TURBULENT FLUIDIZATION AND HEAT TRANSFER
IN A PRESSURIZED FLUIDIZED BED COMBUSTOR
TURBULENT FLUIDIZATION AND HEAT TRANSFER IN A PRESSURIZED FLUIDIZED BED COMBUSTOR

Proefschrift

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Aan Arnout en Tsjibbe
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- the students of different departments and laboratories who participated in the course of their graduate studies and the names of which are found in the list of references;
This thesis deals with the heat transfer in a pressurized fluidized bed combustor, with an emphasis on the transfer of heat to immersed vertical tubes. In order to gain a better understanding of the phenomena, also the gas-particle flow as such is investigated.

From experimental- and theoretical findings it is concluded, that the turbulent fluidization regime applies to pressurized fluidized bed combustors at operating pressure beyond 5 bars. Therefore ample attention is devoted to the physical modeling of that regime.

Experiments and calculations indicate that a turbulent fluidized bed consists of a central region, that is actually turbulently fluidized and a surrounding wall layer, that is minimally fluidized. In the central region porosity waves move upward and induce a net upward motion of the particles. The resulting transport of solid material is compensated for the downward transport of particles in the wall layer.

Fast porosity measurements show that the rise velocity of the porosity waves is rather similar to that of slugs. Contrary to slugs, however, porosity discontinuities and particle free regions do not occur with porosity waves.

To describe the gas-particle flow, the well known Ergun relation for packed beds is modified, so as to make it application to expanded beds. Further the consequences of hydrodynamic dispersion, i.e. the mixing resulting from the zig-zag motion of the gas through the interstices between the particles, are investigated. Starting from the modified Ergun relation a one-dimensional turbulent fluidized bed model is developed, that gives information about the mean velocities of the gas and the particles.

The properties of the porosity waves are investigated by applying the linearized theory of continuity waves and continuity shock waves on the model just mentioned. The results permit conclusions to be drawn about the rise velocity, amplitude and frequency of the porosity fluctuations.

Additionally some information about fluidization regimes is obtained.

Using the turbulent fluidized bed model, a heat transfer model is derived that encompasses four transport mechanisms: convection, radiation, dispersive conduction and boundary layer renewal. Scraping by the moving particles is taken as the cause of the renewal of the boundary layer. To calculate the gas velocity profile in the enhanced wall porosity layer,
the modified Ergun relation is completed by a correction factor yielding zero velocity at the wall.

Measurements taken in a 1.5 MW, 0.485 m diameter pressurized fluidized bed combustor serve a two fold purpose in the investigations. On the one hand they provide some starting points for the development of the theory, on the other hand they permit verification of theoretical predictions. The measurements taken, partly by means of newly developed techniques, concern:

- local and overall pressure gradients along the bed, both average values and fluctuations;
- amplitude, frequency and propagation velocity of the local porosity fluctuations, and
- mean value and dynamical components of the local heat transfer to an immersed surface, and the statistical properties of the dynamical components.

The results of the theoretical turbulent fluidization model and the related heat transfer model are confirmed by the experimental findings and are not in conflict with the scarce data found in literature concerning pressurized fluidization of large particles.

SAMENVATTING

Onderwerp van dit proefschrift is de warmteoverdracht in een wervelbed-drukvuurhaard, in het bijzonder de warmteoverdracht aan verticale pijpen. Om tot een beter begrip van deze warmteoverdracht te komen is bovendien de gasdeeltjesstroming onderzocht. Uit experimentele- en theoretische bevindingen blijkt dat in wervelbeddrukvuurhaarden het turbulente fluidisatie regime heerst, bij een werkdruk boven ongeveer 5 bar. Aan de verdere fysische modellering van dit regime wordt daarom ruime aandacht gegeven. Experimenten en berekeningen wijzen erop dat een turbulent wervelbed bestaat uit een centraal deel dat turbulent gefluidiseerd is, omgeven door een wandlaag die minimaal is gefluidiseerd. In het centrale gedeelte van het bed bewegen zich porositeitsgolven naar boven en induceren daarbij een netto opwaarts gerichte beweging van bedmateriaal. Deze beweging wordt gecompenseerd door het neerwaartse transport van deeltjes in de - bij benadering homogene - wandlaag.

Uit dynamische porositeitsmetingen is gebleken, dat de porositeitsgolven behouden kunnen worden als gedegenereerde, met deeltjes beladen slugs. Om de gas-deeltjesstroming te beschrijven wordt de algemeen bekende Ergun-relatie voor gepakte bedden gemodificeerd. Ook worden de gevolgen van de zogenaamde hydrodynamische dispersie (de menging als gevolg van de quasi turbulente zig-zag beweging van het gas dat door de tussenruimten tussen de deeltjes stroomt), nader onderzocht.

Uitgaande van de gemodificeerde Ergun-relatie wordt een één-dimensionaal model van een turbulent gefluidiseerd bed ontwikkeld, dat informatie geeft over de gemiddelde snelheden van zowel het fluidiserende gas als de gefluidiseerde deeltjes in het centrale turbulente deel en in de wandlaag. Verder kunnen er, uitgaande van de theorie van continuïteitsgolven en continuïteitsschokgolven en door toepassing van een gelineariseerd porositeitsgolfconcept op de continuïteits- en impulsvergelijkingen conclusies getrokken worden met betrekking tot fluidisatie regimes, en de eigenschappen van porositeitsgolven in het bed. Laatstgenoemde eigenschappen betreffen de amplitude, de frequentie en de stijgsnelheid van de porositeitsfluctuaties.

Uitgaande van hun turbulent fluid bed model wordt vervolgens een warmteoverdrachtsmodel afgeleid. Dit model, waarin convectie, straling en dis-persie geleding in beschouwing worden genomen, wordt primair gekarakteriseerd door een wandlaag-verversingsmechanisme. De schrapende werking van langs de wand bewegende deeltjes wordt als de oorzaak van de verversing beschouwd.
Uitgaande van de gomodificeerde Ergun-relatie, die in de nabijheid van de wand gecorrigeerd wordt, wordt het gasnelheidsprofiel in de wandlaag berekend.

De experimentele aanknopingspunten van de ontwikkelde theorieën zijn verkregen uit metingen in een 1,5 MW wervelbeddrukvuurhaard met een inwendige diameter van 0,485 m, uitgerust met een verticale warmtewisselaar. Daarnaast hebben enkele meetresultaten voor verificatie gedien.

De metingen, gedeeltelijk uitgevoerd met nieuw ontwikkelde technieken, betreffen:
- de gemiddelde- en de dynamische component van de drukval in het wervelbed, zowel lokaal als overall;
- de amplitude-, frequentie- en voortplantingsnelheid van lokale porositeitsfluctuaties;
- de lokaal gemiddelde- en de dynamische component van de warmteoverdracht.

De theoretische resultaten van het turbulente fluidisatiemodel en de hieraan gerelateerde warmteoverdrachtsmodel worden bevestigd door de experimentele bevindingen en zijn niet in tegenspraak met de gegevens, die in de literatuur zijn gevonden over de fluidisatie onder druk van grove deeltjes.
CHAPTER 3 TURBULENT FLUIDIZATION AND HEAT TRANSFER

3.1. Introduction
3.2. Turbulent fluidized bed model
3.3. Turbulent fluidization model
  3.3.1. Introduction
  3.3.2. Continuity waves
    3.3.2.1. Introduction
    3.3.2.2. Continuity waves
    3.3.2.3. Continuity shock waves
  3.3.3. Dynamics of gas-particle flow
    3.3.3.1. Introduction
    3.3.3.2. Continuity and momentum equations
    3.3.3.3. Dynamics of gas-particle flow
  3.3.4. Conclusions
3.4. Turbulent fluidized bed heat transfer model
  3.4.1. Introduction
  3.4.2. Turbulent fluidized bed heat transfer model
    3.4.2.1. Introduction
    3.4.2.2. Results and conclusions
  3.4.3. Results and conclusions
    3.4.3.1. Introduction
    3.4.3.2. Time constant
    3.4.3.3. Wall layer properties
    3.4.3.4. The components of turbulent fluidized bed heat transfer
    3.4.3.5. Turbulent fluidized bed heat transfer

CHAPTER 4 EXPERIMENTAL SET-UP

4.1. Introduction
4.2. Pressurized fluidized bed combustor
4.3. Operating the pressurized fluidized bed combustor
  4.3.1. Start-up procedures
  4.3.2. Steady state operation
  4.3.3. Safety precautions
  4.4. Instrumentation
    4.4.1. Introduction
    4.4.2. Local mean heat transfer measurements
    4.4.3. Bed pressure differential analyser
    4.4.4. Local dynamical porosity measurements
      4.4.4.1. Introduction
      4.4.4.2. High temperature impedance probe
    4.4.5. Local dynamical heat transfer measurements
      4.4.5.1. Introduction
      4.4.5.2. Local dynamical heat transfer probe
  4.5. Data acquisition and reduction
    4.5.1. Introduction
    4.5.2. Operating conditions and steady state heat transfer
    4.5.3. Data acquisition of the BPDA
    4.5.4. Data acquisition of the HTIP
    4.5.5. Data acquisition of the LDHTP
  4.6 Data processing
    4.6.1. Introduction
    4.6.2. Data processing of the heat transfer measurements
    4.6.3. Data processing of the BPDA signal
    4.6.4. Data processing of the HTIP signal
    4.6.5. Data processing of the LDHTP signal

CHAPTER 5 EXPERIMENTAL RESULTS

5.1. Introduction
5.2. Porosity fluctuations
  5.2.1. Introduction
  5.2.2. BPDA porosity fluctuations
  5.2.3. HTIP porosity fluctuations
    5.2.3.1. Introduction
    5.2.3.2. Nature of the HTIP signal
    5.2.3.3. APDF of the HTIP signal
    5.2.3.4. Spectral analysis of the HTIP signal
    5.2.3.5. CCF of the HTIP signal
  5.3. Heat transfer
**LIST OF SYMBOLS**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
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<tr>
<td>A</td>
<td>cross sectional area</td>
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<td>Ar</td>
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**Greek symbols**

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<td>thickness</td>
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<td>emission coefficient</td>
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<td>ε</td>
<td>relative permittivity</td>
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<td>thermal conductivity</td>
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<td>υ</td>
<td>dynamic viscosity</td>
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<td>frequency</td>
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<td>specific density</td>
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<td>s</td>
<td>conductivity</td>
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<td>o</td>
<td>Stephan Boltzmann constant</td>
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<td>τ</td>
<td>time constant</td>
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<td>ψ</td>
<td>heat sink</td>
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<td>ω</td>
<td>radial frequency</td>
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**Subscripts**

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<td>a</td>
<td>air</td>
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<tr>
<td>as</td>
<td>asymmetric slug</td>
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<td>B</td>
<td>bubble</td>
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<td>Bb</td>
<td>bubble phase</td>
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<td>bed</td>
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dense phase
distributor
dynamic
effective
fluid
gas convection
hydrodynamic
contact
longitudinal
mixture
minimum fluidization
orifice
particle
packed bed
particle convection
radiative
shock
slug
solid
transversal
turbulent zone
wall
wall layer
wall return layer
wave

Acronyms
ACF Auto Correlation Function
AFBC Atmospheric Fluidized Bed Combustor
APDF Amplitude Probability Density Function
PBDA Bed Pressure Differential Analyser
CCF Cross Correlation Function
FBC Fluidized Bed Combustion
FFT Fast Fourier Transformation
HTIP High Temperature Inpedance Probe
LDHTP Local Dynamical Heat Transfer Probe
MEM Maximum Entropy Method
PFBC Pressurized Fluidized Bed Combustor
RHS Right Hand Side
TZ Turbulent Zone
WLR Wall Return Layer
WR Wall Region
CHAPTER 1 TURBULENT FLUIDIZATION AND HEAT TRANSFER

1.1. Introduction.

In spite of extensive and worldwide research on fluidized bed combustion, the design of fluidized bed combustors depends still largely on empirical knowledge.

To improve the design, operation and control of a fluidized bed combustor, more fundamental information is required about, among others:

- the fluid dynamical nature of the gas particle flow
- the heat- and mass transport, including mixing of both the fluidizing fluid and the fluidized particles and
- the chemical reaction kinetics.

Such knowledge would permit the solution of practical problems such as:

- scaling-up of fluidized beds
- economic optimization taking combustion efficiency and heat exchanger configuration into account
- reducing the emission level of SO\textsubscript{x}, NO\textsubscript{x} and fly-ash and
- operating and controlling of such combustion system.

The main topics of this thesis are the dynamics of gas particle flow and heat transfer to tubes immersed in a pressurized fluidized bed combustor. In the first chapter, the historical development of fluidization is briefly reviewed, after which some aspects of fluidized bed combustion in connection with power generation are touched upon and the objective of this thesis is formulated.

The classical "bubbling" fluidized bed models and the related heat transfer are scrutinized in Chapter 2.

Based on a fundamental approach a new turbulent fluidized bed model, a theory concerning the dynamics of gas particle flow and a turbulent fluidized bed heat transfer model are developed in Chapter 3. This approach includes wave theory, hydrodynamic dispersion, porosity waves and a surface renewal mechanism on particle size scale. Related to this approach, some elementary gas particle flow properties, leading to amongst others a modified Ergun equation, are discussed in Appendix A2.

In Chapter 4 the testrig and some standard and newly developed measurement techniques are described, that served the experimental investigation of
the average and dynamical aspects of the turbulent fluidization regime and its heat transfer. The results of the experiments are presented in Chapter 5. And finally, in Chapter 6 the theoretical and experimental results are compared with each other and literature.

1.2. Historical review.

Although the principles of fluidization were discovered in Germany in the 1920’s and the first application, in cat cracking, was introduced in the 1940’s, it lasted till the 1960’s before combustion of coal in a fluidized bed got reasonable interest. However, owing to the low prices of natural gas and oil, which were easy to obtain and to use, and the expectations regarding nuclear power in those days, the interest in the firing of coal decreased and, consequently, most fluidized bed combustion research was cancelled. Only the British Coal Research Association continued its research on the burning of low grade coals and waste materials. In 1968 the first pressurized fluidized bed combustor was erected at Leatherhead, ROY (1969).

Due to the increasing concern about air quality in the late 1960’s and the oil crises of 1973 and 1979, research on different types of energy systems, including fluidized bed combustion, was undertaken on a larger scale, see BOSSMA (1977), PATTERSON (1978) and VERBEEK (1978). Research on fluidized bed combustion in The Netherlands dates back to 1965, VAN KOPPEN (1965). In 1976 professor Stuart Mitchell started an investigation at Delft University of Technology to apply this technique at pressurized conditions to fire heavy, vanadium containing oil residues in marine gas turbines, but a few years later, combustion of coal became of major interest. Also in those years, a national research program for coal fired combustion systems, including both atmospheric and pressurized fluidized bed combustion, were proposed, DRAYER (1981). However, due to the economic recession, the energy research funding had to be restricted to short term goals, among which only atmospheric fluidized bed combustion. As a consequence, only the Delft University of Technology is presently executing research on pressurized fluidized bed combustion in The Netherlands. This in spite of the fact, that its expected capital and running costs are lower than for any other energy conversion system with desulphurization equipment, see BOSSMA (1978) and ROY (1983). Elsewhere pressurized fluidized bed combustion pilot plants have been erected amongst others by Curtiss Wright, MOSKOVITZ (1978) and at Grimethorpe, BROADBENT (1975) and ROY (1983), and at the Aachen University of Technology, ANONYMUS (1982), all with substantial government support. Recently PODOLSKI (1983) has reviewed the state of the art of the pressurized fluidized bed combustion technology.

1.3. Present views and research on fluidized bed combustion.

Characteristics of fluidized bed combustion are the high volumetric heat capacity of the bed (≈ 1 MJ/m$^3$ K), the well mixed conditions of the bed and, for solid fuel, a long residence time of the particles. Due to these properties, a fluidized bed combustor is thermally very stable, operates at nearly isothermal conditions and permits almost every type or combination of gaseous, liquid or solid fuel to be fired, even at a relatively low temperature of about 1150 K. As a consequence this low combustion temperature, the NOx emission level is low and relatively low quality construction materials can be used. Further the fluidized bed heat transfer rates are high, which makes the combustor compact compared to for example pulverized coal combustors. Finally it is very important that SO2 and possibly even V2O5, NEWSTEAD (1972), can be trapped by adding calcium containing minerals or artificial sorbents to the fluid bed.

Pressurized fluidized bed combustion permits an even more compact design than atmospheric combustion, and may, with advanced hot gas clean-up, be incorporated in highly efficient gas turbine power generating cycles, ROY (1983) and figure 1-1. Much of the current work on fluidized bed combustion concentrates on the achievement of higher combustion efficiencies at sufficient pollutant capture. Due to the nearly perfect mixing of the solids in a fluidized bed and the occurrence of gas by pass at the usual fluidization velocities, high combustion and/or sulphur capture efficiencies are only achievable in deep beds. Alternatively circulating the solid particles in the bed and/or recirculating the entrained particles can contribute to this end. Internal circulation in the bed can be obtained by non-uniform air supply distribution, Ormston, see PAYNE (1980) and STAB (1983). It may serve an increased residence time of small particle if fed into the down coming region of such a bed and, possibly, the firing of very low grade fuels, see figure 1-2.

A recirculating fluidized bed combustion system, in which for example the fly-ash from the first cyclones is fed back into the bed, STEVEN (1983),
FIGURE 1-1.
Simplified schematic of a pressurized fluidized bed combustor.

FIGURE 1-2.
Schematic of a circulating fluidized bed combustor.
High local air supply (relatively large $M_a$) causes bed content to circulate.
After Ormston, see PAYNE (1980) and STAB (1983).

FIGURE 1-3.
Schematic of a recirculating fluidized bed combustor, after STEVEN (1983).

provides a more complicated but, as yet, more certain way to achieve these effects, see figure 1-3. Supplying the fly-ash to a carbon-burn up cell, is losing interest now, because of the disadvantage that a second combustion system is added to the primary one.

An environmental problem of fluidized bed combustion, which is still waiting for a solution, is the waste disposal. The fly-ash and any superfluous material taken out of the bed, are not vitrified. Consequently excess due to lixiviation of these two by products, the environment may be polluted by trace elements, among which heavy metals.

As regards the construction of the combustor, the corrosion and erosion problems of heat exchanging tubes immersed into the bed, are not fully solved as yet, PATTERTON (1978) and POBOLSKI (1983).

These problems do arise from attrition by the fluidized particles and from trace elements in the fuels and additives used, and are presumably aggravated by thermal stress.

The opinion sometimes expressed, that the response time of a fluidized bed combustion system is too large to meet the required flexibility of a thermal power generation station, has recently been falsified for pressurized fluidized bed combustors by KOOL (1983). However, widening the control range of fluidized bed combustors is an issue still worth pursuing.
1.4. Objective.

The objective of this thesis is to contribute to the understanding of fluidization and heat transfer in a pressurized fluidized bed combustor. To this purpose the fluidization and heat transfer are modeled in terms of the turbulent fluidization regime. The resulting models are based on a fundamental approach of the gas-particle flow, including amongst others porosity waves, hydrodynamic dispersion, a modified Ergun equation, and for the heat transfer a boundary layer renewal mechanism.

To verify the theoretical assumptions, a 1.5 MW pressurized fluidized bed combustor has been erected and new measurement techniques have been developed and applied to obtain information about the just mentioned phenomena, both as regards mean and dynamical aspects.

Remark.

It should be noted, that quite a number of acronyms are used in this thesis, see the list of symbols.

CHAPTER 2 FLUIDIZATION AND HEAT TRANSFER

2.1. Introduction.


In literature two ways are found to describe the flow phenomena in fluidized beds:

- purely empirical black box approaches, and
- bubble models, in which the parameters are semi-empirical functions of the operating conditions.

In the next sections only the latter approach is taken into consideration. Firstly the classical bubble models are shortly reviewed. Afterward its two modified versions, the so-called fast- and slow bubble models, are critically discussed.

Subsequently some general properties of slugging fluidized beds are given. As regards fluidized bed heat transfer, two groups of correlations can again be distinguished in literature:

- purely empirical heat transfer correlations, and
- semi-empirical heat transfer correlations, which are composed of a particle convective, a gas convective and a radiative component.

Only the last mentioned approach is relevant for the approach taken in this thesis, and will be discussed here.
2.2. Gas solid fluidization.

2.2.1. Introduction.

Fluidized bed flow phenomena can be described in terms of homogeneous (particulate or bubble free) or heterogeneous (aggregative or bubbling) fluidization, see amongst others DAVIDSON (1963), (1971) and KUNII (1969). Homogeneous fluidization is specific for small, cohesive particles with a size below 0.1 mm, which are fluidized in the lower part of their fluidization velocity range. In such circumstances cohesive particle forces stabilize the bed. In other words a locally generated porosity disturbance dies out due to "damping" forces, RIETEMA (1973), (1982). As these circumstances do not occur in PFBC's, homogeneous fluidization is not discussed here.

Heterogeneous fluidization starts, when destabilizing forces become dominant. Local porosity disturbances then grow and manifest themselves in voids, commonly designated as bubbles or slugs. Slugs have a horizontal dimension of the same order of magnitude as the bed diameter.

The properties of single, isolated bubbles or slugs and the mechanism(s), which determine(s) their maximal size are more or less known. See for a compilation DAVIDSON (1971). However, the way in which bubbles in a swarm coalesce, split up, and interact with the surrounding dense phase is at present not fully understood, BAR-COHEN (1978) and RIETEMA (1982). Consequently all efforts to describe heterogeneous gas-solid fluidization from first fluid mechanical principles have failed, as yet.

The semi-empirical "bubbling" fluidized bed models presented in literature, suffer from similar imperfections and shortcomings. Because of this and as a consequence of the uncertainty on the influence of the bed diameter, it is hazardous to apply the published experimental results for scaling up. Bubble models can only be used to scale up, if the maximum bubble size is an order of magnitude smaller than the bed diameter, see amongst others WERTHER (1972). Slugs escape scaling up rules nearly completely, see e.g. Hovmand, see DAVIDSON (1971).

Nevertheless, the modelling of gas particle flow has been strongly improved, due to the numerous experimental efforts in the past decade.

These have resulted amongst others in fluidization charts, permitting a rough prediction of the fluid dynamical behaviour of the fluidized bed. Most notable are the powder classification diagram of GELDART (1973), which has recently been improved by MOLERUS (1982b) and the fluidization regime charts of REH (1977), BAR-COHEN (1977), STRAUB (1978) and VAN DEEMTER, see GRACE (1980). Following the classification of Geldart, the bed content of fluidized bed combustors, predominantly sandlike particles with a specific density of 1000 to 2600 kg/m³ and a size up to about 1 mm, fluidizes as so-called B- and D-powders. Starting from these fluidization charts (see LA NAUZÉ (1982) for a review), the "bubbling" bed models, dominating in current literature are discussed in this section. The approaches are the fast- and the slow bubble model, both modified classical bubble models, and the slug flow model.

Together these models cover most of the fluidization conditions of practical interest in PFBC. The adjustments required to make them applicable to a fluidized bed combustor, are discussed in the sections 2.2.2 and 2.2.3.

However, first the fundamentals of the models are given. The fast- and also the classical bubble approach are characterized by a substantially particle free void (bubble) surrounded by a so-called cloud and followed by a wake. The rise velocity of the void is larger than the interstitial gas velocity in the surrounding dense phase, BAR-COHEN (1978), (1981). Characteristic for the slow bubble is, again consisting of a substantially particle free bubble void with a wake, that the gas velocity in the bubble exceeds both the rise velocity of the visible void and the interstitial gas velocity in the surrounding dense phase, BAR-COHEN (1977), (1981). Most authors assume the dense phase to be minimally fluidized, but some assume a moderate expansion.

Due to coalescing and/or dense phase degassing, the bubble diameter generally increases with height above the air distributor. When the bubble diameter becomes of the order of magnitude of the bed diameter, the so-called slugs develop. In beds consisting of particles of low density, the slugs are essentially particle free and the dense phase moves downward on either edge of the void. In beds consisting of large particles of high density, however, particles rain downwards through the slug. This phenomenon can be interpreted as a first indication of the occurrence of porosity waves in fluidized bed combustors, as discussed in Chapters 3. As far as presently known for both types of slugs, the surrounding dense phase is minimally fluidized. Because the boundary delimitation between the two types of slugs is not very sharp, the designation "slug flow regime" usually covers bot types.
2.2.2. Modelling of bubbling fluidized beds.

2.2.2.1. Introduction.

To analyse gas-particle fluidization, the counter current model of VAN DEEMTER (1967) is widely used. A counter current model is characterized by a lean phase, consisting of rising bubbles or voids and a dense phase, which is taken to be minimally fluidized. The rising bubbles cause an upwards drift of solid particles, whereas the solids in the dense phase are moving downwards. Originally VAN DEEMTER (1967) proposed a fluidized bed to encompass one single circulation (counter current) cell. Recently RIETEMA (1982) has shown a fluidized bed model encompassing several circulation cells to be closer to reality, if the smallest cell size is made to correspond to a single void and its cloud and wake. Historically the main application of the counter current model and the fast and slow bubble model was the modelling of cat crackers and therefore the applicability of these models for fluidized bed combustors (FBC) was doubtful from the start, in particular because of the much larger particle size in FBC's.

In order to establish the limits of applicability more precisely, all three models are subsequently applied to a FBC in the next sections. Also some modifications are tried out. From the results the shortcoming of these classical models for the analysis of FBC's are deduced.

Finally some reasons are given why the modelling of FBC's in terms of turbulent fluidization is to be preferred.

It is remarked that in the analyses silica sand with a particle size of about 1 mm is taken as representative for the solid material in FBC's.

2.2.2.2. Bubble model.

In most classical bubble models, the bubbles are assumed to be particle free and the wake porosity to be equal to the dense phase porosity.

Further the rise velocity of the gas and the solids in the wake is taken equal to that of the bubble. So the (mean) mass balances of the fluidizing gas and the fluidized particles over a cross section of the fluidized bed read (see figure 2-1, total upward and downward solid flows are equal):

\[(1 + a \epsilon_g) \epsilon_B U_B + (1 - \epsilon_B(1+a)) \epsilon_d U_d = J\]

(2-1)

\[(1 - \epsilon_d) a \epsilon_B U_B + (1 - \epsilon_B(1+a)) (1 - \epsilon_d) V_d = 0\]

(2-2)

For reasons of completeness, it is remarked, that for particle laden bubbles with an internal bubble porosity \(\epsilon_{B_1}\) the mass balance read slightly different:

\[\epsilon_B \left( \epsilon_{B_1} + a \epsilon_d \right) U_B + \left(1 - \epsilon_B(1+a)\right) \epsilon_d U_d = J\]

(2-3)

and:

\[\epsilon_B \left(1 - \epsilon_{B_1} + a(1-\epsilon_d)\right) U_B + \left(1 - \epsilon_B(1+a)\right) (1 - \epsilon_d) V_d = 0\]

(2-4)

However, YOSHIDA (1978) observes that \(\epsilon_{B_1}\) has to be larger than 0.95, because only then the bubble is transparent. As a consequence final results from (2-3) and (2-4) are nearly identical to those from (2-1) and (2-2). Therefore (2-1) and (2-2) are used for further consideration.
To apply (2-1) and (2-2), the various quantities have to be specified. The wake fraction, i.e. the volume ratio of the wake and the particle free fraction bubble void, is defined as (KUNII (1969)):

\[ a = \frac{\text{Vol}_W}{\text{Vol}_B} \quad (2-5) \]

For bubble swarms GELDART (1973) recommends a wake fraction of 0.2. Such to circumvent excessively high calculated solid flow rates.

