Some pages of this thesis may have been removed for copyright restrictions.

If you have discovered material in AURA which is unlawful e.g. breaches copyright, (either yours or that of a third party) or any other law, including but not limited to those relating to patent, trademark, confidentiality, data protection, obscenity, defamation, libel, then please read our Takedown Policy and contact the service immediately.
ADVANCES TOWARDS A PRESSURISED
ROTATING FLUIDISED BED COMBUSTOR

by

GARETH WILLIAM OSKAM

A thesis submitted for the
DEGREE of DOCTOR OF PHILOSOPHY

Department of Mechanical Engineering
The University of Aston in Birmingham
April 1983
ADVANCES TOWARDS A PRESSURISED ROTATING FLUIDISED BED COMBUSTOR

by

Gareth William Oskam

A THESIS SUBMITTED FOR THE DEGREE OF DOCTOR OF PHILOSOPHY

APRIL 1983

SUMMARY

Rotating Fluidised Beds offer the potential for high intensity combustion, large turndown and extended range of fluidising velocity due to the imposition of an artificial gravitational field. Low thermal capacity should also allow rapid response to load changes. This thesis describes investigations of the validity of these potential virtues.

Experiments, at atmospheric pressure, were conducted in flow visualisation rigs and a combustor designed to accommodate a distributor 200mm diameter and 80mm axial length. Ancillary experiments were conducted in a 6” diameter conventional fluidised bed.

The investigations encompassed assessment of; fluidisation and elutriation, coal feed requirements, start-up and steady-state combustion using premixed propane and air, transition from propane to coal combustion and mechanical design.

Assessments were made of an elutriation model and some effects of particle size on the combustion of premixed fuel gas and air.

The findings were:-

a) More reliable start-up and control methods must be developed. Combustion of premixed propane and air led to severe mechanical and operating problems. Manual control of coal combustion was inadequate.

b) Design criteria must encompass pressure loss, mechanical strength and high temperature resistance. The flow characteristics of ancillaries and the distributor must be matched.

c) Fluidisation of a range of particle sizes was investigated. New correlations for minimum fluidisation and fully supported velocities are proposed. Some effects on elutriation of particle size and the distance between the bed surface and exhaust port have been identified. A conic distributor did not aid initial bed distribution. Furthermore, airflow instability was encountered with this distributor shape. Future use of conic distributors is not recommended. Axial solids mixing was found to be poor. A coal feeder was developed which produced uniform fuel distribution throughout the bed.

The report concludes that small scale inhibits development of mechanical design and exploration of performance. Future research requires larger combustors and automatic control.

ROTATING FLUIDISATION ELUTRIATION COMBUSTION DESIGN
ACKNOWLEDGEMENT

The author would like to take this opportunity to thank all the individuals and organisations who have contributed to the project. In particular, sincere thanks go to:-

Dr J.R. Howard, Project Supervisor, for his constant enthusiasm and intelligent guidance;

D. Green and M. Cox, Mechanical Engineering Department Workshop staff, for their skillful efforts in the manufacture of the experimental combustor and other models;

P. Hickman and G. Hackett, Mechanical Engineering Department laboratory technical staff, for their patience, good humour and skill in the construction, maintenance and operation of the experimental rigs; and

M. Kitson, trainee technician, for constructing the electronic circuits.

The author would also like to express appreciation to the staff of the Mechanical Engineering department, University of Aston; to Mr G. Polonski, Mr J. Cavannah and Mr M. Subzwari, Mechanical Engineering Laboratories, G.E.C. Power Engineering; and to Prof. J Switchenbank, University of Sheffield, for many fruitful discussions.

The author also thanks Mr Subzwari for his computer program for particle trajectories, from which the author's own computer program was developed.

Grateful thanks go to Cullum Detuners Ltd., Derby, and Wellman-Selas Ltd., Manchester, for supplying, free of charge, consultation, designs and equipment used in this project.

The author also wishes to thank the S.E.R.C.; the Mechanical Engineering Laboratories, G.E.C. Power Engineering; Stone-Platt Fluidfire; and the University of Aston in Birmingham for their financial support of this project.

Thanks also go to Kaldair Ltd. for the use of their word processor on which this thesis was prepared and to Nicki Cave, Barbara Bowden and Marcella Stubbs for typing the manuscript.

Finally, I wish to thank my friends and family, especially Julie, my wife, for their constant support during the project.
CHAPTER 1
INTRODUCTION

1.1 Background 1
1.2 Brief History of Fluidisation Technology 5
1.3 Required Development Steps 9
1.4 Final Choice of Programme 10
Figures 11-13

CHAPTER 2
INDUSTRIAL APPLICATIONS

2.1 Fluidised Bed Combustors and Boilers 14
2.1.1 General 14
2.1.2 Control via particle size, density and material type 19
2.1.3 Multi Zone Units with sequentially positioned zones 19
2.1.4 Multi Zone Units with independently operated zones 20
2.1.5 Variation of immersed heat transfer surfaces 20
2.1.6 Pressurised fluidised bed combustion 21
2.1.7 Variation of effective bed weight 22
2.1.8 Summary of section 2.1 24
2.2 Power Cycles 24
2.2.1 Conventional combined cycle plant 25
2.2.2 Fluidised bed power cycles 27
2.2.3 Summary of section 2.2 29
CHAPTER 3
LITERATURE SURVEY

3.1 Introduction 37
3.1.1 Previous Reviews of RFB Literature 37
3.2 Choice of Particle Size 39
3.2.1 Particle Size Choice for the RFB 39
3.3 Minimum Fluidising Velocity Umf 41
3.3.1 Umf in Stationary Fluidised Beds 41
3.3.2 Prediction of Umf in the RFB 45
3.3.3 Effect of Gas Temperature on Umf 49
3.3.4 The Effect of Pressure On Umf 50
3.4 Bubbling and Mixing 51
3.4.1 Bubbling and Mixing in Stationary Fluidised Beds 53
3.4.2 Bubbling and Mixing in the RFB 54
3.5 Elutriation 57
3.5.1 Elutriation from Stationary Fluidised Beds 58
3.5.2 Elutriation from the RFB 60
3.6 Heat Transfer 62
3.6.1 Gas to Particle Heat Transfer 62
3.6.2 Heat Transfer to the Walls and Immersed Surfaces 64
3.6.3 Heat Transfer in the RFB 65
3.7 Combustion 66
3.7.1 Combustion of Gaseous Fuels 67
3.7.1.1 Gas Combustion in Stationary Fluidised Beds 67
<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.7.1.2 Gas Combustion in the RFB</td>
<td>68</td>
</tr>
<tr>
<td>3.7.2.1 Combustion of Liquid Fuels in Stationary Beds</td>
<td>70</td>
</tr>
<tr>
<td>3.7.2.2 Combustion of Liquid Fuels in the RFB</td>
<td>71</td>
</tr>
<tr>
<td>3.7.3 Combustion of Solid Fuels in Stationary Fluidised Beds</td>
<td>71</td>
</tr>
<tr>
<td>3.7.4 Combustion of Solid Fuels in the RFB</td>
<td>75</td>
</tr>
<tr>
<td>3.8 Emissions from Fluidised Bed Combustors</td>
<td>78</td>
</tr>
<tr>
<td>3.8.1 Emissions of nitrogen and sulphur oxides from Stationary Fluidised Bed Combustors</td>
<td>78</td>
</tr>
<tr>
<td>3.8.2 Emissions of Particulates from Stationary Fluidised Beds</td>
<td>82</td>
</tr>
<tr>
<td>3.8.3 Emissions from Rotating Fluidised Bed Combustors</td>
<td>83</td>
</tr>
<tr>
<td>3.9 Mechanical Design Considerations</td>
<td>85</td>
</tr>
<tr>
<td>3.9.1 Distributor design in the RFB</td>
<td>85</td>
</tr>
<tr>
<td>3.9.2 Heat transfer tubes</td>
<td>86</td>
</tr>
<tr>
<td>3.10 Principal conclusions</td>
<td>87</td>
</tr>
<tr>
<td>3.10.1 Fluidisation characteristics</td>
<td>87</td>
</tr>
<tr>
<td>3.10.2 Mixing characteristics</td>
<td>88</td>
</tr>
<tr>
<td>3.10.3 Elutriation</td>
<td>88</td>
</tr>
<tr>
<td>3.10.4 Heat transfer</td>
<td>88</td>
</tr>
<tr>
<td>3.10.5 Combustion of solid fuels</td>
<td>89</td>
</tr>
<tr>
<td>3.10.6 Emissions Figures</td>
<td>89</td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>90-93</td>
</tr>
</tbody>
</table>
CHAPTER 4

COMBUSTOR DESIGN

4.1 Introduction 94
4.2 Plenum Chamber 96
4.3 Distributor 97
4.4 Distributor Upper End Plate 101
4.5 Exhaust Port Chimney 102
4.6 Coal injector 105
4.7 Combustor Taper Rings 105
Figures 108-114

CHAPTER 5

RIG FACILITY

5.1 Introduction 115
5.2 Gas Analysis Requirements 117
5.2.1 General Requirements 117
5.2.2 Gas Sampling Probe Design 118
5.2.3 Probe Cooling System 119
5.2.4 Sample Transfer Line 119
5.2.5 Sampling Station and Frequency 120
5.3 Exhaust Gas Cleaning 121
5.4 Pressure Let Down and Exhaust Silencing 121
5.4.1 Pressure Let Down 121
5.4.2 Exhaust Silencing 122
5.5 Solids Handling System 124
5.5.1 Metering and Transfer Methods 125
CHAPTER 6

FLUIDISATION AND RELATED EXPERIMENTS CONDUCTED AT ATMOSPHERIC TEMPERATURE AND PRESSURE

6.1 Introduction 138
6.2 Apparatus 139
6.3 Experimental Procedure 142
6.3.1 Leakage and Distributor Flow Characteristics 142
6.3.2 General Start-up Procedure 142
6.3.3 Pressure Loss and Fluidisation Quality 143
6.3.4 Coal Particle Mixing in Silica Sand Beds 144
6.3.5 Distributor Evaluation 145
6.3.6 Exhaust Port Diameter 145
6.4 Results and Discussion 146
6.4.1 Bed Spreading During Start-up 146
6.4.2 Fluidisation Characteristics – Pressure Loss 147
6.4.2.1 Presentation of Data 147
6.4.2.2 Fluidisation Results 149
6.4.2.3 Discussion of Fluidisation Results 150
6.4.3 Minimum Operating Condition 153
6.4.3.1 Alternative Definitions 153
6.4.3.2 Prediction of Minimum Operating Condition 154
6.4.3.3 Proposal for New Correlations 155
6.4.3.4 Effect of changing the Exhaust Port Diameter on Minimum Operating Condition 156
6.4.4 Aspects of Elutriation 156
6.4.5 Bed Instabilities 161
6.4.5.1 Suggested Mechanism of Pulsation 163
6.5 Particle Mixing 164
6.5.1 Particle Size and Fluidising Velocity Criteria 165
6.5.1.1 Effects of Particle Size Ranges and Fluidising Velocity 166
6.5.1.2 Effect of Non-Homogeneous Initial Mixture 166
6.5.2 Coal Particle Feeder Development 168
6.5.2.1 Spreader Type Feeder 168
6.5.2.2 Pneumatic Feeder 170
6.5.3 Combustor Fuel Feeder 172
6.6 Elutriated Particle Capture 173
6.7 Summary 174
   Tables 176-183
   Figures 184-221

CHAPTER 7
EXPERIMENTS AT ELEVATED TEMPERATURE

7.1 Introduction 222
7.1.1 Test Objectives 223
7.1.2 Overview of Combustion Trials 223
7.2 Initial Rig Set-Up 225
7.3 Initial Combustion Trials 227
7.3.1 Scope 227
<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.3.2 Fluidisation Experiments</td>
<td>229</td>
</tr>
<tr>
<td>7.3.3 Air-Cooled Labyrinth Seal Development</td>
<td>229</td>
</tr>
<tr>
<td>7.3.4 Start-Up Trials</td>
<td>232</td>
</tr>
<tr>
<td>7.3.5 Coal Combustion</td>
<td>235</td>
</tr>
<tr>
<td>7.3.5.1 Bed Fusion</td>
<td>237</td>
</tr>
<tr>
<td>7.3.5.2 Distributor Mechanical Integrity</td>
<td>237</td>
</tr>
<tr>
<td>7.4 Development of the Stand Pipe Distributor</td>
<td>238</td>
</tr>
<tr>
<td>7.4.1 Mechanical Design</td>
<td>238</td>
</tr>
<tr>
<td>7.4.2 Combustion of Gas and Coal</td>
<td>239</td>
</tr>
<tr>
<td>7.4.3 RFB Stand-Pipe Distributor Tests</td>
<td>242</td>
</tr>
<tr>
<td>7.4.3.1 Fluidisation</td>
<td>242</td>
</tr>
<tr>
<td>7.4.3.2 Combustion Trials</td>
<td>243</td>
</tr>
<tr>
<td>7.5 Porous Distributor</td>
<td>245</td>
</tr>
<tr>
<td>7.5.1 Mechanical Design</td>
<td>245</td>
</tr>
<tr>
<td>7.5.2 Fluidisation Experiment</td>
<td>246</td>
</tr>
<tr>
<td>7.5.3 Combustion Experiments</td>
<td>247</td>
</tr>
<tr>
<td>7.5.3.1 Gas Combustion</td>
<td>247</td>
</tr>
<tr>
<td>7.5.3.2 Transient Response</td>
<td>250</td>
</tr>
<tr>
<td>7.5.3.3 Coal Combustion</td>
<td>251</td>
</tr>
<tr>
<td>7.6 Summary of Findings</td>
<td>252</td>
</tr>
<tr>
<td>7.6.1 Mechanical Design Aspects</td>
<td>252</td>
</tr>
<tr>
<td>7.6.2 Combustion Aspects</td>
<td>254</td>
</tr>
<tr>
<td>Tables</td>
<td>257-264</td>
</tr>
<tr>
<td>Figures</td>
<td>265-291</td>
</tr>
</tbody>
</table>
CHAPTER 8
THEORETICAL STUDIES OF OBSERVED PHENOMENA

8.1 Introduction 292
8.1.1 Particle Dynamics 292
8.1.2 Flame Stability and Fuel Gas Bypass 293
8.2 Particle Dynamics in the Combustor Freeboard 294
8.2.1 Summary of section 8.2 302
8.3 Fluidising Velocity/Flame Speed Matching 302
8.4 Fuel-Gas Bypass in Bubbles 304
8.4.1 Prediction of Air:Fuel Ratio 309
8.4.2 Summary of section 8.4 312
   Tables 313-314
   Figures 315-357

CHAPTER 9
CONCLUSIONS AND RECOMMENDATIONS

9.1 Review of Results 358
9.1.1 Combustor Design 358
9.1.1.1 Distributor Design 358
9.1.1.2 Exhaust Port Seal 359
9.1.1.3 Exhaust Port Diameter 360
9.1.1.4 Solids Feeding 360
9.1.2 Fluid Dynamics 361
9.1.2.1 Operating Regime - Descriptive Parameters 361
9.1.2.2 Initial Run-up 361
9.1.2.3 Minimum Fluidisation and Fully Supported Fluidisation 362
9.1.2.4 Elutriation 362
9.1.2.5 Instabilities 363
9.1.2.6 Solids Mixing 364
9.1.3 Combustion 365
9.1.3.1 Start-up 365
9.1.3.2 Gas Combustion 366
9.1.3.3 Coal Combustion 368
9.1.4 Important Findings 369
9.2 Conclusions 371
9.3 Recommendations 374
9.3.1 Combustion Applications 374
9.3.2 Other Applications 375

APPENDIX A
HEAT TRANSFER MODELS FOR AIR COOLED CHIMNEY AND AIR COOLED DISTRIBUTORS

A.1 Air-Cooled Chimney 376
A.1.1 Background 376
A.1.2 Heat Transfer Model 377
A.2 Air-Cooled Distributors 380
A.2.1 Background 380
A.2.2 Heat Transfer Model 381

APPENDIX B
RFBC ENGINEERING DRAWINGS 383-391
APPENDIX C

EQUIPMENT SPECIFICATIONS

Headings Listing

C.1. H.P. Compressor 393
C.2. Gas Analysers 394
C.2.1 Oxygen 394
C.2.2 Oxides of Carbon: CO and CO2 397
C.2.3 Unburned Hydrocarbons UHC 401
C.2.4 Oxides of Nitrogen NO and NO2 (NOx) 404
C.2.5 Gas Sample Transfer Line 407
C.2.6 Oxides of Sulphur SO2 and SO3 408
C.2.7 Particulates 409
C.2.8 Other Constituents 411
C.2.9 New Techniques 412
References 414

