. The fan was of the centrifugal type, a Secomak 492/2, and its performance characteristic is shown in Figure 5.1. The maximum pressure available of about 12 kPa was ample for any realistic bed, but the maximum air flow of 0.2 $m^3\ s^{-1}$ presented a limitation. To be able to attain a fluidising velocity of 1.5 $m\ s^{-1}$ at room temperature, the area of the bed could only be 0.13 m^2. However half of the bed will be operating at an elevated temperature so the fluidising air in that region will expand. The total area of the bed when constructed was 0.16 m^2 which provided sufficiently large fluidising velocities.

The second major constraint on the unit was the form of heating to be used to produce the hot air stream. Electric heating appeared to be the easiest to control and a Secomak 441/2 heater was used. This could supply up to 9 kW with a maximum outlet temperature of 400 ^oC. This was upgraded to a Secomak 15/2, rated at 18 kW, after several element failures in the earlier heater. In fact, at the air flows used, the outlet temperature was under 250 ^oC. This was a limitation in that the necessary heat transfer measurements would have been easier to carry out if the temperature differences had been larger.

FIGURE 5.1 PERFORMANCE CURVE OF SECOMAK FAN 492/2



Thus the size of the unit and its operating temperature were essentially fixed beforehand and the experimental programme was somewhat restricted as a result. In particular the heat exchanger could not be operated at temperatures in excess of 500 ^{o}C which would have tested the model more rigorously and possibly exposed other problems with the unit.

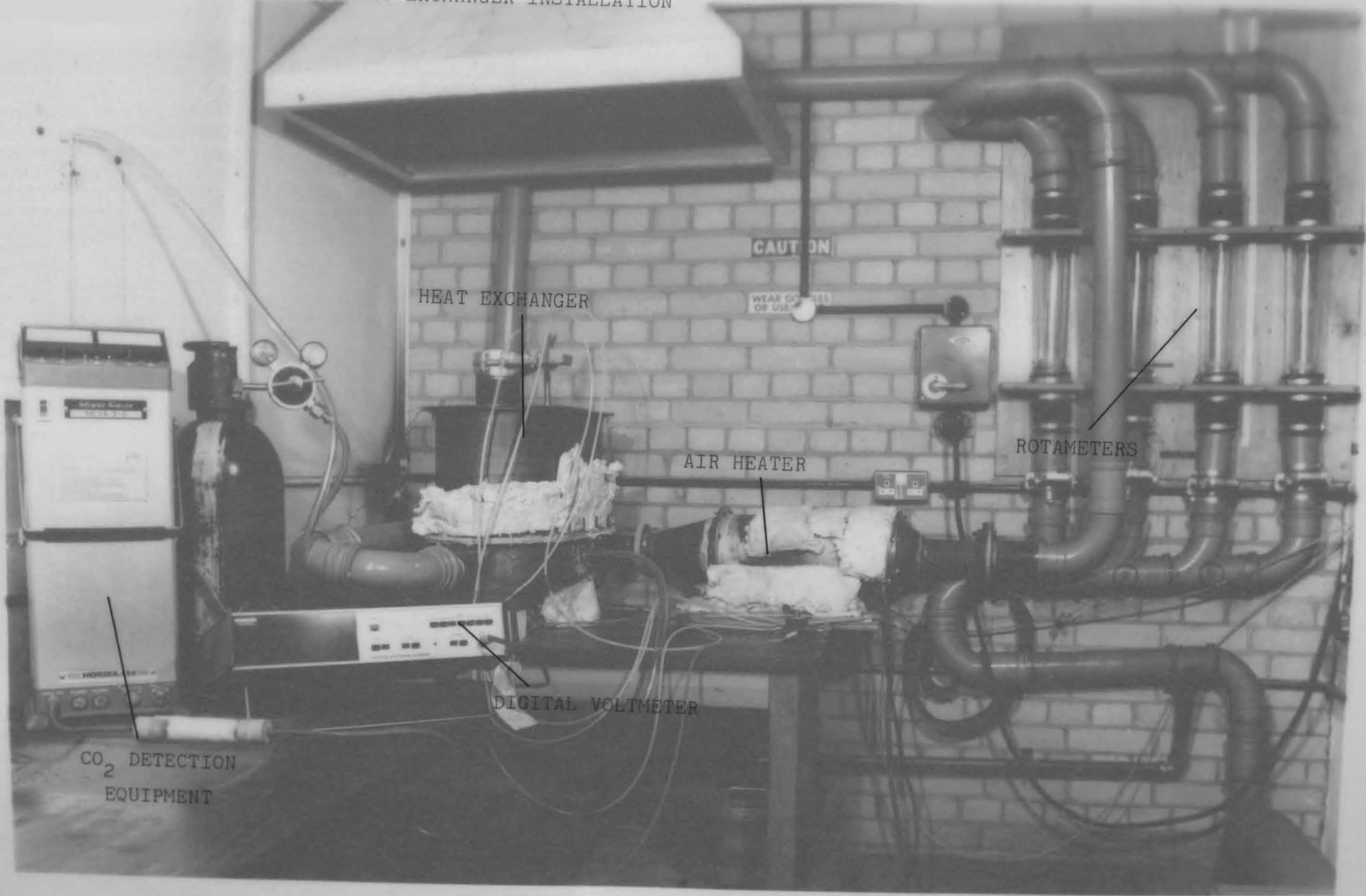
5.4 EXPERIMENTAL UNIT

The experimental unit was assembled at the University from sections fabricated both there and at Fluidfire. Figure 5.2 is a photograph of the complete installation which is shown schematically in Figure 5.3 and Figure 5.4 is a photograph of the heat exchanger alone. The air flow generated by the fan is controlled by gate valves and metered by the Rotameters. One air stream is heated upstream of the plenum chamber which is beneath the distributor. The gate valves on top of the bed containment allow pressure differences to be developed above the two parts of the bed. In this section the construction of the heat exchanger will be considered and the metering equipment will be dealt with in Section 5.5.

5.4.1 Plenum chamber

The plenum chamber acts as a settling volume which removes velocity and pressure profiles from the incoming flow before the air passes through the distributor. In the heat exchanger the plenum is split into two sections, one supplying hot air

FIGURE 5.2 EXPERIMENTAL HEAT EXCHANGER INSTALLATION



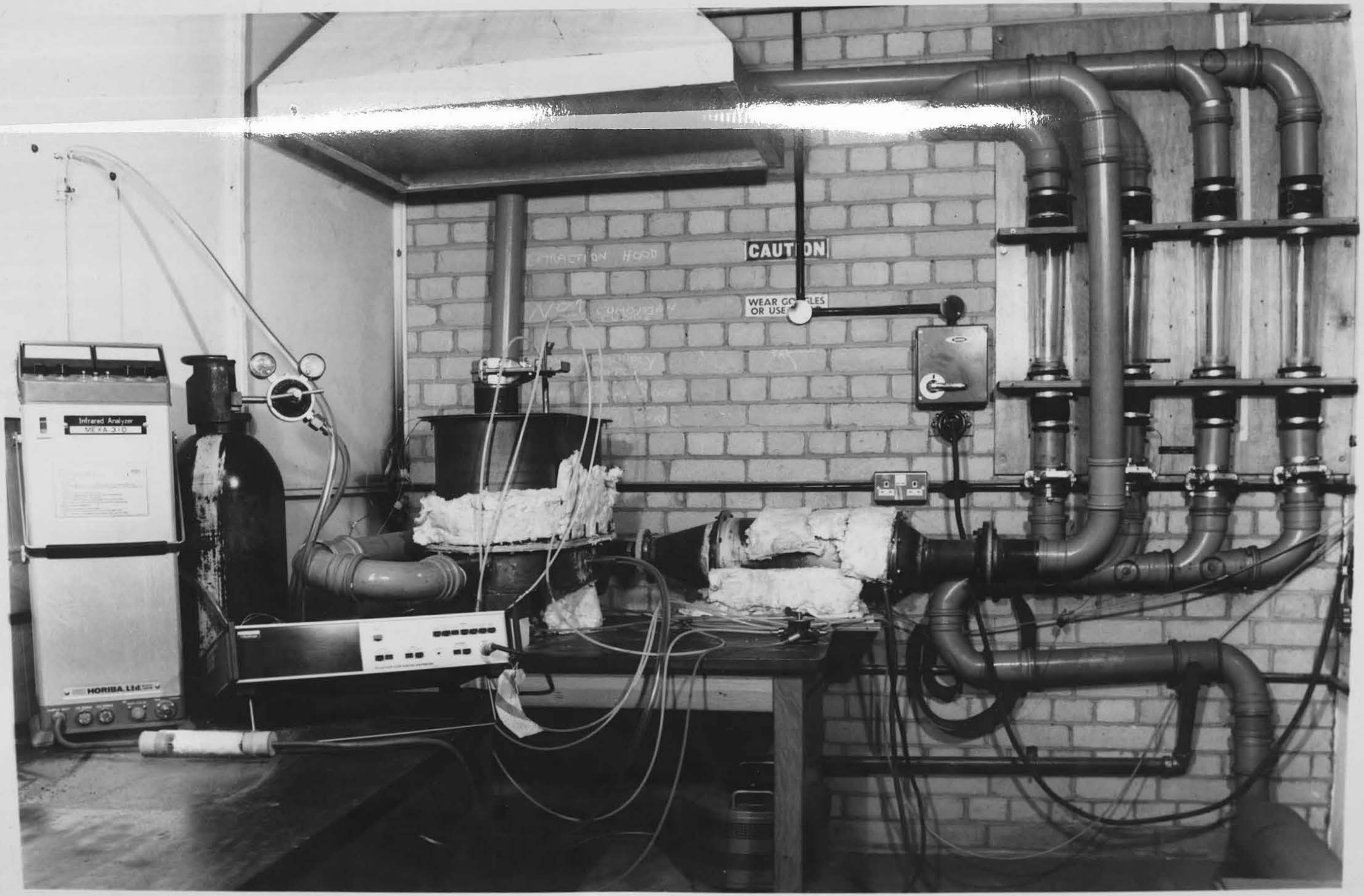
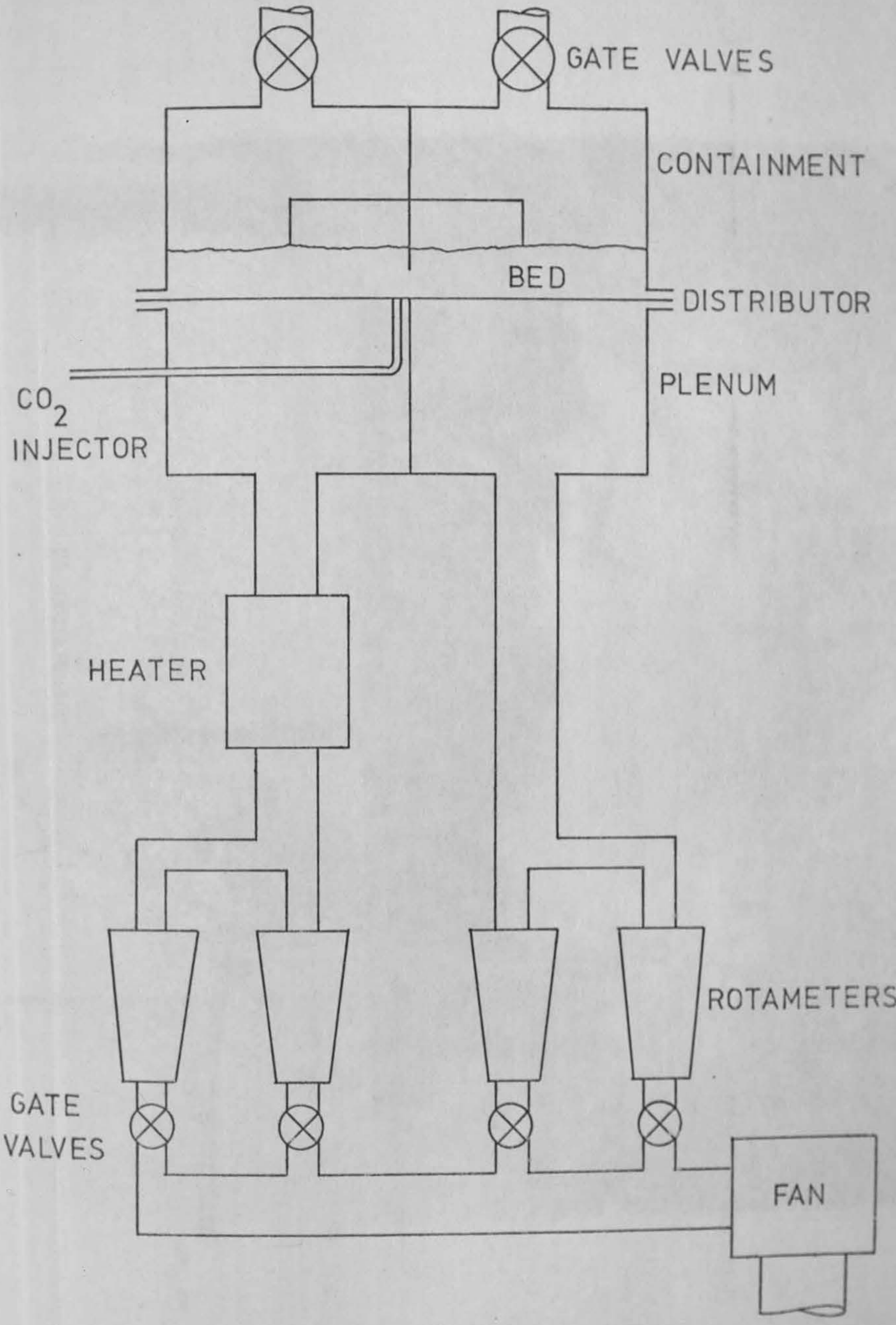
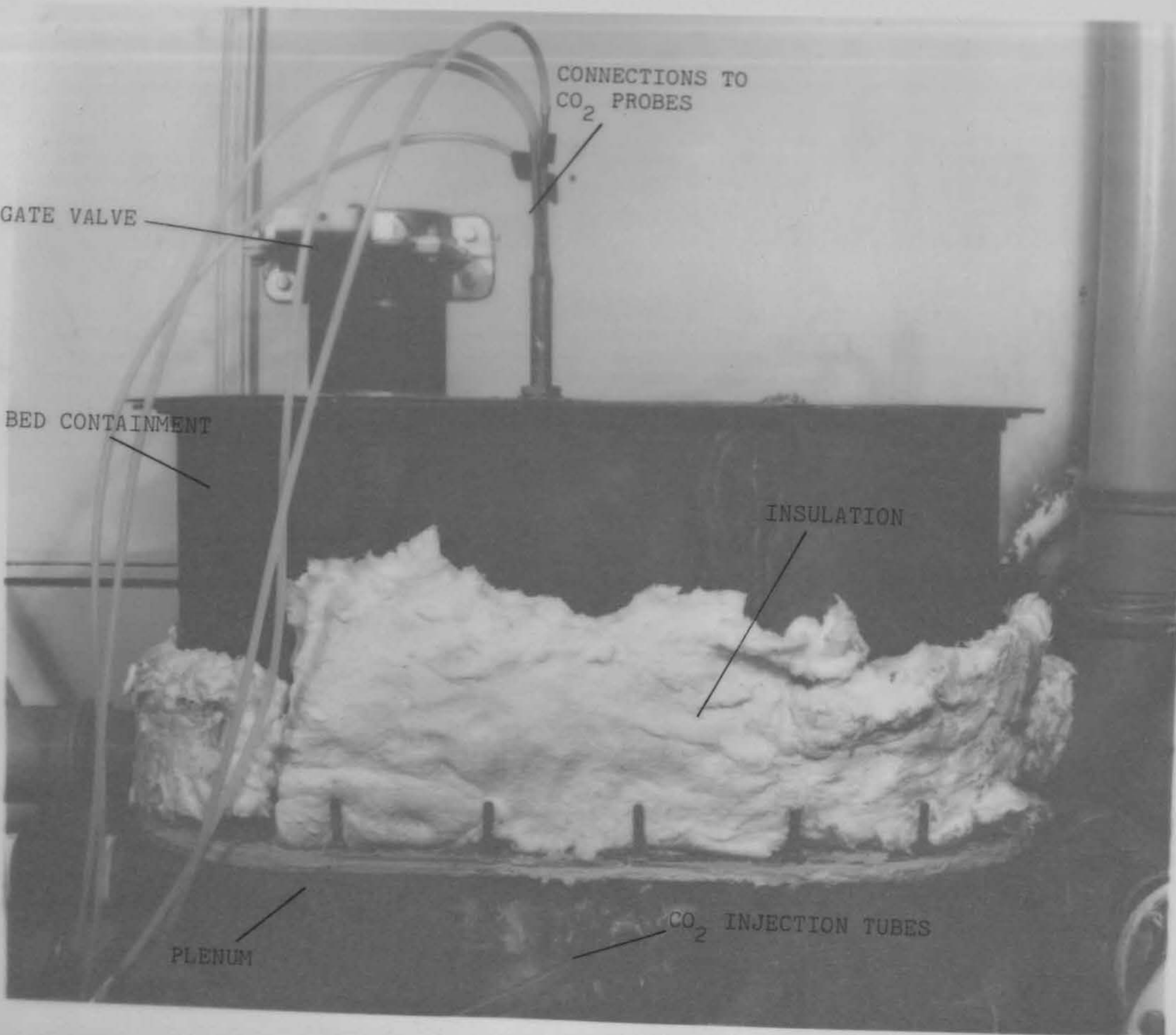
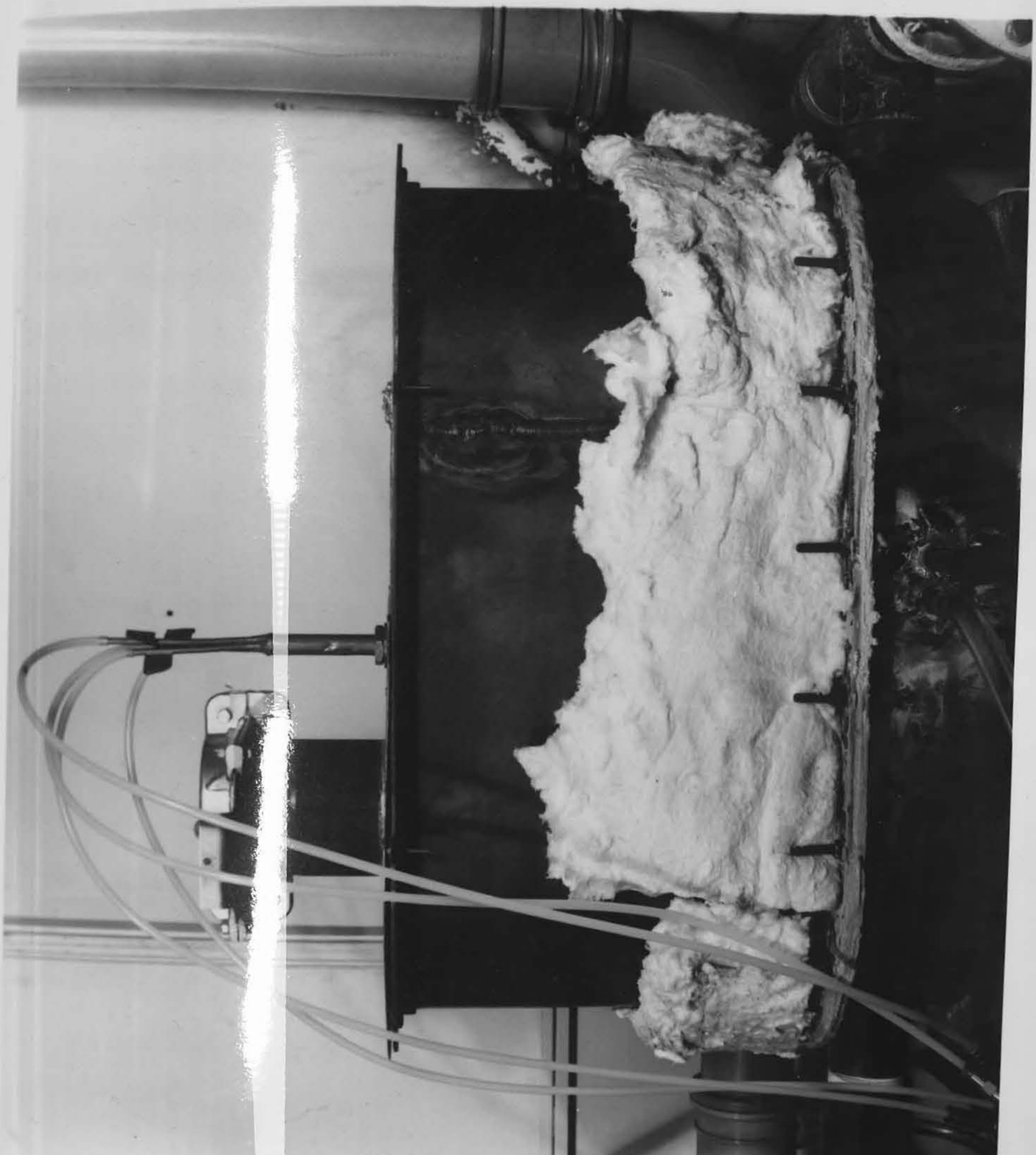

CAUTION
Infrared Analyzer
MEXA-310
HORIBA.Ltd

FIGURE 5.3 SCHEMATIC DIAGRAM OF EXPERIMENTAL SYSTEM




CONNECTIONS TO
CO_2 PROBES
GATE VALVE
BED CONTAINMENT
INSULATION
PLENUM
CO_2 INJECTION TUBES

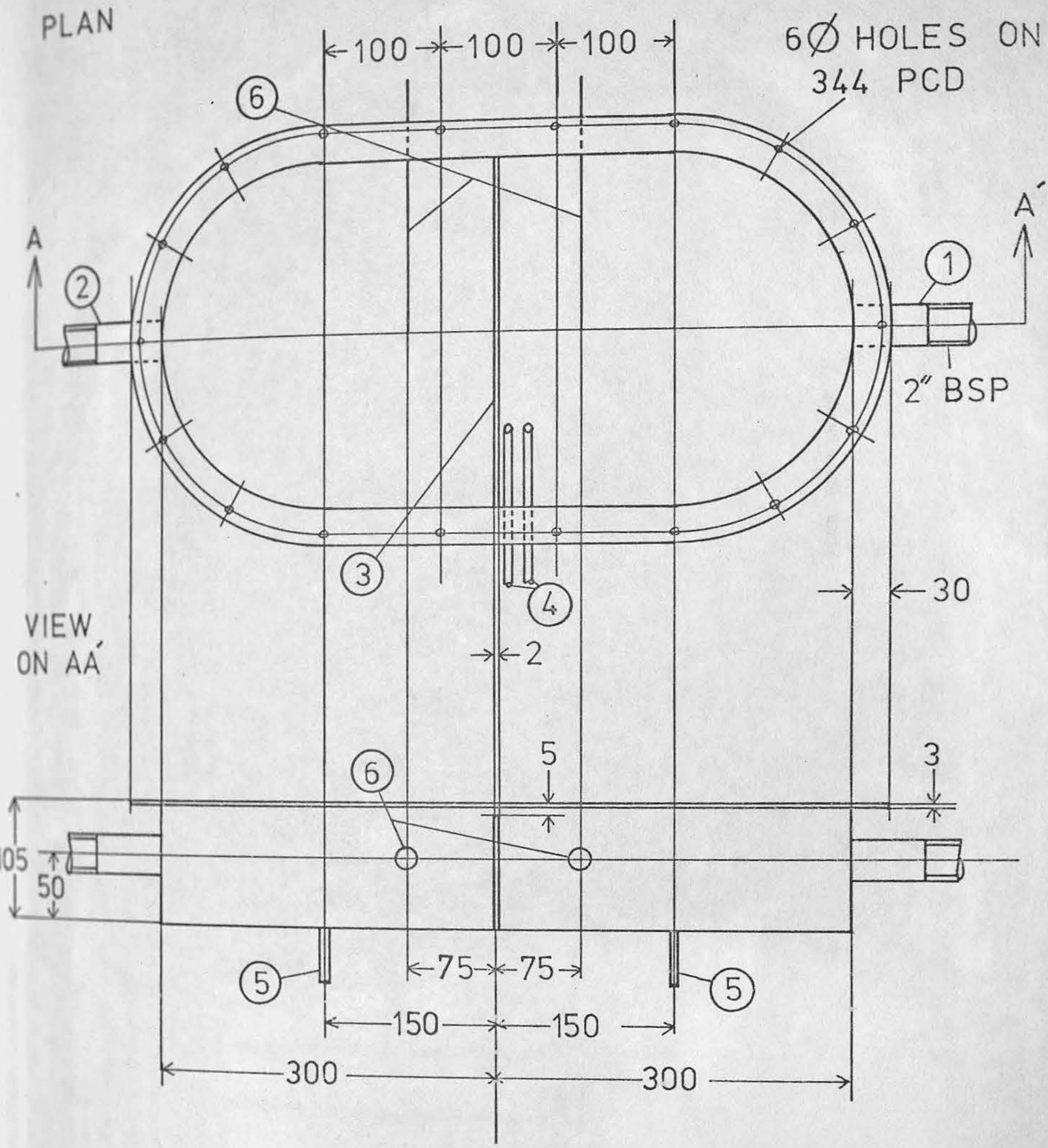


and the other cold air, as shown in Figures 5.5 and 5.6. These two sections are nominally of equal size because the distributor is also divided into two equal areas. However the distributor, and hence the plenum, can be split into two unequal zones should the heat exchanger duty require it.

The air enters the plenum from a 2 inch (50 mm) diameter pipe, after having been metered. The static pressure in each section of the plenum is measured at the two 6 mm pressure tappings in the base of the plenum. The inlet air temperatures are determined using two chromel/alumel thermocouples which penetrate the wall of the plenum. The flange on the top of the plenum seals against the underside of the distributor with an asbestos gasket.

The plenum divider is the most critical component in the plenum chamber. Ideally this should seal directly against the underside of the distributor to prevent leakage between the two sections of the plenum. This is difficult to achieve in practice, particularly in a small development rig where ease of dismantling is important. Thus a simple labyrinth seal was used, the form of which will be explained in the next section, and this performed adequately. From the point of view of the plenum, the pertinent dimensions are the thickness of the divider (2 mm) and the gap of 5 mm between the top of the divider and the face of the flange.

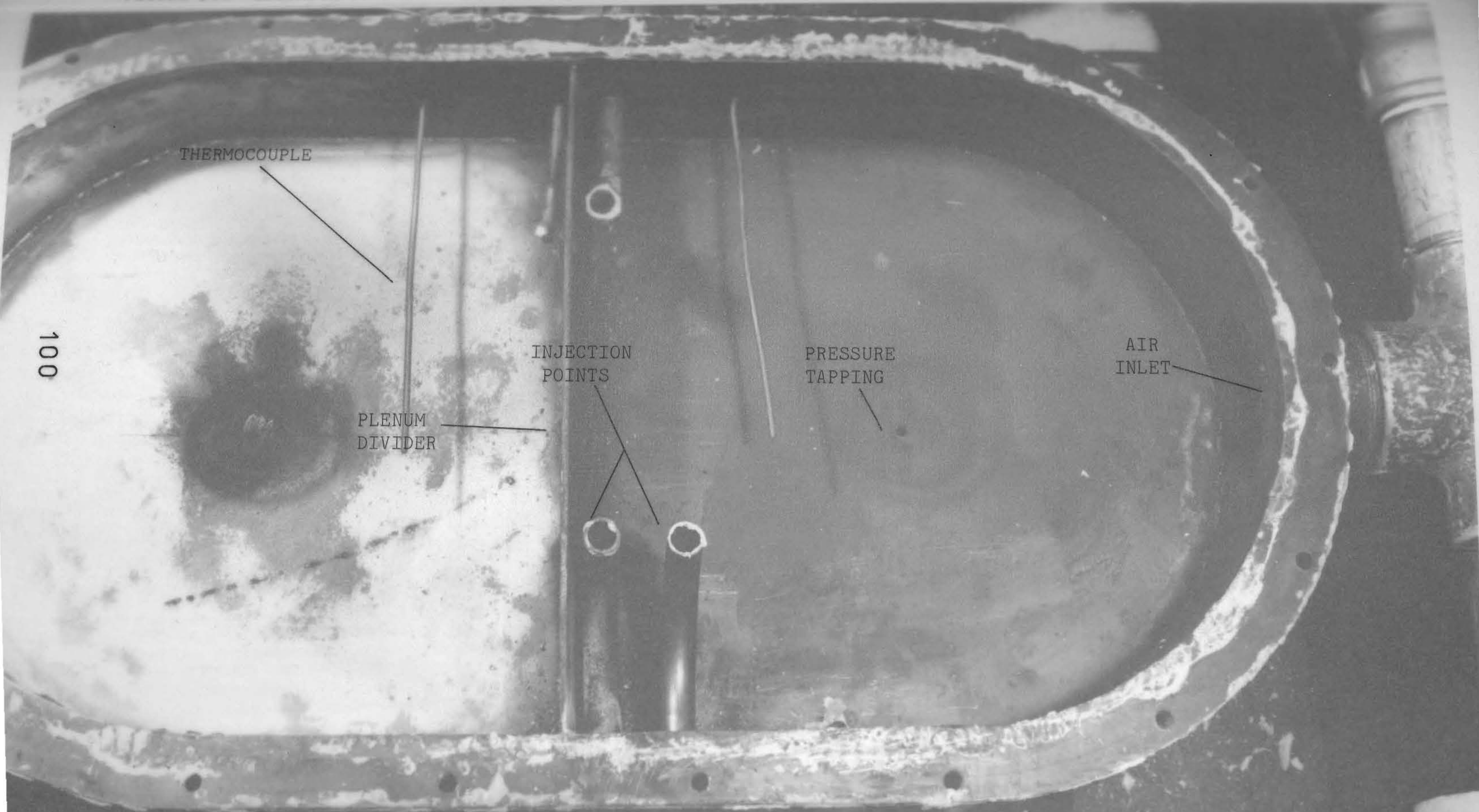
FIGURE 5.5 PLENUM CHAMBER DRAWINGS



DIMENSIONS : mm
SCALE : 1/6 FULL SIZE
MATERIAL : 1 mm STEEL
UNLESS SPECIFIED

1. HOT AIR INLET
2. COLD AIR INLET
3. PLENUM DIVISION
4. CO_2 INJECTORS (FIGURE 5.14)
5. 6 mm PRESSURE TAPPINGS
6. THERMOCOUPLES

FIGURE 5.6 INTERIOR OF PLENUM CHAMBER

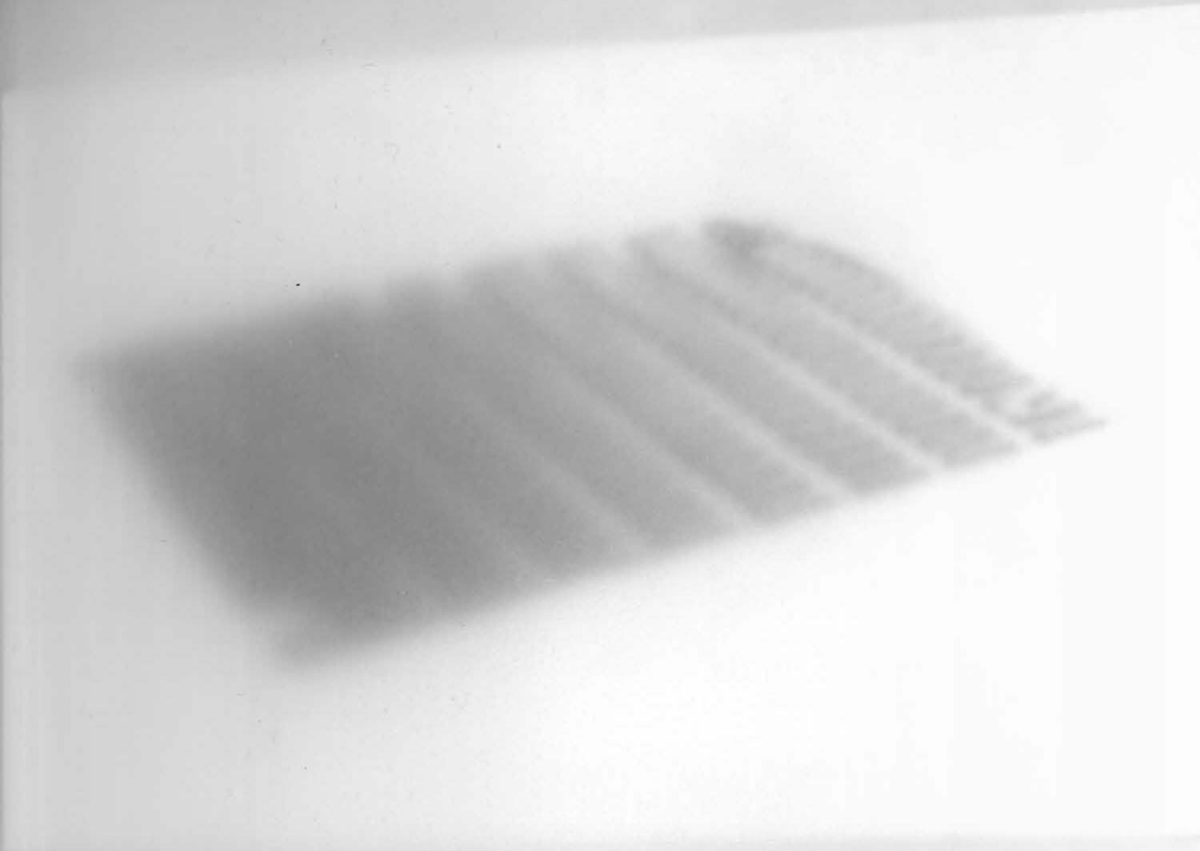


5.4.2 Distributor plate

The distributor plate provides a uniform supply of air to the bed of particles above it. In the heat exchanger the distributor also produces a directed air stream to move the particles as explained in the patent specification (Section 3.2). The distributor material was supplied by N. Greenings Ltd. of Warrington although identical material is now made by Stone-Platt Electrical Ltd. for Fluidfire. Photographs of a small piece of the material are shown in Figure 5.7 and its critical dimensions are given in Figure 5.8. The plate is formed by punching stainless steel sheet with a specially made die. This process is difficult to control accurately so the slot width of 0.4 mm is subject to considerable variation, possibly as much as 20%. The overall percentage open area of the distributor is a nominal 4.4%.

The distributor plate for the heat exchanger was made from four pieces of Greenings' sheet to form a complete loop. This is seen in the drawing of the distributor in Figure 5.9 and the photograph of the distributor when installed, Figure 5.10. The pieces of the distributor were butt-welded to each other to minimise the disruption to the fluidising air flow. The central wall separates the two opposing solids' flows along the straight sections of the loop.

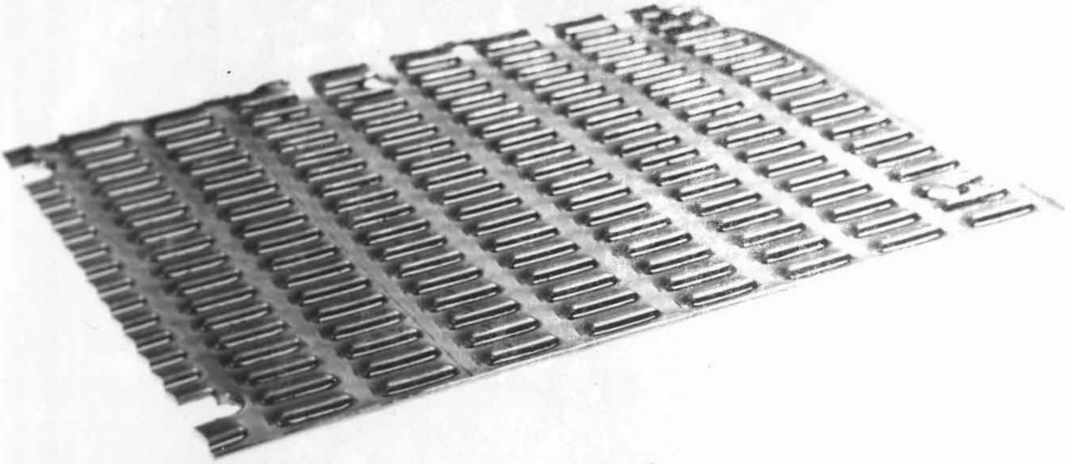
The two sheets beneath the distributor (Figure 5.9) fit inside the plenum and over the plenum divider which produces a simple labyrinth seal between the two fluidising air steams. The seal worked very



SMALL PIECE OF DISTRIBUTOR PLATE MATERIAL

DETAILED SECTION OF DISTRIBUTOR

FIGURE 5.7 DETAIL OF DISTRIBUTOR PLATE



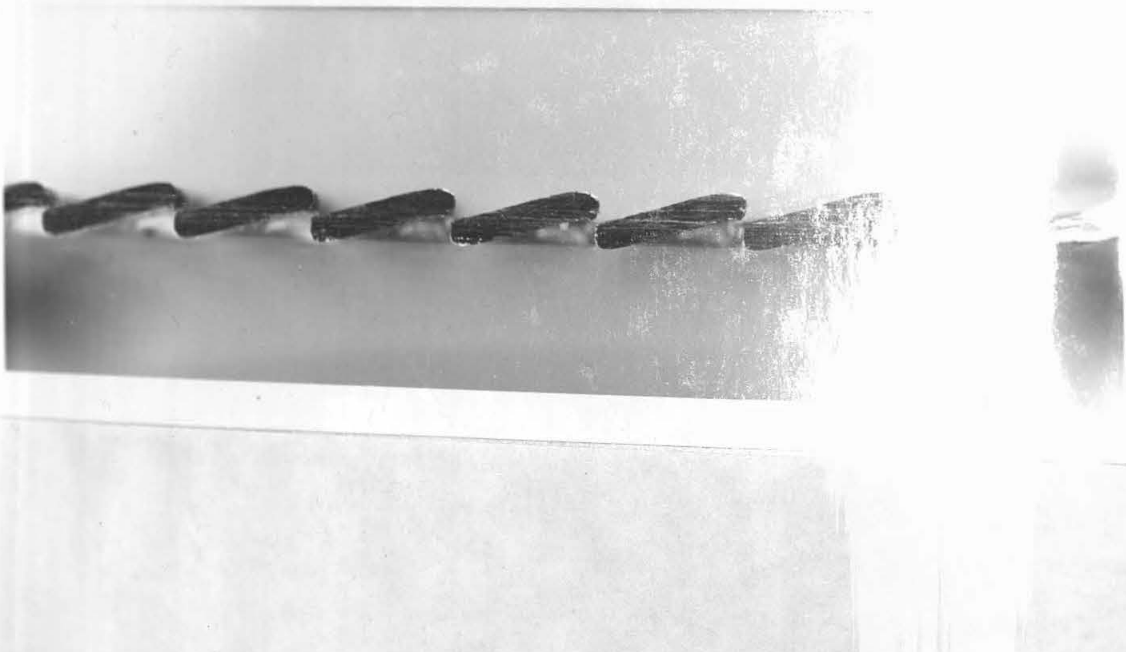
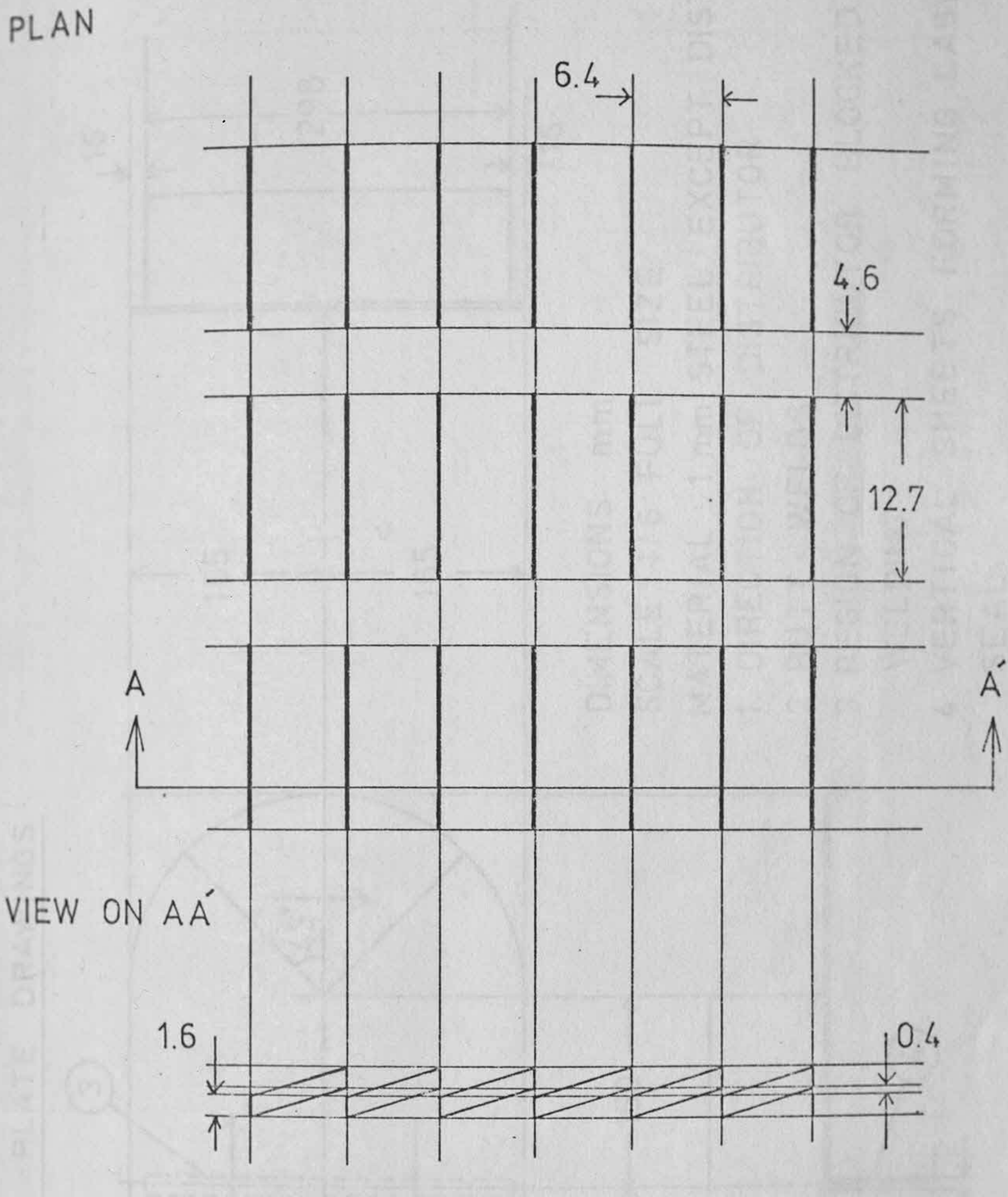


FIGURE 5.8 DISTRIBUTOR PLATE CRITICAL DIMENSIONS



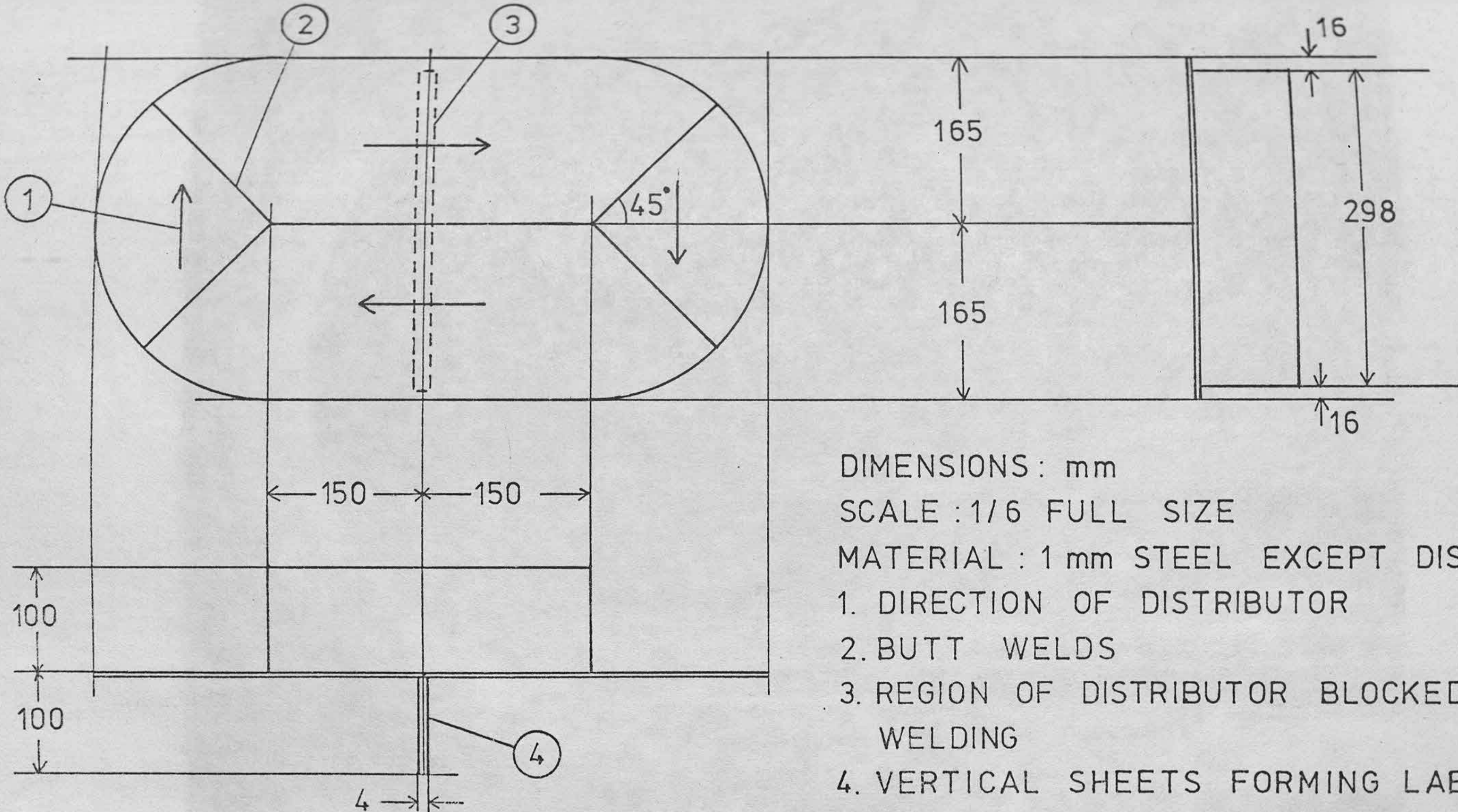
DIMENSIONS : mm

SCALE : TWICE FULL SIZE

MATERIAL : 316 STAINLESS STEEL

SUPPLIER : N. GREENINGS LTD., WARRINGTON

FIGURE 5.9 DISTRIBUTOR PLATE DRAWINGS



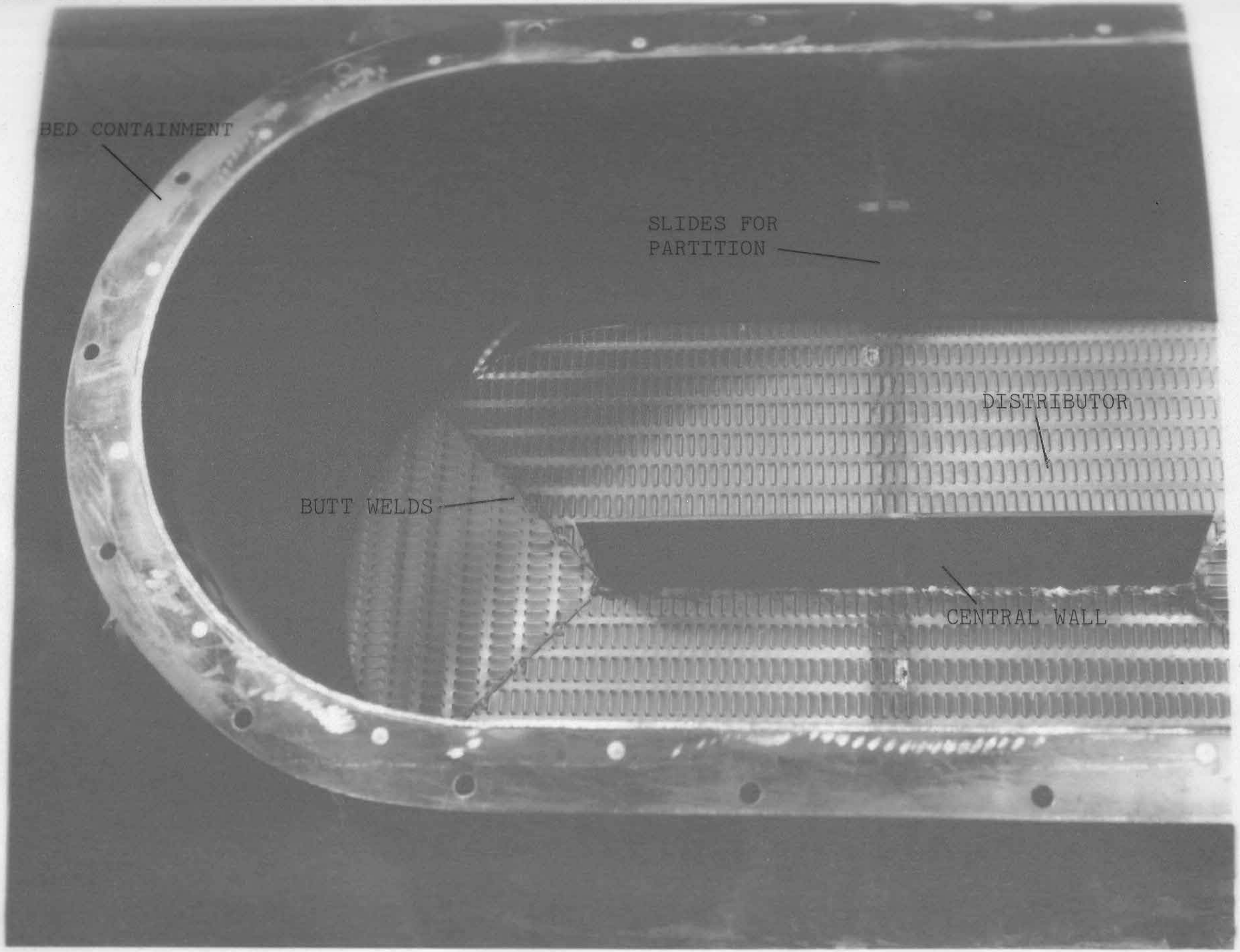
DIMENSIONS: mm

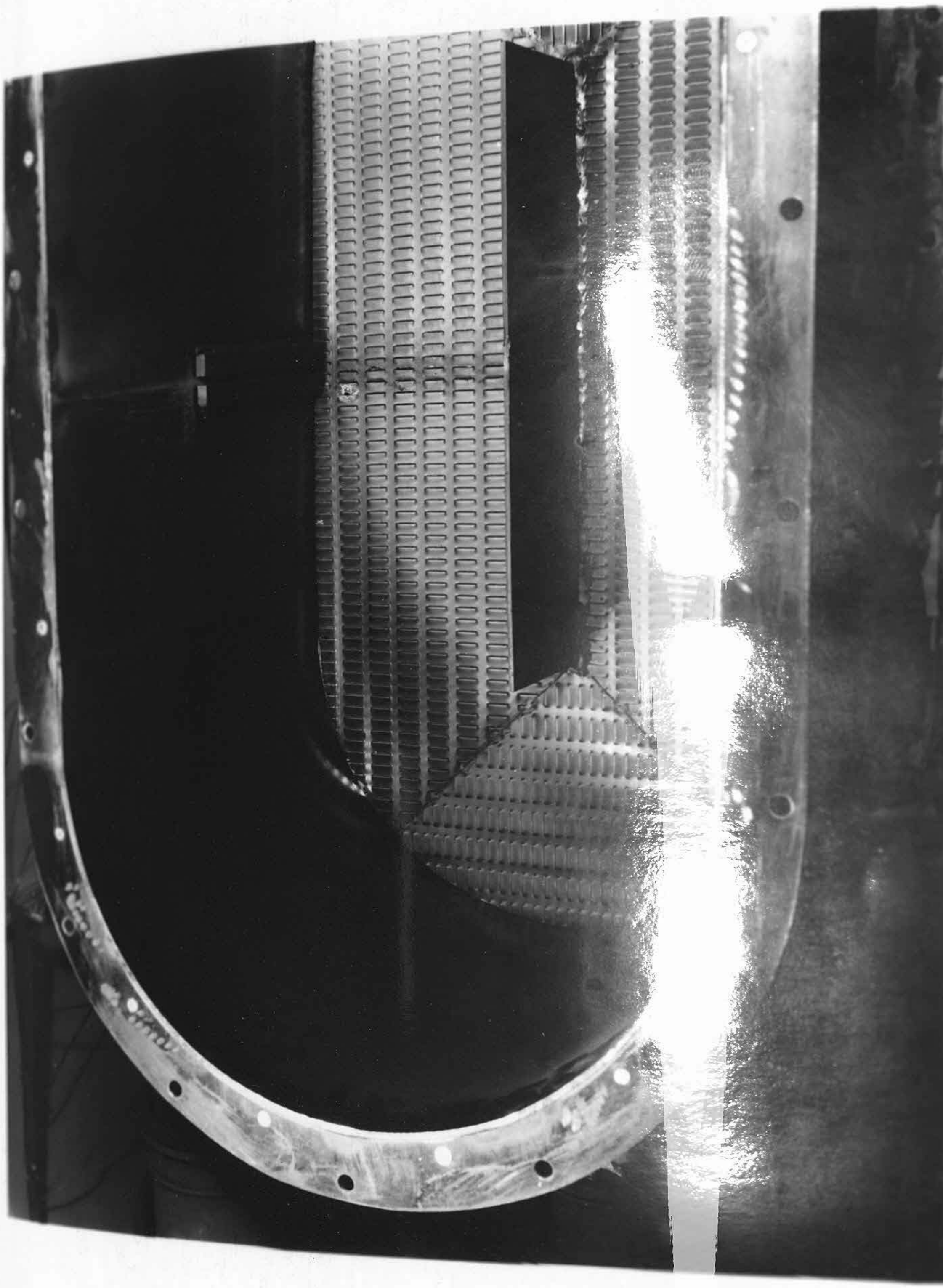
SCALE: 1/6 FULL SIZE

MATERIAL: 1 mm STEEL EXCEPT DISTRIBUTOR

1. DIRECTION OF DISTRIBUTOR
2. BUTT WELDS
3. REGION OF DISTRIBUTOR BLOCKED BY WELDING
4. VERTICAL SHEETS FORMING LABYRINTH SEAL

FIGURE 5.10 DISTRIBUTOR PLATE INSIDE HEAT EXCHANGER





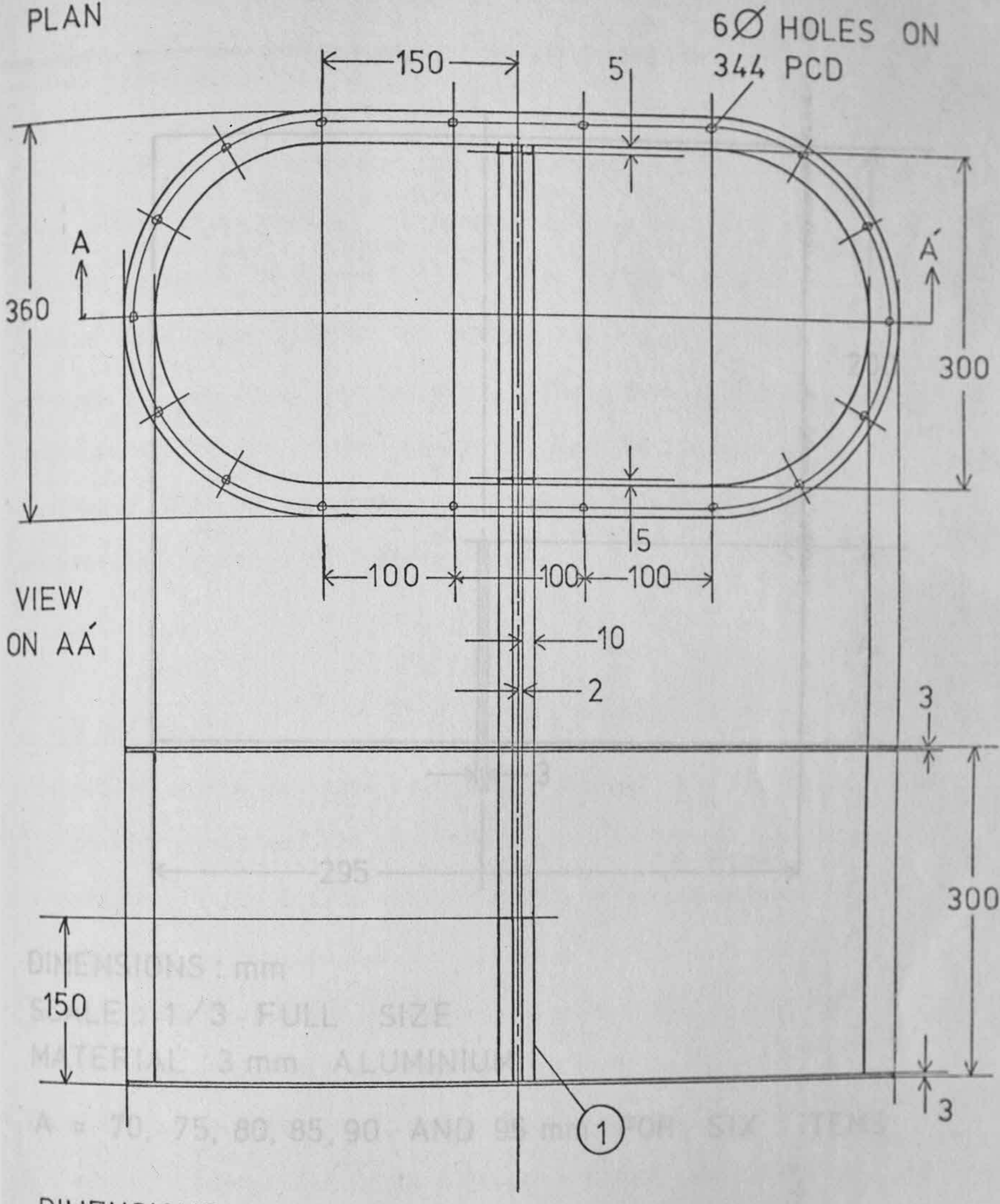
well in practice with no measurable leakage across it. Since these sheets were welded to the underside of the distributor, air was prevented from passing through a strip of the distributor about 8 mm wide. It will be seen later that this small area of blocked off distributor may affect some aspects of the heat exchanger's performance significantly.