Starting from the experimental results of WERTHER (1976), VAN DER POST (1979) has derived an equation for the dense phase porosity, including dense phase degassing. It reads:

\[ \varepsilon_d = 0.41 + 0.0155 \left( \frac{J - J_{mf}}{J_{mf}} \right) - 3h/H_{mf} \]

The slip relation used in nearly all bubble models reads:

\[ U_d = V_d = \frac{J_{mf} - J}{J_{mf}} \quad (2-7) \]

In older bubble models the bubble rise velocity is set equal to (after DAVIDSON (1963)):

\[ U_B = (1 \pm 1.2) (J - J_{mf}) + 0.71 (g D_B)^{0.5} \]

(2-8)

Following the conclusions of BAK (1979) regarding the semi-empirical correlations for the bubble diameter, for further calculations the correlation of WERTHER (1976) is preferred. This correlation reads:

\[ D_B = 0.0033 \left[ 1 + 2.73 (J - J_{mf})^{0.33} \left( 1 + 6.4 h \right)^{1.21} \right] \]

(2-9)

It should be noted that equations (2-5), (2-6), (2-8) and (2-9) were only verified for small particles, say \( d_p \leq 0.2 \) mm.

The minimum superficial fluidization velocity is calculated by the well known Ergun equation, see Appendix A2.

Solving equations (2-1), (2-2) and (2-5) through (2-9) for the fluidization of large particles leads to unacceptable results. The bubble phase porosity becomes nearly equal to 1.0 and the bubble point frequency, which is calculated from the general identity:

\[ \nu_b = 1.5 \frac{\epsilon_B U_B}{D_B} \]

(2-10)

becomes an order of magnitude larger than the value observed by CRANFIELD (1974), see (2-14).

To prevent these unrealities, the approach may be modified by multiplying the first RHS term of equation (2-8) by a coefficient \( \gamma \), \( \gamma > 1.2 \). Herewith the model is transformed into a fast bubble model, further described in the following section.

Another way to solve this set of equations is to replace (2-9) by the general identity:

\[ \left( 1 - \epsilon \right) = \left( 1 - \varepsilon_B \right) \left( 1 - \varepsilon_d \right) \]

(2-11)

and to apply the semi-empirical bed expansion equation of BABU (1978) which reads:

\[ \epsilon = 1 - 0.6 \left[ 1 + \frac{8.4 (J - J_{mf})^{0.738} d_p 1.006 0.376}{D_B^{0.937} 0.126} \right] \]

(2-12)

However, this modification brings the bubble rise velocity up to the order of magnitude of the terminal velocity of the fluidized particle. Such a bubble rise velocity would go with spouting-like phenomena which have not been experimentally observed.

### 2.2.3. Fast bubble model

As already indicated in the preceeding section various authors (VAN DER POST (1980), VAN VREKE (1980) and DAVIDSON (1980)) have modified the classical bubble model by applying a coefficient \( \gamma \), \( \gamma > 1.2 \) to the first term of the RHS of equation (2-8). The equation for the bubble rise velocity then reads:

\[ U_B = \gamma (J - J_{mf}) + 0.71 (g D_B)^{0.5} \]

(2-13)

and the model is transformed into a so-called fast bubble model. Remark: DAVIDSON (1980) notes that equation (2-13) can often be approximated by the simple expression \( \varepsilon_B U_B = J - J_{mf} \).

To illustrate the ins and outs of the fast bubble model, it is elaborated in two ways in this section.
For silica sand with a particle size of about 1 mm, the equations (2-1), (2-2), (2-5), (2-6), (2-7), (2-9) and (2-13) only lead to reasonable results, if it is assumed, that $2.5 < Y < 6.0$, VAN DER POST (1979) and VAN VREDE (1980). For $Y < 2.5$ the bubble phase porosity becomes larger than 0.5 and the bubble point frequency becomes an order of magnitude too large. For $Y > 6.0$ the bubble rise velocity becomes of the order of 10 m/s, which is unrealistically high and would go with spouting like phenomena.

However, keeping $Y$ within the range indicated does not lead to completely satisfactory results as well. Tables 2-1 and 2-2 illustrate this point for the case of $Y = 3.5$, as elaborated by VAN VREDE (1980).

Even with the adapted value of $Y$, it appears from table 2-1, that the (downward) dense phase solid velocity is about 0.5 m/s. Locally and momentarily such a high solid velocity may be possible, but as a mean value it is an order of magnitude too large.

Further at high superficial fluidization velocities, the bubble rise velocity exceeds the terminal velocity of the particles by a factor of 2, which is unrealistic. Such high bubble rise velocities, which would go with extreme upward solid mass flows, have never been observed and also look unrealistic from the point of view of slugging, see section 2.2.3.

TABLE 2-1.
Fast bubble model properties of a fluidized bed as a function of height at a superficial fluidization velocity of 2 m/s, after VAN VREDE (1980).

<table>
<thead>
<tr>
<th>$H$ (m)</th>
<th>$e$</th>
<th>$U_B$ (m/s)</th>
<th>$D_B$ (m)</th>
<th>$\epsilon_B$</th>
<th>$V_B$ (m/s)</th>
<th>$U_d$ (m/s)</th>
<th>$V_d$ (m/s)</th>
<th>$\epsilon_d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>0.60</td>
<td>5.8</td>
<td>0.056</td>
<td>0.28</td>
<td>44</td>
<td>0.75</td>
<td>-0.50</td>
<td>0.84</td>
</tr>
<tr>
<td>0.3</td>
<td>0.58</td>
<td>6.0</td>
<td>0.11</td>
<td>0.28</td>
<td>23</td>
<td>0.76</td>
<td>-0.49</td>
<td>0.42</td>
</tr>
<tr>
<td>0.5</td>
<td>0.57</td>
<td>6.2</td>
<td>0.18</td>
<td>0.27</td>
<td>13</td>
<td>0.77</td>
<td>-0.48</td>
<td>0.41</td>
</tr>
<tr>
<td>0.7</td>
<td>0.56</td>
<td>6.4</td>
<td>0.25</td>
<td>0.26</td>
<td>10</td>
<td>0.77</td>
<td>-0.47</td>
<td>0.41</td>
</tr>
</tbody>
</table>

TABLE 2-2.
Fast bubble model properties of a fluidized bed as a function of superficial fluidization velocity at a height of 0.3 m, after VAN VREDE (1980).

<table>
<thead>
<tr>
<th>$J$ (m/s)</th>
<th>$e$</th>
<th>$U_B$ (m/s)</th>
<th>$D_B$ (m)</th>
<th>$\epsilon_B$</th>
<th>$V_B$ (m/s)</th>
<th>$U_d$ (m/s)</th>
<th>$V_d$ (m/s)</th>
<th>$\epsilon_d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>0.58</td>
<td>2.4</td>
<td>0.07</td>
<td>0.26</td>
<td>14</td>
<td>1.0</td>
<td>-0.19</td>
<td>0.41</td>
</tr>
<tr>
<td>2.0</td>
<td>0.58</td>
<td>6.0</td>
<td>0.11</td>
<td>0.28</td>
<td>23</td>
<td>0.64</td>
<td>-0.49</td>
<td>0.42</td>
</tr>
<tr>
<td>4.0</td>
<td>0.59</td>
<td>13</td>
<td>0.14</td>
<td>0.28</td>
<td>39</td>
<td>0.16</td>
<td>-1.1</td>
<td>0.43</td>
</tr>
</tbody>
</table>

And finally an increasing bubble point with increasing superficial fluidization velocity is not in agreement with literature, see amongst others DARTON (1977). Combining the mass balances (2-1) and (2-2) with (2-7), (2-10), (2-11), (2-12) and (2-13) and starting from the bubble point frequency as given by CRANFIELD (1974), the fast bubble model may also be solved, according to Cranfield:

$$v_B = 0.61 h^{-0.72}$$

With equation (2-14) and the just mentioned set of equations, $D_B$ and $Y$ become the unknowns to be calculated. The results are presented in table 2-3.

This table shows, that the bubble rise velocity and the dense phase solid velocity are again too large and that the bubble diameter takes unacceptable values. Further the bubble rise velocity is constant, which is improbable for varying frequency and diameter.

To the author's opinion, see BOELENS (1982), the problems with the fast bubble approach can be partially circumvented by uncoupling the bubble gas- and bubble solid transport. This leads to the slow bubble approach as will be discussed in the following section.
TABLE 2-3.
Alternative fast bubble model properties of a fluidized bed as a function of heights.
Operating conditions: Ambient, \( J = 2 \text{ m/s}; J^\prime \gg 0.5 \text{ m/s}; \)
\( d_p = 910 \text{ \( \mu \)m}; h^\prime = 0.5 \text{ m}; \) \( \varepsilon_{mf} = 0.40; \varepsilon = 0.56; \varepsilon_B = 0.26; \)
\( U_B = 6.3 \text{ m/s}; U_d = 0.84 \text{ m/s}; V_d = -0.45 \text{ m and} \alpha = 0.2. \)

2.4. Slow bubble model.
KUNII (1969) has indicated that the gas balance of the fast bubble model can be improved by assuming the gas to use the bubbles as fast by-passes or short cuts. This approach leads to the slow bubble concept. BAR-COHEN (1977), (1981), BOELENS (1982) and VAN SWAAIJ (1983).
Bar-Cohen states that for the sake of a realistic gas balance, the fast- and slow bubble concept should be used for respectively the small \( (d_p = 0.1 \text{ mm}) \) and large \( d_p = 1 \text{ mm} \) particle size range.
Recently Van Swaalj has experimentally verified in a two dimensional bed, that for sand like particles at ambient conditions the fast slow bubble transition occurs with a size of about 460 \( \mu \)m. For particles with a larger size, a coloured tracer gas was demonstrated to meander upward from void to void.
Bar-Cohen's model is incomplete, as far as no mass balance is given for the solids. Further it is incorrect as regards the overall porosity; in contrast to experimental results of SEKI (1980) and LI (1981), Bar-Cohen finds this porosity to be a function of height.
BOELENS (1982) distinguishes between bubble gas- and bubble void rise velocity, which lead to the following stationary mass balances over a cross section of the fluidized bed, see figure 2-2.

\[
(1 + \alpha \varepsilon_d) \varepsilon_B U_B + (1 - \varepsilon_B(1+\alpha)) \varepsilon_d U_d = J \tag{2-15}
\]
and:
\[
\alpha (1 - \varepsilon_d) \varepsilon_B V_B + (1 - \varepsilon_B(1+\alpha)) (1 - \varepsilon_d) V_d = 0 \tag{2-16}
\]
The internal bubble porosity is taken equal to 1.0 (see equations 2-3 through 2-4).

The bubble void rise velocity is taken equal to:
\[
V_d = J - J^\prime + 0.71 \left( g D_B \right)^{0.5} \tag{2-17}
\]
The dense phase porosity is 0.4. And according to CRANFIELD (1974), for large particles the wake fraction of the bubble is 0.1.
The results of Boelens' approach, which are obtained by combining the mass balances (2-15) and (2-16) with (2-7), (2-10), (2-11), (2-12), (2-14) and (2-17), are given in tabel 2-4 and 2-5.
TABLE 2-4. Slow bubble properties of a fluidized bed as a function of height, after BOELENS (1982).
Operating conditions: Ambient; \( \dot{J} = 1.0 \text{ m/s} \); \( \dot{J}_{mf} = 0.38 \text{ m/s} \); \( d_p = 750 \mu \text{m} \); \( \alpha = 0.1 \); \( H_{mf} = 0.5 \); \( \epsilon_d = 0.4 \); \( \epsilon = 0.5 \); \( \epsilon_B = 0.17 \).

<table>
<thead>
<tr>
<th>( h )</th>
<th>( U_B )</th>
<th>( V_B )</th>
<th>( D_B )</th>
<th>( V_B )</th>
<th>( U_d )</th>
<th>( V_d )</th>
</tr>
</thead>
<tbody>
<tr>
<td>m</td>
<td>m/s</td>
<td>m/s</td>
<td>m</td>
<td>mm</td>
<td>m/s</td>
<td>m/s</td>
</tr>
<tr>
<td>0.1</td>
<td>4.04</td>
<td>1.34</td>
<td>0.11</td>
<td>3.2</td>
<td>0.92</td>
<td>-0.027</td>
</tr>
<tr>
<td>0.3</td>
<td>4.05</td>
<td>1.87</td>
<td>0.31</td>
<td>1.5</td>
<td>0.91</td>
<td>-0.038</td>
</tr>
<tr>
<td>0.5</td>
<td>4.07</td>
<td>2.30</td>
<td>0.57</td>
<td>1.0</td>
<td>0.90</td>
<td>-0.047</td>
</tr>
<tr>
<td>0.6</td>
<td>4.07</td>
<td>2.50</td>
<td>0.72</td>
<td>0.9</td>
<td>0.90</td>
<td>-0.051</td>
</tr>
</tbody>
</table>

TABLE 2-5. Slow bubble model properties as a function of superficial fluidization velocity, after BOELENS (1982).
Operating conditions: Ambient; \( h = 0.3 \text{ m} \); \( \dot{v}_s = 1.5 \text{ s}^{-1} \); \( \dot{J}_{mf} = 0.38 \text{ m/s} \); \( d_p = 750 \mu \text{m} \); \( \alpha = 0.1 \); \( H_{mf} = 0.5 \); \( \epsilon_d = 0.4 \); \( \epsilon = 0.5 \).

<table>
<thead>
<tr>
<th>( J )</th>
<th>( \epsilon )</th>
<th>( U_B )</th>
<th>( V_B )</th>
<th>( D_B )</th>
<th>( \epsilon_B )</th>
<th>( U_d )</th>
<th>( V_d )</th>
</tr>
</thead>
<tbody>
<tr>
<td>m/s</td>
<td>-</td>
<td>m/s</td>
<td>m/s</td>
<td>m</td>
<td>-</td>
<td>m/s</td>
<td>m/s</td>
</tr>
<tr>
<td>1.0</td>
<td>0.50</td>
<td>4.1</td>
<td>1.9</td>
<td>0.31</td>
<td>0.17</td>
<td>0.91</td>
<td>-0.038</td>
</tr>
<tr>
<td>2.0</td>
<td>0.57</td>
<td>6.1</td>
<td>4.0</td>
<td>1.1</td>
<td>0.28</td>
<td>0.79</td>
<td>-0.16</td>
</tr>
<tr>
<td>4.0</td>
<td>0.64</td>
<td>9.9</td>
<td>7.5</td>
<td>3.0</td>
<td>0.40</td>
<td>0.42</td>
<td>-0.53</td>
</tr>
</tbody>
</table>

These tables indicate that for increasing height and/or superficial fluidization velocity, the bubble diameter becomes rather large. Except for very large bed diameters, such bubbles indicate slugging behaviour. The bubble gas- and bubble void rise velocity are about a factor 2 lower than the bubble rise velocity in the fast bubble approach, but remain rather high from the point of view of terminal velocity and slugging. (The bubble gas rise velocity takes even higher values, if the dense phase porosity is tentatively to some extent in order to improve the model).

2.2.3. Slugging.

All three bubble models treated in the foregoing exhibit amongst others (too) large bubble diameters at greater heights in the bed. When the dimensions of a bubble become larger than about a quarter of the bed diameter, then drag forces along the wall of the bed govern the bubbling behaviour and slugging sets in, BAEGENS (1974).

In practice slugging is generally avoided because of the large pressure fluctuations and mechanical loads and the poor chemical conversions going with it. Consequently the related research mainly concerns the onset of slugging.

In literature only few data are available on the properties of slugging fluidized beds and no satisfactory slugging model has been developed as yet. The most interesting data are summarized below; all of them were obtained at ambient pressure and temperature.

Based on experiments BAEGENS (1974) gives to indicate slugging a visual affirmed pressure fluctuation criterium. As a rule of thumb, a bed is slugging, if the pressure fluctuations constitute 5-10 percent of the pressure drop through the bed. Inspirated by Hovmand (see DAVIDSON (1971)), BAEGENS (1974) gives the following interesting equation.

Slugging always occurs, if the bed height satisfies:

\[ H > H_s = 1.34 D_b^{0.175} \]  

(2-18)

For smaller bed heights slugging occurs, if the superficial velocity satisfies:

\[ J_s \geq J_{mf} + 0.07 \left( g D_b^{0.5} + 0.16 \left( H_s - h \right) \right) \]  

(2-19)

This equation actually predicts, that nearly every bed is slugging. Another consequence of equation (2-19) is, that in terms of excess air \( J_s = J - J_{mf} \) slugging is independent of operating conditions.

According to Baeyens the slug frequencies amount to:
\[ v_s = 0.61 \frac{b}{D_b}^{0.143} \]

if \( h \geq H_s \) \hspace{1cm} (2-20)

and:

\[ v_s = 0.86 (H - h) + 0.61 \frac{b}{D_b}^{0.143} \]

if \( h < H_s \) \hspace{1cm} (2-21)

and finally BAEYENS (1974) has indicated, that for PFBC bed material \([d = 1.0 \text{ mm} \text{ and } p > 1000 \text{ kg/m}^3]\), the rise velocity of the so-called asymmetric slug should be applied. This slug rise velocity reads:

\[ U_{as} = J - J_{mf} + 0.35 \left( \frac{2 g D_b}{b} \right)^{0.5} \] \hspace{1cm} (2-22)

Due to the lack of information about the dimensions and the solid transport mechanism of slugs, see amongst others BROADHURST (1975), no slug model can be presented. Besides it should be remarked, that the slugging behaviour of a fluidized bed is, as indicated by the equation given in this section, strongly dependent of the bed diameter.

Applying the afore mentioned correlations to the PFBC, in which our experiments were executed, meets with the difficulty, that the bed diameter is not uniquely defined here. The diameter of the area between the heat exchanger tubes (0.09 m), as well as the diameter of the free central part (0.18) or the vessel diameter (0.185) may be considered as representative in this respect. To circumvent this difficulty \( J_s \), \( V_s \) and \( U_{as} \) were calculated from equations (2-19), (2-21) and (2-22) for each of these values.

The results are presented in table 2-6.

<table>
<thead>
<tr>
<th>( D_b ) m</th>
<th>( J_s ) m/s</th>
<th>( V_s ) m/s</th>
<th>( U_{as} ) m/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.09</td>
<td>0.24</td>
<td>1.36</td>
<td>1.36</td>
</tr>
<tr>
<td>0.18</td>
<td>0.28</td>
<td>1.38</td>
<td>1.55</td>
</tr>
<tr>
<td>0.485</td>
<td>0.38</td>
<td>1.44</td>
<td>1.98</td>
</tr>
</tbody>
</table>

TABLE 2-6.

Slugging properties as a function of (bed) diameter at PFBC conditions as applied in this thesis.

Operating conditions: \( T_b = 1150 \text{ K}, d_p = 780 \text{ mm}, p = (0.4 - 0.9) \text{ MPa}, J = 1.05 \text{ m/s and } J_{mf} = 0.19 \text{ m/s} \).
2.3 Fluidized bed heat transfer.

2.3.1. Introduction.

In literature the fluidized bed heat transfer is described in literature primarily as a function of superficial fluidization velocity, particle size and operating pressure.

As regards the influence of superficial fluidization velocity on heat transfer, generally three regions are distinguished, see Gel'perin (DAVIDSON (1971)) and figure 2-3:

I the packed bed region where the heat transfer coefficient is low and essentially constant;

II the region of beginning fluidization where the heat transfer coefficient rises rapidly up to a maximum;

III the region of intermediate fluidization with a decreasing heat transfer coefficient as a consequence of increasing void fraction.

At high superficial fluidization velocities, sometimes a fourth region is distinguished, characterized by a gradual increase of heat transfer due to an increasing gas convective component. In a review, AINSHEIN (1966) concludes, that only the correlations for the maximum of the heat transfer coefficient are reliable. Nusselt correlations for other regions show mutual discrepancies. The usually bad defined fluidized bed conditions are responsible for these discrepancies (WUNDER (1980)).

As regards the influence of particle size, the three characteristic regions are best distinguished according to the grain classification of GELDART (1973) (and see MOLERUS (1982a)) and see figure 2-4:

α the small particle size region, so-called C and (fine) A powders. For decreasing particle size fluidization and consequently heat transfer are suppressed due to increasing cohesive interparticle forces (RIETEMA (1973));

β the medium particle size region, so-called B and (fine) D powders, where the surface renewal process governs the so-called "particle convective" heat transfer;

γ the large particle size region, so-called (coarse) D powders, where the gas convective heat transfer becomes dominant because of high interstitial gas velocities.

![Figure 2-3](image)

**FIGURE 2-3.**
General trend of the heat transfer coefficient as a function of superficial fluidization velocity after Gel'perin, see DAVIDSON (1971).

![Figure 2-4](image)

**FIGURE 2-4.**
Maximum heat transfer coefficient as a function of particle size after WUNDER (1980).
For fluidized bed combustion, the regions $\alpha$ and $\gamma$ apply. It should be mentioned that the particle size distribution has a significant influence on the heat transfer, WUNDER (1980) but systematic knowledge is lacking here.

The influence of pressure on fluidized bed heat transfer is illustrated in figure 2-5, after BOTTERILL (1975). At increasing pressure, the dynamical nature of the gas particle flow changes, such as to give a more homogeneous particle packing and a more intensive particle flow close to the surface. Also the gas convective heat transfer becomes of increasing importance.

![Figure 2-5](image)

**FIGURE 2-5.** Heat transfer coefficient as a function of superficial fluidization velocity for different operating pressures, after BOTTERILL (1975).

Bed material: copper shot, size 625 μm.

In semi-empirical approaches, the heat transfer coefficient of an immersed tube in a fluidized bed is usually taken to be composed of a particle convective - a gas convective - and a radiative component or:

$$h = h_{pc} + h_{gc} + h_R$$  \hspace{1cm} (2-23)

The three heat transfer components are discussed separately in the next sections. Other possible contributions such as fluid-particle, particle-particle and particle-heat exchanger surface heat transfer are probably not relevant for particles of the order of magnitude of 1 mm, BOTTERILL (1975), (1977).

The main advantage of distinguishing between the separate contributions is that an analytical description of each can be given, at least approximately. This may lead to a better understanding of the fundamental aspects of the heat transfer and consequently to a better predictability. However, because of the complexity of each of the transfer processes, a numerical approach will often be required for final calculation. The various heat transfer models, available at present, are reviewed by KOLAR (1979) and SAXENA (1981). Notwithstanding the disputability of the semi-empirical correlations, which are generally determined at ambient temperatures, it should be noticed that the predictive power of for example models of BOCK (1980), (1981), (1983) and GANZHA (1983), (BORODULYA (1983)) seem rather satisfying, at least for the conditions under which they are arrived at, mostly ambient. The possible criticism also includes the assumed gas-particle flow regime, as discussed in the preceding sections. This aspect is now elaborated somewhat further.
3.2. Particle convective heat transfer.

The excellent heat transfer properties of a fluidized bed are a consequence of its high volumetric heat storing capacity, as compared to a gas and the fairly strong motions of particles and gas. At a wall, single particles or agglomerates are continually exchanged with the bulk of the bed by a bubble induced surface renewal mechanism and act as heat sources or sinks. This surface renewal mechanism, which can be considered as a "convective" process, not unlike turbulence in fluid flow, goes with transient heat transfer by agglomerates of particles. Fundamentally three types of surface renewal heat transfer models can be distinguished: the stationary semi-infinite approach of MICKLEY (1955), the stationary finite boundary layer approach of VAN HEERDEN (1953) and the instationary finite layer approach of YOSHIDA (1969). The differential equation governing the latter model reads:

\[
\frac{\partial T}{\partial t} = \lambda_{\text{eff}} \frac{\partial^2 T}{\partial x^2}, \quad 0 \leq x \leq \ell_{\text{eff}} \tag{2-24}
\]

For small and large Fourier numbers \(\text{Fo} = t/(\rho C_p \ell_{\text{eff}})\), Yoshida has shown his approach to be identical to respectively the Mickley- and the Van Heerden one, see e.g. KUNII (1969).

The heat transfer coefficients of Mickley and of Van Heerden read respectively:

\[
h_{\text{po}} = \lambda_{\text{eff}} (\rho C_p)_{\text{eff}} S^{0.5} \quad \text{Fo} < 0.8 \tag{2-25}
\]

and:

\[
h_{\text{po}} = \lambda_{\text{eff}} / \ell_{\text{eff}} \quad \text{Fo} < 1.2 \tag{2-26}
\]

In both cases the heat transfer coefficient is a function of the effective physical properties and the dynamics of the gas particle flow.

In a review of the semi-empirical equations for the effective conductivity of the bulk of the fluid bed, HOELEN (1976) has shown this parameter to be a strong function of the conductivity of the solid particles. The effective volumetric heat capacity is, for obvious reasons, given by:

\[
(\rho C_p)_{\text{eff}} = (1-\varepsilon) \rho_s C_{ps} \tag{2-27}
\]

IRVINE (1978) has shown, that the effective volumetric heat capacity amount to about 2.2 MJ/m\(^3\) K for many different types of material. In Yoshida's approach, the fluid dynamical behaviour of the bed is taken into account respectively via the thickness of the effective surface layer \(\ell_{\text{eff}}\) and the stirring factor \(S\). For the effective surface layer thickness no equation is generally accepted, HOELEN (1976).

The stirring factor expresses the renewal rate of the particles. Assuming an uniform surface renewal age distribution function, the stirring factor becomes according to BOCK (1980), (1981):

\[
S = \frac{4}{\pi} \frac{t}{\tau} \tag{2-28}
\]

The mean agglomerate residence time \(\tau\) is:

\[
\tau = (1 - \varepsilon_B)/\nu_B \tag{2-29}
\]

For a wide range of experimental conditions, BOCK (1981) gives empirical equations for both \(\varepsilon_B\) and \(\tau\).

To exclude extremely high heat transfer coefficients for very small Fourier numbers, BASKAKOV (1973) has introduced a heat contact resistance \(R_k\), see the next section.

Now for a constant wall temperature, the analytical expression of the particle convective heat transfer coefficient can be approximated according to Gel'perin, see DAVIDSON (1971), by:

\[
h_{\text{po}} = 1/(R + 0.5 R_k) \tag{2-30}
\]

The fluid bed bulk resistance \(R\) is the inverse of (2-25) or (2-26). For dominant contact resistance equation (2-30) reduces to:

\[
h_{\text{po}} = 1/R_k \tag{2-31}
\]

Based on the local dynamical heat transfer experiments of MICKLEY (1961), it is generally held that for small particles \((d_p < 1000 \mu m)\) the bubble phase does not participate in the particle convective heat transfer. Consequently the particle convective heat transfer coefficient is multiplied by a factor of \([1 - \varepsilon_B]\).
A main weakness of the particle convective heat transfer theory is the very strong dominating influence of the bubble phase porosity; under circumstances $\varepsilon_B$ can become equal to or larger than 0.5, see section 2.2 and MARTIN (1980) and WUNDER (1980). The particle convective heat transfer approach of MARTIN (1980) does not fall in any of the categories mentioned above. It is discussed separately in Appendix A1.

2.3.2.1. Heat contact resistance.

Two groups of heat contact resistance correlations are found in literature (HOELEN (1976), SAXENA (1981)):
- gas gap correlations and
- enhanced wall porosity correlations.

Both groups are shortly reviewed in this section.