APPENDIX D

COMBUSTION RIG REQUIREMENTS

D.1 Gas Analysis System Requirements 416
D.2 Exhaust Gas Cleaning 420
D.3 Scrubbing Water Supply and Treatment Plant 422
D.4 Exhaust Silencing 423
D.5 Solids Feed System - Valve Selection and Sequence Control 426
D.5.1 Selected Coal Feeder 427
D.5.2 Airlock Valve Sequence Control 428
# NOMENCLATURE

**CHAPTER 3**

<table>
<thead>
<tr>
<th>SYMBOL</th>
<th>DESCRIPTION</th>
<th>UNITS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cd</td>
<td>drag coefficient</td>
<td></td>
</tr>
<tr>
<td>Cp</td>
<td>specific heat of fluidising gas at constant pressure</td>
<td>kJ/kgK</td>
</tr>
<tr>
<td>D</td>
<td>distributor diameter</td>
<td>mm</td>
</tr>
<tr>
<td>Db</td>
<td>bubble diameter</td>
<td>m</td>
</tr>
<tr>
<td>E</td>
<td>bed voidage</td>
<td></td>
</tr>
<tr>
<td>E'g/υg</td>
<td>elutriation rate constant</td>
<td>kg/m²s</td>
</tr>
<tr>
<td>Emf</td>
<td>bed voidage at Umf</td>
<td></td>
</tr>
<tr>
<td>G</td>
<td>mass velocity</td>
<td>kg/m²/s</td>
</tr>
<tr>
<td>Ga</td>
<td>[ \frac{\int (\rho_0-\rho) , dp , \rho_d^3 , d \rho}{\mu_g^2} ]</td>
<td>Galileo number</td>
</tr>
<tr>
<td>Ga'</td>
<td>[ \frac{\int (\rho_0-\rho) , dp , \rho_d^3 , d \rho}{\mu_g^2} ]</td>
<td>modified Galileo number</td>
</tr>
<tr>
<td>H</td>
<td>bed depth</td>
<td>m</td>
</tr>
<tr>
<td>Hmf</td>
<td>bed depth at Umf</td>
<td>m</td>
</tr>
<tr>
<td>Nu</td>
<td>[ \frac{h_{dp}}{\rho g} ]</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>φ</td>
<td>particle sphericity</td>
<td></td>
</tr>
<tr>
<td>ΔP_b</td>
<td>bed pressure loss</td>
<td>N/m²</td>
</tr>
<tr>
<td>P</td>
<td>pressure</td>
<td>N/m²</td>
</tr>
<tr>
<td>Pr</td>
<td>[ \frac{c_p \mu_b}{\rho_d} ]</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>Re_{fs}</td>
<td>particle Reynolds number at Ufs</td>
<td></td>
</tr>
<tr>
<td>Re_{mf}</td>
<td>particle Reynolds number at Umf</td>
<td></td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Unit</td>
</tr>
<tr>
<td>--------</td>
<td>------------------------------------------------------------------------------</td>
<td>--------------------</td>
</tr>
<tr>
<td>T</td>
<td>temperature</td>
<td>°C</td>
</tr>
<tr>
<td>U</td>
<td>linear velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>Ub</td>
<td>bubble rise velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>U_{ff}</td>
<td>gas velocity at full fluidisation</td>
<td>m/s</td>
</tr>
<tr>
<td>U_{fs}</td>
<td>fully fluidised velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>U_g, U_o, U_s</td>
<td>superficial gas velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>U_{mf}</td>
<td>minimum fluidising velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>U_r</td>
<td>radial component of velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>U_s</td>
<td>slugging velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>U_t</td>
<td>tangential component of velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>U_T</td>
<td>particle terminal velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>Z</td>
<td>bed depth (McGaw's correlation)</td>
<td>mm</td>
</tr>
<tr>
<td>ar</td>
<td>radial component of acceleration</td>
<td>m/s²</td>
</tr>
<tr>
<td>dp</td>
<td>particle diameter</td>
<td>microns or mm</td>
</tr>
<tr>
<td>dp</td>
<td>mean particle diameter</td>
<td>microns or mm</td>
</tr>
<tr>
<td>g</td>
<td>acceleration due to gravity</td>
<td>m/s²</td>
</tr>
<tr>
<td>h_p</td>
<td>heat transfer coefficient from gas to particle</td>
<td>W/m²K</td>
</tr>
<tr>
<td>k_g</td>
<td>thermal conductivity of fluidising gas</td>
<td>W/mK</td>
</tr>
<tr>
<td>m_{i}</td>
<td>mass of particles in i-th shell</td>
<td>kg</td>
</tr>
<tr>
<td>n</td>
<td>nondimensional multiplier</td>
<td></td>
</tr>
<tr>
<td>p</td>
<td>pitch of distributor holes</td>
<td>mm</td>
</tr>
<tr>
<td>r</td>
<td>radius</td>
<td>m</td>
</tr>
<tr>
<td>r_{i}</td>
<td>radius of &quot;i&quot;th particle shell</td>
<td>m</td>
</tr>
<tr>
<td>r_m</td>
<td>mean radius</td>
<td>m</td>
</tr>
<tr>
<td>w</td>
<td>angular velocity</td>
<td>rad/s</td>
</tr>
<tr>
<td>w_{i}</td>
<td>weight of &quot;i&quot;th fraction</td>
<td>kg</td>
</tr>
<tr>
<td>y</td>
<td>gas kinematic viscosity</td>
<td>m²/s</td>
</tr>
<tr>
<td>s_g</td>
<td>gas density</td>
<td>kg/m³</td>
</tr>
</tbody>
</table>
\( \mu_g \)  & gas viscosity & Ns/m²  \\
\( \rho_p \)  & particle density & kg/m³  \\

**CHAPTER 6**

<table>
<thead>
<tr>
<th>SYMBOL</th>
<th>DESCRIPTION</th>
<th>UNITS</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A_d )</td>
<td>distributor area</td>
<td>m²</td>
</tr>
<tr>
<td>( C_d )</td>
<td>discharge coefficient</td>
<td></td>
</tr>
<tr>
<td>( G_a )</td>
<td>Galileo number (same as Archimedes number)</td>
<td></td>
</tr>
<tr>
<td>( G_a' )</td>
<td>Modified Galileo number</td>
<td></td>
</tr>
<tr>
<td>( H_d )</td>
<td>axial length of distributor</td>
<td>m</td>
</tr>
<tr>
<td>( M_b )</td>
<td>bed mass</td>
<td>kg</td>
</tr>
<tr>
<td>( \Delta P )</td>
<td>pressure loss</td>
<td>N/m²</td>
</tr>
<tr>
<td>( \Delta P_{(d+b)} )</td>
<td>distributor plus bed pressure loss</td>
<td>N/m²</td>
</tr>
<tr>
<td>( \Delta P_d )</td>
<td>distributor pressure loss</td>
<td>N/m²</td>
</tr>
<tr>
<td>( \rho_p )</td>
<td>plenum pressure</td>
<td>N/m²</td>
</tr>
<tr>
<td>( Q_p )</td>
<td>plenum flow function</td>
<td>( \sqrt{\rho K \cdot s/m} )</td>
</tr>
<tr>
<td>( (Q_p)_e )</td>
<td>plenum flow function at onset of elutriation</td>
<td></td>
</tr>
<tr>
<td>( (Q_p)_{fs} )</td>
<td>plenum flow function at full support</td>
<td></td>
</tr>
<tr>
<td>( Re_{fs} )</td>
<td>Reynolds number at ( U_{fs} )</td>
<td></td>
</tr>
<tr>
<td>( Re_{mf} )</td>
<td>Reynolds number at ( U_{mf} )</td>
<td></td>
</tr>
<tr>
<td>( T_p )</td>
<td>plenum gas temperature</td>
<td>°C or °K</td>
</tr>
<tr>
<td>( U_{fs} )</td>
<td>fully supported fluidising velocity ( m/s )</td>
<td></td>
</tr>
<tr>
<td>( U_{mf} )</td>
<td>minimum fluidising velocity ( m/s )</td>
<td></td>
</tr>
<tr>
<td>( d_p )</td>
<td>particle diameter</td>
<td>microns or ( \text{mm} )</td>
</tr>
<tr>
<td>( m_a )</td>
<td>air mass flow rate</td>
<td>kg/s</td>
</tr>
<tr>
<td>( n_g )</td>
<td>'n' times normal gravity</td>
<td>m/s²</td>
</tr>
<tr>
<td>SYMBOL</td>
<td>DESCRIPTION</td>
<td>UNITS</td>
</tr>
<tr>
<td>--------</td>
<td>---------------------------------------</td>
<td>-----------</td>
</tr>
<tr>
<td>( r_i )</td>
<td>radius at bed free surface</td>
<td>m</td>
</tr>
<tr>
<td>( r_o )</td>
<td>bed radius at inside surface of distributor</td>
<td>m</td>
</tr>
<tr>
<td>( w )</td>
<td>angular velocity</td>
<td>rad/s</td>
</tr>
</tbody>
</table>

**CHAPTER 8**

<table>
<thead>
<tr>
<th>SYMBOL</th>
<th>DESCRIPTION</th>
<th>UNITS</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A )</td>
<td>particle cross-section area</td>
<td>m(^2)</td>
</tr>
<tr>
<td>( AFR )</td>
<td>air to fuel ratio</td>
<td></td>
</tr>
<tr>
<td>( C_d )</td>
<td>drag coefficient</td>
<td></td>
</tr>
<tr>
<td>( C_p )</td>
<td>specific heat at constant pressure</td>
<td>kJ/kg K</td>
</tr>
<tr>
<td>( CV )</td>
<td>fuel calorific value</td>
<td>J/kg or J/m(^3)</td>
</tr>
<tr>
<td>( D_b )</td>
<td>bubble diameter</td>
<td>m</td>
</tr>
<tr>
<td>( E )</td>
<td>bubble hold up</td>
<td></td>
</tr>
<tr>
<td>( F_{ce} )</td>
<td>centripetal force</td>
<td>N</td>
</tr>
<tr>
<td>( F_{co} )</td>
<td>Coriolis force</td>
<td>N</td>
</tr>
<tr>
<td>( F_d )</td>
<td>aerodynamic drag force</td>
<td>N</td>
</tr>
<tr>
<td>( F_g )</td>
<td>gravitational force</td>
<td>N</td>
</tr>
<tr>
<td>( G_a' )</td>
<td>modified Galileo number</td>
<td></td>
</tr>
<tr>
<td>( H_I )</td>
<td>heat input rate</td>
<td>W</td>
</tr>
<tr>
<td>( H_R )</td>
<td>heat removal rate</td>
<td>W</td>
</tr>
<tr>
<td>( L )</td>
<td>bed depth</td>
<td>m</td>
</tr>
<tr>
<td>( L_{mf} )</td>
<td>bed depth at min. fluidisation</td>
<td>m</td>
</tr>
<tr>
<td>( \dot{\theta} )</td>
<td>angular velocity</td>
<td>rad/s</td>
</tr>
<tr>
<td>( \ddot{\theta} )</td>
<td>angular acceleration</td>
<td>rad/s(^2)</td>
</tr>
<tr>
<td>( P )</td>
<td>gas pressure</td>
<td>atm.</td>
</tr>
<tr>
<td>( Q )</td>
<td>gas volume flow rate</td>
<td>m(^3)/s</td>
</tr>
<tr>
<td>Symbol</td>
<td>Definition</td>
<td>Unit</td>
</tr>
<tr>
<td>--------</td>
<td>------------</td>
<td>------</td>
</tr>
<tr>
<td>$Re_{mf}$</td>
<td>Reynolds number at min. fluidisation</td>
<td></td>
</tr>
<tr>
<td>$T_b$</td>
<td>bed temperature</td>
<td>°C</td>
</tr>
<tr>
<td>$T_g$</td>
<td>gas temperature</td>
<td>°K</td>
</tr>
<tr>
<td>$T_i$</td>
<td>gas inlet temperature</td>
<td>°C</td>
</tr>
<tr>
<td>$</td>
<td>U</td>
<td>$</td>
</tr>
<tr>
<td>$U_g(r)$</td>
<td>tangential velocity at radius $r$</td>
<td>m/s</td>
</tr>
<tr>
<td>$U_b$</td>
<td>bubble rise velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>$U_{mf}$</td>
<td>minimum fluidisation velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>$U_{r}(r)$</td>
<td>radial velocity at radius $r$</td>
<td>m/s</td>
</tr>
<tr>
<td>$U_s$</td>
<td>flame propagation velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>$U_t$</td>
<td>tangential velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>$U_z(r)$</td>
<td>axial velocity at radius $r$</td>
<td>m/s</td>
</tr>
<tr>
<td>$V_b$</td>
<td>bubble volume</td>
<td>m$^3$</td>
</tr>
<tr>
<td>$dp$</td>
<td>particle diameter</td>
<td>m</td>
</tr>
<tr>
<td>$f$</td>
<td>fraction of gas bypass</td>
<td></td>
</tr>
<tr>
<td>$g$</td>
<td>acceleration due to gravity</td>
<td>m/s$^2$</td>
</tr>
<tr>
<td>$m$</td>
<td>particle mass</td>
<td>kg</td>
</tr>
<tr>
<td>$n$</td>
<td>numerical multiplier (n*g)</td>
<td></td>
</tr>
<tr>
<td>$r$</td>
<td>radius</td>
<td>m</td>
</tr>
<tr>
<td>$i$</td>
<td>radial velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>$i$</td>
<td>radial acceleration</td>
<td>m/s$^2$</td>
</tr>
<tr>
<td>$r_e$</td>
<td>exhaust port radius</td>
<td>m</td>
</tr>
<tr>
<td>$r_i$</td>
<td>radius at &quot;i&quot;</td>
<td>m</td>
</tr>
<tr>
<td>$r_s$</td>
<td>bed surface radius</td>
<td>m</td>
</tr>
<tr>
<td>$t$</td>
<td>time</td>
<td>sec</td>
</tr>
<tr>
<td>$w$</td>
<td>angular velocity</td>
<td>rad/s</td>
</tr>
<tr>
<td>$z$</td>
<td>axial velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>$z$</td>
<td>axial acceleration</td>
<td>m/s$^2$</td>
</tr>
<tr>
<td>$\rho_a$</td>
<td>air density</td>
<td>kg/m$^3$</td>
</tr>
<tr>
<td>$\rho_p$</td>
<td>particle density</td>
<td>kg/m$^3$</td>
</tr>
</tbody>
</table>
**APPENDIX A**

<table>
<thead>
<tr>
<th>SYMBOL</th>
<th>DESCRIPTION</th>
<th>UNITS</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q_{1,2,3}$</td>
<td>heat fluxes</td>
<td>W</td>
</tr>
<tr>
<td>$T_c$</td>
<td>cooling air temperature</td>
<td>°K</td>
</tr>
<tr>
<td>$T_g$</td>
<td>exhaust gas temperature</td>
<td>°K</td>
</tr>
<tr>
<td>$T_i$</td>
<td>metal temperature on gas side</td>
<td>°K</td>
</tr>
<tr>
<td>$T_o$</td>
<td>metal temperature on cool side</td>
<td>°K</td>
</tr>
<tr>
<td>$T_f1, T_f2$</td>
<td>initial and final film temperatures</td>
<td>°K</td>
</tr>
<tr>
<td>$h$</td>
<td>film heat transfer coefficient</td>
<td>W/m²K</td>
</tr>
<tr>
<td>$k_{1,2,3,4}$</td>
<td>thermal conductivities</td>
<td>W/mK</td>
</tr>
<tr>
<td>$l$</td>
<td>length</td>
<td>m</td>
</tr>
<tr>
<td>$r$</td>
<td>radius</td>
<td>m</td>
</tr>
<tr>
<td>$t_1$</td>
<td>cooling air film thickness</td>
<td>m</td>
</tr>
<tr>
<td>$t_2$</td>
<td>metal thickness</td>
<td>m</td>
</tr>
<tr>
<td>$\mu_g$</td>
<td>gas viscosity</td>
<td>Ns/m²</td>
</tr>
</tbody>
</table>