5.4.3 Bed containment

The containment section of the heat exchanger forms the outside wall of the bed and defines the freeboard of the unit as shown in Figure 5.11. It also holds the partition in place and supports the thermocouples and the gate valves in the air outlet flows. The lower flange is sealed on to the top of the distributor with an asbestos gasket. The complete plenum to containment seal is made airtight by filling the edge with jointing compound which hardens under operating temperatures and has proven to be an excellent seal.

The partition above the bed separates the two air outlet flows. It is directly above the plenum divider and is positioned by the slides on the inside of the containment. A series of interchangeable partitions was made as shown in Figure 5.12. The six different items with the various values of dimension A correspond to a series of gaps between the bottom of the partition and the distributor of 5, 10, 15, 20, 25 and 30 mm. The size of this gap was more reproducible with six different

FIGURE 5.11 BED CONTAINMENT DRAWINGS



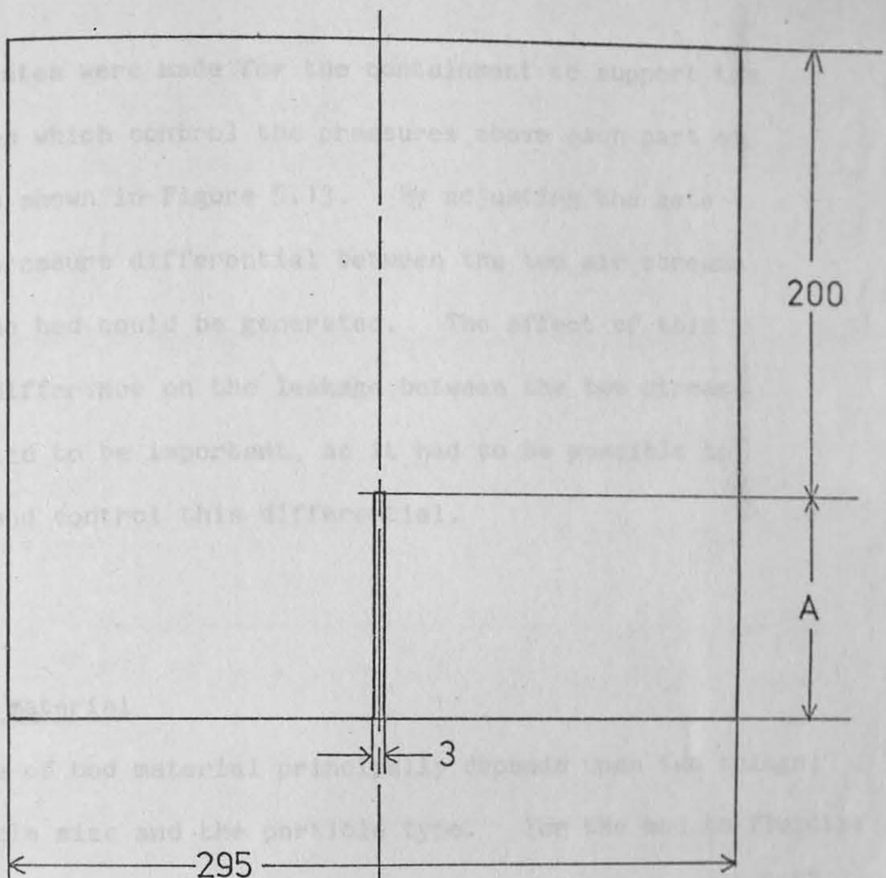
DIMENSIONS : mm

SCALE : 1 / 6 FULL SIZE

MATERIAL : 2 mm STEEL UNLESS SPECIFIED

1. SLIDES FOR PARTITION

FIGURE 5.12 PARTITION DRAWINGS



DIMENSIONS : mm

SCALE : 1/3 FULL SIZE

MATERIAL : 3 mm ALUMINIUM

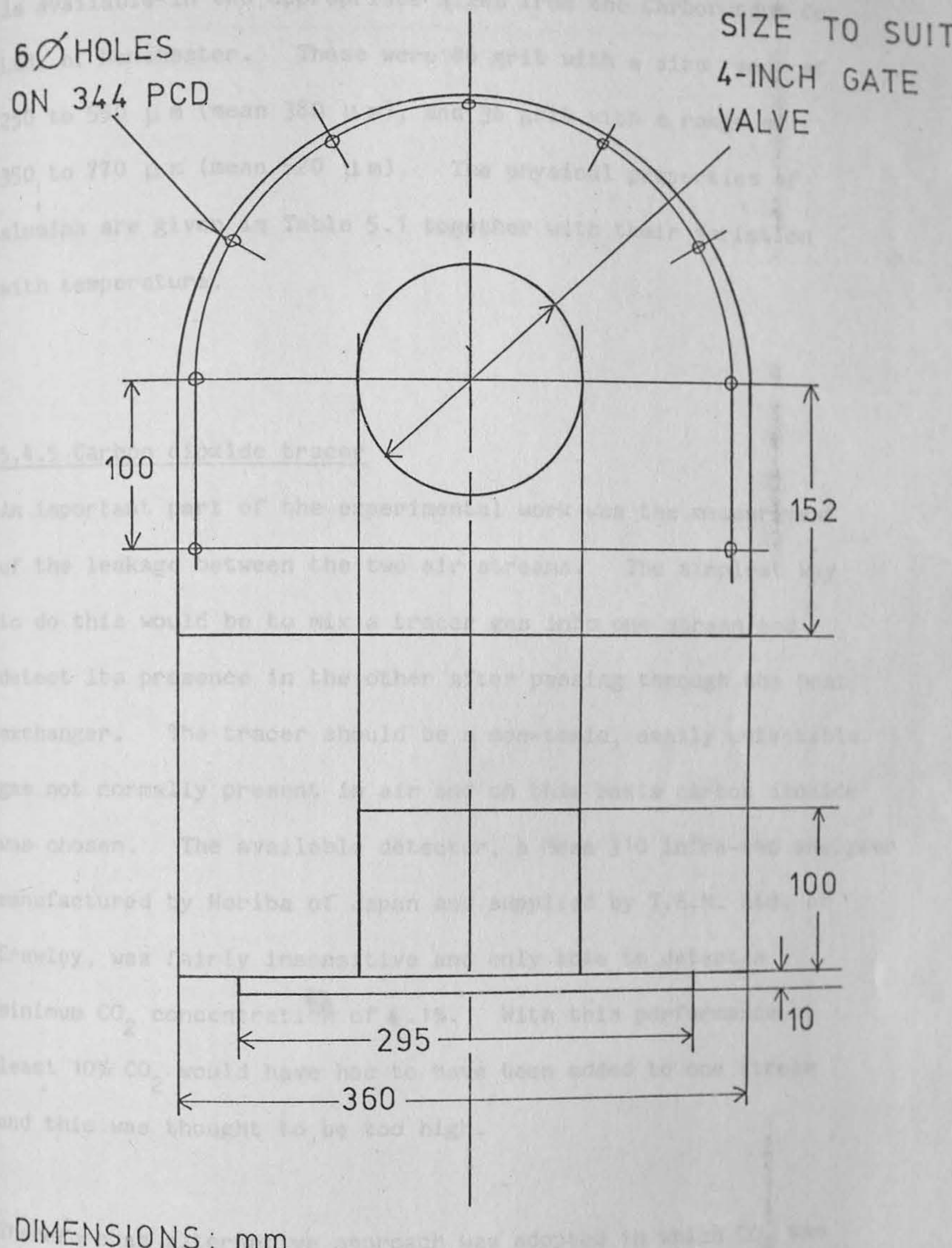
A = 70, 75, 80, 85, 90 AND 95 mm FOR SIX ITEMS partitions than if one movable partition had been adjusted each time. The slot in the centre of the partition is a push fit over the central wall on the distributor.

Two top plates were made for the containment to support the gate valves which control the pressures above each part of the bed as shown in Figure 5.13. By adjusting the gate valves a pressure differential between the two air streams leaving the bed could be generated. The effect of this pressure difference on the leakage between the two streams was expected to be important, so it had to be possible to generate and control this differential.

5.4.4 Bed material

The choice of bed material principally depends upon two things; the particle size and the particle type. For the bed to fluidise satisfactorily the solids must fall into Geldart's Group B [16] as described in Section 2.2.3. It will be recalled that the characteristics of Group B require that the particle mean size should be 40 to 500 μm and the density 1400 to 4000 kg m^{-3}. These limits are not exact, but at least they indicate some of the required properties. In addition the particles need to be resistant to corrosion by hot exhaust gases, to attrition generally and in particular to that induced by thermal cycling. For use in a commercial product the material must be readily available and as cheap as possible. With reference to these requirements, the discussion of Section 3.6 concluded that

FIGURE 5.13 TOP PLATE FOR BED CONTAINMENT



DIMENSIONS: mm
SCALE: 1/4 FULL SIZE
MATERIAL: 1 mm STEEL

alumina was the most suitable bed material. Fused alumina is available in two appropriate sizes from the Carborundum Co. Ltd. of Manchester. These were 46 grit with a size range of 250 to 550 μm (mean 380 μm), and 36 grit with a range of 350 to 770 μm (mean 520 μm). The physical properties of alumina are given in Table 5.1 together with their variation with temperature.

5.4.5 Carbon dioxide tracer

An important part of the experimental work was the measurement of the leakage between the two air streams. The simplest way to do this would be to mix a tracer gas into one stream and detect its presence in the other after passing through the heat exchanger. The tracer should be a non-toxic, easily detectable gas not normally present in air and on this basis carbon dioxide was chosen. The available detector, a Mexa 310 infra-red analyser manufactured by Horiba of Japan and supplied by T.E.M. Ltd. of Crawley, was fairly insensitive and only able to detect a minimum CO_2 concentration of 0.1%. With this performance at least 10% CO_2 would have had to have been added to one stream and this was thought to be too high.

Therefore an alternative approach was adopted in which CO_2 was injected at a single point close to the boundary between the hot and cold air streams and detected on the other side of the boundary. The experimental arrangement is shown in detail in Figure 5.14. The injectors were made from 12 mm copper tube

Table 5.1 Physical properties of alumina

Density 3970 kg m^{-3}

Temperature (K)	Specific heat [99] ($J\ kg^{-1}\ K^{-1}$)	Thermal conductivity [96] ($W\ m^{-1}\ K^{-1}$)
300	777	36.0
400	959	26.4
500	1050	20.2
600	1105	15.8
700	1143	12.6
800	1172	10.4
900	1196	8.9
1000	1217	7.9
1100	1236	7.1
1200	1253	6.6
1300	1269	6.1

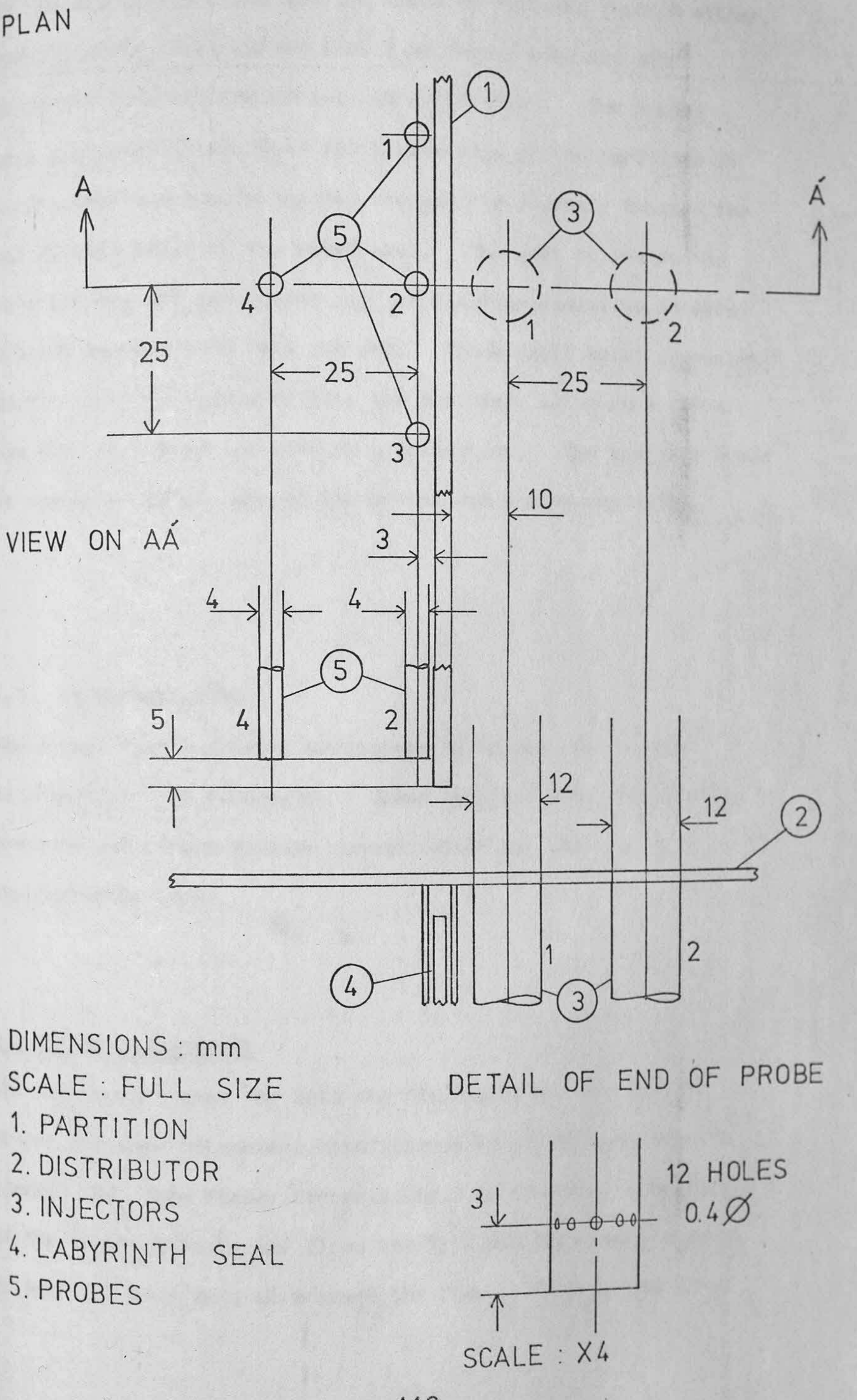


FIGURE 5.14 CARBON DIOXIDE INJECTION AND DETECTION

and the ends were sealed with jointing compound to the underside of the distributor, and the CO_2 could be switched through either. The CO_2 probes were formed from 4 mm copper tube and were suspended from the top of the bed containment. The probes were positioned 5 mm above the bottom edge of the partition to ensure that the sample gas had crossed the boundary between the air streams prior to its withdrawal. The ends of the probes were blocked off and twelve 400 μm diameter radial holes were drilled in each 3 mm from the end. These small holes prevented particles being sucked up into the analyser, although a filter was also placed in the line as a precaution. The analyser could be connected to any one of the probes via a five-way valve.

5.5 INSTRUMENTATION

There were four different parameters to be measured on the experimental heat exchanger. These were air flow, temperature, pressure and carbon dioxide concentration and these will be considered in turn.

5.5.1 Flow measurement

The flow meters used for both the fluidising air and the CO_2 tracer gas were Rotameters manufactured by GEC-Marconi Process Control Ltd. (now Fisher Controls Ltd.) of Croydon. For each of the two fluidising air flows two Type 65K Rotameters mounted in parallel were used to measure the flow. Each of these was calibrated in turn against a corner-tapped orifice made to BS 1042 [100] by Honeywell Controls Ltd. It was found that none of the four Rotameters differed by more than 4% from the manufacturer's curve. The standard curve for a 65K Rotameter for air is shown in Figure 5.15, although the individual calibrations were used to calculate the flows. The air temperature and pressure at the inlet to the Rotameters were measured and the reading corrected.

The leakage measurements required that the CO_2 velocity at the exit of the injector was the same as the fluidising velocity. Thus the CO_2 flow had to be metered and a 7K Rotameter was used for this. The Rotameter had to be re-calibrated to measure CO_2 rather than air and this was done using corrections recommended by the manufacturer. The CO_2 calibration of the 7K Rotameter is given in Figure 5.16.

5.5.2 Temperature measurement

All the temperature measurements were made using chromel/alumel junction thermocouples supplied by BICC Ltd. The thermo-electric e.m.f. was measured relative to an ice/water bath at 0 ^{o}C. using a Solartron A200 digital voltmeter which was checked against a standard Weston cell using a potentiometric bridge. The thermocouple calibration tables were taken from Kaye and Laby [101] and are shown graphically in Figure 5.17.

Nine thermocouples were used to measure the temperatures around the system. The air temperatures were measured at the inlet

FIGURE 5.15 STANDARD CALIBRATION OF 65K ROTAMETER

from manufacturer's data sheet RP. 2322-A

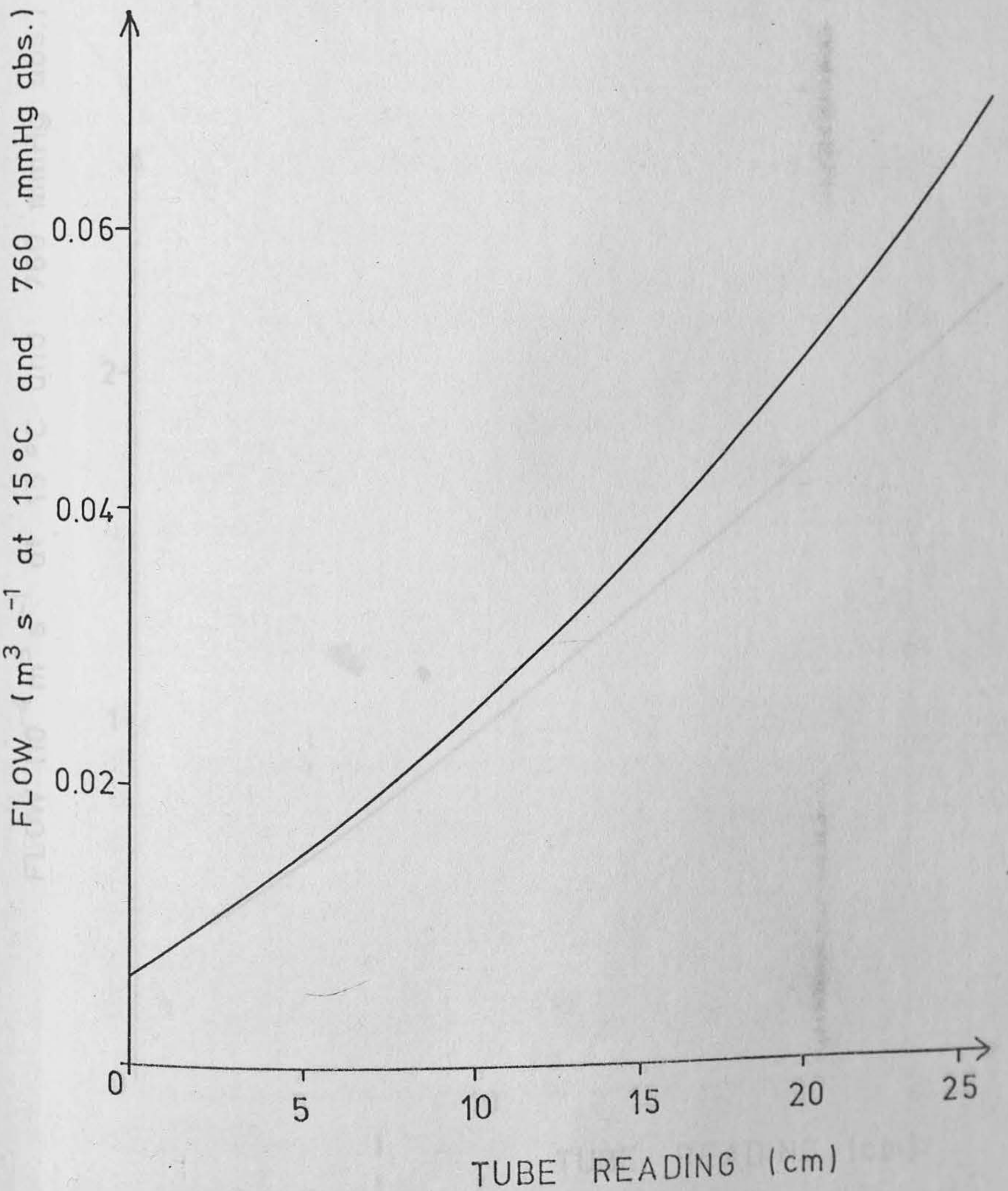


FIGURE 5.16 CALIBRATION OF 7K ROTAMETER FOR CARBON DIOXIDE

from manufacturer's data sheets RP. 2307-A and RP. 2552

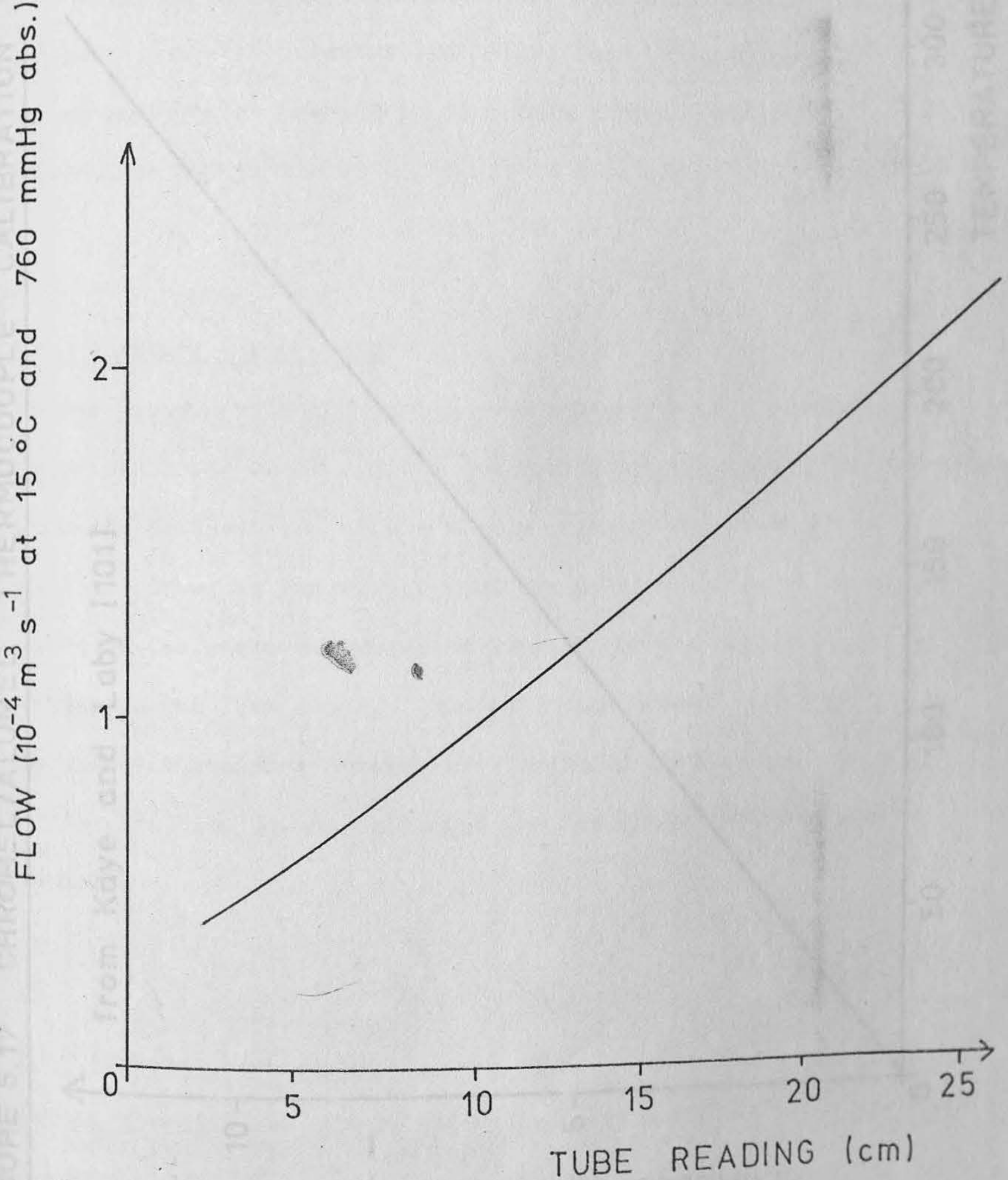
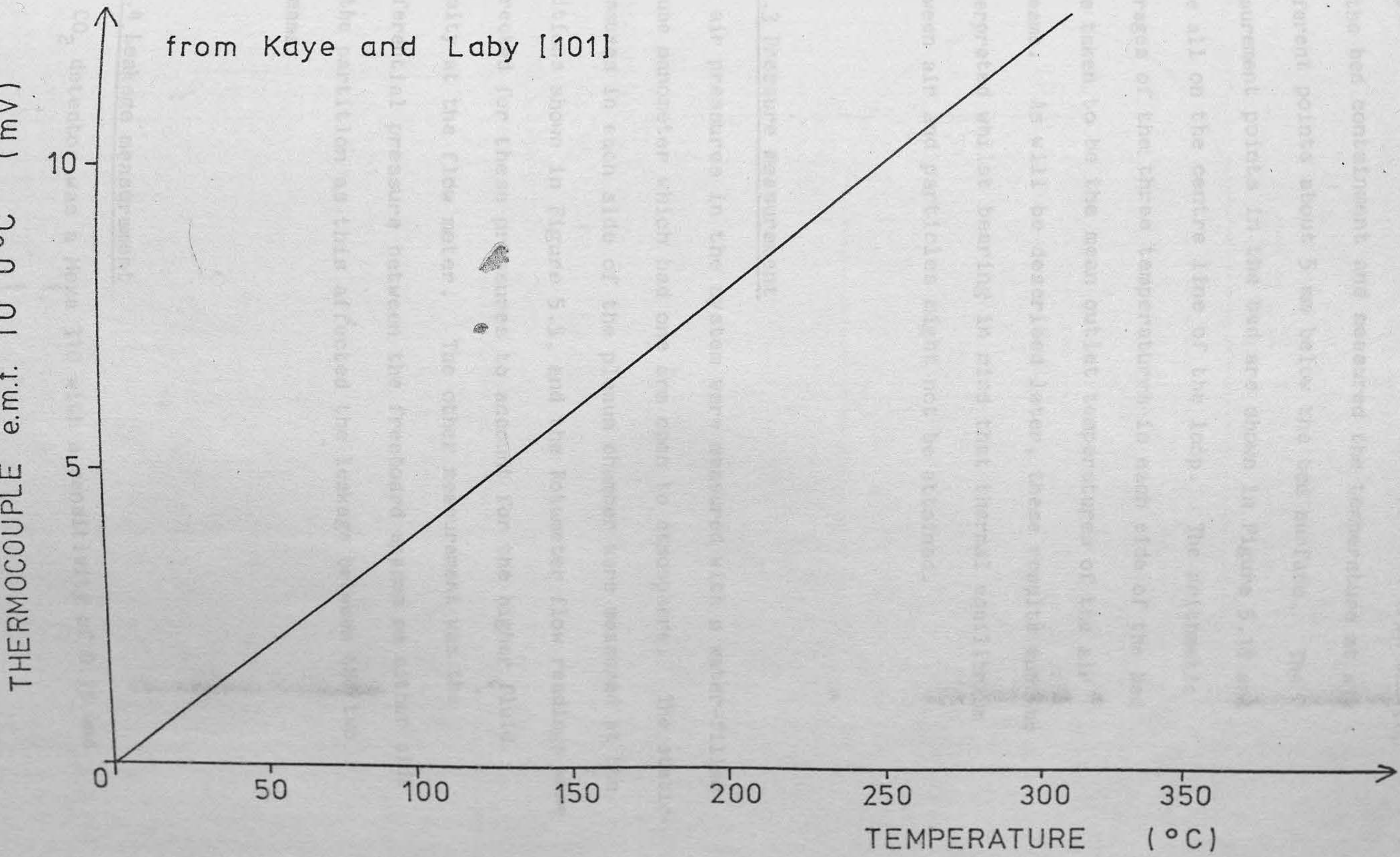


FIGURE 5.17 CHROMEL/ALUMEL THERMOCOUPLE CALIBRATION



manifold to the 65 K Rotameters and in each side of the plenum chamber, these latter being the hot and cold air inlet temperatures. Six thermocouples were supported from the top of the bed containment and measured the temperature at six different points about 5 mm below the bed surface. The measurement points in the bed are shown in Figure 5.18 and were all on the centre line of the loop. The arithmetic averages of the three temperatures in each side of the bed were taken to be the mean outlet temperatures of the air streams. As will be described later, these results must be interpreted whilst bearing in mind that thermal equilibrium between air and particles might not be attained.

5.5.3 Pressure measurement

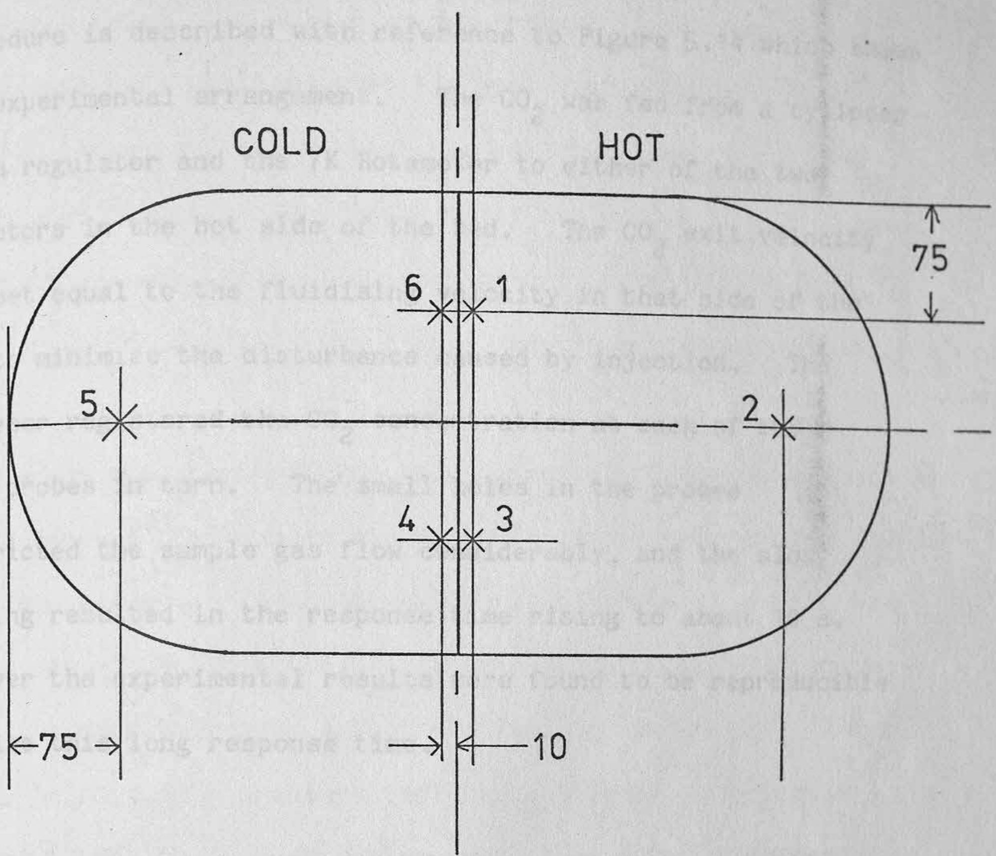
The air pressures in the system were measured with a water-filled U-tube manometer which had one arm open to atmosphere. The static pressures in each side of the plenum chamber were measured at the positions shown in Figure 5.5, and the Rotameter flow readings were corrected for these pressures to account for the higher fluid density at the flow meter. The other measurement was the differential pressure between the freeboard spaces on either side of the partition as this affected the leakage between the two streams.

5.5.4 Leakage measurement

The CO_2 detector was a Mexa 310 with a sensitivity of 0.1% and

FIGURE 5.18 TEMPERATURE MEASUREMENT POSITIONS

PLAN VIEW OF BED



DIMENSIONS : mm
SCALE : 1/5 FULL SIZE
THERMOCOUPLES : X
SOLIDS' FLOW : CLOCKWISE

a full scale deflection of 15%. Under some conditions the meter showed concentrations in excess of this and the scale was extrapolated approximately to 25%. The measurement procedure is described with reference to Figure 5.14 which shows the experimental arrangement. The CO_2 was fed from a cylinder via a regulator and the 7K Rotameter to either of the two injectors in the hot side of the bed. The CO_2 exit velocity was set equal to the fluidising velocity in that side of the bed to minimise the disturbance caused by injection. The analyser registered the CO_2 concentration at each of the four probes in turn. The small holes in the probes restricted the sample gas flow considerably, and the slow purging resulted in the response time rising to about 10 s. However the experimental results were found to be reproducible despite this long response time.

5.6 EXPERIMENTAL PROCEDURE

Having chosen the particle size, bed depth and the gap between the partition and the distributor, the following procedure was adopted in the experimental work on the heat exchanger. The hot side mass flow was selected along with the maximum cold side flow and these were set on the Rotameters. The air heater was then switched on and the heat exchanger approached a steady-state. It took about 1000 s for a steady-state to be attained and the temperatures in the system then oscillated slowly by about 0.5 K.

To plot a performance curve, the hot mass flow was maintained constant and the cold mass flow reduced in small steps. At each point all the thermocouple voltages and air pressures were noted before the cold flow was reduced to the next value. The changes were small so that the heat exchanger never drifted far from the steady-state which was re-established within 100 s of the step change. The reduction of the cold flow ceased when the particles no longer circulated around the heat exchanger, so then a new hot mass flow was chosen and the procedure repeated.

The leakage measurements were made separately from the performance curves because the leakage study formed the second phase of the experimental work. A steady-state was established and CO_2 was fed to one of the injectors. At each value of the cold mass flow, the gate valves on top of the bed containment were closed slowly to develop a pressure difference between the air outlet streams. The CO_2 concentrations were then noted at each of the probes for different values of the differential pressure.

In the experimental work care always had to be taken to ensure that a steady-state had been reached. By keeping the hot mass flow constant to plot a performance characteristic, the power supplied and the hot air inlet temperature were held constant also. If the hot flow had been varied and the cold flow been fixed instead this would not have been the case. Interpretation of the data would then have been more difficult and the trends harder to uncover.

CHAPTER SIX

OPERATIONAL CHARACTERISTICS OF THE EXPERIMENTAL HEAT EXCHANGER

CHAPTER SIX

OPERATIONAL CHARACTERISTICS OF THE EXPERIMENTAL HEAT EXCHANGER

6.1 INTRODUCTION

This chapter describes the work carried out on the experimental heat exchanger, the design of which was detailed in the previous chapter. The aim of the experiments was to find how the heat exchanger performed, not to study the reasons for its behaviour. For instance the momentum transfer from the fluidising air to the particles which makes the bed circulate and the mechanisms involved were not considered. The experiments focused on two aspects of the operation of the unit, namely the effectiveness of heat exchange and the leakage from one air stream to the other. Before describing these the basic practical properties of the distributor plate and the particles will be discussed.

6.2 DISTRIBUTOR PLATE PRESSURE DROP CHARACTERISTICS

The distributor plate used in the heat exchanger was formed from sections of Greenings' sheet. The specification and nominal dimensions of the sheet are shown in Figure 5.8. The pressure drop across the bed, Δp_d, increases with the air velocity through it and the plot of this characteristic is shown in Figure 6.1.* The two curves are for each part of the distributor since the hot and cold sides of the heat exchanger can be regarded as distinct distributors though Δp_d was measured under ambient conditions on both sides.

It can be seen that the hot side pressure drop is some 15% greater than that of the cold side. This is presumably due to differences between the two parts of the distributor and there are two possible explanations. Firstly the butt welds in the plate (Figure 5.9) may block off different areas of each side of the distributor. Secondly there may be non-uniformity of the slot size in the plate, though the distributor was fabricated from pieces from a single sheet. Many distributors exhibit a power law relationship of the form $\Delta p_d \propto U_f^n$, and analysis of the curves in Figure 6.1 shows that both sides of the distributor have the same value of n, namely n = 2.0. Since in this sense the two sides are the same, the difference between them may be caused by the blockage of some slots rather than by real differences in the form of the slots themselves. No large blockages could be seen, however, by careful visual observation and the effect might result from the existence of small burrs on the edges of the slots since no special attention was given to their removal. Whatever the cause,

* The Figures in this chapter are to be found at the end of the chapter.

this disparity is unlikely to affect the performance of the heat exchanger significantly and serves only to highlight the differences in flow characteristics which can arise between supposedly identical distributors.

6.3 MINIMUM FLUIDISING VELOCITY OF THE PARTICLES

The minimum fluidising velocity, U_{mf}, of each size of alumina grit used in the heat exchanger was determined in a small separate bed 140 mm in diameter. This had a porous tile distributor and compressed air was used to produce fluidisation. The distributor plate characteristic was measured and the pressure drop across the bed, Δp_b, was found by subtraction from the total pressure drop. In these experiments the fluidising velocity was gradually reduced from well above U_{mf} to zero. The resulting graphs are shown in Figures 6.2 and 6.3. The beds were shallow, only 11 and 13 mm deep, so that the distributor plate pressure drop was always much larger than that of the bed. This ensured that the fluidisation was stable [102] and also that the bed depth was much less than the diameter of the bed. As Pillai has reported [6] there is a tendency in shallow beds for Δp_b to be less than the weight per unit area of the bed. There were indications in the current experiments also of this effect, although it was not studied as the intention was only to measure U_{mf}.

The determination of U_{mf} from the pressure drop data has been described earlier with reference to Figure 2.4 and this can be done quite accurately for Figures 6.2 and 6.3. The values of U_{mf} for 46 and 36 grit alumina are 0.22 and 0.32 $m\,s^{-1}$ respectively. Below these values of fluidising velocity the bed would not be fluidised so it is impossible for the heat exchanger to operate below these limits. It will be shown later that fluidising velocities significantly in excess of U_{mf} are needed to initiate the circulation of the particles which is essential to the operation of the heat exchanger.

6.4 THERMOCOUPLE MEASUREMENTS IN A FLUIDISED BED

The question of whether a thermocouple immersed in a fluidised bed measures particle or gas temperature has been raised already. There is no difficulty if the gas and particles are in thermal equilibrium as their temperatures are then the same. However the theoretical analysis outlined in Section 4.4 shows that thermal equilibrium will not have been attained at the top of the bed in the heat exchanger because the bed is so shallow. Singh and Ferron [39] used a radiation pyrometer to measure particle temperature, T_p, and an energy balance to deduce the gas temperature, T_g. They postulated that the bed temperature as measured by an immersed thermocouple, T_b, was given by the simple relation

$$T_b = \alpha T_g + (1-\alpha) T_p \tag{6.1}$$

where α is an empirical constant defined to fit this equation. They inferred that α was about 0.2 and therefore the immersed thermocouple measured a temperature equivalent to 80% of the particle temperature plus 20% of the gas temperature.

We can write

$$T_p = T_g + \delta \tag{6.2}$$

where δ is the temperature difference between the solid and the gas at the measurement point. The theoretical model provides an estimate of δ, so combining equations (6.1) and (6.2) gives

$$T_g = T_b - 0.8\,\delta \tag{6.3}$$

Thus we can account for the fact that the gas and particles are not in thermal equilibrium and can obtain ~~a~~ an estimate of the true gas temperature from the measured bed temperature.

6.5 RANGE OF VARIABLES COVERED IN THE EXPERIMENTS

The range of variables covered in the experiments is given in Table 6.1 along with the air inlet conditions. The upper limits of the fluidising velocities were fixed by the restricted flow rate available from the fan. However

Table 6.1 Range of variables

Variable	Range
Hot side fluidising velocity	0 to 1.3 $m s^{-1}$
Cold side fluidising velocity	0 to 1.0 $m s^{-1}$
Bed depth	15 to 50 mm
Partition gap	5 to 30 mm
Particle diameter	380 and 520 μm (46 and 36 grit)
Hot air inlet temperature	200 to 250 ^oC
Cold air inlet temperature	35 to 45 ^oC

above these velocities particle entrainment would have become very severe and operation fruitless. The covering patent [7] specifies that the bed depth should be between 12 and 75 mm, but early experiments quickly established that depths greater than 50 mm were of decreasing value. The partition gap between the lower edge of the partition and the distributor (Figure 3.2) was varied in 5 mm steps from 5 to 30 mm. This provided a complete range of flow phenomena when the condition of no partition being present was also included. The alumina particles were available in two sizes, 46 and 36 grit, from the Carborundum Co. Ltd. of Manchester. The hot gas inlet temperature was limited to 250^{o}C by the power output of the heater and the cold gas was raised slightly above room temperature by the action of the fan.

6.6 PARTICLE FLOW AROUND THE HEAT EXCHANGER

The operation of the heat exchanger requires that the particles circulate around the closed loop defined by the distributor plate. It is necessary to obtain some estimate of the circulation period so that an average value of the solids' flow velocity, U_p, can be entered into the theoretical model. Figure 6.4 is a photograph of the bed and its turbulent nature can be seen readily, although it is not clear that the bed is actually flowing in a clockwise direction when viewed from above. The measurement of the solids' velocity is therefore difficult as the bed is too turbulent for Botterill's turbine device [68] to survive. Yet the use of a float has been criticised by McGuigan [64] as being unreliable and unrepresentative of the solids' flow velocity. Timing the passage of a float is, however, the only simple method of determining U_p so it was used despite its drawbacks.

A 10 mm cube of perspex was used as the float and the circulation period was the average of ten complete circuits. The variation of the period with fluidising velocity is shown in Figure 6.5. These measurements were taken with no partition present, since if there were one, it would stop the float, and with the same fluidising velocity in each side of the bed. It can be seen that U_f has to be about 2 U_{mf} for circulation to start and that the shallower bed flows more easily. The onset of circulation is quite sudden and the maximum solids' flow velocity is achieved at a U_f of about 0.8 m s^{-1}. The minimum period of 5 s corresponds to a solids' velocity of 0.2 m s^{-1}. This maximum value of U_p is the same for both bed depths and cannot be increased with fluidising velocity. It is governed by the momentum transfer to the particles, and hence is a function of distributor plate design and the properties of the particles and fluidising gas.

There are two flow phenomena which have been seen in the solids' circulation around the heat exchanger. The first of these is the "hydraulic jump", shown in Figures 6.6 and

6.7, which is produced occasionally near to the partition. It is seen with beds less than about 30 mm deep and with small partition gaps. The particles move very rapidly just beyond the partition if the fluidising velocity is appropriate, and the hydraulic jump so produced is quite stable. The second observation is that stagnant, recirculating zones can be created in the bed in the lee of the central wall as shown in Figure 6.8. These zones form because the distributor only moves the particles in a straight line and cannot accelerate them around curves. So some of the particles are left behind after the sharp corners at the ends of the central wall. This reduces the heat exchange between the two air streams and so should be avoided if possible.

6.7 HEAT RECOVERY PERFORMANCE

The measure of the heat exchanger's performance is its effectiveness, η , which is defined in equation (4.15) and derived in Section 4.7. The experimental procedure which was used to obtain the performance curves is described in Section 5.6. Before presenting the experimental data, a typical curve will be described and its shape discussed.

Figure 6.9 shows the general shape of a performance curve, and it will be seen later that all the curves exhibit this general form. If the fluidising velocity is less than U_{mf} then the bed does not circulate, but circulation quickly improves with increasing fluidising velocity (Figure 6.5) and so the effectiveness rises sharply at A in Figure 6.9. The effectiveness approaches a maximum value at B because the circulation becomes the most rapid attainable. Finally the curve flattens out at C and no further increase in η can be effected. The curve levels at $\eta = 1.00$ so the unit operates as if it were similar to a perfect parallel flow heat exchanger.

The complete set of experimental data is shown in Figures 6.10 to 6.49. A specimen calculation showing how the effectiveness is derived from the initial measurements is given in Appendix 2. The range of the experimental variables is given in Table 6.1 and this was wide enough to allow a thorough investigation of the heat recovery performance of the heat exchanger. The various trends which can be seen in the large amount of data will be considered in turn following a general review of the data.