A gas gap heat contact resistance is based on the (debatable) assumption that a gas film occurs between the fluidized particles and the heat exchanging surface. A simple expression for the heat resistance due to the gap is (HOELEN (1976)):

$$R_g = \frac{a}{d_p} \frac{1}{\lambda_f}$$

(2-32)

Starting from and modifying the gas gap contact resistance after SCHLÜNDER (1971) (based on the Knudsen effect), BOCK (1980), (1981) proposes the following expression for the contact resistance:

$$R_k = \frac{a}{d_p} \left(1 - \varepsilon_{mf}\right) \frac{\text{Nu}_{p,\text{max}}}{\lambda_f}$$

(2-33)

with:

$$\text{Nu}_{p,\text{max}} = \text{Nu}_{p,\text{max}} \left( \frac{1}{\varepsilon_{mf}} \right)$$

(2-34)

$$\varepsilon_{mf} = 2\varepsilon \delta / d_p$$

(2-35)

and:

$$2\varepsilon = 4A (2/\gamma - 1) + K$$

(2-36)

A, $\lambda$, and $K$ are respectively the mean gas film thickness near the wall, the mean free path of the gas molecules and an empirical particle roughness factor. BOCK (1981) assigns to $K$ the value of 9.53 $\mu$m, but MARTIN (1980) puts $K$ equal to 0. For air the accommodation coefficient $\gamma$ follows from MARTIN (1980):

$$\log \left( \frac{1}{\gamma} - 1 \right) = 0.243 - \frac{357}{T}$$

(2-37)

It deserves noting that, for particles of about 1 mm, this heat contact resistance becomes dominating. As a consequence the agglomerate surface renewal mechanism plays no role anymore. This means that only indirectly via the porosity, the velocity of the fluidizing fluid influences the heat transfer. Here it should be remarked that the approach of WUNDER (1980) can be considered as a special case of the gas gap heat contact resistance. Based on an extensive set of very well defined experiments, WUNDER (1980) concluded that the particle convective heat transfer should be described by an unsteady conductive heat transfer mechanism, in which the (laminar) fluid boundary layer plays a dominant role. This fluid boundary layer is "periodically" stripped off by the intensive motion of the solid particles. An additional point of view is that the boundary layer is self stirring, because of the tortuous motion of the gas in the interstices near the wall. The governing equation for the boundary mechanism is identical to (2.21), except that the physical properties are now equal to those of the fluidizing fluid. Assumming no bubble phase contribution and uniform surface renewal mechanism, the heat transfer coefficient becomes on an analogous way as described in the preceding section, see (2-25):

$$h_{po} = \left[1 - \varepsilon_B\right] \frac{\lambda_f}{d_p} \frac{C_p f \frac{u_B}{\delta}}{\pi^{0.5}}$$

(2-38)

According to Wunder the mean residence time $\tau$ on particle size scale is:

$$\tau = \frac{d_p}{U_B}$$

(2-39)

However, Wunder states that for Reynolds number larger than 10, no reliable correlations are available for the bubble phase porosity $\varepsilon_B$ and the bubble rise velocity $U_B$. Further it is argued in Chapter 3 of this thesis that the mean residence time for this boundary layer renewal mechanism should be written:

$$\tau = \frac{d_p}{U_B}$$

(2-39)
where $\overline{V}$ is the mean velocity of the particles moving along the heat exchanger surface.

For the enhanced wall porosity type of heat contact resistance, BASKAKOV (1973) gives the general expression:

$$R_k = \delta_w^{1/2} \frac{d_p}{\lambda_p}$$  \hspace{1cm} (2-41)

The effective conductivity of the wall layer is to be taken equal to the effective conductivity of the bulk of the fluidized bed corrected for the enhanced porosity of the wall layer, as proposed by DAVIDSON (1971). Representative for the present state of development of this approach is the recent model of GANZHA (1982). For large particles ($d_p \geq 1 \text{ mm}$), GANZHA proposes a combined instationary gas-solid heat transfer mechanism, governed by the differential equations analogous to equation (2-24):

$$\rho_f C_p \frac{\partial T}{\partial t} = \lambda_f \frac{\partial^2 T}{\partial x^2}$$  \hspace{1cm} (2-42)

and:

$$\rho_a C_p \frac{\partial T}{\partial t} = \lambda_a \frac{\partial^2 T}{\partial x^2}$$  \hspace{1cm} (2-43)

The local temperature is taken equal for gas and solids. In equations (2-42) and (2-43), $\delta$ and $d_o$ respectively correspond to an effective thickness of a lens like gas gap near the wall and an equivalent particle diameter. GANZHA shows that the analytical solution of (2-42) and (2-43) is:

$$h_{pc} = 8.95 \left( 1 - \epsilon_w \right)^{0.67} \lambda_f / d_p$$  \hspace{1cm} (2-44)

where $\epsilon_w$, the mean porosity of the wall layer, is taken from GOROLIK (1967), see (3-47). It is interesting to note that (2-44) is nearly identical to the empirical correlation of ZABRODSKY (1981), who found a factor 7.2 instead of 8.95. Further GANZHA shows for large particles that except for helium and hydrogen as fluidizing fluid, the influence of the residence time of particles on the surface can be neglected. Consequently the velocity of the fluidizing fluid again only has an indirect influence on the heat transfer (via the porosity), which is in agreement with Bock, see this section.

2.3.3. Gas convective heat transfer.

For particles larger than about 1 mm and/or elevated pressures, the interstitial gas convective heat transfer becomes of interest XAVIER (1978), XAVIER and BORODULYA, see both GRACE (1980) and BORODULYA (1983). Up till now no well established theoretical approach of the gas convective fluidized bed heat transfer exists. The main reason is, that the local gas particle flow conditions are not well known. Generally turbulent local flow conditions are assumed in experimental investigations. Consequently the empirical gas convective Nusselt numbers are assumed to be proportional to $Re^{0.8}$, BASKAKOV (1972), BUTTERILL (1977), DECKER (1980), GANZHA (1982) and BORODULYA, see GRACE (1980).

2.3.4. Radiative heat transfer.

Up till now views on the radiative contribution to FBC heat transfer are rather conflicting. Based on a review, SAXENA (1981) suggests to be conservative, a radiative contribution of 20 to 30%, but lower values are also mentioned, KARCHENKO (1964) and YOSHIDA (1974). In this section the radiative heat transfer is shortly reviewed and the points to which literature is contradictory are indicated. In general the radiative heat transfer coefficient is expressed as the sum of a dense- and a bubble phase contribution or:

$$h_R = (1 - \epsilon_B) h_{Rd} + \epsilon_B h_{RB}$$  \hspace{1cm} (2-45)

For both phases it is generally accepted following ZABRODSKY (1966), that the fluidized bed can be considered as a black source. The radiative temperature of the (particle free) bubble phase is taken equal to that of the bed. However, because of the temperature profile near the surface, it is mostly assumed that the temperature of the dense phase particles facing the surface is sufficiently low to neglect dense phase radiation. As a consequence of these assumptions, the radiative heat transfer coefficient becomes:

$$h_R = \epsilon_B \epsilon_R \sigma (T_b - T^\star)/\left( T_b - T_w^\star \right)$$  \hspace{1cm} (2-46)
The emission coefficient $\varepsilon_R$ of the surface depends on the material. As a function of temperature it can be found in handbooks, such as PERRY (1973).

To the author's opinion, the not well known bubble phase porosity is the main cause of the conflicting radiative contribution to FBC heat transfer, see section 2.2. Besides the assumption made concerning the effective temperature of the dense phase particles presumably only holds for heat exchangers which use water as the coolant. Further as discussed in the following chapter, porosity waves are present in a PFBC bed. As a consequence the extremes of particle free bubbles and dense phase do not apply. This means that the temperature profile of the particles facing the surface and also the temperature of the surface itself have to be reconsidered.

CHAPTER 3 TURBULENT FLUIDIZATION AND HEAT TRANSFER

3.1. Introduction.

Starting from the findings concerning fluidization and heat transfer as discussed in the preceding chapter and own experimental results, see Chapter 5, the two phase flow and heat transfer in a PFBC are described in this section in terms of the concept of turbulent fluidization, Wen, see CHEREMISINOFF (1983). The term turbulent fluidization was first introduced by CANADA (1978). In his definition a bed is turbulently fluidized if the amplitude of the bed pressure drop fluctuations decreases with increasing fluidization velocity. He observed such a behaviour with beds of large particles at high fluidization velocity. A similar behaviour was found for the PFBC in our own experiments, see section 5.2.2. So it stands to reason to assume turbulent fluidization to occur in a PFBC too. Our measurements on heat transfer, section 5.3.1, further indicate that the fluidized bed can be divided in a turbulent zone (TZ) in the centre and a wall return layer (WRL), see figure 3-1.

![Figure 3-1. The turbulent bed model after BOELENS (1983).](image)

 Actually $V_w$ is negative and $U_w$ may be so.

On the average, the TZ can be interpreted as a region with an enhanced porosity compared to the overall fluidized bed porosity. However to explain the observed local dynamical fluctuations of porosity and heat transfer in the TZ, it is assumed, that porosity waves are present in this
part of the bed. Such porosity waves can physically be interpreted as rising particle laden voids, functioning as "rapids" for the fluidizing gas and as decelerating regions for the solid particles. Nevertheless the voids are the driving mechanism for upward drift of the particles. The mixing of the particles is assumed to occur mainly in these porosity waves (and in the splashing zone). According to Chen, see CHEMISINOFF (1983), the flow regime in the TZ may be classified as fluidized flow. Because of the low heat transfer coefficient measured near the wall of the PFBC, see section 5.3.1, the flow regime in the WRL is assumed to be moving bed flow. That is to say that the solids move downward in dense packing, possibly dragging the gas downward with them. The three parts, which can be distinguished in this chapter are:

* the turbulent fluidized bed model
* the turbulent fluidization model
* the turbulent fluidized bed heat transfer model.

In the turbulent fluidized bed model the average mass balances of the fluidizing fluid and the fluidized particles are formulated (section 3.2). Further the Ergun equation (ERGUN (1952)) is modified to permit prediction of the expansion of a large-particle fluidized bed, (Appendix A2). Combining these equations and applying a relation between the overall porosity and the porosities of WRL and TZ yields the averaged parameters of the turbulent fluidized bed.

Subsequently the dynamical aspects of gas particle flow are analysed in the turbulent fluidization model (section 3.3). Starting from the continuity wave- and continuity shock wave theory, relations are derived to describe some void properties. These properties concern the void rise velocity and the amplitude of the porosity fluctuation, going with these voids. Combining the continuity and momentum equations of the fluidizing fluid and the fluidized particles yields, after linearization, a dispersive wave equation, from which the state of the fluidization and the frequency of the rising voids can be derived. It further permits some conclusions about the homogeneity of a slugging fluidized bed. These results and the own experimental findings, discussed later in Chapter 5, indicate sine longitudinal porosity waves to be present in the turbulent zone of the fluidized bed.

The turbulent fluidized bed heat transfer model is developed in section 3.4. Fundamentally the model can be considered as a heat contact resistance model with an enhanced wall porosity layer. Two surface renewal mechanism are distinguished in the model: a particle agglomerate scale mechanism and a single particle scale one. The first mechanism has its origin in the renewal of agglomerates of particles due to rising voids. This renewal causes an easy transport of heat from the bulk of the bed to close to the surface. The second mechanism is characterized by the renewal (scraping) of the gas film of the enhanced wall porosity layer by the moving particles. It is shown, that in the TZ the time constant of the particle size scale mechanism is about two orders of magnitude smaller than that of the particle agglomerate one. Because of this, it is assumed that the thermal properties of the fluidized particles play no significant role in the bed to surface heat transfer mechanism of the TZ.

The complete heat transfer model encompasses convective, radiative and dispersive conductive heat transfer components. Not included are conduction of heat in the particles, exchange of heat due to particle-particle- and particle-wall interactions, heat transfer due to circulating fines in the pores of the fluidized bed and any heat transfer effects due to chemical reaction. It should be noted, that because of the omission of the heat conduction in the particles, the model is not suitable for heat transfer calculations in the WRL or at low superficial fluidization velocity in general.

As already indicated, the proposed model is partly based on experimental data obtained from the PFBC. However it does not follow from these data in a straightforward way. Actually theoretical considerations play a more important role and for this reason, it was judged better to utilise the experimental results as a verification of the theory and to present the model first.

Most calculations were made at the following conditions of the PFBC, further referred too as standard operating conditions:

- mean size \( d_p \) of silica sand bed material: 780 \( \mu \)m
- bed temperature \( T_B \): 1123 K
- operating pressure \( p \): 0.6 MPa
- superficial fluidization velocity \( U \): 1.05 m/s
- beddiameter \( D_B \): 0.485 m
- turbulent zone fraction of the bed \( F \): 0.8
- porosity of the WRL \( \varepsilon_{wR} \): 0.4
Additional conditions and any deviation(s) from standard conditions are indicated ad hoc. Further it should be remarked, that for these standard operating conditions in the operating pressure range of 0.4 - 1.0 MPa, the superficial fluidization velocity at minimum fluidization conditions \( J_{mf} \) (A2-5) is approximately constant and varies from 0.20 to 0.18 m/s. For the, sometimes referred to, silica sand bed material with a mean size of 1500 \( \mu \)m, \( J_{mf} \) varies from 0.58 to 0.47 m/s. All quantities, used in this thesis are assumed to be independent of height. So the theories developed in this chapter should not be extended to average porosities beyond 0.8 (LI, 1981).

3.2. Turbulent Fluidized Bed Model.

Starting from the mean mass balances of the fluidizing fluid and the fluidized particles, the mean turbulent fluidized bed behaviour is described in this section. As indicated in Appendix A9, the turbulent fluidized bed model is a first simplified approach in which shear stress and so momentum exchange between TZ and WRL are not taken into consideration explicitly.

The averaged one-dimensional mass balances read, BOELENS (1983) and see figure 3-1:

\[
F \varepsilon_T U_T + (1-F) \varepsilon_W U_W = J
\]

and:

\[
F(1-\varepsilon_T) V_T + (1-F)(1-\varepsilon_W) V_W = 0
\]

Here \( F \) is the turbulent zone fraction of the fluidized bed. According to CRANFIELD (1978), BOCK (1980) and own experimental heat transfer measurements, see section 5.3.1, \( F \) is equal to 0.8. The relation between the overall mean porosity of the fluidized bed and the averaged porosities of the WRL and TZ is:

\[
(1-\varepsilon) = F(1-\varepsilon_T) + (1-F)(1-\varepsilon_W)
\]

The mean overall porosity is calculated from the modified Ergun equation, see Appendix A2. The average porosity of the WRL is assumed to be constant and equal to 0.4. Its precise value has only a minor influence on the particle mass balance. The moving bed flow in the WRL is assumed to be governed by the slip relation, or

\[
U_W = V_W = \frac{\varepsilon_W}{\varepsilon_W} \frac{J_{mf}}{\varepsilon_W}
\]

As a first approximation the coefficient \( c \) is taken equal to 1.0. Observations by SCHMALFELD (1976), WERTHER (1976) and STAUB (1979) all indicate that the downward velocity of the solids near the wall is fairly small. These observations were made at ambient conditions, but there are no obvious reasons why a pressurized fluidized bed should behave differently in this respect. No further experimental data being available, we assume the downward solid velocity in the wall return layer to amount to:
\[ V_w = -0.2 \left( J - \frac{F}{m} \right) \]  
\hspace{1cm} (3-5)

The coefficient 0.2 represents an average value for the observations just mentioned.

The results of this approach are given in tables 3-1 and 3-2.

From table 3-1 and 3-2 it follows that for increasing superficial fluidization velocity and/or operating pressure the turbulent fluidized bed behaves more and more like a counter current reactor, in which in the turbulent zone the gas is in plug flow and the particles are nearly perfectly mixed.

A remarkable feature of the model developed so far is that void coalescence and consequent variation of \( V_T \) and \( U_T \) can be simulated by decreasing \( F \) as a function of height. This is demonstrated in table 3-3, see also section 3.3.3 (figure 3-16).

**Table 3-1.**
Mean turbulent fluidized bed properties, as a function of superficial fluidization velocity.
Standard operating conditions but for \( J \).

\[
\begin{array}{|c|c|c|c|c|c|c|}
\hline
J \hspace{0.5cm} & \epsilon & U_T \hspace{0.5cm} & V_T \hspace{0.5cm} & \epsilon_T & U_W \hspace{0.5cm} & V_W \\
\hline
0.39 & 0.46 & 0.92 & 0.011 & 0.48 & 0.44 & -0.039 \\
0.59 & 0.51 & 1.26 & 0.026 & 0.54 & 0.40 & -0.079 \\
0.80 & 0.56 & 1.61 & 0.045 & 0.60 & 0.36 & -0.12 \\
1.03 & 0.60 & 1.93 & 0.073 & 0.65 & 0.31 & -0.17 \\
1.23 & 0.63 & 2.19 & 0.10 & 0.69 & 0.28 & -0.20 \\
1.43 & 0.66 & 2.44 & 0.13 & 0.72 & 0.23 & -0.25 \\
1.63 & 0.69 & 2.66 & 0.18 & 0.76 & 0.19 & -0.29 \\
\hline
\end{array}
\]

**Table 3-2.**
Mean turbulent fluidized bed properties, as a function of operating pressure.
Standard operating conditions but for \( p \).

\[
\begin{array}{|c|c|c|c|c|c|c|}
\hline
p \hspace{0.5cm} & \epsilon & U_T \hspace{0.5cm} & V_T \hspace{0.5cm} & \epsilon_T & U_W \hspace{0.5cm} & V_W \\
\hline
0.4 & 0.58 & 2.04 & 0.069 & 0.63 & 0.32 & -0.17 \\
0.6 & 0.60 & 1.95 & 0.073 & 0.65 & 0.31 & -0.17 \\
0.8 & 0.62 & 1.87 & 0.075 & 0.67 & 0.30 & -0.17 \\
1.0 & 0.63 & 1.85 & 0.085 & 0.70 & 0.29 & -0.17 \\
\hline
\end{array}
\]

**Table 3-3.**
Mean properties of the turbulent zone of the turbulent fluidized bed model, as a function of the turbulent zone fraction.
Standard operating conditions.

Table 3-3 illustrates that for decreasing turbulent zone fraction, the bed takes spouting-like properties.
3.3. Turbulent fluidization model.

3.3.1. Introduction.

In this section the dynamics of the gas particle flow in the turbulent zone of a fluidized bed of large particles (\( d_p > 1 \) mm) is described. Assuming that the velocity of the fluidized particles is small compared to the gas velocity and that particle-particle interaction forces are not significant, MOLERUS (1982a) and RIETEMA (1973), (1982), we suppose that on the scale of a volume element a heterogeneously fluidized bed is similar to a homogeneously fluidized bed. Such a bed is characterized by a relatively well ordered matrix of solid particles and by turbulent flow in the interstices.

Applying the theory of continuity waves and continuity shock waves, the generation of particle free and dust laden voids (continuity shocks) and their stability can be explained. Further expressions are found for the rise velocity and the amplitude of porosity fluctuation going with such voids.

Starting from packed beds, BEAR (1972) and homogeneously fluidized beds, RIETEMA (1973), (1982) and NEEDHAM (1983) the continuity- and momentum equations of the fluidizing fluid and the fluidized particles are derived on the scale of a volume element (section 3.3.2). The dimensions of such a volume element are an order of magnitude smaller than the wave length of any porosity fluctuation and an order of magnitude larger than the size of the fluidized particles, RIETEMA (1982). A volume element contains a large number of fluidized particles, typically in the order of a thousand.

Based on the flow phenomena in packed beds, hydrodynamic dispersion, as discussed in detail in Appendix A2, is introduced for fluidized beds in this thesis. Hydrodynamic dispersion is the macroscopic, averaged result of mixing on a microscopic scale. It is the sum of molecular self diffusion and mechanical dispersion (i.e. the mixing caused by the transverse component of the interstitial motion).

Combining the continuity- and momentum equations finally yields a dispersive wave equation. Substituting a harmonic porosity fluctuation into this wave equation yields relations describing the state of fluidization, the frequency of the porosity fluctuations and permitting some conclusions about the homogeneity of a heterogeneously fluidized bed. It should be remarked that a main assumption of this approach is that the amplitude of the porosity gradient is obtained by linearization of the porosity fluctuation.

In order to separate fluctuations and mean values, every property is decomposed in a mean- and a fluctuating component:

\[
\mathbf{g} = \mathbf{\bar{g}} + \mathbf{g}' \quad \text{with} \quad \mathbf{\bar{g}} = 0
\]  

3.3.2. Continuity waves.

3.3.2.1. Introduction.

In this section the continuity wave and the related continuity shock wave are discussed.

Starting from the continuity wave velocity (group velocity), the interaction of continuity waves, resulting in void (in)stabilities, is described. A (continuity) roof shock (wave) and a degenerating wake shock are distinguished.

Finally expressions are given for the amplitude and the upward propagation velocity of the porosity fluctuations associated with these shocks.

3.3.2.2. Continuity waves.

According to WALLIS (1969), continuity waves are quasi steady state phenomena, which occur whenever there is a relationship between flow rate and concentration (porosity). Applying the continuity principle on an infinitely small increase of the porosity in the turbulent zone of a fluidized bed, Wallis obtains for the continuity wave velocity (group velocity) the expression:

\[
W_c = \frac{2J}{\mathbf{\bar{U}}} \quad (3-7)
\]

For \( U >> V \), see section 3.2, it is assumed that the \( T_z \) superficial fluidization velocity \( J = c_T U_p \) is given by equation (A2-9). As will be shown later (cf. figures 3-4 and 3-5), the general trend of the continuity wave velocity as a function of the porosity is as depicted in figure 3-2. From figure 3-2 it is easily understood, that due to dynamic forces (see following section), the "quasi steady state" continuity waves originating from different porosities interact in the way indicated in figure 3-3.
This figure illustrates that small porosity perturbations degenerate (a) or overtake each other to form a single hump (shock) (b), if $\frac{\partial W}{\partial \varepsilon}$ and $\Delta \varepsilon$ have respectively the same or the opposite sign.

The two situations to be discerned correspond with the bubble (in)stabilities cited in literature, see for a review amongst others GUEDES DE CARVALHO (1976). To the authors opinion, however, the interpretation should be changed in the framework of turbulent fluidization, such in the following way:

- a. Degenerating wake of the void. This degeneration is unstable and difficult to observe, because any particles entrained in the void, fall back due to gravitation.
- b. Tendency to generate a sharp porosity front at the roof of the void, further designated as roof shock.

**FIGURE 3-2.**
General trend of continuity wave velocity as a function of porosity.

**FIGURE 3-3.**
Interactions of small porosity perturbations $\Delta \varepsilon$ as a consequence of propagation velocity differences of continuity waves. Time $t_2 > t_1$.

**FIGURE 3-4.**
Continuity wave velocity as a function of superficial fluidization velocity for different operating pressures. Standard operating conditions, but for $p$ and $J$. 
In figures 3-4 and 3-5, the continuity wave velocity is depicted as a function of some operating conditions.

Figure 3-4 illustrates that due to increasing operating pressure, the continuity wave velocity decreases over the full superficial fluidization velocity range. Figure 3-5 shows that increasing particle size leads to a higher continuity wave velocities.

![Graph showing continuity wave velocity as a function of superficial fluidization velocity for different particle sizes.](image)

**FIGURE 3-5.**
Continuity wave velocity as a function of the dimensionless superficial fluidization velocity for different particle sizes.
Standard operating conditions, but for \( \phi_p \) and \( J \).

3.3.2.3. Continuity shock waves.

As described in the previous section, the interacting continuity waves result in roof shocks. According to WALLIS (1969), the rise velocity of a weak shock wave is equal to:

\[
W_g = (J_2 - J_1) / (e_2 - e_1)
\]

The general trend of the velocity of a weak shock wave as a function of porosity is identical to that of a continuity wave, see figure 3-2. The only difference is that instead of the slope of the tangent, now the slope of a small chord is applied. From figure 3-2 it is easily understood, that due to dynamic forces, see following section, the weak shocks are interacting, see figure 3-6.

![Diagram showing interactions of weak roof shocks as consequence of propagation velocity differences of such shocks.](image)

**FIGURE 3-6.**
Interactions of weak roof shocks as consequence of propagation velocity differences of such shocks. Time \( t_2 > t_1 \).
Because $\Delta \epsilon_{3w}$ and the porosity differences $\Delta \epsilon_1$ and $\Delta \epsilon_2$ have the opposite sign, weak roof shocks overtake each other. The maximum porosity jump going with a roof shock is equal to:

$$\Delta \epsilon_{\text{roof}} = \epsilon_{mf} - 1$$  \hspace{1cm} (3-9)

Taking in mind the two situations, as distinguished in the previous section, a void can now be visualized, as depicted in figure 3-7. The marks a and b correspond with respectively a degenerated wake and a roof shock.

Starting from figure 3-7, it is suggested that the positive and negative amplitude going with voids, are respectively equal to:

$$\epsilon_{o,\text{max}} = 1 - \epsilon$$  \hspace{1cm} (3-10)

and:

$$\epsilon_{o,\text{min}} = \epsilon - \epsilon_{mf}$$  \hspace{1cm} (3-11)

Because generally $\epsilon_{o,\text{max}} > \epsilon_{o,\text{min}}$, the amplitude density distribution function of $\epsilon_o$ will be asymmetric. This is in agreement with own experimental findings, see section 5.2.3.3.

For increasing superficial fluidization velocity and/or operating pressure equations (3-10) and (3-11) indicate $\epsilon_{o,\text{max}}$ to decrease and $\epsilon_{o,\text{min}}$ to increase. Based on scarce data known from literature, LI (1981) and own experimental findings, see section 5.2.2 and 5.2.3.3, it is felt by the author, however, that at such circumstances the porosity of the dense part of the void and, as a consequence, also the local superficial gas velocity, should both increase, see table 5-2. Also, as discussed in the following section, the roof of a void becomes unstable at high porosities (Taylor instability). Consequently the upper porosity value will be lower than 1.0. Actually the "dense phase" expansion, the Taylor instability and the amplitude of the porosity fluctuation can only be explained by non-linear theory, which is not covered in this thesis.

For calculation purposes, it is further assumed in this thesis that the mean amplitude of the porosity fluctuation, going with a void, is as a first approximation given by:

$$\epsilon_o = \frac{1 - \epsilon_{mf}}{2}$$  \hspace{1cm} (3-12)

The rise velocity of a void then follows from (see equation (3-8)):

$$W = \frac{(J - J_{mf})}{\epsilon_o}$$  \hspace{1cm} (3-13)

The void rise velocity, i.e. the propagation velocity of the porosity fluctuation, is further designated as phase velocity.
In figures 3-8 and 3-9, the phase velocity is depicted as a function of operating conditions.

Both figures show that the phase velocity increases with increasing superficial fluidization velocity. Figure 3-8 shows that the phase velocity is (nearly) independent of operating pressure. This is not in agreement with own experimental findings. The contradiction may be caused by the "dense phase" expansion in the TZ, discussed earlier. If so, $\epsilon_{NF}$ in equation (3-12) would take a higher value at higher operating pressure, leading to a higher value for $W_u$. Figure 3-9 shows the phase velocity to increase with increasing particle size.

The phase velocity is of the same order of magnitude as both the rise velocity of an asymmetric slug, see equation (2-23) and own experimental findings, see section 5.2.3.5.
3.3. Dynamics of gas particle flow.

3.3.1. Introduction.

In the first part of this section, the continuity- and momentum equations of the fluidizing fluid and the fluidized particles are shortly reviewed. In the second part, these equations are combined, which leads, after substituting of a dynamical state, to a dispersive wave equation. Substituting a plane harmonic longitudinal porosity perturbation into this wave equation and linearizing the amplitude of the porosity gradient yields expressions for the dispersion of the amplitude and the frequency. The amplitude dispersion determines the nature of the fluidization: heterogeneous or homogeneous. The frequency dispersion relation yields the void frequency as a function of operating conditions. Based on the growth rate distance, which is related to the amplitude dispersion and the wave length of the porosity fluctuation, one can draw conclusions about the apparent homogeneity of fluidized beds, that are actually heterogeneous fluidized.

3.3.2. Continuity and momentum equations.

According to BEAR (1972) and RIETEMA (1973), (1982) the fluid continuity equation of a volume element in vertical direction can be written as:

\[ \frac{\partial (\epsilon \rho_f U)}{\partial t} - \frac{\partial}{\partial z} \left( \epsilon \rho_f \frac{D_f}{\partial z} \right) - \frac{\partial (\epsilon \rho_f U)}{\partial t} = 0 \]  
\[ \text{(3-14)} \]

with \( D_f \) the coefficient of hydrodynamic dispersion, see Appendix A2. For constant fluid density the fluid continuity equation reduces to:

\[ \frac{\partial \epsilon}{\partial t} \frac{\partial (\epsilon U)}{\partial z} = 0 \]  
\[ \text{(3-15)} \]

The solids continuity equation at these conditions reads, RIETEMA (1973):

\[ \frac{\partial (1-\epsilon) V}{\partial t} - \frac{\partial ((1-\epsilon) V)}{\partial z} = 0 \]  
\[ \text{(3-16)} \]

It may be noted that adding of (3-15) and (3-16) yields a constant upward volume velocity of the mixture.