**APPENDIX E**

<table>
<thead>
<tr>
<th>SYMBOL</th>
<th>DESCRIPTION</th>
<th>UNITS</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>area</td>
<td>m²</td>
</tr>
<tr>
<td>$\Delta P$</td>
<td>pressure loss</td>
<td>N/m²</td>
</tr>
<tr>
<td>$P$</td>
<td>pressure</td>
<td>N/m²</td>
</tr>
<tr>
<td>$Q = \frac{m}{T}$</td>
<td>flow function</td>
<td></td>
</tr>
<tr>
<td>$\frac{m}{T}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$T$</td>
<td>air temperature</td>
<td>°K</td>
</tr>
<tr>
<td>$m$</td>
<td>air mass flow rate</td>
<td>kg/s</td>
</tr>
</tbody>
</table>

xvii
\[ \rho_a \] \quad \text{air density} \quad \text{kg/m}^3

**APPENDIX F**

<table>
<thead>
<tr>
<th>SYMBOL</th>
<th>DESCRIPTION</th>
<th>UNITS</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A_d )</td>
<td>distributor area</td>
<td>m²</td>
</tr>
<tr>
<td>( A, A' )</td>
<td>exhaust port areas</td>
<td>m²</td>
</tr>
<tr>
<td>( \Delta P_b )</td>
<td>bed pressure loss</td>
<td>N/m²</td>
</tr>
<tr>
<td>( P_1, P'_1 )</td>
<td>plenum total pressures</td>
<td>N/m²</td>
</tr>
<tr>
<td>( P_2, P'_2 )</td>
<td>total pressures between bed and distributor</td>
<td>N/m²</td>
</tr>
<tr>
<td>( P_c, P'_c )</td>
<td>freeboard total pressures</td>
<td>N/m²</td>
</tr>
<tr>
<td>( U_b )</td>
<td>gas velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>( V_g )</td>
<td>gas volume flow rate</td>
<td>m³/s</td>
</tr>
<tr>
<td>( m_g, m'_g )</td>
<td>gas mass flow rates</td>
<td>kg/s</td>
</tr>
<tr>
<td>( P_c, P'_c )</td>
<td>combustor static pressures</td>
<td>N/m²</td>
</tr>
<tr>
<td>( \phi_c, \phi'_c )</td>
<td>combustor gas densities</td>
<td>kg/m³</td>
</tr>
</tbody>
</table>
CHAPTER 1

INTRODUCTION

1.1 Background

During the last decade it has become increasingly obvious that we cannot, in the future, depend so greatly on oil and natural gas as our primary sources of energy. This situation has, popularly, became known as the "Energy Crisis" and there is much speculation over which of the many available sources of energy will be utilised to take us into the Twenty First Century and beyond (1). In fact, we are, quite literally, surrounded by "available" energy: what is lacking is the technology necessary to convert this energy into sources of power and heat which can be utilised for domestic and industrial purposes at acceptable cost and minimum pollution. There is, therefore, a "Technology Gap" rather than an "Energy Crisis". One technology which has been developing for the direct and indirect production of power via the combustion of coal and low grade fuels is that based on the phenomenon of "Fluidisation" (2, 3, 4, 5). The contents of this thesis present information about one application of this remarkable phenomenon.

Essentially, fluidisation is the process by which a mass of particulate matter, commonly called a bed, is suspended in a fluid flow within a confined zone. A pictorial representation is shown on Fig. 1.1 A full description of the phenomenon may be found in standard texts(8). Briefly, a bed of material is supported on a plate called the distributor through which the gas can pass, located between retaining
walls. Fluid, gaseous or liquid, passes through pores or perforations in the distributor and then through the bed. The function of the distributor is to present as even as possible an air flow to the bed as is practicable with, in the main, the minimum possible pressure loss. At a particular flow rate, fluid drag forces are just sufficient to support the weight of the particles and overcome interparticle forces. The particles then become suspended in the fluid flow and this condition is known as incipient or minimum fluidisation. Increase in the fluid flow rate will cause the bed volume to expand. If the supporting fluid is a liquid, bed expansion continues as flow rate increases until, at a sufficiently high flow rate, particles are entrained in the flow and carried out of the containment.

If the fluid is a gas, increase in flow rate causes the formation of cavities, similar to bubbles in a boiling liquid, containing few solids. These bubbles carry the gas flow in excess of that required for minimum fluidisation (8) and cause the particles to mix which, as will be seen later, has extremely important consequences on the technological application of fluidised beds. Further increase in gas flow rate may result in turbulent breakdown of the gas flow (8) which, again, affects the nature of particle mixing and gas-particle contacting. Finally, at sufficiently high flow rates, particulate matter is entrained and carried out of the containment. This particulate loss, called elutration, is an important limitation to the operation of fluidised beds. Further increase in gas flow rate leads to pneumatic transport of the solids.

For combustion purposes, the supporting fluid would be gaseous (normally air) and further references to a "fluid" will imply a gas.
Fluidisation using a liquid medium is the domain of Chemical Engineering and is not relevant to the current work.

The principal features that make fluidisation attractive as the basis of a technology for power generation, combustion and heat transfer application are two-fold.

The particulate matter suspended in the fluid flow is in constant motion due to bubbling and turbulence. This motion can be far ranging and violent and is, to some extent, controllable by variation of the rate of fluid flow and the mechanical design of the containment. The motion results in the intimate mixing of the gas and solid phases resulting in large rates of mass transfer and hence solid/solid and gas/solid chemical reactions can be brought to completion rapidly. This feature is of considerable importance in the combustion of solid fuels and it has been found that fluid bed combustors can burn a very wide range of fuels eg. commercial grade coal through to municipal refuse (4, 6).

The second important feature of fluidisation is that high rates of heat transfer are available between the particulates and supporting fluid; between the 'hot' and 'cold' particulates; between particulates and immersed objects and between particulates and the containment walls. The high heat transfer rates help to reduce temperature gradients and many beds operate in near isothermal conditions.

These features play an important role in the combustion process, ensuring that combustible matter reaches ignition temperature rapidly after entering the bed. Also, fluidised beds may be utilised for
heating, rapidly and thoroughly, immersed objects for heat treatment purposes (7) or immersed pipework carrying water for steam production (4). Fluidised beds also provide a means by which combustion temperature may be controlled by removal of heat from the process.

Gas to solid heat transfer rates have been determined in the range 6-23 W/m²/°C (8) and solid to solid heat transfer rates in the range 200-600 W/m²/°C (2, 8). Although the gas to solid heat transfer rates are modest, the actual particulate surface and through which the heat passes is considerable. Botterill (8) states that 1 m³ of material of 100 microns average particle diameter would have a total particle surface area of 30,000m². Thus a relatively modest size of bed has the potential for transferring very large amounts of heat form the fluidising gas to the bed solids and from the solids to immersed objects or pipework.

Fluidised bed combustion has been actively researched because of the great potential for utilising a wide range of fuels. This feature removes much of the fuel specificity of conventional combustion means and allows the use of low grade fuels (ie low C.V. and high ash.) (3, 4). Further, waste disposal via FBC is being actively pursued when the waste will intrinsically support the combustion process or when the waste contains components that are either toxic, corrosive or both. In the latter case addition and removal of receptors/neutralisers for the toxic or corrosive components is a simple procedure.

The ease of addition of chemical receptors to the combustion zone promises great potential of FBC for meeting low levels of acid emissions (SO₂, SO₃) when coals high in sulphur content are burned. Exhaust
gas scrubbing plant should then be unnecessary resulting in considerable saving in capital and running costs and opening the way for the utilisation of vast reserves of high sulphur solid fuels in existence around the world.

Finally, since the combustion in FBC takes place at temperatures that are generally lower than those at which ash softens and alkali salts contained in the ash devolatilise there should be considerably less fouling of boiler tubing downstream of the combustion zone. The nett result is that boilers fired by FBC should suffer shorter downtimes for cleaning at less frequent intervals.

Before continuing to a description of current fluidised bed applications to combusters and boilers and their application to power generation cycles a brief history of fluidisation technology is appropriate.

1.2 Brief history of Fluidisation technology

Probably, the earliest reference to a process which could be described as fluidisation dates from the 16th century (in 8) and is concerned with the roasting of mineral ores.

Contemporary fluidisation technology originates from the work of Winkler (9) where fuel gas was manufactured from small coal particles, d=10mm, previously discarded as waste. The process was patented in 1922 and achieved commercial operation in 1926. A U.S. patent was granted in 1928 (10) but it was some time before the catalytic cracker was fully developed. A co-operative venture between the Standard Oil Development
Company, M. W. Kellogg Company and the Standard Oil Company of Indiana and based on the work of Lewin and Gilliland culminated in the first Fluid Catalytic Cracking process (11). Since then, fluidisation has become an important process in large scale chemical engineering.

The combustion of coal was attempted in the mid 1950's with patents awarded to Badische Anilin (12, 13) Union Carbide (14), and Combustion Engineering Inc. (15, 16).

Meanwhile, fluidisation technology was used for many applications in process industries such as catalytic cracking (17, 18), gasification (19, 20), calcination (21, 22), desulphurisation (23, 24) and drying and dehydration, (25).

Thermal drying of particulates can be accomplished using hot air. This technique could be applied to extract water from sewage, peat, brown coal and colliery washery tailings to produce useful fuel from, otherwise, waste material.

Heat treatment of metals (7) is not confined to temperature treatment. The advantages being rapid heating, uniformity of product quality due to the isothermal nature of the bed and the potential for fuel economy. The metal may also be subjected to oxidising or reducing atmospheres if a fuel gas is burnt in the treatment bed with suitably adjusted fuel/air ratio. Such processes are under continuous development.

Fluidised bed heat exchangers have also been developed. Originally these units were immersed in deep beds sometimes several feet deep.
This, however, produced a severe pressure drop in the fluidising gas and significant power was required to develop the necessary gas flow. The cost of this power outweighed the "value" of the heat recovered and it was not until Elliott and Hulme (26) showed that shallow fluidised beds, having bed depths less that one foot, could operate as effective heat exchangers that commercial development became economic. Research at the University of Aston in Birmingham (27, 28, 29) provided the data and design basis for commercial development of self contained units.

A shallow fluidised bed heat exchanger has been coupled to a vertical chimney through which hot gases from a process pass upwards and solid particles ran downwards. Heat is transferred to the particles which fall into the FB heat exchanger, there to give up the collected heat to tubes carrying water. This "Falling Cloud" heat exchanger, due to Elliott (30), has been used effectively for removing heat from dirty, dust-laden gases. Other examples of dilute phase systems for gas solid contacting are found in chemical engineering applications (11).

A comparatively recent variation of fluidised combustion devices comes from the work of Hatch et al (31) who proposed a fluidised bed, supported on a cylindrical distributor which was rotated about the axis of symmetry to produce a high gravitational field on the bed. In this application, fluidising gas flows radially inwards. Such a unit might be able to operate at higher fluidising velocities than stationary beds and be capable of operating over a very large range of gas flow rates because the rotational speed can be used as an additional control variable. A schematic representation of the Rotating Fluidised Bed (RFB) is shown in Fig. 1.2. This technique was seen as a means of overcoming the turndown problems associated with conventional fluid bed
combustors and boilers (3) as well as possible size reduction. Combustion of gaseous fuel in a rotating fluidised bed (RFB) was first reported by Broughton and Elliott (32). Since then a series of research programmes have developed rotating fluidised bed combustion (RFBC) at atmospheric pressure (33, 34, 35, 36, 37, 38, 39, 40, 41, 42). The work undertaken in these programmes investigated a number of features of the RFB. The early work (32) demonstrated the feasibility of gas combustion and heat transfer in a small RFB using propane as the fuel burned in a bed of small (average size, $\bar{d}_p$, $\approx$200 microns) sand particles. This work was expanded by Metcalfe (34, 41) who presented data on the mechanical design of a gas fired RFB and identified some of the relevant operating limitations for "thin" beds (radial thickness, $t_r$, $\approx$1cm.) of small particles, $\bar{d}_p$ = 200 microns.

Demircan (35, 40) and, more recently, Subzware (42) have extended RFBC experience to "thick" beds ($t_r$=5cm) of large particles, $\bar{d}_p$ = 1mm, in which gas, oil and coal have been successfully burned.

A number of papers have been published examining the feasibility of the RFB for application to rocket propulsion (31, 33) and for coal in power generation schemes (37, 38) whilst others (eg. 36, 39) have examined some of the fundamental aspects of flow characteristics, and bubble and entrained particle dynamics. Combustion at elevated pressure for gas turbine applications could be expected of follow the pattern of combustion in stationary beds.
1.3 Required development steps

Any programme to develop a pressurised rotating fluidised bed combustor (PRFBC) would have the following objectives:

1. To provide information on operating characteristics and assess control requirements.
2. Specification of coal preparation and handling equipment.
3. Design of start-up equipment.
4. Design of pressurised coal feeding equipment.
5. Design of pressurised ash extraction equipment.
6. Design of bed reclamation system.
7. Design and specification of airfeed and rotating mechanism.
8. Design of pressure vessel containment.

This thesis gives an account of the design and development of equipment used to explore items 1, 3, 4 and 5 and presents new information which has exposed problems hitherto unexpected. Some of the mechanisms of RFB combustion have been investigated and the relationship of these mechanisms to RFBC operation have been assessed. Attempts have been made to characterise the operating regions of the RFB in terms of gas velocity, particle size, bed thickness and combustor geometry and to examine some of the aspects of one limitation to performance ash elutration. Much effort has been put in to trying to establish operating guide lines for start-up technique and for transition from gas firing to coal firing. Further, a new combustor design philosophy has been suggested which avoids mechanical integrity problems encountered with previous combustor designs.

Coal preparation requirements will depend upon the results of a study
of the effects on operating characteristics of various grades of fuel. Design of such equipment is beyond the scope of this project.

Bed reclamation is not dependent upon the operation of a RFBC and is under study elsewhere. (eg. 75,76, 77)

The specification of airfeed and rotating mechanisms will depend upon the specific application of the PRFB and as such is not in the field of general research whilst the design of pressure vessel containment is a well established technology.

1.4 Final Choice of Programme

Complete assessment of the items in the above programme was not possible in the time allowed and effort was concentrated on the following selected items:-

1. the building of a suitable test facility for the study of pressurised RFBC.
2. the mechanical design of a laboratory-scale PRFB.
3. The effects on operating pressure loss and the flow conditions at the onset of elutriation of varying bed mass, particle size range and combustor geometry.
4. The effectiveness of inert/reactive particle mixing and the optimisation of fuel feeder design.
5. Development of a pressurised coal feeding system.
6. Combustion trials at atmospheric pressure to attempt to establish satisfactory mechanical design, control of start-up and transition from gas firing to coal firing.
Combustion at elevated pressure was prevented by unexpected difficulties in the start-up trials where successful transition from gas to coal firing was not achieved although limited success was achieved with gas firing alone.
FIG 1.1  SCHEMATIC OF A GAS FLUIDISED BED
CHAPTER 2

INDUSTRIAL APPLICATIONS

2.1 FLUIDISED BED COMBUSTORS AND BOILERS

2.1.1 General

It has already been stated in section 1.1 that fluidisation technology provides a means by which low grade or difficult fuels may be effectively utilised as alternatives to oil and gas as primary energy sources. This is due to the high rates of mass and heat transfer available in a fluidised bed.

The high heat transfer rates obtainable in a fluidised bed may be exploited to produce a boiler which is much more compact than conventional coal fired boiler designs(3). The compactness is due to significantly reduced tubing requirements because the effective heat transfer to immersed tubes can be as much as six times greater than that available in normal gas convective processes (43).

Conventional boiler design dictates that the fuel be burnt at near stoichiometric fuel:air ratios to maximise gas temperatures at boiler inlet and thus heat transfer rates, and to minimise the quantity of heat carried away by exhaust gases and the power required for pumping air into the combustion and heat transfer zones. Typical values of "excess air" (that greater than the stoichiometric requirement) are between 2% and 10% (44). These requirements do not apply to a combustor whose
application is to provide hot gas for a process. A typical example would be a gas turbine whose full power air:fuel ratio would be in the range 40:1 - 50:1, using kerosene, i.e. excess air levels of 200% to 250% (44). In the gas turbine, air fuel ratios near to stoichiometric would result in turbine entry temperatures too high for the turbine to withstand.

The design of a fluidised bed unit will depend upon whether it is to be a boiler or combustor. The operating limitations will depend upon the design, the gas throughput requirements and particle properties. The operating condition at minimum duty will be that at which the bed is just satisfactorily fluidised. This gas flow rate may be greater than that necessary for incipient fluidisation due to requirements for gas cooling of the distributor. If the bed is composed of particles of widely differing sizes, as in the case of all practical systems, there will be a gradual transition from the packed to the fully fluidised state and, for such a bed, Metcalfe (41) suggests the use of the "fully fluidised" condition, i.e., that at which the whole bed is fluidised, as the minimum practical operating condition. The maximum gas throughput is related to particle terminal velocity but is complicated by the effects of bubbles bursting at the bed free surface and, in the case of R.F.B.'s by the interaction of complex, rotating flow fields, and centripetal, fluid drag and Coriolis forces.