6.7.1 General review of the data

It is important to look first at the data qualitatively and comment upon their accuracy and consistency. The graphs show that η decreases steadily with decreasing fluidising velocity and this monotonic trend is only rarely disturbed as for example in Figure 6.11, $U_{fh} = 1.23\ m\,s^{-1}$ at $U_{fc} = 0.79\ m\,s^{-1}$. Since this is an isolated case it may be concluded that η was measured regularly to within

$\pm$ 0.01. The scatter of the points on the curves is less for smaller partition gaps. In particular the curves with no partition, such as Figure 6.30, tend to be quite confused. A possible explanation for this is that the partition stabilises the solids' flow by impeding it and so reduces the fluctuations in temperature which in turn decrease the accuracy of the measurement of effectiveness. The set of curves for the larger 520 μ m particles (Figures 6.36 to 6.49) is not complete like that of the smaller 380 μ m material. The intention was only to gather sufficient data to establish whether the patterns in the data for the smaller particles were also present when the larger grit was employed.

6.7.2 Effect of fluidising velocity

The performance characteristics shown in Figures 6.10 to 6.49 are each for a fixed bed depth and partition gap but for several different hot side fluidising velocities, U_{fh}. It can be seen that U_{fh} affects the η versus cold side fluidising velocity, U_{fc}, curves and all the results show the same trend to varying degrees. Figure 6.18 for a 25 mm deep bed of 46 grit and a partition gap of 15 mm is an especially good example of this.

The effectiveness falls more sharply for smaller values of U_{fh} because the circulation depends on both fluidising velocities. Circulation can be maintained at lower values of U_{fc} if U_{fh} is large. In fact with $U_{fh} = 1.27 \ ms^{-1}$ the solids continue to circulate with U_{fc} almost as low as U_{mf}. This effect is almost certainly dependent upon the size and geometry of the bed, because in large beds the flow of the solids on one side could be driven less readily by that on the other. Therefore as the bed increases in size the performance curves are expected to tend toward that for the lowest value of U_{fh}.

The second effect of fluidising velocity is that η reaches a slightly higher value if U_{fh} is small. This is because the particles and air will more nearly reach the same temperature at low fluidising velocities because the air residence time in the bed is then greater. More heat is then exchanged between air and particles and so η is increased if U_f is low. The results show that it only needs the fluidising velocity on one side of the heat exchanger (in this case U_{fh}) to change for this effect to manifest itself.

6.7.3 Effect of bed depth

The heat recovery performance of the heat exchanger is affected by the depth of the bed as shown in Figure 6.50. These characteristics are for a bed of 46 grit alumina, a partition gap of 15 mm and $U_{fh} = 1.03 \ ms^{-1}$. The curves are the appropriate ones selected from the complete set of data, although other similar groups show the same trend of with bed depth.

It can be seen that η is larger for shallower beds except in the case of the shallowest, 20 mm. Deep beds circulate less easily than shallow ones for given fluidising velocities and so η is reduced. In the case of the 20 mm bed η does not increase because the limiting factor is no longer particle circulation but the fact that the air residence time is too short for the heat to be transferred fully. At the lowest values of U_{fc}, however, η is greatest for the 20 mm bed because the deeper beds have ceased to circulate.

In Section 4.2 it was shown that the optimum gas flow pattern in the bed for heat transfer between gas and particles was plug flow. This pattern is approached more closely in shallow beds so η is reduced in deeper beds. Thus the data imply that shallow beds should be used in the heat exchanger, but depths less than 25 mm are not satisfactory.

6.7.4 Effect of partition gap

Figure 6.51 shows the influence of the partition gap on the effectiveness for a 40 mm deep bed of 46 grit and $U_{fh} = 1.03\ ms^{-1}$. As with the bed depth effects, the trend shown in Figure 6.51 is repeated in similar sets of data for both particle sizes and the curves are taken from the complete set of data. The curves show that η falls off at higher values of U_{fh} for small partition gaps. This is not a surprising result for the gap impedes circulation and the solids' flow cannot be sustained at low

U_{fc} with small gaps. With no partition present (that is effectively an infinite gap) the circulation of the 40 mm bed continues even only just above U_{mf}. At high values of U_{fc} the trend becomes confused and it is not clear if the partition gap significantly affects the maximum effectiveness attainable. It should not, because similar circulation rates are obtained for any partition gap over 15 mm provided the fluidising velocities are greater than 0.9 $m s^{-1}$.

It is important to consider the relative effects of bed depth and partition gap on the effectiveness. This can be seen in Figure 6.52 where the three curves are for cases in which the penetrations of the partition into the bed are the same. There is little difference between these curves above $U_{fc} = 0.5\ m s^{-1}$; below this the 30 mm deep/10 mm gap curve falls off as circulation ceases. Since the characteristics are similar, it may be inferred that the advantages of using a shallow bed for good circulation are balanced by the restriction of a small partition gap. Conversely a larger gap in a deeper bed is no better because of the inherently poor flow properties of the deeper bed. For the geometry and size of the experimental unit, it would appear that if the penetrations are the same, different combinations of depth and gap have similar heat recovery performances.

There is no reason to believe that this is due to something other than chance, so the conclusion must be applied with caution to future, larger units.

6.7.5 Effect of particle size

The minimum fluidising velocity of the larger 520 μm particles is clearly larger than that of the smaller 380 μm material. Hence higher fluidising velocities are needed to operate the heat exchanger with the larger particles, a result demonstrated by the comparison of appropriate pairs of characteristics, for example Figures 6.22 and 6.42. It is, however, more useful to plot η against multiples of the relevant minimum fluidising velocity for each of the particle sizes as shown in Figures 6.53 to 6.55. The curves for each of the two values of U_{fh} now align more closely in the three instances shown, but the agreement is not good. This suggests that U_{mf} is not a suitable characterisation of the particles for describing the solids' flow in the heat exchanger. This is not a surprising result considering the novel form of the distributor plate material which has not been studied before. Of equal practical importance is the unexpected finding that the effectiveness appears to increase more sharply with U_{fc} for the larger particles, though it is not clear if a higher maximum value can be attained.

To show that this effect is not confined by chance to just one particular combination of bed depth and partition gap, three different cases are shown in Figures 6.53 to 6.55. In all of these η rises more quickly for the larger particles and this implies that the circulation rate also increases more sharply with fluidising velocity for larger particles. It is not clear why this is so, but it is presumably a result of the complex interaction between the air and the particles which transfers the momentum to the solids from the air. A detailed study of this interaction was beyond the scope of the work reported here and the understanding of this phenomenon will have to await that study.

6.8 LEAKAGE BETWEEN THE AIR STREAMS

To determine the cross-contamination between the two air streams, carbon dioxide was used as a tracer gas as described in Section 5.4.5. The experimental data for the measurements of leakage are shown in Figures 6.56 to 6.63 in which the CO_2 concentration at each probe is plotted against Δp_{air}, the pressure difference between the two air streams above the bed. The experimental arrangement is shown in Figure 5.14, and all the graphs are for injector 1 and probes 1, 2 and 3. Under no conditions did the concentration at probe 4 exceed 1% which was much less than at any of the other three. Also injector 2 did not produce concentrations above 1% at any of the four probes, so these results are not plotted.

The major point to be made about the leakage graphs is that they are not very accurate. The concentrations fluctuated considerably over a period of about 15 s, particularly at the higher values. Broadly, the deviation was about half of the actual value, thus the measurement might be 10 $\pm$ 5%. That this fluctuation occurs is evidence of the confused nature of the bed at the boundary between the two air streams. The construction of the distributor plate (Figure 5.9) inevitably produces a de-fluidised zone because a small area of the distributor is blocked off as shown in Figure 6.64. It is possible that this zone is not completely stable and that it breaks down occasionally which leads to a sudden increase in the leakage. Further work will be necessary to establish the influence of the de-fluidised zone on the leakage, though the overall effect is likely to be to reduce leakage.

With these reservations in mind, the leakage results will be discussed. Probe 2 is that which is directly in line with the injector, so it is not surprising that this registers the highest concentrations. Probes 1 and 3 should be disposed symmetrically and so read the same, but probe 1 is consistently the higher. This implies asymmetry in the system which is probably due to the injector covering parts of two slots in the distributor rather than just one. Since the errors of these experiments are large, only general conclusions can be drawn about the effect of the bed parameters on the leakage. The maximum value of Δp_{air} which could be obtained was limited by the spray of particles from the partition gap which could be contained in the freeboard. The leakage tended to remain roughly constant until at some value of Δp_{air} it fell sharply. Thus the higher pressure above the cold side with respect to the hot could be employed successfully to inhibit leakage from hot to cold.

The critical value of Δp_{air} at which the leakage was reduced was related to the bed parameters in the following manner:-

(i) U_{fc} has little effect on the critical value;

(ii) Comparison of Figures 6.56, 6.59 and 6.63 suggests that the critical value is higher with a deeper bed for the same partition gap;

(iii) Figures 6.57, 6.59 and 6.61 imply that the critical value is smaller for a larger partition gap. This is contrary to the initial expectation that a smaller gap should reduce leakage. It may be because the de-fluidised zone is unstable with a small gap whereas a larger gap provides space for both the flowing solids and a stable de-fluidised region;

(iv) The effect of U_{fh} can be seen by comparing Figures 6.58 to 6.60 which indicates that the critical value is lower for smaller U_{fh}. The leakage comes from close to the partition so if the air velocity is reduced it has a smaller velocity pressure and so is more easily inhibited;

and

(v) Figure 6.62 shows that the leakage is not increased by reversing the pressure difference and pressurising the hot freeboard.

It is possible to estimate the overall leakage of the heat exchanger because the source of the contaminating gas is localised. Since negligible leakage was detected from injector 2, all the leakage gas comes from within 25 mm of the partition. The concentrations at the front three probes imply that almost 40% of the CO_2 reached the other air stream when Δp_{air} was small. This could be reduced to one fifth of this value if Δp_{air} were increased sufficiently. For the heat exchanger as a whole, the leakage could be decreased from 2% to 0.4% by the provision of a pressure difference between the outlet air streams. This result applies only to the size and geometry of this particular experimental unit, but for larger units the leakage should be less as the area producing the contaminating gas will be proportionately less. Also the idea of staggering the partition away from the plenum division (Figure 3.4) appears to be useful if the partition is moved about 25 mm downstream of the division.

During the leakage measurements the effectiveness of the heat exchanger was monitored to find out if it was affected adversely by the pressure difference being applied to reduce leakage. It was found that η was decreased only slightly by increasing

Δp_{air} from, for example, 0.99 to 0.95. Thus it would seem that the pressure differential does not impede the solids' flow significantly, so the heat exchanger continues to operate satisfactorily.

6.9 EXPERIMENTAL VERIFICATION OF THE THEORETICAL MODEL

There are two predictions of the theoretical model which can be verified using the data from the experimental unit. These are the steady-state temperature differences of the solids as they flow around the circuit, and the time required to reach that steady-state. The relevant theory is described in Sections 4.6.1 and 4.6.2, and the two predictions will be considered in turn.

The steady-state temperature distribution is affected most by the solids' flow velocity, U_p, as shown in Figures 4.6 to 4.8. As discussed in Section 6.6 it is very difficult to measure U_p in the heat exchanger so the comparison of experiment and theory is almost impossible. All that can be stated reliably is that the measured temperature differentials are about 4 to 6 K at maximum effectiveness. This is broadly in line with what would be expected if U_p were about 0.15 m s^{-1} which gives a circulation period of 6 s. This agrees with the data shown in Figure 6.5, so theory and experiment are roughly consistent.

It is easier to verify the theory in respect of the time required for the heat exchanger to attain a steady-state. This time is affected most by the depth of the bed, that is its thermal capacity, as shown in Figures 4.9 and 4.10. The heat-up experiment was carried out in a 30 mm deep bed of 380 μm alumina with a partition gap of 15 mm and the results are shown in Figure 6.65. The hot and cold air inlet temperatures were not constant during the experiment, so the accompanying theoretical curve is calculated point by point to account for the changing temperatures. This also meant that the fluidising velocities changed since the mass flows were constant, but the theoretical result assumes that U_p was unaffected by these changes. After the 900 s measuring period the experimental curve was rising faster than that for the theory, so they may reach the same final value. The agreement is fair, but if the thermal capacity of the bed were increased by 25% the curves would almost coincide. This increase is not unreasonable since the theory does not take into account the inevitable heating of the distributor and bed containment. Thus it may be concluded that the theory is realistic, but that it is not possible to test it rigorously because U_p is so difficult to determine reliably.

6.10 SUMMARY OF OVERALL PERFORMANCE

The work on the experimental unit shows conclusively that the heat exchanger functions such that its effectiveness approaches that of a perfect parallel flow unit. The effects of bed depth, partition gap and fluidising velocity can all be explained in terms of their influences on particle circulation. Thus any change which impedes the solids' flow reduces the effectiveness, so η decreases with increasing bed depth, decreasing partition gap and decreasing fluidising velocity. The leakage between the two streams is about 2% by volume and it all originates within about 25 mm of the boundary between the streams. The leakage from hot to cold can be reduced by the application of a pressure differential such that the freeboard above the cold bed is pressurised with respect to the hot.

In the next chapter it will be seen how these results affect the likely market for the heat exchanger by comparing the required performance with that of which the system is capable. This will reveal the probable success of the product in the market place and establish whether or not it is likely to be a successful innovation.

FIGURE 6.1 PRESSURE DROP CHARACTERISTICS OF THE DISTRIBUTOR

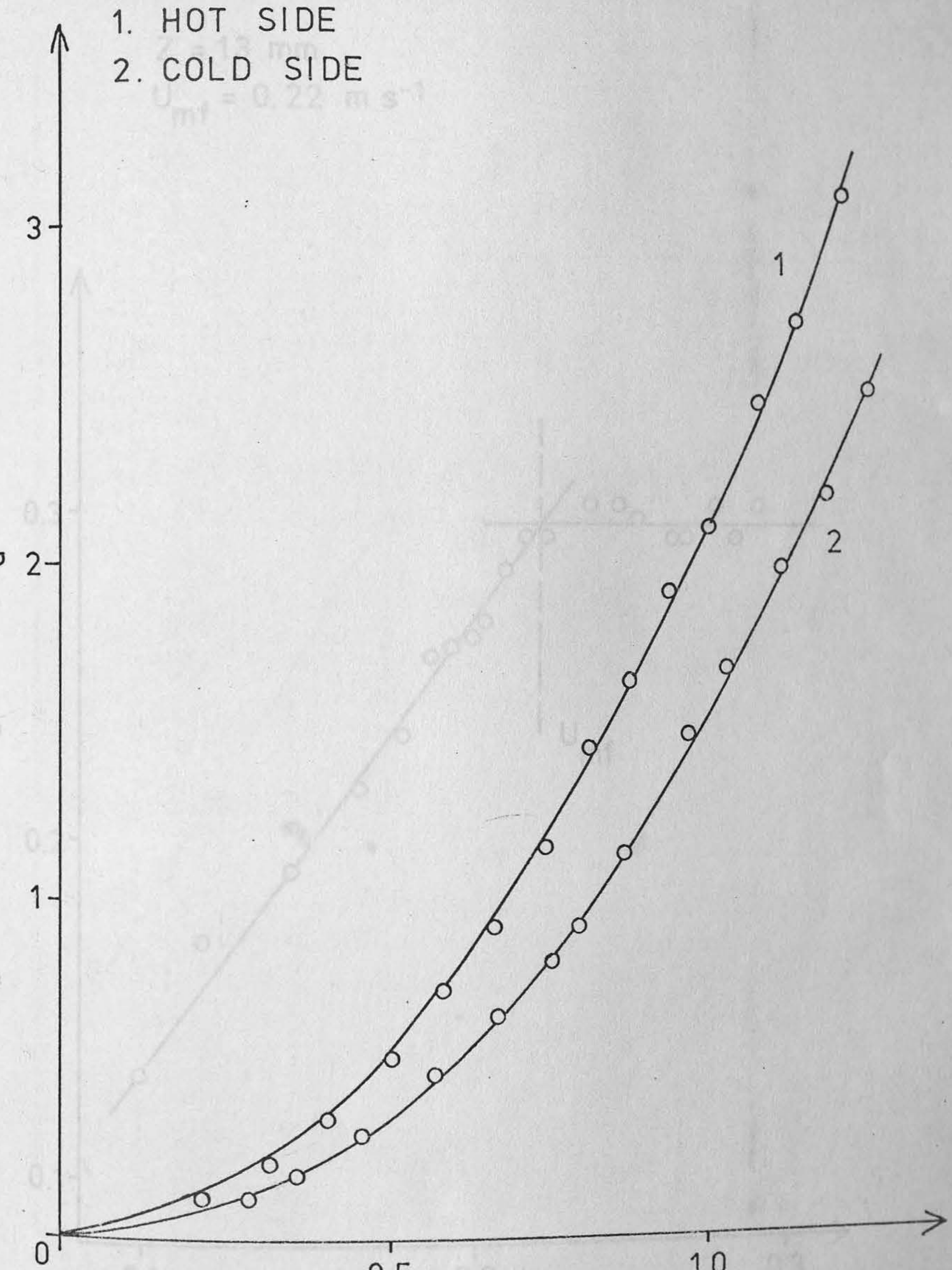


FIGURE 6.2 MINIMUM FLUIDISING VELOCITY OF 46 GRIT ALUMINA

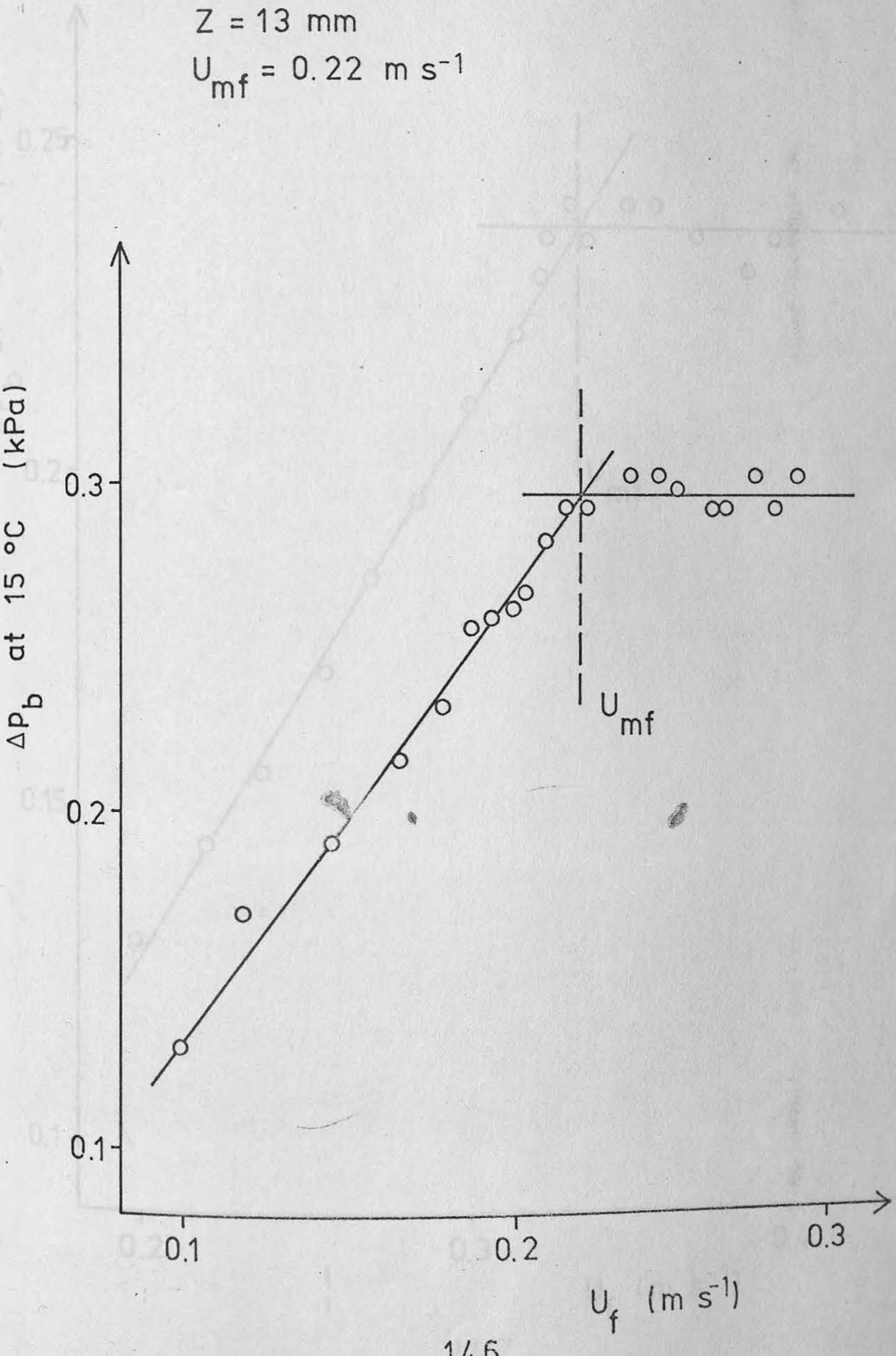


FIGURE 6.3 MINIMUM FLUIDISING VELOCITY OF 36 GRIT ALUMINA

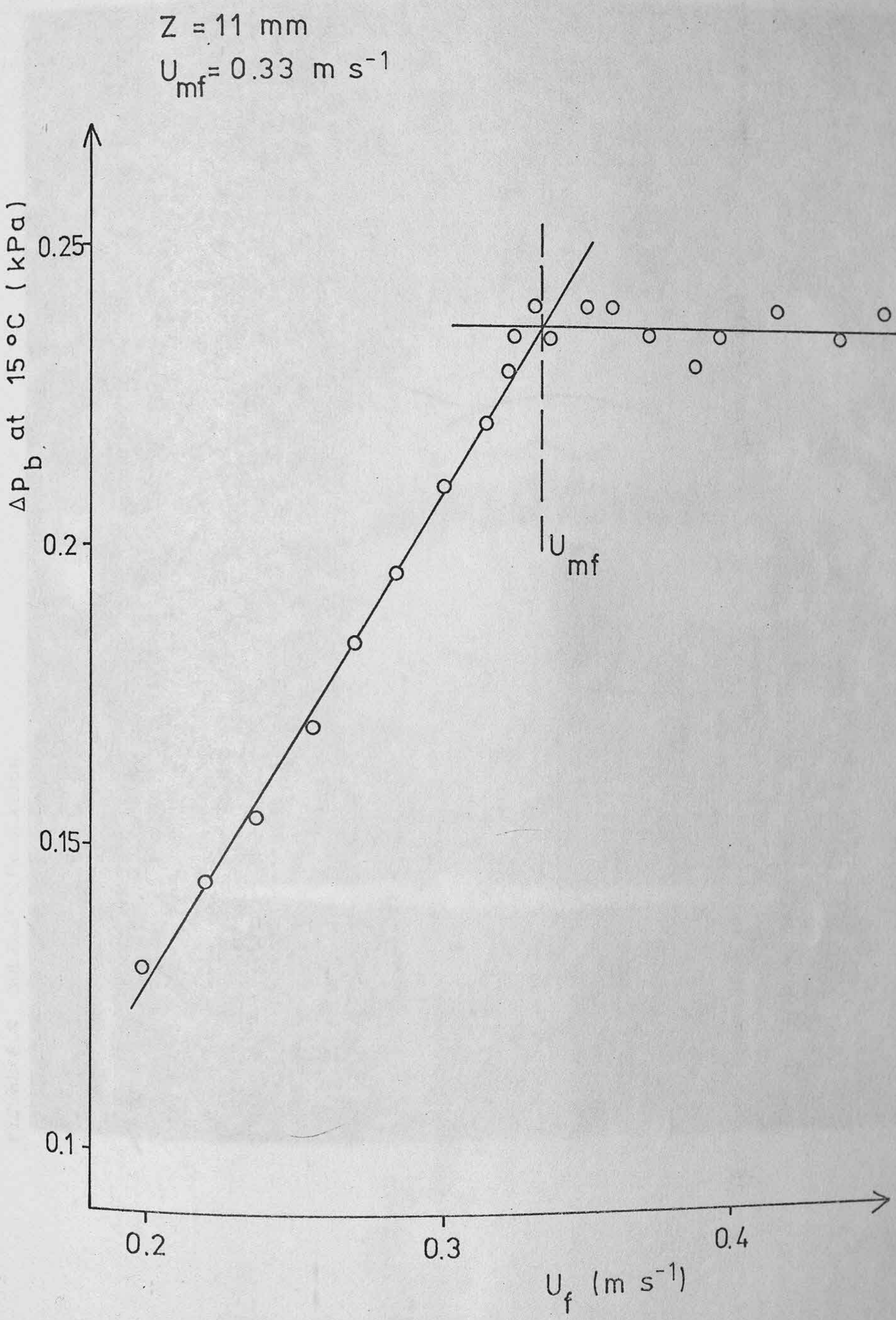
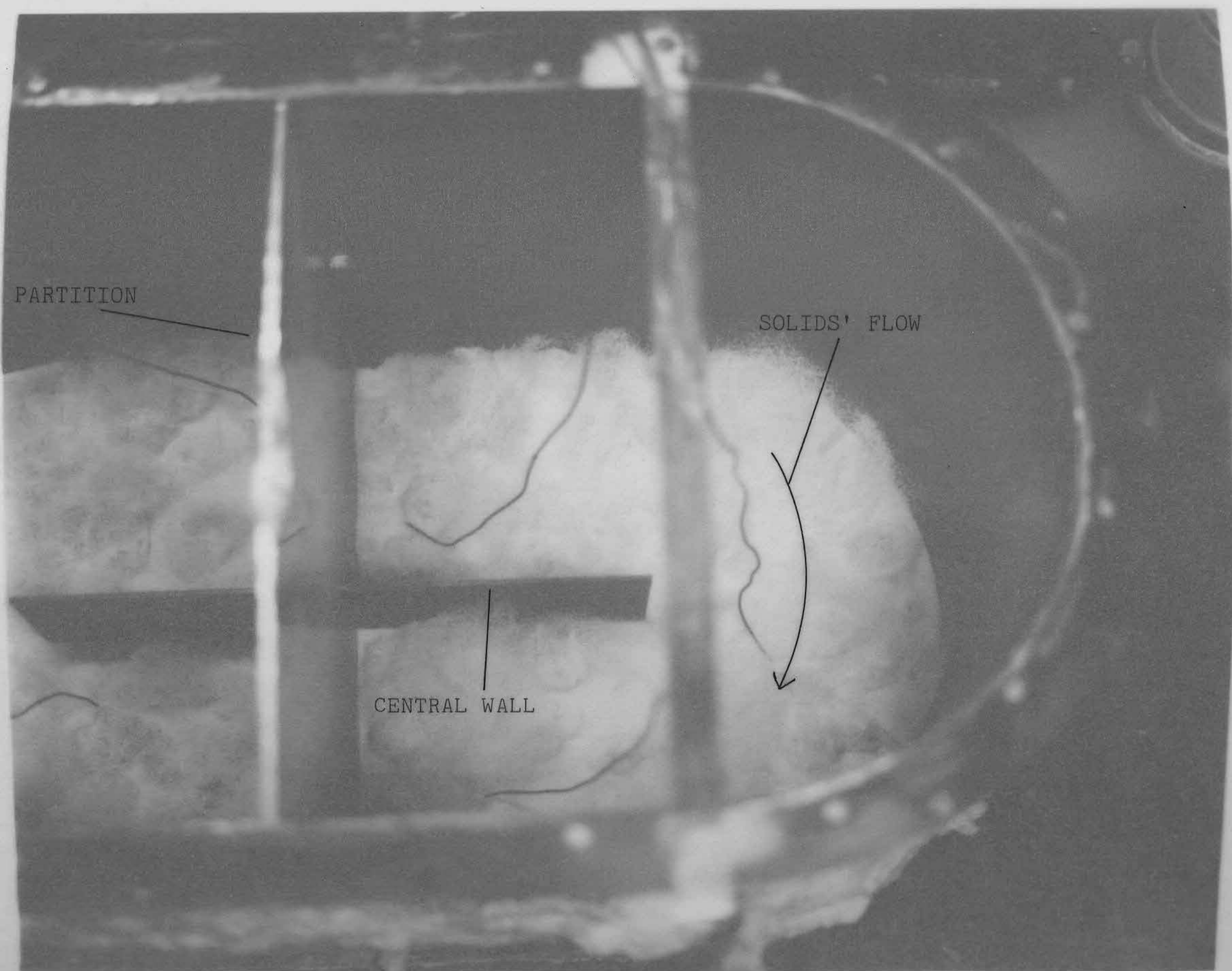


FIGURE 6.4 SOLIDS' CIRCULATION



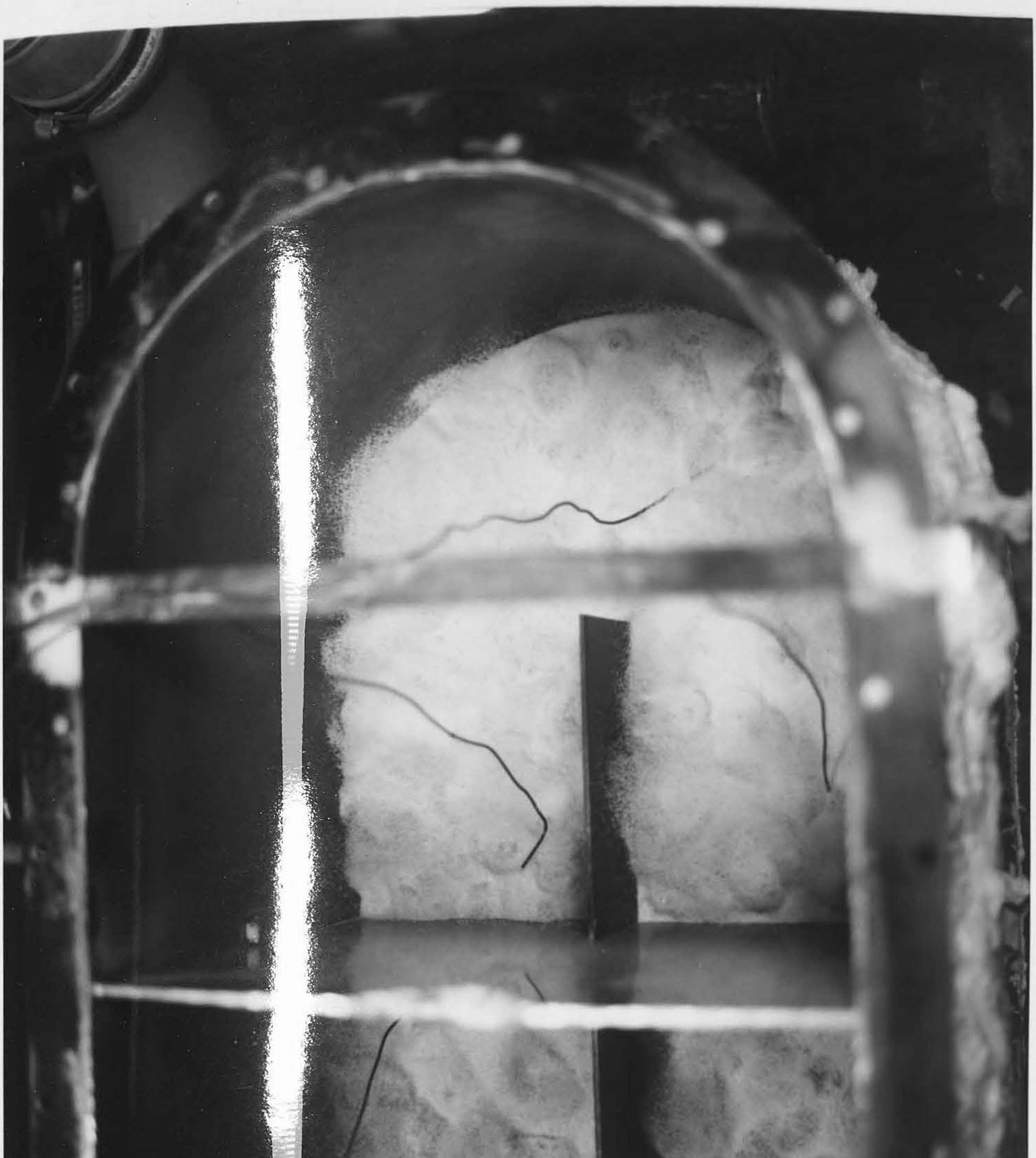


FIGURE 6.5 CIRCULATION PERIOD OF THE PARTICLES

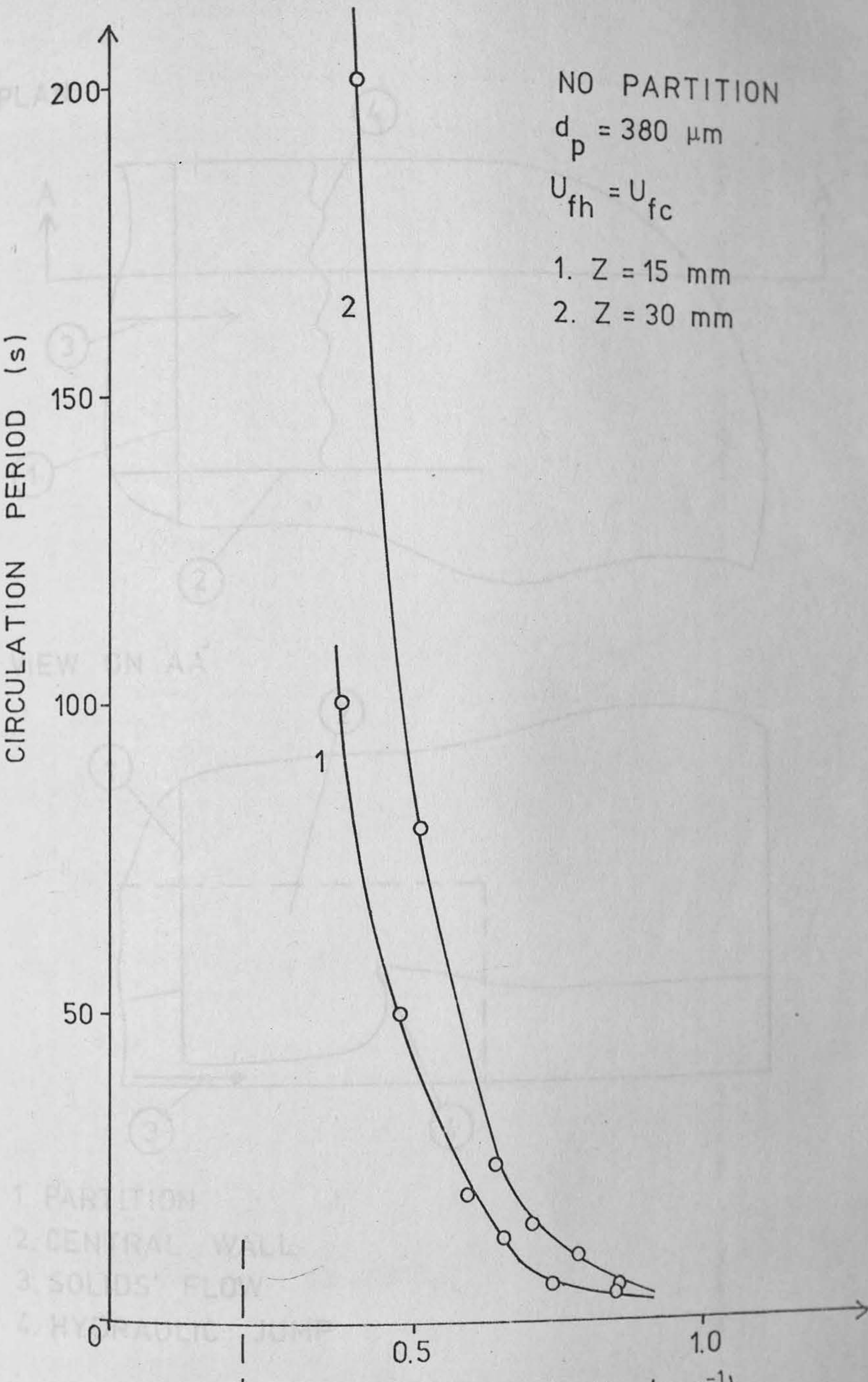
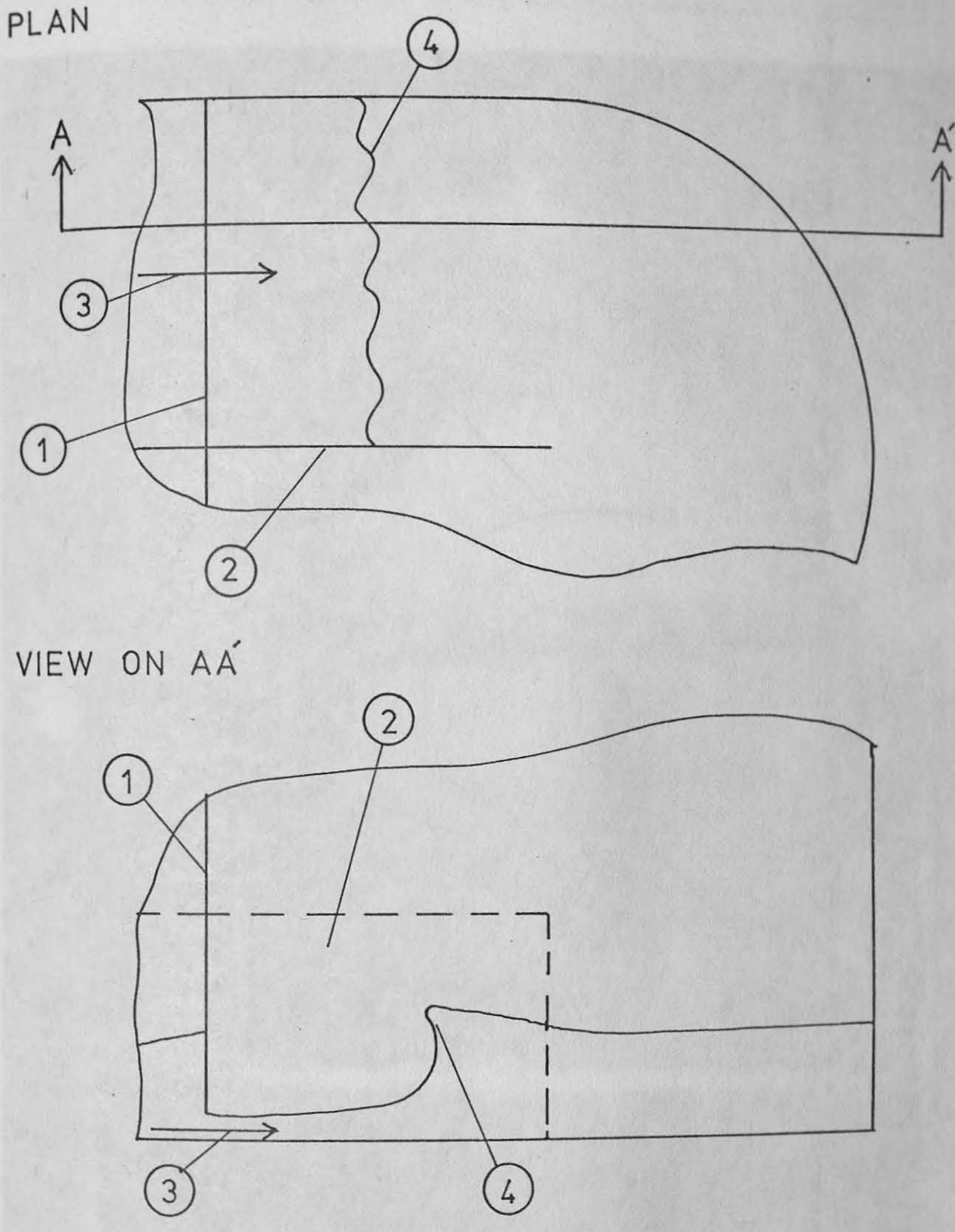
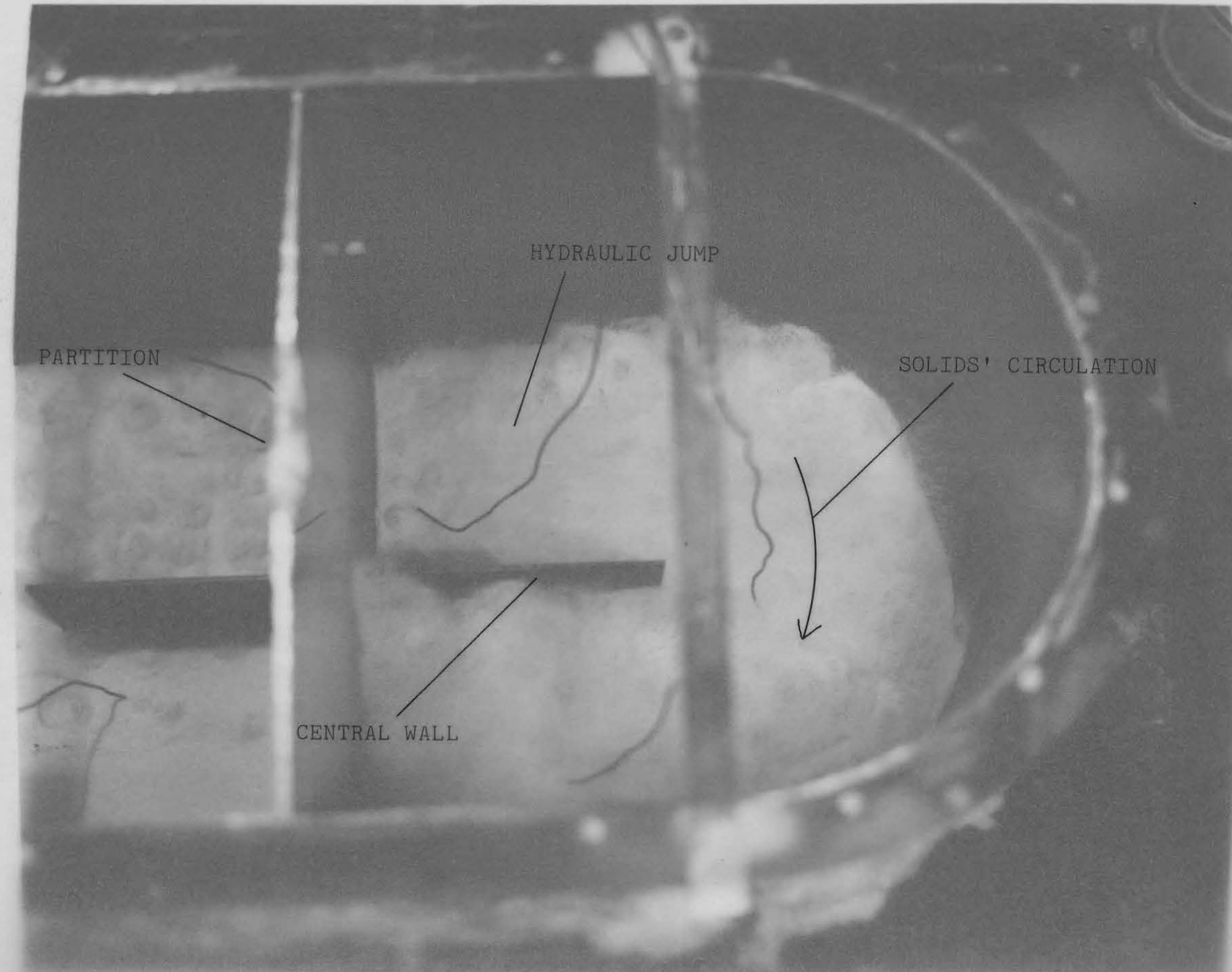


FIGURE 6.6 SKETCH OF HYDRAULIC JUMP



1. PARTITION
2. CENTRAL WALL
3. SOLIDS' FLOW
4. HYDRAULIC JUMP

FIGURE 6.7 PHOTOGRAPH OF HYDRAULIC JUMP



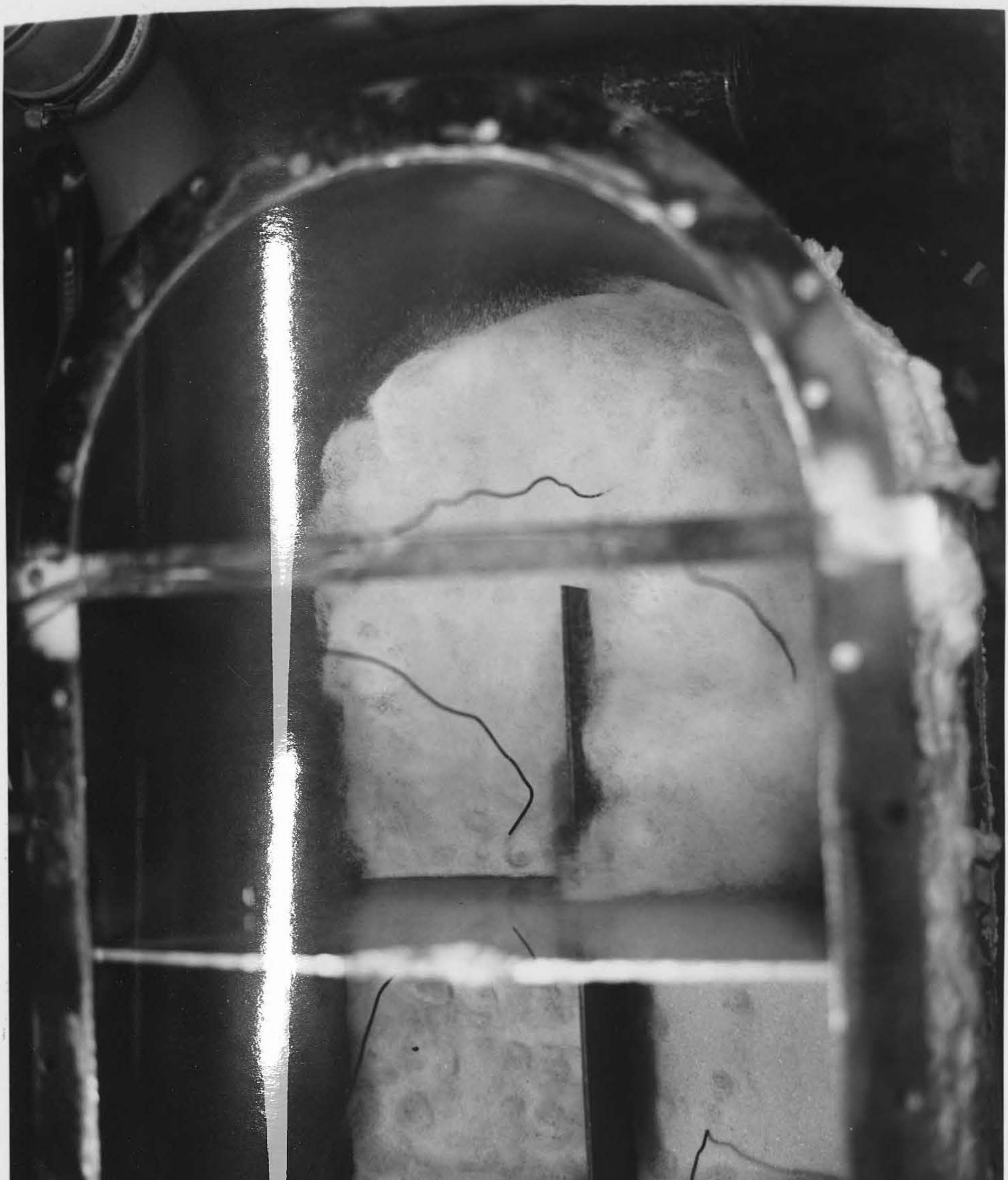
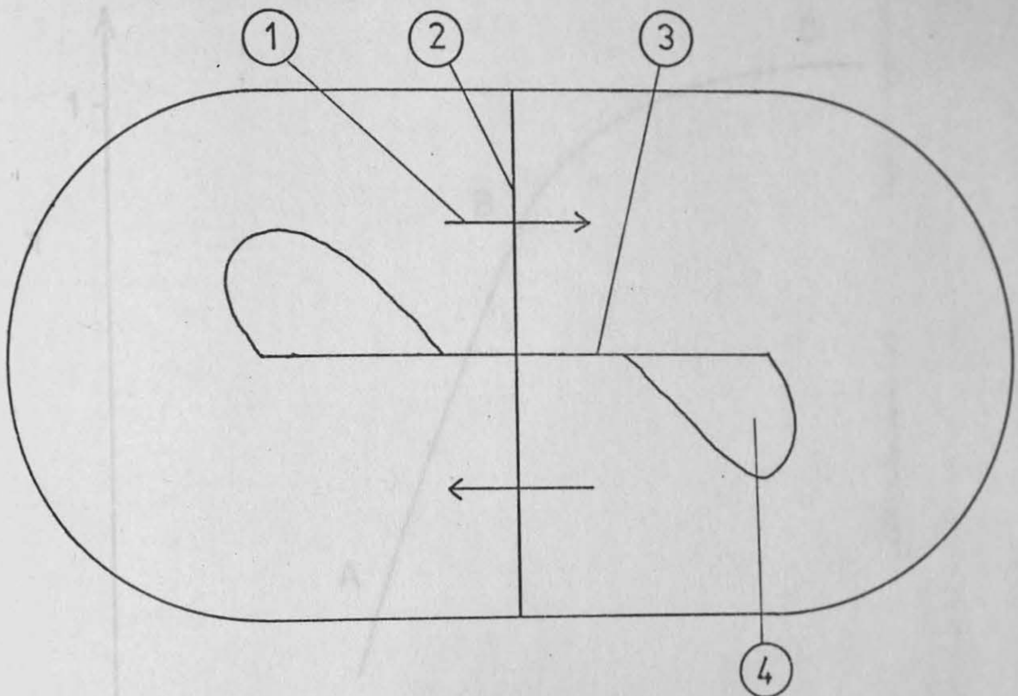