For the fluid and particle momentum equations, the cohesive particle-particle forces and collision forces are not taken into account. The cohesion forces are very small compared to other forces for the particle size considered, MOLERUS (1982a) and RIETEMA (1973), (1982). The collision forces are taken to cause only small scale random "Brownian" movements of the fluidized particles. Combined with the continuity equations, the fluid and solid momentum equations become respectively, according to BEAR (1972) and RIETEMA (1982):

\[ \epsilon \rho_f \frac{D_f U}{Dt} = - \frac{\partial}{\partial z} \left( \epsilon \frac{D_f}{\partial z} \right) - \epsilon \rho_f g + F_E + F_a \]  
\[ \text{(3-17)} \]

and:

\[ (1-\epsilon) \rho_s \frac{D V}{Dt} = -(1-\epsilon) \frac{\partial}{\partial z} \left( \rho_s \frac{D}{\partial z} \right) - (1-\epsilon) \rho_s g + F_E - F_a \]  
\[ \text{(3-18)} \]

The RHS terms of these two equations are forces per unit volume. The buoyancy- and gravitational forces are represented by the terms containing respectively pressure gradient and gravitational acceleration. \( \tau_f \) is the local average fluid shear stress tensor, which takes into account the non-uniformity of the fluid velocity distribution. This non-uniformity is due, among others, to hydrodynamic interaction of the flow fields around neighbouring particles. The "Reynolds" stress tensor of the fluid \( R_f \) describes the exchange of momentum between adjacent fluid layers as a consequence of the quasi-turbulent zig-zag motion of the gas in the interstices between the particles. The associated turbulent eddy viscosity is a function of a mixing coefficient, which is related to the hydrodynamic dispersion, see Appendix A2.

The fluid-solid interaction force is assumed to be described by the modified Ergun equation (A2-8):

\[ F_g = \frac{1}{K} \left( 1 + \frac{p_s K_f \epsilon}{\mu} (V-U) \right) c^2 (V-U) \]  
\[ \text{(3-19)} \]

\( K \) is the permeability and \( K_f \) the Forchheimer coefficient. The particle-particle shear force in the TZ is equal to, see Appendix A9:

\[ F_\tau = (1-F) (\epsilon - \epsilon_m) (p_s - p_f) g \]  
\[ \text{(3-20)} \]
Note: In this chapter E is the porosity of the TZ.

If the slip velocity (U-V) is not constant in time, which is a consequence of porosity waves, the relative acceleration of the so-called added mass of the fluidized particles should be taken into account. According to RILETEMA (1982), this force reads:

\[ F_a = \frac{c(1-c) \rho_f D(V-U)}{2} \] (3-21)

Because U >> V, see tables 3-1 and 3-2, the fluid momentum equation can now be written as:

\[ P_f \frac{DU}{Dt} = -2 \frac{\partial P_f}{\partial z} + \frac{3K_f}{\eta} + \frac{1}{\varepsilon} \frac{\partial \varepsilon}{\partial z} - \rho_f g - F_f/(1-c) \] (3-22)

For \( \rho_s >> \rho_f \), which holds for PPFC, the relative acceleration force \( F_a \) can be neglected in the solid momentum equation, so:

\[ P_s \frac{DV}{Dt} = - \frac{\partial P_s}{\partial z} - \frac{3K_f}{\eta} - \rho_s g - F_f/(1-c) - F_s/(1-c) \] (3-23)

The RHS's of (3-22) and (3-23) represent the dynamic forces per unit volume of the concerned phase, acting on respectively the fluid and the solids.

They are further designated as \( f_f \) and \( f_s \).

### 3.3.3.3 Dynamics of gas particle flow.

WALLIS (1969) has shown that combination of the one-dimensional continuity and momentum equations of fluid and particles (preceding section) and substitution of a dynamic state, defined as:

\[
\begin{align*}
\varepsilon &= \varepsilon + \varepsilon' \quad ; \quad \varepsilon' T = 0 \\
U &= \bar{U} + U' \quad ; \quad \bar{U}' T = 0 \\
V &= \bar{V} + V' \quad ; \quad \bar{V}' T = 0
\end{align*}
\] (3-24)

yield the following linearized equation in \( \varepsilon' \):

\[ \frac{\partial^2 \varepsilon'}{\partial t^2} + A \frac{\partial^2 \varepsilon'}{\partial z^2} + 2\dot{V} \frac{\partial \varepsilon'}{\partial t} + B \left( \frac{\partial \varepsilon'}{\partial t} + \dot{W} \frac{\partial \varepsilon'}{\partial z} \right) = 0 \] (3-25)

This dispersive wave equation, WHITHAM (1974), describes the fluctuations (the dynamical component) of the porosity and is the starting point for our consideration of the dynamical aspects of the turbulent gas-solid fluidized bed.

Before substituting a harmonic porosity perturbation into (3-25), we discuss the coefficients as finally obtained from the original expressions of Wallis.

According to Wallis A reads:

\[ A = \frac{(1-c) \rho_f U^2}{\varepsilon \rho_f} + V^2 + \frac{(1-c)}{\rho_f} \rho_f \] (3-26)

with:

\[ \psi = f / - \frac{\partial \psi}{\partial z} \] (3-27)

As discussed later in this section the linearized amplitude of the porosity gradient reads:

\[ \frac{\partial \psi}{\partial z} = - \varepsilon \omega/\dot{W} \] (3-38)

The dynamic force per unit volume \( f \) reads (see end of preceding section):

\[ f = f_f - f_s \] (3-28)

Writing out this equation, it becomes clear that the influence of the local average fluid shear and the "Reynolds" stress tensor are both small and can be neglected compared to the solid-solid shear stress, and that the relative acceleration force \( F_s \), see (3-21) is of minor importance. The dynamic force, which causes the so-called dynamic wave, accelerates material as a result of the concentration (porosity) gradient, WALLIS (1969). The dynamic wave velocity \( W_d \) is given by the expression:

\[ W_d^2 = V^2 - A \] (3-29)

The weighted mean velocity reads:

\[ V_o = V + U \rho_f (1-c)/\rho_s c \] (3-30)
The positive coefficient $B$ in (3-25), which represents the damping in a fluidized bed (see WALLIS (1969)) reads:

$$B = B_E + B_P$$  \hspace{1cm} (3-31)

with:

$$B_E = \frac{2\rho F K}{\mu(1-\varepsilon)} \left[ 1 + \frac{2\rho F K (U-V)}{\mu} \right] \left( \frac{-\varepsilon^2 + \varepsilon + 2}{3-\varepsilon} \right)$$ \hspace{1cm} (3-32)

and as a first approximation:

$$B_P = 2V(1-\varepsilon)/(1-\varepsilon)^2(1-F)$$  \hspace{1cm} (3-33)

The continuity wave velocity $W_c$ (group velocity) has been discussed earlier in section 3.3.2.2.

The general plane harmonic longitudinal porosity perturbation to be substituted into (3-25) is:

$$\varepsilon = \varepsilon_0 e^{at} + iw(t-z/W)$$  \hspace{1cm} (3-34)

$W_c$ and $e^{at}$ represent respectively the propagation velocity (phase velocity), see section 3.3.2.3 and the amplification of the porosity wave.

Because of the gradual fluidized bed - splashing zone - freeboard transition, the influence of waves, reflected at the bed surface, is considered to be of minor importance.

After substitution of (3-34) into (3-25), one obtains from the imaginary- and the real part of the exponent of $e$ respectively the amplitude amplification factor $a$ (amplitude dispersion):

$$a = \frac{B}{2} \left( \frac{W_c}{W_p} - 1 \right)$$  \hspace{1cm} (3-35)

and the circle frequency:

$$\omega^2 = \frac{B}{4} \left( \frac{W_c^2 - W_p^2}{W_c - W_p} \right)$$  \hspace{1cm} (3-36)

The circle frequency of the harmonic perturbation as predicted by this frequency dispersion relation should be real. Consequently either:

$$W_c^2 < W_p^2 < W_d^2$$  \hspace{1cm} (3-37)

or:

$$W_d^2 < W_c^2 < W_p^2$$

For $W_c < W_d$, the perturbation is damped because $a < 0$. Under these conditions the gas particle flow is stable, which means that voids tend to disappear and the bed tends to fluidize homogeneously.

For $W_c = W_d$, perturbations grow, because $a > 0$. The gas particle flow is now unstable and consequently voids are generated. The bed is said to fluidize heterogeneously.

The amplitude $\varepsilon_0$ and propagation (phase) velocity $W_c$ of the porosity fluctuation were estimated earlier in section 3.3.2.3. The expressions read respectively:

$$\varepsilon_0 = \frac{1 - \varepsilon_0}{2}$$  \hspace{1cm} (3-12)

and:

$$W_c = \frac{(J - J_{ml}^2)}{\varepsilon_0}$$  \hspace{1cm} (3-13)

Finally, neglecting the amplitude dispersion in a volume element, the linearized amplitude of the porosity gradient $\frac{\partial \varepsilon}{\partial z}$ can be approximated as:

$$\frac{\partial \varepsilon}{\partial z} = -\varepsilon_0 w/W_c$$  \hspace{1cm} (3-38)

The set of equations given above can now be solved.

The amplitude amplification factor $a$ (exponent of $e$ in equation (3-34)) is depicted as a function of operating conditions in figures 3-10 and 3-11. The figures illustrate that increasing $J$ and/or $p$ and/or $d_p$ has a stabilizing effect on fluidization.

Decreasing amplitude amplification factor may result, however, in void instability. Due to second order effects, such as the Taylor instability, see GUEDES DE CARVALHO (1976) particles rain like fingers through the void. This means that instead of particle free voids particle laden voids will be observed. Actually the latter were found in the PFBC, see section 5.2.3.
In literature only Rowe, see GRACE (1980) and CHITESTER (1984), report a transition from particle free into particle laden voids at higher superficial fluidization velocity and operating pressure for small and generally light particles ($d_p \leq 370 \mu m$ and $\rho_p \leq 1250 \text{ kg/m}^3$).

The stabilizing influence of higher fluidization velocity is contrary to findings for small particles ($d_p < 0.1 \text{ mm}$). This can be explained by the stabilizing influence of cohesive particle-particle forces, which are not relevant for PFBC conditions.

This influence causes small particle beds to fluidize stably in the lower part of the superficial fluidization velocity range (NIRTEH (1973)).

The dynamic wave velocity is depicted as a function of operating conditions in figures 3-12 and 3-13. Both figures show that the dynamic wave velocity increases with increasing superficial fluidization velocity. Figure 3-12 further shows that increasing operating pressure has only a minor influence. Figure 3-13 illustrates that the dynamic wave velocity increases with increasing particle size.
The frequency of the rising voids is depicted as a function of operating conditions in figures 3-14 and 3-15. The figures show that:

- the void frequency decreases with increasing superficial fluidization velocity
- the void frequency decreases with increasing particle size, and that
- the operating pressure has only a minor influence on the void frequency.

The first two trends are similar to the findings at ambient conditions of VERLOOP (1974) for particle sizes up to 1200 µm and DARTON (1977) and Rowe, see GRACE (1980) for small particles (dp = 100 µm).

The last trend is in agreement with VERLOOP (1974) and own experimental findings, see sections 5.2.3.4 and 5.3.2.4.

No void frequency data or relations for PFBC conditions were found in literature. The calculated frequency, however, is of the same order of magnitude as the frequency measured in own experiments, see sections 5.2.3.4 and 5.3.2.4 and the void frequency reported in AFBC literature, LORD (1982), MITTMANN (1982), YOSHIDA (1982) and ZHANG (1982).
Decreasing turbulent zone fraction $F$, which would go with coalescence of the voids with height, results in an increasing superficial fluidization velocity see table 3-3, and so in a decreasing frequency, see figure 3-16. Such a trend is in agreement with own experiments, see section 5.3.2.4 and is reported for small particles, among others, by Rowe, see GRACE (1980). The dependency of void frequency on operating pressure may be related to a decrease of the turbulent zone fraction. However, no such a dependency of the turbulent zone fraction on operating pressure has been reported in literature.

The homogeneity of fluidized beds can also be discussed in terms of the growth rate distance of the amplitude of the porosity wave and the wave length of the porosity fluctuation. The growth rate distance $\lambda_a$ is defined as the distance in which the amplitude increases by a factor of $e$. From equation (3-34), it is easily calculated that:

$$\lambda_a = \frac{W}{a}$$  \hspace{1cm} (3-39)

If the amplitude amplification factor goes to zero, the growth rate distance becomes very large. The general trend of the growth rate distance is depicted in figure 3-17. If the growth distance is an order of magnitude larger than the bed height it may be conjectured that heterogeneity cannot develop sufficiently within the bed. So an apparently homogeneous bed will be observed.

The wave length of the porosity fluctuation reads:

$$\lambda_p = \frac{W}{\nu}$$ \hspace{1cm} (3-40)

The general trend of the wave length as a function of operating conditions is depicted in figure 3-17. For $\lambda_p \gg H$, the fluidized bed is, seemingly, homogeneous. On the other hand for $\lambda_p < H$, the bed is heterogeneous in both nature and appearance. If $\lambda_p = 2H$, standing waves might be present in the bed, but to the authors knowledge, no observations support this presumption.

An apparently homogeneously fluidized bed will be observed if both the growth rate distance $\lambda_a$ and the wave length $\lambda_p$ are an order of magnitude larger than the bed height.

FIGURE 3-16.
Void frequency as a function of turbulent zone fraction $F$.
Standard operating conditions, but $J = 1.05 \text{ m/s}$.

FIGURE 3-17.
Growth rate distance of voids $\lambda_a$ and wave length of porosity fluctuations $\lambda_p$ as a function of superficial fluidization velocity.
Standard operating conditions, but for $J$.
- - - - - : $\lambda_a$ equation (3-39)
- - - - - : $\lambda_p$ equation (3-40).
3.3.4. Conclusions.

The main conclusions to be drawn from the theory describing the dynamics of gas particle flow in the PFBC are the following:

- At high superficial fluidization velocity both the growth rate distance and the wavelength of the porosity fluctuations are an order of magnitude larger than the bed height; consequently the fluidized bed, though essentially unstable, exhibits seemingly homogeneous fluidization.

- At increasing superficial fluidization velocity and/or pressure a transition from particle-free voids to particle laden voids occurs; no distinct value for the point frequency can be given as yet.

- The amplitude probability density function of the porosity fluctuation should show a skewed distribution. For calculation purposes the amplitude has been assigned a constant value. However, actually the porosity fluctuations "disappear" and the amplitude tends to low values at high porosity.

- The trends of the phase (rise) velocity of the porosity waves as a function of both superficial fluidization velocity and particle size agree satisfactorily with experimental observations. The theoretical influence of operating pressure on the phase velocity is opposite to the experimentally observed one.

- The influence of superficial fluidization velocity, operating pressure and particle size on void frequency agree satisfactorily with empirical data.

3.4. Turbulent fluidized bed heat transfer model.

3.4.1. Introduction.

As already mentioned in the introduction of this chapter, the turbulent fluidized bed heat transfer model is fundamentally an enhanced wall porosity model, see section 2.3. In our approach a particle agglomerate scale surface renewal mechanism and a particle size scale "gas film" renewal mechanism in the boundary layer are assumed.

The first mechanism and so its time constant find their origin in the renewal of agglomerates of particles due to rising voids, of the kind described in the preceding section. This mechanism transports heat from the bulk of the bed to close to the surface, with negligible heat resistance. The latter in agreement with the nearly perfect mixing of solids in turbulent fluidization. The assumed particle size scale heat transfer mechanism implies, that the turbulent gas film (boundary layer) near the heat exchanging surface is continually renewed (scraped, stripped off) by the moving particles. At this point, it should be remarked that by assuming a turbulent gas film, our approach differs from classical ones, which all assume a laminar boundary layer, see Gel'perin (DAVIS, 1971, p. 476). The high Reynolds numbers occurring with PFBC are the main reason for our assumption.

The particle size scale mechanism, suggested in this thesis is characterized by three assumptions:

- in the enhanced wall layer, the modified Ergun equation is combined with a boundary layer velocity weight function to determine the gas velocity profile;

- the gas renewal length is taken equal to the mean distance between the particles of the layer nearest to the heat exchanging surface.

- the renewal time constant is taken equal to the ratio of the gas renewal length and the particle velocity $\bar{V}_T$ of the TZ, as given earlier in section 3.2.

The underlying picture for these assumptions is a whirling gas motion around the particles touching the surface and, by their movements, also renewing the surface gas layer.
3.3.4 Conclusions.

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- The amplitude probability density function of the porosity fluctuation should show a skewed distribution. For calculation purposes the amplitude has been assigned a constant value. However, actually the porosity fluctuations “disappear” and the amplitude tends to low values at high porosity.

- The trends of the phase (rise) velocity of the porosity waves as a function of both superficial fluidization velocity and particle size agree satisfactorily with experimental observations. The theoretical influence of operating pressure on the phase velocity is opposite to the experimentally observed one.

- The influence of superficial fluidization velocity, operating pressure and particle size on void frequency agree satisfactorily with empirical data.

3.4 Turbulent fluidized bed heat transfer model.

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- the gas renewal length is taken equal to the mean distance between the particles of the layer nearest to the heat exchanging surface.

- the renewal time constant is taken equal to the ratio of the gas renewal length and the particle velocity \( \bar{V}_i \) of the TZ, as given earlier in section 3.2.

The underlying picture for these assumptions is a whirling gas motion around the particles touching the surface and, by their movements, also renewing the surface gas layer.
As will be shown, the time constant of the particle size scale mechanism is about two orders of magnitude smaller than the time scale of the particle agglomerate mechanism, which is equal to the inverse of the void frequency $v$, see section 3.3.3.3. As a consequence the thermal properties of the fluidized particles play no significant role, because for PFBC conditions the particle Fourier number $<< 1$.

The turbulent fluidized bed heat transfer model includes dispersive conductive heat transfer. For packed beds this contribution has already been introduced by means of a Péclet number, see ZEHNER (1973) for a review, or in the form of thermal dispersion by GUNN (1981), PLUMB (1981) and CHEUNG (1981). For fluidized beds, however, only AEROV, (DAVIDSON (1971), p. 487), gives a contribution of gas convection to the heat transfer, which is similar to dispersive conduction. Dispersive conduction (thermal dispersion) is due to hydrodynamic dispersion, see Appendix A2 and improves the effective conductivity of the fluidizing fluid significantly, in particular for large particles and high flow velocities. The heat transfer model to be discussed in the next sections, also encompasses turbulent convection and radiation. Excluded are conduction of heat in the particles, heat exchange due to particle-particle and particle-wall interactions, heat transfer due to circulating fines in the pores of the fluidized bed and any heat transfer effects due to chemical reactions. Due to the omission of heat conduction in the particles ($P_0 << 1.0$), the model is not suited for calculations in the WRL, nor in the TZ if as shown, the Reynolds number is too low. In the following section, the turbulent fluidized bed heat transfer to vertical surfaces, immersed into a fluidized bed of large particles ($d_p = 1$ mm) is further described. Subsequently the results of calculations are presented and some conclusions are drawn.

3.4.2. Turbulent fluidized bed heat transfer model.

Assuming local thermal equilibrium between fluidizing fluid and fluidized particles, the particle size to be an order of magnitude smaller than the tube diameter, and the heat flow near the wall to be essentially normal to the wall, yields the following one dimensional energy equation for the turbulent fluidized bed heat transfer model:

$$\frac{P_f C_{pf}}{\frac{\partial T}{\partial t}} + U \frac{\partial T}{\partial x} = \lambda_{eff} \frac{\partial^2 T}{\partial x^2} + \phi_R + \phi_{Bw} \tag{3-41}$$

The radiative terms are further explained with (3-50) and (3-51).

Initial and boundary conditions are:

$$\begin{align*}
T &= T_w \quad x = 0 \quad \text{all } t \\
T &= T_b \quad x > 0 \quad t = 0 \\
T &= T_b \quad x = 0 \quad t > 0
\end{align*} \tag{3-42}$$

It should be noted that the local particle surface temperature is assumed equal to the gas temperature at the same distance from the wall. The local particle surface temperature determines the local radiation.

The equation implies that the thickness of the enhanced porosity wall layer is equal to $0.56 \, d_p$. The fluidizing fluid velocity parallel to the vertical surface is determined in first instance from the modified Ergun equation, see Appendix A2. For obvious reasons the mean vertical pressure gradients in the wall layer and the bulk of the bed are equal:

$$\frac{\partial P}{\partial z} = \frac{\partial P}{\partial z} \hat{w}_T \tag{3-44}$$

Because near the wall the porosity approaches 1.0, equation (3-44), yields the terminal velocity there. To prevent such unrealistic flow phenomena the first layers of the fluid velocity profile are multiplied by a correction factor $M$, which reads:
The correction coefficient is restricted to a layer of 0.32 d, because else the velocity profile is too strongly corrected, in the authors opinion.

It may be argued that M should somehow be related to a Reynolds number characterizing the enhanced porosity layer or, as an alternative, the boundary layer along the wall. Because of the scarcity of experimental data concerning the enhanced porosity layer, it was felt, however, that such an approach would be premature. The trend of the fluid velocity profile as calculated by equation (3-44), its correction equation (3-45) and the final profile are depicted as a function of dimensionless distance in figure 3-18.

Locally the upward velocity surpasses the average upward velocity $\bar{U}$. The real occurrence of such a high velocity can, by the way, visually be observed as the annular "fluidization" of the bed surface around a tube, which is vertically immersed into a minimally fluidized bed, consisting of large particles ($d_p = 1$ mm). The velocity profile is in agreement with the only experimental result, known from literature, Korolev (see BOTTERILL (1977)).

In our approach the overall time constant of the heat transfer mechanism is taken equal to:

$$\tau = 2\tau_v \tau_p / (\tau_v + \tau_p)$$ (3.46)

$\tau_v$ and $\tau_p$ are respectively the particle agglomerate and the particle size scale time constant. The particle agglomerate time constant is equal to the inverse of the frequency $v$, of the rising voids, as discussed in the preceding section. Equation (3-46) permits a smooth transition from situations with relatively small $\tau_v$, to situations with relatively small $\tau_p$. The gas film near the wall is supposed to be renewed by particles sliding along the surface. The time constant of this particle size scale surface renewal mechanism is equal to:

$$\tau_p = d / \bar{V}_T$$ (3.47)

The gas renewal length $d$ is taken equal to the centre diagonal of a cubic particle matrix or:

$$d = 1.73 \sqrt[3]{\frac{\tau}{(1-\varepsilon) \varepsilon}}$$ (3.48)

The assumptions expressed in equations (3-46), (3-47) and (3-48) comply qualitatively with the general picture of the gas and particle motion near the wall presented earlier. The values of the coefficients in the equations are interconnected and can hardly be verified separately. For calculation purposes, both a layer to layer and a layer to wall radiative heat sink are introduced. Such in order to accommodate formally for the increasing porosity near the wall. The energy emitted per unit volume of the layer to layer heat sink reads:

$$\phi_p = \frac{\delta q_R}{3\kappa}$$ (3.49)

Absorption of radiation by the gas between the particles and/or heat exchanging surface can be neglected, because the particle/particle and particle-surface distances are very small.
The radiative flux \( q_R \) is written, assuming black body radiation for the particles:

\[
q_R = \varepsilon_R \sigma F(T_R - T_w)
\]  

(3-50)

The emission coefficient \( \varepsilon \) of the wall (oxidized steel) is equal to 0.9, PERRY (1973). According to ZABRODSKY (1966), see also section 2.3.4, the emission coefficient of the fluidized bed is 1.0. The view factor of the radiative layer to layer heat sink, see figure 3-19, is equal to 1.0.

The effective thermal conductivity of the fluid is:

\[
\lambda_{eff} = \lambda_f + E_f 
\]  

(3-52)

The coefficient of thermal dispersion \( E_f \) is associated with the fluid percolating through the interstices between the particles near the wall. Analogously to packed beds, ZEHNER (1973), the coefficient of thermal dispersion is taken proportional to the product of the volumetric heat capacity of the fluidizing fluid and the transversal hydrodynamic dispersion as corrected for fluidized beds in Appendix A2, or:

\[
E_f = 0.10 \rho_f C_{pf} D_T
\]  

(3-54)

In the calculations all thermal properties of the fluidizing fluid are taken equal to those of air. The reason for this approximation is two fold. Firstly chemical aspects are not taken into consideration in this thesis. And secondly during our experiments, the carbon dioxide concentration was low, only about 7%, which means that the error due to deviating gas composition is small.

To solve the one dimensional energy equation (3-41), the central difference method of Crank-Nicolson is applied, see for example THRING (1975). After substitution of the initial and boundary conditions and discretisation, the set of simultaneous difference equations, characterized by a tridiagonal matrix, is solved by elimination. With reference to the grid, it should be remarked that the time step is taken equal to 1/20 of the time constant (3-46) and that the wall layer is divided into sublayers with a thickness of 0.08 \( d_p \). The minimum number of sublayers used is 10. The number is incremented stepwise with 8 (up to a maximum of 100) until the temperature of the last sublayer is equal to the temperature of the bulk of the bed. The results of the calculations and some conclusions about the turbulent fluidized bed heat transfer model are presented in the following section.
3.4.3. Some numerical results.

3.4.3.1. Introduction.

As regards the numerical results obtained with the constructed model, the four subjects which can be distinguished in this section, concern respectively:

- particle size scale and overall time constant
- properties of the enhanced wall porosity layer
- influence of convection, radiation and dispersive conduction
- heat transfer coefficient as a function of operating conditions.

3.4.3.2. Time constants of renewal mechanism.

In this section the inverse of the time constant of particle size scale renewal (particle size scale frequency) and the overall renewal time constant are discussed.

In figures 3-20 and 3-21 the particle size scale frequency is depicted as a function of operating conditions. Both figures show that the frequency increases with increasing superficial fluidization velocity which is in agreement with MARTIN (1980), see Appendix A1.

Figure 3-20 shows that an increasing operating pressure results in an increased frequency, only in the upper part of the superficial fluidization velocity range. Figure 3-21 illustrates that the particle size scale frequency increases both for increasing particle size and increasing (dimensionless) superficial fluidization velocity. Comparing this frequency with the particle agglomerate (void) frequency, see section 3.3.3.3, two conclusions can be drawn for PFBC conditions (medium to high superficial fluidization velocities). Firstly the particle size scale renewal frequency is at least one order of magnitude larger than the particle agglomerate one. And secondly the trend of the particle size scale renewal frequency is opposite the particle agglomerate one.
The overall time constant after equation (3-46) is depicted in figures 3-22 and 3-23 as a function of operating conditions. Both figures show the overall time constant to decrease with increasing superficial fluidization velocity. Figure 3-22 shows that increasing operating pressure results in a slightly decreased overall time constant. And figure 3-23 illustrates that for increasing particle size, the time constant decreases as a function of the dimensionless superficial fluidization velocity. This last trend is in agreement with ERNST (1959).

Neglecting the influence of heat conduction in the particles near the heat exchanging surface is permitted, if the Fourier number is smaller than 0.1. As a consequence the time constant should be smaller than 90 ms for silica sand with a size of 780 μm. To fulfil this requirement, the superficial fluidization velocity should have a medium to high value, as found by PETROVIC (1985). Also our experiments (see Chapter 5) are in accordance with this theoretical result.

FIGURE 3-22.
Overall time constant (3-46) as a function of superficial fluidization velocity for different operating pressures.
Standard operating conditions, but for \( p \) and \( J \).

FIGURE 3-23
Overall time constant (3-46) as a function of dimensionless superficial fluidization velocity for different particle sizes.
Standard operating conditions, but for \( d_p \) and \( J \).

FIGURE 3-24
Porosity profile in wall layer as a function of dimensionless distance from the wall.
Standard operation conditions, but \( J = 0.95 \text{ m/s} \).
3.4.3.3. Wall layer properties.

The calculated porosity, local fluid velocity and temperature profile in the wall layer are presented in this section.

The porosity profile is depicted in figure 3-24 and the velocity profile of the fluidizing fluid in figure 3-25, given as a function of dimensionless distance from the wall in the both cases.

As a consequence of the correction factor equation (3-45), the velocity profile has a maximum for $x/d_p = 0.32$. Further (not shown in figures), the ratio $U(x/d_p = 0.32) / U$ decreases with increasing superficial fluidization velocity, as to be expected.

The temperature profiles of the enhanced porosity wall layer are depicted as a function of the dimensionless distance for different operating conditions in figures 3-26, 3-27 and 3-28.