Many fluidised bed boilers currently under development utilise beds with depths in the range 1-8 feet (0.3 - 2.8m) (45,46,47), whilst some use very deep beds, up to 4m, for example, the NCB PFBC Research facility at Grimethorpe (48). It has been found that deep beds are necessary if complete combustion of gaseous fuel is to be achieved
within the bed if the fuel is injected directly (49). However, the cost penalty incurred by pumping large quantities of air through the large pressure drop of a deep bed can be high. Furthermore deep beds have other disadvantages arising from the tendency to develop large, unstable bubbles and to "slug". Slugging is the lifting, by a pocket of fluidising gas, of large unfluidised portions of the bed.

Shallow beds avoid slugging, because bubbles cannot grow so large, whilst retaining excellent mixing properties vital for the combustion of solid fuels. Furthermore, pressure loss penalties are lower, resulting in an important saving in fan power. Gaseous fuels, however, must be premixed with the fluidised air to ensure complete combustion (51).

The design of the distributor is always critical in fluidised beds because it must promote even fluidisation and bubble formation. When burning premixed gas in the bed, the flow rate of the air/gas mixture must be such that flash-back of flame into the plenum chamber cannot occur. When the fuel gas is premixed with all of the combustion air, alteration of the fuel/air ratio can no longer be used to prevent flash-back (52) so it must be ensured that the velocity of the mixture through the distributor plate is greater than the flame propagation velocity of the fuel/air mixture. Broughton (52) suggests that an effective open area of 4% is required to ensure jet velocities in excess of flame velocities, when using perforated distributors. This condition alone is not sufficient to prevent burning back in the distributor because the distributor must be kept sufficiently cool to prevent ignition. One solution is to use a porous distributor. Heat transfer from the large internal wetted area helps maintain the distributor at a sufficiently low temperature without the gas temperature reaching
ignition point. This type of distributor has been used successfully in small and medium scale combustion applications (2) but can have the limitation of high pressure loss unless the open area is large. Apart from the prime advantages of the fluidised bed combustor/boiler i.e. the ability to burn a wide range of fuels and considerably reduced heat transfer surface requirements, fluidised bed units have some very useful characteristics which are associated with the temperature of combustion obtained in the bed.

Combustion temperatures in fluidised beds are generally in the range 850°C to 1000°C (2,3,4.). These temperatures are considerably below the normal combustion temperatures found in conventional combustor/boilers. As a result, atmospheric nitrogen is not oxidised to the extent normally encountered and so emissions regulations for NOx can be accommodated more easily (57). Some fossil fuels also contain sulphur and this can be a considerable problem in coal from some parts of the world, giving rise to large outputs of acid gas emissions (44). At present, high sulphur coals can only be used if flue gas desulphurisation is installed (44). It has been found that, by the addition of suitable sulphur receptors, and control of the bed temperature to 850°C (59), emissions of oxides of sulphur can also be reduced to acceptable levels (57).

At temperatures below 1000°C, the ash formed during coal and oil combustion does not fuse; it is friable and does not stick to heat transfer surfaces within or above the bed (4). Fouling of boiler tubes by molten ash is a considerable problem in conventional boiler designs and limits the allowable ash content of various fuels (coal in particular) and the application of the fluidised bed to boilers is, again, welcome. Furthermore, the ash that is formed in fluidised beds
seems to be highly receptive to alkali metals which cause severe corrosion problems in boiler tubing - the implications are obvious. The ash, being soft and friable, does not normally cause erosion problems to immersed tubing and the scrubbing action of the particulate matter in the bed tends to give tubing a bright polish, thus maintaining high transfer coefficients.

All fluidised beds suffer from rather a narrow range of operating velocities which limits the operational flexibility of a plant because of limited turndown from the maximum output (3). Fig. 2.1 shows a typical operating envelope for a fluidised bed combustor. It will be seen that the upper and lower boundaries are associated with bed temperatures. It has been found (3) that when burning coal at atmospheric pressure without cooling tubes in the bed, combustion intensities of $1 \text{MW/m}^3$ and a turndown of 3:1 are possible. Higher intensities, $2 \text{MW/m}^3$, but reduced turndown, 2:1, are obtained if cooling tubes are immersed in the bed. The most common method that have been investigated as a means of improving turn down and operational flexibility are as follows:

i) Control of particle size, density and material type.

ii) Multi Zone Unit with sequentially positioned zones.

iii) Multi Zone unit with independently operated zones.

iv) Variation of immersed heat transfer surface area by mechanical means or change of bed depth.
v) Operation at elevated gas pressures.

2.1.2 Control Via Particle Size, Density and Material Type

As an alternative to limiting the maximum gas velocity the fluid bed unit might be designed to operate at very high velocities, 10m/s or more. Here the particulate matter is circulated between two or more containments and the process is termed "entrained bed" fluidisation or "dilute phase" fluidisation. Kunii and Levenspiel (11) show several examples of this process applied in Chemical Engineering. Yerushalmi and Crankurt (53) have suggested this process as suitable for large scale combustion plant and Nack et al (54) describe the small scale development of such a unit. Other examples include the Lurgi (143) recycling fluidised bed. This type of fluidised bed uses variations in fluidisation properties due to particle size and density to achieve high throughputs and potentially large turndown ratios (143). Separate heat transfer fluidised beds can be used to further increase the range of turndown available(144). Recycling of fines, which is a concomitant feature of the recycling fluidised bed, helps restore lost combustion efficiency due to carbon carryover. The recycling bed thus incorporates a feature that is often necessarily added on to conventional FBC boiler.

2.1.3 Multi-Zone Units with Sequentially Positionned Zones

Maximum gas velocity in gas fired fluidised bed combustors is limited by the quality of fluidisation in that, at high flow rates, significant fuel gas bypass can occur in the bubbles of a shallow bed (52). Broughton (52) and Pillai (37) have developed models to predict bypass.
Multi zone units can reduce losses due to unburnt fuel. However, the danger of burning or clogging of the downstream distributors in sequentially zoned units makes this approach rather impractical. In solid fuel fired fluidised beds, velocities are limited by excessive elutriation of unburnt fuel or by slugging. The former may lead to the problem of downstream distributor blockage.

2.1.4 Multi-Zone Units with Independently Operated Zones

Multi zone units with independently operated zones are receiving attention due to potential ease of mechanical design, manufacture and operation. One example is the atmospheric fluid bed combustor (AFBC) under development at West Virginia University (46). Here, several cells, of basically the same design, are to provide information on the integration of separate units as well as development of unit components. The multicell approach has capital cost advantages in that it may prove cheaper to manufacture several simple units than one complicated one and may ultimately prove to have operating advantages. The principal disadvantage is the time required to heat up the moderately deep beds, typically 1 ft. deep, during start-up.

2.1.5 Variation of Immerged Heat Transfer Surface

Variation of immersed heat transfer surface can be achieved by positioning the surfaces, eg tubes, just above the defluidised bed surface (37). The tubes will then only come into contact with particles after bed expansion occurs and thus is controlled by fluidising velocity. The disadvantage of this method is that burning of the heat transfer surfaces can occur or soot can be deposited if volatiles are
not completely burnt out when the surfaces are not fully immersed.

Rather than rely on bed expansion it is possible to increase or decrease the bed depth by adding or removing bed material. This is not difficult in conventional fluidised beds and is a method used for start up and control in very large units (48,55).

2.1.6 Pressurised Fluidised Bed Combustion

Operation at elevated pressure offers two distinct advantages. First, it is possible to pass an increased mass of gas through a given flow area at a given temperature and velocity. It should therefore be possible to burn an increased mass flow rate of fuel. Second, work may be extracted from hot, pressurised gases by expansion through a turbine. These two features have led to considerable effort in the development of pressurised fluidised bed combustor (eg. 47, 48, 60, 62, 64).

Operation at elevated pressure has both advantages and detrimental effects on fluidisation characteristics. Less gas passes through the bed in bubbles which reduces the mixing effectiveness (56), but reduced bubbling improved the "quality" of fluidisation thus improving heat transfer coefficients between gas and particles. This increase is due to the large proportion of gas in the emulsion phase of the bed (8).

For a given mass throughput, increase of the gas pressure will result in reduced bed cross sectional area, assuming that the gas velocity is to remain constant. The reduced size of unit may reduce capital cost though this must be balanced against the increase in cost due to the
production of pressure vessels and pressurised fuel transport systems, etc. Increased pressure also increases combustion intensities and larger heat transfer surface areas may required to ensure both uniformity and control of bed temperature. The resulting tube packing density may cause problems associated with defluidised zones between tubes. These problems can be reduced if temperature control is achieved by variation of excess air level. It is feasible that in-bed tubing can be dispensed with altogether but this extreme also has control problems (62). A compromise of combined steam and hot gas generation may prove to be advantageous because of the high efficiency of the resulting thermodynamic cycle coupled with a lower technical risk and more flexible control of bed temperature by variation of water/flow rate in the bed steam tubing (62).

2.1.7 Variation of Effective Bed Weight

If a means can be found by which a very large range of gas flow rates can be accommodated then a very flexible heat exchanger or combustor can be produced. The minimum flow rate is that at which the bed will first operate satisfactorily in the fluidised state. The maximum is dependent upon the physical characteristics of the bed material and the containment design. To avoid the mechanical complications of a "variable" containment we look for physical characteristics that may be varied at will.

It has been seen that when the gas flow rate exceeds a certain value for a given bed of material in a given containment, particles will be entrained. If it were possible to increase the apparent weight of these
particles then a correspondingly higher gas flow rate would be required before entrainment occurred. A simple method by which the weight of a bed of particles can be varied at will is to situate the bed in a mechanically generated centrifugal field. If the distributor plate were cylindrical in form with suitable end boundary plates and this assembly is rotated about its axis of symmetry, a vessel has been produced by which a variable centrifugal field and hence a variable weight bed are available. This idea, first reported by Hatch et al (31) and later patented by Elliott (149) in the U.K., gave birth to the Rotating Fluidised Bed (RFB). Fig. 2.2 shows a diagrammatic representation of this device.

Several applications of the RFB have been investigated at laboratory scale, including: (a) exploratory work that might lead to a nuclear reactor for spacecraft propulsion (31), (b) a gas/coal combustor and fume incinerator (34), and (c) a gas/oil/coal combustor and steam generator (35). Power cycle analysis (13) has suggested that a combustor of a size suitable for power generation work could be economically feasible and theoretical studies (36,37,38) have described some of the fundamental design criteria.

Thus far, however, only limited experience (12) has been gained with regard to pressurised rotating fluidised bed combustion (PRFBC) and the aims of the work described in this thesis were to investigate some of the unknown areas associated with PRFBC. The scope of this work has been outlined in Chapter 1.
2.1.8 Summary of Section 2.1

a) Considerable research has shown how fluidised beds may be used as the basis for combustor and boilers that can burn a wide range of fuels whilst meeting emissions regulations.

b) There would appear to be definite advantages to using shallow beds from design, operation and cost considerations.

c) Turndown and combustion intensities are limited for a given design of bed but careful choice of bed material and careful design can reduce the impact of these limitations.

d) Operation at elevated pressures offers the prospects of increased combustion intensities and smaller plant size for a given duty.

2.2 POWER CYCLES

The type of power cycle, to which a fluidised bed unit is to be applied, significantly affects the design and the achievable operating envelope and characteristics. The choice of cycles will depend upon economic factors, eg. units with low capital cost or low running costs or both, or it may depend upon practical factors eg, "difficult" fuels, plant size, flexibility and turndown.
2.2.1 Conventional Combined Cycle Plant

Combined cycle generation (CCG) is a well established idea and most major industrial gas turbine manufacturers have published schemes quoting cycle efficiencies between 40 - 47% (61). Conventional CCG appears to be based upon a gas turbine fired with gas or oil and whose exhaust is ducted to steam raising plant and associated steam turbines, where additional fuel may be burnt.

Reference 61, a survey of co-generation schemes describes three possible cycles. In the "exhaust fired" or "high efficiency" cycle, the gas turbine provides the smaller proportion of power and its exhaust is used as heated combustion air for a conventional boiler furnace. In the "unfired" or "recuperation" cycle the gas turbine provides the higher proportion of power and the exhaust goes to heat recovery boilers to generate steam for steam turbines. The former has high capital costs and improvement of gas turbine performances has only a marginal effect on cycle efficiency but relatively low grade fuel may be burnt in the boiler stage. The latter requires high grade fuel for the gas turbine which provides about 70% of the output power (61).

Overall efficiency of the recuperation cycle is governed by the (gas) turbine entry temperature, stack temperatures, condenser vacuum and the amount of steam in the steam turbine. Use of steam for regenerative heating is not appropriate as this reduces the amount available for producing power in the steam turbine. Feed heating is achieved via economiser recirculation. Component efficiencies are important and improvements in gas turbine technology is the main reason for the high cycle efficiency achievable, 55 - 60% (61).
The supplementary fired recuperation cycle is a combination of the two described above which allows easier matching of steam conditions to those required by standard steam turbines and reduces the cost per kW of electricity generated. However, because the supplementary fuel is burnt only in the steam boiler cycle, when temperatures are lower, cycle efficiency is reduced.

Where suitable fuel is available and the overriding consideration is to conserve fuel, Coats (61) suggests that it is the unfired recuperation cycle that is the natural choice. Coats (61) gives examples of both plant philosophies and points out the following advantages for gas turbine plant:-

1) Modular construction and standardised manufacture.

2) Short manufacture cycle, installation time and commissioning time.

3) Minimum number of operating staff or automatic operation.

4) Reliability and availability

Reference 61, however, does not comment on the disadvantages pertaining to gas turbine plant which include:-

a) Stingent requirements of fuel composition

b) Noxious emissions are possible due to inefficient combustion at low power, and high temperatures at high power.
c) Complex fuel control systems: this is offset by much more rapid response to load change than possible with steam turbines.

2.2.2 Pressurised Fluidised Bed Power Cycles

Broadly power cycles suitable for the application of fluidised bed combustion fall into two groups (62).

a) Air Cycles: In this group see Fig. 2.3 air from a compressor is split into two streams. One stream provides the fluidising and combustion air whilst the other passes through in-bed tubing to be heated indirectly. After gas cleaning, the combustion stream is remixed with the bypass and the mixture then expands through the turbine. The proportion of flow shared between the two streams leaving the compressor will depend upon required operating characteristics. The amount of air ducted through the in-bed tubing may vary from zero (high excess air) to an amount which leaves only sufficient flow for the fluidisation and combustion air to be supplied to the bed (ca. 20% excess air) (62). The possible cycle efficiencies are lower than those of the conventional gas turbine due to expected higher pressure losses through the combustor and clean-up systems although this effect could be reduced by increased gas turbine pressure ratio (62).

A variant of the air cycle is the gas turbine in a closed circuit, Fig. 2.4. This variant would, in general, operate at low pressure ratios but very high pressure levels with a subsequent increase in heat transfer coefficient between air and bed tubing walls. This would reduce the tubing requirements. In these units combustion may take place at atmospheric or elevated pressures (63).
Highley (64) states that the open cycle would operate at cycle efficiencies in the region of 25%, depending upon pressure ratio, when operated in high excess air conditions whilst the closed cycle would operate at higher efficiencies than the open cycle due to reduced excess air requirements. Brown (65) suggests cycle efficiencies approaching 45% for closed circuit plant.

b) Supercharged Steam/Gas Turbine Cycles: In this group see Fig. 2.5 bed temperature is maintained at 800-900°C by generating steam in tubes in the bed. Combustion air is expanded through a gas turbine after cleaning. The gas turbine drives the compressor as well as providing external power. There is some scope for variation in the disposal of steam tubing but, according to Roberts et al (62), there are fewer possible cycle variations than with air cycles. This scheme is basically the same as the "high efficiency" cycle of conventional CCG and the main scope for improvement lies in transferring as much heat as possible from the gas turbine exhaust into the steam cycle.

In conventional steam generating systems there is always an incentive to operate at low excess air levels, but the reverse appears to be true for combine cycles. Roberts (62) quotes a cycle efficiency of 40% at 20% excess air rising to 41.5% at 100% excess air. The increase is due to more heat being incorporated at high temperature in the cycle. Further, the literature shows that high levels of excess air is benificial to combustion efficiency, reduces corrosion and improves turndown (3). However, increases in capital costs on pressure vessels and clean up systems are inevitable.
Hoy (66) suggests that the supercharged steam cycle plant would have greater control problems than the air cycle plant but fewer technical risks due to lower temperature to which in-bed tubing would be subjected.

2.2.3 Summary of Section 2.2

Considerable effort is being expended on developing fluidised bed combustion systems suitable for use in CCG.

Based on the foregoing information the author believes that the open cycle gas turbine in the conventional recuperation cycle has greater advantages. High excess air levels produce beneficial combustion characteristics whilst recuperative heat recovery allows high cycle efficiencies. This conclusion significantly influenced the author's design of pressurised rotating fluidised bed combustor.