FIGURE 6.8 SKETCH OF STAGNANT ZONES

PLAN



1. SOLIDS' FLOW
2. PARTITION
3. CENTRAL WALL
4. STAGNANT ZONE

FIGURE 6.9 TYPICAL PERFORMANCE CHARACTERISTIC

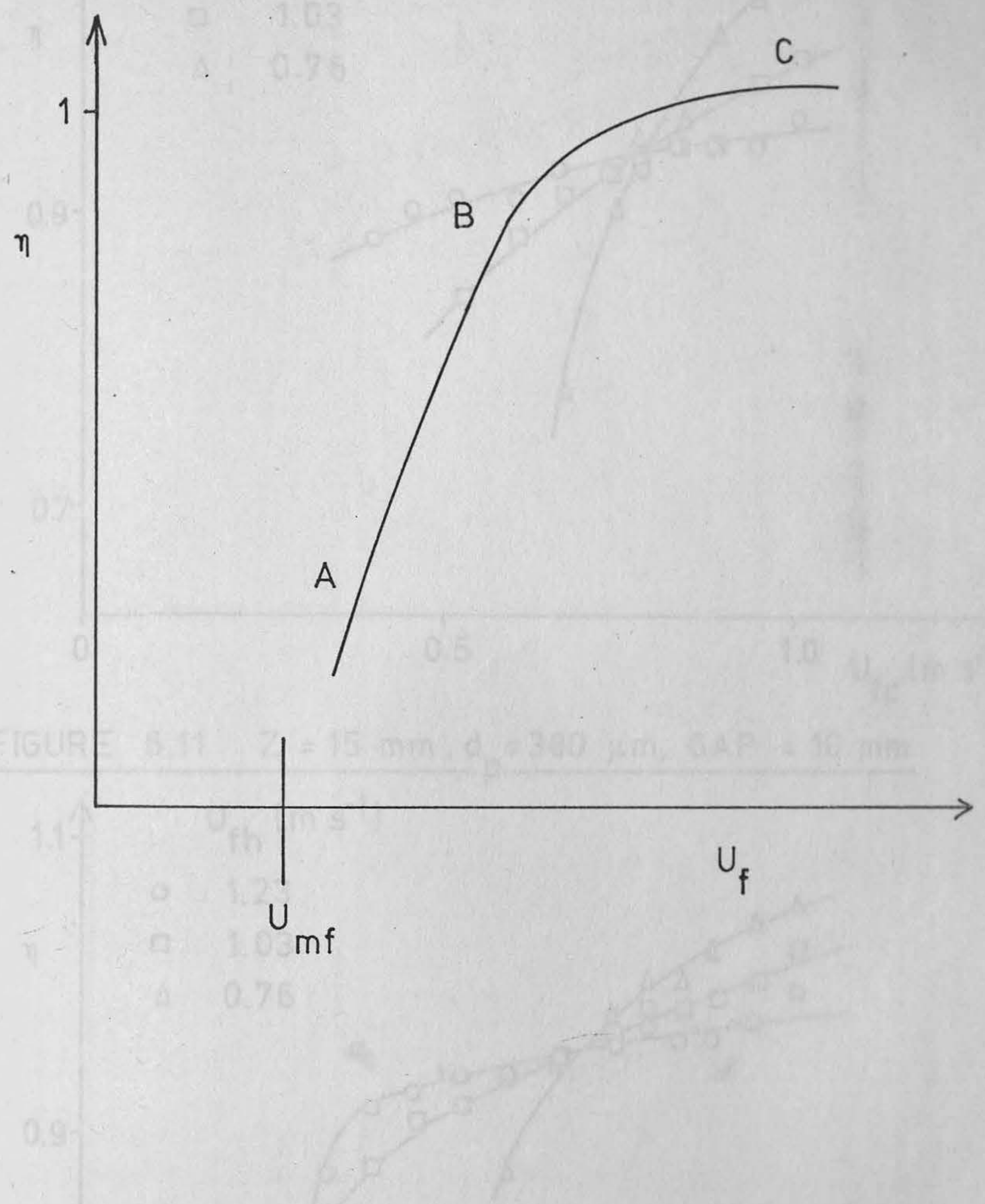


FIGURE 6.10 $Z = 15$ mm, $d_p = 380$ µm, GAP = 5 mm

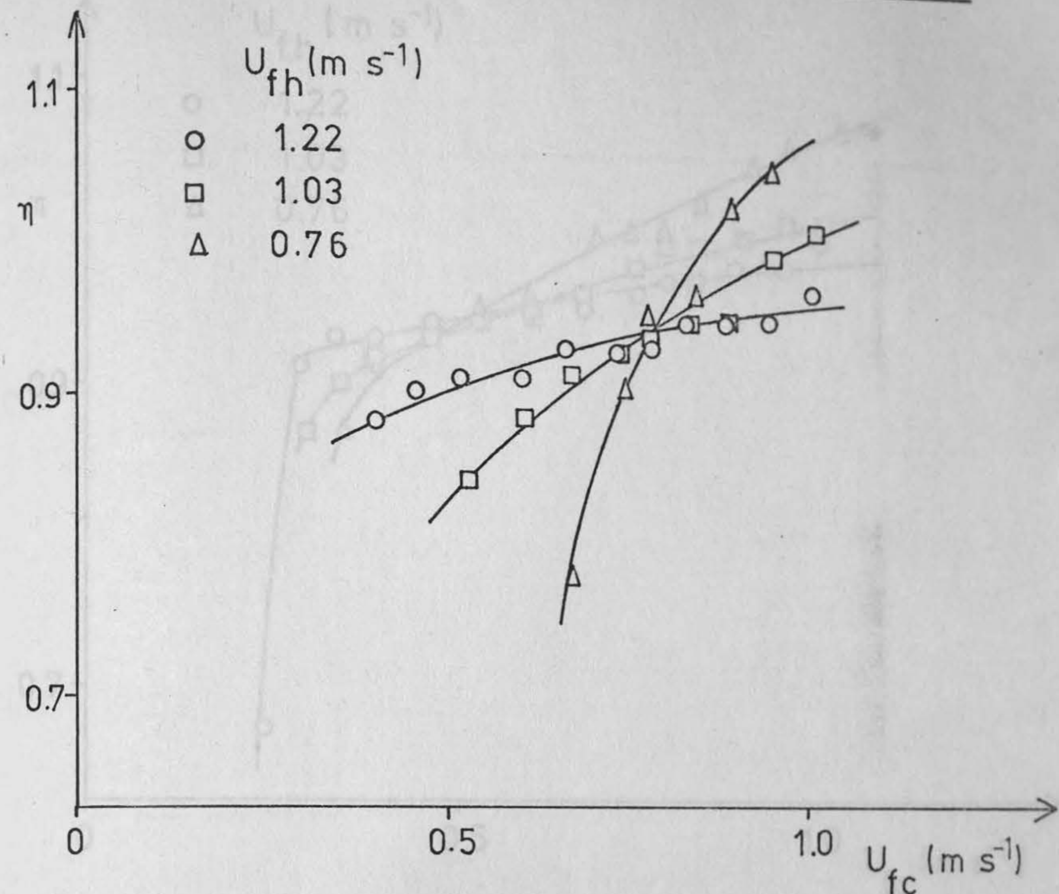


FIGURE 6.11 $Z = 15$ mm, $d_p = 380$ µm, GAP = 10 mm

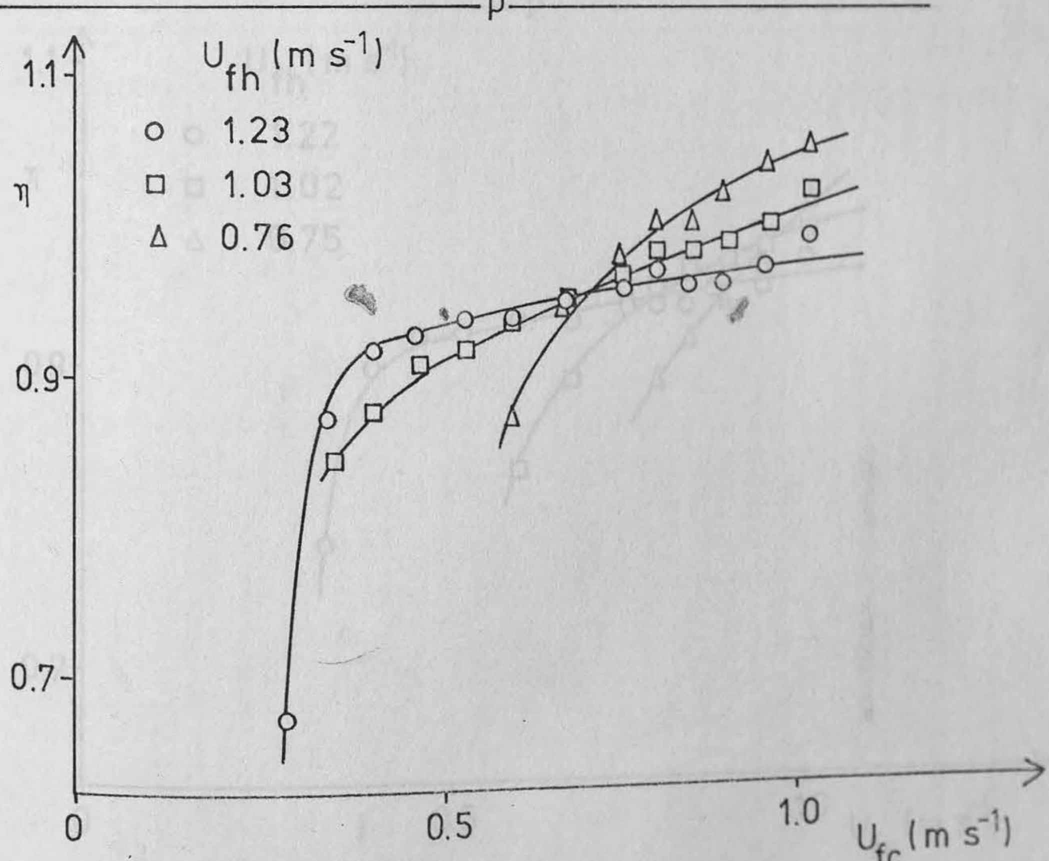


FIGURE 6.12 Z = 15 mm, d_p = 380 µm, NO PARTITION

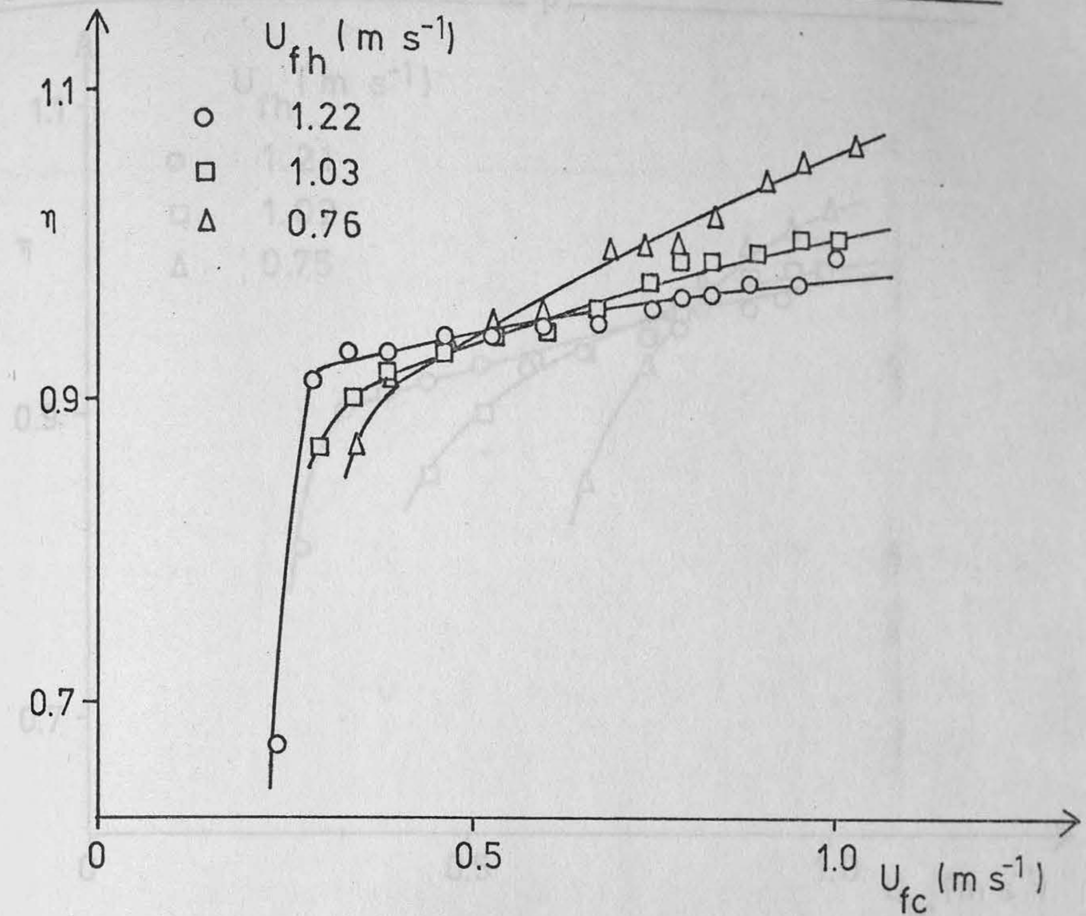


FIGURE 6.13 Z = 20 mm, d_p = 380 µm, GAP = 5 mm

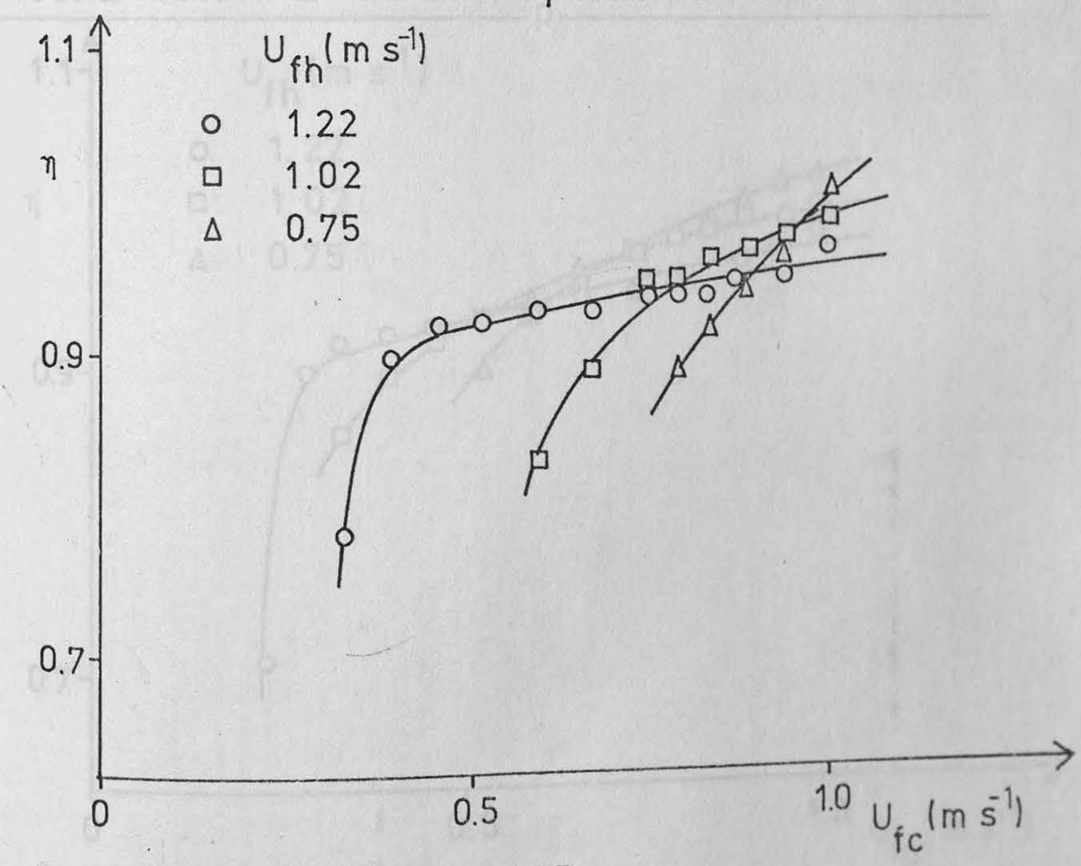


FIGURE 6.14 $Z = 20$ mm, $d_p = 380\ \mu m$, GAP = 10 mm

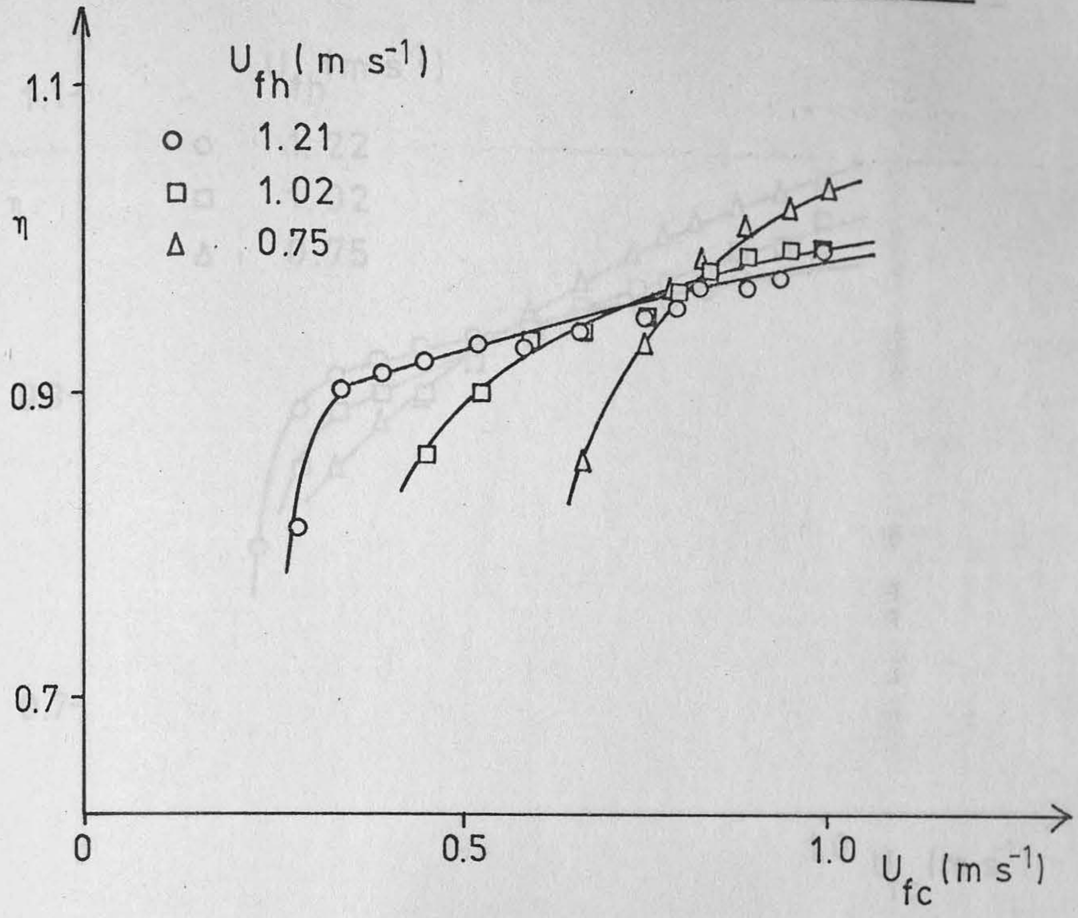


FIGURE 6.15 $Z = 20$ mm, $d_p = 380\ \mu m$, GAP = 15 mm

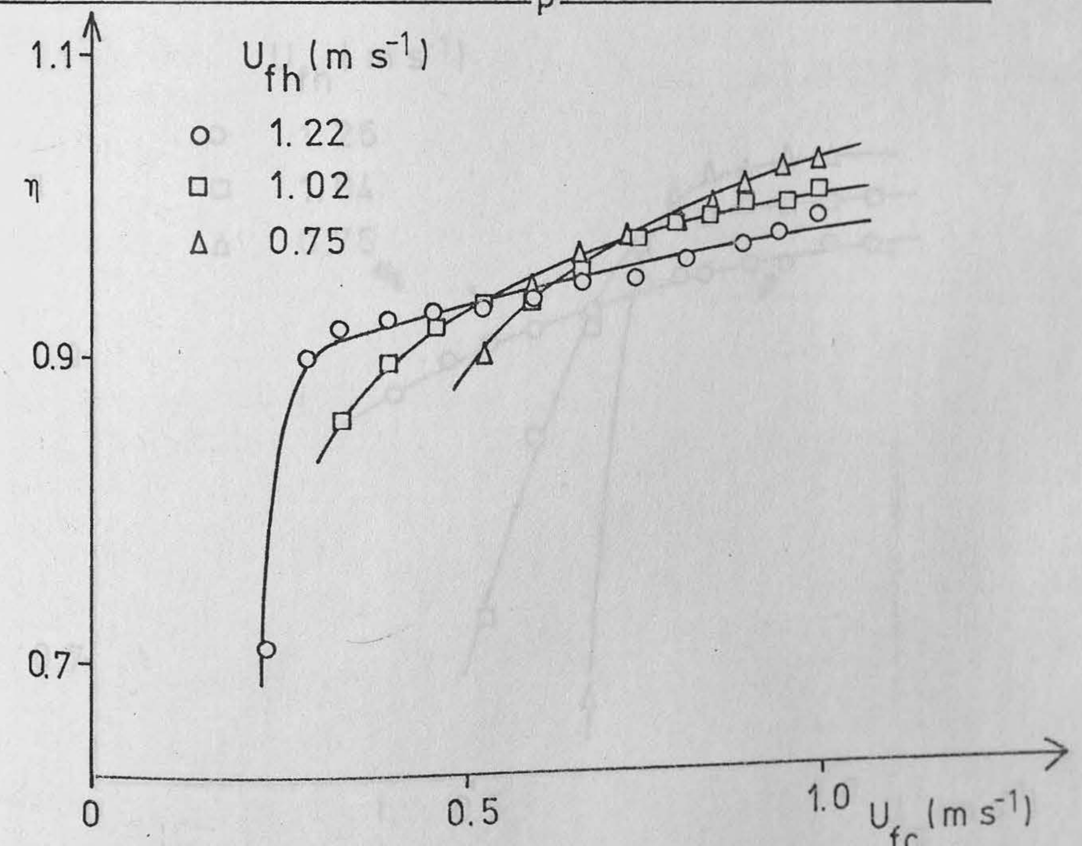


FIGURE 6.16 Z = 20 mm, d_p = 380 μm, NO PARTITION

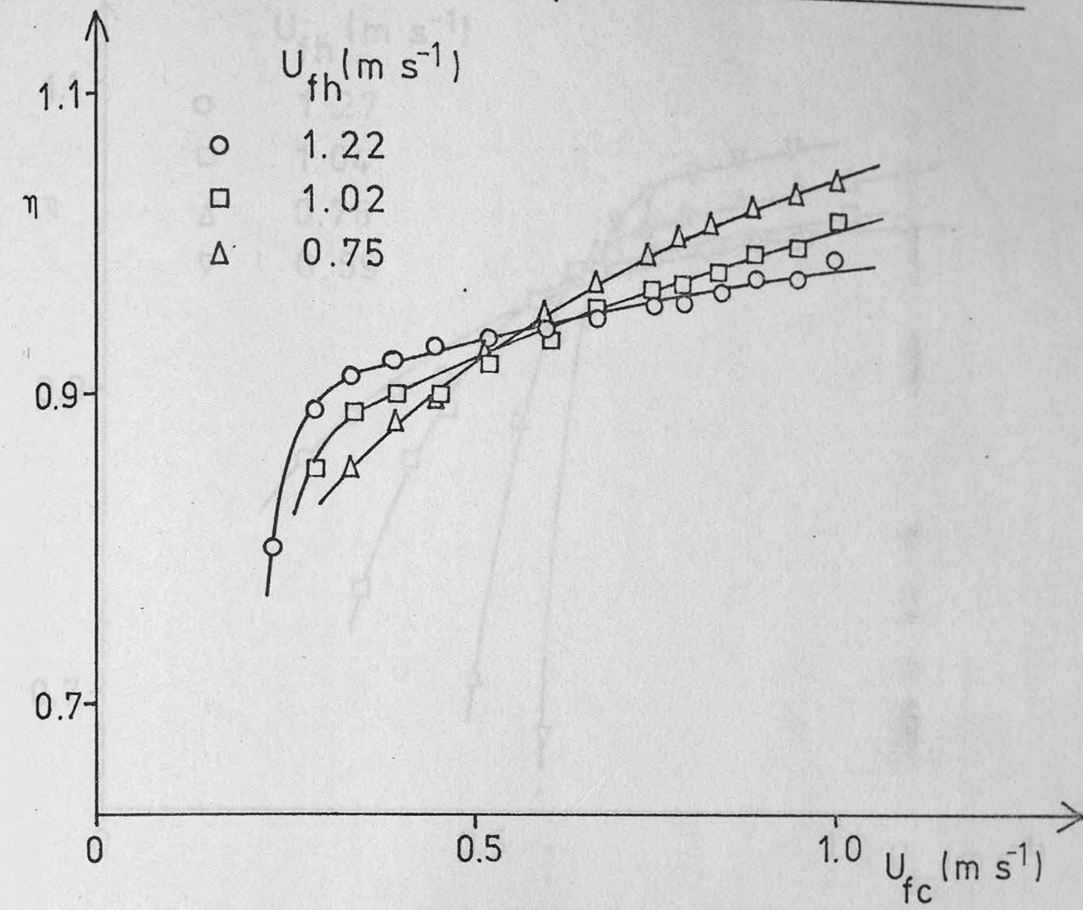


FIGURE 6.17 Z = 25 mm, d_p = 380 μm, GAP = 10 mm

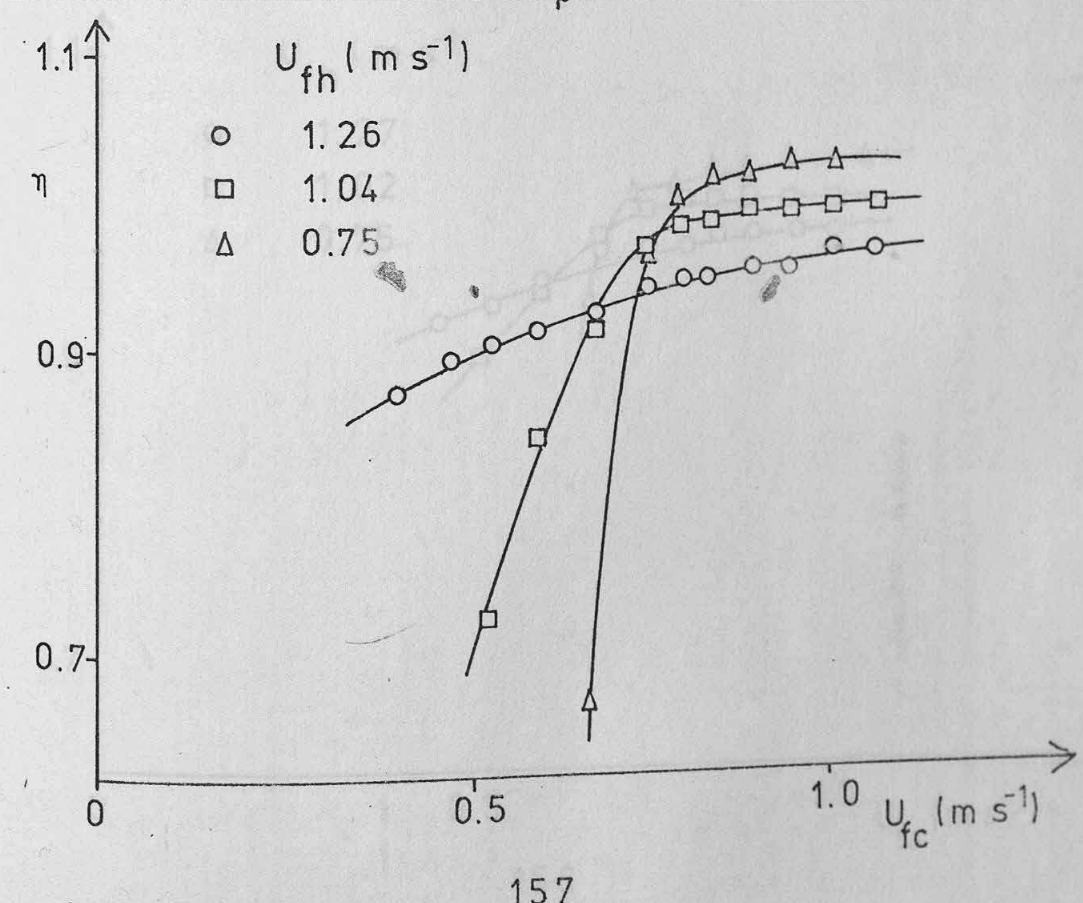


FIGURE 6.18 Z = 25 mm, d_p = 380 μm, GAP = 15 mm

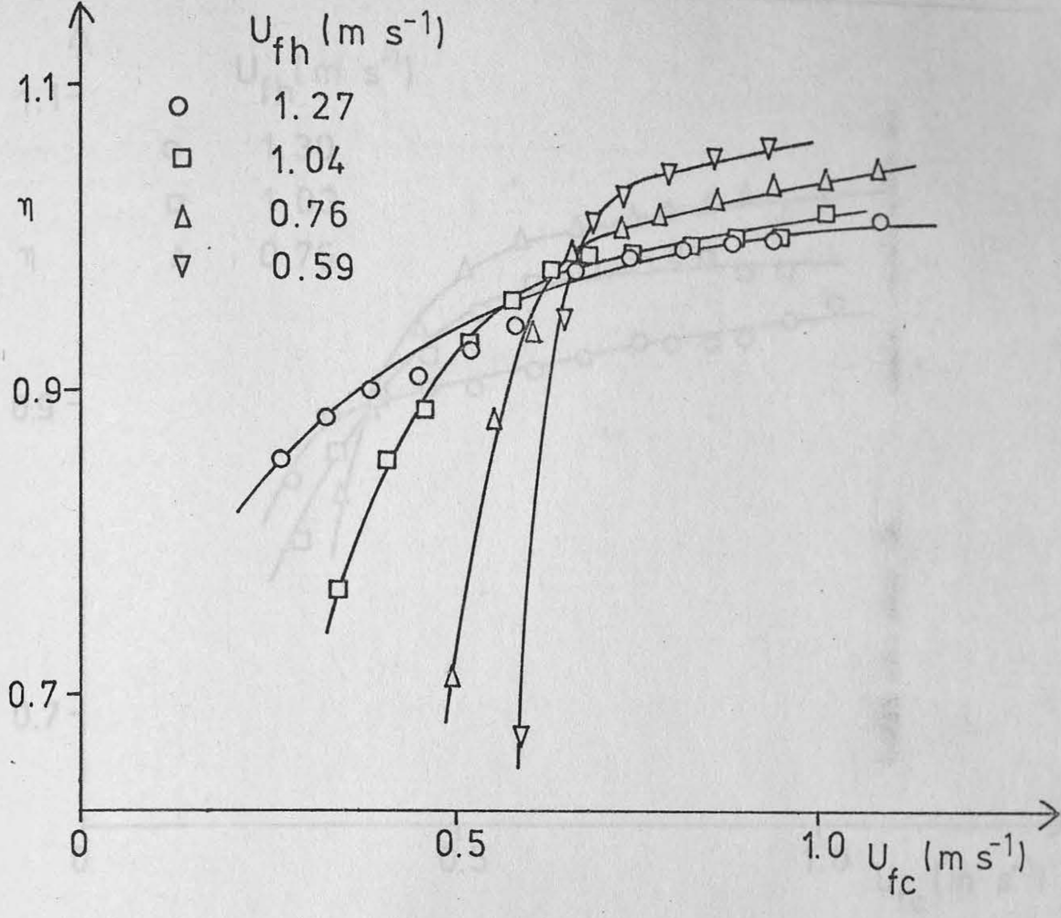


FIGURE 6.19 Z = 25 mm, d_p = 380 μm, GAP = 20 mm

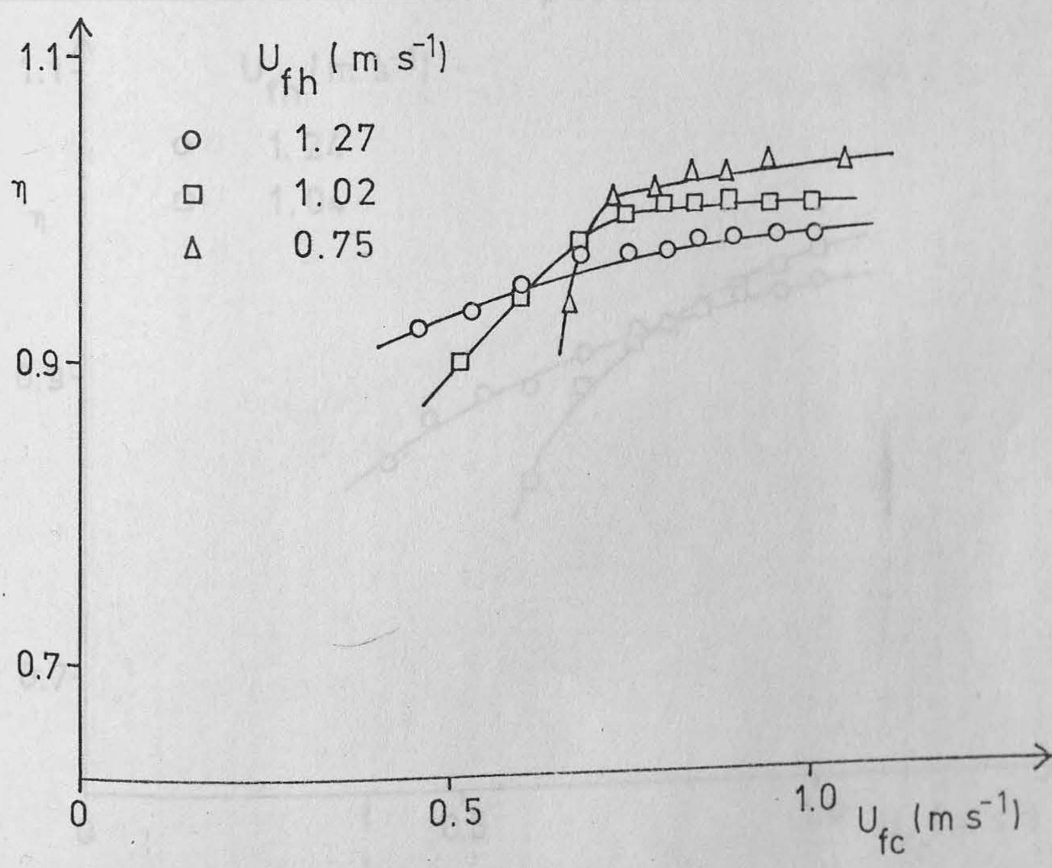


FIGURE 6.20 Z = 25 mm, d_p = 380 µm, NO PARTITION

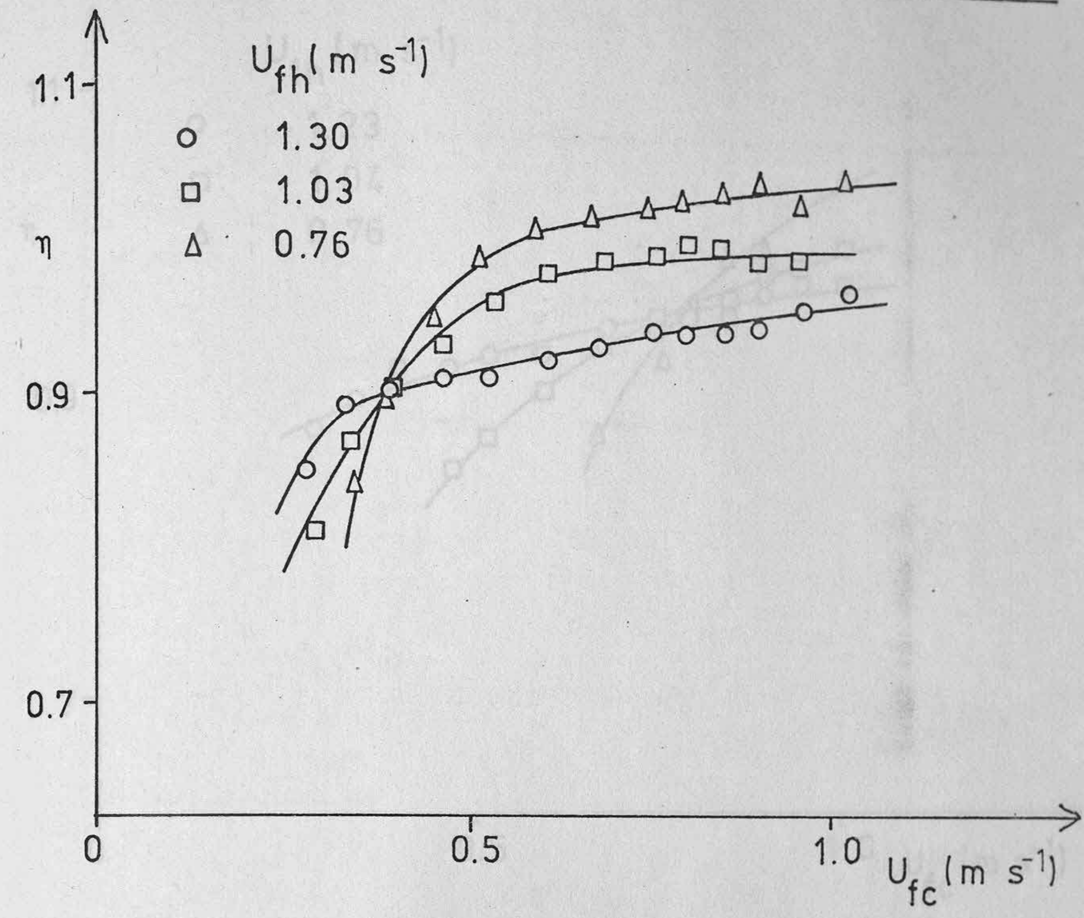


FIGURE 6.21 Z = 30 mm, d_p = 380 µm, GAP = 10 mm

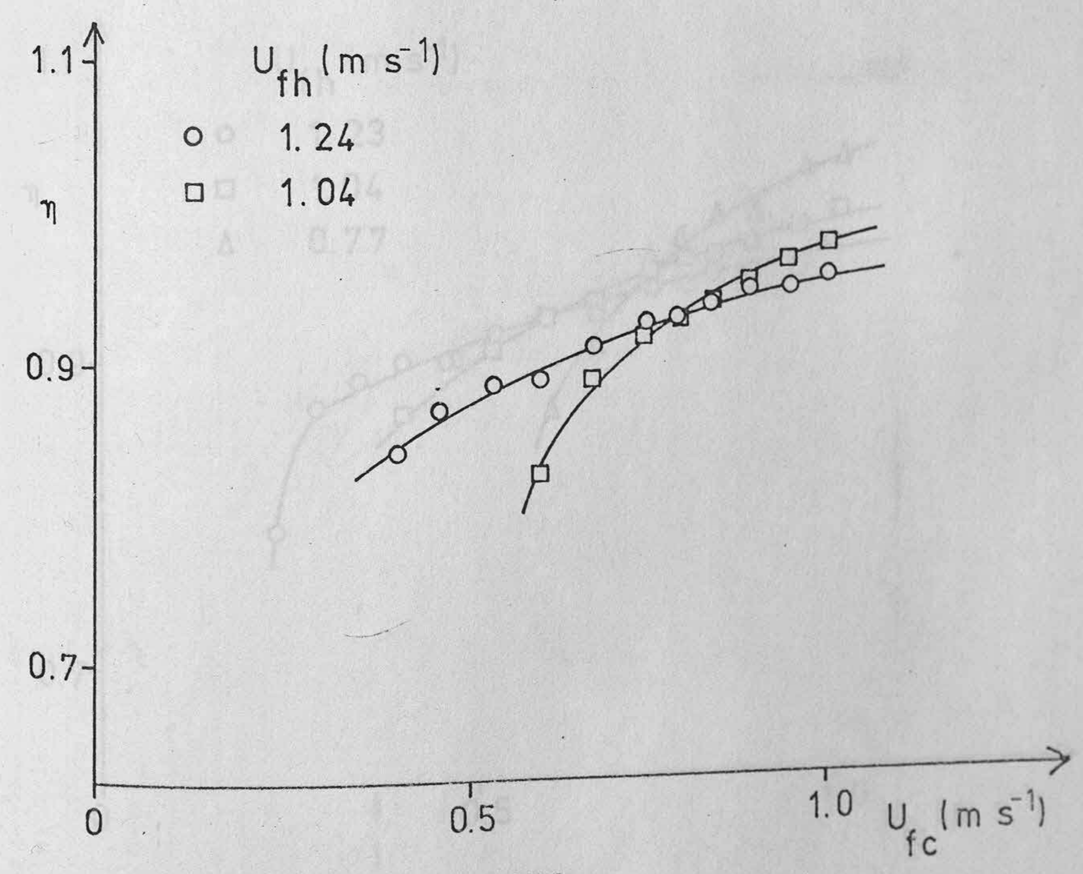


FIGURE 6.22 $Z = 30$ mm, $d_p = 380$ µm, GAP = 15 mm

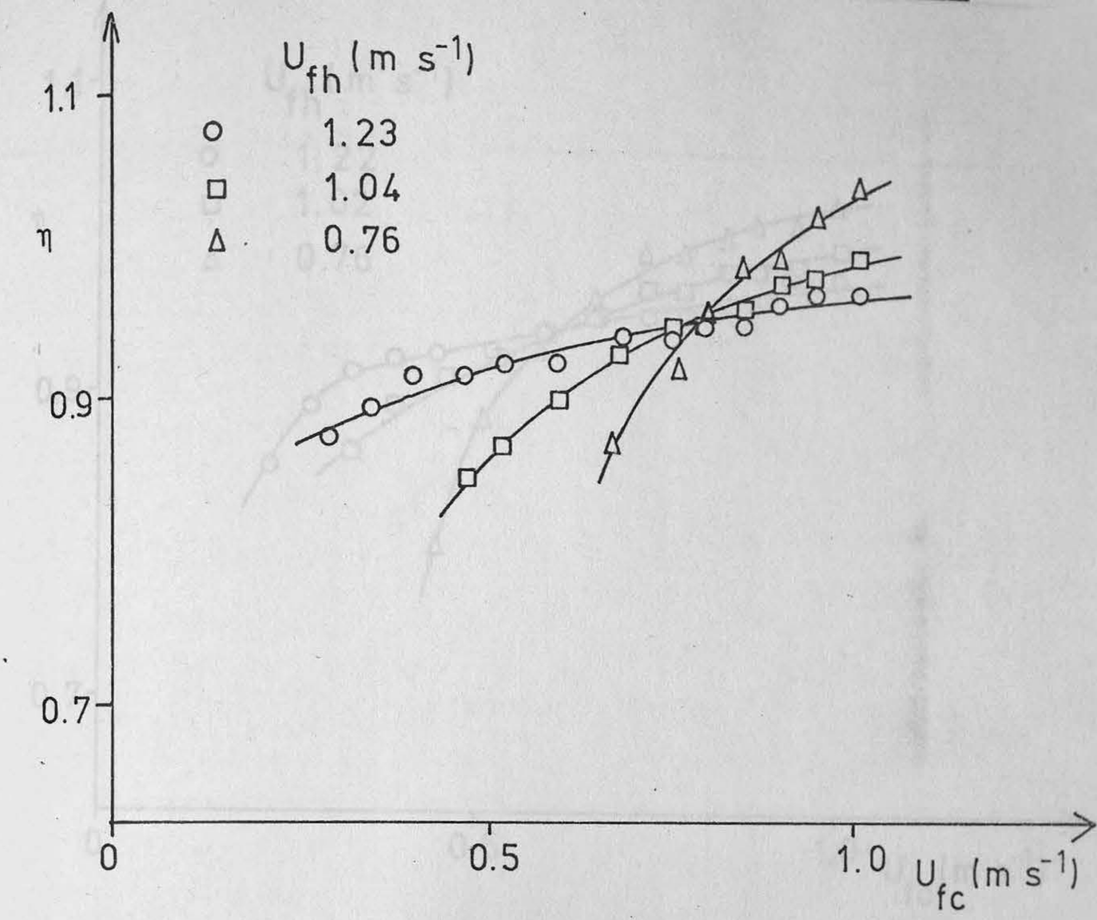


FIGURE 6.23 $Z = 30$ mm, $d_p = 380$ µm, GAP = 20 mm

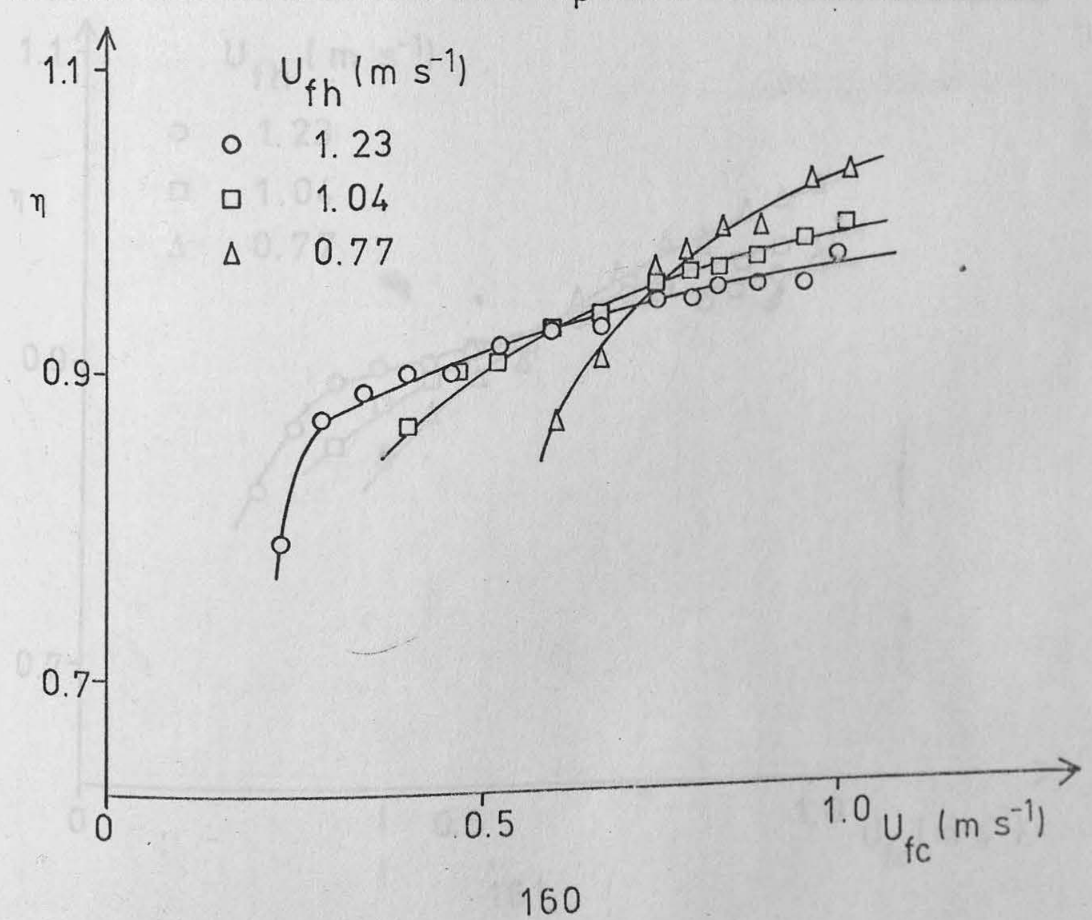


FIGURE 6.24 Z = 30 mm, d_p = 380 µm, GAP = 25 mm

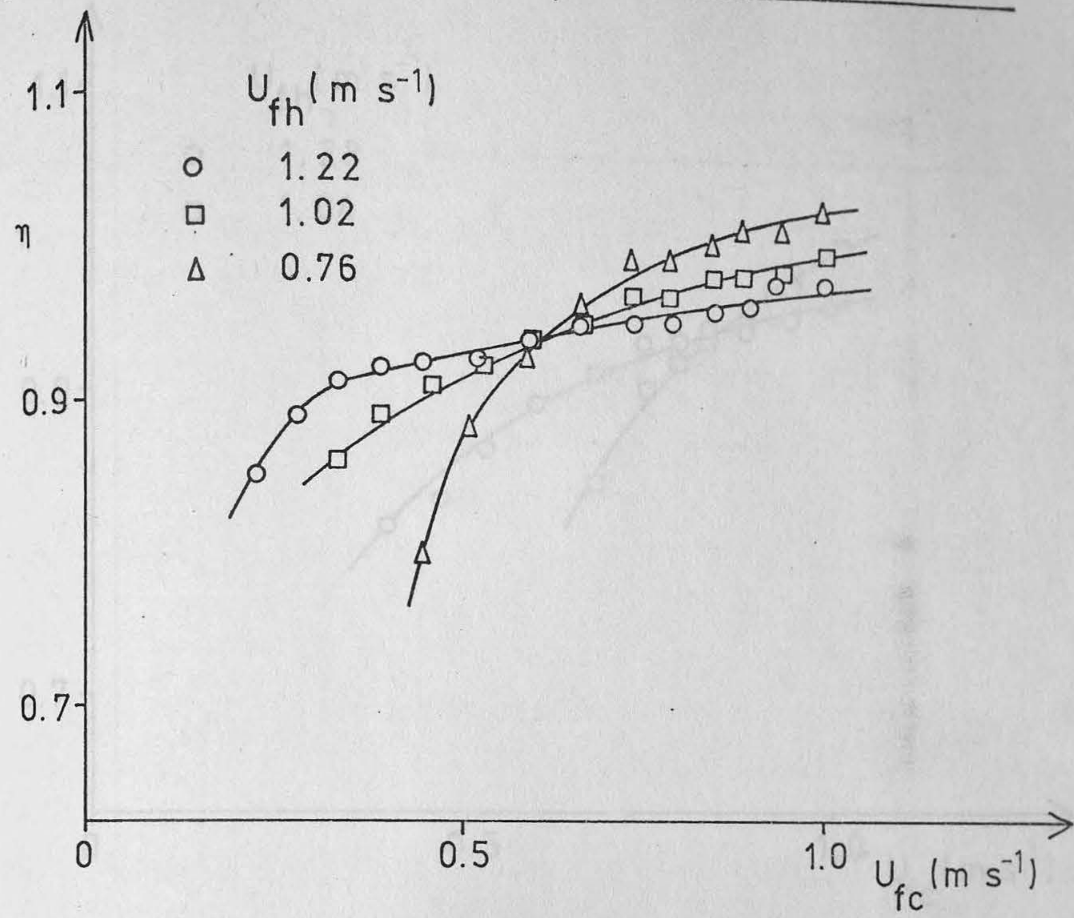


FIGURE 6.25 Z = 30 mm, d_p = 380 µm, NO PARTITION

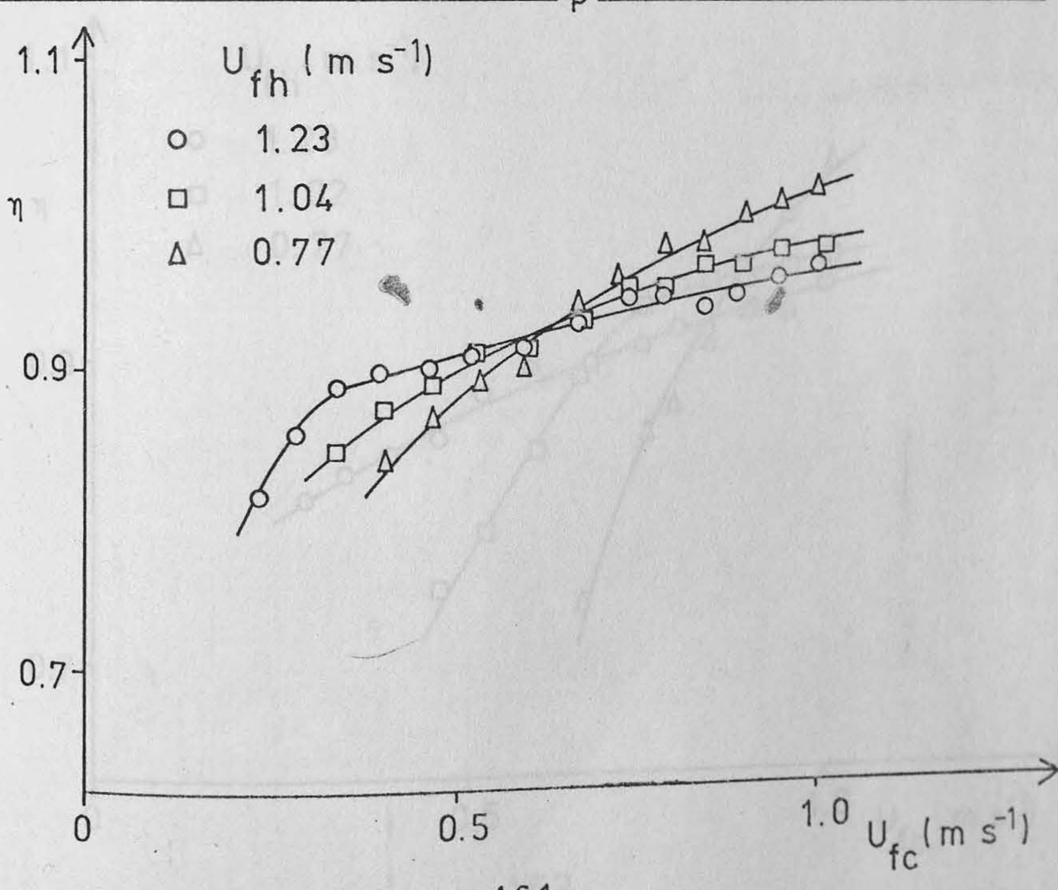


FIGURE 6.26 Z = 40 mm, d_p = 380 μm, GAP = 10 mm

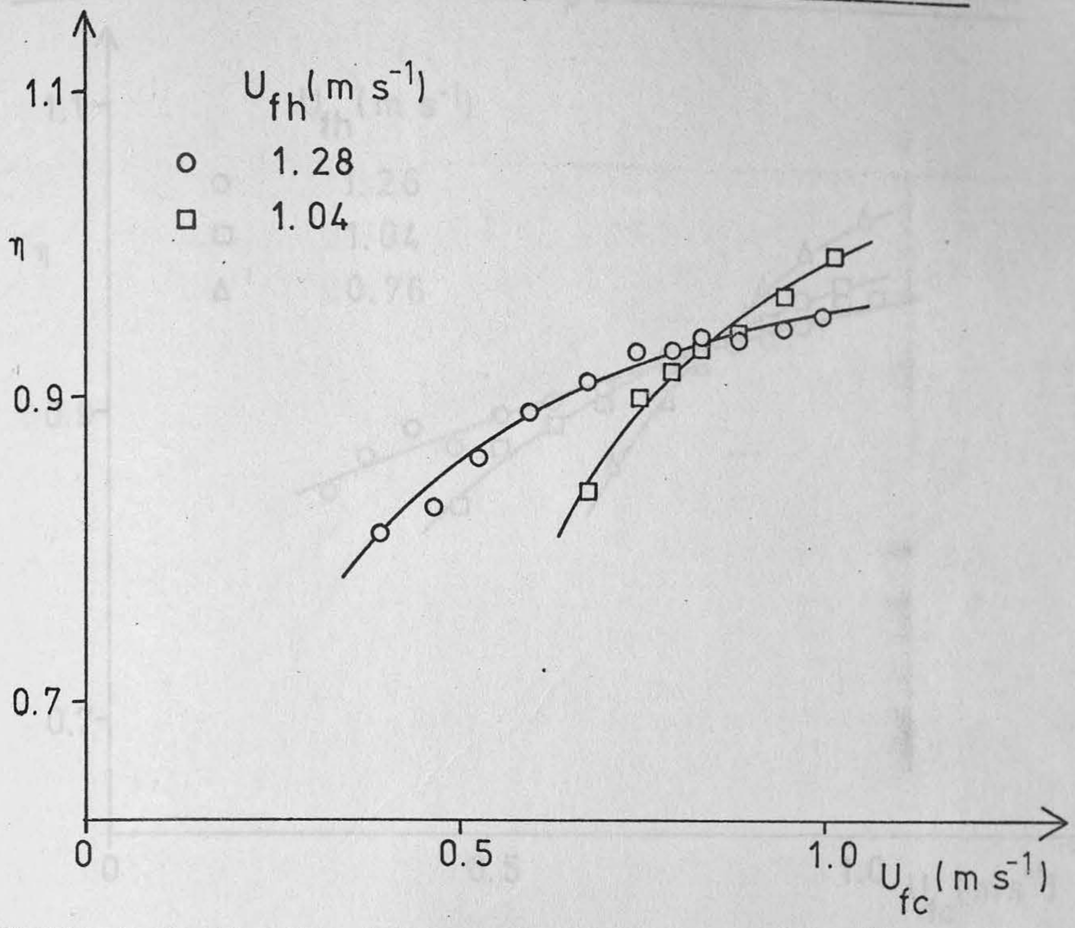


FIGURE 6.27 Z = 40 mm, d_p = 380 μm, GAP = 15 mm

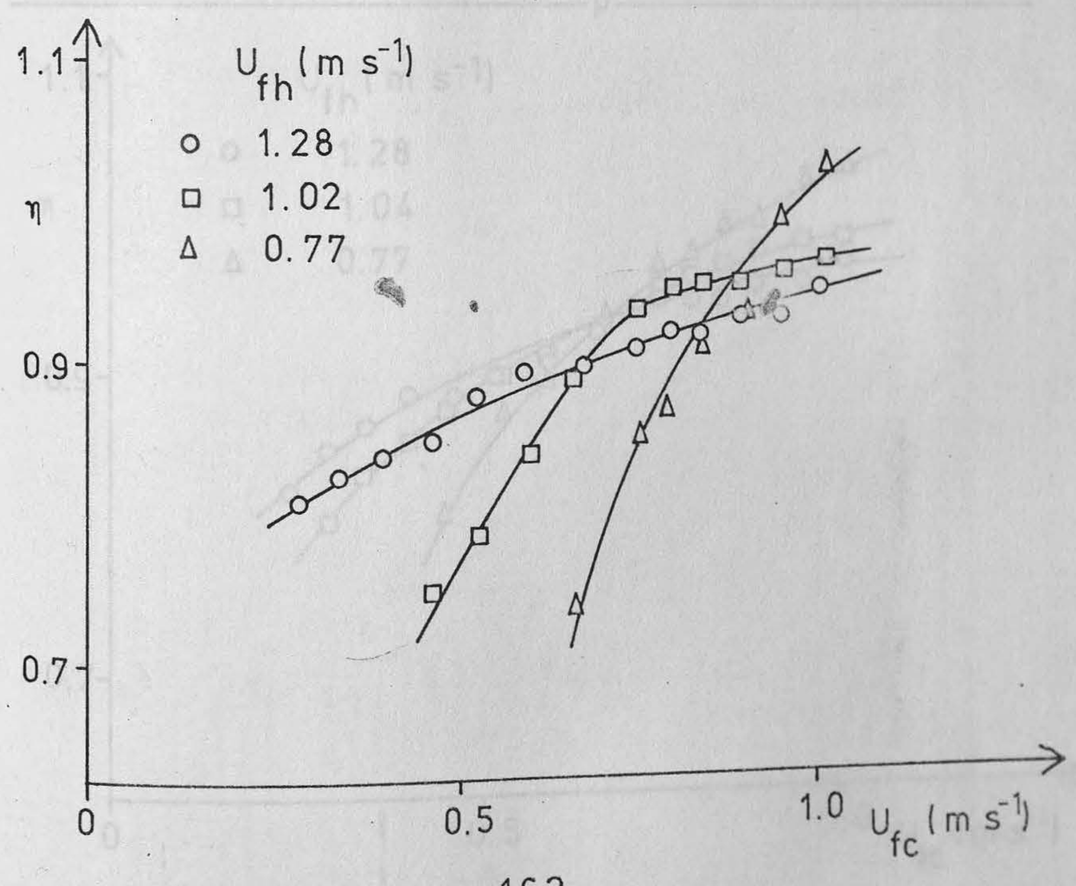


FIGURE 6.28 Z=40 mm, d_p = 380 µm, GAP = 20 mm

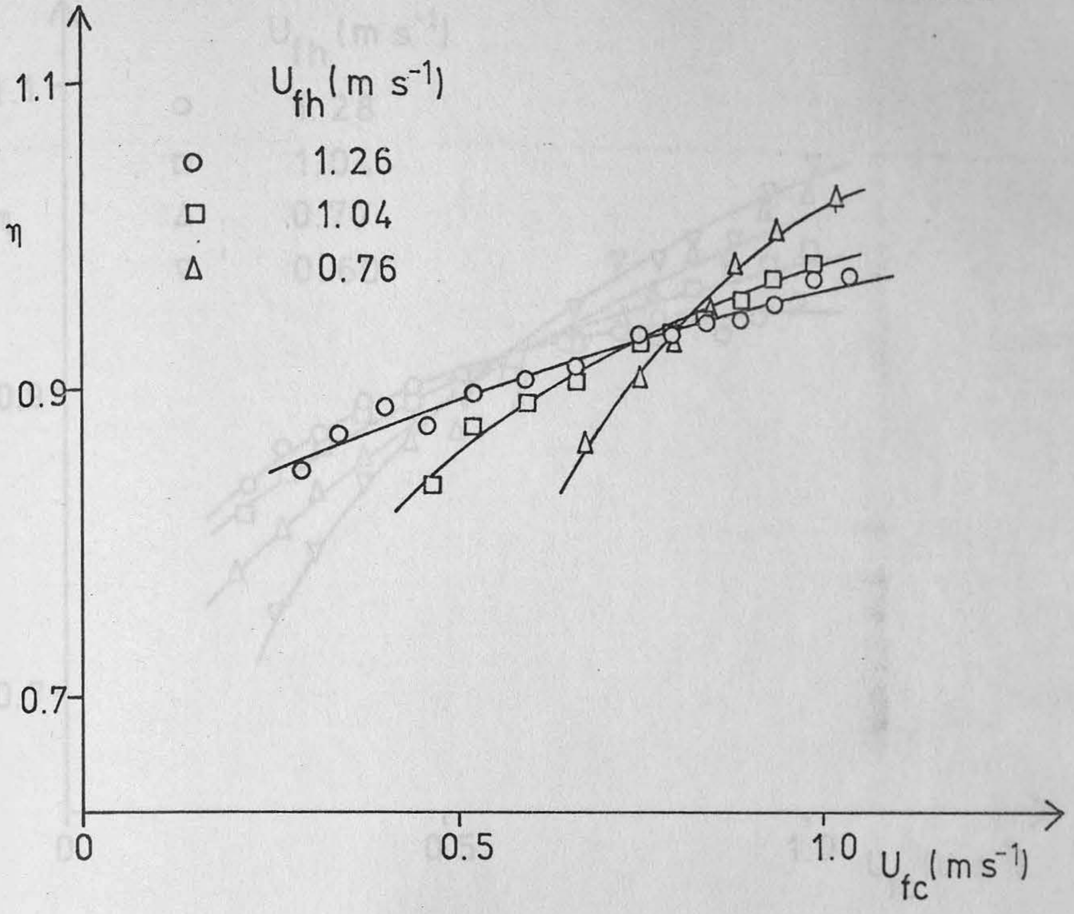


FIGURE 6.29 Z = 40 mm, d_p = 380 µm, GAP = 30 mm

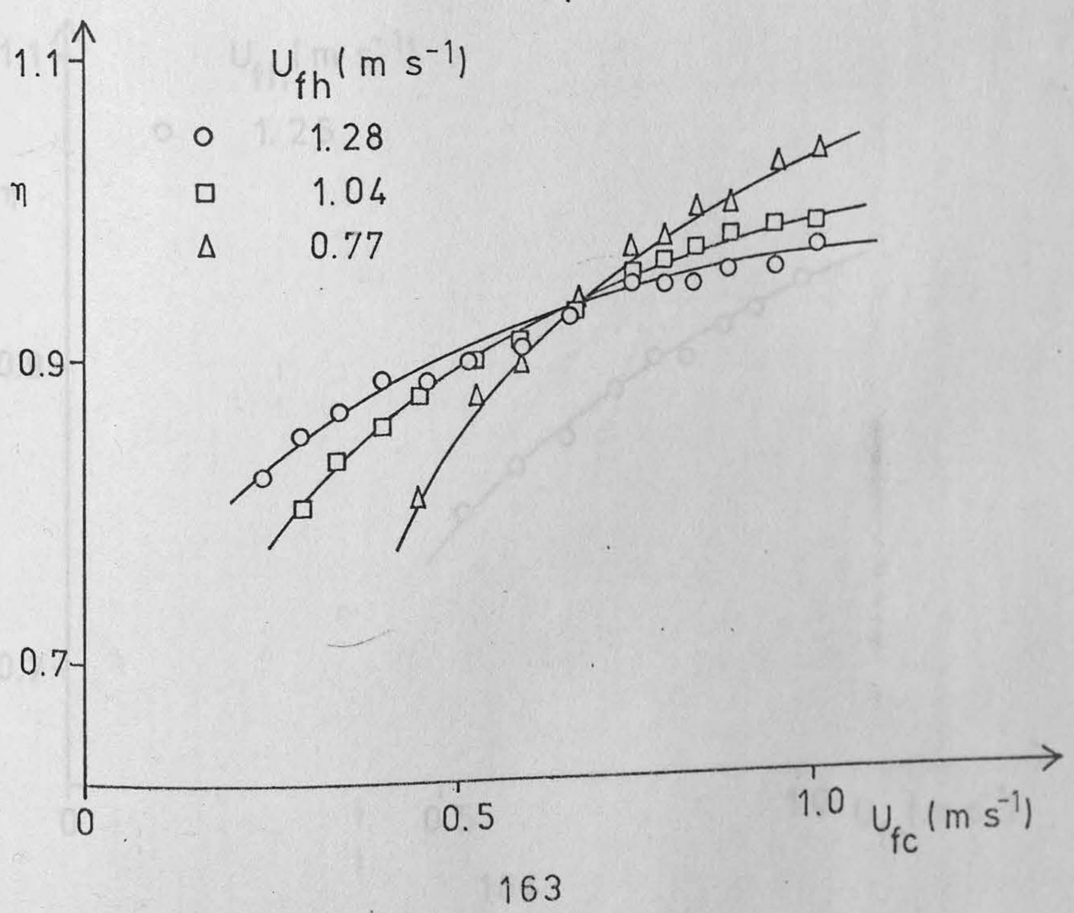


FIGURE 6.30 Z = 40 mm, d_p = 380 μm, NO PARTITION

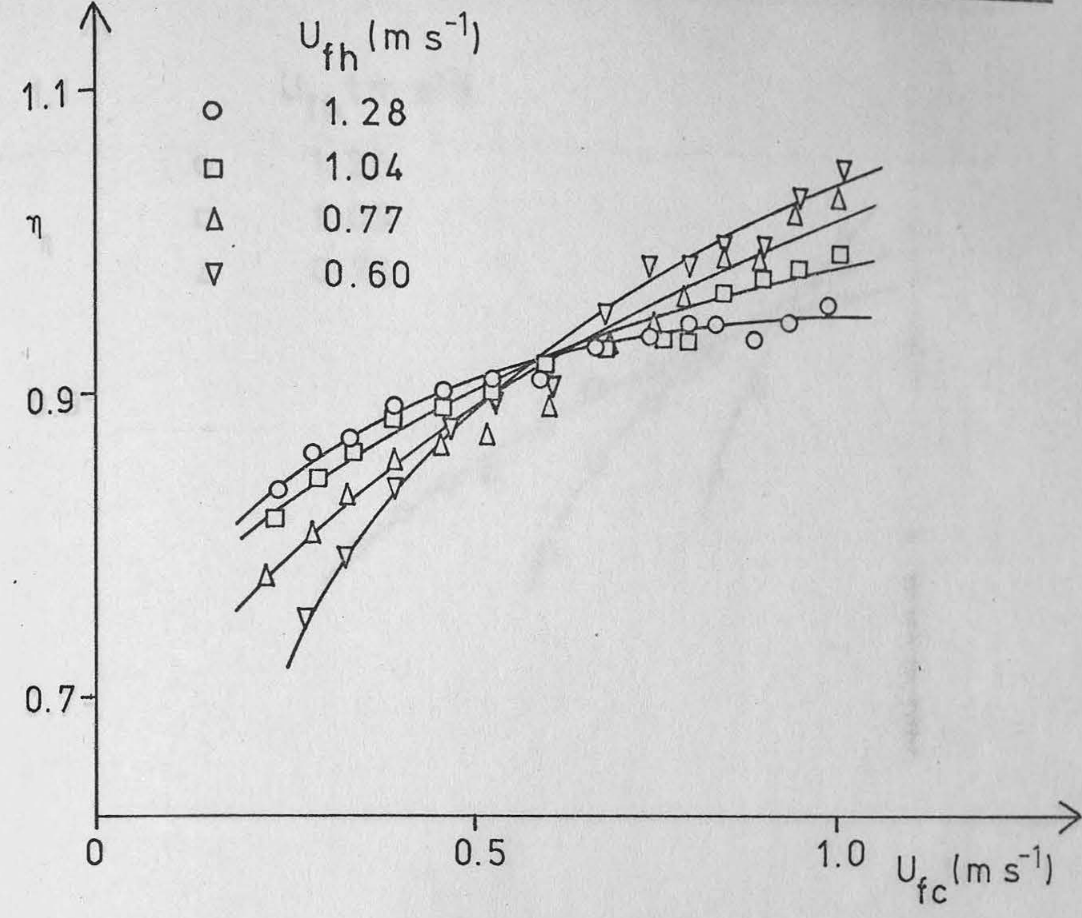


FIGURE 6.31 Z = 50 mm, d_p = 380 μm, GAP = 10 mm

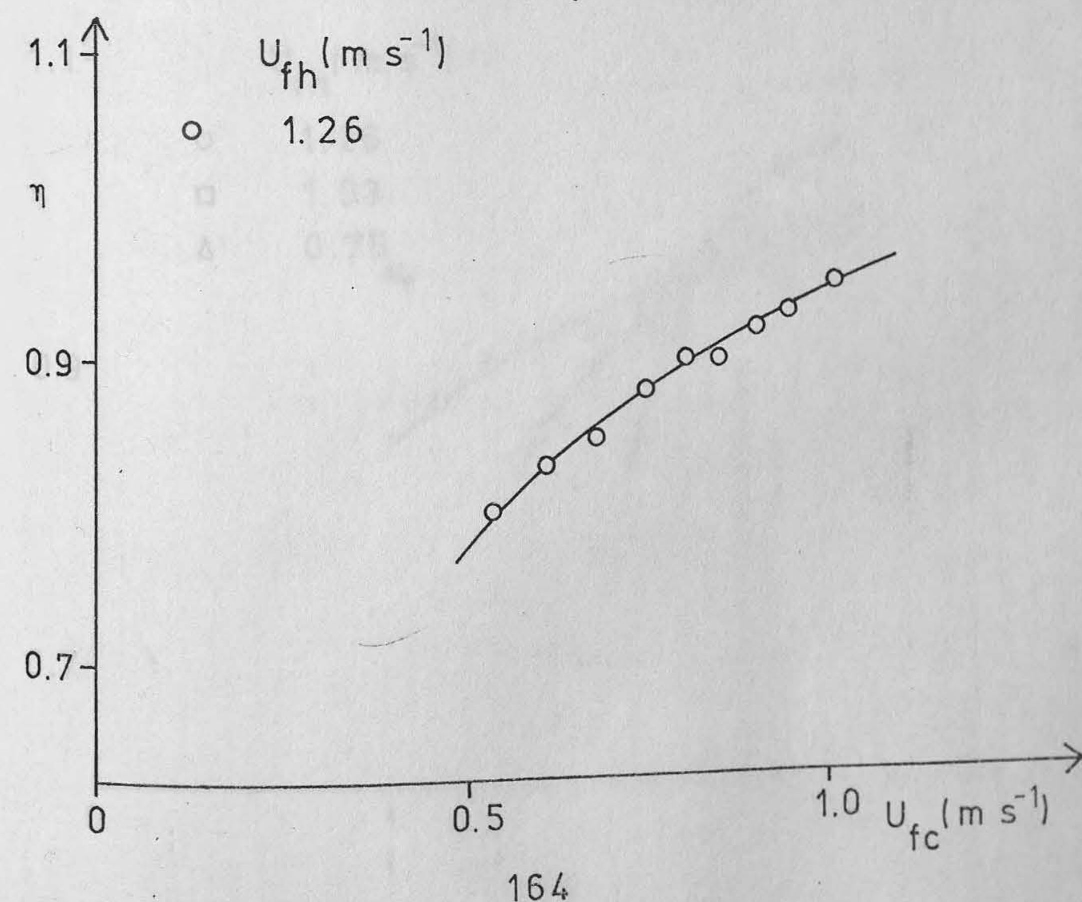


FIGURE 6.32 $Z = 50$ mm, $d_p = 380$ μm, GAP = 15 mm

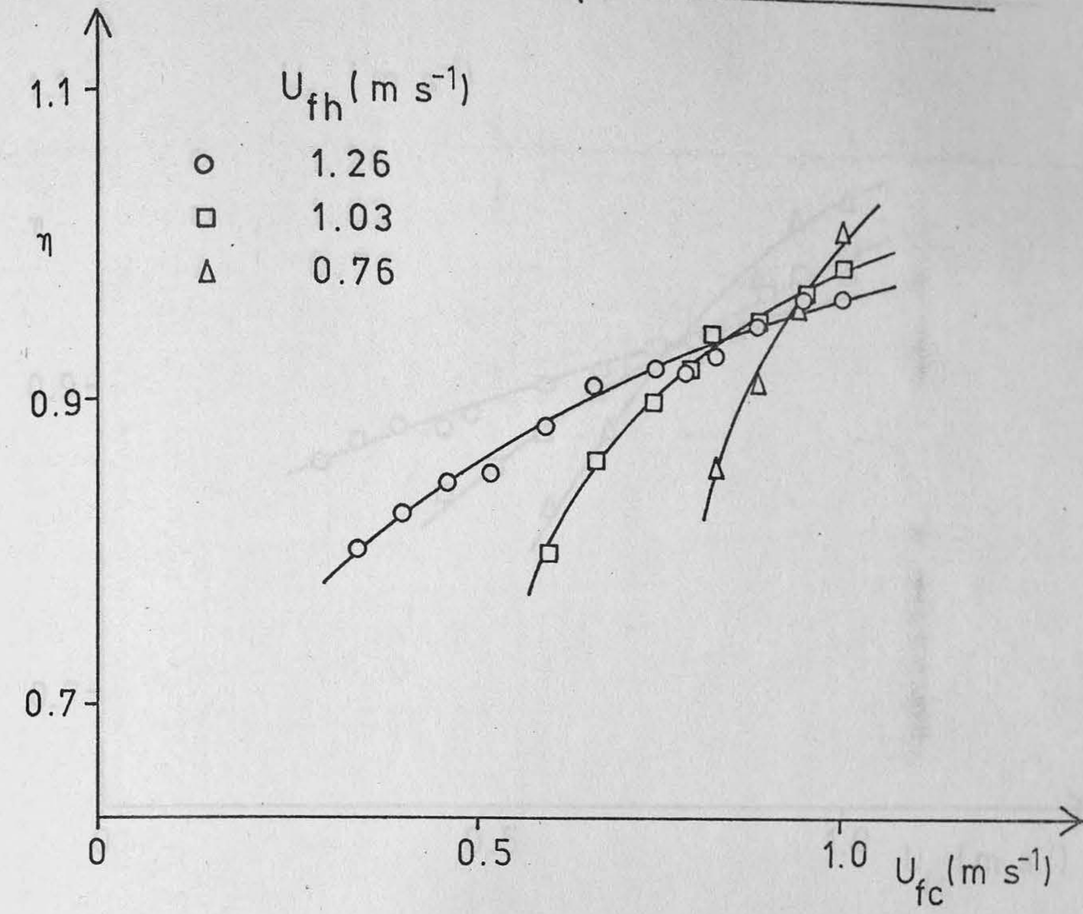


FIGURE 6.33 $Z = 50$ mm, $d_p = 380$ μm, GAP = 20 mm

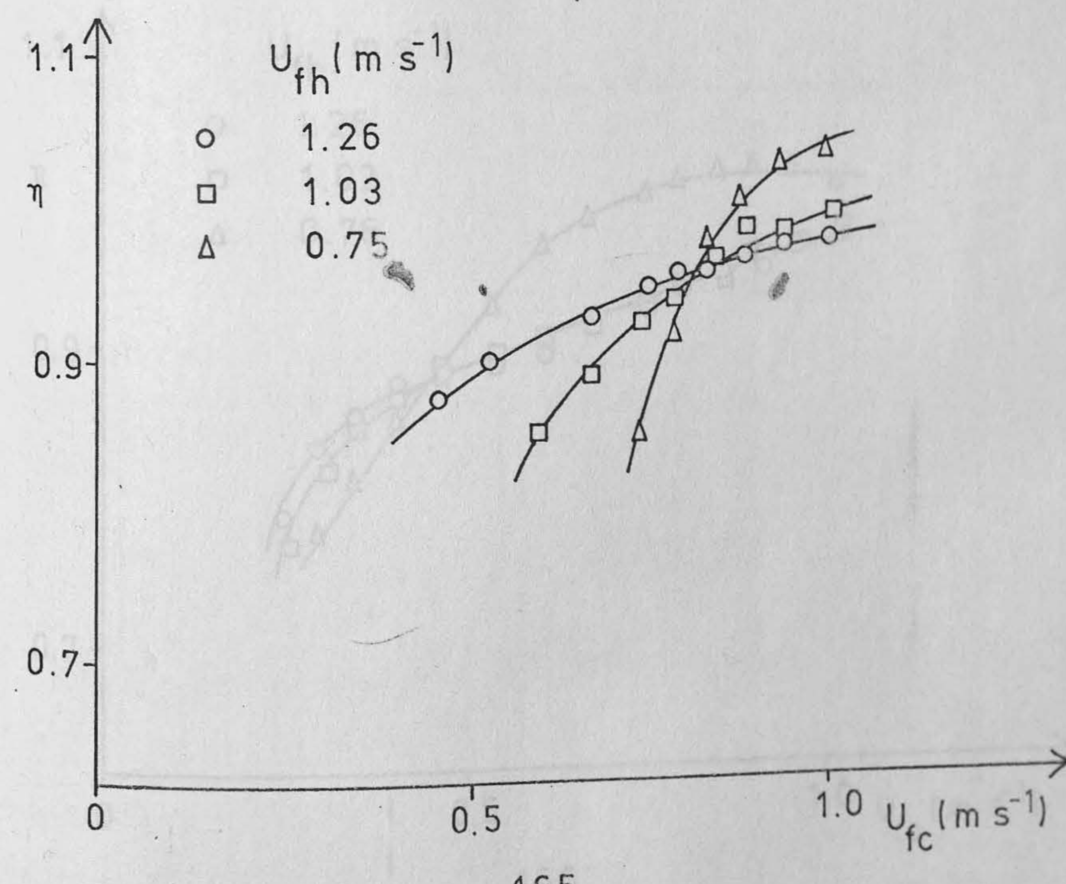


FIGURE 6.34 $Z = 50$ mm, $d_p = 380$ μm, GAP = 30 mm

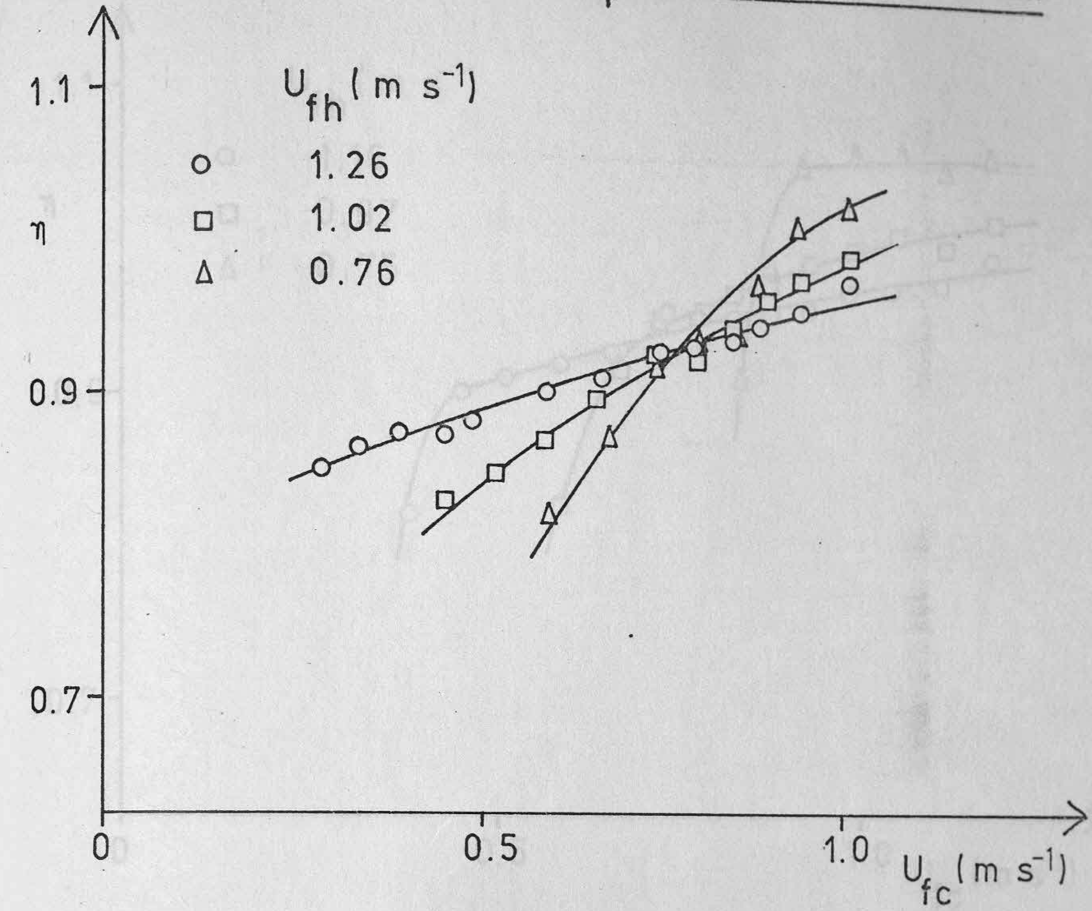


FIGURE 6.35 $Z = 50$ mm, $d_p = 380$ μm, NO PARTITION

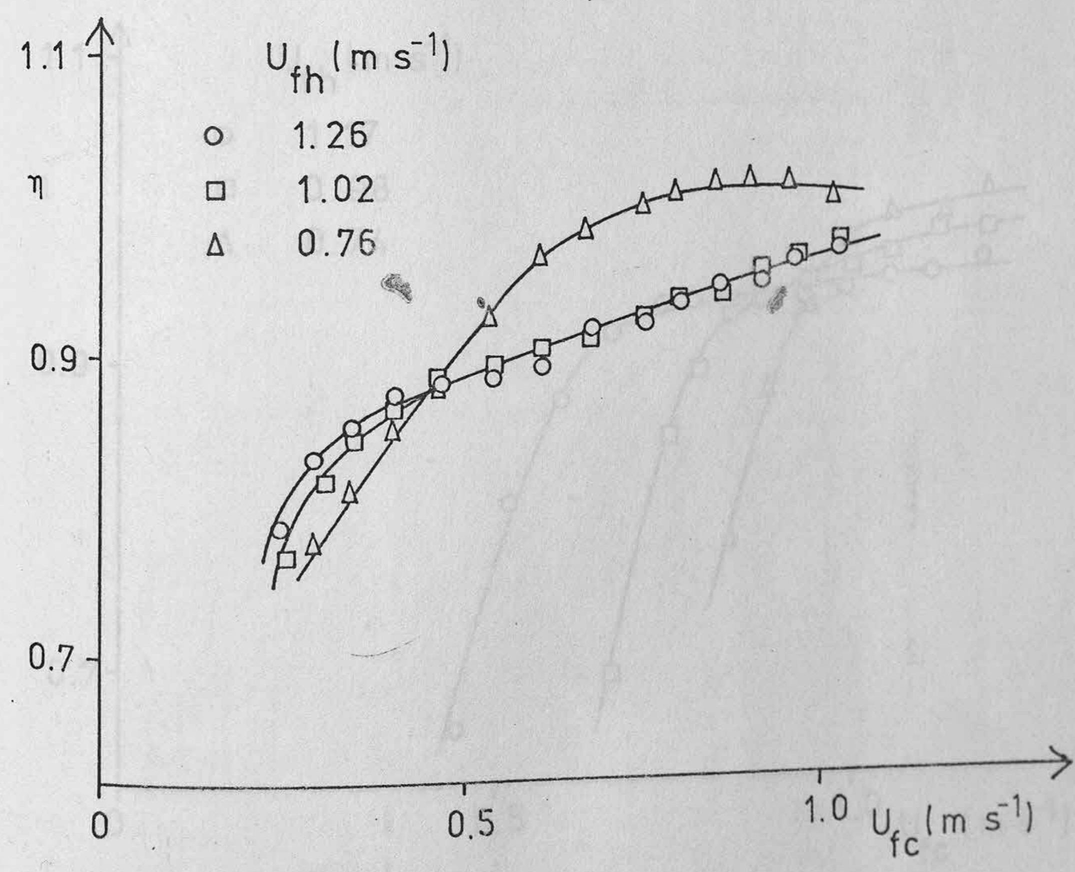


FIGURE 6.36 Z = 15 mm, d_p = 520 µm, GAP = 10 mm

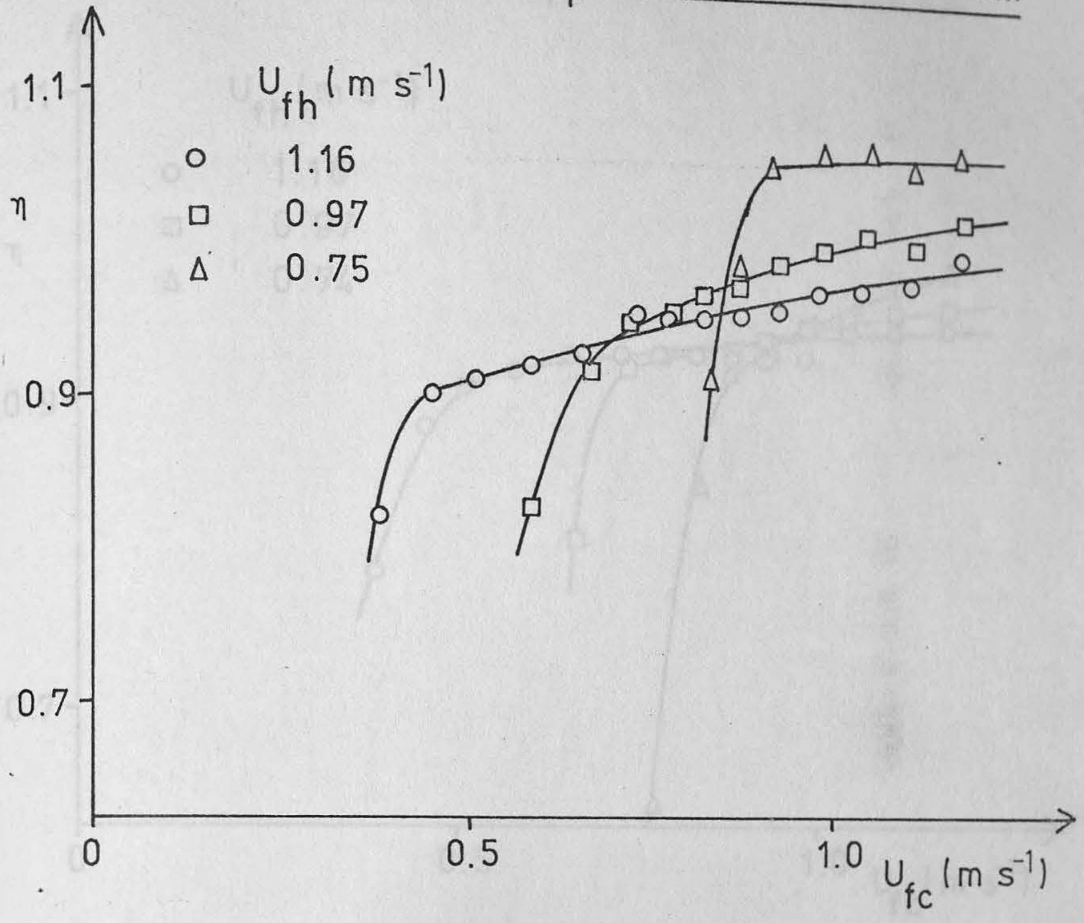


FIGURE 6.37 Z = 20 mm, d_p = 520 µm, GAP = 10 mm

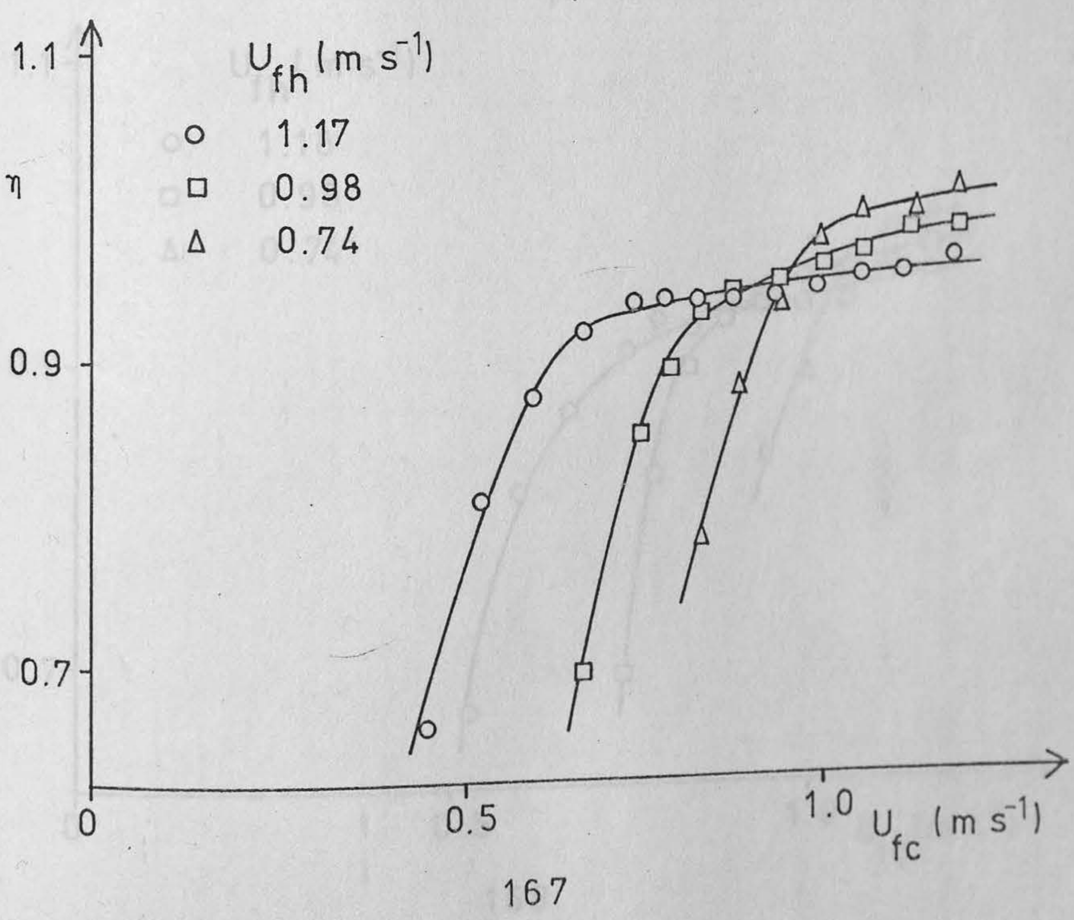


FIGURE 6.38 Z = 20 mm, d_p = 520 µm, GAP = 15 mm

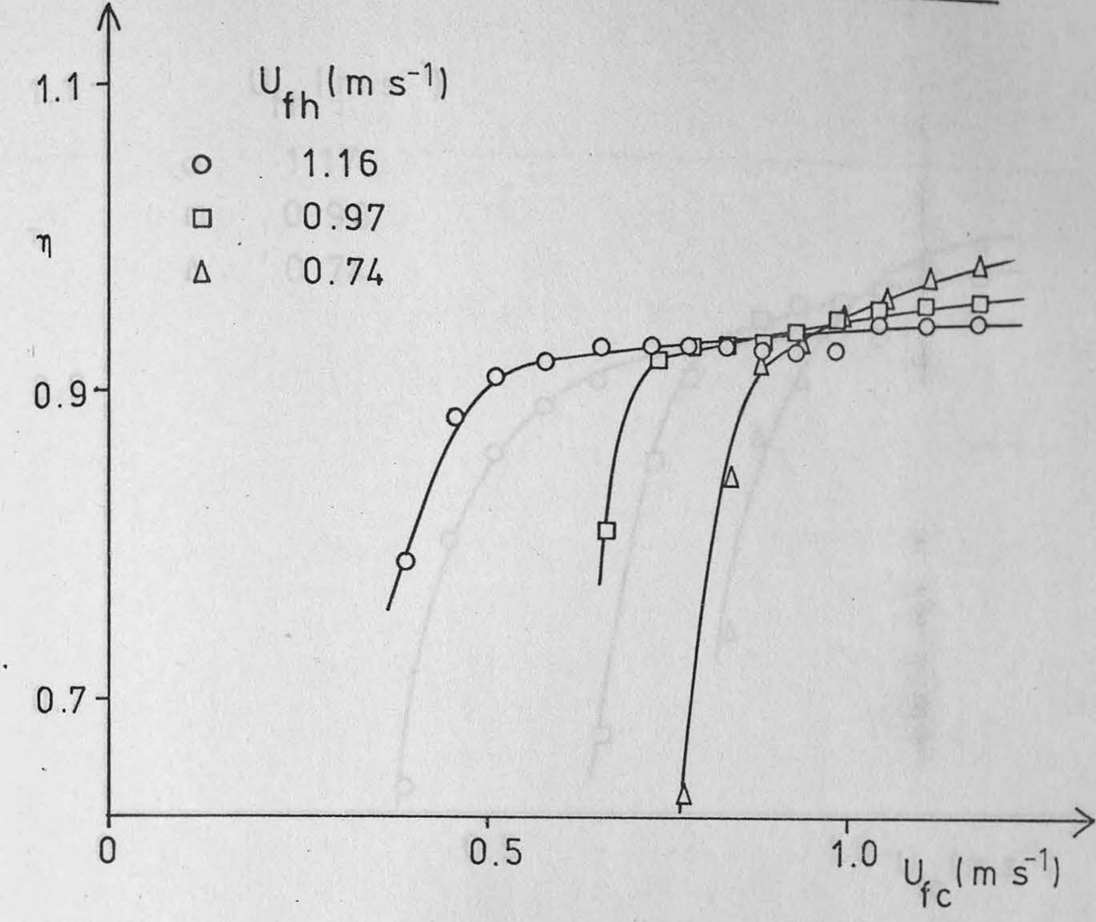


FIGURE 6.39 Z = 25 mm, d_p = 520 µm, GAP = 10 mm

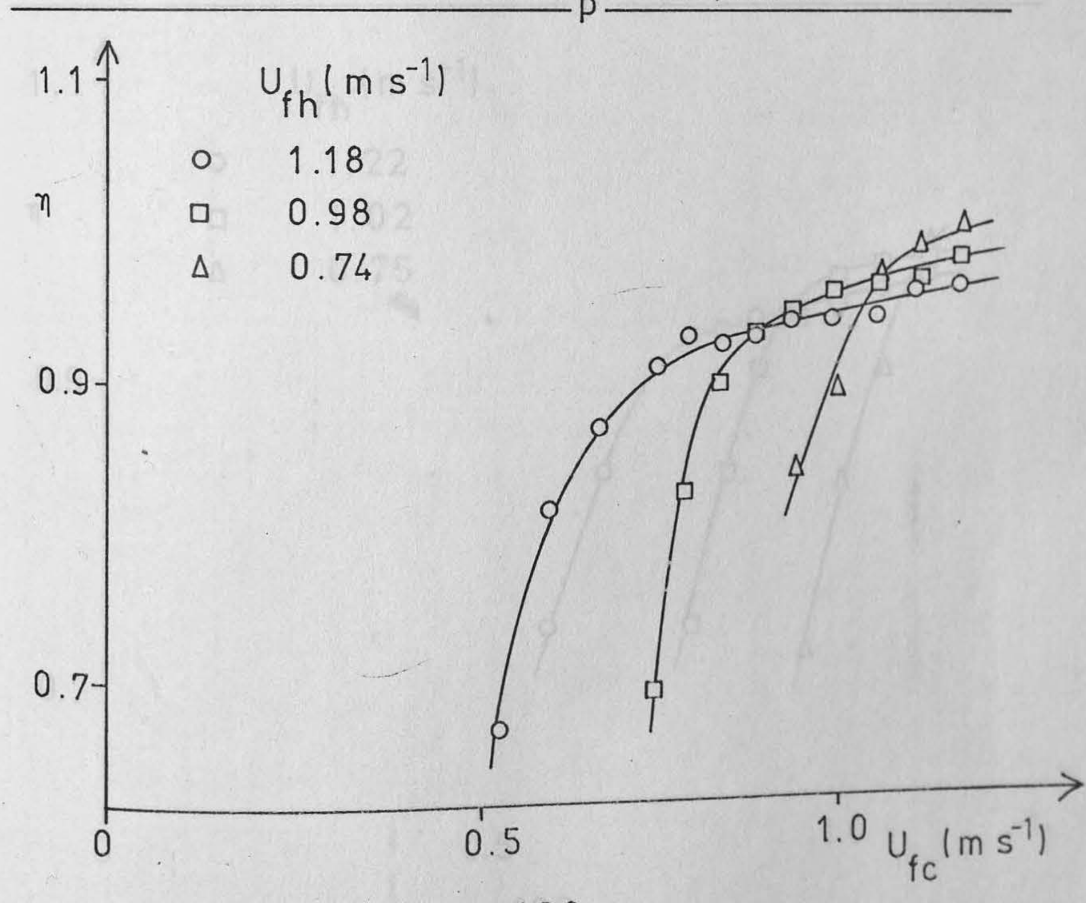


FIGURE 6.40 $Z = 25$ mm, $d_p = 520$ µm, GAP = 15 mm

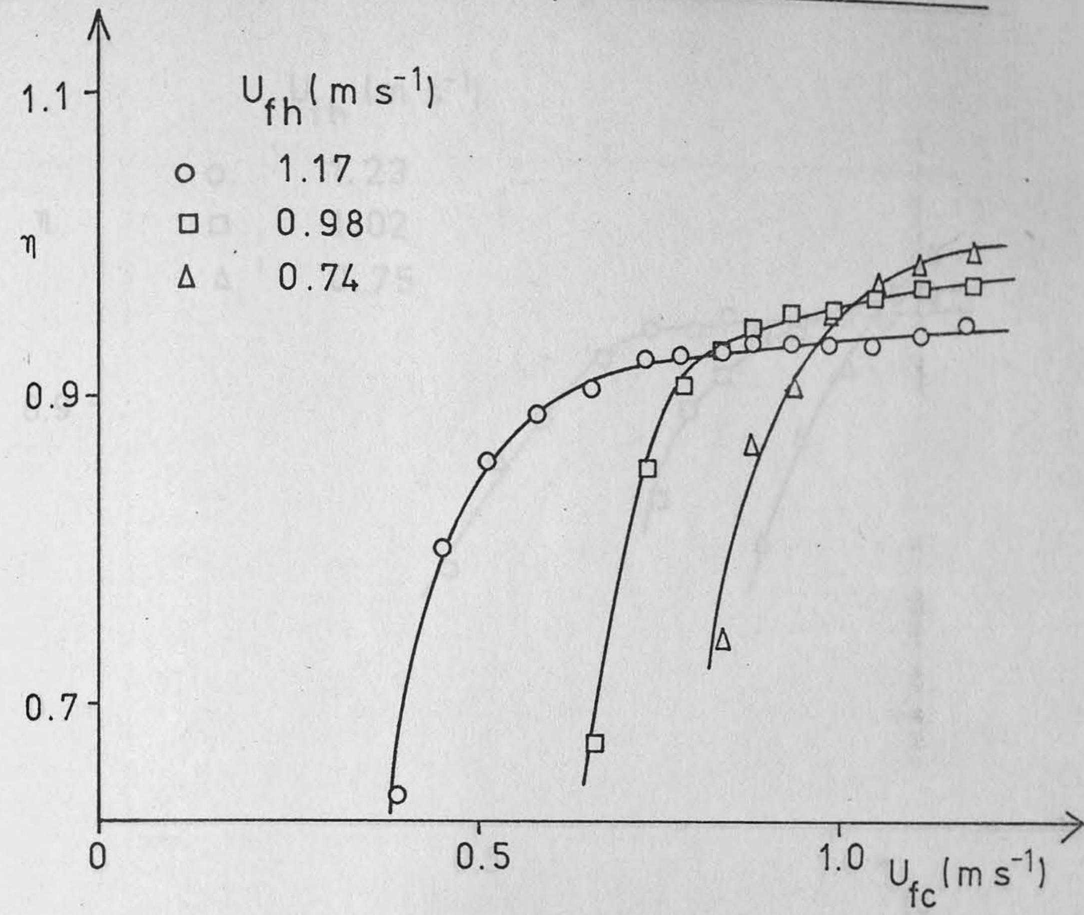


FIGURE 6.41 $Z = 30$ mm, $d_p = 520$ µm, GAP = 10 mm

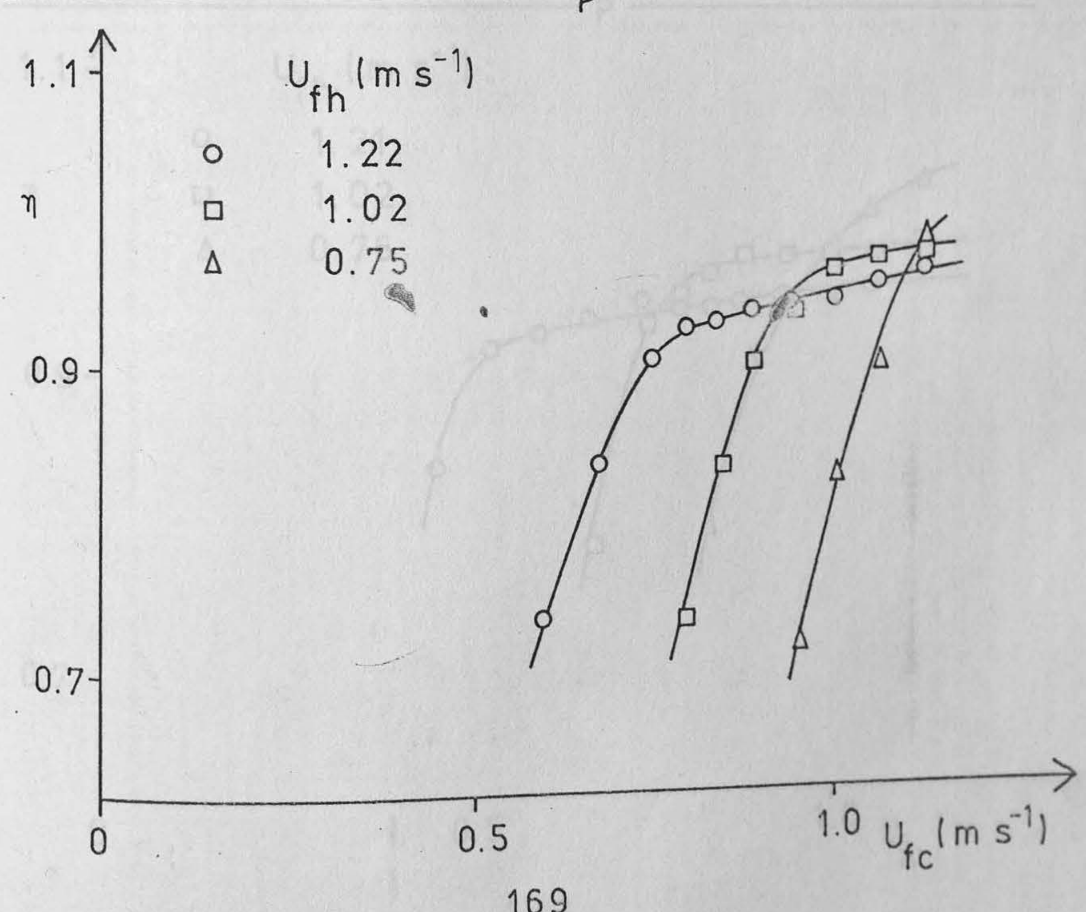


FIGURE 6.42 $Z = 30$ mm, $d_p = 520$ µm, GAP = 15 mm

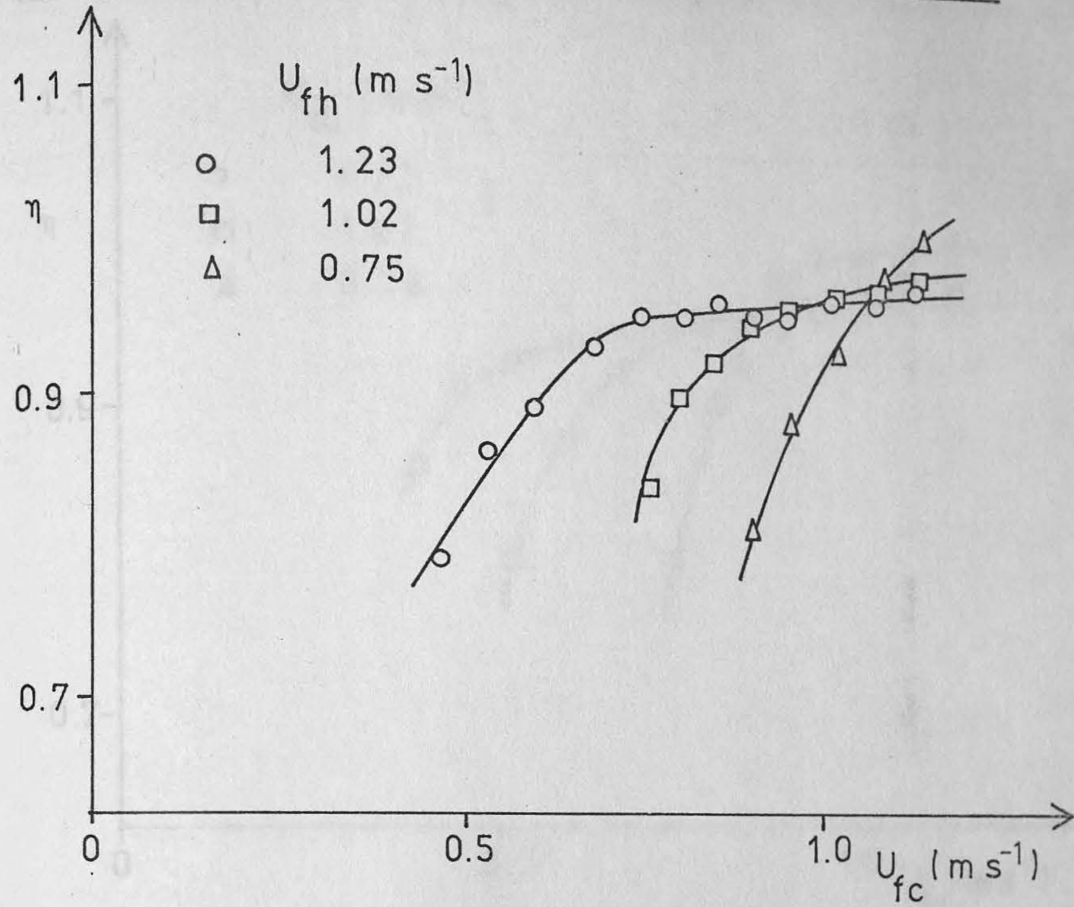


FIGURE 6.43 $Z = 30$ mm, $d_p = 520$ µm, GAP = 20 mm

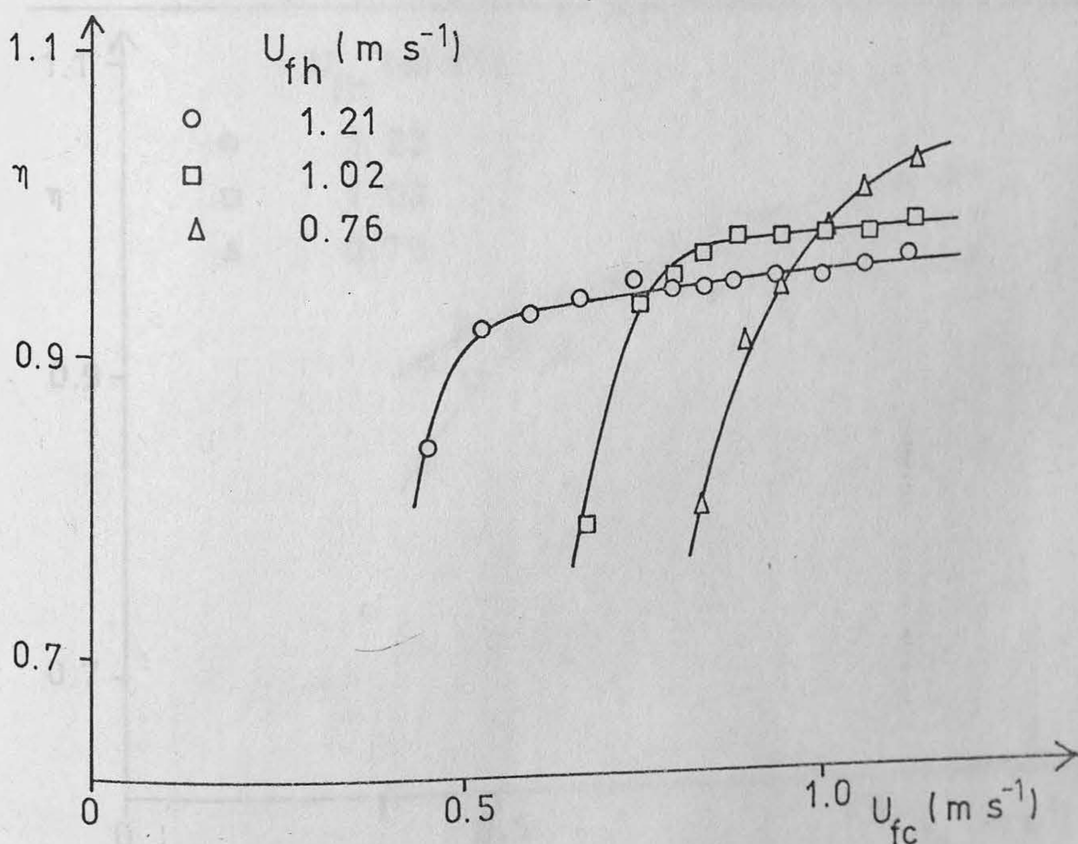


FIGURE 6.44 Z = 30 mm, d_p = 520 μm, GAP = 25 mm

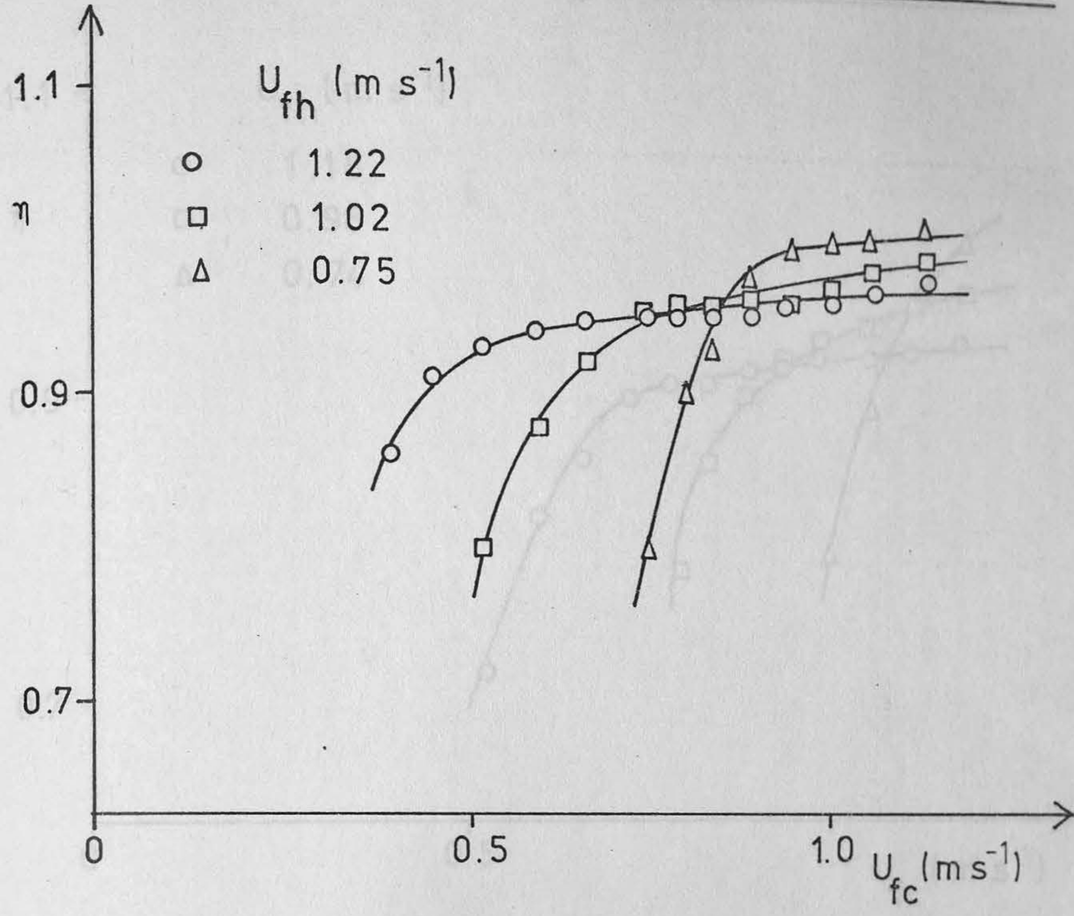


FIGURE 6.45 Z = 30 mm, d_p = 520 μm, NO PARTITION

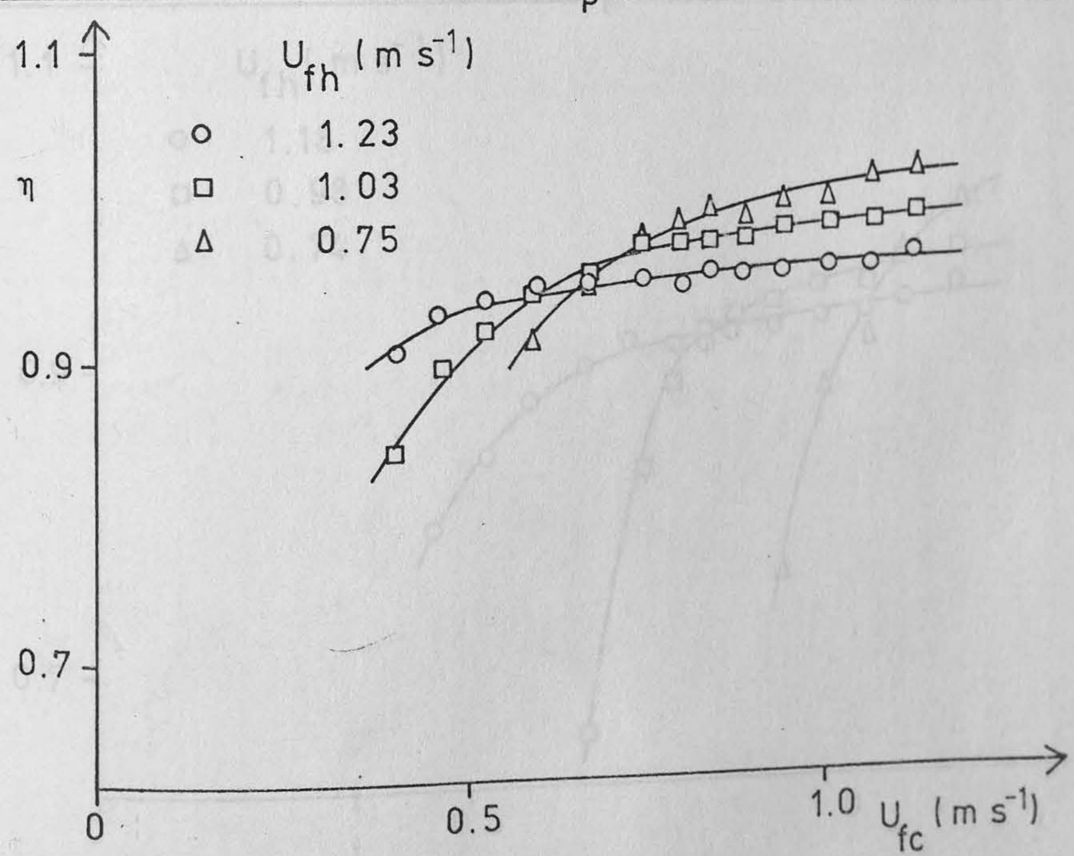


FIGURE 6.46 Z = 40 mm, d_p = 520 μm, GAP = 15 mm

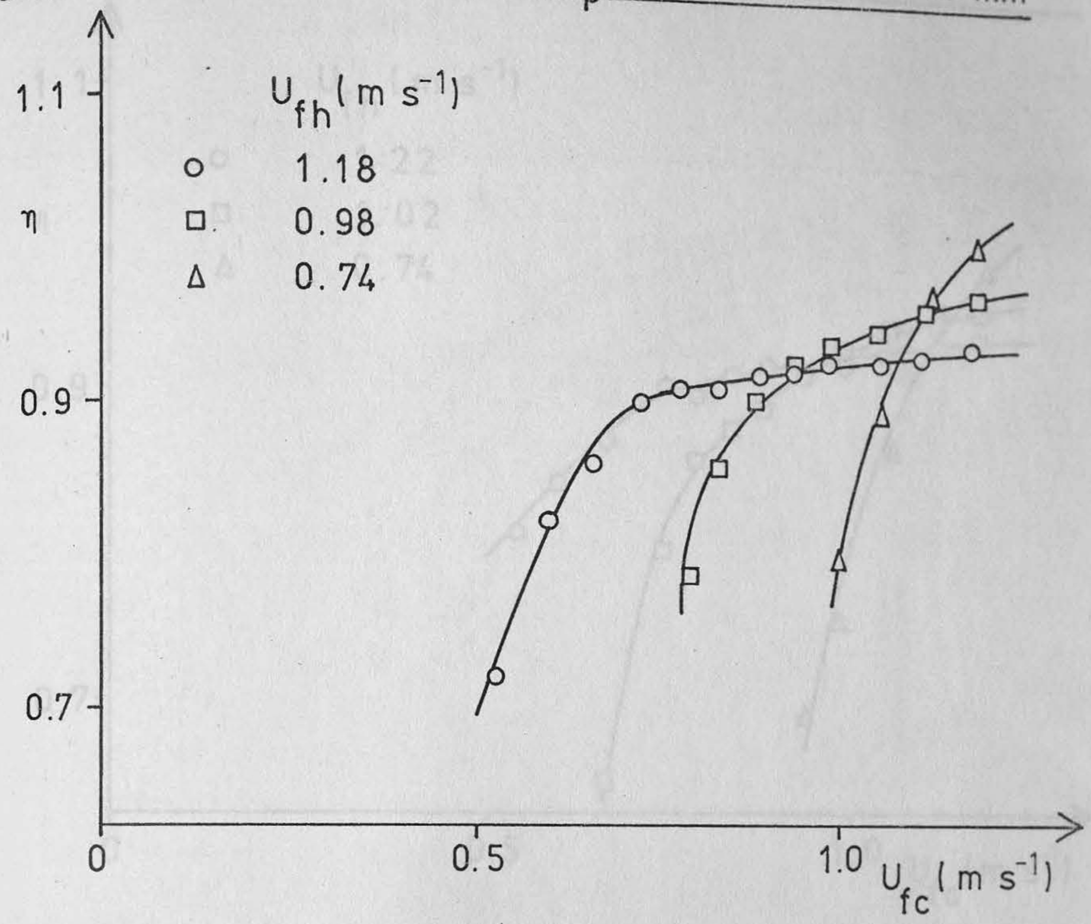


FIGURE 6.47 Z = 40 mm, d_p = 520 μm, GAP = 20 mm

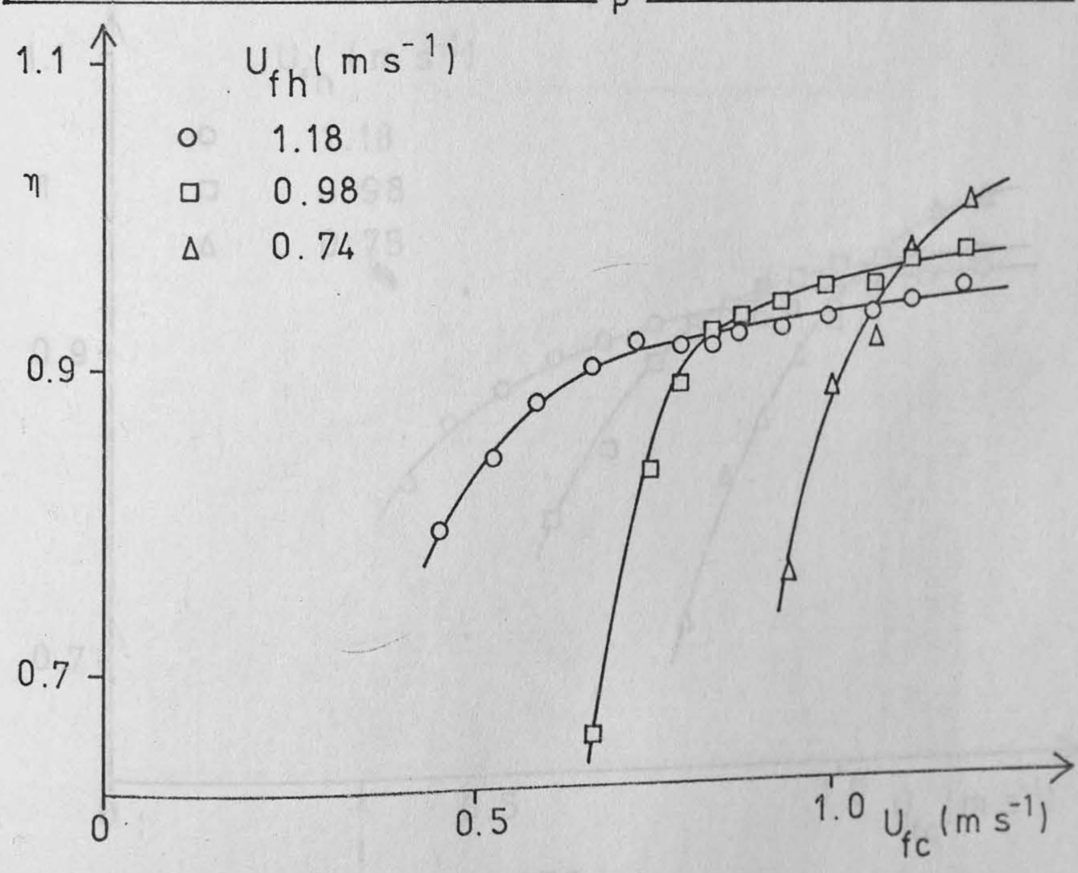


FIGURE 6.48 Z = 50 mm, d_p = 520 μm, GAP = 15 mm

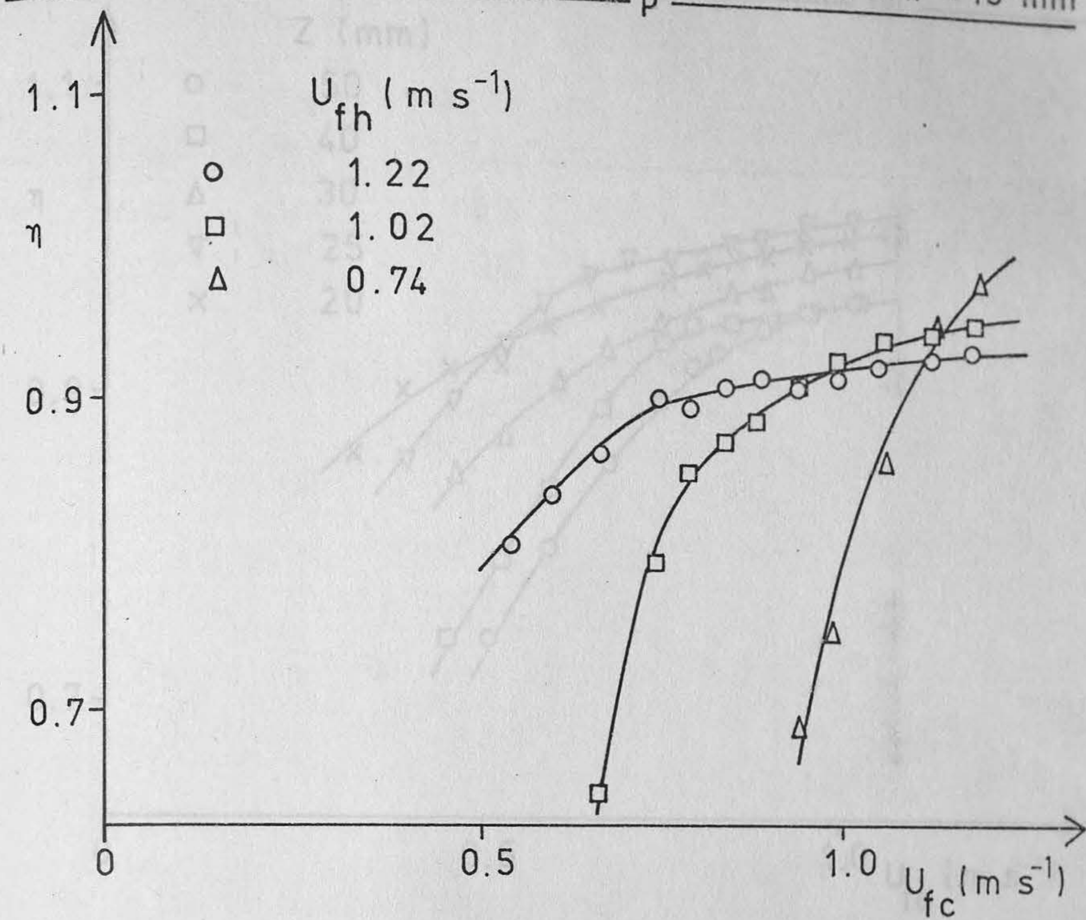


FIGURE 6.49 Z = 50 mm, d_p = 520 μm, GAP = 30 mm

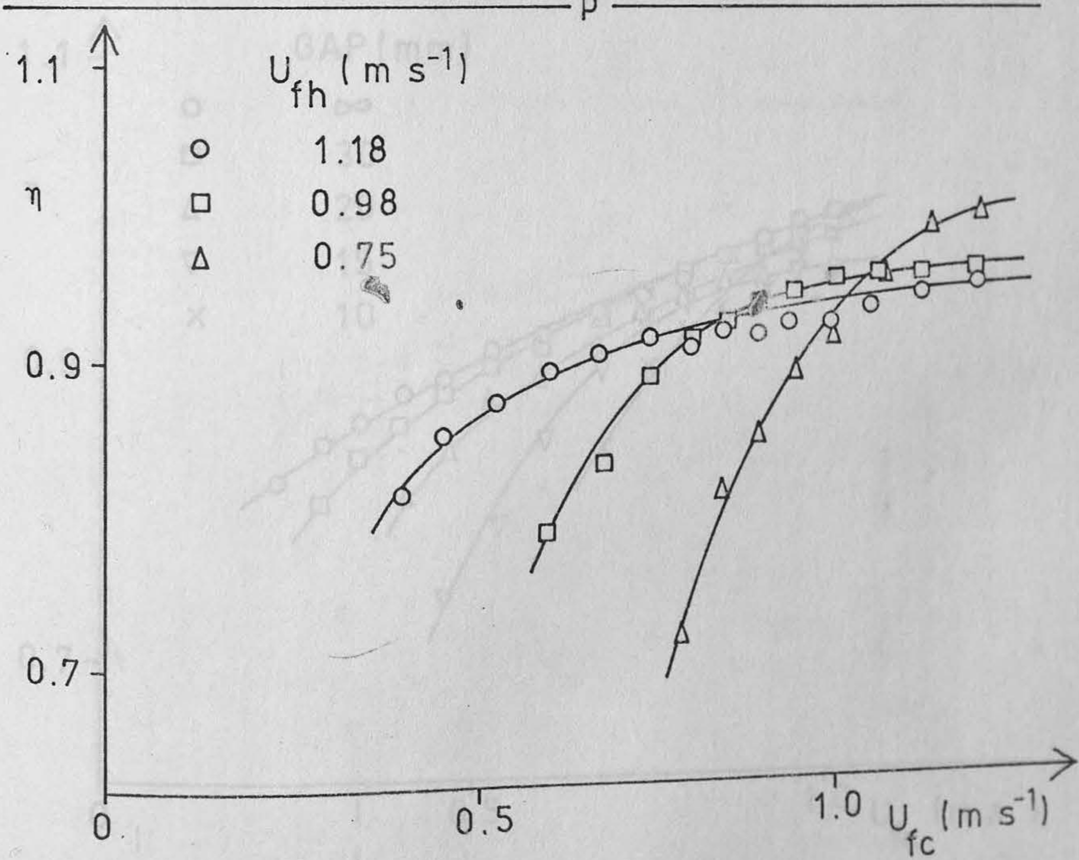


FIGURE 6.50 $d_p = 380\ \mu m$, GAP = 15 mm, $U_{fh} = 1.03\ m\ s^{-1}$

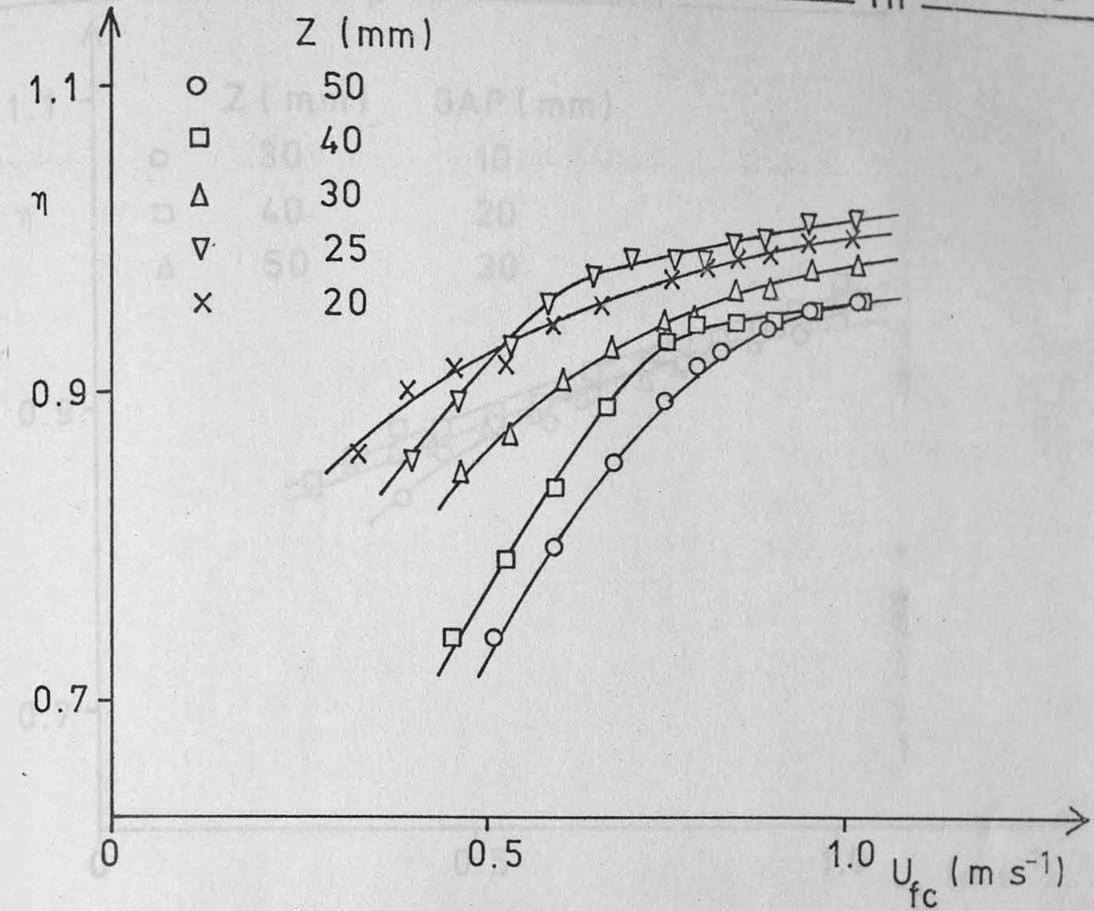


FIGURE 6.51 Z = 40 mm, $d_p = 380\ \mu m$, $U_{fh} = 1.03\ m\ s^{-1}$

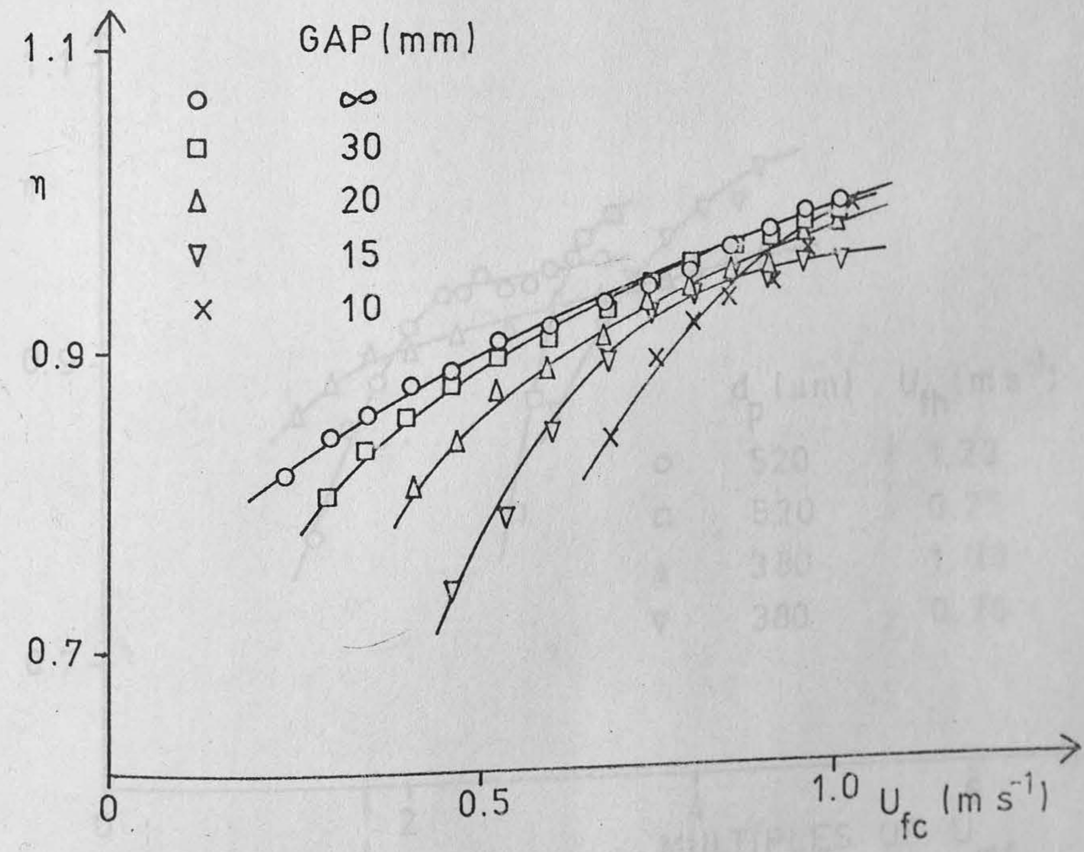


FIGURE 6.52 $d_p = 380\ \mu m$, $U_{fh} = 1.26\ m\ s^{-1}$

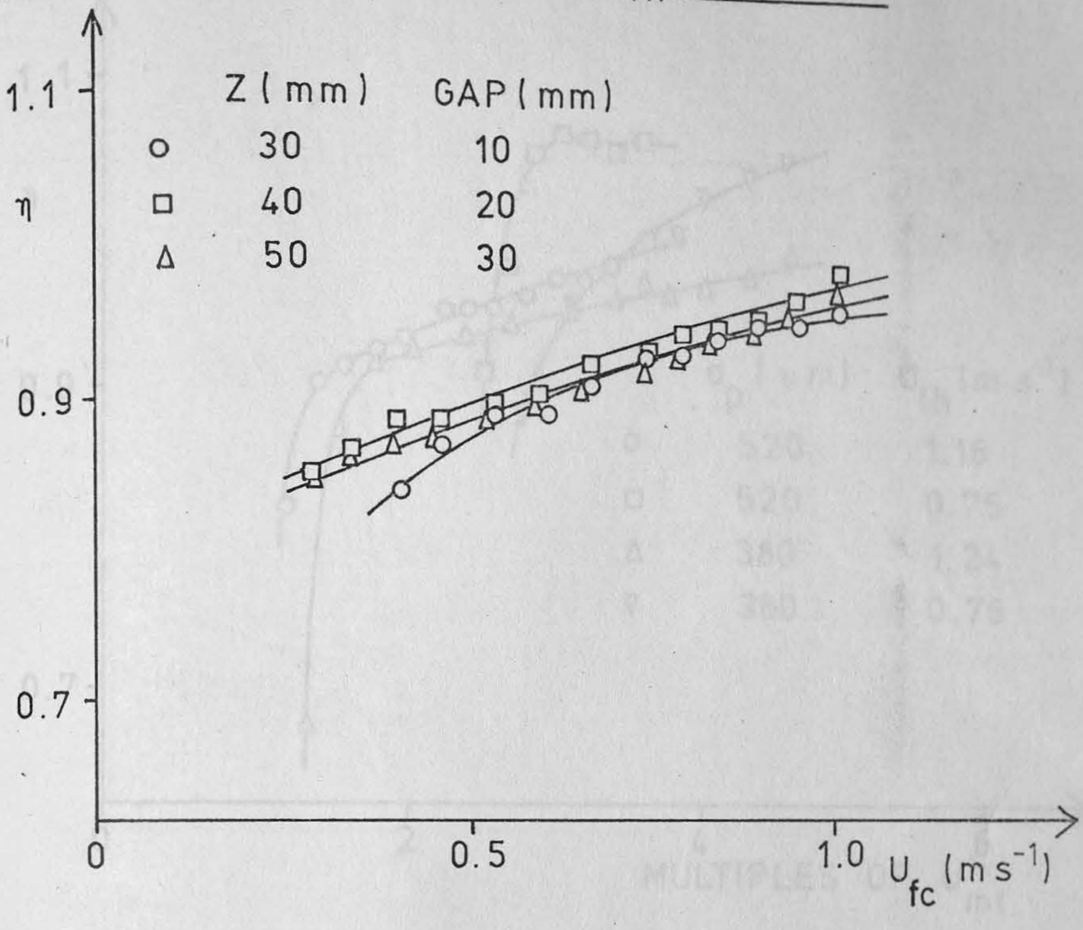


FIGURE 6.53 Z = 30 mm, GAP = 15 mm

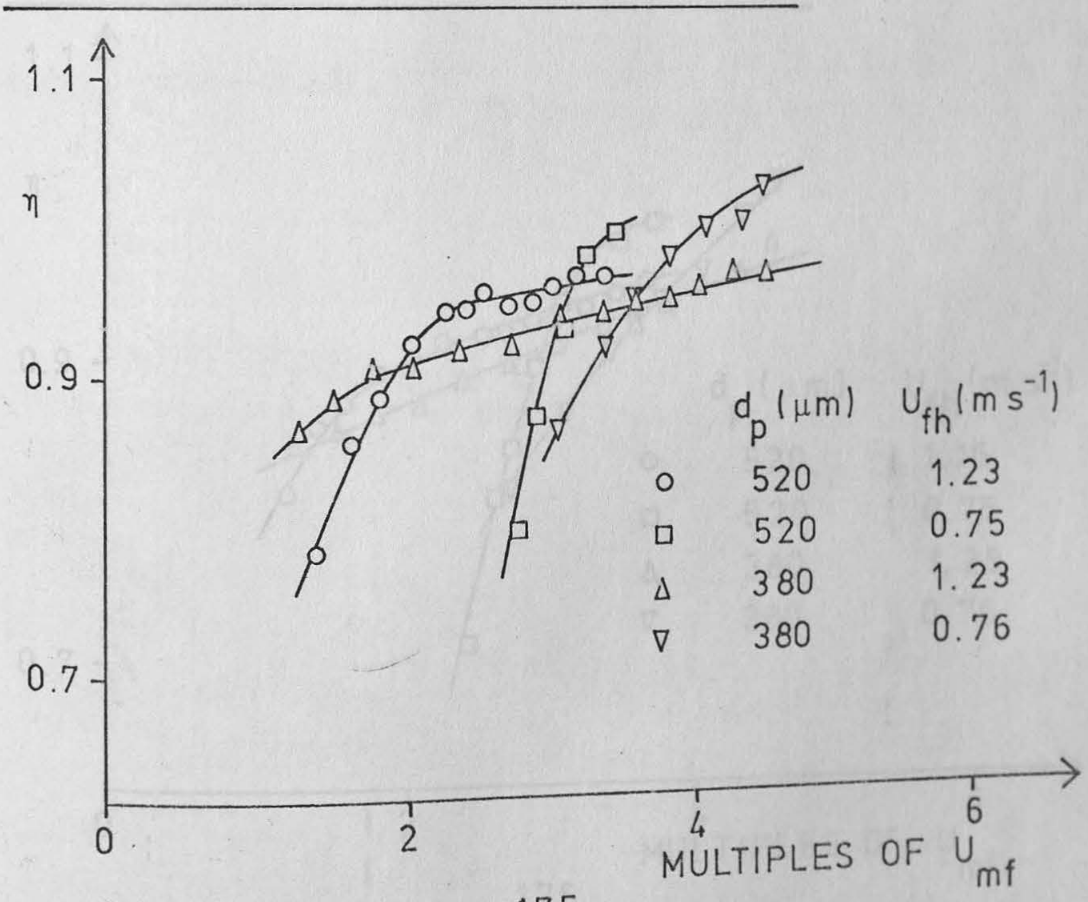


FIGURE 6.54 Z = 15 mm , GAP = 10 mm

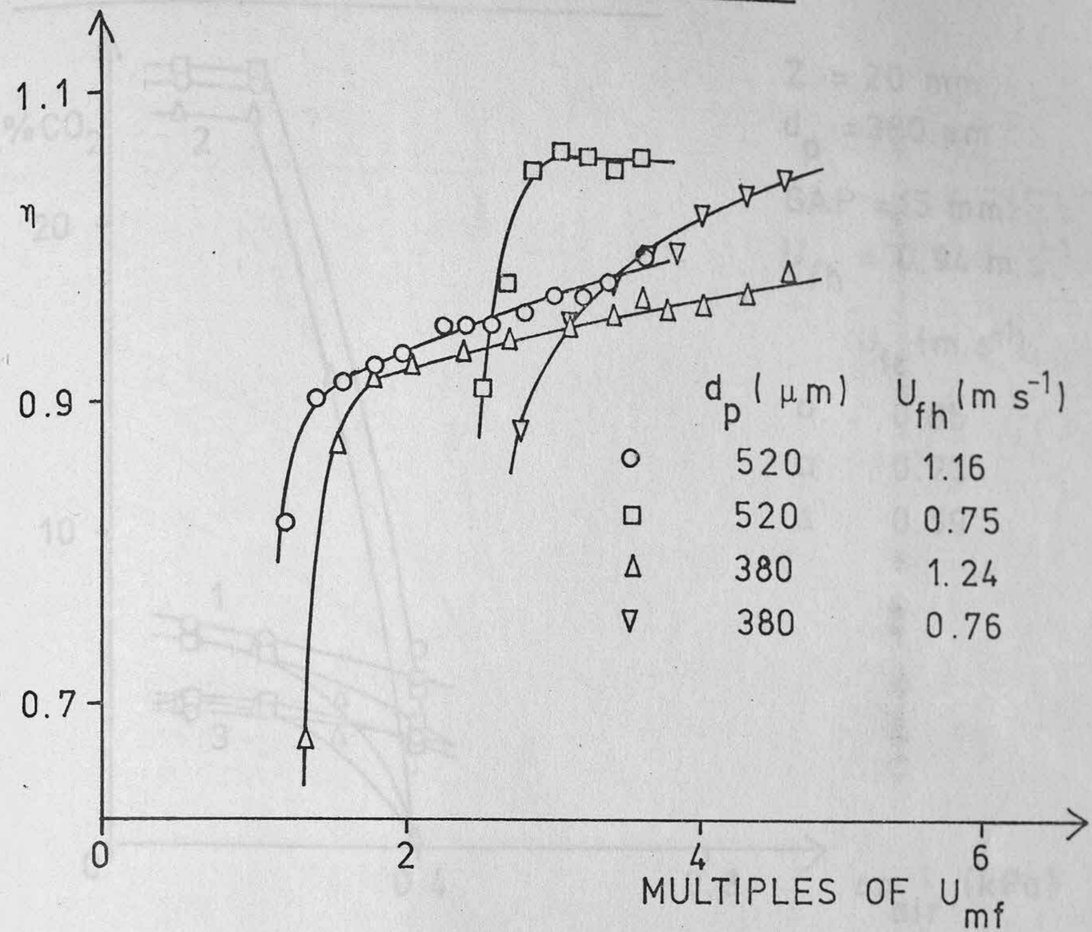


FIGURE 6.55 Z = 50 mm , GAP = 30 mm

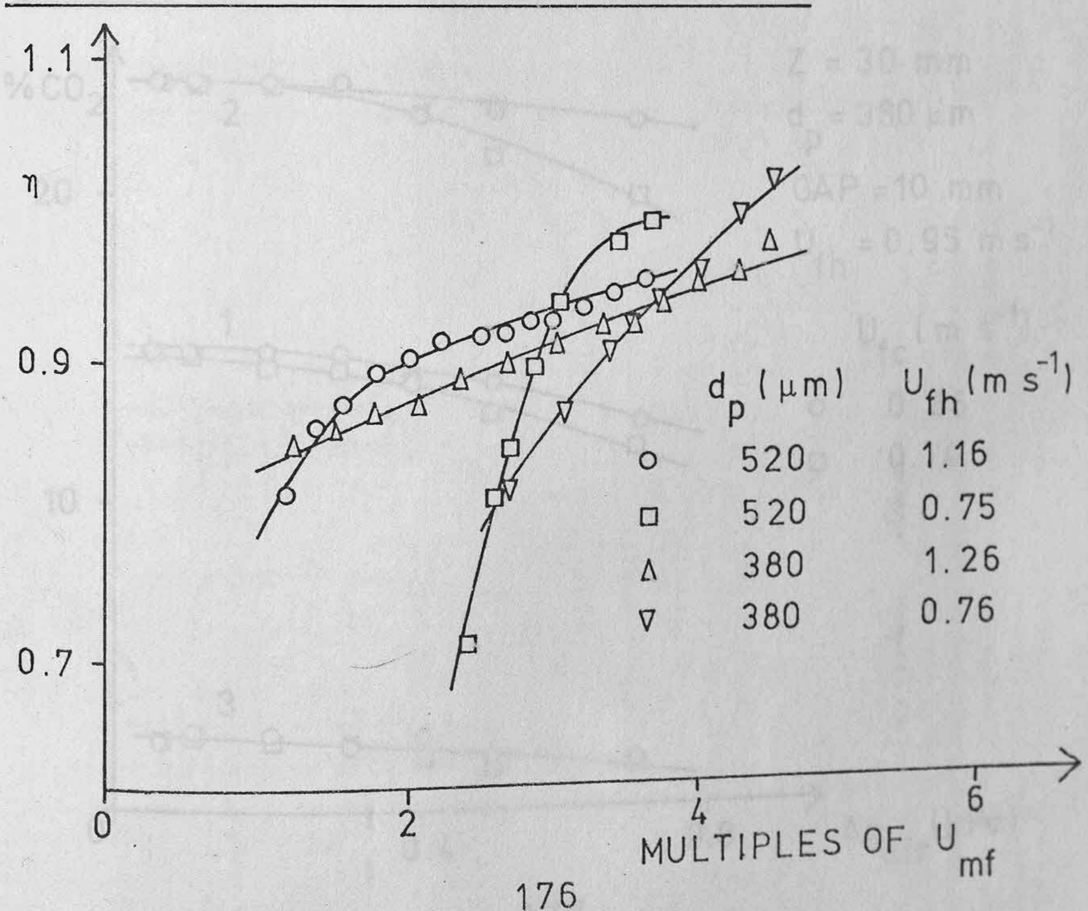


FIGURE 6.56 LEAKAGE DATA

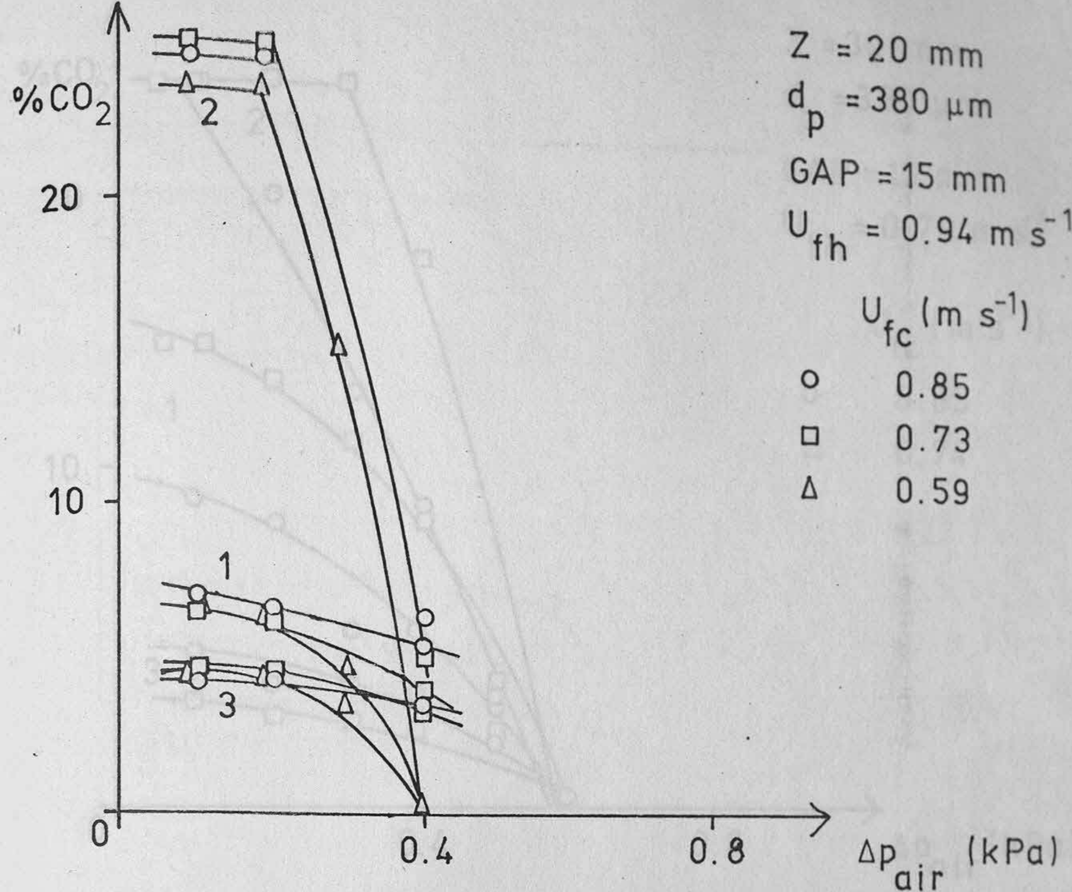


FIGURE 6.57 LEAKAGE DATA

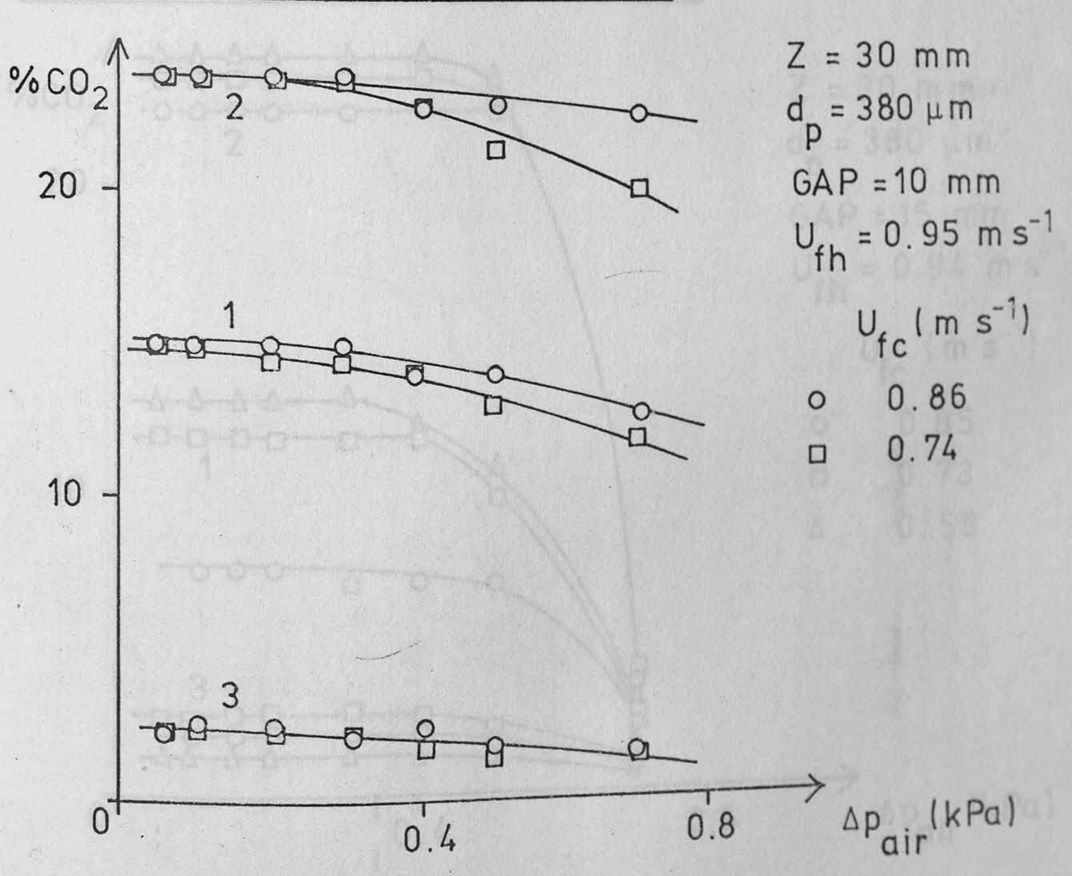


FIGURE 6.58 LEAKAGE DATA

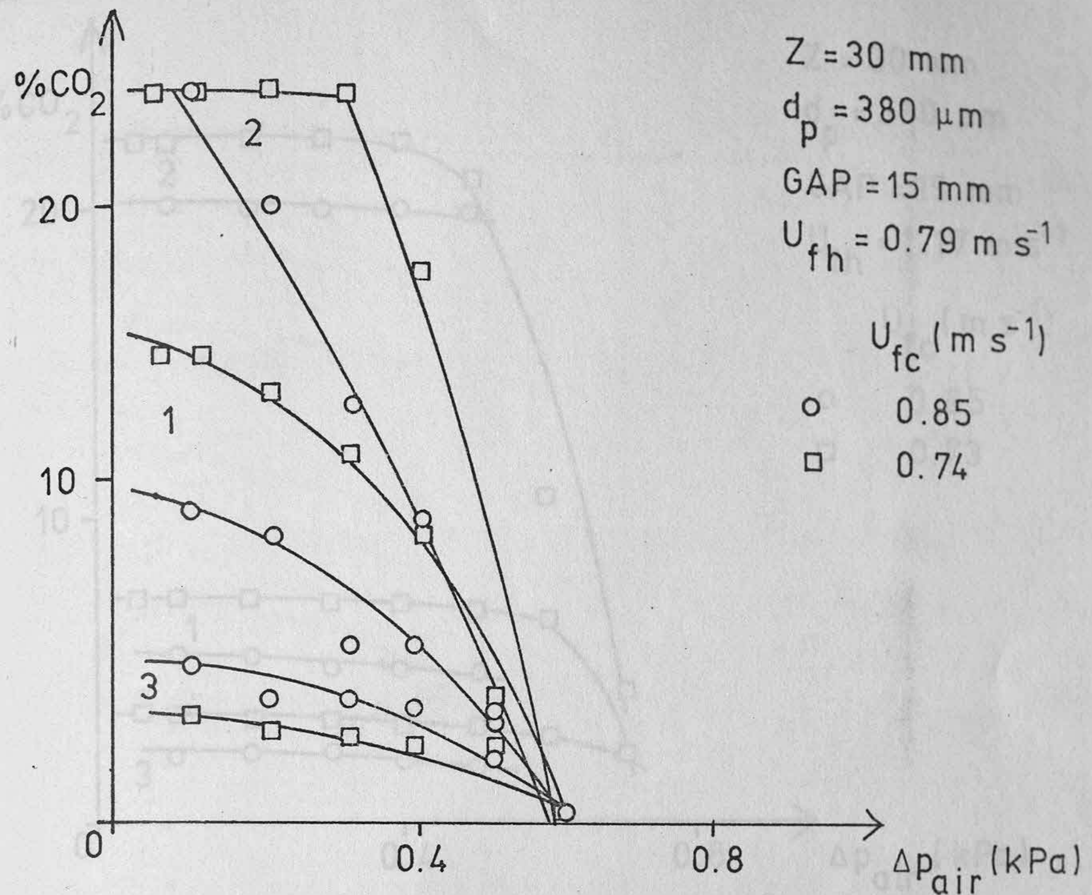


FIGURE 6.59 LEAKAGE DATA

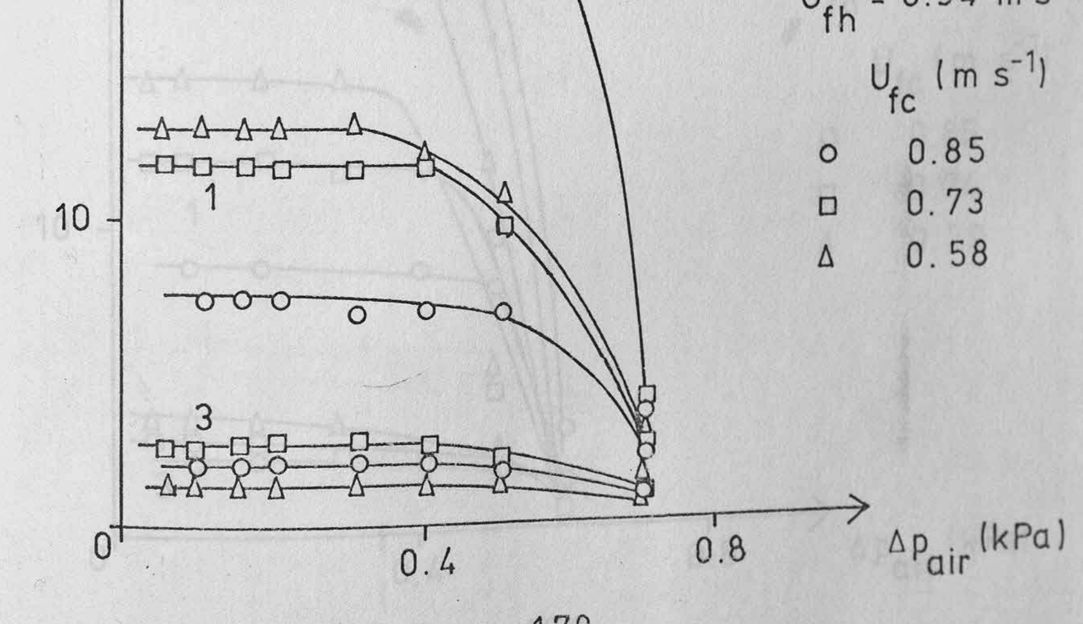


FIGURE 6.60 LEAKAGE DATA

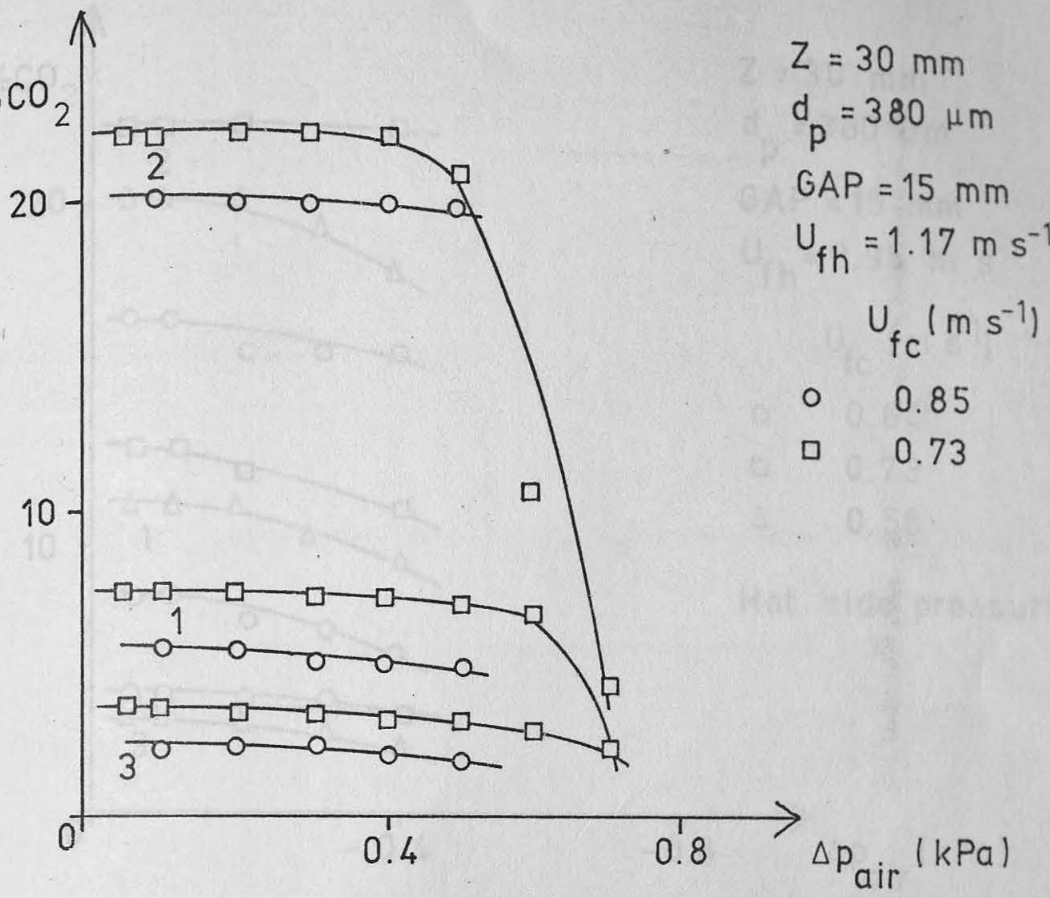


FIGURE 6.61 LEAKAGE DATA

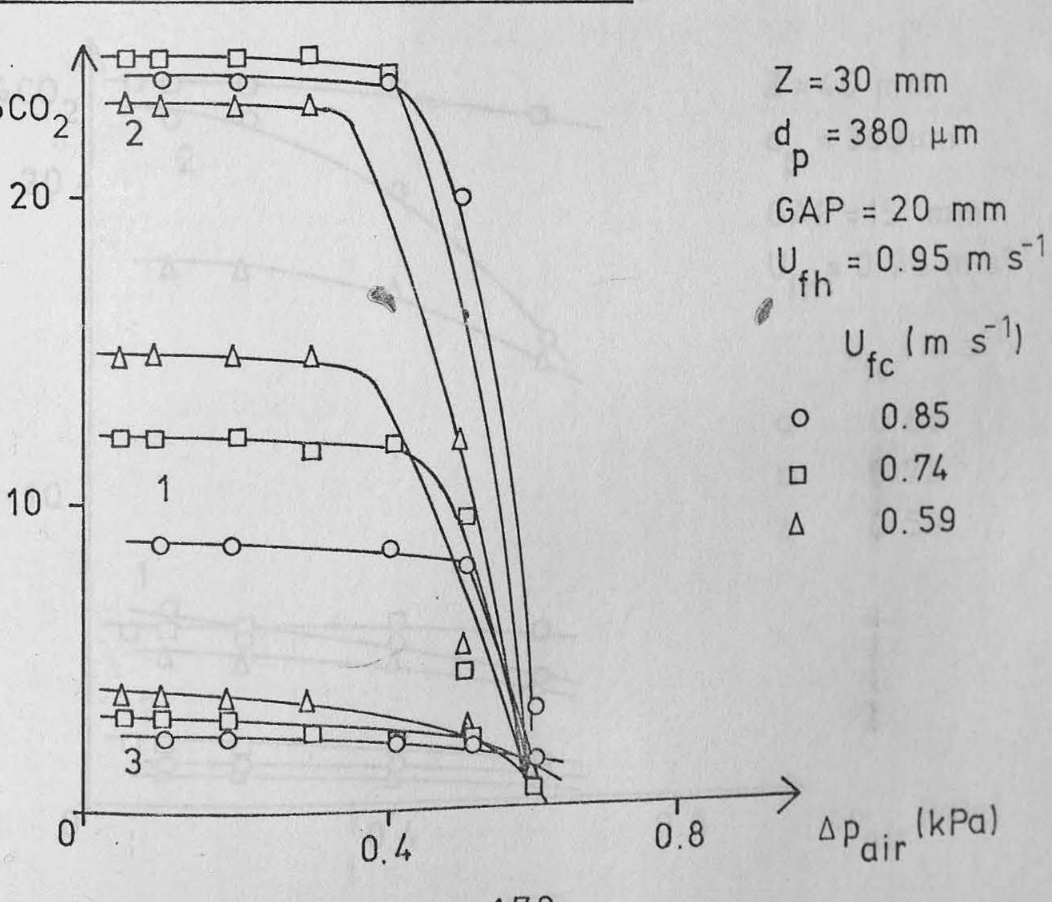


FIGURE 6.62 LEAKAGE DATA

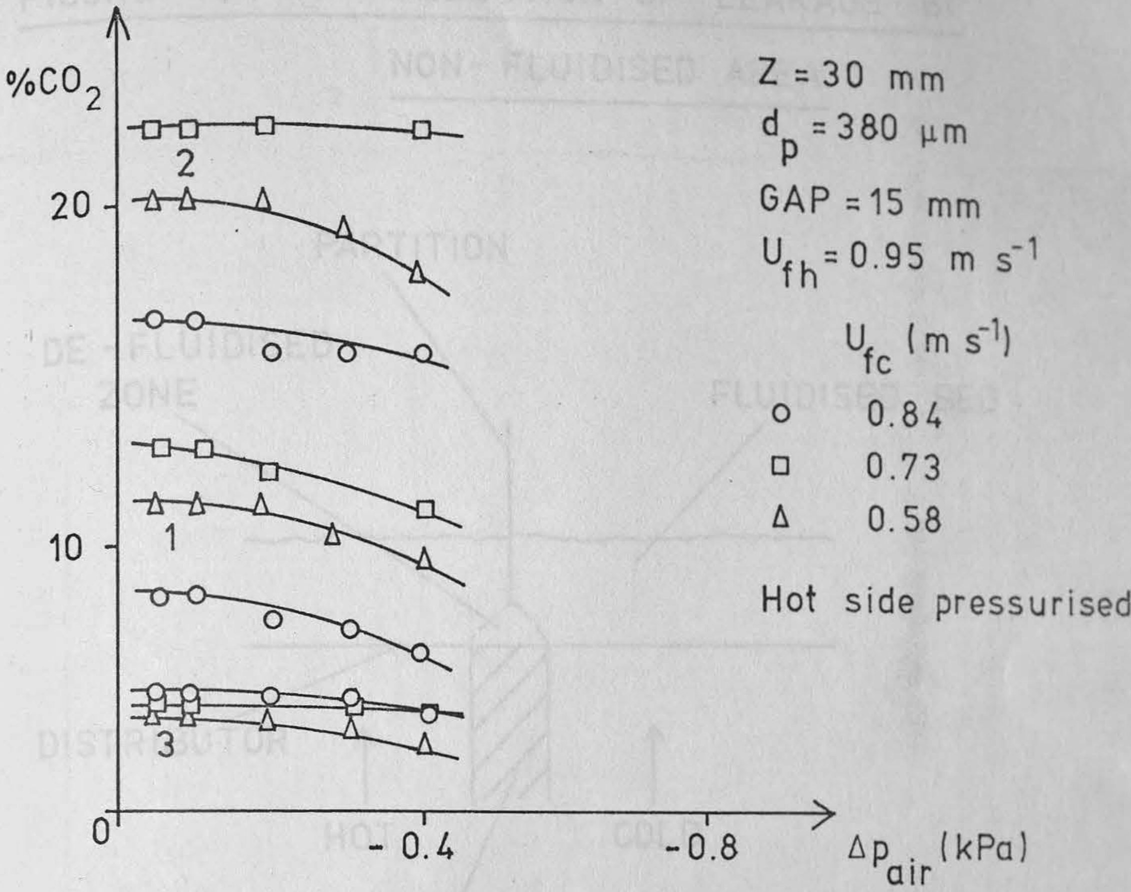


FIGURE 6.63 LEAKAGE DATA

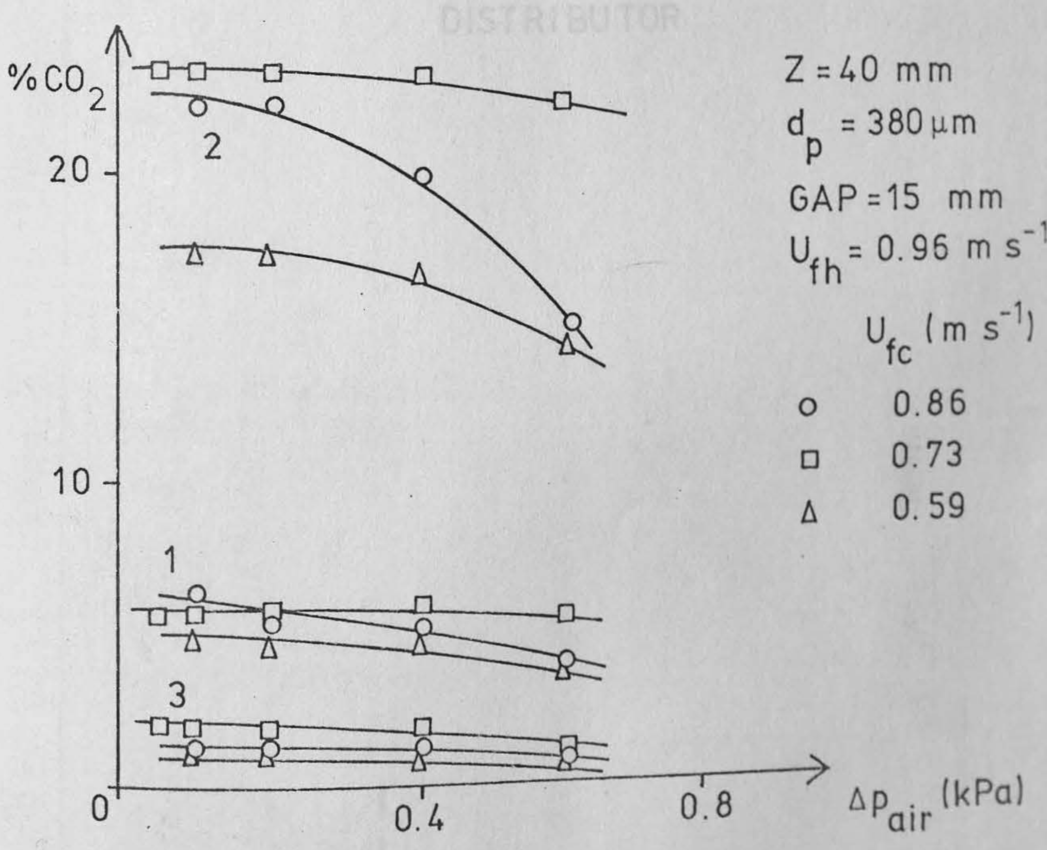


FIGURE 6.64 REDUCTION OF LEAKAGE BY NON-FLUIDISED AREA

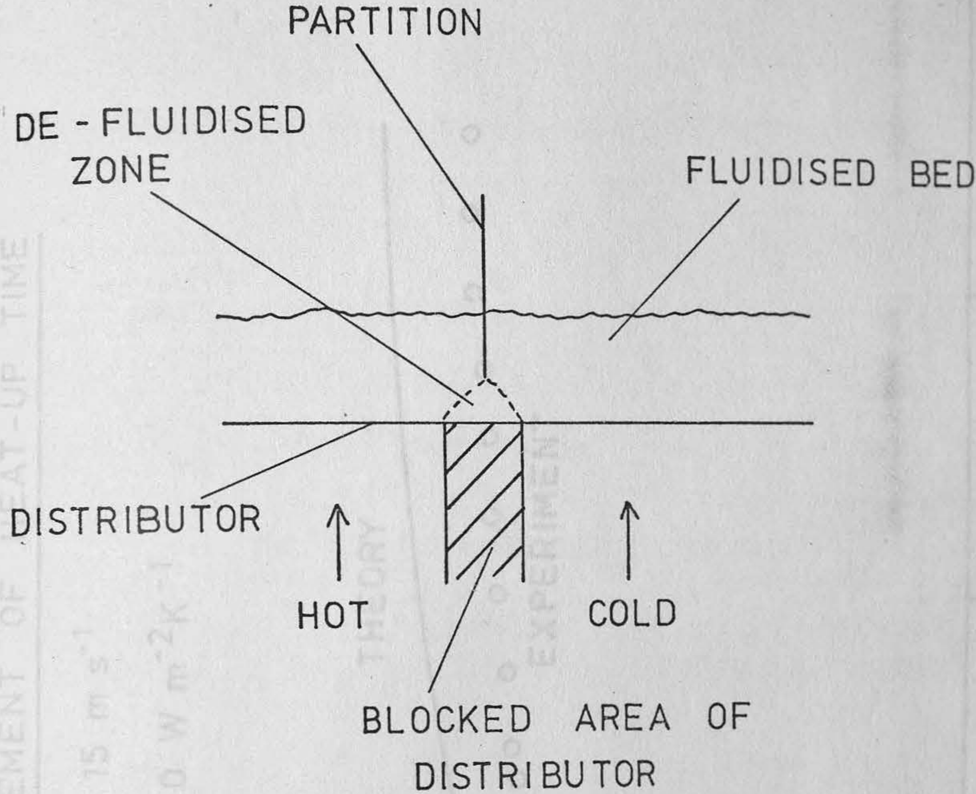
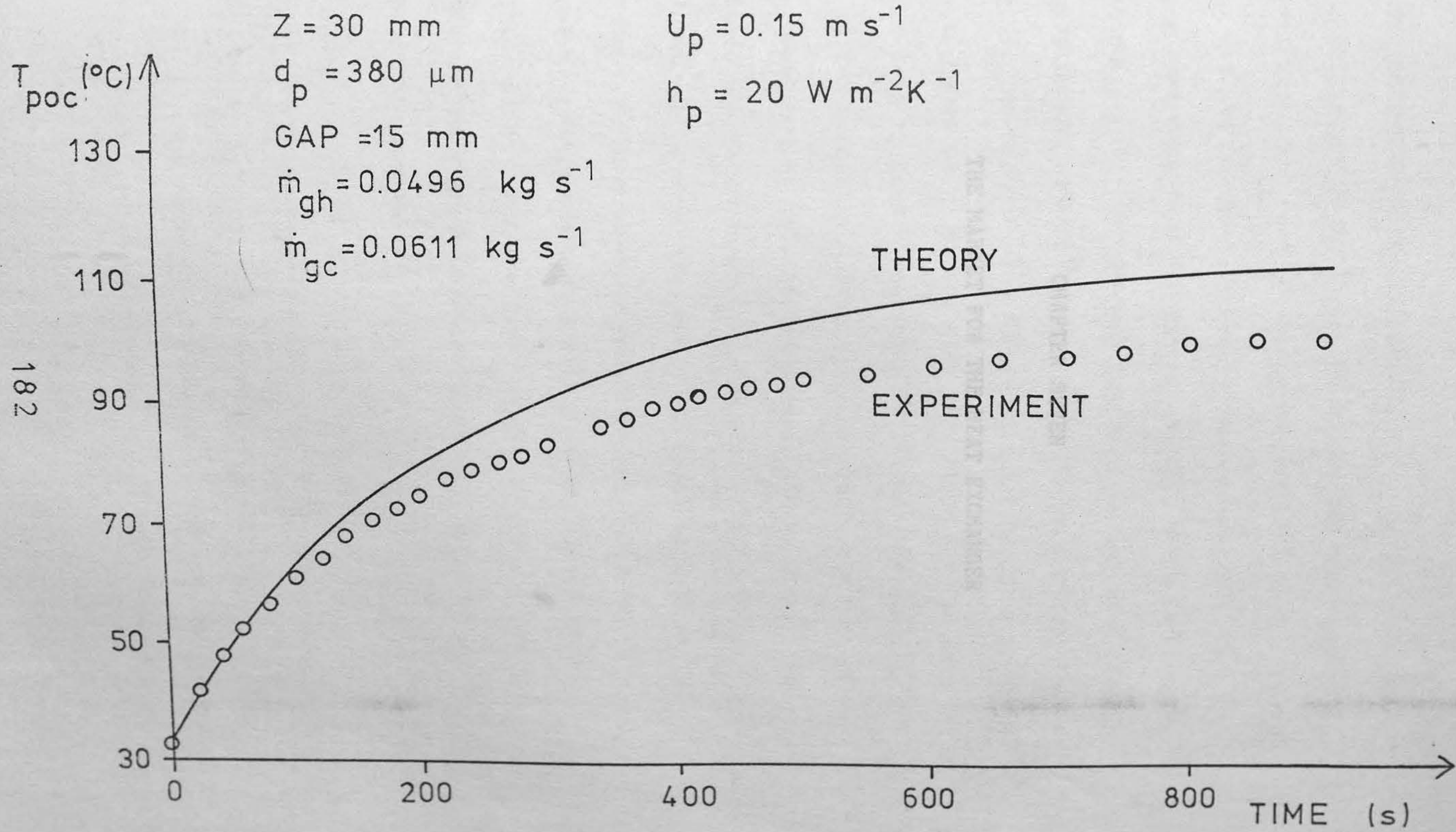


FIGURE 6.65 EXPERIMENTAL MEASUREMENT OF HEAT-UP TIME



CHAPTER SEVEN

THE MARKET FOR THE HEAT EXCHANGER

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THE MARKET FOR THE HEAT EXCHANGER

7.1 THE NEED FOR WASTE HEAT RECOVERY

To demonstrate why it is important to develop new types of heat exchanger, the role of energy conservation in the United Kingdom will be reviewed. Fuel - particularly oil - is becoming more expensive and scarce and this trend is likely to continue for the foreseeable future. During 1978 security of supply became another, more urgent, reason for conservation as the Iranian Revolution suddenly put part of the world's oil supply in jeopardy. At the time of writing the war between Iran and Iraq is also threatening to disrupt the supply. Hence it is increasingly important and profitable for users to conserve fuel by all the means available to them. One particular aspect of energy conservation is the recovery and subsequent use of heat that would otherwise be lost to the environment. This is waste heat recovery (WHR) and it encompasses energy sources such as gaseous, liquid and solid effluents. Air-conditioning exhausts and dirty gas from combustion processes are particularly important potential sources in the context of this thesis.

It is only worth recovering waste heat if it can be used nearby at the time it is available, otherwise the installation is uneconomic. The recovery of waste heat, however, can be of national interest yet be uneconomic as a simple heat recovery process; this point will be discussed further below. The potential value of WHR is none the less enormous because one quarter of British industry's energy consumption is accounted for by losses in warm or hot exhaust gases [103]. This is almost 10% of the nation's energy usage so 8 x 10^5 TJ per annum is lost in this way with a total value of about £400 million assuming 70% is recovered.

Unfortunately there is, at present, no simple way to recover the majority of this energy, and even if it can be recovered, there is often no use for it near the point of recovery. Estimates of how much it is currently feasible to recover are vague. Much can be done by "good housekeeping", for example shutting doors and windows, requiring little or no capital outlay [104] but it is the remainder which concerns us here. An analysis by the Department of Energy [105] suggests the potential recovery of waste heat from manufacturing processes and space heating is 2 x 10^5 TJ per year. The Department also states that "almost all of this (waste heat recovery) is cost-effective at today's energy prices". Although the technology is available, suitable heat exchanger systems are not fully developed at present. Her Majesty's Government and the Commission of the European Communities regard the demonstration of new WHR systems for exhaust gases to be of the utmost importance and are providing funds for this purpose. These are the Demonstration Projects Scheme from the Department of Energy which will have allocated £21.5 million by 1981, and the European Commission's Energy Research and Development Programme for Energy Conservation which runs until 1985.