In all calculations the wall temperature is taken 343 K. The steepness of the temperature profiles exhibits the same trend as the overall time constant (figures 3-22 and 3-23). Figure 3-26 illustrates that with increasing superficial fluidization velocity, the bed temperature is reached for somewhat smaller dimensionless distance. Comparing figures 3-26b and 3-27 it appears that increasing the operating pressure has a similar effect.

![Graphs showing fluid velocity and temperature profiles](image-url)
Comparing figures 3-26a and 3-28 it appears that for increasing particle size but constant dimensionless superficial fluidization velocity, the bed temperature is reached for a smaller dimensionless distance from the wall. However, in spite of the mentioned trends, it should be remarked that the steepness of the temperature profile changes only marginally as a function of the operating conditions.

![Graph showing temperature profile](image)

**FIGURE 3-28.**
Temperature profile in wall layer as a function of dimensionless distance from the wall.
Standard operation conditions, but \( d_p = 1500 \text{ \mu m} \) and \( J/J_m = 5 \).

3.4.3.4. The components of turbulent fluidized bed heat transfer.

The contributions of the convective, radiative and dispersive conductive components and the total turbulent fluidized bed heat transfer are discussed in this section. Figures 3-29, 3-30 and 3-31 show some results of calculations, which serve as starting points. As the components influence each other, it is impossible to single out the contribution of a component separately. Instead the various contributions are estimated by just omitting the various effects one for one in the calculations. Following this indirect approach, it appears from the three figures that the contribution of convection increases with increasing superficial fluidization velocity, increasing operating pressure and increasing particle size. For high values of these three parameters it becomes of more interest than the contribution of radiation.

![Graph showing calculated turbulent fluidized bed heat transfer coefficient](image)

**FIGURE 3-29.**
Calculated turbulent fluidized bed heat transfer coefficient as a function of dimensionless superficial fluidization velocity for different theoretical assumptions.
Standard operating conditions.

- : all components included
- - - : without convection
- - - : without radiation
- - - - - : without dispersive conduction.

Besides it should be remarked that these trends of the convective contribution are opposite to the trends of the time constant and the steepness of the temperature profile in the wall layer, discussed in sections 3.4.3.2 and 3.4.3.3.
The radiative contribution to the turbulent fluidized bed heat transfer increases slightly with increasing superficial fluidization velocity. Comparing figures 3-29 and 3-31, it appears that the radiative contribution decreases with increasing operating pressure.
From figures 3-29 and 3-30 it can be seen that for constant dimensionless superficial fluidization velocity, the radiative contribution of the turbulent fluidized bed heat transfer slightly decreases with increasing particle size. These trends are somewhat confusing, but starting from the one dimensional energy equation (3-42) they can be explained, see Appendix A8.

The trends of the contribution of the dispersive conduction on the turbulent fluidized bed heat transfer are identical to those of convection.

FIGURE 3-30.
Calculated turbulent fluidized bed heat transfer coefficient as a function of dimensionless superficial fluidization velocity for different theoretical assumptions.
Standard operating conditions, but \( d_p = 1500 \) \( \mu \)m.
- - - : all components included
- - - : without convection
- - - : without radiation
- - - - : without dispersive conduction.

3.4.3.5. Turbulent fluidized bed heat transfer.

In figures 3-32 and 3-33 the heat transfer coefficient is depicted as a function of operating conditions. The figures show that the heat transfer coefficient increases with increasing superficial fluidization velocity and increasing operating pressure. Figure 3-33 further shows the heat transfer coefficient to increase with increasing particle size. The trends just mentioned agree with the experimental findings at ambient temperature of Xavier and Borodulva, see both GRACE (1980) and BORODULVA (1983). The theoretical results are (more or less) in satisfactory agreement with our own experimental findings, see section 5.3.
Further it should be remarked that the turbulent fluidized bed heat transfer model is not applicable for low- and high superficial fluidization velocities. The lower applicability limit is determined by the particle Reynolds number, which should be larger than about 20. So as to make the Fourier number small enough (less than 0.05) to neglect any contribution of the particle conductive heat transfer, independent of the Biot number. From this point of view the turbulent fluidized bed heat transfer model developed in the foregoing, can be considered as a gas convective heat transfer approach. This is in agreement with the views expressed by Borodulya, see GRACE (1980). For PFBC fluidization conditions the particle Reynolds number is generally larger than 20. The turbulent fluidized bed heat transfer model is not valid at high superficial fluidization velocity, because, as indicated in Appendix A2, some of the relations applied in the model are not valid for porosities beyond 0.80.
CHAPTER 4 EXPERIMENTAL SET-UP

4.1. Introduction.

In this chapter the experimental set-up for the experimental verification of the model of the two-phase flow and the heat transfer in a pressurized fluidized bed combustor is discussed.

Firstly a general description of the combustor is given. Then the start-up procedure, the steady state operating and the safety precautions of the test rig are described.

Secondly the instrumentation is described. This instrumentation encompasses heat transfer measurement segments, mounted on the heat exchanger and a newly constructed local dynamical heat transfer probe, providing information on the influence of the local fluctuations in the bed on the heat transfer. Further details are given on the bed pressure gradient measurement technique and the newly constructed high temperature impedance probe, serving the determination of local porosity fluctuations.

The third part of this chapter is related to the acquisition and reduction of the obtained data.

Finally the processing of the data of the different used measurement techniques is given.

4.2. Pressurized fluidized bed combustor.

Figure 4-1 shows a schematic flow sheet of the PFBC. The bed diameter is 0.485 m and the total bed freeboard height is 2.00 m. Silica sand with a mean diameter of 780 \( \mu \text{m} \) is used as bed material, see Appendix A3.

Compressed air from two high pressure compressors of the nearly Laboratory of Aero- and Hydrodynamics enters the bed through a distributor, after having been preheated in the annulus between the water cooled pressure vessel and the inner vessel. The flue gas clean-up is achieved by a high temperature, high pressure cyclone and an atmospheric wet cyclone. The heat exchanger - and pressure vessel cooling system - is a closed circuit, filled with demineralized water. Coal and gas or oil can be supplied to the combustor respectively via a top- and under bed feed system. By means of two solid feed systems, coal and inert bed material are dumped through a chute on the top of the fluidized bed. By a pressure lock system, the pressurized storage bunker is periodically filled from an atmospheric one.
The pressurized bunker and the screw feeder were implemented at the start of the second part of the research program, to get a more accurate supply of solids, to enlarge the feed rate and to improve the reliability.

Two different heat exchangers and air distributors are used. Initially the pressurized fluidized bed combustor was provided with a bed freeboard U-tube heat exchanger, see figure 4-2. Based on practical experiences, the U-tube heat exchanger has been replaced by a bayonet pipe heat exchanger with half the heat exchanging capacity. First a perforated plate air distributor was used, see figure 4-2. This distributor consisted of two staggered perforated plates to prevent leakage during down time. Simultaneously with the heat exchanger change, the distributor has been changed into a nozzle distributor. In every nozzle an atomizer is mounted to supply natural gas or oil to the fluidization air. During these reconstructions, a bed ash off take has been mounted as well. See figure 4-3 for the modified testrig.

4.3. Operating the pressurized fluidized bed combustor.

In this section the start-up procedures, the steady state operating and the safety precautions of the testrig are described.

4.3.1. Start-up procedures.

The start-up procedure of the PFBC has shown a strong development in the course of the years.

Figure 4-4 depicts the starting procedure as empirically developed for the original PFBC testrig, with the U-tube heat exchanger and the perforated plate air distributor, VAN DEN BURGH (1979). This start-up procedure was controlled by hand. In order to avoid too strong cooling of the fluidized bed by the U-tube heat exchanger during the heating-up, the bed height had to be restricted to about 0.16 m, until the working temperature of 800 °C had been achieved by means of a premixed gas burner, inserted from above to 0.12 m into the bed, and the intermittent feeding of coal after a bed temperature of 500 °C had been surpassed. After that the height of the bed and the pressure could be gradually increased to the desired test conditions. This start-up procedure took about 4 hours.

The long duration of this procedure was the main reason for changing the distributor, as described in section 4.2. A further important improvement was obtained by the implementation of the automatic on line control.
FIGURE 4-2.
Original pressurized fluidized bed combustor.

FIGURE 4-3.
Modified pressurized fluidized bed combustor.
Start-up procedure for the fluidized bed combustor with U-tube heat exchanger and perforated plate air distributor. Controlled manually.

developed by KOOL (1983). Figure 4-5 depicts the starting procedure development for the test rig with the bayonet pipe heat exchanger and the nozzle air distributor, VAN DEN BURGH (1981). The start-up time of the fluidized bed at atmospheric pressure, which is hand controlled, takes 6 minutes. This is in the order of magnitude of the temperature response time of a fluidized bed which, according to GIBBS (1975), equals:

$$\tau = \frac{m_s C_p}{M_a C_p}$$  \hspace{1cm} (4-1)

After changing over to oil, the on line control takes over. Within 45 minutes the bed is raised-up to the minimal required test conditions.

Start-up procedure for the fluidized bed combustor with bayonet pipe heat exchanger and nozzle air distributor. Automatically on line control.

which are characterized by a bed height of 0.15 m and a pressure of 0.1 MPa. For coal, an adapted procedure is carried out. At a bed pressure of 0.2 MPa the coal supply is started and manually increased, while the oil mass flow is decreased automatically. After cutting-off the oil flow, the coal feed system is embodied in the on line control system.
4.3.2. Steady state operation.

Except for the high temperature, the high pressure outlet- and the bypass inlet valve, the testrig is controlled from a control room. Originally the test conditions, e.g. mass flow, fluidization velocity, bed pressure, bed temperature etc. are presented on a television screen by means of a continuously scanning Doric Digitrend 240 process monitor and an IBM 5100 minicomputer, KASSELS (1980) and LIEFHEBBER (1980). For operating purposes, the flue gases are sampled after the outlet valve and the oxygen-, the carbon dioxide- and the carbon monoxide percentages are recorded. The solid feed system is controlled by a PC-100 personal computer. The computer displays the feed rate and the total amount of coal consumed. The change of the pressurized bunker is on-off controlled by a minimum- and maximum level indicator. In the second part of the research program the bed pressure differential analyzer, see section 4.4.3. which measures amongst others the total bed pressure drop, is used to control the bed content. During the experiments, the solid bed content and consequently the total bed pressure drop were kept constant, the latter on 7 kPa. This corresponds at minimum fluidization conditions to a static bed height of $(0.45 \pm 0.02) \text{ m}$ for the silica bed material used and the bed porosity at such conditions of 0.4 (see Appendix A3). This height is in agreement with the overall in- and output balance of bed material of the testrig.

In the last stage of the research program, the on line control of K00L (1983) has taken over all these control- and display functions. The range of possible test conditions of the combustor is given in table 4-1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Superficial fluidization velocity</td>
<td>$0.9 &lt; J &lt; 1.5 \text{ m/s}$</td>
</tr>
<tr>
<td>Bed pressure</td>
<td>up to 1.0 MPa</td>
</tr>
<tr>
<td>Bed temperature</td>
<td>$\approx 1100 \text{ K}$</td>
</tr>
<tr>
<td>Maximum air flow</td>
<td>0.65 kg/s</td>
</tr>
</tbody>
</table>

**TABLE 4-1.**

Range of possible test conditions of the PFBC.

The minimum- and maximum conditions are determined by respectively non-homogeneous mixing at too low velocity, which can result in sintering, and entrainment of bed material at too high velocity.

4.3.3. Safety precautions.

To protect the testrig against excessive temperatures and/or pressures, some safety precautions are taken, VAN DER BURGH (1979), (1981). In case of flame failure of the start-up burner, which is detected by an ionisation cell, a minimum waiting time is imposed prior to new ignition. The fuel feed is cut-off for temperatures lower than 650 °C and higher than 950 °C. Excessive temperatures up to 1100 °C only occurred during the first try-outs of the start-up procedure, when too much coal was fed into the bed. In such situations, generally severe sintering occurred.

The design pressure of the testrig is 5.0 MPa to resist any explosions. Other pressure safety precautions are pressure relief valves in respectively the air mass flow measurement segment and the air inlet line. The high temperature, high pressure outlet valve is spring loaded. For bed pressures higher than 1.2 MPa, the fuel feed and air flow are cut-off. This happened once, when the bed pressure line was blocked. In case of mechanical failure of the cooling water pump, the fuel feed is cut-off and to prevent too excessive temperatures in the heat exchanger during this emergency and melt down of the testrig, a prepressurized water tank has been incorporated in the closed cooling water circuit. In case of electrical power end, the whole equipment of the testrig turns down. Subsequently the testrig is taken out of service in the same way as in the case of failure of the cooling water pump just mentioned. The air supply remains guaranteed because of the high pressure buffer vessels, which are part of the high pressure air compressor system.
4.4. Instrumentation.

4.4.1. Introduction.

The instruments, used to control and to operate the pressurized fluidized bed combustor have been discussed in the preceding section. In this section, the instrumentation to obtain general- or specific fluidized bed information will be considered.

This instrumentation encompasses:
- the equipment to measure the bed temperature profile;
- the heat transfer measuring segments of both the U-tube- and bayonet pipe heat exchanger;
- the bed pressure differential analyser (BPDA), serving the determination of the fluidized pressure drop and pressure gradient;
- the high temperature impedance probe (HTIP), developed to measure the local porosity fluctuations, and
- the local dynamical heat transfer probe (LDHTP) for measurement of the heat transfer fluctuations.

4.4.2. Local mean heat transfer measurements.

To obtain the mean fluidized- and freeboard heat transfer coefficient, the bed and freeboard temperatures and the properties of the cooling medium of the heat transfer measurement segments should be determined.

To measure the bed- and the freeboard temperature profile, thermocouples have been mounted on a column, fixed on the distributor plate, and on the heat exchanger. The freeboard thermocouples are provided with a radiation shield.

The mean bed- and freeboard heat transfer coefficients are derived from the temperature increase of the coolant and the cooling water flow. To this end, some segments of different U-tubes respectively bayonet pipes are provided with strategically located thermocouples. See figures 4-2 and 4-6 for the U-tube and figures 4-3 and 4-7 for the bayonet pipe. Because of the Prandtl number, which is about 4 and the Reynolds number, which is about 30,000 it can be assumed, that the cooling water in the measurement segments is perfectly mixed, see e.g. KAYS (1980). By a sonar flow meter it has been checked, that the cooling water flow is evenly distributed over the U-tubes of the heat exchanger.
The cooling water flow of the bayonet pipe heat exchanger is evenly distributed, because of the high pressure drop in the annulus of the bayonet pipe.

For both types of heat transfer measurement segment, the absolute measurement error of the heat transfer coefficient is about 20%. This is mainly caused by the temperature difference of the coolant, which is small and difficult to measure.

**FIGURE 4-8.**
Schematic of the bed pressure differential analyser (BPDA).
### Measurement Principles

<table>
<thead>
<tr>
<th>Measurement Principle</th>
<th>Reference</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>A. Pressure Probes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1) Single pressure tap</td>
<td>ZHANG (1982)</td>
<td>a) Not well defined measurement volume because the porosity fluctuation is surrounded by a pressure field</td>
</tr>
<tr>
<td>2) Double pressure tap</td>
<td>LORD (1982)</td>
<td>a) Same as A(1)</td>
</tr>
<tr>
<td>B. Optical Probes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1) Fiber optic</td>
<td>LORD (1982)</td>
<td>a) Because of the cooling required, the dimensions of the probe may be too large for disturbance free measurements in the fluidized bed</td>
</tr>
<tr>
<td></td>
<td></td>
<td>b) Relationships between light attenuation and porosity are ill defined, YOSHIDA (1978)</td>
</tr>
<tr>
<td>C. Micro Waves</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1) Micro wave scattering</td>
<td>EASTLUND (1982)</td>
<td>a) No material contact with particles, non-perturbing</td>
</tr>
<tr>
<td></td>
<td></td>
<td>b) Only suitable for overall measurements</td>
</tr>
<tr>
<td></td>
<td></td>
<td>c) The utility of micro wave diagnostic is dependent on the refraction index of the used media</td>
</tr>
<tr>
<td>D. Impedance Probes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1) Electrical discharge</td>
<td>YOSHIDA (1982)</td>
<td>a) Electrical discharge phenomena at high temperatures and elevated pressures are not well known, LANDOLT (1957)</td>
</tr>
<tr>
<td>2) Single needle capacitor</td>
<td>MITTMANN (1982)</td>
<td>a) Length of the signal carrying conductor is limited by parasitical capacitance</td>
</tr>
<tr>
<td>3) Double needle, three point impedance probe</td>
<td>BOELENS (1985)</td>
<td>a) Length of the signal carrying conductor is not limited</td>
</tr>
</tbody>
</table>

**TABLE 4-2.** Dynamic probes for fluidized bed combustion research.

#### 4.4.2. High Temperature Impedance Probe

The sensor of the high temperature impedance probe, which in the first instance has been developed in co-operation with the Department of Applied Physics of the Delft University of Technology, VAN DER STAR (1979), essentially consists of two parallel needles, see figure 4-9, mounted on a water cooled fin. The time varying amount of bed particles between these needles, gives rise to a changing electromagnetic impedance. For an oil fired pressurized fluidized bed combustor, both silica sand and flue gases can be considered as a dielectric with a significant conductivity, BOELENS (1985). The inductance component of the impedance is negligible for the used non-magnetic silica sand.

![FIGURE 4-9. Drawing of the high temperature impedance probe (HTIP).](image)

The electronic measurement system of the probe is based on an alternating current driven bridge in conjunction with the principle of coherent detection as commonly used in lock-in amplifiers, see figures 4-10 and 4-11. The driving frequency of the bridge is 20 - 200 kHz. For a relative probe admittance change due to a varying amount of bed material between the needles, the transfer function of the bridge can be written as, see figures 4-10 and 4-11:

\[
H = \Delta Y/(Y_1 + Y_2 + Y_D)
\]  

(4-2)
FIGURE 4-10.
Schematic of electronic measurement system of the HTIP. The location of the probe admittances between the pins a and b is given in figure 4-11, BOELENS (1985).

In practice, the admittance of the denominator of equation (4-2) mainly consists of the parasitical capacitance $Y_3$ of the probe coaxcables (about 50 pF/m). Starting from the analytical expression for two infinitely long parallel wires, the admittance change of the sensor due to porosity fluctuations is:

$$
\Delta Y = \Delta C - j \frac{\Delta G}{\omega}
$$

The capacitance- and conductance changes are respectively:

$$
\Delta C = \pi \left( \epsilon_{r1} - \epsilon_{r2} \right) \epsilon_{\text{vacuo}} \frac{1}{k}
$$

and:

$$
\Delta G = \pi \left( \sigma_1 - \sigma_2 \right) \frac{1}{k}
$$

For a mixture of a gas and non-spherical particles, the logarithmic mixing rule appears to be a good approximation to determine both the relative permittivity $\epsilon_r$, KNOE (1980) and the conductivity $\sigma$, ZWIKKER (1966). For a binary mixture of silica sand and flue gases, this rule reads:

$$
\ln \epsilon_{rm} = (1-c) \ln \epsilon_{rS} + c \ln \epsilon_{rG}
$$

The relative permittivity of the silica sand and the flue gases, which are both independent of the temperature, are respectively 3.9 and 1.0, BOELENS (1985). The relative permittivity of the flue gases is independent of its composition. With these values and the in figure 4-9 given dimension, the capacity change, corresponding with the transition of a dense phase to a void with respectively a porosity of 0.4 and 1.0 is equal to 0.34 pF. The conductivities of the silica sand and the flue gases, which are strongly temperature dependent, are respectively 1.0 m($\Omega$ m)$^{-1}$ and 0.50 m($\Omega$ m)$^{-1}$ at 1150 K. Similarly the change in conductance is 1.5 m($\Omega$)$^{-1}$. 
Because the conductance change is strongly dependent on the composition of the bed material, the HTIP can also serve for solid residence time distribution measurements. This may be illustrated by the ratio of the conductance change of a bed with or without a volume fraction $\gamma$ of solid tracer, which reads, BOELENS (1985):

$$\frac{\Delta G_t}{\Delta G} = \left(\frac{\sigma_t}{\sigma}\right)^\gamma$$  \hspace{1cm} (4-9)

For example, the ratio of the conductivities of silica sand and coal ash is of the order of magnitude of 10,000. Consequently a small amount of ash gives a detectable change of the probe signal.

The capacitance change of the probe can be used to determine the porosity in a fluidized bed as a function of time, because of its insensitivity to temperature fluctuations. The calibration of the high temperature impedance sensor and its theoretical time constant are discussed in Appendix A5. The probe, which is provided with two sensors just above each other, can be traversed vertically in the PFBC, see figures 4-3 and 4-9.

### 4.4.5. Local dynamical heat transfer measurements.

#### 4.4.5.1. Introduction.

Today's fluidized bed heat transfer models suffer from a lack of information on the local mean- and momentary heat transfer. To a considerable extent, the heat transfer is correlated with the local dynamical nature of the gas particle flow, see Chapters 2 and 3. Experimental data on the local dynamical heat transfer are very scarce. The dynamical nature of the heat transfer in a fluidized bed manifests itself in surface temperature fluctuations, going with the heat transfer. To measure such temperature fluctuations, MICKLEY (1961) and recently MILLER (1982) have developed a low thermal capacity heater, made out of platinum foil, with a time constant of about 8.5 m s, HINZE (1975), and suitable for atmospheric fluidized bed combustor conditions. However, as shown by Miller, the calibration of such a complex probe is rather hazardous at fluidized bed conditions.

Because of this, a local dynamical heat transfer probe (LDHTP) probe has been developed to measure the surface temperature fluctuations.

The new local dynamical heat transfer probe (LDHTP) is a thick walled bayonet pipe heat flux meter with small surface thermocouples, see figure 4-12 and HEESTERBEEK (1982), (1983).

To determine the mean- and varying component of the surface temperature, thin chromel-alumel thermocouples with a diameter of 0.25 mm are embedded in a groove with a depth of 0.30 mm. To prevent an excessive cooling medium pressure drop, only the measuring part of the instrument, which consists of AISI 316 stainless steel, is provided with a thick wall. Parasitical heat flows are suppressed by a well defined cool water flow, an insulating groove at the circumference of the pipe wall, filled with low thermal conductivity ceramical material, Aremco Ultra-Temp ($\lambda = 0.75$ W/m K), and by a low thermal conductivity ceramic cap at the tip of the probe. Using the probe as a heat flux meter, the local averaged heat...
transfer coefficient can be obtained. Based on the amplitude and frequency of the surface temperature, the local dynamical heat transfer coefficient can be determined. In the frequency range that is of interest, 1-5 Hz, the time constant, which is about 2 ms, and the dimensions of the probe are such as to enable an undistorted measurement of the local dynamical heat transfer, see Appendix A6. Traversing the probe vertically in the testrig, see figure 4-3, the local dynamical heat transfer of both the fluidized bed and the splashing zone can be measured.

4.5. Data acquisition and reduction.

4.5.1. Introduction.

The three different data acquisition and reduction systems, which can be distinguished, are related to:
- the operating conditions and steady state heat transfer measurements
- the BPDA
- the HTIP and LDHTP

The data acquisition and reduction systems are discussed in the following sections.

4.5.2. Operating conditions and steady state heat transfer measurements.

The data on the operating conditions of the testrig and/or the data of the heat exchanger are sampled on request 5 times by a Doric Digitrend 240 process monitor and stored in an IBM-5100 minicomputer, KASSELS (1980) and LIEFHEBBER (1980). The mean value and the standard deviation of these data are calculated by the minicomputer and stored on tape. Afterwards the reduced data are transmitted to and stored on a data set of the Amadahl computer of the Computer Centre of the Delft University of Technology. Processing these data sets, the steady state heat transfer coefficients can be obtained as a function of the pressurized fluidized bed conditions.

4.5.3. Data Acquisition of the BPDA.

A PC-100 personal computer, of which the output electronics have been modified to control the magnetic valve system, is the central device for the data acquisition of the BPDA, COPPENS (1981), and see figure 4-13. The PC-100 scans the A/D converter 10 times with a sample frequency of 1 Hz. To prevent any zero bias of the pressure differential transducer, the pressure connections are switched halfway the scanning period. Further the PC-100 is programmed in such a way, that on request, one or all pressure differences are scanned continuously in a loop. The moving average(s) and standard deviation(s) are presented on a television screen and printed if desired. A RAM is added to the PC-100 to permit quick loading or reloading of the program. Purging the whole system with air, the blocking of the measuring linear is prevented.
4.5.4. Data Acquisition from the HTIP.

The signals from the high temperature impedance probe are processed on a LSI 11/23 computer with a RT-11 operating system. The specific operating conditions, copied from the control monitor and the sampled data, are stored on a floppy disk, see figure 4-14. For the analysis of the probe signal, the gas particle flow phenomena in a fluidized bed are considered to be a second order ergodic random process in a frequency range up to about 8 Hz. The following functions are used to analyse the signals in the amplitude-, time-, and frequency domain:

- Amplitude Probability Density Function (APDF)
- Auto- and Cross Correlation Function (ACF and CCF)
- Fast Fourier Transformation (FFT) and Maximum Entropy (MEM) power spectrum estimate.

The APDF gives information about the signal level distribution from a statistic point of view and, possibly, indicates periodicity of the signal concerned, BENDAT (1980). The amplitude density histograms are obtained by the summing of the observations from 40 equal class intervals. The number of class intervals has been selected to satisfy the 5 percent level of significance of a Chi-square goodness of fit test, BENDAT (1966).

The ACF is a computation technique, which makes it easy to determine the period of a possible periodical component in the more or less random signal and also yields its mean and variance. The numerical expression for the ACF, BENDAT (1966), is:

$$R_{xx}(r) = \frac{1}{N-r} \sum_{n=1}^{N-r} x_n x_{n+r} \quad r = 0,1,2 \ldots, m \quad (4-10)$$

To obtain an accurate estimated autocorrelation, the time step $\tau$ and the number of data points should be sufficiently large. Generally the number of points $m$, over which the autocorrelation is shifted, is taken to one tenth of the number of data points. The sampling frequency and the number of data points are respectively 45 Hz and 512 points.

The CCF is used to determine the time delay between the two sensors of the HTIP. The numerical expression for the CCF, BENDAT (1966), is:

$$R_{xy}(r) = \frac{1}{N-r} \sum_{n=1}^{N-r} x_n y_{n+r} \quad r = 0,1,2 \ldots, m \quad (4-11)$$
In general terms, the requirements for an accurate estimated CCF are the same as for the autocorrelation. However, a higher sampling frequency and a larger number of data points are necessary. The sampling frequency and the number of data points are respectively 330 Hz and 1052 points.

Recently CHILDERS (1978) and KAY (1981) have given an extensive review on spectral analysis, which covers several aspects, such as: consistency, variance, noise sensitivity and frequency resolution. In this thesis only the general aspects of the Fast Fourier Transformation (FFT) and the Maximum Entropy power spectrum estimate are discussed.

The FFT is widely used as a computationally efficient method to determine the power spectral density function. The general form of the power spectral density function is well visualized by this method. However, it has a poor ability to resolve spectral peaks. Extensive averaging techniques, implying a large number of data points (long recording times), are required to obtain a satisfactory accuracy. The discrete FFT power spectral density function, KAY (1981), is:

\[
P(v_m) = \frac{1}{N} |X_m|^2\]  (4-12)

where the discrete Fourier transformation is defined as:

\[
X_m = \sum_{n=0}^{N-1} x_n \exp(-j2\pi mn/N)\]  (4-13)

and the frequency \(v_m\) is equal to:

\[
v_m = m/2N \quad ; \quad r = 0,1,2 \ldots, N-1\]  (4-14)

The sampling frequency of 45 Hz used for the ACF is sufficient to meet the requirements of the FFT power spectral density function. The FFT power spectral density function is acquired by averaging the separate FFT's of 16 subsequent segments of 128 data points, each zero padded with 128 zeros. To reduce leakage effects, a Hanning window is applied.

To discover the probable occurrence of a significant frequency peak in the power spectrum, the maximum entropy (MEM) spectral estimation is applied. The main advantage of this technique over the more conventional FFT method is an improved frequency resolution for a relative short data record length. The MEM is essentially based on a more general class of procedures, collectively known as autoregression.