2.3 GASIFICATION

The production of gaseous fuels from solid and liquid feedstocks using fluidised bed methods which originated from the work of Winkler in the 1920's has been under development in the U.K. and the USA for some time (24,67, 68, 69, 70). The importance of gasification is that a fuel free from ash and sulphur can be produced. This fuel may be burnt in conventional gas turbine combustion chambers with no modifications necessary to the combustor or turbine and no gas cleanup plant is required between combustor and turbine. This policy offers the prospect of relatively low development and capital costs for gas turbine power generation plant. The major development costs and technical risks arise
in the development of very large scale gasification plant. Also, the
technical development of gas turbines expected in the next decade in the
direction of increased Turbine Entry Temperature, T.E.T, from 1000°C
(current maximum) to 1200°C could only be achieved using ash-free fuels.

On the economic side however, it was reported, early in 1979 (72),
that pressurised fluidised bed combustion may be significantly cheaper
than gasification. Using the fuel costs of solid fuel firing in a
conventional boiler without exhaust gas clean up as a datum of 100
units, the consulting engineering group of Kennedy and Donkin compared
the fuel costs of a number of proposals for a 2 x 300 MW power station
in Britain sited 40 miles from a coal mine producing coal of 5.025
MJ/KG:-

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Low energy gasification</td>
<td>: cost 108 units</td>
</tr>
<tr>
<td>2. Low energy gasification in combined cycle</td>
<td>: cost 92 units</td>
</tr>
<tr>
<td>3. High energy gasification</td>
<td>: cost 116 units</td>
</tr>
<tr>
<td>4. High energy gasification in combined cycle:</td>
<td>cost 99.5 units</td>
</tr>
<tr>
<td>5. Pressurised fluidised bed in combined cycle</td>
<td>: cost 86 units</td>
</tr>
</tbody>
</table>

Gasifier efficiency was assumed to be 85%. Had it been assumed at
90% it is reported that cost estimates for low energy gasification with
combined cycle and PFBC would be roughly the same. However, the study
considered that capital costs for PFBC plant would be less than gasified
plant and perhaps less than a conventional P.F. boiler. The report
recognised the attraction of the higher T.E.T. feasible with
gasification but suggested that the availability of the high technology
gas turbines might be poor, offsetting the advantages. The issue is
unresolved because no commercial PFBC or CCG plants have been built at
the time of writing. It was suggested that, in the long term, a
national methane grid be established, based on large high-energy
gasifiers sited at pitheads.

At about the same time in 1979 (73) it was reported that many U.S.
manufactures were abandoning their gasification programmes. However,
Texaco, Shell-Koppers and the N.C.B are firmly committed to
gasification (72). Indeed, Shell-Koppers has announced successful
operation of their pilot entrained bed gasifier sited near Amsterdam,
Holland (74). These factors plus the preference for gasification shown
by the CEBG in Britain will serve to keep gasification a strong
contender for electricity within Europe for the foreseeable future.
FIG 2.1  TYPICAL FBC OPERATING ENVELOPE
FIG 2.2 SCHEMATIC OF A ROTATING FLUIDISED BED
FIG 2.3 SIMPLIFIED AIR-CYCLE GENERATION PLANT
FIG 2.4  CLOSED-CIRCUIT GENERATION PLANT
FIG 2.5  SIMPLIFIED STEAM AND GAS TURBINE GENERATION PLANT
CHAPTER 3

LITERATURE SURVEY

3.1 Introduction

For brevity, the literature reviewed here is that pertaining to the design and operation of fluidised bed combustors. Although there has been extensive research in fluidisation technology, the rotating fluidised bed combustor (RFB) is a significant departure from the main stream of previous research and so there is only a limited body of directly applicable information available. For the most part, assessments must be made from information obtained from conventional fluid bed technology and due recognition must be given to its limitations.

3.1.1 Previous Reviews of RFB Literature

Metcalfe (41) gives a comprehensive review of RFB literature published before 1977. This literature describes exploratory experiments designed to investigate the fluid dynamics, combustion and heat transfer performance of the RFB. Work also reviewed described various theoretical studies including $SO_2$ absorption and economic assessments.
Metcalfe's survey extracted the following important points from the literature:

1. Fluidisation in the RFB occurs in layers, being initiated at the inner free surface and then progressing towards the distributor as fluidising velocity increases.

2. The nonuniform nature of fluidisation renders conventional theoretical predictions of minimum fluidisation velocity inappropriate so that the definition of $U_{mf}$ for stationary fluidised beds and correlations for its prediction are inadequate for the equivalent condition in the RFB.

3. Bubbling in the RFB had received no theoretical attention and practical knowledge was limited.

4. The pressure drop across the bed displayed the same broad characteristics as found in stationary beds although the magnitude of the pressure loss required to achieve full support of the bed may be less than the weight of bed per unit area.

5. Exploratory experiments had shown that the combustion of premixed propane/air mixtures is possible in the RFB, but no systematic study had been performed to establish design or operating limitations.
3.2 Choice of Particle Size

Not all particles are readily fluidised. Some, due to small size, agglomerate whilst others due to very large size or high density cannot be easily supported by fluid drag forces. Baeyens and Geldart (78) suggest a classification of particles based on mean diameter and density which allows selection of suitable combinations of the two parameters for fluidisation purposes under normal gravitational conditions. The classification gives four groups, identified by the letters A, B, C, D and is shown diagramatically in Fig. 3.1 Groups A and C contain very small particles \( \bar{d_p} \leq 100 \) micron which are impractical for combustion applications. Groups B and D contain particles of size range 40-500 micron and \( > 600 \) micron respectively with densities in the range 1400 kg/m\(^3\) to about 4000 kg/m\(^3\). Particles described by these two groups are those most commonly used in fluidised bed combustors. In operation, a fluidised bed may contain a range of particle sizes and densities embracing groups A, B and C.

3.2.1 Particle Size Choice for the RFB

Metcalfe (41) found little trouble in fluidising group 'B' particles of sand (density= 2630 kg/m\(^3\)) and zircon (4000 kg/m\(^3\)) with size ranges 90 - 180 micron, 180 - 250 micron and 250 - 355 micron in a rotating fluidised bed under imposed radial accelerations of between 5 and 30 times normal gravity. Demircan (40) and Subzwaari (42) have both successfully fluidised sand particles of about 1mm diameter at similar applied accelerations. It would appear that the Geldart classification applies to rotating bed fluidisation but the boundaries between divisions may need to be revised due to increases in particle weight.
The effects of particle size on combustion characteristics are not fully understood but certain relevant points must be borne in mind. First, there may be practical limits on minimum and maximum size for stable combustion of some fuels. Broughton (108) has suggested an analysis for the combustion of gaseous fuels in conventional fluidised beds based on gas bypass in bubbling beds. Comparing the results from this analysis with measured data, Broughton concluded that gas bypass was too great to support combustion in a methane-fuelled conventional fluidised bed if the mean particle diameter was less than about 200 micron. Metcalfe (41), however, has shown that stable propane combustion is possible with sand particles of about 130 micron mean diameter in a rotating fluidised bed combustor, but noted that temperature control became difficult when the mean particle size was raised to 300 micron. Clearly, applied radial acceleration caused the bed to take on the characteristics of a conventional bed made up of larger particles and supports, the earlier suggestion of the need to shift the Geldard classification boundaries in order to describe rotating beds. Metcalfe's observation of unstable temperature does not necessarily mean that there is an upper limit to the size of particle for which gas combustion is possible; the reasons for the lack of control need to be investigated. Demircan (40) successfully burnt propane in beds containing particles of about 1mm diameter. His rotating bed was equipped with water cooling tubes while that used by Metcalfe was not. It is possible that the extra control obtained by heat removal in the bed was the reason that stable combustion could be achieved but evidence from two markedly dissimilar combustors is insufficient to support this argument and further work is necessary to clarify this point.
Careful choice of the size of particles to be used in a combustor is vital to the ultimate success because the attractive features of a fluidised bed are all in some way, related to the size of the particles as will be seen in succeeding sections of this chapter.

3.3 Minimum Fluidising Velocity $U_{mf}$

The minimum fluidising velocity, $U_{mf}$, is of considerable importance for two reasons. First, it describes the flow condition at which the 'average' particle within a bed is just supported and thus gives the very minimum flow condition for operating a bed in a near-fluidised condition. Second, many bed phenomena important to combustor design, viz., bubbling, bed expansion, heat and mass transfer and elutriation have been described in terms of $U_{mf}$. It is, therefore, a very useful quantity to evaluate for any combination of bed particles and combustor design.

3.3.1 $U_{mf}$ in Stationary Fluidised Beds

The usual definition of $U_{mf}$ (8) comes from a consideration of the plot of bed pressure loss $\Delta P_b$, versus fluid flow rate (linear velocity $U$ or mass velocity per unit area $G$), see Fig. 3.2. Estimation of $U_{mf}$ is determined from the intersection of straight lines, one drawn through the packed bed characteristic, the other drawn through the fluidised bed characteristic as shown in Fig. 3.2. The flow velocity $U$, at which this intersection occurs may be significantly lower than that required to fully fluidise the bed, $U_{fs}$, and Richardson (in 80) suggests that it is probably more repeatable to determine $U_{mf}$ as $U_{fs}$ could be influenced by the initial bed packing and containment. It must
be appreciated, though, that $U_{fs}$ is the practical minimum operating condition and should, therefore, be determined. In fact, since, the pressure loss characteristic is best determined by starting with a well fluidised bed and then reducing the flow rate progressively, a repeatable characteristic curve is obtained, as demonstrated by Deinken (97) and Metcalfe (41), in the RFB, avoiding the problems suggested by Richardson.

Considerable effort has been expended in efforts to find ways of predicting $U_{mf}$ for given particle and gas conditions. Unfortunately, purely theoretical methods require knowledge of two factors that are very difficult to determine, viz, bed voidage and particle sphericity. Under or over estimates of either quantity can result in very poor estimates of $U_{mf}$ whilst spatial variations of these quantities through the bed worsens the reliability of the estimate.

The best known equation is due to Ergun (81).

$$\int g \left( \phi_p - \phi_g \right) dp^3_g = \frac{150(1-\text{Emf})}{\mu^2} \left( U_{mf} \text{ dp} \phi_g \right) + \frac{1.75}{\text{Emf}^3} \left( U_{mf} \text{ dp} \phi_g \right)^2 \quad 3.1$$

Substituting $Ga = \int g \left( \phi_p - \phi_g \right) dp^3_g$ and $Re_{mf} = \frac{U_{mf} \text{ dp} \phi_g}{\mu}$
we get

\[ Ga = 150 \frac{(1 - \text{Emf}) \text{Re}_{mf}}{\text{Emf}^3} + 1.75 \frac{\text{Re}_{mf}^2}{\text{Emf}^2} \]  \hspace{1cm} (3.2)

This equation assumes spherical monosized particles, but can be used for non-spherical particles if the diameter used is equal to that based on the specific surface of the particles.

Inclusion of particle sphericity changes equ. 3.2 to:

\[ Ga = 150 \frac{(1 - \text{Emf}) \text{Re}_{mf}}{\text{\rho} \text{Emf}^3} + 1.75 \frac{\text{Re}_{mf}^2}{\text{\rho}^2 \text{Emf}^3} \]  \hspace{1cm} (3.3)

Wen and Yu (83) made estimates of the factors \((1-\text{Emf})/\text{\rho} \text{Emf}^3\) and \(1/\text{\rho}^2 \text{Emf}^3\) based on the results of 284 experimental determinations and found that

\[
(1-\text{Emf})/\text{\rho} \text{Emf}^3 = 11 \quad \text{and} \quad 1/\text{\rho}^2 \text{Emf}^3 = 14
\]

Substituting in 3.3 and rearranging we find

\[ \text{Re}_{mf} = (33.7^2 + 0.0408 \text{ Ga})^{1/2} - 33.7 \]  \hspace{1cm} (3.4)

A standard deviation of \(\pm 34\%\) was found to apply over a range of Ga from \(10^{-3}\) to \(10^3\).

Eq 3.2 may, however, be split into two parts where

\[ Ga = 150 \frac{(1-\text{Emf}) \text{Re}_{mf}}{\text{Emf}^3} \]  \hspace{1cm} (3.5)
or \[ \text{Ga} = \frac{1.75 \text{Re}^{2}}{\text{Em}^{3}} \]

Equation 3.5 applies when \( \text{Re} < 20 \). Here, viscous flow dominates i.e. laminar flow.

Equation 3.6 applies when \( \text{Re} > 100 \) where kinetic flow considerations dominate i.e. turbulent flow.

Between these limits of \( \text{Re} \), transition flow regimes occur within the bed and the full equation must be used.

Using Wen and Yu's factors

Equation 3.5 gives

\[ U_{mf} = \frac{d}{p} (\frac{d}{p} - \frac{2a}{a}) \frac{g}{a} \]

\[ = 1.650 \]

Equation 3.6 gives

\[ U_{mf}^{2} = \frac{d}{p} (\frac{d}{p} - \frac{2a}{a}) \frac{g}{a} \]

\[ = 24.5 \mu \]

A variety of values have been suggested for the constant in 3.7 (89). It is likely that the value of the constant for any situation will depend on the design of the fluid bed containment. Broughton (89) suggests that 1420 gives reasonable estimates of \( U_{mf} \) in most cases.
3.3.2 Prediction of $U_{mf}$ in the RFB

The characteristic pressure loss curve of a rotating fluidised bed has been shown to have the same features as that of a stationary fluidised bed with an identifiable minimum fluidising velocity (eg. 97,109) occurring at a pressure loss approximately equal to the bed weight, see Fig. 3.3

Metcalfe (41) compared predictions of $U_{mf}$, using eq. 3.7, with values of $U_{mf}$ determined in a rotating fluidised bed but found that the predicted values were not good. He suggested reasons for the discrepancies, viz, invalid assumptions for $dp$ and $Emf$ and variation of flow regime through the bed. These highlight the uncertainties in using theoretical or empirical prediction methods and support the general feeling that it is best to measure $U_{mf}$ or $U_{fs}$ in any fluidised bed application.

For the RFB, the prediction of $U_{mf}$ is complicated by the acceleration of the fluid flow because the flow area reduces as it passes from the distributor radially inwards through the bed and due to the variation of applied radial acceleration through the bed. However, Gelperin (86) showed that $U_{mf}$ could be calculated with reasonable confidence from:

$$Re_{mf} = \frac{Ga^\prime}{1400 + 5.22\sqrt{Ga^\prime}}$$ 

3.9
where
\[ G\alpha' = \frac{dp^3 \delta g (\delta p - \delta g) ng}{\mu g^2} \]

and \( ng \) = applied radial acceleration.

Donaldson (87) has shown that the radial component of \( Umf \), \( Ur \), can be derived from the Ergun equation by

\[ \frac{dp}{dr} = \frac{150 (1-E)^2 \mu g U_r + 1.75 (1-E) \delta f U_r^2}{E^3 \frac{dp}{dr} E^3 \frac{dp}{dr}} \]

If the radial acceleration imposed is large enough for the effect of the Earth's gravitational acceleration to be insignificant then equation 3.10 reduces to

\[ \frac{dp}{dr} = (1-E) (\delta p - \delta g) ar \]

Shakespeare et al (35) combined 3.10 and 3.11 with

\[ \text{Emf} = 0.4 \text{ and } Ur = Ut^2 \text{ with the result that} \]

\[ (Ur \frac{dp}{dr} g) = 25.7((1 + 5.53 \times 10^{-5} (\frac{1}{\nu} - 1) \frac{dp}{dr} Ut^2 / \nu)^{1/2} - 1) \]

(\( \frac{g}{\nu^2} \))mf
Which may be re-written as

$$Re_{mf} = 25.7 \ast ((1 + 5.53 \ast 10^{-5} \text{Ga'})^{1/2} - 1)$$  \hspace{1cm} (3.13)$$

This is the same as that derived by Richardson (in 80) for \(U_{mf}\) in a stationary bed with the modified Galileo number, \(\text{Ga'}\), in place of \(\text{Ga}\).

We see that variation of tangential velocity will affect the minimum fluidisation velocity, \(U_{mf}\), and for this reason the thickness of the bed in an RFB should be controlled to prevent fluidisation occurring at the free surface before it occurs at the distributor. This situation could limit the available operating range of the RFB.