Given the premise that the recovery of this energy is economic, it must be asked why such a quantity is not conserved. For most industries except steel, cement and petrochemicals, energy forms only a small proportion of total product cost, perhaps 2 to 3 per cent at most [105]. So energy saving is unlikely to reduce the cost of a product significantly and hence is not thought worthwhile. Therefore although energy conservation may be vital to the national interest to maintain a secure fuel supply, it is a much less important factor to individual companies. In an effort to counteract this problem the Department of Industry offers grants to companies to assist them in taking energy conservation measures [106].

Thus national policy is to support both the development and installation of WHR systems. To find out if the fluidised bed gas-to-gas heat exchanger will be successful, its performance must be compared with that which the customer requires and that of existing gas-to-gas heat recovery systems. Adequate market research is essential to identify the most likely market and match the new product to that market [107]. If an appropriate WHR system can be produced at the right price, then present Government policy and financial incentives will improve its chances of being implemented.

7.2 CUSTOMER REQUIREMENTS FOR WASTE HEAT RECOVERY

The first and foremost reason for any company to install WHR equipment is to improve the rate of return on capital for the process as a whole. To establish if an investment in WHR is sound, the company must identify the places where energy is lost and quantify the losses, and the techniques for doing this are often called energy accounting [108]. This requires specialist advice and the Department of Industry set up the Industrial Energy Thrift Scheme to assist companies in the manufacturing sector [109, 110]. The Scheme also provides information on the overall energy usage in the economy and aids the development of a coherent European energy policy [111]. One of the results of these studies has been to show that much energy is lost in hot flue gases and that the technology needed to recover the energy is already known but not proven [112].

After showing that energy is available for recovery, companies then define a required rate of return on their investment in WHR. Typically a WHR installation will be expected to give a rate of return about twice that of new production equipment, or at least 30% per annum [113]. This is because there is a belief that "money saved is worth less than money made". The demonstration of a sufficiently high rate of return is not easy [114, 115], and often rests on the predictions of future fuel prices, and of the availability and magnitude of Government assistance grants. These are subject to uncertainties because they are controlled essentially by political decree or influence.

This is particularly true in the United States where home-produced oil prices are restrained artificially below international levels.

Once a company considers that investment in WHR is an attractive proposition, several questions relating to the actual equipment will appear. Often the installation will be on existing plant so it must be simple and quick to install with a minimum of disruption to equipment and production. The system must be reliable, especially if it is fitted into a continuous process where the failure of one segment leads to a total shut-down. Similarly maintenance should be simple with long intervals between servicing. Where dirty gases are involved the fouling of the heat exchanger must be minimised and any necessary cleaning automatic.

A less important consideration for a customer will be the magnitude of the pressure drop across the heat exchanger. For a given duty there is a compromise to be made between the pressure drop, which determines the pumping power and running costs, and the size which fixes the initial cost [116]. Also a high pressure drop can cause problems with installations in existing systems where the previous pressure balances may be disturbed with adverse consequences.

Hence it can be seen that if a customer establishes that WHR is economic (difficult, given current attitudes and fuel prices) then he will demand a guarantee of reliability. The question of reliability is central to the marketing of any waste heat exchanger, so it will have to be demonstrated during the development of the new heat exchanger prior to commercial production. These problems apply to any WHR system and are not peculiar to gas-to-gas waste heat exchangers such as the fluidised bed system described in this thesis.

7.3 APPLICATIONS OF GAS-TO-GAS WASTE HEAT EXCHANGERS

It has already been shown in the previous section that much energy is lost in warm or hot exhaust gases, and it is often desirable to transfer this heat to another gas stream. There are broadly three categories of gas-to-gas WHR, namely air-conditioning, space heating from hot exhaust, and combustion air pre-heating. These will be considered in turn by drawing largely on the works of Reay [117] and Boyen [118] which are amongst the best in the whole field of industrial energy conservation.

7.3.1 Air-conditioning

In the ventilation of buildings warm, humid air is discharged and fresh air is introduced. The incoming air needs to be heated or cooled, depending upon the climate, and it is advantageous to interchange some of the energy between the inlet and the exhaust air streams. The energy content of the moist exhaust air is apportioned about equally between the enthalpy of the air and the enthalpy of evaporation of the water content. Therefore any heat recovery system for this application must condense some of the water vapour to recover its enthalpy of evaporation and has to operate with a small temperature difference between the air streams. Also it should be reversible so that in areas with large climatic variations it can conserve cold (that is maintain a lower temperature than the surroundings) in summer and heat in winter. It should be noted that the recovery of the enthalpy of evaporation of the moisture inevitably entails condensation and even freezing so the system must be designed to continue operating reliably when these phenomena occur.

The rate of heat transfer must be controllable as lights, people and equipment all provide unsteady heat inputs to buildings, yet the temperature of the air within the building has to be held reasonably constant. Some heat recovery systems offer an inherent leakage path between the two air streams. In air-conditioning, however, this is not normally important as the exhaust air is neither dirty nor toxic and leakage only affects the heat exchanger effectiveness. In special cases such as hospitals and clean rooms leakage could not be tolerated, but these form only a small minority of air-conditioned buildings.

7.3.2 Space heating from hot exhaust

In many factories heat from various processes is carried away in flue gases which are often hotter than necessary for the prevention of condensation in the stack. It may be desirable to recover some of this heat and use it for the space heating of the factory and offices. This application leads to different requirements for the heat recovery equipment compared with air-conditioning. Primarily the leakage must be minimal, although the tolerable limit depends upon the constitution of the hot flue gas. The exhaust may be from a combustion process and so contain CO, CO_2 and NO_X, or be contaminated with vapours from solvents. If the concentration of a contaminant in the exhaust is known, then a maximum allowable leakage can be calculated as the acceptable concentrations in breathing air are documented [89] and should be referred to when specifying a design. Such toxic components must be excluded as well as possible from the incoming fresh air in the heat exchanger. Unless the levels of these toxic products in the heated fresh air are satisfactory, that particular piece of WHR equipment will be unacceptable to a prospective customer.