Essentially the MEM aims at obtaining the unknown discrete terms of the autocorrelation function \(R_x((m+1)t)\), \(R_x((m+2)t)\), etc from the known terms \(R_x(0)\), \(R_x(t)\), \(\ldots, R_x(m)\). This extrapolation takes place such that the 'entropy' of the resulting power spectrum sampled with a frequency \(v_s\):

\[
\int_{-v_s/2}^{v_s/2} \ln P(v) \, dv
\]  (4-15)

is maximal, Burg, see KAY (1981). This means, that the fewest constraints are imposed on the unknown terms of the autocorrelation by maximizing its randomness and consequently giving a minimum bias solution. In other words, the chosen spectrum compromises itself minimally with the unknown part of the autocorrelation. The power spectrum is found by maximizing equation (4-15) and assuming that the Wiener-Kinchin relation is valid for the known terms of the autocorrelation. Using the Lagrange multiplier technique, the solution of the Wiener-Kinchin relation is found to be:

\[
P(v) = \frac{P_m}{\prod_{m=1}^{M} \left|1 - \sum_{n=1}^{M} a_{mn} \exp(-j2\pi mn)\right|^2}\]  (4-16)

\(P_m\) and the prediction coefficient \(a_{mn}\) are determined by using the Yule-Walker equation, which reads in matrix form, see e.g. ANDERSON (1974):

\[
\begin{pmatrix}
R_{xx}(0) & R_{xx}(1) & R_{xx}(m) \\
R_{xx}(1) & R_{xx}(0) & R_{xx}(m-1) \\
R_{xx}(m) & R_{xx}(m-1) & 0
\end{pmatrix}
- \begin{pmatrix}
a_{m1} \\
a_{m2} \\
\vdots
\end{pmatrix} = \begin{pmatrix}
P_m \\
0 \\
0
\end{pmatrix}\]  (4-17)

The prediction (reflection) coefficients \(a_{mn}\) are obtained by the Burg algorithm. The essence of this algorithm is, that the forward- and backward prediction energy \(e_m\) of a linear prediction error filter is minimized. ANDERSON (1974):
Using the Levinson recursion, KAY (1981), $S_m$ can be determined. Now for increasing model order $m$, a solution is found. The optimum estimated power spectrum is defined as the one, which yields, on basis of an estimation criterium, the minimal loss of information. For this purpose the Akaike information criterium is used, AKAIKE (1971). The MEM has been applied to 256 data points, sampled at a frequency of 45 Hz.

### 4.5.5. Data acquisition of the LDHTP

For the determination of the local averaged heat transfer coefficient, the interior- and exterior thermocouples of the probe are directly connected with the LSI 11/23 computer, see figure 4-15. After averaging the outside temperature the thermocouple data are stored, together with the specific operating condition data, on a floppy disk. Because of limited measuring range of the LSI 11/23, the dynamical component of the exterior thermocouple signal is processed prior to its being presented to the LSI 11/23 computer, see figure 4-15.

![Diagram](image-url)

**FIGURE 4-15.**

Data acquisition of the LDHTP.
4.6. Data processing.

4.6.1. Introduction.
Subjects of this section are the data processing of the heat transfer measurements, the BPDA, the HTIP and the LDHTP. Because the data processing of the operating conditions of the PFBC is rather straightforward, it will not be discussed.

4.6.2. Data processing of the heat transfer measurements.
The temperature profile of the fluidized bed and its freeboard are visualized by printing the temperatures in a scheme of the test rig.

The overall heat transfer coefficient of the bed - and freeboard measurement segments of the U-tube and the bed - and splashing zone measurement segment of the bayonet pipe is given by:

$$h = \frac{M_{H_2O} C_{pH_2O} \Delta T_{H_2O}}{A \Delta T_{ lm}} \quad (4-19)$$

Because of the isothermal bed and nearly isothermal splashing zone condition and because:

$$\frac{\Delta T_{H_2O}}{(T_b - T_{H_2O, out})} \ll 1 \quad (4-20)$$

the log mean temperature difference of the heat transfer measurements concerned can be approximated by:

$$\Delta T_{ lm} = T_b - T_{H_2O, out} \quad (4-21)$$

4.6.3. Data processing of the BPDA signal.
The data processing of the BPDA will now be discussed. The air distributor pressure drop is equal to:

$$\Delta P_d = C_d \frac{1}{2} \rho \frac{V^2}{2} \quad (4-22)$$

See for a further discussion of the air distributor Appendix A4.

4.6.4. Data processing of the HTIP signal.

Based on the new theory, which describes the dynamical nature of the gas particle flow, see Chapter 3, and own experimental findings, see Chapter 5, the porosity fluctuations are given in terms of a plane harmonic longitudinal wave or:

$$\epsilon = \bar{\epsilon} + \epsilon_0 \sin \omega (t - \frac{Z}{W_w}) \quad (4-25)$$

Calibrating the probe, see Appendix A5, the averaged porosity $\bar{\epsilon}$ and the amplitude of the porosity fluctuation $\epsilon_0$ can be obtained from the APDF. The radial frequency $\omega$ is defined by means of the maximum of the maximum entropy power spectrum. The propagation velocity of the wave $W_w$ is equal to:

$$W_w = \frac{d}{\tau} \quad (4-26)$$

d is the distance between the two high temperature impedance sensors, see figure 4-9. The time delay $\tau$ of a wave passing these two sensors is obtained from its CCF.

4.6.5. Data processing of the LDHTP signal.
4.6.5. Data processing of the LDHTP signal.

The local averaged heat transfer coefficient is determined by the temperature difference between the mean outer- and inner measured temperature or:

\[ h = \frac{\bar{q}_w}{(T_b - T_u)} \quad (4-27) \]

with:

\[ \bar{q}_w = \lambda_w \left( \frac{r_1 - r_2}{\ln (r_1/r_2)} \right) \quad (4-28) \]

The indices 1 and 2 correspond with respectively the exterior and interior positioned thermocouples, see figure 4-12. Assuming the wall to be a semi-infinite medium and the local dynamical fluidized bed heat transfer to be a harmonic function of time, then the alternating component of the bed wall heat transfer coefficients reads, CARSLAW (1959):

\[ h_0(t) = \frac{q_w(x=0,t)}{(T_b - T_w(x=0, t))} \quad (4-29) \]

with:

\[ T_w(x,t) - T_w(x=0) \exp \left(-x \left(\frac{nu}{a}\right)^{0.5}\right) \sin \left[ut - x \left(\frac{nu}{a}\right)^{0.5} \right] \quad (4-30) \]

\[ q_w(x,t) = q_w(x) \sin \left[ut - x \left(\frac{nu}{a}\right)^{0.5} \right] \quad (4-31) \]

and:

\[ q_w(x) = \lambda_w T_w(x=0) \left(\frac{2nu}{a}\right)^{0.5} \exp \left(-x \left(\frac{nu}{a}\right)^{0.5}\right) \quad (4-32) \]

Assuming \( T(x=0,t) << T \) equation \(4-29\) reduces to:

\[ h_0(t) = \frac{q_w(x=0)}{(T_b - T_w)} = h_0 \sin \left(ut + \frac{x}{\frac{a}{2}}\right) \quad (4-33) \]

Determining the amplitude and the frequency of the temperature fluctuations on depth \( x \) by means of respectively the APDF and the maximum of the maximum entropy power spectrum, the local dynamical heat transfer coefficient is known. See for the calibration of the probe Appendix A6.

CHAPTER 5 EXPERIMENTAL RESULTS

5.1. Introduction.

In this chapter the experimental results concerning the dynamical behaviour of the gas-particle flow in terms of porosity fluctuations, and the closely related dynamical heat transfer are discussed, respectively in sections 5.2 and 5.3.

Most measurements were taken at the following conditions of the PFBC, further referred to as standard operating conditions:

- superficial fluidization velocity \( J \) based on the free horizontal cross section area of the bed: 1.05 m/s
- mean size \( d_p \) of silica sand bed material: 780 \( \mu \)m
- bed temperature \( T_b \): 1123 K
- static bed height \( \bar{H} \): 0.45 m

Additional conditions (operating pressure) and any deviation from the standard conditions are indicated ad hoc.
5.2. Porosity fluctuations.

5.2.1. Introduction.

In the second part of the research program some experiments were made on porosity fluctuations in the PFBC by means of the BPDA and the HTIP. In the following sections the results are given and compared with correlations known from literature.

5.2.2. BPDA porosity fluctuations.

The BPDA measures the pressure field surrounding a void. Because of its undefined "measurement volume" the BPDA is only suitable to determine the mean pressure difference and possibly the frequency of the voids. Figure 5-1 gives a typical registration of the signal of the BPDA concerning the total bed pressure drop. Both a low frequency fluctuation with a rather large amplitude and a (quasi) random fluctuation of higher frequency are observable.

Data on the porosity fluctuations are obtained from the BPDA pressure difference measurements between the second and third pressure tap, which are located respectively at \( h = 0.25 \) and \( 0.35 \) m, see figure 4-3. The mean of at least seven separate measurements (each encompassing 10 measuring points) is taken to be representative for the mean porosity. Further the relative standard deviation of the porosity is tentatively assumed to be equal to the relative standard deviation of the pressure drop.

Table 5-1 gives the results thus obtained. For comparison the porosities as obtained from the HTIP, see section 5.2.3, and the porosities as calculated from the semi-empirical correlation of Babu (1978) and the modified Ergun equation (A2-9) are also presented.

<table>
<thead>
<tr>
<th>( p ) (MPa)</th>
<th>( N )</th>
<th>( \varepsilon ) (BPDA) mean st.dev</th>
<th>( \varepsilon ) after Babu (2.12)</th>
<th>( \varepsilon ) after modified Ergun (A2-9)</th>
<th>( \varepsilon ) (HTIP) mean amplitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.4</td>
<td>42</td>
<td>0.51 0.07</td>
<td>0.59</td>
<td>0.58</td>
<td>0.54 0.14</td>
</tr>
<tr>
<td>0.5</td>
<td>28</td>
<td>0.52 0.09</td>
<td>0.59</td>
<td>0.59</td>
<td>0.54 0.12</td>
</tr>
<tr>
<td>0.6</td>
<td>28</td>
<td>0.54 0.12</td>
<td>0.59</td>
<td>0.60</td>
<td>0.54 0.09</td>
</tr>
<tr>
<td>0.7</td>
<td>7</td>
<td>0.56 0.09</td>
<td>0.59</td>
<td>0.61</td>
<td>0.54 0.09</td>
</tr>
<tr>
<td>0.8</td>
<td>7</td>
<td>0.58 0.08</td>
<td>0.59</td>
<td>0.62</td>
<td>0.55 0.08</td>
</tr>
</tbody>
</table>

TABLE 5-1.
Measured BPDA mean porosity with standard deviation, calculated mean porosities and measured mean and amplitude of the HTIP porosity at \( h = 0.3 \) m (see following section) as a function of the operating conditions. \( N \) is the number of BPDA measurements used for averaging.

The measured porosities are in good agreement. For increasing operating pressure the standard deviation of the PBDA passes through a maximum, corresponding to large pressure fluctuations. The amplitude of the HTIP porosity behaves differently and decreases steadily with increasing pressure. Both observations confirm the turbulent fluidized bed model as described in Chapter 3 and are in agreement with Canada (1978), (1978a).

The measured porosities differ considerably from the calculated ones. For increasing operating pressure the PBDA and calculated mean porosities tend to agree better.
5.2.3. HTIP porosity fluctuations.

5.2.3.1. Introduction.

The porosity fluctuations as measured by the HTIP are discussed in this section. The following four subjects are treated:

* the nature of the HTIP signal
* the amplitude probability density function (APDF)
* the autocorrelation function (ACF) and the FFT and MEM spectral density function
* the cross correlation function (CCF).

5.2.3.2. Nature of the HTIP signal.

Figure 5-2 gives a typical registration of the signal of the HTIP as obtained in the pressurized fluidized bed combustor.

![Signal Registration](image)

The signal indicates a lower- and upper porosity level of about 0.40 and 0.80 respectively. The lower porosity corresponds with the packed bed porosity. The lowest measured porosity of 0.3 seems not acceptable.

However, the following two arguments deny this point of view. Firstly the dense rhombohedral packings have a porosity of 0.26, HAUGHHEY (1969), and secondly small particles may fill up the interstices between the larger ones. The upper porosity indicates, that the "voids" in the bed are not particle free. The same conclusion follows from the APDF of the HTIP signal, see section 5.2.3.3. No amplitude corresponding to a porosity of 1.0 is found, in other words no particle free voids occur. Also the local dynamical heat transfer experiments, to be described in section 5.3.3, indicate, that particle free voids do not occur at PFBC conditions. That is to say at least at the pressure range 0.4 to 0.9 MPa, a height of 0.15 to 0.6 m and the particle size of 780 µm investigated.

This finding is in conflict with most of the results published until now. However, (see KUNII (1969), p. 121) Rowe has demonstrated, that in a two dimensional bed, stalactites of particles are raining through undisturbed bubbles and Toei has shown that small obstacles may cause similar phenomena. Recently Rowe, see GRACE (1980) and CHITESTER (1984), has observed at ambient temperatures, that respectively at high superficial fluidization velocities and increased operating pressures, bubbles tend to "fill" with particles. And further it is in agreement with the turbulent fluidization model as described in section 3.3.

The main cause of the moderate void porosity is - in the authors opinion - the large particle size in the pressurized fluidized bed combustor; \( d_p = 780 \) µm. At ambient conditions and for large particles (\( d_p = 1000 \) µm) bed material is raining through slugs, see section 2.2.3.

And finally it should be noted, that since the measurement volume of the probe (= 30 mm \(^3\)) is nearly two orders of magnitude smaller than the volume element (= 1 cm \(^3\)) as assumed in Chapter 3, the obtained HTIP information is representative for the micro scale phenomena in the bed.

5.2.3.3. APDF of the HTIP signal.

The APDF of two HTIP signal recordings are depicted in figure 5-3. The figure indicates, that the porosity fluctuations in the fluidized bed can be considered as a sine wave in Gaussian noise, BENDAT (1980). The mean porosity and the negative and positive sine wave amplitudes based on hand drawn enclosing curve (see figure 5-3) are given in tables 5-2 and 5-3 as a function of operating conditions. Because of the position of the vertical traversing line of the HTIP in the bed the turbulent zone porosity is measured, see figure 4-3.
FIGURE 5-3.
Typical APDF of the HTIP signal.
Standard operating conditions, but h = 0.3 m and a) J = 1.25 m/s and p = 0.4 MPa
b) J = 1.05 m/s and p = 0.9 MPa.

TABLE 5-2.
The mean porosity and the negative- and positive sine wave amplitude of the porosity fluctuation as a function of operating pressure and height, as measured by the HTIP in the pressurized fluidized bed combustor.
Standard operating conditions, but J = 1.25 m/s and p = 0.4 MPa.

<table>
<thead>
<tr>
<th>p (MPa)</th>
<th>h (m)</th>
<th>e_T^-</th>
<th>e_To-^-</th>
<th>e_To+^-</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.4</td>
<td>0.25</td>
<td>0.58</td>
<td>0.21</td>
<td>0.16</td>
</tr>
<tr>
<td></td>
<td>0.35</td>
<td>0.58</td>
<td>0.20</td>
<td>0.15</td>
</tr>
<tr>
<td></td>
<td>0.45</td>
<td>0.58</td>
<td>0.21</td>
<td>0.15</td>
</tr>
<tr>
<td></td>
<td>0.55</td>
<td>0.58</td>
<td>0.18</td>
<td>0.18</td>
</tr>
<tr>
<td></td>
<td>0.65</td>
<td>0.58</td>
<td>0.18</td>
<td>0.18</td>
</tr>
<tr>
<td>0.7</td>
<td>0.15</td>
<td>0.59</td>
<td>0.17</td>
<td>0.12</td>
</tr>
<tr>
<td></td>
<td>0.20</td>
<td>0.59</td>
<td>0.17</td>
<td>0.11</td>
</tr>
<tr>
<td></td>
<td>0.60</td>
<td>0.59</td>
<td>0.14</td>
<td>0.10</td>
</tr>
<tr>
<td>0.8</td>
<td>0.15</td>
<td>0.58</td>
<td>0.15</td>
<td>0.10</td>
</tr>
<tr>
<td></td>
<td>0.30</td>
<td>0.58</td>
<td>0.14</td>
<td>0.09</td>
</tr>
<tr>
<td></td>
<td>0.45</td>
<td>0.57</td>
<td>0.14</td>
<td>0.08</td>
</tr>
<tr>
<td></td>
<td>0.60</td>
<td>0.55</td>
<td>0.11</td>
<td>0.11</td>
</tr>
<tr>
<td>0.9</td>
<td>0.30</td>
<td>0.57</td>
<td>0.12</td>
<td>0.09</td>
</tr>
</tbody>
</table>

TABLE 5-3.
The mean porosity and the negative- and positive sine wave amplitude of the porosity fluctuations as a function of height as measured by the HTIP in the pressurized bed combustor.
Standard operating conditions, but J = 1.25 m/s and p = 0.4 MPa.

As already mentioned in the preceding section, these results are in good agreement with the BPDA measurements. Based on these two tables it may be concluded, that the mean porosity of the turbulent zone is independent of the operating conditions in the range investigated. For a turbulent zone fraction of 0.80 and a WLR porosity of 0.40 the overall mean porosity of the bed is equal to 0.54.

At constant superficial fluidization velocity both e_To-^- and e_To+^- tend to decrease slightly both for increasing height at constant pressure and increasing pressure at constant height. The first trend can be explained by a decreasing turbulent zone fraction with increasing height, see section 3.3.2.3. The second trend is a consequence of decreasing amplitude amplification factor with increasing operating pressure, see section 3.3.3.3.

Comparing table 5-2 with 5-3 it appears, that both e_To-^- and e_To+^- decrease slightly with increasing superficial fluidization velocity which is in agreement with the turbulent fluidization model, see section 3.3.2.3. The remarkable fact, that e_To^-^- is larger than e_To+^- can be explained by the turbulent fluidization model.
5.2.3.4. Spectral analysis of the HTIP signal.

The ACF and the FFT- and MEM spectral density function of the HTIP signal are discussed in this section. These two functions are the power spectra in respectively time and frequency domain. Further the dominating frequency, as defined by the MEM power spectrum, is presented as a function of operating conditions.

In this thesis the ACF is used to detect in an easy way any periodical component in the HTIP signal. It is presented in figure 5-4. The dominating frequency following from the rather weak periodicity is about 5 Hz.

In figure 5-5 the FFT spectral density function is depicted. The spectrum shows, that the HTIP signal can be considered as a low frequency band noise with a significant contribution in the lower frequencies. It should be noted, that the lowest frequencies, which manifest themselves in practice as a signal drift, may partly have been caused by oscillations of the probe in the fluidized bed. Because of the poor frequency resolution and the variance of the estimator of the FFT technique, no well defined peak(s) is (are) found in these FFT power spectra. The MEM spectral density estimation technique is better suited to this purpose. Figure 5-6 gives examples of the MEM spectral density function for two operating conditions. Because the MEM technique is very sensitive to signal drift.
only about half of the recorded data files actually showed a peak in the spectral density function. The author considers the maximum of the MEM power spectrum as the best available indication of the dominating frequency of the porosity fluctuations. The frequencies thus found are depicted in figures 5-7 and 5-8 for various operating pressures and heights. The frequency decreases with increasing height as well as operating pressure.

5.2.3.5. CCF of the HTIP signal.

Figure 5-9 gives the CCF of the lower and upper sensors of the HTIP, see figure 4-9.

Because of a too low scanning frequency the curve has a flat peak and as a consequence the time delay, which is defined by the maximum, is afflicted with a possible error of 13%. The mean rise velocity of the voids is equal to the ratio of the distance between these two sensors and the time delay. The rise velocities thus calculated are depicted in figures 5-10 and 5-11 as a function of height and operating pressure. Figure 5-10 show the rise velocity to increase with height up to h = 0.3 m. Both figures suggest, that the rise velocity increases slightly with increasing operating pressure.
Rise velocity of the porosity wave as a function of operating pressure.

Standard operating conditions, but for $h$ and $p$:
- $p = 0.4$ MPa
- $x = 0.5$ MPa
- $p = 0.6$ MPa.

5.3. Heat transfer.

In the first part of this section the steady state heat transfer coefficients as measured by using of the U and bayonet pipe measurement segments and the LDTH are presented.

In the second part the dynamical component of the fluidized bed heat transfer coefficient is discussed, as measured by the LDTH.

5.3.1. Mean PPDC heat transfer.

5.3.1.1. Introduction.

The mean heat transfer coefficients as measured by the just mentioned instruments are discussed in this section. Because the heat transfer experiments are executed at different positions in the PPDC, it is more accurate to consider the results as local mean heat transfer coefficients of vertical tubes immersed into a fluidized bed.

5.3.1.2. U-tube heat transfer.

As described in section 4.2.4 a U-tube is provided with thermocouples to measure the heat transfer in the bed and in the splashing zone freeboard region. The heat transfer coefficients as measured in the U of the fluidized bed are presented in table 5-8. The heat transfer coefficients increase with increasing operating pressure, but the influence of the superficial fluidization velocity is not significant. The results may have been influenced by the geometry of the U-tube heat exchanger configuration, which favours the passage of gas through the U in the centre of the bed, see figure 5-2. It should be noted, that because of the manual control of the testrig the accuracy of the temperature and pressure settings of the fluidized bed does not exceed 2%. As a consequence the superficial fluidization velocity has a possible error of 5%. Due to these not well defined operating conditions the heat transfer coefficient is afflicted with a significant error. The total possible error of the heat transfer coefficient is 20%. The error finds its origin mainly in the difficult to measure small temperature increase of the coolant of the heat exchanger. The results of the measurements in the WR and the splashing zone freeboard region are given in Appendix 7. The heat transfer coefficients found in these regions are equal to about 100 W/m$^2$ K, with a total possible error of 40%.
5.3.1.3. Bayonet pipe heat transfer.

The bayonet pipe heat exchanger is provided with bed- and splashing zone heat transfer measurement segments at four different distances from the centre of the fluidized bed, see section 4.2.4. Due to technical failure of a number of thermocouples reliable results were only obtained for the bed segments at the radii of 88.5 and 132.5 mm. The results are depicted in figure 5-12 as a function of the operating pressure. Compared to the U tube results the bayonet pipe heat transfer coefficients show a much smaller, if any, dependency on operating pressure. The heat transfer coefficient increases slightly toward the axis of the fluidized bed. This providing (weak) support to the existence of a "turbulent zone" in the bed.

Table 5-4.

<table>
<thead>
<tr>
<th>J [m/s]</th>
<th>p [MPa]</th>
<th>h [W/m² K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5</td>
<td>0.5</td>
<td>460</td>
</tr>
<tr>
<td></td>
<td>0.6</td>
<td>430</td>
</tr>
<tr>
<td></td>
<td>0.7</td>
<td>570</td>
</tr>
<tr>
<td>1.6</td>
<td>0.5</td>
<td>460</td>
</tr>
<tr>
<td></td>
<td>0.6</td>
<td>570</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>580</td>
</tr>
<tr>
<td></td>
<td>0.9</td>
<td>570</td>
</tr>
<tr>
<td>1.7</td>
<td>0.5</td>
<td>460</td>
</tr>
<tr>
<td></td>
<td>0.6</td>
<td>520</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>660</td>
</tr>
<tr>
<td>1.8</td>
<td>0.5</td>
<td>500</td>
</tr>
<tr>
<td></td>
<td>0.6</td>
<td>500</td>
</tr>
</tbody>
</table>

5.14

**TABLE 5-4.**

TZ heat transfer coefficient as measured by means of the U tube measurement segment as a function of superficial fluidization velocity and operating pressure.

Standard operating conditions, but for J and p and T_b = 1100 K and H_mf = 0.4 m.

Table 5-4 gives some heat transfer coefficients measured at 0.4 MPa and two different superficial fluidization velocities.

<table>
<thead>
<tr>
<th>J [m/s]</th>
<th>h at r = 88.5 mm [W/m² K]</th>
<th>h at r = 132.5 mm [W/m² K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.05</td>
<td>590</td>
<td>610</td>
</tr>
<tr>
<td>1.25</td>
<td>600</td>
<td>630</td>
</tr>
</tbody>
</table>

**TABLE 5-5.**

Bayonet pipe heat transfer coefficients as a function of superficial fluidization velocity at an operating pressure of 0.4 MPa. Standard operating conditions, but for J.

It is indicated, that the heat transfer increases slightly with increasing superficial fluidization velocity. However, the small number of data does not justify any hard conclusion.
The accuracy of the bayonet pipe measurements is to some extend superior to that of the U tube measurements. Such because of the automatic control of the testrig during the bayonet pipe measurements, virtually leading to exact setting of the operating conditions and the temperature increase of the coolant of the bayonet pipe, which is larger and better to measure. The total possible error of the measured bayonet pipe heat transfer coefficient is 20%. Even so it makes little sense to enter upon any further trends, that might be read from figure 5-12, or to carry comparison with the U tube results beyond the remarks made above.

5.3.1.4. Mean LDHTP heat transfer.

Traversing the local dynamical heat transfer probe (LDHTP) vertically in the centre of the bed the mean local heat transfer of both the fluidized bed and the splashing zone was determined. The experimental results are depicted in figure 5-13, 5-14 and 5-15, HEESTERBEEK (1983). They show a weak influence of both the superficial fluidization velocity and the operating pressure.
As a function of height the heat transfer coefficient shows a flat peak near the top of the fluidized bed, which coincides with the static bed height.

The mean fluidized bed heat transfer coefficient as measured with the LDHTP is approximately equal to 500 W/m² K for all operating conditions investigated. In the small operating interval of 0.4 to 0.6 MPa, the mean splashing zone heat transfer coefficient, i.e. at a height of 0.6 m, is nearly constant and equal to 350 W/m² K; as expected it is lower than the heat transfer coefficient in the fluidized bed. In an atmospheric fluidized bed combustor, NAUDE (1980) has found similar trends for the heat transfer coefficients.

The total possible error of the measured LDHTP heat transfer coefficients is 6%. The LDHTP heat transfer coefficients are generally lower than those measured with the bayonet pipes. However, their error intervals overlap. A possible explanation of the difference is, that the LDHTP attracts voids or influences the turbulent two phase flow in a way different from the bayonet pipes. Such because the bayonet pipes are inserted into the fluidized bed from above and the LDHTP from below.

5.3.2. Dynamical component of the PFBC heat transfer.

5.3.2.1. Introduction.

Subject of this section is the dynamical component of the fluidized bed heat transfer as measured by means of the LDHTP, HEESTERBEEK (1983). Subsequently the following topics are discussed:

* the nature of the LDHTP signal
* the APDF
* the ACF and the FFT and MEM spectral density functions.

5.3.2.2. Nature of the LDHTP signal.

Figure 5-16 shows a typical record of the signal of the LDHTP. The signal exhibits remarkable differences with the experimental results of MICKLEY (1961), an example of which is given in figure 5-17. Neither the discrete levels, nor the exponential decay curves are observed with the LDHTP. These differences have to be a consequence of the differences in the operating and measuring conditions as listed in table 5-6.
TABLE 5-6. Main operating and measuring conditions of the local dynamical heat transfer measurements of MICKLEY (1961) and this thesis.