Refering to equ. 3.12 it would probably be better to express \((Ut^2)/r\) as \(w^2/r\) to avoid the uncertainty of determining \(Ut\). This, of course, assumes that the gas flowing through the bed itself is in solid body rotation rather than free vortex flow. Kroger et al (112) have shown that there is evidence to support this approach. Thus the modified Galileo number suggested by Gelperin (86):

$$\text{Ga'} = \frac{dp^3 \int g (fp - fg) \rho g}{\mu^2}$$  \hspace{1cm} (3.14)

might be better written as

$$\text{Ga'} = \frac{dp^3 \int g (fp - fg) w^2 r}{\mu^2}$$  \hspace{1cm} (3.15)
where \( w \) = angular velocity
and \( r \) = 'average' radius of bed particles.'

The average bed radius is very difficult to define but could, perhaps, be represented by the mass weighted mean radius

\[
\frac{r_m}{m_i} = \sum \frac{r_i m_i}{m_i}
\]

Where \( r_i \) and \( m_i \) are the radii and masses of a number of shells that, together, make up the bed.

Metcalfe (41) examined five correlating equations for the prediction of \( U_{mf} \) in the RFB based on his own measurements. These equations were:

\[
Re_{mf} = (33.7^2 + 0.0408 \text{ Ga})^{1/2} - 33.7
\]

3.17

\[
Re_{mf} = (3.389 + 0.0014 \text{ Ga})^{0.752}
\]

3.18

\[
Re_{mf} = 4.1854 + 0.0004 \text{ Ga}
\]

3.19

\[
Re_{fs} = (1.708 + 0.0024 \text{ Ga})^{0.689}
\]

3.20

\[
Re_{fs} = 9.7579 + 0.0003 \text{ Ga}
\]

3.21

His general conclusion was that, whilst some of the specific equations appeared to fit his measured data better than others, the modified Wen and Yu correlation 3.17, was sufficiently accurate for
Initial estimation of $U_{mf}$ up to $Ga = 7 \times 10^4$.

### 3.3.3 Effect of Gas Temperature on $U_{mf}$

Gas density and viscosity both change as gas temperature changes and so $U_{mf}$ would be expected to be temperature dependent. At low $Re \leq 10$ we see, from the Ergun equation, that

$$U_{mf}d \frac{1}{\mu}$$

or

$$U_{mf}d \frac{1}{\tau^n}$$

whilst at high $Re \geq 10000$

$$U_{mf} \propto \sqrt[4]{Ga}^{-0.5}$$

Doheim and Collinge (88) and Broughton (89) have both published results that confirm the validity of the former equation whilst Saxena and Vogel (90) have shown that in the transition region between $Re \geq 10$ and $Re < 1000$, $U_{mf}$ is roughly independent of temperature at constant pressure. Saxena and Vogel showed that, based on their experimental data, Wen and Yu's correlation, eq 3.4 should be modified to

$$Re_{mf} = (25.28^2 + 0.0571 \text{ Ga})^{1/2} - 25.28$$

\[3.22\]
in order to take account of large changes in temperature and pressure of the fluidising gas stream.

The importance of these results is that, if constant pressure operation is undertaken, the minimum operating condition reduces as the temperature of the fluidising gas stream increases, for small particles where \( \text{Re} < 10 \), but that it is relatively stable for larger particles where \( 10 < \text{Re} < 1000 \). Thus, choosing moderately sized particles with \( \bar{d}_p \) between 500 micron and 800 micron may ease operational control requirements.

3.3.4 The Effect of Pressure On \( \text{Umf} \)

Increasing the static pressure of the fluidising gas stream increases the density of the gas and hence the buoyancy component of \( \text{Umf} \) may change significantly, particularly when very high pressure, ca. 10bar, are employed.

Saxena and Vogel (90), when fluidising dolomite particles with \( \bar{d}_p = 650 \) micron and 704 micron found that, at a given bed temperature, \( \text{Umf} \) decreases with increasing pressure. However, at a fixed pressure they claimed that \( \text{Umf} \) was roughly independent of temperature in the range 291K to 700K when the particle Reynold's Number was transitional, \( 10 < \text{Re} < 100 \). These results would then be of considerable importance when operating a pressurised fluidised bed. If constant pressure operation is assumed, at a fixed rotational speed for an RFBC, then the minimum operating conditions will be relatively unaffected by changes of temperature for moderately sized particles and \( 10 < \text{Re} < 100 \). As was seen in the previous section, there may be a slight reduction possible at
high temperatures due to viscosity changes. However, with fixed
temperature and variable pressure operation problems of elutriation
control may arise as pressure is increased and $U_{mf}$ decreases. It may
be necessary, therefore, to increase rotational speed as pressure is
increased in order to counter possible increases in elutriation rate
when running at high gas velocities.

Knowlton (91) investigated the high pressure fluidisation of several
coal derived materials and found that the most accurate prediction of
minimum fluidisation velocity resulted if the 'minimum' fluidisation
velocity was taken to be that at which the bed was fully supported, in
the same way as Metcalfe (41). This velocity was predicted using Wen
and Yu's correlation for narrow ranges of particle size for the bed
under consideration. The $U_{mf}$ predicted for each size fraction was
weighted by the weight fraction of the size cut. The sum of the
products gave $U_{fs}$ ie.

$$U_{fs} = \Sigma w_i U_{mf} i$$  \hspace{1cm} (3.23)

The agreement between measured and predicted values using this method
depended on the material fluidised but was always better than
conventional methods. These results lend considerable weight to the
arguments put forward by Metcalfe on the characteristics of minimum
operating condition.

3.4 **Bubbling and Mixing**

Effective solids mixing in a fluid bed combustor is of vital
importance for successful operation. It is well established that mixing
between the bed surface and the distributor is due, mainly, to bubbling, in the absence of artificially induced circulation. Considerable effort has been expended studying the bubbling properties of fluidised beds (eg. 8,80,92). The important aspects for combustor design are:-

(a) The estimation of minimum bubbling condition

(b) Rates of solids mixing

(c) Rates of bubble growth

(d) Maximum stable bubble size

(e) Instabilities.

The latter three are of interest as they may have a direct bearing on minimum and maximum permissible bed depths. The first two are of principal interest as either may define the minimum operating condition. In fact, Metcalfe (41) found that, in a small RFB, the mixing of small coal particles \( \bar{d}_p \) ca. 800 microns, in bed of fine sand, \( \bar{d}_p = 212 \) microns, was satisfactory above a fluidisation index, \((U/U_{mf})\), of 2, although he did not attempt to measure mixing rates. Similar results have been reported by Subzware (42) for larger sand particles, \( d_p = 1 \) mm.

Direct measurement of mixing rates is extremely difficult and so rates are usually estimated indirectly from measurements of heat transfer rate. These types of study have shown that mixing in the direction perpendicular to the bulk gas flow may be very small in
comparison to that in the direction of fluid flow (93) in conventional fluidised beds.

3.4.1 Bubbling and Mixing in Stationary Fluidised Beds

According to Botterill (8) there is no general agreement on the relationships which describe mixing rates, however, rates do appear to increase with decreasing mean particle diameter and with increasing bubble volume flow rate. Lewis et al (93) showed that the degree of solids mixing tends to increase with increases in viscosity and density of the fluidising gas. This suggests that mixing rates will benefit from the combination of increased temperature and pressure in pressurised combustion whereas, with atmospheric combustion, density reduction at increased temperature might effect mixing rate increases due to viscosity increases. The inclusion of in-bed tubing can have both advantageous and detrimental effects. In-bed tubes may enhance bubble production which could improve mixing, but Botterill (8) reports that Mori and Nakamura have found that a high packing density of tubes impedes particle flow and hence mixing.

The generation of bubbles, bubble sizes, volume flow rates and instabilities are strongly influenced by the design of distributor, the interactions between distributor and bed particles and the properties of the fluidising gas and bed particles. Many correlations and models have been proposed to explain bubbling behaviour eg. (8, 80, 92) and studies have demonstrated methods of estimating bubble size, velocity and distribution eg. Yoshida et al (92).

Summarising the above work, simple relationships have been proposed
which describe some bubble properties, viz.,

\[
D_b = \frac{1}{g} \left( \frac{H_{mf}}{(U_{mf} - U)} \right)^2 \left[ \frac{(U - U_{mf})}{0.711} \right]
\]

Bubble Rise Velocity \( U_b = U_0 - U_{mf} + 0.711 (g D_b)^{1/2} \)

Slugging velocity \( U_s = 0.35 (g D_b)^{1/2} \)

Size distributions require complex description. Yoshida et al (92) have successfully predicted measured bimodal distributions and state that their model is suitable for situations when other types of distribution might be encountered.

All of the preceding comments apply to relatively deep beds where the average bed diameter \( D_b \) is considerably less than the bed depth, \( H_b \) or where the bed depth is such that there is sufficient gas residence time for bubbles to develop and evolve.

3.4.2 Bubbling and Mixing in the RFB

In cases where \( H_b \) is less than \( D_b \) or where residence times are small, less than 1 sec, bubbles may develop and evolve in different ways or possibly not at all. In the radially thin bed RFB both conditions apply and, as applied acceleration increases and so increases the effective particle density, bubble formation and growth and bubbling regimes may change drastically. Little work has been published on bubble phenomena in the rotating fluidising bed. Those which do exist (39, 42) suggest that spontaneous bubble generation may occur as the fluidising gas passes through the bed. Also, high fractions of gas as
bubbles may occur at inner radii. This calls into question the assumptions of solid body rotation at these locations.

Spontaneous bubble generation may have been the cause of an instability observed by Deinken et al (97) who, in the mid 1950’s studied the RFB as the basis of a compact high gas flow rate nuclear reactor for space craft propulsion. This instability referred to as 'bed bounce' was explained in terms of an imbalance between bed weight, voidage and pressure loss but the factors necessary to initiate the condition were not considered. Spontaneous bubble generation at the inner radii in the bed could have been the event that first produced the increased expansion and voidage that triggered the bounce sequence of rapid expansion and contraction of the bed. If it is possible to predict the onset of spontaneous bubble formation it may be possible to avoid operating conditions at which bed instabilities could occur.

The prediction that high fractions of gas may occur as bubbles at the inner radii suggest that the flow dynamics associated with free vortex flow may be dominant in the surface layers of the RFB. If this is so, the operating characteristics of the freeboard of an RFB combustor may be more akin to those associated with cyclone and tangential swirl combustors than those associated with the conventional dense phase fluidised bed combustor. Elutriation can be expected to be associated with particle dynamics similar to those describing the operation of the cyclone separator and models describing elutriation from conventional fluid bed combustors may have little bearing on that actually observed.

Mixing in the RFB has been investigated by several workers (40, 41, 41, 127). Metcalfe (41) studied mixing in a small RFB at atmospheric
temperature and pressure. He found that good radial mixing occurred when the fluidisation index of the bed particles was approximately 2. He found that it was not necessary to match the minimum fluidising velocity of the bed and coal particles. Good mixing was achieved with coal particles, typically 300 - 500 micron, and small sand particles, 180 - 250 microns, with a fluidising index approximately 2. Axial mixing was, however, poor requiring an even distribution of fuel from the fuel feeder in order to achieve good mixing across the bed area.

Demircan (40) found satisfactory mixing of coal particles 1 - 6mm diameter in a bed of sand particles 0.7-1.0 mm diameter at similar fluidising indexes to those used by Metcalfe.

Subzwari (42) also found good radial mixing of large coal particles with smaller sand particles and concluded, as did Metcalfe, that even spreading of the coal from the feeder was essential to achieve satisfactory lateral mixing in the bed.

Kroger et al (127), on the other hand, found that when similar sized glass beads were mixed in an RFB good lateral mixing could be achieved. This conclusion was reached from tests on beds composed of a mass of clear glass beads and a mass of coloured beads, the coloured beads being separated from the clear beads at the beginning of each test.

Tangential mixing via the temporary entrainment of bed particles thrown into the freeboard was also observed, supporting one aspect of the theoretical predictions that can be made from Chevray's (39) bubble and entrainment model. Predictions of this type were also made by Subzwari (42) for both bed and fuel particles from which he concluded
small fuel particles (ca. 200 microns) could return to the surface of an RFB provided that gas flow rates and rotational speeds were chosen appropriately.

Clearly, mixing in the RFB is not a simple phenomenon and would appear to be a function of particle density, shape and size distribution as well as fluidising velocity and rotational speed. For all practical purposes it would seem prudent to assume poor lateral mixing and design a fuel feeder that produces a wide distribution of fuel particles.

3.5 Elutriation

The third major flow regime of interest in fluidised bed fluid dynamics is that in which material is carried away from the body of the bed and lost in the exhaust stream. The phenomenon is known as elutriation and may restrict the maximum operating condition. As the gas flow rate through the bed increases small particles may be entrained in the gas flow because the flow velocity exceeds their terminal velocity, $U_t$, whilst larger particles may be thrown into the freeboard space above the bulk of the bed by violent bursting of bubbles. In both cases, conditions may occur in which the particles, now in the freeboard, may be lost from the system. Prediction of when these conditions will occur and the rate governing equations is extremely difficult, however as improved relationships and models become available, the design of fluid bed combustors with specified operating regimes should be possible with greater confidence.
3.5.1 Flutriation from Stationary Fluidised Beds

Terminal velocity depends upon particle diameter and sphericity. It is important that the expected maximum gas velocity does not exceed the terminal velocity of the smallest particle required to remain in the bed. For a spherical body \( U_T \) may be obtained from:

\[
U_T = \left( \frac{4g dp (\rho_p - \rho_g)}{3 \rho g C_d} \right)^{1/2}
\]

which is derived from fluid mechanicial considerations. \( C_d \) is a drag coefficient that will vary from particle to particle. In the range \( R_e = 0.4 \) to 5000 which applies to most fluidised bed applications an empirical relationship for spherical particles is

\[
C_d = \frac{10}{Re}
\]

hence

\[
U_T = \left( \frac{4 (\rho_p - \rho_g) \rho g^2}{225 \rho g^2} \right)^{1/3} dp
\]

Non-spherical particles require totally empirical methods eg (95). The non sphericity, angularity and tumbling motion of the particles make it impossible to determine easily applicable relationships for terminal velocity. As a first approximation, however, equation 3.34 may be used with \( dp \) approximated from the specific volume.

The onset and rate of eluatration may not however be solely related
to $U_t$. Lin et al (95) quote a number of correlations for the rate of elutriation. These correlations use dimensionless groups such as \( (\mathbf{f}_p - \mathbf{f}_g)/(\mathbf{f}_g), \) \( (\mathbf{f}_g \ U_t \ \text{dp})/\langle \mathbf{H} \rangle, \) \( (U_g - U_t)^2/(\mathbf{f}_g \ \text{dp}), \) \( U_{mf}/(U-U_{mf}) \) and \( U_g^2/(\mathbf{g} \ \text{dp} \ \mathbf{f}_g^2) \) and, in most cases, have a limited range of application. These correlations were compared with measured elutriation rates from a bed composed of silica sand and char particles. From the measured data a simple power law correlation was proposed relating elutriation rate constant, \( E'/\langle \mathbf{f}_g \ U_g \rangle \), to particle Froude number \( (U_g^2)/(\mathbf{f}_g \ \text{dp}) \) viz:

\[
E' = 9.43 \times 10^{-4} \left( \frac{U_g^2}{\mathbf{f}_g \ U_g} \right)^{1.65}
\]

\[
\mathbf{f}_g \ U_g \quad (\mathbf{f}_g \ \text{dp})
\]

This correlation appeared to work well for small particles, ca. 75 microns and low velocities ca. 30 cm/sec and a variety of freeboard heights. With particles larger than 75 microns the particle transport disengagement height changes sufficiently to require modifications of the correlation in order to fit the measured rates.

The initiation of elutriation has been linked to the bursting of bubbles at the free surface of the bed, the projected particles either coming from the bulge formed as the bubble reaches the surface or from ejection of particles caught in the bubble wake (113). The former mechanism has been used as the basis of a model (96) proposed as a first stage in the analysis of entrainment. The model successfully predicted solids projection rates from the bed surface provided that not slugging occurred. It was pointed out that solids projection is only the first step in the elutriation process and that the other stages, viz, particle dynamics between the bed surface and transport disengagement height and
particle dynamics above this height require further investigation.

3.5.2 Elutriation from the RFB

In the case of the rotating fluidised bed, a model has been proposed by Chevray et al (39) and investigated by Subzware (42). This model examines the dynamics of bubbles and particles within the rotating system at a fundamental level and as such, is not dependent on the design of the combustion space (whereas all correlations are specific to a given combustor).

Chevray’s model predicts two important parameters which control elutriation, namely,

i) The distance between the bed surface and the exit port.