There are several other requirements for this application and these are as follows:-

(i) control will be necessary to regulate the temperature of the space heating air and this is accomplished most easily by diluting the heated air with further fresh air;

(ii) the exhaust may be contaminated with dust and liquid droplets, so any heat exchanger surfaces exposed to the exhaust will be liable to foul. Thus the system should have its susceptibility to fouling reduced and have a simple means of cleaning if necessary;

and

(iii) the heat exchanger must be made resistant to corrosion by the exhaust, and at combustion exhaust temperatures normally encountered this implies almost inevitably the use of stainless steel.

The recovery of waste heat from hot exhaust for space heating often entails the transport of energy since the points of supply and demand may be distant, perhaps some hundreds of metres. In these cases the heat is transferred to a thermal fluid such as pressurised hot water or water/glycol mixtures which in turn gives up its energy to the space heating air. These systems are sometimes called "runaround coils" though they are not single entities but two connected gas-to-liquid heat exchangers which are some distance apart. They are not considered further as they are not direct competitors of the new fluidised bed unit, though such systems may be the most appropriate for some installations.

7.3.3 Combustion air pre-heating

The best application of WHR is where the recovered heat can be returned directly to the process creating the hot exhaust. The heat is then always needed when it is available, and source and demand are adjacent. This situation arises where the enthalpy of combustion products is used to heat the incoming air prior to combustion. The overall process efficiency is then increased and fuel is conserved.

This application leads to slightly different requirements for the heat exchanger as compared with space heating from hot exhaust. The provisions for preventing fouling and corrosion still apply, but some leakage is tolerable as the system is closed. Control will be needed to keep the pre-heated combustion air at an appropriate temperature for the burner. At present gas burners cannot accept air hotter than about 400 °C and somewhat higher for oil burners, though the development of suitable high temperature heat exchangers may also stimulate work to increase this.

7.4 EXISTING TYPES OF GAS-TO-GAS HEAT EXCHANGERS

There are four existing types of gas-to-gas heat exchangers all of which can be used in one or more of the WHR applications outlined above [117]. These are the heat pump, the rotary regenerator (sometimes called the heat wheel), the recuperator and the heat pipe heat exchanger. These will be described in turn with particular reference to their suitability for various applications.

7.4.1 Heat pump

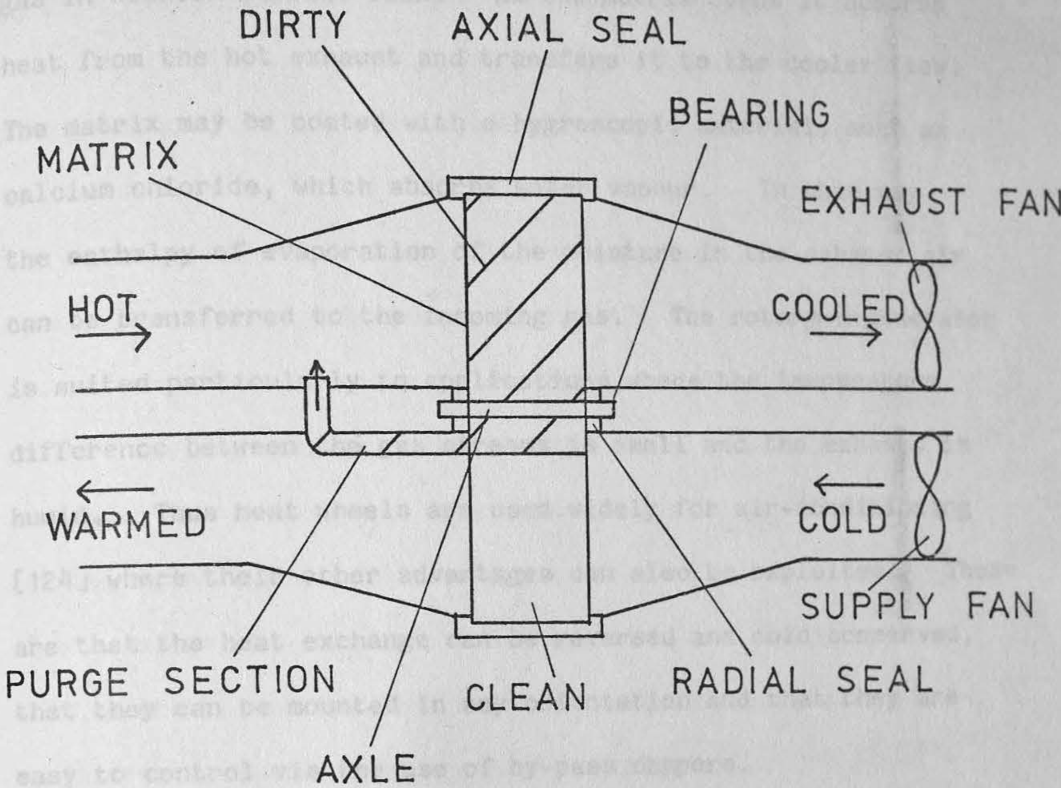
The heat pump is essentially a device which enables heat to be upgraded in temperature and transferred to wherever it may be needed. The heating cycle can deliver several times more heat than the electrical energy needed to drive the compressor of the working fluid. The development of the heat pump has been chronicled by Fearon [119] and its use in WHR is described by Trenkowitz [120]. It is only suitable for air-conditioning applications because currently the upper delivery temperature is about 60 ^{o}C, the heat being removed from the warm exhaust air.

Much of the present work is being carried out by the Building Research Establishment [121] who are developing domestic air-to-air heat pumps with power outputs up to about 15 kW [122]. The potential of the heat pump has only recently been considered seriously, and doubtless this will stimulate further work.

7.4.2 Rotary regenerator

The rotary regenerator is the best-selling type of gas-to-gas heat exchanger [123] in terms of money, but some of the units are very large so this type of system does not form the majority of devices delivered. The principle is explained with reference to Figure 7.1 which shows a schematic diagram of a rotary regenerator. The matrix is rotated by a small

FIGURE 7.1 ROTARY REGENERATOR



electric motor with a chain drive and the regenerator spans two ducts, one carrying the exhaust and the other the supply gas in counter-current flow. As the matrix turns it absorbs heat from the hot exhaust and transfers it to the cooler flow. The matrix may be coated with a hygroscopic material, such as calcium chloride, which absorbs water vapour. In this way the enthalpy of evaporation of the moisture in the exhaust air can be transferred to the incoming gas. The rotary regenerator is suited particularly to applications where the temperature difference between the gas streams is small and the exhaust is humid. Thus heat wheels are used widely for air-conditioning [124] where their other advantages can also be exploited. These are that the heat exchange can be reversed and cold conserved, that they can be mounted in any orientation and that they are easy to control via the use of by-pass dampers.

There are, however, two severe disadvantages of the rotary regenerator, these being fouling of the matrix and leakage between the gas streams through the moving seals. The static pressure difference between the two gas streams implies that leakage can occur, though the fan arrangement shown in Figure 7.1 means that this is from clean to dirty and not vice versa. Many manufacturers use a purge section to flush the matrix with clean gas which is discharged back into the dirty exhaust. This greatly improves the effective performance of the radial seals, but the axial seals at the periphery are not aided. MacDuff and Clark [125] discuss the problem at length and emphasise the deleterious effects that thermal cycling has on the seals at high temperatures. At air-conditioning temperatures the seals operate adequately, but above 400 ^{o}C the difficulties of keeping the leakage within specification over long periods are severe and not satisfactorily resolved.

The second problem, fouling, is also exacerbated at higher temperatures [126, 127] because corrosion rates are greater and places where corrosion has occurred are more liable to foul. Fouling increases the pressure drop through the heat exchanger and reduces its efficiency. The nature of the matrix, which may be steel wool or natural fibre, makes it almost impossible to clean. In large units cleaning and maintenance has to take the form of replacing segments of the matrix. These very large units accept up to 200 $m^3 s^{-1}$ of gas in each stream and are used largely in power stations for pre-heating combustion air. It is not anticipated that the new fluidised bed system will ever compete in this part of the market because of the size.

There are many manufacturers who make smaller heat wheels of all types for gas flows up to about 40 $m^3 s^{-1}$ and these will be direct competitors of the fluidised bed unit. Table 7.1 lists those companies whose products are available in the United Kingdom and their product ranges. All the heat wheels have quite low pressure drops in the range 100 to 300 Pa, operate with gas approach velocities of between 1 and 7 $m s^{-1}$

Table 7.1 Manufacturers of rotary regenerators

Manufacturer	Gas flow (m^3s^{-1})	Operating Temperature (^oC)
Acoustics and Environmetrics Ltd., Claygate, Surrey	0.5 - 23	<200
Bahco Ltd., Banbury	0.5 - 23	<100
Curwen and Newbery Ltd.,	0.16 - 40	<100
Westbury, Wiltshire	0.16 - 40	<205
	0.16 - 40	<425
	0.16 - 40	<980
Fläkt Ltd., Staines	1 - 15	<80
Wing Co. (Armca Specialities	30	<50
Co. Ltd., Ilford, Essex)	30	<150
	0.2 - 20	<400
	0.2 - 8.5	<815

and require pumping powers of up to 2 kW per unit face area. A typical rotation rate is 10 min^{-1}, and the electric motor for turning the wheel is unlikely to consume more than 500 W so the running costs are low. The efficiencies of heat exchange range up to 85% though these depend upon the ratio of the two mass flows. Theoretical attempts to predict the percentage heat recovery are most complicated [128] and are of little practical value. Instead manufacturers rely on previous operating experience to predict the performance of heat wheels.