<table>
<thead>
<tr>
<th>Operating and measuring conditions</th>
<th>MICKLEY (1961)</th>
<th>this thesis LDHTP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluidized bed diameter [m]</td>
<td>0.104</td>
<td>0.485</td>
</tr>
<tr>
<td>Fluidized bed height [m]</td>
<td>0.75 to 1.5</td>
<td>0.45</td>
</tr>
<tr>
<td>Operating conditions</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Superficial fluidization velocity [m/s]</td>
<td>up to 0.55</td>
<td>up to 1.25</td>
</tr>
<tr>
<td>Temperature [K]</td>
<td>ambient</td>
<td>1123</td>
</tr>
<tr>
<td>Pressure [MPa]</td>
<td>ambient</td>
<td>0.4 to 0.9</td>
</tr>
<tr>
<td>Bed material mean [μm]</td>
<td>glass beads</td>
<td>silica sand</td>
</tr>
<tr>
<td>Probe design</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Type</td>
<td>low thermal</td>
<td>thick wall heat</td>
</tr>
<tr>
<td>Capacity meter</td>
<td>surface</td>
<td>flux meter with</td>
</tr>
<tr>
<td></td>
<td>thermocouples</td>
<td></td>
</tr>
<tr>
<td>Exterior diameter [mm]</td>
<td>6.35</td>
<td>30</td>
</tr>
<tr>
<td>Time constant [ms]</td>
<td>8.5</td>
<td>2</td>
</tr>
</tbody>
</table>

As indicated in Chapter 2, the experiments of Mickley were executed in the particle free bubbling or slugging two phase flow regime. The heat transfer to tubes immersed in such a fluidized bed is characterized by a lower- and upper level corresponding respectively with a bubble and a slug, the latter going with a fresh agglomerate of bed material. The exponential decay as a function of time of the upper level indicates, that the heat transfer mechanism of such an agglomerate can be described by a penetration theory.

The LDHTP experiments are executed in the turbulent two phase flow regime, see Chapter 3, the dynamical behaviour, is best described by means of sinusoidal porosity waves. The porosity fluctuations have to a considerable extent, a random character as well as the heat transfer fluctuations resulting from them. Therefore no well defined lower and upper level of the heat transfer is found and virtually no indication of the exponentially decaying heat transfer, characteristic for a penetration heat transfer mechanism.

Instead the fluctuations of the LDHTP probe signal point to a dispersive conductive- and to a certain extent convective heat transfer mechanism following from the turbulent fluidized bed heat transfer model developed in section 3.4. According to this model the maxima and minima of the recorded signals correspond with the highest and lowest porosity in the fluidized bed.

5.3.2.3. APDF of the LDHTP signal.

A typical APDF of the LDHTP signal is depicted in figure 5-18. As expected (preceeding section) no discrete levels can be distinguished in this density function.

FIGURE 5-18. APDF of the LDHTP signal.

Standard conditions, but p = 0.9 MPa and h = 0.3 m.

In figure 5-19 and 5-20 the amplitude of the dynamical component of the heat transfer is shown as a function of respectively height and operating pressure. Because sinusoidal porosity waves are assumed the amplitude is calculated as \(a = \sqrt{2} \sigma\), with \(\sigma\) the standard deviation of the surface temperature from its mean value. Both figures show that the amplitude is almost independent of height and operating pressure. At a height of \(h = 0.35\) m the amplitude of all the executed experiments shows a flat maximum of about 150 W/m².

The trends of the amplitude of the dynamical component of the heat transfer are further scrutinized in Chapter 6.
5.3.2.4. Spectral analysis of the LDHTP signal.

In this section the spectral properties of the LDHTP signal are discussed, as reflected in the ACF and the FFT and MEM spectral density functions. In the last part of this section the dominating frequency, as defined by the MEM spectrum, is given as a function of operating conditions.

Figure 5-21 gives the ACF of the LDHTP signal. The ACF indicates a dominating frequency of about 3 Hz.

The FFT spectral density function is depicted in figure 5-22. The spectrum indicates that the LDHTP signal can be considered as a low frequency band noise. However, as a consequence of the poor frequency resolution and the variance of the estimator no well defined peak frequency is found in the FFT power spectra. Because the LDHTP is better adjusted than the HTIP, see figure 4-3, less low frequency noise has been detected.

Applying the MEM technique, which accentuates peaks in the power spectrum, a maximum is found in the power spectrum density function, see figure 5-23. As it has been said before (section 5.2.3.4) this maximum is considered to be the best available indication of the dominating frequency of the porosity waves and associated phenomena.
In figure 5-24, 5-25 and 5-26 the dominating frequency is depicted as a function of the operating conditions. Up to the static bed height of 0.45 m these figures show the dominating frequency to be almost independent of height, operating pressure and for a bed pressure of 0.4 MPa, also of the superficial fluidization velocity in the range investigated. Above 0.45 m, in the so-called splashing zone, the dominating frequency decreases slightly.

The independency of the LDHTP frequency of height is not in agreement with the HTIP findings, see section 5.2.3. As already remarked before, this can possibly be explained by the fact that the LDHTP attracts voids or influences the turbulent two phase flow in a way, different from the HTIP. Such because the HTIP is inserted into the fluidized bed from above and the LDHTP from below.

The trend of the LDHTP frequency as a function of operating pressure is in agreement with the theory of dynamics of gas particle flow, see section 3.3.3.3.

Referring to the just mentioned theory, the frequency should decrease with increasing superficial fluidization velocity, however, this is not confirmed by experiments.
FIGURE 5-25.
Dominating frequency of the LDHTP signal as a function of height.
Standard operating conditions, but \( J = 1.25 \) m/s and \( p = 0.4 \) MPa.

FIGURE 5-26.
Dominating frequency of the LDHTP signal as a function of operating pressure.
Standard operating conditions, but \( h = 0.30 \) m.

5.4. Conclusions.

In the following chapter the experimental results are scrutinized and compared with the ones as obtained from the "classical" theory and the newly developed turbulent fluidization and heat transfer model, see respectively Chapter 2 and Chapter 3.
CHAPTER 6  DISCUSSION OF RESULTS; RECOMMENDATIONS.

6.1. Introduction.

In this section, the experimental findings as given in the previous chapter, are briefly scrutinized from the point of view of both the "classical" theories (Chapter 2) and the newly developed turbulent fluidization and heat transfer approach (Chapter 3). Subsection 6.2 concerns the fluidization and subsection 6.3 the fluidized bed heat transfer. Finally in section 6.4 some general conclusions and recommendations are presented.

6.2. Fluidization.

6.2.1. Introduction.

In this section, the experimental findings as presented in Chapter 5, are discussed from the point of view of bubbling and/or slugging and the turbulent fluidization model. Successively the following subjects are discussed:

* porosity fluctuations
* void frequency
* void rise velocity
* bubble diameter.

6.2.2. Porosity fluctuations.

Mainly the nature of the HTIP porosity fluctuations and their APDF are discussed in this section. Afterwards some remarks are made about the mean value of the overall porosity. Most remarkable in the measurements is the fact, that the upper porosity level is equal to about 0.80, see figure 5.2. The LDHTP signal supports this, because the heat transfer mechanism cannot be explained by a penetration theory. The APDF of the HTIP indicates, that the porosity fluctuations can be considered as a sine wave in Gaussian noise. The more or less artificially defined amplitude of the porosity fluctuations (see figure 5-3) decreases with height, operating pressure and superficial fluidization velocity, (see tables 5-2 and 5-3). The APDF of the LDHTP cannot confirm or contradict this, because its signal to noise ratio is too low, (see figure 5-18).
The just mentioned properties of PFBC porosity fluctuations cannot be interpreted in terms of (fast or slow) bubble models. The observed phenomena can only be explained by slugs, through which particles rain downward. The pressure fluctuations, such as measured with the BPDA, confirm this conclusion. However, the observation that the standard deviation of the BPDA passes through a maximum at an operating pressure of 0.6 MPa, cannot be explained by conventional slugging data, (see table 5-1). Only the turbulent fluidization model is able to explain both the observed upper porosity level as well as the APDF profile of the HTIP signal. The maximum of the standard deviation of the BPDA measurements should then be interpreted as an indication of the transition from heterogeneous (bubbling) fluidization into a more homogeneous fluidization.

As indicated in section 5.2.2, the measured averaged BPDA and HTIP overall porosities are in agreement with each other. However, the measured- and calculated porosities only agree for increased operating pressure. In Appendix A2 it is shown, that the semi-theoretical modified Ergun equation agrees well with the empirical correlations of Babu and Denloye.

6.3.2. Void frequency.

The experimental findings, concerning void frequency are not identical for the HTIP and the LDHTP. The HTIP void frequency decreases both with increasing height and operating pressure, (see figures 5-7 and 5-8). The LDHTP void frequency is of the same order of magnitude as the HTIP one, but is hardly influenced by these two parameters, (see figures 5-24, 5-25 and 5-26). As already mentioned in Chapter 5, this difference can possibly be explained by the fact, that the LDHTP attracts voids or influences the two phase flow in a way different from the HTIP. Such because the HTIP is inserted into the fluidized bed from above and the LDHTP from below. The approximately constant LDHTP void frequency of 2.3 s⁻¹ indicates, that due to "void attraction of the probe" (coalescence) bubbles get their maximum size or slugging is fully developed at relative low bed heights. Due to coalescence, the HTIP frequency approaches this value of 2.3 s⁻¹ in the upper part of the bed. Increasing operating pressure has a stimulating influence on this coalescence. This interpretation of the HTIP frequency is confirmed by the observed void rise velocity, as discussed in the following section. As just indicated, the trends of the experimentally determined void frequency can be interpreted in terms of bubbling or slugging.

The predicted void frequencies of the turbulent fluidization model are of the same order of magnitude as the experimental ones. The dependency of HTIP frequency on height and/or operating pressure, can be explained by a conjectured decrease of the turbulent zone fraction.

6.3.4. Void rise velocity.

Due to presumably coalescence, the voids accelerate up to a height of about 0.3 m, reaching a constant velocity there. An increase of operating pressure results in an increasing rise velocity in only the lower part of the operating range, (see figure 5-11). Assuming a constant bubble size for heights larger than 0.3 m, the trends just mentioned can be explained by a bubble model. Taking this bubble size equal to the diameter of the free central part of the combustor, then the coefficient Y (2-13) varies from 0.9 to 1.8 for respectively low- and high operating pressures. This value of Y, however, is much smaller than the theoretically assumed one, which is equal to 3.5. In the upper part of the operating pressure range, the slow bubble concept does not hold, because bubbles with a size larger than the bed diameter then have to be assumed. So these rise velocity data are only compatible with the fast bubble fluidization regime.

The trends also indicate, that for h > 0.3 m, slugging should occur in the PFBC testrig. As indicated in section 2.2.3, the rise velocity of an asymmetric slug with a horizontal diameter determined by the bed diameter, corresponds quite well with the experimental findings.

The void rise velocity, as predicted by the turbulent fluidization model, is of the same order of magnitude as the experimental one. The dependency of rise velocity on height can be explained by a conjectured decrease of the turbulent zone fraction. However, this model fails to explain the observed dependency of the rise velocity on operating pressure.
6.2.5. Bubble diameter.

In this section the empirically determined bubble diameter is discussed, initially assuming that all observed voids are bubbles. As a start, the necessary relations are briefly reviewed. For both fast- and slow bubble models, the expression for the bubble diameter reads:

\[ D_B = 1.5 \frac{c_B}{U_B/v} \]  

(2-10)

The void frequency and rise velocity, both as determined by the HTIP, are respectively given in the figures 5-7 and 5-8 and the figures 5-10 and 5-11. The bubble phase porosity follows from the general identity:

\[ (1-\epsilon) = (1-\epsilon_B) (1-\epsilon_d) \]  

(2-11)

The dense phase porosity \( \epsilon_d \) is equal to 0.4. And the overall porosity is taken equal to the one as measured with the BPDA, (see table 5-1).

In tables 6-1 and 6-2, the bubble diameter calculated from these relations is presented as a function of height and operating pressure. It should be noted, that the data given in these tables were taken from the lines, fitted by a computer in the mentioned figures. Consequently the tables show some small discrepancies.

<table>
<thead>
<tr>
<th>p = 0.4 MPa</th>
<th></th>
<th>p = 0.6 MPa</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( h )</td>
<td>( v_B (= v) )</td>
<td>( u_B (= U_B) )</td>
</tr>
<tr>
<td>[m]</td>
<td>after fig.5-7</td>
<td>after fig.5-10</td>
</tr>
<tr>
<td>0.15</td>
<td>5.3</td>
<td>1.4</td>
</tr>
<tr>
<td>0.30</td>
<td>4.1</td>
<td>1.7</td>
</tr>
<tr>
<td>0.45</td>
<td>2.9</td>
<td>1.7</td>
</tr>
</tbody>
</table>

TABLE 6-1.

Bubble diameter as a function of height and operating pressure and for experimentally obtained void frequency and rise velocity.

Standard operating conditions, but for \( p \).

Table 6-1 shows, that due to coalescence, the bubble diameter increases with height. Both tables indicate, that for operating pressures larger than or equal to 0.6 MPa, the bubble diameter exceeds both the slugging criterion of Baeyens, \( D_B < 1/\sqrt{D_b} \), see section 2.2.3, as well as the diameter of the free central part of the testrig. The conclusion is, that at increasing height and/or operating pressure, a transition from bubbling to slugging occurs. This is confirmed by the BPDA measurements, which show that at 0.6 MPa the standard deviation passes through a maximum (see table 5-1).

The turbulent fluidization model indicates the possibility of a transition of the kind just described. This model shows that, presumably due to a decreasing turbulent zone fraction as a function of height and/or increasing operating pressure, the fluidization regime changes from particle free- into particle laden bubbling or slugging in the range investigated. In practice the operating pressure of a PFBC will generally be high.

6.2.6. Conclusions.

The main conclusion from the foregoing is that, in terms of "classical" empirical theories, the fluidization regime changes from bubbling into slugging for increasing height and/or operating pressure. However, the internal porosity of the voids deviates considerably from the classical
value of 1.0. Therefore it is, to the author's opinion, better to consider
the fluidization as degenerated bubbling or slugging.
To the author's knowledge, no classical fluidization theory is available,
which can only explain the observed void properties in a straight forward
way. Only the observed void rise velocity might be explained by assuming
fast bubbles to occur in the bed.
The turbulent fluidization model, developed in section 3.3, predicts the
order of magnitude and most of the observed trends quite well. In
particular the transition from predominantly particle free bubbling or
slugging to predominantly particle laden bubbling or slugging at
increasing height and/or operating pressure is satisfactorily predicted by
this theory. Moreover it offers an explanation for the internal porosity
of the voids. Further the trend of the observed HTIP void frequency at
increasing height and operating pressure can be explained by this theory,
if a decreasing turbulent zone fraction is assumed. The trend of the void
rise velocity at increasing height may be explained if a decreasing
turbulent zone fraction is assumed.
For reasons indicated in Chapter 3, the model fails to explain the trends
observed for the void rise velocity as a function of operating pressure.
Obviously the sine longitudinal porosity wave, applied in the turbulent
fluidization model is only a first linear approximation.
As regards the semi-theoretical modified Ergun equation, the author feels
that it may re-open a way to explain the expansion of fluidized beds as
occurring in PFBC's. Further experimental substantiation of this equation
is required, however.

6.3. Heat transfer.

6.3.1. Introduction.

The theoretical- and experimental findings concerning PFBC heat transfer
are summarized in this section. The following subjects are treated:
- stationary (local mean) component of the heat transfer
- dynamical component of the heat transfer.

6.3.2. Local mean heat transfer.

As discussed in section 5.3.1, the local mean heat transfer coefficient of
vertical tubes immersed into the bed, as measured by the U-tube, the
bayonet pipe and the LDHTP exhibit differences, but their error intervals
overlap. The influence of operating conditions (mainly operating pressure)
on the heat transfer was found to be weak.
For PFBC conditions the heat transfer models of both Bock and Ganzha (see
Chapter 2) predict heat transfer coefficients, which are of the same order
of magnitude as the experimental ones. The trends in both models are
weaker, but agree with the trends following from the turbulent fluidized
bed heat transfer approach. Via the particle convective heat transfer
component, both Bock and Ganzha take the thermal properties of the
particles into account. However, for PFBC conditions the gas gap heat
contact resistance is dominant and the influence of the thermal properties
of the particles can be eliminated. Also for a dominating heat contact
resistance, the overall porosity is the governing variable.
The predictions of the turbulent fluidized bed heat transfer model are of
the same order of magnitude as the experimentally determined heat transfer
coefficients. The theoretical influence of the operating condition on heat
transfer is not confirmed by the experiments. It is important to note,
that the turbulent fluidization theory is only applicable for particle
Reynolds number larger than 20, because conductive heat transfer in the
particles is neglected. Unfortunately own experimental findings are too
limited and well defined data in literature too scarce as yet to draw any
hard conclusions regarding the validity of the turbulent fluidized bed
heat transfer approach for mean heat transfer predictions.
6.3.3. Local dynamical heat transfer.

The implications of the observed nature and APDF of the dynamical component of the LDHTP heat transfer coefficient are discussed in this section.

The nature of the LDHTP signal shows (see figure 5-16), that convection and dispersive conduction contribute significantly to PFBC heat transfer. In the turbulent fluidized bed heat transfer model, the contribution of convection is taken into account in a way similar to the approaches of Bock and Ganzha, as described in Chapter 2. In first instance all these three theories are directed towards the calculation of the mean heat transfer coefficient. Consequently they offer no direct way to determine the dynamical component of the heat transfer. The turbulent fluidized bed heat transfer model, however, permits determination of the dynamical component because of its information regarding porosity fluctuations. As pointed out by Wunder and Ganzha (Chapter 2) and Martin (Appendix A1), and similarly adopted in the turbulent fluidized bed heat transfer model, the parameter governing heat transfer is the residence time of the particles close to the surface. Under PFBC conditions, this residence time is roughly two orders of magnitude smaller than the particle agglomerate residence time (equal to the inverse of the void frequency). So the agglomerate renewal due to rising voids, can be taken as a quasi-stationary process, when considered from particle renewal point of view. So the dynamical component of heat transfer follows directly from the changing local flow conditions, going with the passing of porosity waves. This view is confirmed by the observation, that the trends of the amplitude of the dynamical heat transfer component are nearly identical to the trends of the LDHTP frequency (see figures 5-24 and 5-26). Starting from the porosity and its fluctuations as outlined in respectively sections 3.2 and 3.3, the amplitude of the dynamical component of the heat transfer coefficient can be calculated from the turbulent fluidized bed heat transfer model. For an amplitude of the porosity fluctuations of \( \varepsilon_0 \) (see equation 3-12) the results are given in table 6.3 with the operating pressure as parameter. The table indicates, that the amplitude depends weakly on operating pressure and is nearly equal to the experimentally determined value of about 150 W/m² K (see figures 5-19 and 5-20). Concerning this table it should be remarked that in the lower part of the pressure operating range, the lower value of the heat transfer coefficient cannot be calculated, because the Reynolds number is smaller than 20.

<table>
<thead>
<tr>
<th>( p ) [MPa]</th>
<th>( \varepsilon_0 ) (3-3) [-]</th>
<th>( \varepsilon_0 ) (3-12) [-]</th>
<th>( \bar{n} (\varepsilon_0) ) [W/m² K]</th>
<th>( h (\varepsilon_0 + \varepsilon_0^2/4) ) [W/m² K]</th>
<th>( h (\varepsilon_0 - \varepsilon_0^2/4) ) [W/m² K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.4</td>
<td>0.63</td>
<td>0.3</td>
<td>540</td>
<td>690</td>
<td>-</td>
</tr>
<tr>
<td>0.5</td>
<td>0.64</td>
<td>0.3</td>
<td>570</td>
<td>730</td>
<td>-</td>
</tr>
<tr>
<td>0.6</td>
<td>0.65</td>
<td>0.3</td>
<td>610</td>
<td>770</td>
<td>-</td>
</tr>
<tr>
<td>0.7</td>
<td>0.66</td>
<td>0.3</td>
<td>650</td>
<td>820</td>
<td>510</td>
</tr>
<tr>
<td>0.8</td>
<td>0.67</td>
<td>0.3</td>
<td>690</td>
<td>870</td>
<td>540</td>
</tr>
<tr>
<td>0.9</td>
<td>0.68</td>
<td>0.3</td>
<td>720</td>
<td>890</td>
<td>570</td>
</tr>
</tbody>
</table>

Table 6.3.
The mean, upper and lower value of the turbulent fluidized bed heat transfer coefficient as a function of operating pressure.

Standard operating conditions (see Chapter 3), but for \( p \).

It is important to note, that both Bock's and Ganzha's model yield too low an estimate of the amplitude of the dynamical heat transfer component, if this amplitude is calculated from the changes in velocity of the fluidizing fluid, corresponding to the porosity fluctuations. The cause of this deficiency is that dispersive conduction is overlooked in both models.

According to the turbulent fluidized bed heat transfer model, the peak values of the PFBC heat transfer are fully explained by the high momentary velocity of the fluidizing fluid and dispersive conduction occurring with the upper value (about 0.8) of the porosity. The contribution of radiative fluctuations to the dynamical heat transfer component appears to be insignificant.
6.4. General conclusions and recommendations.

The results of our investigations on PFBC can be summarized in the following general conclusions and recommendations:

* The overall properties of the fluidized bed, such as the velocities of fluidizing fluid, fluidized particles and rising voids are incompatible with classical bubble models and are best explained by turbulent fluidization. Verification of the just mentioned phenomena by means of respectively gas- and solid residence time distribution experiments are necessary to perfect this picture.

* The experimental determination and theoretical verification of the fluidized bed expansion is still an important item for fluidized bed combustion research. Also further research is needed to determine the actual value of the turbulent zone fraction as a function of operating conditions.

* The turbulent fluidization model, combined with linear density wave theory correctly predicts the experimentally established transition from particle free- to particle laden bubbling or slugging, at increasing operating pressure and/or particle size and/or fluidization velocity. Further this approach permits explanation of most of the observed dynamical properties of PFBC fluidization in terms of sine longitudinal porosity waves. In order to develop this model further, more experiments on the dynamical phenomena of fluidization should be executed and non-linear effects, e.g. regarding the porosity fluctuations, should be introduced in the theory.

* Classical heat transfer theories as well as the turbulent fluidized bed heat transfer model predict the order of magnitude of the mean heat transfer coefficient reasonable well. The observed (weak) influence of operating conditions on the mean heat transfer coefficient is not correctly predicted, however.

* At PFBC conditions, the particle replacement close to a heat exchanging surface, plays a more important role in heat transfer than the agglomerate renewal due to rising voids.

* The dynamical properties of the PFBC heat transfer can only be explained by the turbulent fluidized bed heat transfer model. For this explanation, the dynamics of heat transfer have to be related to porosity fluctuations.

* Further theoretical- and experimental research concerning the influence of properties of the enhanced porosity wall layer, such as gas- (and) solid flow conditions, conduction of heat in particles near the wall and radiation is recommended.
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APPENDIX A1

Effective particle convective heat transfer after MARTIN (1980).

Somewhat similar to JACOB's (1957) approach for turbulent flow, MARTIN (1980) assumes particle motion and heat transfer in a gas fluidized bed to be similar to molecular motion and heat transfer. Martin's semi-empirical particle convective heat transfer coefficient reads:

\[ h = \frac{1}{C} \rho_s \frac{C_p s}{\rho s s_p} V_p \]  

(A1-1)

The accommodation coefficient \( \gamma_p \) is, after Knudsen, see McADAMS (1954), defined as the ratio of actual- and maximum possible energy exchange. Borrowing from heat exchanger theory, Martin derives:

\[ \gamma_p = 1 - \exp \left( -6h_{p,\text{max}} \frac{t_k}{(\rho_p C_p \sigma s_p d_p)} \right) \]  

(A1-2)

with:

\[ t_k = \frac{4d_p}{C_k s_p} \]  

(A1.3)

\( h_{p,\text{max}} \) is already given in section 2.3.2.1. The empirically determined coefficient \( C_k \) equals 3. Martin postulates a free fall mechanism over the mean free path of the particles. Again borrowing from kinetic gas theory, he derives for the velocity of the particles:

\[ V_s = \left( \frac{g d_p F_M}{M} \right)^{0.5} \]  

(A1-4)

with:

\[ F_M = \frac{\epsilon - \epsilon_{\text{mf}}}{\left[ 5 \left( 1-\epsilon_{\text{mf}} \right) (1-\epsilon) \right]} \]  

(A1-5)

Substituting in the numerator of equation (A1-5) for particle free bubbles the well known expansion relation:

\[ (1-\epsilon) = (1-\epsilon_B) \left( 1-\epsilon_{\text{mf}} \right) \]  

(A1-6)

then \( F_M \) becomes:

\[ F_M = \frac{\epsilon_B}{5(1-\epsilon)} \]  

(A1-7)
This equation indicates, that Martin's criticism on the use of the bubble phase porosity in fluidized bed heat transfer models is not valid, because also his own approach is afflicted with this shortcoming.

The velocity of the particles, \( A1-4 \) and the residence time \( t_k \) of the particles near the wall, which can be considered as a particle size scale time constant, are both in agreement with own theoretical findings, see Chapter 3. Admittedly the results of Martin's model are in good agreement with literature.

APPENDIX A2

Elements of gas particle flow.

A2.1 Introduction.

In this appendix some elements of one-directional gas flow in packed beds are modified to make them applicable to fluidized beds. To this end, the fluidized bed is considered as a uniformly expanded packed bed, and the consequences of this expansion for the gas flow are investigated. The gas flow properties considered are the tortuosity, the pressure drop as given by the Ergun equation and the hydrodynamical dispersion.

The tortuosity, accounts for the meandering of the fluidizing fluid, percolating through the interstices of the bed. Uniform expansion of a bed leads to a reduction of the tortuosity. A tortuosity term is introduced, which accounts for this effect.

The Ergun equation, which gives the pressure drop per unit bed height of a porous medium, is modified by means of the just mentioned tortuosity correction. The modified Ergun equation, then permits calculation of the expansion of beds, consisting of large particles \( d_p > 1 \text{ mm} \).

The hydrodynamic dispersion is the macroscopic average result of the mixing of a fluid on a microscopic scale. It is the sum of mechanical dispersion, which results from tortuosity and molecular diffusion.

A2.2 Tortuosity.

Starting from packed beds, the tortuosity is modified for nearly homogeneously fluidized beds. For a heterogeneous bubbling or slugging bed, such an approach is impossible as yet.

The tortuosity or directional tensor of a porous medium is an operator, which comprises the meandering and the expansions and contractions of a fluid flow path, winding through the pores of a packed bed. Tortuosity is extensively described for packed beds by BEAR (1972). According to CARMAN (1956), the tortuosity can often be considered as a scalar, equal to:

\[
T = \left( \frac{L}{L_e} \right)^2 \leq 1
\]

\[A2-1\]

where \( L \) is the length of a straight line, which connects the ends of a tortuous tube of length \( L_e \). Winsauer and Archie, see BEAR (1972), p. 115 have shown that the tortuosity is equal to:
As a simple approximation, the tortuosity is further assumed to be equal to \( \varepsilon \). The porosity of a packed bed is generally about 0.4 and so its tortuosity \( T_{pb} = 0.4 \).

In a packed bed, the stream lines have a fixed geometry, because every elementary channel is a stream tube fixed in space.

For nearly homogeneously fluidized beds, which are characterized by a more or less regular matrix of fluidized particles, it is assumed that a similar approach is also valid. In other words we assume, that the average tortuosity of a fluidized bed is equal to its porosity:

\[ T = \varepsilon \]  

(A2-3)

A2.3. Modified Ergun equation.

According to ERGUN (1952), the pressure drop going with one dimensional flow through a packed porous medium, is given by:

\[ \frac{dp}{dz} = \frac{\mu}{K} \left( 1 + \frac{\rho_f K_F \varepsilon U}{\mu} \right) \varepsilon U \]  

(A2-4)

The first and second term of the RHS of this equation correspond to respectively viscous- and inertial energy dissipation.

Using the tortuosity concept, the permeability \( K \) and the Forchheimer coefficient \( K_F \) become respectively, according to SCHEIDEGER (1960):

\[ K = d_p^2 \varepsilon^3 T_{mf} \]  

(A2-5)

and:

\[ K_F = 0.00738 \frac{d_p}{(1-\varepsilon) T_{mf}^{0.5}} \]  

(A2-6)

with: \( T_{mf} = T_{pb} = 0.8 \)

Replacing in equation (A2-4) the fluid velocity by the slip velocity between fluid and fluidized particles, because the latter may be moving, one obtains for a (nearly) homogeneously fluidized bed:

\[ \frac{dp}{dz} = \frac{\mu}{K} \left( 1 + \frac{\rho_f K_F \varepsilon |V-U|}{\mu} \right) \varepsilon (V-U) \]  

(A2-7)

The permeability and Forchheimer coefficient are given by equations (A2-5) and (A2-6), but with \( T = \varepsilon \) (equation (A2-3)).