This determines the smallest particle that will not elutriate. As an example, Subzware showed that, for a bed 200mm I.D. and 80mm axial length with an exit port diameter of 120mm, at conditions where 60 micron particles are predicted to elutriate from the bed surface furthest from the exit port, 100 micron particles may elutriate from the bed surface nearest to the port. In an extreme case this imbalance could lead to very poor combustion efficiency due to significant amounts of the fuel particles being lost before they had burnt down to an acceptably small size.

ii) Exit Port Diameter

As the exit port diameter is reduced, so the maximum size of particle
that will be elutriated at any given flow condition will also reduce. This occurs because, the rotational viscous core in the freeboard, centred on the axis of symmetry of the RFB also reduces (since its size is directly related to port diameter). According to the model, only particles that penetrate this viscous core will be elutriated; others will either return to the bed or remain suspended in the free Vortex flow region that appears to be located between the inner layers of the bed and the viscous core boundary. As the viscous core shrinks, smaller particles are found to be retained in the freeboard region. Hence for a combustor design where the exit is a plain hole, reduction of the hole diameter can be expected to increase the fluidising velocity at which a given size of particle begins to elutriate.

Elutriation has also been investigated by Levy et al (128). Using a simple model, based on the terminal velocity of the particles, they obtain qualitative agreement between predictions and experiment data on loss rates for round glass beads. They found that the airflow rate at which elutriation was initiated was a strong function of angular velocity and bed thickness. Once elutriation began, they found that particle loss was a function of airflow rate as well as angular velocity and bed thickness. Their model overpredicted the airflow rate at which elutriation was seen to initiate and this was attributed to enhanced particle freeboard velocities due to bubble bursting and also to vertical variations in the radial velocity of the gas.

Clearly, then, particle entrainment and elutriation are complex phenomena. Further research is needed to evaluate the effects of changing combustor geometry and to reconcile experimental results obtained with round glass beads, with elutriation data obtained from
beds of sand and other combustor bed materials.

3.6 Heat Transfer

Fluidised bed reactors were first developed to capitalise on the mass and heat transfer properties associated with dense phase fluidisation. In section 3.4 mass transfer due to turbulent mixing and bubbling was described. Heat transfer is an equally complex phenomenon although it is somewhat more amenable to quantification and a considerable body of knowledge has been built up about it. An extensive subject in itself, only the more relevant features of heat transfer will be discussed here in relation to combustion of solid, liquid and gaseous fuels. Much of the work described here is drawn from Botterill (8) whose review of fluidised bed heat transfer research is one the of most comprehensive available.

3.6.1 Gas to Particle Heat Transfer

Published data for gas to particle heat transfer shows great variation. Kunii and Levenspiel (11) report a thousand fold variation between different observers and this is the source of some uncertainty in any discussion of gas to particle heat transfer.

Gas to particle heat transfer is usually described by a Nusselt number, \( h_d p/k_g \), \( (\text{Nu}) \). The ideal case of an isolated sphere in an infinite medium is described by the equation given by Ranz and Marshall (104):

\[
\text{Nu} = 2 + 0.6 \text{Pr}^{1/3} \text{Re}^{1/2}
\]
At zero velocity when \( Re = 0 \), \( Nu \) should be equal to 2 where heat transfer is by conduction and the authors demonstrated this result experimentally (105). However, there are many examples of measured values of \( Nu \) of less than 0.001 (21). Newey (98) suggests that consensus of opinion is that the theoretical model is not truly representative and the explanation for measured Nusselt numbers less than 2 is due to the fact that real particles are not spherical and the real effective heat transfer area is considerably less than the total surface area of the particles. This occurs because, at the points of contact between particles, the thermal boundary layers surrounding neighbouring particles merge and gas outside these layers cannot easily penetrate and reach the particle surfaces. As gas velocity increases the boundary layer thickness reduces and particle motion tends to break down the boundary layer, more of the particles' surface area becomes available for heat transfer and \( Re \) tends towards the predictions of the simple sphere equation.

A first approximation for \( Nu \) can be obtained from a relationship due to Kuni and Levenspeil (11) viz,

\[
Nu = 0.003 \, Re^{1.3} \quad (3.31)
\]

This result does not take account of variations of heat transfer that may be due to the physical design of the apparatus noted by many workers eg. (106) and so any correlation should be used with great caution.
Typical is that due to McGaw (106)

\[ \text{Nu} = 0.353 \text{ Re}^{0.9} \left( \frac{\overline{d}}{D} \right)^{0.47} \left( \frac{\overline{d}}{D} \right)^{0.47} \left( \frac{\overline{d}}{D} \right)^{-0.19} \]  

3.32

\[ \left( z \right) \left( P \right) \left( D \right) \]

This relates to a shallow bed, 33mm deep, in which large particles (a few mm in diameter) were fluidised on a distributor having a triangular array of holes. Significant changes in physical dimensions or distributor design will certainly invalidate the correlation. Gas to particle heat transfer coefficients are always small, in the range 3 to 50W/m² K (8) but when this is taken in combination with the very large surface area of the particles, total heat transfer is seen to be very large i.e. heat is transferred very rapidly and gas and particles temperatures will be very similar within a short distance from the distributor. Zabrodsky (103) has suggested that very little temperature difference remains after the gas has passes through a monolayer of particles. Hence the gas follows particle temperature and not vice-versa.

3.6.2 Heat Transfer to the Walls and Immersed Surfaces

Newey (98) describes two models of bed to surface heat transfer. One, due to Mickley and Fairbanks (99), uses an analogue of gas to surface convective heat transfer. Here, small volumes of gas and solid approach the surface and heat transfer occurs until they are removed by mixing processes and replaced by new volumes of material and gas. This theory assumes that the thermal properties of the gas and solid are uniform; this is not realistic.
Botterill and Williams (100) considered the heat transfer between a single sphere surrounded by gas and a surface. This more closely resembles the heterogeneous dense phase fluid bed. This model has been extended (101) to involve two particles which better accounts for differing gas and particle properties.

Newey states that the latter models are superior to Mickley's when the residence time of the packet is short but the packet theory has been improved by Kubie and Broughton (102) by the inclusion of a boundary layer at the surface. Correlations of bed to surface heat transfer are numerous but these vary considerably due to the wide range of apparatus and beds employed. Botterill (8) suggests that, for group B particles, a correlation for maximum heat transfer due to Zabrodsky (103) is adequate for temperatures up to 900 K after which radiant heat transfer is significant. Zabrodsky's correlation is

\[ h_s = 38.8 \frac{g}{p^{0.2} k_g^{0.6} d_p^{-0.36}} \] 3.33

To avoid underestimation of heat transfer surface area Botterill suggests that 70% of the derived value be used in initial design estimations.

3.6.3 Heat Transfer in the RFB

Heat transfer rates in the RFB were measured by Broughton and Elliot (32) who found that the most significant parameter affecting bed to tube heat transfer coefficient was that of Galileo number, Ga (which is the same as Archimedes number Ar). This suggests that heat transfer rates are dependent principally, upon particle size and applied acceleration. It would be expected from this finding that highest heat
transfer rates would occur with large particles and high radial accelerations (provided that the bed was fluidised).

Demircan (40) has also measured heat transfer rates in the RFB. He showed that heat transfer rates increased linearly as particle size was increased from 0.6 to 2.4mm and, to a lesser extent, as bed thickness was reduced. The former result is consistent with the findings of Broughton and Elliot. The reasons for the latter effect were not discussed but may be due to increased particle mobility in the shallower beds.

Typical heat transfer rates from bed to tubes appear to be in the range 50 - 250 W/m²K (32, 40) for particles in the size range 0.14mm - 2.4mm. The lowest coefficients occur with small particles and vice versa.

3.7 Combustion

A wealth of information is available on the characteristics of combustion in fluidised beds. This information relates to both deep and shallow beds burning a wide variety of fuels. Coal combustion has probably received the major research effort and this allows some reasonable generalities to be offered in its description. Much of the information has been generated from pilot and full scale plant constructed in a large number of designs and thus it should be possible to distinguish combustion features which appear independent of mechanical design from those which change from one design to another.

Alongside the practical efforts of many research groups and
manufacturers, considerable effort has been made in attempts to model the combustion of solid fuel in fluidised beds. The efforts range from the modelling of single particles combustion e.g. (115) to models of the gross behaviour of solid fuel fired combustors (eg. 115) and boilers (eg. 116).

3.7.1 Combustion of Gaseous Fuels

3.7.1.1 Gas Combustion in Stationary Fluidised Beds

The combustion of gaseous fuels in fluidised beds has received attention as it affords an easy method by which a bed may be heated to a temperature at which the solid fuels will burn continuously.

The work has shown two important aspects that must be considered when opting for a gas fired start-up cycle. The first is that if complete combustion is take place within the bed, then the bed depth should be large with respect to the mean bubble size expected at the start-up fluidising velocity (108). This condition is especially true when the fuel gas is injected into the fluidisation air upstream of the plenum chamber. In this case, combustion will take place efficiently in a shallow fluidised bed provided that there is sufficient residence time for adequate gas transfer between the bubble and emulsion phase to allow complete consumption of the fuel gas (108).

Broughton (108) has proposed a model to describe the combustion of premixed gas air mixtures in shallow fluidised beds which shows that gas bypass, in bubbles, will limit the minimum average particle size that
can be used in the bed if self-sustaining combustion is to occur. This lower limit is different for different fuel gases depending upon the gas calorific value and its reactivity.

3.7.1.2 Gas Combustion in the RFB

This subject has been studied by several workers. Broughton and Elliot (32) found that stable combustion of propane could be achieved in a laboratory scale RFB operating between up to 10g with mean particle sizes between 141 and 300 microns. It is interesting to note that the smallest mean particle size used in these tests is rather smaller than the smallest size that Broughton had found suitable for premixed combustion of methane in a conventional bed (108). This bears out the comments made pertaining to particle size and gas bypass in the previous section.

Broughton and Elliott found no problems in start up when a pilot flame was used as the ignition source, the initial fuel air mixture ratio was slightly substoichiometric and the initial fluidising velocity was such that only the surface layers of the bed were incipiently fluidised. The only problem encountered in gas combustion was near the lean limit where some temperature instability occurred. No reason was offered for this effect.

Metcalfe (41) studied the combustion of propane extensively in two laboratory scale RFB combustors which did not utilise in bed cooling tubes. The experiments covered the combustion of propane in beds of
approximately 1cm radial thickness composed of particles having mean sizes between 212 microns and 298 microns. It was found that, depending upon particle size and imposed radial acceleration between 5g and 30g, stable propane combustion could be achieved for volumetric air to fuel ratios between 30:1 and 70:1. A typical operating envelope is shown in Fig. 3.4.

The lower limit on fluidising velocity was chosen to be four times the predicted hot $U_{mf}$ resulting in minimum velocities between 0.75 and 1.5 m/s, hot. The upper velocity limit was set by combustion inefficiency. When the ratio for $CO: (CO_2 + CO)$ reached 0.02 the upper operating limit was deemed to have been reached. This resulted in maximum operating velocities between 2 and 5.5 m/s, hot.

The rich fuel air ratio operating limit was set as that to achieve a bed temperature of 1150°C thus allowing a safety factor of 50°C below the sand particle fusion temperature. The lean fuel to air ratio operating limit was set as that at which combustion was lost from the bed, viz. approx. 900°C.

Metcalf reported temperature fluctuations in the bed when using the largest particles but offered no explanation for this occurrence.

The typical turndown Metcalf achieved at a fixed applied acceleration was 3:1. However, by increasing the radial acceleration from 10g to 30g an overall turndown of 5:1 was achieved between the maximum allowable bed temperature condition and a minimum operating velocity of 10 times $U_{fs}$. 

69.
Demircan (40) also performed gas combustion trials in a small RFB. This unit incorporated in bed cooling tubes for added control of bed temperature. The bed particle sizes used were 0.6 - 1.2mm and 1 - 2.4mm (mean sizes 0.85 mm and 1.55mm respectively). Whilst the operation of this combustor was somewhat different to those operated by Metcalfe the results obtained by Demircan substantiated those obtained by Metcalfe. One interesting difference is that Demircan did not report any temperature instabilities with the generally larger particles he used. It is possible that the extra control afforded by the inclusion of in bed cooling tubes was sufficient to minimise any fluctuations.

Typical operating air to fuel ratios used by Demircan ranged from about 30:1 to 50:1. He found that by using smaller particles stable combustion could be achieved over a wider range of air to fuel ratios supporting the findings of Metcalfe. Further, turndown ratios up to 5:1 were obtained by varying the rotational speed as well as the air to fuel ratio.

3.7.2.1 Combustion of Liquid Fuels in Stationary Beds

The combustion of liquid fuels in fluidised beds has not received extensive attention since most liquid fuels can be burned with little problem in conventional equipment. However because of the ability to retain fuel sulphur in the bed, the fluidised bed combustor has attractions for burning low grade high sulphur oils. Typical of the work is that done by the British Petroleum Research Centre and at BCURA (17). Here a variety of fuel oils have been burned in beds of sand and alumina. The early problems of liquid coatings on particles quenching the combustion reactions have been overcome by careful design of the
distributor and fuel injection systems. Combustion efficiencies approaching 100% were achieved using excess air ratios above 20% and bed temperatures between 850 and 900°C. Higher bed temperatures were found unnecessary and excess air ratios greater than 60% produced no measurable improvement in performance.

Roberts et al (64) reported that heavy fuel oils can be burned with efficiencies approaching 100%. However, no details of the combustor used were given but the authors note that, in order to prevent fuel vapour bypass in bubbles, it is vital that the initial fuel distribution is as uniform as technically possible.

3.7.2.2 Combustion of Liquid Fuels in the RFB

Demircan (40) has studied the combustion of gas oil in a small RFB. The work, although very limited in scope, showed that light oils could be burned successfully in the RFB. Preheating of the bed to about 500°C was necessary before oil feeding was started. Steady state oil combustion was achieved at a bed temperature of 700°C with a gravimetric air to fuel ratio of about 47:1. These results were obtained with large particles (1.55mm) fluidised under an applied acceleration of about 18g at velocities of 0.59 and 0.64 m/s.

3.7.3 Combustion of Solid Fuels in Stationary Fluidised Beds

This section of the literature review will examine the range of fuel types that have been used in FBC plant and the considerations necessary for efficient combustion. Emissions, and their control, will be examined in section 3.8.

71.
The types of solid fuels that have been burned in FBC plant range from char (e.g. 144,188) which is essentially pure carbon, through high grade coals (e.g. 45) to high ash and high sulphur content coals (e.g. 119) and colliery tailings (120). Further other 'solid' fuels such as oil shales and tar sands have been burned successfully in FBC units (e.g. 121).

Generally speaking, combustion efficiency is good for all types of coal provided certain constraints are met. Highley (64) reports that, at atmospheric pressures, between 2% and 10% of the carbon content of the fuel may be elutriated. This can be minimised by the use of high excess air ratios (greater than 20%), 'high' bed temperatures (typically 850°C) and low fluidising velocities. The fluidising velocity is controlled, principally, by the required heat output from the bed and so, to minimise carbon losses, the recycling of fly ash and elutriated carbon can be used (122). However, very high recycling rates (up to four times the fuel feed rate (112)) are required for relatively modest improvements (about 2%) in combustion efficiency: these may prove impractical or uneconomical.

Highley (64) also reports that combustion at elevated pressures improves carbon burn-up. This is attributed to the improved air:solid contact due to smaller bubbles in the bed.

Roberts et al (62) report that the improvement in combustion efficiency due to increased pressure reduces as the excess air ratio is increased. Carbon utilisation of about 92% at 20% excess air was achievable whilst operating at pressures of 350 KN/m². An increase in
utilisation of about 1% could be achieved by increasing the excess air ratio to 30% but the improvement over atmospheric combustion reduced from about 2% to about 1.5%. If this trend were to continue at higher excess air rates, atmospheric and pressurised combustion could achieve equal utilisation of carbon at an excess air rate of about 5%. Whilst this trend may apply to coal combustors generally, the excess air rates and achieved efficiencies could vary considerable from one combustor to another. Nevertheless, the data is useful as a guide to the level of combustion efficiency and excess air ratio to be set as a target for the RFB in the present work. [It should be noted that the bed depth used in the plant reported on by Roberts was, typically, several feet and this would aid carbon burn-up. The thin bed RFB would be required to achieve similar carbon utilisation with beds of a few centimetres thickness. This may prove impractical in a once through system and fines recycle may be required for efficient combustion.]

A further variable in carbon utilisation reported by Highley is the ash content of the coal. It seems that, as the ash content of the coal increases, the rates of carbon loss also increase. Similar effects were noted by Beacham (45) who also reports that reductions in carbon utilisation are also associated with increases in sulphur content but to a much lesser extent than found with increases in ash content.

Haque et al (123) report poor combustion efficiency with high sulphur Indian coals. Efficiency was seen to fall with increasing excess air ration from about 90% at 5% excess air to about 70% at 20% excess air. The reducing efficiency was attributed to reduced coal particle residence time and increased cooling rates arising from increased fluidising velocity rather than sulphur content. These
findings are in accord with those reported by Roberts (45) and Highley (62).