According to the manufacturers' specifications leakage can be reduced to 0.1% by volume of the exhaust flow if a purge section is used in units operating at temperatures below 200 °C. However at higher temperatures the purge sections are usually fan-assisted rather than relying solely on the static pressure difference between the gas flows. In this case the leakage is about 1%, although the purge flow can be up to 10% of the exhaust flow so there is a consequent loss in efficiency. Heat wheels are generally constructed so that they can pass 0.5 mm particles without fouling. The matrix can be made specially to cope with 1 mm particles if necessary, but the efficiency is reduced because of the smaller heat transfer area.

Overall rotary regenerators can be used successfully in air-conditioning where the leakage is not a problem. At higher temperatures where the one gas stream is from a combustion process the thermal cycling of the wheel can lead to the breakdown of the moving seals and subsequent leakage. This means that the heat wheel is not suitable for space heating from hot exhaust applications. In the pre-heating of combustion air leakage is much less of a problem, but the heat wheel is often susceptible to fouling which militates against its use in this application also.

7.4.3 Recuperator

The recuperator consists of a framework containing an arrangement of plates or vanes which separate the hot and cold gas streams. The vanes form channels about 2 mm wide which alternately carry hot and cold gas. Heat is transferred through the vanes by conduction and they can be made of metal, glass or even plastic. The major advantages of the recuperator are that there are no moving parts, no leakage between the gas streams and the heat exchanger can be made quite compact. In addition the recuperator can be designed such that condensation is removable from one side, though the unit is then non-reversible and must be orientated in a specific manner. The main disadvantage is that large areas of material are exposed to the gas and these may corrode if the gas is hot and dirty.

More recuperators are sold than any other type of gas-to-gas heat exchanger [123] and these sales are for either air-conditioning or pre-heating of combustion air applications.

The latter type obviously operates at high temperatures and generally has stainless steel vanes which are therefore expensive. The manufacturers given in Table 7.2 all produce units which operate at face velocities of 2 to 3 $m\ s^{-1}$ and with pressure drops of 100 to 300 Pa. With the exception of Acóustics and Environmetrics Ltd., all produce high temperature as well as air-conditioning units in all of which control is maintained by the use of dampers in a by-pass duct.

The use of recuperators at temperatures above 700 °C is being studied actively at present. The advantages of recuperators over heat wheels, in particular the absence of moving seals, make them better-suited to high temperature applications. Their use on metallurgical process furnaces has been discussed by Ernest and White [129], and recuperators for gas turbines are described by Cuffe et al. [130]. These recuperators have high-grade stainless steel plates and are used for pre-heating combustion air.

In some industries it is desirable to recover energy from very dirty exhausts at up to 1300 °C, and the development of a suitable recuperator has been conducted by the British Steel Corporation [131]. These units have ceramic tubes carrying the supply air over which the hot exhaust flows and are marketed now by Encomech Engineering Services Ltd., Epsom, Surrey. However much work remains on the development of reliable joints between the ceramic and other materials in such hostile conditions.

Table 7.2 Manufacturers of recuperators

Manufacturer	Gas flow ($m^3 s^{-1}$)	Operating temperature (^oC)
Acoustics and Environmetrics Ltd., Claygate, Surrey	0.6 - 14	<80
Curwen and Newbery Ltd., Westbury, Wiltshire	16	<750
Fläkt Ltd., Staines		as required
United Air Specialists Ltd., Leamington Spa, Warwickshire	0.7 - 19	<840

7.4.4 Heat pipe heat exchanger

The heat pipe is a sealed container holding a wick and a suitable working fluid as shown in Figure 7.2. If one end is heated, the liquid evaporates and the vapour is transferred by the pressure difference between the ends of the pipe. At the cooler end the vapour condenses and the liquid is returned to the heated end by the capillary action of the wick. The energy is transferred as the enthalpy of evaporation of the fluid, and the heat pipe can operate with only a small temperature difference between its ends. In recent years gas-to-gas heat exchangers containing banks of heat pipes have been developed [132]. The theoretical characteristics of such heat exchangers have been studied by Wakiyama et al. [133] and many empirical data are given by Lee and Bedrossian [134].

Heat pipe heat exchangers have several advantages such as having no moving parts and zero leakage. They are fully reversible, can be designed with very low pressure drops and are compact because of the high thermal conductivity of heat pipes. The main disadvantage is that finned heat pipes are needed to utilise the heat pipe's large heat transfer capability, and the fins are liable to corrode and foul. It is possible to control the heat exchange by tilting the unit because the properties of heat pipes are sensitive to the angle of operation although the problems of tilting up to a tonne of equipment should not be overlooked.

FIGURE 7.2 SCHEMATIC DRAWING OF A HEAT PIPE

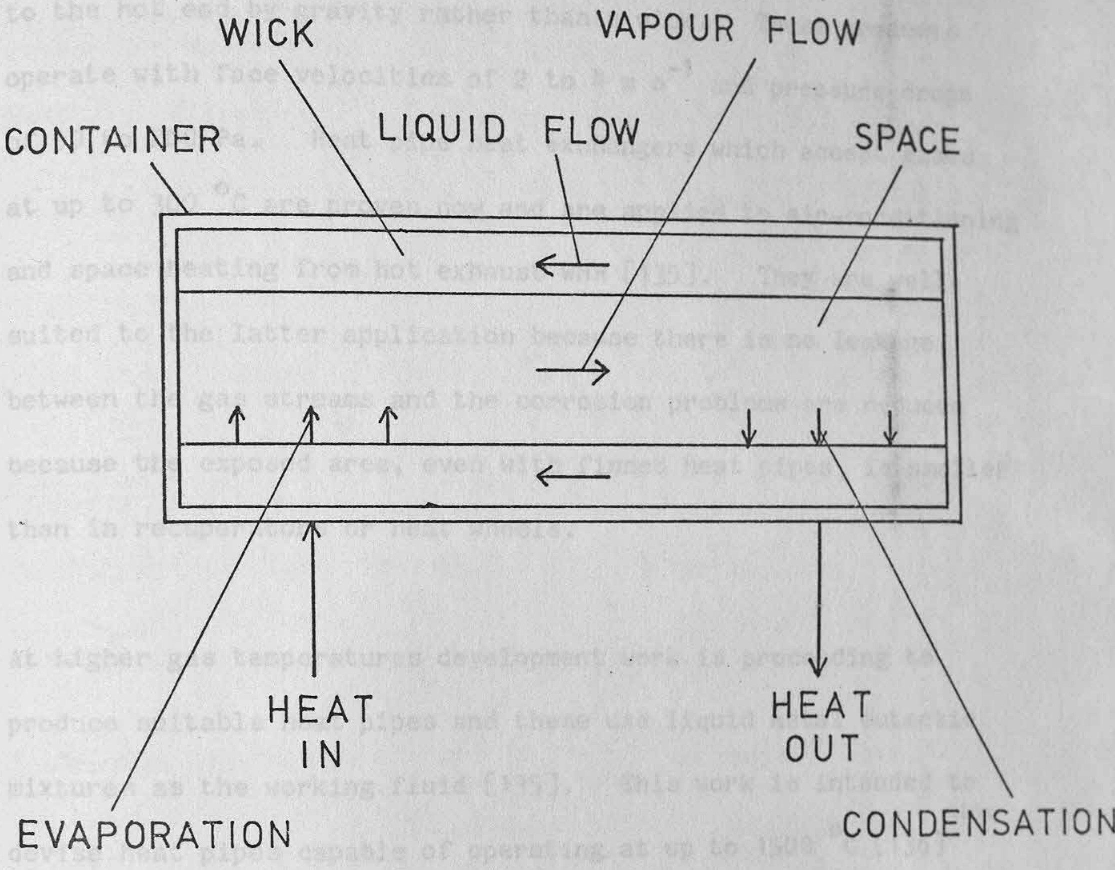


Table 7.3 lists the manufacturers of heat pipe gas-to-gas heat exchangers whose products are available in the United Kingdom. The units made by Fläkt Ltd. are not true heat pipes but thermosyphons in which the condensate is returned to the hot end by gravity rather than a wick. These products operate with face velocities of 2 to 4 m s^{-1} and pressure drops of 50 to 200 Pa. Heat pipe heat exchangers which accept gases at up to 300 ^{o}C are proven now and are applied to air-conditioning and space heating from hot exhaust WHR [135]. They are well-suited to the latter application because there is no leakage between the gas streams and the corrosion problems are reduced because the exposed area, even with finned heat pipes, is smaller than in recuperators or heat wheels.

At higher gas temperatures development work is proceeding to produce suitable heat pipes and these use liquid metal eutectic mixtures as the working fluid [135]. This work is intended to devise heat pipes capable of operating at up to 1500 ^{o}C [136] which is the topmost end of the industrial exhaust gas temperature range. When this work has been completed successfully then heat pipes can be used as the basis of a range of WHR systems covering all exhausts and applications, something which recuperators and heat wheels are never likely to provide.

7.4.5 Summary of existing equipment

The presently available gas-to-gas heat exchangers are of four

Table 7.3 Manufacturers of heat pipe heat exchangers

Manufacturer	Gas flow ($m^3 s^{-1}$)	Operating temperature (^oC)
Curwen and Newbery Ltd.,	5	<65
Westbury, Wiltshire	5	<205
	5	<260
	5	<315
Fläkt Ltd., Staines	1 - 6	<800
International Research and Development Co. Ltd., Newcastle-upon-Tyne	as required	
Isothermix Ltd., London	as required	
Q-Dot Corp. (Burke Thermal Engineering Ltd. , Alton, Hampshire)	0.3 - 27	<720

types none of which is capable of being applied successfully to all the applications outlined in Section 7.3. The heat pump is still under development and for the foreseeable future it will be usable only for air-conditioning because of its limited temperature range. The heat wheel has the inherent weakness of having moving seals which are liable to leak, particularly under arduous thermal cycling at high temperatures. In addition their large surface areas may corrode and then foul, a problem they share with recuperators. At lower temperatures, less than about 400 °C, both these types are suited to all three applications.