From this equation, the fluid solid interaction force per unit volume, \( F \), can be calculated because, see also section 3.3.2:

\[ F = \varepsilon \frac{dp}{dz} \]  

(A2-8)

For \( |V| \ll |U| \), see section 3.2, and because the pressure drop per unit bed height is equal to gravitational minus buoyancy force per unit volume, the averaged porosity of the fluidized bed can be calculated from:

\[ (1-\varepsilon)(\rho_s - \rho_f) g = \frac{\mu}{K} \left( 1 + \frac{\rho_f K_F J}{\mu} \right) J \]  

(A2-9)

\( J \) is the superficial fluidization velocity and is equal to \( \varepsilon U \).

This equation is in agreement with CARMAN (1956), who indicated that such a relation is valid for liquid-solid fluidized beds.

Up to porosities of 0.8, the bed pressure drop per unit bed height is independent of height, LI (1981). So the bed expansion equation (A2-9) is valid for average porosities up to the just mentioned value.

FIGURE A2.1.

Porosity as a function of superficial fluidization velocity after two authors. Standard operating conditions (see Chapter 3), but \( J \) and \( \varepsilon \): BABU (1978)

: Boelens, this thesis equation (A2-9).
Figures A2-1 and A2-2 show, that up to porosities of 0.80 the results of the semi-theoretical modified Ergun equation (A2-9) are in agreement with BABU (1978) and DENLOYEE (1982). The correlation of Babu (2-12) is based on a compilation of experimental data of different authors. The Babu equation should predict the porosity up to 0.72 within 12%.

Neither Babu nor Denloye give any information on the shear forces that may occur in the fluidized beds investigated. So a more complete comparison of (A2-9) with literature, taking into account shear forces, is impossible as yet.

The average porosity of the fluidized bed is depicted as a function of operating conditions in figures A2-3 and A2-4. The figures show that:

$$10^5 < Ar = \frac{d^2}{p} \left( \frac{\rho_s - \rho_f}{\rho_f} \right) \frac{\rho_f g}{\eta} < 10^7$$  \hspace{1cm} (A2-10)
- the porosity increases with increasing superficial fluidization velocity
- the porosity increases with increasing particle size, and
- the porosity increases slightly with increasing operating pressure.

Equation (A2-9) is in acceptable agreement with own experimental findings, see Chapter 5.

A2.4 Hydrodynamic dispersion.

The hydrodynamic dispersion describes the irreversible spreading of a fluid flowing through a porous medium. Hydrodynamic dispersion is the macroscopic result of the actual movements of the fluid on a molecular- and particle size scale. The first movement results from molecular self-diffusion and the second from fluid flow forces due to, amongst others, interaction between the fluid and the solid matrix of the porous medium.

Consequently the hydrodynamic dispersion \( D_h \) reads:

\[
D_h = D_{\text{MS}} + D_M \quad (A2-11)
\]

\( D_{\text{MS}} \) is the coefficient of molecular self-diffusion, which can be found in standard text books such as BIRD (1960). The convective diffusion \( D_M \), often called mechanical dispersion, is a consequence of the variations in magnitude and direction of the velocity, due to inhomogeneities of the tortuous pore structure. Both variations on particle size scale and variations due to changes in the permeability from one part of the porous medium to the next, play a role. At high pore Reynolds numbers, mixing due to turbulence also contributes to the mechanical dispersion.

The mechanical dispersion (essentially a second rank symmetric tensor) has been analyzed by various investigators, see for packed- and fluidized beds respectively BEAR (1972) and RIETEMA (1982). In general terms, the mechanical dispersion depends on the velocities and the basic medium properties and can be written as:

\[
D_M = a_1 \bar{U} + a_2 \bar{U}^2 \quad (A2-12)
\]

The porous medium properties \( a_1 \) and \( a_2 \) are respectively called the geometric- and dynamic dispersivity and are both fourth rank tensors.

In a packed bed, the longitudinal- and transverse mechanical dispersion on particle size scale amount to, SAPPMAN (1959):

\[
D_L = 0.76 \, U \, d_p \quad (A2-13)
\]

and:

\[
D_T = 0.20 \, U \, d_p \quad (A2-14)
\]

As stated earlier, mechanical dispersion is the result of the spreading of the fluid, flowing through the interstices between the particles. Because of this, the mechanical dispersion on particle size scale in an expanded bed can be obtained by correcting for the decreased number of pores per unit area. This number of pores per unit area \( N \), amounts to, see KOK (1973):

\[
N = \frac{2 (1-\varepsilon)^2}{\varepsilon} \quad 1.5 \quad (A2-15)
\]

So the longitudinal- and transversal mechanical dispersion on particle size scale of a turbulent fluidized bed become respectively:

\[
D_L = 0.76 \, U \, d_p \, N/N_{mf} \quad (A2-16)
\]

and:

\[
D_T = 0.20 \, U \, d_p \, N/N_{mf} \quad (A2-17)
\]

\( N_{mf} \) is the number of pores per unit area at minimum fluidization condition, obtained from (A2-15) with \( \varepsilon_{mf} = T_{mf} = 0.4 \).
APPENDIX A3

Properties of bed material.

In this appendix the properties of the bed material used in our experiments and its minimal fluidization velocity at ambient conditions are summarized.

The sieve analysis of the silica sand bed material is given in table A3.1.

<table>
<thead>
<tr>
<th>size range [μm]</th>
<th>weight fraction</th>
</tr>
</thead>
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<tr>
<td>&gt; 1000</td>
<td>0.0038</td>
</tr>
<tr>
<td>1000 - 850</td>
<td>0.0929</td>
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<td>0.6876</td>
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<td>0.1159</td>
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<td>0.0007</td>
</tr>
<tr>
<td>&lt; 180</td>
<td>0.0000</td>
</tr>
</tbody>
</table>

TABLE A3-1
Sieve analysis of the silica sand bed material.

The mass mean particle size of the mixture is 780 μm.

The specific density of the bed material, as determined with a pycnometer, is 2630 kg/m³.

After experiments of VERBRAAK (1982), the minimal superficial (air) velocity of the material is 0.36 m/s at ambient conditions and the bed porosity after turning down the air supply amounts to 0.40.

Applying Ergun's equation (A2-4) yields a minimal superficial fluidization velocity of 0.39 m/s under said conditions.

Verbraak's measurements were taken in a bed with a diameter of 0.4 m and a static height of 0.6 m.
**APPENDIX A4**

**Distributor.**

A4.1 **Introduction.**

In this appendix some details on the construction and performance of the distributor are presented. The discussion is limited to the two aspects of distributor design that might affect the reliability of measurements:

- the uniformity of fluidization, and
- the penetration of gas jets into the bed.

During the research program, both a perforated plate and a nozzle distributor have been applied. As a consequence both distributors with vertical- and horizontal orifices will be discussed. At the end of the discussion, concerning the uniformity of fluidization, also the pressure drop of a nozzle distributor is scrutinized.

A4.2 **Uniformity of fluidization.**

For both vertical- and horizontal orifices, the ratio of distributor and bed pressure drop is generally accepted as the criterion for uniformity of fluidization. Out of the various equations proposed, we here use the equation recently put forward by QURESHI (1979) for a perforated plate:

$$\left(\frac{\Delta P_d}{\Delta P_b}\right)_{\text{min}} \geq 0.01 + 0.2 \left(1 - \exp \left(-\frac{D_b}{H}\right)\right)$$  \hspace{1cm} (A4-1)

For given bed material and diameter and height of the bed, this equation yields the minimum acceptable distributor pressure drop.

The distributor pressure drop reads:

$$\Delta P_d = \frac{1}{2} \rho_{or} U_{or}^2 = C_d \frac{M^2}{D_b}$$  \hspace{1cm} (A4-2)

The open area ratio \( \phi \) is equal to:

$$\phi = N \frac{d_{or}^2}{D_b^2}$$  \hspace{1cm} (A4-3)

\( N \) is the number of orifices. For the pressure drop coefficient \( C_d \), QURESHI (1979) gives the empirical expression (for a perforated plate):
\[ C_d = 1.04 \left( \frac{d_{or}}{t} \right)^{0.25} \]  
(A4-4)

\( t \) is the thickness of the distributor plate. In this thesis, it is assumed that equations (A4-1) and (A4-4) also apply to distributors with nozzles, which have horizontal orifices. During our experiments, equation (A4-1) was always satisfied, even during start-up of the PFBC testrig. The just mentioned assumption, concerning the pressure drop coefficient is positively confirmed by experiments of VERBRAAK (1980) at ambient conditions in a bed with a diameter of 0.4 m and a static height of 0.6 m. During the second part of the research program, the distributor pressure drop was measured at PFBC conditions, see table A4.1. The table shows that experiment and theory are in good agreement with each other.

<table>
<thead>
<tr>
<th>( p )</th>
<th>( T_{or} )</th>
<th>( \Delta P_{d,exp} )</th>
<th>( \Delta P_d ) after Qureshi</th>
</tr>
</thead>
<tbody>
<tr>
<td>MPa</td>
<td>K</td>
<td>kPa</td>
<td>kPa</td>
</tr>
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<td>540</td>
<td>21</td>
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<td>540</td>
<td>27</td>
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<td>0.7</td>
<td>520</td>
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<td>39</td>
</tr>
<tr>
<td>0.8</td>
<td>510</td>
<td>44</td>
<td>43</td>
</tr>
</tbody>
</table>

**TABLE A4-1.**

Experimentally determined pressure drop of the nozzle air distributor and theoretical pressure drop, as calculated after QURESHEY (1979) as a function of operating pressure of the PFBC testrig. Standard operating conditions (see section 5), but for \( p \). Distributor properties: \( d_{or} = 3 \text{ mm}, t = 2.6 \text{ mm} \) and \( \phi = 0.38^\circ \).

A4.3. Jet penetration.

In the context of this thesis jet penetration is of interest with regard to sand blasting of bed internals, to possible channeling in the bed and to possible influencing of local measurement of heat transfer, pressure and porosity. Based on orifice jet momentum dissipation, several correlations are given in literature for the penetration depth \( P \) of vertical- and horizontal jets in fluidized or to fluidize beds. To fluidize beds at ambient conditions, ZENZ (1968) gives the empirical correlations:

\[ P_{vert} / d_{or} = 69.4 \log \left( \frac{U_{or}}{d_{or}} \right) - 97.0 \]  
(A4-5)

and:

\[ P_{hor} / d_{or} = 22.1 \log \left( \frac{U_{or}}{d_{or}} \right) - 34.4 \]  
(A4-6)

The half jet angle \( \phi \) of both types of jets is equal to about \( 6^\circ \). At ambient conditions (A4-5) has been confirmed by VERBRAAK (1980). Up till now no orifice jet penetration depth correlations are available, which have been validated at elevated pressures or which take the temperature of the entrance zone into account. For want of something better, it is assumed in this thesis that both equations are also suitable at PFBC conditions.

For vertical jets, the maximum penetration depth should be substantially smaller than the static bed height or:

\[ P_{vert,max} < c H_{st} \]  
(A4-7)

According to KUNII (1969), the constant \( c \) is equal to 0.5. But to prevent measurement problems in our experimental situation, the vertical penetration depth at maximum PFBC fluidization conditions has to be smaller than 0.2 m.

To preclude undesired vertical jet interaction, which can cause non-uniform initial fluidization, the pitch \( S \) of the orifices should be larger than:

\[ S > 2 \frac{P_{vert}}{\tan \phi} \]  
(A4-8)

For maximum PFBC fluidization conditions, \( S \) becomes about 0.04 m, which is equal to the rectangular pitch of the perforated plate air distributor. During the second part of the research program, a nozzle air distributor with horizontal orifices has been used. To prevent nozzle-nozzle jet interaction, which can result in non-uniform fluidization, the distance between the nozzles should be \( > 2 \frac{P_{hor}}{\phi} \). At maximum PFBC fluidization conditions, \( P_{hor} = 6.7 \text{ cm} \) and the just mentioned condition is satisfied. However, to prevent sand blasting of the wall, some of the orifices in the outer nozzles, see figure 4-3, have been omitted. The angle between nearly orifices was made \( 40^\circ \), so as to exclude interaction of the jets.
The height of the orifice above the plate on which the nozzles are mounted, was taken 4 cm, to prevent sand blasting of the plate, and any warpage due to thermal stresses.

APPENDIX A5

Calibration of the high temperature impedance probe.

The theoretical time constant and the calibration of the signal in terms of porosity of the high temperature impedance probe are discussed in this appendix. For detailed information about the probe, see BOELENS (1985). The theoretical time constant can be obtained from the transfer function of the measurement system. In its general form, the transfer function \( H \) reads:

\[
H = \frac{\Delta Y}{Y}
\]  

(5-1)

\( \Delta Y \) and \( Y \) being respectively the admittance change between the needle tips of the probe (see section 4.4.4.2) and the parasitical capacitance of the probe coax cables.

Working out this function, one obtains:

\[
H = \frac{(\Delta G - j\Delta G/\omega)}{C}
\]  

(5-2)

The capacity and conductance change are equal to respectively 0.34 pF and 1.5 \( \mu F \), see section 4.4.4.2.

The theoretical time constant \( \tau \) is equal to:

\[
\tau = \frac{\Delta G}{\omega^2 \Delta C}
\]  

(5-3)

For the values just mentioned and a driving frequency of 100 kHz, \( \tau \) amounts to 11 \( \mu s \). This is sufficiently small for undistorted measurements.

Because the relative permittivity of silica sand is not a function of temperature (BOELENS (1985)), the transfer function can be calibrated in terms of only porosity. The calibration was executed in a silica sand fluidized bed at atmospheric conditions. Taking the porosity in the freeboard as the zero level and applying the logarithmic mixing rule, for the overall relative permittivity of an agglomerate of particles, one obtains for the average capacity change between bed and freeboard:

\[
\Delta C = \text{constant} \left( \epsilon_{rs}^{1/2} - 1 \right)
\]  

(5-4)

The relative permittivity \( \epsilon_{rs} \) of the silica sand bed material is 3.9.
Averaging the HTIP signals, the constant and the mean porosity can be determined by means of two step functions, see figure A5.1.

**FIGURE A5.1.**
Dynamical and average component of the HTIP signal and its step response from fluidized bed level to respectively packed bed and freeboard.

Operating conditions:
- \( J = 0.6 \) m/s, \( p = 0.1 \) MPa, \( d_m = 780 \) \( \mu \)m; \( T_b = 1113 \) K and \( D_b = 0.25 \) m.

The step from the mean porosity level to the packed bed is made by cutting off the air mass flow. The step from the mean porosity level to the freeboard porosity is obtained by traversing the probe vertically from the bed into the upper part of the splashing zone. The irregular bed to freeboard step function (figure A5.1) has its origin in the manual vertical traversing.

The packed bed freeboard capacity change of 0.10 pF is in good agreement with the previously mentioned theoretical one of 0.34 pF. Assuming a packed bed porosity of 0.4, the constant in (A5-4) becomes equal to 0.32 pF. The mean porosity in the centre of the fluidized bed is found to be equal to 0.58. Assuming a TZ fraction of 0.8, the corresponding overall porosity of the fluidized bed becomes 0.54 (see section 3.2). This mean overall bed porosity is in good agreement with the value determined simultaneously by means of the bed pressure differential analyser, which yields \( \bar{\epsilon} = 0.54 \pm 0.03 \).

The mean overall bed porosity as calculated after BABU (1978), see equation (2-12) and from the modified Ergun equation (see Appendix 2) are both 0.51 under the same conditions.
APPENDIX A6

Calibration of the Local Dynamical Heat Transfer Probe.

The classical theory of penetration of heat waves into a solid body surface layer permits calculation of the performance of the LDHTP, if the position of the thermocouples, their calibration curves and the way in which they locally disturb the probe geometry and composition are accurately known. The calibration of the thermocouples is known, and based on a certificate (THD-82006), the geometric errors of the thermocouples are negligible, HEESTERBEEK (1982). However, the inner- and outer thermocouples are respectively based in a long drilled hole and embedded in slits in the surface. Due to these local disturbances of the probe, the accuracy of any calculation is only moderate. So additional calibration of especially the outer thermocouples is required. Laboratory calibrations executed by HEESTERBEEK (1983) were not very successful, mainly because of the too low heat fluxes that could be imposed on the probe surface. The calibrations were executed by means of a laboratory black body radiator, operating at a temperature of 1123 K.

The stationary heat transfer measurement was calibrated by positioning the LDHTP in the centre of an oven. Because the emission coefficient of the outer polished surface of the probe is about 0.3, the (constant) absorbed radiant heat flux was about 25 kW/m². This resulted in a temperature difference between outer- and inner thermocouple of only 12 K. This temperature difference is too small for accurate calibration. As a consequence the error of the stationary heat transfer measurement amounts to 6%, HEESTERBEEK (1982).

Assuming a negligible geometric error, then dynamical calibration mainly serves determination of the local composition disturbances, caused by the thermocouple. For this calibration, a rotating disk with a slit was positioned between the laboratory black body radiator and the probe, see figure A6-1. By varying the speed of the disk different, approximately square incident energy waves were generated locally on the probe surface. The Bode diagram derived from the thermocouple response was found to have a band width of about 4 Hz, which is almost an order of magnitude smaller than expected, HEESTERBEEK (1982).

However, due to the too low imposed heat flux and parasitical heat flows in the probe, this calibration is not very accurate (the parasitical heat flows are a consequence of the fact that only one side of the probe was radiated).
FIGURE A6-1.
Scheme of the experimental set up to determine the time constant of the surface thermocouple of the LDHTP. Temperature of the laboratory black body radiator 1123 K. Radial frequency of the rotating disk is variable between 2.26 to 62.87 rad/s.

As a consequence of this poor calibration, the statistical signal properties of the LDHTP could not be corrected in a reliable way. As concluded in Chapter 5, however, the frequency band of both the LDHTP and the HTIP signal is 1 to 5 Hz with a maximum at about 2.5 Hz. So distortion of the LDHTP signals is acceptable.

APPENDIX A7

Heat transfer in the WRL and in the splashing zone/freeboard.

Information about the heat transfer in the WRL and the wall region (WR) of the splashing zone/freeboard of a PFBC is very scarce. Therefore the local mean heat transfer coefficients in both regions, as determined by the vertical U-tube heat transfer measurements (see figure 4-6) are given in table A7-1, in spite of their possibly large measurement error of 40%.

<table>
<thead>
<tr>
<th>J (m/s)</th>
<th>p (MPa)</th>
<th>$h_{WRL}$ (W/m² K)</th>
<th>$h_{WR}$ (W/m² K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5</td>
<td>0.5</td>
<td>110</td>
<td>90</td>
</tr>
<tr>
<td>0.6</td>
<td>0.6</td>
<td>110</td>
<td>100</td>
</tr>
<tr>
<td>0.7</td>
<td>1.10</td>
<td>110</td>
<td>110</td>
</tr>
<tr>
<td>0.6</td>
<td>0.5</td>
<td>100</td>
<td>90</td>
</tr>
<tr>
<td>1.1</td>
<td>0.8</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>0.8</td>
<td>0.9</td>
<td>100</td>
<td>120</td>
</tr>
<tr>
<td>0.5</td>
<td>1.0</td>
<td>90</td>
<td>90</td>
</tr>
<tr>
<td>1.7</td>
<td>0.6</td>
<td>110</td>
<td>90</td>
</tr>
<tr>
<td>0.8</td>
<td>0.8</td>
<td>80</td>
<td>90</td>
</tr>
<tr>
<td>0.5</td>
<td>1.8</td>
<td>130</td>
<td>80</td>
</tr>
<tr>
<td>0.6</td>
<td>0.6</td>
<td>90</td>
<td>90</td>
</tr>
</tbody>
</table>

TABLE A7-1.

Measured heat transfer coefficient in the WRL and the WR of the splashing zone/freeboard of the PFBC, at various superficial fluidization velocities and operating pressures.

Operating conditions $T_b = 1100$ K, $d_p = 780$ μm and $H_{mf} = 0.4$ m.

The heat transfer coefficient in the WRL of the bed amounts to about 100 W/m² K, which indicates that this zone is very quiet and most probably only just minimally fluidized.

The results of the measurements in the WR of the splashing zone/freeboard of the testrig are presented without further comment. Due to thermocouple failure, the splashing zone/freeboard heat transfer in the central part of the testrig could not be determined.
APPENDIX A8

The radiative component of turbulent fluidized bed heat transfer.

As indicated in section 3.4.3.4, the trends of the radiative contribution to the turbulent fluidized bed heat transfer are somewhat confusing. However, starting from the two radiative terms of the energy equation (3-45), this trends can be explained.

The contribution of the radiative bed wall heat sink $R_{bw}$ is independent of operating conditions (apart from temperature). The reason for this is that the steepness of the temperature profile changes only marginally as a function of operating conditions, see section 3.4.3.3.

To illustrate the influence of the layer to layer radiative heat sink we simplify energy equation (3-45) to:

$$\rho_f \frac{C_p}{T^3} \frac{dT}{dT} = \Phi_R \quad (A8-1)$$

After substituting equations (3-48) and (3-49) this expression becomes, in dimensionless form:

$$\frac{2\beta}{\alpha} = \frac{2\theta}{\alpha} \quad (A8-2)$$

with:

$$\beta = 4\epsilon \frac{C_p}{T^3} \frac{1}{\rho_f c_{pf}} \frac{1}{l_e} \quad (A8-3)$$

$l_e$ is the thickness of the temperature profile near the wall.

The coefficient $\beta$, which can be interpreted as "radiative conduction", increases slightly with increasing superficial fluidization velocity. And it decreases respectively significantly and slightly for both increasing operating pressure and particle size. So the trends of the "radiative conduction" give an explanation for the nature of the radiative component of the turbulent fluidized bed heat transfer.
APPENDIX A9

Shear forces in a turbulent fluidized bed.

The solid-solid shear forces in a turbulent fluidized bed are briefly discussed in this appendix. Assuming vertical shear forces to occur between TZ and WRL, as well as between WRL and wall, the solid shear stress profile and the solid velocity profile of a fluidized bed are derived for the turbulent fluidization model as presented in Chapter 3. The pressure gradient of the bed as a whole is the starting point for this derivation. However, comparing these shear stresses with the scarce information known from literature leads to poor agreement. The general trends of solid shear stress and solid velocity profile in a turbulent fluidized bed are depicted in figure A9-1.

FIGURE A9-1.
General trends of solid shear stress profile (a) and solid velocity profile (b) in a turbulent fluidized bed.

Assuming the solid velocities in TZ and WRL to be uniform, and the shear stresses to be concentrated in the contact surface between TZ and WRL and near the wall, figure A9-1 is reduced to the simplified picture sketched in figure A9-2. This schematic picture permits an estimate to be made of the velocities and shear stresses in the turbulent fluidized bed.

Combining (A2-9) and (3-3) yields the pressure gradient of the bed as a whole:

\[
\frac{dp}{dz} = \left[F \left(1-\varepsilon_f\right) + (1-F) \left(1-\varepsilon_w\right)\right] \left(\rho_s - \rho_f\right) g \tag{A9-1}\]

A9.1
FIGURE A9-2.
Simplified picture of solid shear stress and solid velocity in a turbulent fluidized bed.

This pressure gradient also applies to TZ and WRL. The RHS of (A9-1) may be rearranged in two ways:

\[
\frac{dp}{dz} = [(1-\epsilon_T) (\rho_s - \rho_f) g + (1-F) (\epsilon_T - \epsilon_W) (\rho_s - \rho_f) g \ (A9-2)
\]

and:

\[
\frac{dp}{dz} = [(1-\epsilon_W) (\rho_s - \rho_f) g + F (\epsilon_T - \epsilon_W) (\rho_s - \rho_f) g \ (A9-3)
\]

(A9-2) permits a global analysis of the vertical force equilibrium in the TZ. The first term in the RHS compensates the (net) gravity force on the solids. Consequently some other downward force(s) has (have) to account for the second term. Similarly it appears from (A9-3) that in the WRL some upward force(s) is (are) required to compensate for the second term in the RHS. Because of the direction of these forces, it may be conjectured that they originate from the shear stress, due to momentum exchange, between rising solids in the TZ and descending solids in the WRL and from the friction between the latter solids and the wall.

From (A9-2) it is easily calculated that the shear stress required to restore vertical force equilibrium for the TZ amounts to:

\[
\tau_{TW} = (1-F) (\epsilon_T - \epsilon_W) (\rho_s - \rho_f) g F^{0.5} D_b/a \ (A9-4)
\]

From (A9-3) it follows, on the same grounds, that:

\[
\tau_{TW} = \tau_{TW} F^{0.5} = F (\epsilon_T - \epsilon_W) (\rho_s - \rho_f) g (1-F) D_b/a \ (A9-5)
\]

According to STAUB (1979):

\[
\tau_{TW} = \omega (V_T - V_W) \ (A9-6)
\]

\(\omega\) is the lateral solids mass velocity between the TZ and WRL in one direction. Neglecting the TZ-WRL gas shear stress, such in agreement with Chen, see CHEREMISINOFF (1983), Staub finally obtains:

\[
(1-\epsilon)^2 F^{0.5} D_b (1-\epsilon_T) \rho_s V_T^2
\]

Substituting standard conditions, section 3.1, and turbulent fluidized bed model properties, section 3.2, it is easily shown, that the shear stresses (A9-4) and (A9-7) differ about one order of magnitude. To the author's opinion, several effects playing a role in the equilibrium of forces, e.g. acceleration in the TZ as well as the WRL, Staub's not verified TZ-WRL solid mass exchange etc. may cause these unsatisfying results. The author feels that momentum exchange between TZ and WRL and friction between WRL and wall of the bed deserves high priority in future research.

As indicated by (A9-6) momentum exchange and friction are closely related to the velocities of solids in the TZ and WRL. Consequently it may govern the circulation rate, residence time, etc. of the particles in the fluidized bed.
Stellingen bij het proefschrift

TURBULENT FLUIDIZATION AND HEAT TRANSFER
IN A PRESSURIZED FLUIDIZED BED COMBUSTOR

door
G. Boelens

1. Hoewel de fluidisatie in een fluid bed drukvuurhaard heterogeen is, vertoont zij gezien de amplitude-versterkingsfactor en de golflengte van de porositeitsfluctuaties een homogeen karakter.
   Dit proefschrift, sectie 3.3.

2. De veronderstelling van Kool betreffende de maximale in de THD-1 wervelbed drukvuurhaard voorkomende beldiameter is niet gerechtvaardigd en leidt tot een overwaardering van het "fast bubble" model.
   Dynamic modeling and identification of a coal fired pressurized fluidized bed combustor.
   Dissertation, Delft University of Technology.
   b) Dit proefschrift, sectie 6.2.

3. Tussen de warmtetransporten door geleiding, conveetie en straling treden in een wervelbed belangrijke wisselwerkingen op. Het is daarom niet zinvol om, zoals vele onderzoekers doen, voor de warmteoverdracht te schrijven
   \[ h = h_p + h_v + h_g \]
   Deze optelling mag alleen in differentiaalvorm worden gebruikt.
   Dit proefschrift, secties 2.3 en 3.4.

4. Martin veronderstelt ten onrechte, dat de bellenfasevolumefractie in zijn wervelbedwarmteoverdrachtetheorie buiten beschouwing is gelaten.
   Dit proefschrift, Appendix A1.

5. Dispersieve geleiding van warmte draagt essentieel bij tot de warmteoverdracht in een wervelbeddrukvuurhaard.
   Dit proefschrift, sectie 3.4.

6. De experimenteel bepaalde dynamische component van de warmteoverdrachtscoëfficiënt in een wervelbeddrukvuurhaard wordt verklaard door de locale gas (en) vaste stof stromingscondities en niet zoals vele onderzoekers aannemen door de bijdrage van de straling tot de warmteoverdracht.
   Dit proefschrift, secties 3.4 en 2.3.

7. Gezien de enorme vlucht, die de audio-visuele technieken recentelijk hebben genomen, zou de presentatie van een proefschrift in de vorm van een videoband tot de mogelijkheden moeten gaan behoren.


9. De RSV kolengraver-affaire is een klassiek voorbeeld geworden van het mislukken van een branche-vreemde activiteit.