Lastly there are heat pipe heat exchangers which have no leakage paths, low pressure drops and smaller surface areas exposed to the gas. They appear to be the best type for applications other than air-conditioning where the heat wheel's ability to recover the enthalpy of evaporation of the moisture in the exhaust is vital. Heat pipes operating above 300 °C are not fully proven, but development work is likely to extend this upper limit to well over 1000 °C. Thus heat pipe exchangers may be the first commercially available WHR systems to work reliably in high temperature exhausts. However these will probably be expensive and may not be saleable because unless the economic criteria are satisfied, no heat exchanger can be marketed successfully. With this in mind the economic considerations relevant to WHR equipment will now be discussed.

7.5 ECONOMIC CONSIDERATIONS

Many people consider that energy conservation is based on sound technical grounds, but that implementation is often obstructed for financial reasons or board level priorities [137]. With fuel price and availability fluctuating over periods of weeks, it is vital that decisions on the purchase of WHR equipment should be taken quickly, but the structure of executive decision-making militates against this. In an attempt to overcome this much work is being sponsored by national governments, both in this country [106] and in the United States [138]. These funds are intended to stimulate the development of new types of equipment, especially in existing high-technology companies, and to encourage the usage of WHR systems. Despite these efforts there is still great difficulty in selling energy conservation systems, and the decision about possible purchase is often taken on economic grounds alone.

Klaschka [103] presents a detailed study of this problem and recognises that industry currently demands a payback period of two years or less for WHR equipment. A simple payback method is used rather than, for example, discounted cash flow, because it is the method most commonly found in sales literature. The heat exchanger itself represents only 15 to 35% of the total price of a system, the remainder being made up of ducting, fans, control equipment and installation costs. The penalty of shutting down a process during installation also needs considering when continuously operating machinery is involved. On this basis Klaschka [103] presents the target heat exchanger prices which are required to render the units saleable as given in Table 7.4.

These target prices are calculated for a fuel price of £1.60 to £1.80 per GJ which was the average in mid-1979. The current price can be obtained from the most recent issue of Energy Trends [139] which provides much up-to-date information on fuel price and availability. It is useful to look at the prices of various fuels as supplied to industry, though it must be stressed that these are average prices including cheap long-term supplies. These are shown in Table 7.5 where the real change is the actual change deflated by the wholesale price index excluding fuel.

It can be seen from Table 7.5 that the economic acceptability of a WHR system depends upon which fuel is conserved. Electricity is the most expensive form of energy, but the sharp increase in fuel oil price, over 50% in real terms last year, is of more interest because electricity is often only a minor component in a company's fuel usage. Those industries that use a great deal of electricity, such as steel and paper making, have special tariffs which are much cheaper than the

Table 7.4 Target heat exchanger prices

Process hours per year	Price (£ per kW recovered)
2000 (day-time batch)	4 to 8
7000 (semi-continuous)	15 to 30
8600 (continuous)	20 to 40

Table 7.5 Industrial fuel price trends

Fuel	Price	Actual change	Real change
	1st quarter 1980	1st quarter 1979 to 1st quarter 1980	1st quarter 1979 to 1st quarter 1980
	(£ per GJ)	(%)	(%)
Coal	1.24	34	23
Heavy fuel oil	1.97	72	57
Gas	1.51	30	19
Electricity	6.40	19	9
Total fuel	-	39	27

average. It is the continuing increase in fuel prices, particularly those which are derived from oil, which makes investment in WHR continually more attractive. Since this trend is likely to persist - no one is expecting oil prices to fall significantly - then payback periods calculated at current prices will be pessimistic and the investment recovered more quickly.

7.6 EXISTING WASTE HEAT RECOVERY INSTALLATIONS

Manufacturers and suppliers of WHR equipment often present details of existing installations of their products and provide information relating to the processes to which their systems are attached. It seems reasonable to suppose that the makers only provide case histories of their more successful units, so the sample is biased to present WHR in a favourable light. Table 7.6 shows this sample of installations and indicates their type, application, savings and payback period.

Several trends emerge from this information as follows:-

(i) the payback period is one to two years in the quoted instances, but is substantially longer in the United States where energy is less expensive;

(ii) the energy recovery of a typical installation is in the range 0.2 to 2 MW, so the proposed large prototype discussed in Appendix 3 is representative of industrial needs;

Table 7.6 Economics of existing waste heat recovery installations

Date	Type		Process	Temperature (^{o}C)	Recovery (MW)	Savings (£yr^{-1})	Payback (yr)
1975	Heat wheel	[140]	Board mill	128	0.5	10 000	1.8
-	Heat wheel	[141]	Papermaking	66	2.4	60 000	0.9
1976	Heat wheel	[142]	Brick kiln	210	0.33	13 000	1.5
1974	Heat wheel	[142]	Tunnel kiln	115	0.14	4700	0.8
1977	Heat wheel	[142]	Rubber	283	0.25	7200	1.9
-	Heat wheel	[142]	Textiles	130	0.25	5000	1.0
-	Recuperator	[143]	Fabrication	116	0.48	11 600	1.5
-	Recuperator	[143]	Gypsum board	-	-	23 000	0.9
*1974	Heat pipe	[144]	Paint oven	213	0.29	$4600	2.5
*1975	Heat pipe	[144]	Foundry	357	1.1	$27 000	2.0
* -	Heat pipe	[144]	Dryer	107	0.24	$3400	3.0
1978	Heat pipe	[140]	Dryer	170	0.5	12 000	2.5

* United States installation

and

(iii) no installation operating at over 357 ^{o}C is listed, presumably because those working at higher temperatures are less economic at present.

The case histories do not provide details of what proportion of the time the units were operating, so it is not possible to comment on the reliabilities of the systems. For the installations in Table 7.6 the average initial cost was £33 per kW recovered, though the two most recent systems cost £54 and £62 per kW. As discussed earlier, the heat exchanger accounts for less than a third of the total cost, so the cost of the heat exchanger is around £20 per kW or less. This is in broad agreement with the target prices given in Table 7.4. The absence of high temperature units from Table 7.6 probably indicates that existing systems for such arduous duties are substantially more expensive. This points to the likely success of the new fluidised bed system as it should be cheaper than its rivals.

7.7 APPLICATION OF THE FLUIDISED BED GAS-TO-GAS HEAT EXCHANGER

The major advantage of the fluidised bed gas-to-gas heat exchanger is its simplicity in that there are no moving parts and no large metal surfaces exposed to the gas streams. Hence the heat exchanger should be reliable and the only component liable to corrode and foul is the distributor plate. Since this is a small item, it can be made from an expensive, corrosion-resistant alloy if necessary, but with little effect on product cost. Automatic cleaning of a similar distributor in a hot and dirty gas has been demonstrated on a shipboard fluidised bed [88] so should not present a serious problem. These advantages can be exploited to the full in WHR from hot exhaust gases above about 400 ^{o}C where existing products become too expensive or unreliable.

The disadvantages of the fluidised bed unit lie in several areas, namely leakage, efficiency, orientation, size and pressure drop, the last two being related. The experimental data show that the leakage between the gas streams can be suppressed, and further work may improve this, but initially the leakage will rule out the application of the system to space heating from hot exhaust because of the risk of leakage of toxic gases. Since the heat exchanger is of the parallel flow type, at best it can only recover half of the available heat if the hot and cold side mass flows are the same. This is achieved experimentally because the small cross flow contribution balances the imperfect parallel flow heat exchange. The recovery is, however, still less than that of the other types of gas-to-gas heat exchanger which operate in the counter flow mode, and this will count against the new system if other aspects are equal. It should be noted also that the fluidised bed unit cannot recover the enthalpy of evaporation of the moisture as there are no hygroscopic coatings on the particles, hence the system is not applicable to air-conditioning heat recovery.

The other disadvantages are less important, though they must not be overlooked if the heat exchanger is to be marketed successfully. Obviously the fluid nature of the bed means that the unit can be operated only with the distributor in a horizontal plane. The bed area and the pressure drop are governed by the face velocities once the volume flows are known. The face velocity in the fluidised bed unit will be between 1 and 1.5 m s^{-1} which is only about one third of that of existing products, thus the face area will be larger. Under these conditions the pressure drop will be 1000 to 1500 Pa which is several times greater than that of current units so larger fans will be needed with the penalties of greater running and capital costs. If the face velocity is increased to reduce the bed area then the pressure drop becomes still larger and the erosion the distributor caused by the passage of the fluidising gas can become significant [145].

Overall the fluidised bed gas-to-gas heat exchanger will be suitable for only one of the three applications outlined in Section 7.3, that is pre-heating combustion air. In that instance the reliability and corrosion resistance of the unit to high temperature gases can be exploited fully. Despite the disadvantages of large size and pressure drop, it should be the first commercially available system to cope successfully with the high temperature exhausts encountered in that application. Until the advent of cheap high temperature heat pipes, the fluidised bed unit will be the cheapest and most reliable system in that market sector.

CHAPTER EIGHT

ASSESSMENT OF THE FLUIDISED BED GAS-TO-GAS HEAT EXCHANGER

CHAPTER EIGHT

ASSESSMENT OF THE FLUIDISED BED GAS-TO-GAS HEAT EXCHANGER

8.1 INTRODUCTION

The intention of this chapter is to collate all the information presented earlier on the theoretical model, the experimental work and the market for gas-to-gas heat exchangers. The assessment of this information will lead to five distinct areas for discussion. These are the feasibility of the ideas in the patent, the market openings for gas-to-gas heat exchangers, the commercial potential of the new fluidised bed system, the design criteria for larger units, and recommendations for further work. The following sections will discuss each of these aspects in turn.

8.2 FEASIBILITY OF THE PATENT

As described in Section 3.2, the patent [7] protects a novel fluidised bed gas-to-gas heat exchanger. The experimental work has demonstrated that a heat exchanger built in accordance with the provisions of the patent operates satisfactorily. The heat exchanger performs as if it were a perfect parallel flow unit under appropriate fluidising conditions. The performance of the heat exchanger is described fully elsewhere, but the general conclusion is that it satisfies the technical requirements for a WHR system.

The question of leakage between the two gas streams is raised in the patent and is very important to the marketing and applicability of the heat exchanger. It is recognised in the patent that some leakage is inevitable because of the design of the system, and several methods for reducing leakage are proposed. These have not been studied exhaustively, but the results provide some grounds for the claim in the patent that one or more of these methods can reduce leakage significantly.

8.3 MARKET OPENINGS FOR GAS-TO-GAS HEAT EXCHANGERS

The survey of the products currently available for gas-to-gas WHR in Section 7.4 has shown that there is a need for a cheap and reliable system operating at high temperatures. Existing systems tend to be insufficiently reliable or prohibitively expensive when required to operate above about 400 °C. Under these conditions corrosion-resistant materials are required and moving seals tend to fail due to thermal stresses. A new type of heat exchanger is needed to overcome these problems because it is unlikely that existing types can be developed further and still produce a relatively cheap unit. The work reported here suggests that the fluidised bed gas-to-gas heat exchanger can fulfil this market need.

The demand for WHR equipment is likely to increase in the future as the price and insecurity of supply of fuel increase. Because of the present economic recession in the industrial countries, the great majority of British companies are reviewing their investment programmes. In the face of resolving priorities for investment, energy conservation often rates as a low priority. This is unfortunate because such short-term decisions may make those same companies less competitive when the economy recovers. When that occurs the demand for energy is likely to rise and shortages may result, depending upon the political decisions of the oil-producing countries, which would affect those companies without WHR installations more adversely.

Hence although the long-term future for the demand for WHR equipment seems assured, the short-term requirements will depend heavily upon Governments' support for such installations. The rationale for this assistance is essentially that energy conservation is of value to the nation as a whole in addition to individual companies, and the national interest should be supported by the Government. This view is recognised in much of the European Economic Community and funds are being provided to help companies install WHR equipment. In this way the short-term demand for gas-to-gas heat exchangers will be stimulated because of their role in waste heat recovery.

8.4 COMMERCIAL POTENTIAL OF THE NEW HEAT EXCHANGER

The need for new developments in gas-to-gas heat exchangers for use in high temperature WHR has been demonstrated already. It has been shown also that systems for use in energy conservation must be as cheap as possible so that the payback period for the installation can be reduced. The most important feature in the marketing of any item of WHR equipment is the payback period, particularly in times when interest rates are high and capital expenditure is being scrutinised most critically as at present. Thus initial cost will probably be more important than maintenance and running costs to potential customers. This situation tends to favour the prospects for the novel fluidised bed system. Its simplicity means that the initial cost will be low, though the consequent reduction in maintenance costs may not be as strong a selling point. Both these aspects of lower costs are offset somewhat by the high pressure drop of the unit which requires a larger fan which has higher running costs.

The application to which the fluidised bed gas-to-gas heat exchanger is suited the best is the pre-heating of combustion air by the enthalpy of hot exhaust gases. This exploits the fact that the new system will be the first to be reliable at high temperatures, yet will not expose its disadvantages. In particular leakage between the two gas streams is, within reason, inconsequential in this application, but at the levels found in the work reported here the system is unsuitable for space heating from hot exhaust. If further work can reduce the leakage to acceptable limits, then the market for the heat exchanger will be expanded considerably.

As stated above the capital and installation costs will be the central feature of the marketing strategy for the heat exchanger and target prices are shown in Table 7.4. The prices shown are those needed to achieve a payback period of two years which appears to be the longest acceptable to financial controllers in industry at the moment. The short-term view which is implicit in this limit has been discussed already and that it does not reflect that national interest is served by energy conservation. Obviously the heat exchanger, in common with all other items of plant, should be operated for as many hours per year as possible, thus continuous or semi-continuous processes are the preferred installations. In such cases the capital and installation costs combined must be less than £20 to £40 per kW recovered for units to be saleable as economic items of WHR equipment.

It must be stressed that the main advantage of the fluidised bed system is its simplicity and hence its reliability for high temperature operation. On other grounds the new heat exchanger is markedly inferior to existing types of gas-to-gas heat exchangers. In terms of compactness, which is a measure of energy transferred per unit volume of the heat exchanger, the fluidised bed system is five times the size of an equivalent heat wheel or recuperator. A second criterion is the energy transferred per unit pressure drop and here the new device has over twice the pressure drop of existing types. But it is thought that the advantages of the fluidised bed unit in being the only economic and reliable high temperature WHR system will outweigh these deficiencies.

8.5 DESIGN CRITERIA FOR LARGE UNITS

Since the indications are that the heat exchanger can fulfil a known market need, it is important to extract sufficient information from the data to define the criteria essential for the prediction of the performance of a larger unit. The design of a large prototype capable of recovering 550 kW forms the basis of an application by Stone-Platt Fluidfire Ltd. for funding for the next stage of development. This proposal is discussed in detail in Appendix 3, but here only the principles adopted for scaling-up the heat exchanger design are described.

The starting points of the design are the hot and cold gas inlet temperatures and volume flow rates. From the latter the areas of the two sides of the bed can be determined by setting the actual fluidising velocity at some value. The full set of recommended design values is given in Table 8.1. The ratio of the two mass flows will depend upon the application, but for the pre-heating of combustion air the mass flows will be almost identical. The predicted effectiveness is about 0.98 so the expected heated gas outlet temperature can be calculated from equation (4.15).

Of more importance, however, is the composition and depth of the bed and the configuration of the distributor plate. Alumina has been chosen as the particulate material because of its properties (see Section 3.6) and the larger 36 grit size is preferable because it requires less disengagement space. The

Table 8.1 Recommended design values

For Greenings' type distributor plate as shown in Figure 5.8

Actual fluidising velocity	1.2 to 1.5 m s^{-1}
Bed material	36 grit (520 μm) fused alumina
Bed depth	40 mm
Partition gap	20 mm
Pressure difference between hot and cold side freeboards	500 Pa

bed depth is a compromise between enhanced heat transfer and greater pressure drop for in deeper beds the gas and particles approach thermal equilibrium more closely. However a deeper bed circulates less easily so the optimum depth will depend upon the temperature difference between the two gas streams; deeper beds being required for larger temperature differences.

To reduce leakage it is likely to be beneficial to stagger the partition away from the plenum division, as in Figure 3.4, as well as developing a pressure difference between the two gas streams leaving the bed. The partition should be positioned about 25 mm downstream of the division where the particles flow from the hot side to the cold. Depending upon the relative areas of the two sides of the bed and any external space restrictions on the size of the unit, the shape of the unit may differ from that used in the initial development reported here. This may affect the most suitable position for the partition to minimise leakage, and this is a factor which should be borne in mind before changing the shape of the heat exchanger. Experience has shown that the scaling-up of a fluidised bed system can lead to unexpected difficulties, hence the importance of the next phase of development which is described below.

8.6 FURTHER WORK

Further studies on the heat exchanger will fall into one of two categories. One will be to improve and scale-up the heat exchanger, and the other will gain more knowledge of why it operates as it does. The two groups will be discussed in turn, and doubtless the proposed work itself will suggest other aspects which merit more study.

8.6.1 Improving and scaling-up the heat exchanger

The next stage of the development of the heat exchanger into a commercial product is the construction of a large prototype of typical industrial size. This is the subject of an application for funding from the Commission of the European Communities and is described in Appendix 3. The proposed unit would recover 550 kW from a gas stream at 800 ^{o}C and would demonstrate the system to potential customers.

The technical aims of the proposed work are to study attrition, the effects of prolonged thermal cycling on the particles, distributor life, solids' flow and the quality of fluidisation. All these will be influenced by the larger size and higher operating temperature of the prototype compared with the present small experimental unit. The predictions of the theoretical model will also be checked and improved to aid the design of a full range of production units. In the prototype the hot gas stream will be exhaust from a gas burner and hence contain large quantities of combustion products. This can be used to advantage to determine the leakage more accurately because the tracer gas will be mixed uniformly into one gas stream.

There are no plans in the proposed work to use other types of distributor plates because the high cost of making special plates cannot be justified at this stage. If the heat exchanger proves to be of great interest to industry, then alternative distributor plates and bed geometries should be investigated. Much will depend upon the response of the market place to the new heat exchanger as to what further development and studies are carried out.

8.6.2 Advancing basic understanding

The principal area of basic understanding which was beyond the scope of the present work was the interaction between the fluidising gas and the bed material. This is a most complicated phenomenon which certainly merits further investigation. The profiles of the jets of gas leaving the distributor are determined by many factors, but particularly by the size and shape of the exit holes. Momentum and heat are transferred from the jets to the particles in an interaction that is central to the operation of the heat exchanger. The generation of the solids' flow internally by the fluidising gas has not been reported before and is a new type of flowing fluidised bed. The comparison of this with existing flowing beds such as inclined channels may be valuable. Other areas for further investigation are specific to the heat exchanger and these include a detailed study of the flow patterns at the boundary between the two gas streams in the vicinity of the partition. Also controlled experiments on the effect of thermal cycling on the particles can be conducted more easily in the laboratory than on the prototype heat exchanger.

CHAPTER NINE

CONCLUSIONS

CHAPTER NINE

CONCLUSIONS

9.1 CONCLUSIONS

With reference to the questions posed in Section 1.4 which were to be answered in this thesis, the following conclusions may be drawn:-

(i) a heat exchanger constructed according to British Patent Number 1 500 231 operates sufficiently well to justify further development;

(ii) the performance under optimum fluidising conditions closely approaches that of a perfect parallel flow heat exchanger;

(iii) some aspects of the performance can be predicted from the simple theoretical model of the unit developed in Chapter 4;

(iv) the source of cross-contamination between the two gas streams is confined to a small region near the boundary between them;

(v) the leakage can be reduced significantly by developing a pressure differential between the freeboard zones on either side of the partition;

(vi) the heat exchanger is suited best to the waste heat recovery application in which the enthalpy content of hot exhaust gases from combustion is used to pre-heat the incoming combustion air;

(vii) existing types of gas-to-gas heat exchangers are unreliable and/or expensive when required to operate at temperatures in excess of 400 °C;

(viii) the new system shows promise of being able to be produced cheaply enough to fulfil this market demand at a cost the market can withstand;

and

(ix) a large prototype capable of recovering 550 kW from hot exhaust at 800 °C has been designed from the data collected in the present work and a set of recommended design values is presented.

APPENDIX 1

This is a copy of the covering patent of the gas-to-gas heat exchanger which was filed by Elliott and Virr in 1975. The complete specification was published in 1978 when the invention became fully protected.

PATENT SPECIFICATION (11) 1 500 231

1 500 231

(21) Application No. 18854/74 (22) Filed 30 April 1974
(23) Complete Specification filed 25 July 1975
(44) Complete Specification published 8 Feb. 1978
(51) INT. CL.² F28D 19/02
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F4K 23A2 23A3 23A4 29
(72) Inventors DOUGLAS ERNEST ELLIOT
MICHAEL JOHN VIRR

(54) IMPROVEMENTS IN AND RELATING TO HEAT EXCHANGERS

(71) We, Stone-Platt Fluidfire Limited, formerly Fluidfire Development Limited, a British company of Washington Street, Netherton, Dudley in the County of Worcester, do hereby declare the invention for which we pray that a patent may be granted to us and the method by which it is to be performed to be particularly described in and by the following statement:—

This invention relates to a heat exchanger for transferring heat from a first gas stream to a second gas stream and also to a method of transferring heat between two such streams.

The invention has been devised primarily to meet a requirement for transferring heat from gaeous products of combustion to air which may be used for the combustion of a fuel or for supply to the interior of a building to heat same.

Known apparatus for transferring heat from gaseous products of combustion to air comprises a wheel which is permeable to gases in the axial direction and two ducts, one for conveying the products of combustion and the other for conveying the air. Diametrically opposite parts of the wheel extend across respective ones of the ducts and metallic or ceramic elements in the wheel absorb heat from the products of combustion whilst the elements are within the duct containing same. The wheel is rotated so that the heated elements pass into the duct containing the air where they give up heat to the air. Such apparatus is generally referred to as a heat wheel. Heat wheels which operate satisfactorily are expensive and it is difficult to provide a satisfactory seal between the boundary walls of the ducts and the rotary wheel.

There has also been a proposal, in published specification No. 1,166,761, for the provision of upper and lower fluidised beds through which there are passed respective gas streams between which heat is required to be transferred. According to this proposal, particles are caused to flow from the upper bed to the lower bed by gravity and are raised from the lower bed to the upper bed by a carrier fluid which is supplied to the bottom of a riser passage.

One object of the present invention is to provide an alternative form of heat exchanger for transferring heat between two gas streams which is less expensive than or more efficient than the known heat wheel or the arrangement described in the aforesaid published specification.

According to a first aspect of the invention there is provided a heat exchanger comprising two ducts for conveying respective first and second gas streams between which heat is to be exchanged, two beds of particles, each situated in a respective one of said ducts and arranged for fluidisation by a respective one of said gas streams, and one or more openings in said ducts, which openings provide communication between the beds, the beds being at substantially the same level and the heat exchanger being so arranged that particles migrate from one of said beds to the other bed when the heat exchanger is operating.

The migrating particles carry heat from one of the streams which is at a higher temperature to the other of the streams which is at a lower temperature. Since the beds are at substantially the same level, it is possible to establish migration of the particles without the use of transport means specifically for transporting particles from one bed to the other.

The heat exchanger may comprise more than two beds of particles. For example, four beds may be provided, the beds being arranged in pairs so that the beds of each pair are situated one in each of the ducts and that particles can migrate between the beds of a pair. Furthermore, it would be within the scope of the invention to provide three or more ducts, each containing a respective bed so that heat can be exchanged between three or more gas streams.

According to a further aspect of the invention there is provided a method of transferring heat from a first gas stream to a second gas stream, wherein the gas streams are passed through respective beds of particles in such a manner as to fluidise the beds, the beds being at substantially the same level, and wherein particles are caused to migrate from one of said beds to the other.

The invention will now be described by way of example with reference to the accompanying drawings wherein:

FIGURE 1 shows a cross section in a vertical plane on the line 1-1 of Figure 2 of a heat exchanger in accordance with the invention,

FIGURE 2 shows a horizontal cross section of the heater on the line II-II of Figure 1,

FIGURE 3 shows diagrammatically a cross section in a vertical plane of a further heat exchanger in accordance with the invention,

FIGURE 4 is a fragmentary view on an enlarged scale showing a detail of the heat exchangers of Figure 1 and Figure 3, and

FIGURE 5 is a view similar to Figure 3 illustrating a modification of the heat exchanger shown in Figure 3.

The heat exchanger illustrated in Figures 1 and 2 is intended for transferring heat from hot flue gases to air which is to be supplied to the interior of a building for heating the latter. The heat exchanger comprises a casing 10, the interior of which is divided by upwardly extending walls 11, 12 and 13 into a flue gas duct 14 and an air duct 15. These ducts extend side-by-side upwardly through the heat exchanger.

The gas and air ducts 14 and 15 are subdivided by a horizontal support 16 into upper and lower portions. The support 16 is pervious to gases. A flue gas inlet 17 communicates with the lower portion of the flue gas duct 14 and an air blower 18 communicates with the lower portion of the air duct 15. Adjacent to the air blower 18, the casing 10 is formed with an air inlet 19 through which ambient air is drawn by the blower 18. The upper portion of the flue gas duct 14 communicates with a further blower 20 which is arranged to discharge gases from the flue gas duct into a flue 21. The upper portion of the air duct 15 leads to air outlet openings 22 through which heated air is discharged from the heat exchanger in use.

On that portion of the support 16 which lies within the air duct 15 there is a bed 23 of refractory particles for example alumina or zircon. On that part of the support 16 which lies within the flue gas duct 14 there a further bed 24 of the same refractory particles. The wall 12 which divides the air duct from the flue gas duct is formed with an opening 25 which has the form of a rectangular slot and extends across the full width of the walls 12. The lower boundary of the slot 25 coincides with the underside of the support 16 and the upper boundary of the slot 25 is spaced upwardly from the support but lies below the uppermost surfaces of the beds 23 and 24. The beds communicate with each other through the slot 25.

Each of the beds 23 and 24 is partly divided by a wall 26 which projects upwardly from the support 16, has a height somewhat greater than the depth of the beds 23 and 24 and intersects the wall 12 at right angles. The wall 26 extends along the bed 23 from the wall 12 to a position spaced somewhat from that end of the bed 23 which is remote from the bed 24. Similarly, the wall 26 extends from the wall 12 along the bed 24 to a position spaced somewhat from that end of the bed 24 which is remote from the bed 23.

The support 16 is in the form of a sheet of metal which has been pierced to form slits which are distributed over the entire area of the support 16. As shown in Figure 4, these slits 27 are so formed that air entering the bed 23 or the bed 24 through a slit has a velocity which includes a horizontal component and a vertical component. The gas herefore applies to the particles of the bed an upwardly directed component of force, which maintains the bed in a fluidised condition, and a horizontal component of force which causes the fluidised particles to flow. In a part of the support 16 indicated by the reference 28 in Figure 2, the slits 27 all face in a direction away from the wall 12. In a part of the support indicated by the reference 29 the slits face in a direction parallel to the wall 12. In a part of the support indicated by the reference number 30 and also in a further part of the support which lies below the bed 24 and is indicated by the reference number 31, the slits face in a direction opposite to that of the slits in the part 28. In a further end part 32 of the support, the slits face in the direction opposite to that of the slits in the part 29 and in the part 33 of the support the slits face in the same direction as that of the slits in the part 28. When air is passed through the support 16 into the bed 23 to fluidise the bed, and flue gas is passed through the support 16 into the bed 24 to fluidise that bed, circulation of the particles from the bed 24 through the bed 23 and back to the bed 24 is established. With fluidising velocities of 0.5 to 10 feet per second, average velocities of the particles around this circulatory path of up to 10 feet per second can be attained. Whilst in the bed 24 the particles are heated by the flue gases. When in the bed 23, the particles give up heat to the air.

The apparatus is intended primarily for recovering waste heat from flue gas produced in a boiler or some other separate apparatus. When the heat exchanger is used in this way, the flue gas from the boiler or other apparatus is admitted to the heat exchanger through the inlet 17. The heat exchanger can also be used in circumstances where the flue gas from the boiler or other separate apparatus does not carry sufficient heat, or where no such flue gas is available from separate apparatus. To this end, the heat exchanger includes a burner 33 which is arranged to discharge a burning gaseous or liquid fuel into the lower portion of the flue gas duct 14. The hot exhaust gases from the burner pass through the bed 24 in just the same way as does flue gas admitted through the inlet 17. Air which is to be heated, normally fresh air, is drawn into the heat exchanger through the inlet 19 and is discharged through the outlets 22 either directly into a space which is to be heated or into ducts which convey the heated air to the point of use.

It will be noted that the blowers 18 and 20 are so arranged that there will normally be established in the flue gas duct 14 a pressure slightly below that of the atmosphere and in the air duct 15 a pressure slightly above that of the atmosphere. This arrangement has two advantages. Firstly, any leaks in the flue gas duct will result in ambient air being drawn into this duct, instead of flue gases being discharged into the atmosphere adjacent to the heat exchanger. Secondly, there will be a small flow of air from the duct 15 through the slot 25 into the duct 14 and this flow will prevent flue gas passing into the air duct 15 to be discharged from the outlets 22.

The beds 23 and 24 have the same depth. When the beds are slumped, this depth is between $\frac{1}{2}$ inch and 3 inches. The size of the particles is within the range 100 micron to 3000 micron. The width of the slits 27 is such that the particles cannot pass through the support 16. The length of each slit is approximately $\frac{1}{2}$ inch.

The heat exchanger illustrated in Figures 1 and 2 is a single stage heat exchanger. The highest temperature to which the particles of the bed 24 could, in theory, be heated is the temperature of the flue gas discharged from the heat exchanger. Since the air cannot be heated to a temperature higher than that of the particles leaving the bed 24, and in practice will be heated to a somewhat lower temperature, if the flow rates of the flue gas and air are equal then no more than one half of the heat available in the flue gas will be transferred to the air being heated. Transfer to the air of a larger proportion of the heat carried by the incoming gases can be achieved in a multi-stage heat exchanger. In Figure 3, there is illustrated diagrammatically a two-stage heat exchanger. If the flow rates of flue gas and air are equal, the theoretical maximum amount of heat which can be extracted from the flue gas is 66.7% of the heat carried by the incoming flue gas.

The heat exchanger shown in Figure 3 includes a flue gas duct 40 in which there are upper and lower beds 41 and 42 respectively, of refractory particles. The lower end of the flue gas duct communicates with a flue gas inlet 40*a* and the upper end of the flue gas duct leads to a blower 43 from which flue gas is discharged through a flue. The heat exchanger further comprises an air duct 44 which is separated from the flue gas duct by a vertical wall 45. The air duct also contains upper and lower beds 46 and 47 respectively.

The beds 41 and 46 are supported on a common support 48 and communicate with each other through a slot 49 in the wall 45. The beds 42 and 47 are supported on a common support 50 and communicate with each other through a slot 51 in the wall 45.

The air duct 44 is so arranged that air drawn into the heat exchanger by a blower 52 is forced through the support 48 into the bed 46 and then flows to the underside of the support 50 through which it passes to enter the bed 47. From the bed 47, the air is discharged to a space which is to be heated.

The support 48 is formed of the same pierced sheet metal as is the support 16 hereinbefore described. The support 48 may be arranged in the same way as the support 16 so that circulation of particles from the bed 41 through the bed 46 and back to the bed 41 is established in operation. To this end, a wall corresponding to the wall 26 would be arranged to partly divide the beds 41 and 46. Alternatively, the slot 49 may be formed in two separate portions, disposed adjacent to respective ends of the beds 41 and 46 so that circulation of particles is established along the bed 46 through one portion of the slot 49 into the bed 41, in the reverse direction along this bed and through the further portion of the slot 49 into the bed 46 once more. The beds 42 and 47 and the support 50 would be arranged in the same way as the beds 41 and 46 and the support 48.

In Figure 5, there is illustrated a modification of the heat exchanger shown in Figure 3. Parts shown in Figure 5 which correspond to those already described with reference to Figure 3 are indicated by like reference numerals with the prefix 1 and the preceding description is deemed to apply thereto.

The only difference between the apparatus shown in Figure 5 and the apparatus shown in Figure 3 is that, in the apparatus of Figure 5, that part of the wall 145 which lies above the support 148 is off-set laterally with respect to that part of the wall which lies between the supports 148 and 150 in a direction towards the air duct and that the part of the wall 145 which lies below the support 50 is similarly off-set in a direction towards the flue gas duct 140. This arrangement of the wall 145 causes a small proportion of the air flowing through the air duct 144 to pass through the slots 149 and 151 into the flue gas duct. This prevents flow of flue gas in the opposite direction through the slots 149 and 151 so that the air discharged from the heat exchanger is not contaminated by flue gas. With the arrangement shown in Figure 5, it is not necessary for the blowers to be arranged so as to establish a higher pressure in the air duct than exists in the flue gas duct. The blowers may be arranged to maintain equal pressures in these ducts.

In the accompanying drawings, the openings through which adjacent beds communicate with each other are shown to be submerged below the upper surfaces of the beds concerned when these beds are fluidised. To achieve this, it is not essential for the openings to be submerged below the upper surfaces of the slumped beds since the depth of the fiuidised beds is greater than that of the slumped beds. The depth of the fluidised beds may be between 3 and 4 times that of the slumped beds.

A further modification which may be made to promote flow of particles along the beds is to provide for each bed a respective support which slopes downwardly from one end to the other end of the bed, Adjacent to the lower end of the bed, the support would be adapted to admit gas to the bed at a relatively high rate so that particles in the lower end portion of the bed are thrown upwardly to a higher end portion of an adjacent bed. When fluidised, the particles would tend to flow down the associated support. The beds may slope from the same higher level to the same lower level.

Heat exchangers in accordance with the invention may advantageously be used to transfer heat from hot flue gases to air which is subsequently discharged into a space which is to be heated. The air may be heated to a temperature, for example 90° to 100°F, at which it can be discharged directly into a space occupied by people. Alternatively, the air may be heated to a higher temperature, for example 300°F, distributed to the point use and then diluted with cooler air before being discharged into the space to be heated. Heat exchangers in accordance with the invention may also be used to preheat air used for combustion in boilers, furnaces and other combustion apparatus by transferring heat from the flue gases to the incoming air, to heat air to be used in a drying plant by transferring to that air heat from the gases exhausted from the plant and to heat clean ventilation air by transferring to that air heat from stale air which is to be discharged from a building by a ventilation system.

In certain applications, large pressuure differences between two gas streams between which the transfer is required may be unavoidable. In such cases, the flow of gas and particles through the openings which provide communication between adjacent beds may be controlled by means of a rotary valve. A suitable form of rotary valve comprises a number of vanes radiating from a spindle. The spindle would be mounted in a position above the upper surfaces of the adjacent beds for rotation about a horizontal axis. The vanes below the spindle would extend into the bed to the associated support, there being a slight clearance between the vanes and the support. The vanes and the boundaries of the opening would be so arranged that, when stationary in any position, the valve obstructs the opening. Rotation of the valve, which may be caused by the flow of particles or by driving the spindle, would permit controlled flow of the particles through the opening.

As will be clear from the foregoing description, the references herein and in the appended claims to two beds of particles which are in communication with each other embrace what can alternatively be regarded as two portions of a single bed.

WHAT WE CLAIM IS:—

1. A heat exchanger comprising two ducts for conveying respective first and second gas streams between which heat is to be exchanged, two beds of particles, each situated in a respective one of said ducts and arranged for fluidisation by a respective one of said gas streams and one or more openings in said ducts, which openings provide communication between the beds, the beds being at substantially the same level and the heat exchanger being so arranged that particles migrate from each of the beds to the other during operation.

2. A heat exchanger according to Claim 1 wherein the arrangement is such that, during operation, there is a general flow of particles along a circulatory path through both beds.

3. A heat exchanger according to Claim 1 or Claim 2 wherein the or each opening is submerged below the surface of the associated beds, at least when the beds are fluidised.

4. A heat exchanger according to any preceding claim wherein the depth of each bed, when slumped, does not exceed 3 inches.

5. A heat exchanger according to Claim 4 wherein the depth of each bed, when slumped, is at least ½ inch.

6. A heat exchanger according to any preceding claim which is so arranged that, during operation, a small proportion of the gas which enters one of the beds from one duct passes from that bed into the other duct, whereby contamination of the gas stream in the one duct by gas from the other duct is supressed.

7. A heat exchanger according to Claim 6 wherein the ducts are separated from each other by a lower wall portion below said opening and by an upper wall portion above the opening and the upper wall portion is off-set laterally relative to the lower wall portion in a direction towards said one duct.

8. A heat exchanger according to Claim 6 which includes a first blower arranged to blow gas through the one duct and situated upstream of the bed in that one duct, and a second blower arranged to draw gas through the other duct and situated downstream of the bed in that other duct, whereby the gas in the one duct is maintained under a higher pressure than the pressure maintained in the other duct.

9. A heat exchanger according to any preceding claim wherein each bed is supported on a support which is arranged to admit gas to the bed in such a manner that the incoming gas has a velocity with a component in a horizontal direction, whereby the gas entering the bed promotes flow of particles of the bed across the support.

10. A method of transferring heat from a first gas stream to a second gas stream wherein the gas streams are passed through respective beds of particles in such a manner as to fluidise the beds, the beds being at substantially the same level, and wherein particles are caused to migrate from each of the beds to the other.

11. A method according to Claim 10 wherein the migration of the particles is promoted by the action of the gas streams on the particles.

12. A method according to Claim 11 wherein the only gas streams which promote migration of the particles are those gas streams between which heat exchange is required to take place.

13. A method according to any one of Claims 10, 11 and 12 wherein the gas in one bed is maintained at a pressure greater than the pressure of the gas in the other bed.

14. A method of transferring heat from a first gas stream to a second gas stream, the method being substantially as herein described with reference to the accompanying drawings.

15. A heat exchanger substantially as herein described with reference to and as shown in Figures 1, 2 and 4 of the accompanying drawings.

16. A heat exchanger substantially as herein described with reference to and as shown in Figures 3 and 4 or as described with reference to and as shown in Figures 4 and 5 of the accompanying drawings.

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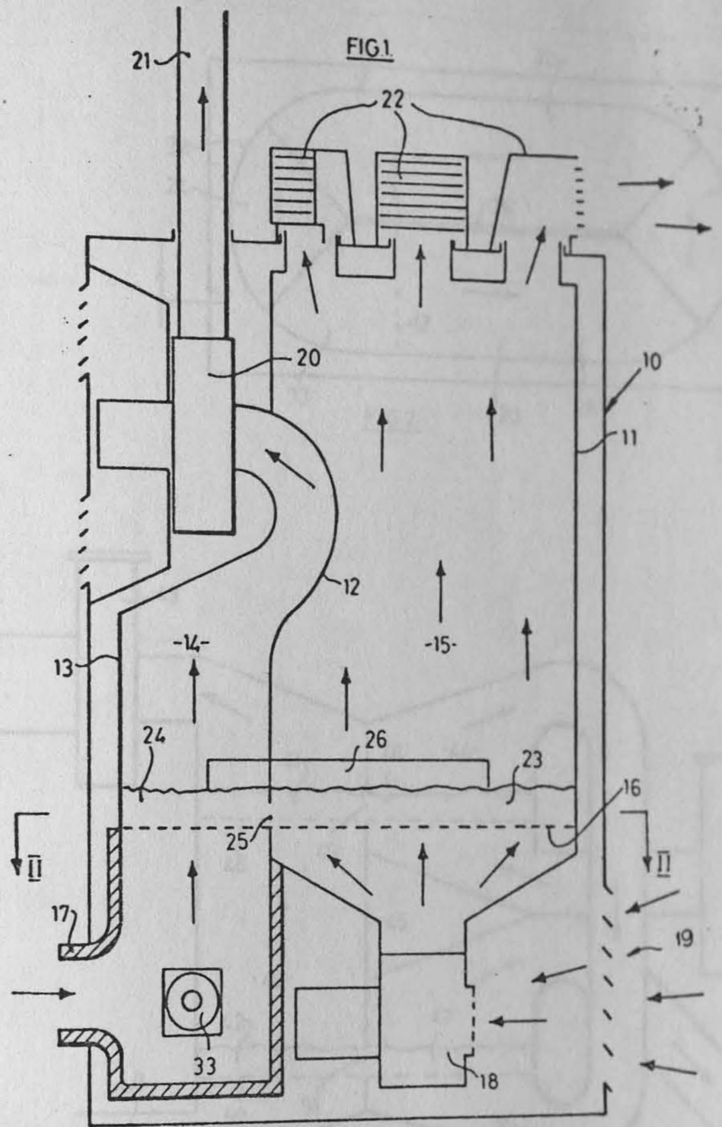
1500231 COMPLETE SPECIFICATION

3 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 1

FIG.1.



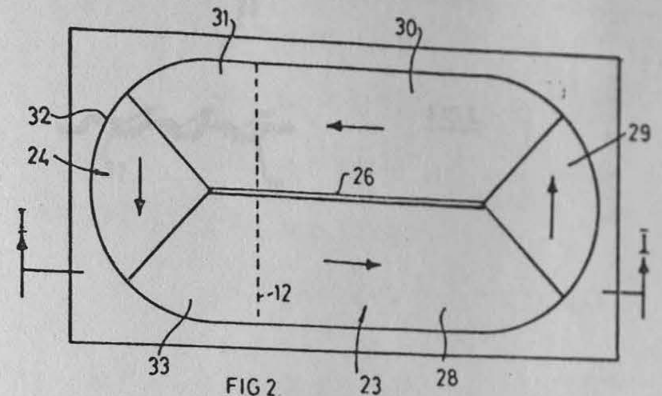

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FIG 2.

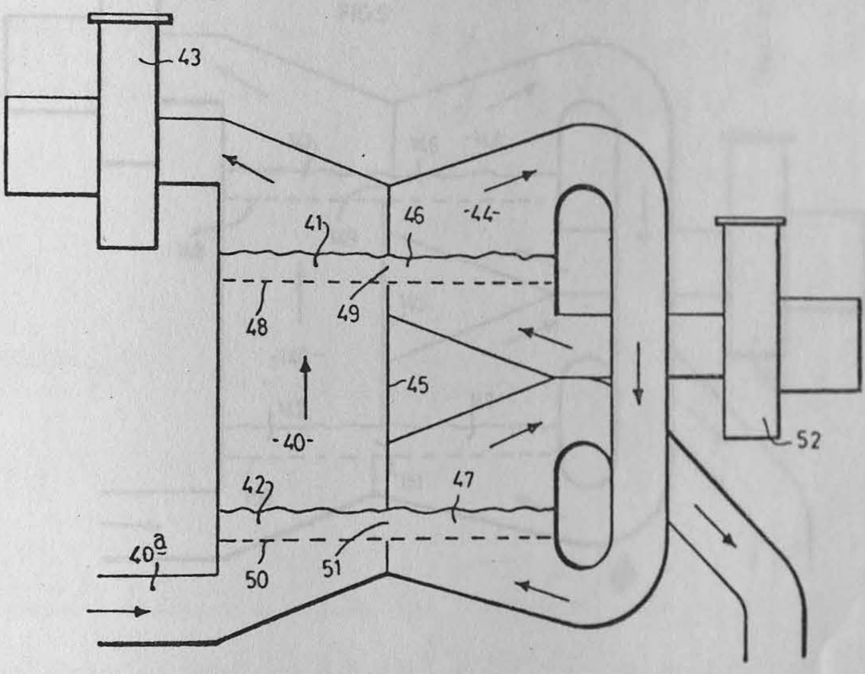
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FIG 3.

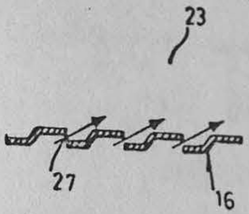

FIG.4.

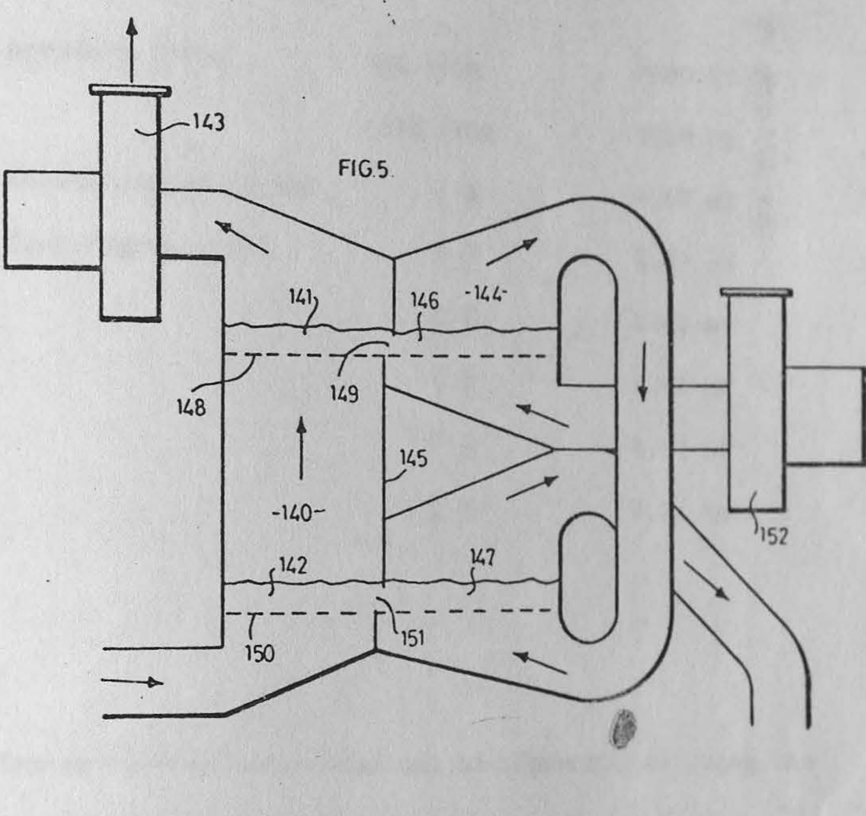

FIG.5.

APPENDIX 2

Specimen calculation of effectiveness.

1. Starting data:-

thermocouple e.m.f.s.	Rotameter inlet		1.65 mV
	hot air inlet		9.87 mV
	cold air inlet		1.68 mV
Rotameter readings	hot flow (R1/R2)		0/15 cm
	cold flow (R3/R4)		10/18 cm
pressure drops	hot side		1500 Pa
	cold side		1990 Pa
thermocouples in bed		1 ~~A~~	4.64 mV
(see Figure 5.18)		2 ~~B~~	4.83 mV
		3 ~~C~~	4.89 mV
		4 ~~D~~	4.90 mV
		5 ~~E~~	4.73 mV
		6 ~~F~~	4.70 mV

2. Convert thermocouple readings to temperatures using the calibration shown in Figure 5.17.

Rotameter inlet temperature	40.4 °C
hot air inlet temperature	243.6 °C
cold air inlet temperature	41.2 °C

3. Calculate Rotameter correction factor.

$$\text{Factor} = \sqrt{\frac{\text{actual density}}{\text{calibrated density}}}$$

$$= \sqrt{\frac{293}{(273 + 40.4)} \quad \frac{(101325 + 1750\ *)}{101325}}$$

$$= 0.975$$

* average pressure drop

4. Calculate flows from Rotameter calibrations.

$$\text{hot air flow} = 1.09\ R1^2 + 113.94\ R1 + 280.9 + 1.38\ R2^2$$

$$+ 108.61\ R2 + 324.4$$

$$= 2545\ /\ \text{factor}$$

$$= 2610 \quad l\ min^{-1}$$

$$\text{cold air flow} = 1.27\ R3^2 + 111.68\ R3 + 313.6 + 1.42\ R4^2$$

$$+ 112.59\ R4 + 298.1$$

$$= 4342\ /\ \text{factor}$$

$$= 4454 \quad l\ min^{-1}$$

5. Calculate actual fluidising velocities.

cold side distributor area $= 0.0801 \text{ m}^2$

hot side distributor area $= 0.0774 \text{ m}^2$

cold side fluidising velocity

$$= \frac{4454 \ (273 + 41.2)}{293} \quad \frac{1}{60\,000 \quad 0.0801}$$

$$= 0.994 \text{ m s}^{-1}$$

hot side fluidising velocity

$$= \frac{2610 \ (273 + 243.6)}{293} \quad \frac{1}{60\,000 \quad 0.0774}$$

$$= 0.991 \text{ m s}^{-1}$$

6. Calculate air outlet temperatures.

The average of thermocouples A, B and C is the hot side outlet temperature; and that of D, E and F is the cold side outlet temperature.

cold side outlet temperature $= 117.0^{\circ}$ C

hot side outlet temperature $= 117.3^{\circ}$ C

To convert these to true air temperatures they must be corrected for the difference in air and particle temperatures shown in Figures 4.3 to 4.5. In Section 6.4 it is shown that the thermocouple immersed in the bed registers a combination of 80% particle and 20% air temperature. Combining this with the results of the theoretical model the air outlet temperatures may be obtained.

The correction depends upon the value of the gas-to-particle heat transfer coefficient, and a reasonable value is 15 W m^{-2} K^{-1}.

Therefore
$$\frac{T_{go} - T_{gi}}{T_p - T_{gi}} = 0.95$$

and the temperature measured at the thermocouple

$$= 0.8\,T_p + 0.2\,T_{go}$$

Hence cold side outlet temperature = 113.7° C

hot side outlet temperature = 122.1° C

7. Heat balance

heat input
$$= \frac{(243.6 - 41.2)\ 2610\ \ 1.196\ \ 1010}{60\ 000}$$

$$= 10\ 640\ \text{W}$$

heat removed from hot air

$$= \frac{(243.6 - 122.1)\ 2610\ \ 1.196\ \ 1010}{60\ 000}$$

$$= 6380\ \text{W}$$

heat transferred to cold air

$$= \frac{(113.7 - 41.2)\ 4454 \quad 1.196 \quad 1010}{60\ 000}$$

$$= 6500 \text{ W}$$

The error between heat removed from the hot air and the amount transferred to the cold is 120W or 2% which is comparable with the experimental error.

$$\text{percentage recovery} = \frac{6500}{10\ 640}$$

$$= 61\%$$

8. Effectiveness.

The effectiveness is calculated according to equation (4.15).

$$\text{Hence } \eta = \frac{(2610 + 4454) \quad (113.7 - 41.2)}{2610 \quad (243.6 - 41.2)}$$

$$= 0.97$$

APPENDIX 3

Proposed large prototype

A proposal was submitted to the Commission for the European Communities in November 1979 for the funding of the building of a large prototype fluidised bed gas-to-gas heat exchanger [146]. This would develop the concept from the work reported in this thesis, and from some small-scale experiments at high temperature which are to be carried out shortly, to a proven system. The prototype would operate in accordance with the recommendations given in Table 8.1, and would be the same shape as the experimental unit used in the present work (Figures 3.1 and 3.2). The bed would be 4 m^2 divided equally between the hot and cold sides which would accept gas at 800 ^oC and 50^oC respectively, the former being supplied by a natural gas burner. The hot and cold mass flows would be 0.9 and 3.1 kg s^{-1} of air respectively and the heat transferred is predicted to be 550 kW, or 77% of that available, if the effectiveness is 0.98.

The large prototype should demonstrate whether or not the heat exchanger can be increased in size successfully. Upon completion of the programme of work on the prototype, the heat exchanger can be demonstrated to be a viable product to prospective customers, and can be marketed. At the time of writing the Commission has agreed in principle to funding this work, though the details of the contract have yet to be finalised. It is hoped that the programme will commence in early 1981 and take two years to complete.

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