Further support for the need to minimise the elutriation of carbon fines is shown by results reported by Anlisio et al (124). Improvements in combustion efficiency from 85% to 95% using fines recyle were found to apply to both pilot and full scale plant for various operating conditions of fluidising velocity and bed temperature. However, it was also reported that carbon utilisation of 99% was achieved by using a separate carbon combustion chamber operating at a low fluidising velocity and a high temperature. Both of these conditions are more conducive to the efficient combustion of small carbon particles than the respective conditions used in the main combustion beds but the extra capital cost of the carbon burn-up cell may render this approach uneconomic.

These combustion experiments were performed in beds of silica sand, limestone, dolomite and coal ash utilising particles of sizes varying from 400 microns to 2.4 mm. Coal particle sizes used were typically a few millimetres. There did not appear to be any significant differences in carbon utilisation for the different bed materials or fuel particle sizes used although as Hoy (125) points out, the carbon utilisation can be significantly improved by burning large lumps of coal rather than particles. He reports carbon utilisation as high as 97% without fines recyle when singles coal (12.5 - 25 mm) was burned in a bed 150 mm deep. However, if the lumps of coal shatter due to thermal shock, as some coals do, this is equivalent to feeding fine material into the bed with attendant effects on efficiency.
When the fuel used has a high volatile content it is important that the initial fuel distribution is as even as possible and deep within the bed in order to minimise the quantity of volatiles that will burn above the bed (126).

Apart from this provision, there do not appear any other considerations to be taken into account at this stage when burning high volatile coal.

3.7.4 Combustion of Solid Fuels in the RFB

This subject has only been studied extensively by two workers, viz. Metcalfe (41) and Demircan (40), who used combustors of different designs. Metcalfe's unit used shallow beds (1 cm radial thickness) of small particles ($\bar{d}_p = 212$ and 297 microns) and had no cooling tubes in the bed volume. Demircan's combustor used deeper beds (5 cm radial thickness) of large particles ($d_p = 1\text{mm}$) and incorporated cooling tubes in the bed volume. These two studies allow some assessment of the merits of the two approaches to be made.

Metcalfe first attempted the combustion of anthracite particles of 600 - 1000 microns in a sand bed of 180 - 250 microns. The bed of sand was first preheated to 950°C using propane combustion. Coal feeding was then started at a rate that, from heat release calculations, was expected to maintain 950°C and the propane fuel gradually shut off.

Initially, Metcalfe noted uneven combustion of the anthracite due to maldistribution of the fuel. This problem was reduced by positioning the pneumatic fuel feeder head near the top of the combustor. However,
even when relatively even combustion was obtained, a high level of carbon monoxide (6.1%) was found to be present in the exhaust gas. This shows that the residence time of this gas in the bed was insufficient and suggests that beds deeper than 1 cm should be used.

Another problem encountered by Metcalfe was the elutriation of carbon fines at applied accelerations of 10g with fluidising velocities just over 1m/s. Increasing the anthracite feed size to 850 - 1000 microns and the applied acceleration to 15g delayed elutriation of fines until a fluidising velocity of 1.97m/s was reached. However, the sand bed had started to elutriate at 1.38m/s, thus reversing the problem.

Localised combustion was another problem encountered to the extent that fusion of bed, coal and ash occurred. When fusion occurred the areas of localised combustion could not be dispersed by increasing the fluidising velocity.

Many of the problems Metcalfe reported were attributed to the relatively poor air distribution in the plenum chamber resulting from ducting the air through the drive shaft. He discovered that the better air distribution, produced in his modified combustor, produced much more even mixing of sand and anthracite particles and hence reduced the tendency for local fusion. However, he does not report any improvement in combustion efficiency or a reduced tendency for elutriation. Fusion was found to occur in the improved combustor and this was attributed to unsteady fuel feed rates and a large inventory of bed carbon although quantities are not given.

An important procedure that Metcalfe developed was a reliable
start-up technique. Initially he tried gradual change-over from propane to coal firing, and sudden propane shut off with simultaneous coal feed initiation but both failed to achieve satisfactory steady state coal combustion. He then found that satisfactory changeover could be obtained by preheating the bed to 950°C, using propane; shutting off the gas flow and starting the coal feed at a rate high enough to counter the cooling of the bed due to the heat transfer from sand particles to fluidising air and then, after 30 seconds, reducing the coal feed rate to the level necessary for continuous combustion at 950°C.

Demirson (40) studied the combustion of a high volatile content coal (volatile content 40% w/w). He reported few problems either in start up or steady state combustion when burning large fuel particles, 1 - 6mm, in beds composed of sand particles 0.7 - 1.0mm and 0.6 - 2.2mm diameter. He noted that there were some instabilities in the combustion of the coal and it was not uniform across the bed surface but adjustment of the cooling water flow rate allowed satisfactory control of bed temperature and prevented fusion occurring. Some combustion of volatiles occurred above the bed even though bed depths up to 4cm were used. Combustion efficiencies were estimated to be in excess of 90% with about 1% carbon loss. The balance of the carbon loss is not accounted for. Emissions of carbon monoxide are reported as between 0.35% and 0.5% by volume which are considerably lower levels than reported by Metcalfe and was found to be dependent on bed temperature, excess air, rate of coal feed and coal particle size. The data shows that CO concentration is a strong function of bed temperature reducing rapidly as temperature is increased. The data also suggests that CO concentration reduced with increasing fluidising velocity. The CO concentration also appears to be a weak function of bed depth,
i.e. residence time, as would be expected.

Satisfactory combustion was achieved with excess air ratios between 20% and 120%. Lower ratios tended to result in particle fusion whilst higher ratios tended to produce excessive elutriation or heat loss.

3.8 Emissions from fluidised bed combustors

One of the principal attractions of the fluidised bed combustor is that noxious emissions have been found to be less than those emitted from conventional combustion plant. This has been achieved by operation at low combustion temperatures, less than 1000°C, and by including chemically active materials in the bed. As a result, these is confidence amongst both research groups and industry that stringent pollution regulations expected to be in force worldwide before the end of the century can be satisfied by various applications of FBC.

Due to the nature of a fluidised bed and its operation, particulate emissions must be controlled by external means. This is necessary for the protection of associated plant (boilers and gas turbines), plant personnel and also the general public. A great deal of work is being done in an effort to characterise particulate emissions and to develop methods for their control.

3.8.1 Emissions of nitrogen and sulphur oxides from stationary fluidised bed combustors

The principal gaseous pollutants that are being, and will be,
regulated are oxides of nitrogen and oxides of sulphur. Oxides of nitrogen and sulphur are known to give rise to "acid rain" and it is believed widely that the oxides of nitrogen could reduce the size of Earth's protective ozone layer. There is however, great controversy about the impact of both effects.

It has been widely reported that by operating fluidised beds at temperatures below 1000°C oxides of nitrogen emissions can be maintained at levels below a few hundred ppm by volume e.g. (3, 4, 23, 34, 35, 40, 41, 45, 55, 57, 62, 64, 76) when burning a wide variety of coals.

In an atmospheric FBC boiler Beacham (45) reports that NOx concentrations are a strong function of temperature, tending towards a plateau above a temperature of 850°C and remaining constant thereafter up to about 950°C. This suggests most strongly that the source of the NOx is fuel-bound nitrogen. Highley(64) reports that there does not appear a strong effect on NOx concentration with increases in excess air ratio.

These findings are supported by other work e.g. (129) and, since the fuels are of differing types, it is likely to be a reasonably general trend.

Emissions of NOx at elevated pressure appear to have a lesser dependence on bed temperature (64), and concentrations generally lower than would be expected from atmospheric combustion. However, the concentrations of NOx increase sharply with increases in excess air ratio. Ruth et al (76) report that, at 10 atm operating pressure, a threefold increase in NOx emission from 0.1 lb/million BTU at 5% excess
air to 0.3 lb/million BTU at 100% excess air. This effect is also reported by Highley (64). Thus careful control of excess air ratio is likely to be necessary for the pressurised RFB.

A second effect which has been observed is the increase in NO\textsubscript{x} emission at elevated pressure with increases in calcium/sulphur mole ratio (130). However, by maintaining the excess air level at a relatively low value, 15\%, NO concentrations were found to be below 300 ppm v/v when burning a bituminous coal having a 2.8\% sulphur content. With a bed temperature of about 900°C combustion efficiency was found to fall sharply with increasing fluidising velocity from about 97\% at 2ft/s to about 91\% at 5ft/s. A corresponding increase in NO concentration occurred from about 190 ppm v/v to about 275 ppm v/v.

Emissions of oxides of sulphur can be suppressed by the addition of sulphur receptors to the bed material. These receptors are normally calcium or magnesium oxides in the form of raw or calcined limestone or dolomite (e.g. 45). A wide body of data suggests that the typical calcium to sulphur mole ratios required to achieve greater than 90\% sulphur absorption lie between 2:1 and 3:1 depending upon the receptor type (45, 57, 123, 131). Other variables that have been found to influence sulphur absorption are bed temperature, receptor particle size and fluidising velocity (123, 131).

Bed temperature seems to have the most significant effect and it has been noted that absorption seems to be optimised by operating the fluidised bed combustor at a temperature between 800°C and 850°C (e.g. 132, 133). This fluidising is so consistent as to suggest that chemical, rather than physical, conditions control the sulphur
absorption with respect to temperature.

Haque et al (123), amongst others, have noted the dependence of sulphur absorption on fluidising velocity and receptor particle size. Both effects seem to be strongly dependent on receptor type and the only general comments that can be made are that desulphurisation tends to reduce with increasing fluidising velocity, indicating a residence time effect, and to peak at a given particle size, indicating an effect dependent on particle porosity and the rates of diffusion of SO₂ into the particle.

Combustion at elevated pressure shows that sulphur retention may be lower when using uncalcined limestone due to less calcination in the higher partial pressure of CO₂ (64). Dolomite and calcined limestone, on the other hand, retain sulphur slightly more effectively at pressure as compared with atmospheric conditions (64); this is possibly due to better gas/solids contacting.

Under pressurised conditions the variation of sulphur retention with temperature is quite different to that which occurs at atmospheric pressure (134, 135). The retention efficiency improved marginally as temperature was increased and, up to bed temperatures of 900°C, these was no recorded fall-off in retention efficiency. Further, when the fluidising velocity was increased these little fall in sulphur retention, provided that the calcium sulphur ratio was greater than 2:1 (134, 135). It was also reported (135) that there was little change in retention efficiency when increasing excess air ratio.
3.8.2 **Emissions of Particulates from Stationary Fluidised Beds**

The problem of particulates emissions from fluidised bed combustors is widely recognised in industry and extensive research is being carried out on methods to measure emission rates (see Appendix C) and on exhaust gas cleaning equipment.

The characterisation of particulates emissions is difficult because the physical design of a combustor has a strong influence on particle motions within the bed and particle dynamics in the freeboard. Thus industrial research is principally concentrated on particle capture rather than the prediction of loss rates from "ideal" combustors.

Particulate capture has been attempted by a number of conventional techniques (136) (cyclones, bag filters, precipitators and scrubbers), and efforts continue in the examination of new techniques (moving granular beds, high temperature ceramic fibre filters and high temperature precipitation).

Discussion of conventional means of gas cleaning can be found in standard texts (e.g. 21, 137) and has been used successfully on a variety of pilot and industrial scale FBC plants, most of which are fitted with one or more high efficiency cyclones (e.g. 48, 76). Henschel (131) states that, in 1980, particulate control measures adequate to meet the EPA New Source Performance Standard of $0.03 \text{ lb/10}^6 \text{ Btu}$ had yet to be demonstrated but suggested that conventional technology should eventually provide the solution for atmospheric combustion system. Henschel suggests that cyclone separators will need to be supplemented with precipitators or fabric filters.
Work on pilot scale plant has shown that ceramic filtration is a feasible method of particulate capture in high pressure, high temperature conditions (e.g. 138). These filters require regular back purging in order to clean the fibre material and maintain a satisfactorily low pressure drop. Tests have shown that ceramic fibres are capable of withstanding up to 50,000 cycles.

Another promising method is that of filtration by moving granular bed. Particulates are captured on the surface of large granules in cross flow or counter flow conditions. The use of a moving granular bed allows continuous cleaning of both gases and dirty bed material, eliminating the need for plant cycling and thus easing the flow control problems. Typical of this work are the projects reported by Yamamura and Terada (139) and Gutfinger et al (140). Submicron capture efficiencies greater than 99% have been reported (140) using filter bed thicknesses of approximately 60 cm and gas flow velocities of 33 cm/sec. Pressure losses were acceptable; between 45 and 80 cm water gauge.

Henschel (131) reports that satisfactory performance has been achieved using multiple cyclone trains on the Exxon Miniplant. Ceramic fibre filters, on the same plant, also gave encouraging results.

3.8.3 Emissions from Rotating Fluidised Bed Combustors

No systematic investigation of the emissions from RFB combustors has been reported in the available literature. The data available from the work of Demircan (40) and Metcalfe (41) suggest that emissions of NOx are very low (less than 20 ppm) when gaseous fuels are utilised.
NOx emissions when burning coal are reported by Demircan. He showed that a steady increase in NO emission between 700°C and 950°C bed temperature of about 90 ppm/100°C whilst the actual emission concentration reduced, by about 60 ppm, when the rotational speed was increased from 400 rpm to 750 rpm.

At a bed temperature of 850°C, at which optimum absorption of SO₂ occurs in conventional FBC, Demircan's data suggest NO levels between 170 and 240 ppm when burning coal with a 1.5% nitrogen content. Unfortunately, Demircan does not identify at what air to fuel ratios these data were gathered and thus it is difficult to compare them with data from conventional combustors.

The low levels of NOx emissions, reported for gas combustion, are, almost certainly, due to the short residence time of fluidising gases in the bed. Short residence times could be expected to give rise to moderately high levels of carbon monoxide in the exhaust gases. Data published by Demircan (40) show that CO emissions as high as 0.17% were measured when burning propane and as high as 0.86% when burning coal. Both of these levels are very high when compared with the emissions from conventional p.f. boiler plant, (ca. 100 ppm), (44).

Metcalfe (41) reports that the ratio of CO:(CO₂ + CO) increases rapidly as fluidisation index is increased when propane fuel was used. No CO was detectable at fluidisation indeces below 10, but, thereafter, the ratio increased rapidly to about 0.09 at a fluidisation index of 42. This clearly demonstrates the adverse effect of short residence time.
Metcalfe measured CO emissions when burning small coal particles (1mm diameter). Concentrations were, typically, 6% by volume. Whilst he does not discuss possible reasons for these high levels of CO, it seems likely that they are a result of using very shallow beds for coal combustion. Comparing Metcalfe's bed depth with those used by Demircan, it is seen that the gas residence time in Demircan's beds was about double that in Metcalfe's. It would appear that a bed depth of about 1.5 cm should be regarded as a minimum for the RFB combustor in order to ensure satisfactory CO burn-up when burning coal.

3.9 Mechanical Design Considerations

The literature allows the examination of two important mechanical design aspects of fluidised bed combustors, viz. the distributor design and the design of heat transfer tubing.

3.9.1 Distributor design in the RFB

A number of investigations have been done which examine the effects of distributor design on fluidisation characteristics (35,36,40,41,112,127). The information in (35),(36),(112) and (127) has shown that a distributor manufactured in the shape of a conic frustrum can aid the initial spreading of bed material during run-up and aid subsequent cross-flow particle mixing. However the information shown in (40) and (41) suggest that adequate run-up distribution and cross-flow mixing may also be achieved provided that, in the former case, some fluidisation air is provided to aid particle movement, and, in the latter case, fluidisation velocities are high enough to produce vigorous bubbling. The positions of the fuel feed nozzles may also be important.
in achieving good mixing. Metcalfe (41) further suggested that the maximum length to diameter ratio for the distributor for uniform fluidisation should be 1:2. Demircan (40), however, experienced no problems with L/D ratio of 1:1.

Two other important design considerations were discussed by Metcalfe. He suggests that the open area of the distributor be chosen such that, when using a premixed air and fuel gas, the mixture velocity through the distributor should be significantly larger than the flame propagation speed of the mixture in order to prevent burn-back into the plenum. Further, he suggests the distributor should be free to expand in the axial direction but that every effort should be made to achieve a gas-tight seal between the distributor and its mountings.

3.9.2 Heat transfer tubes

If the decision is made to fit heat transfer tubing, careful consideration must be given to tube sizing, position and packing density. These factors can affect, significantly, the operational characteristics of fluidised beds (eg.141). Briefly, the packing density and position of the tubing is important in that heat transfer rates to in-bed tubes vary as fluidisation velocities are changed. At low velocities, highest rates were obtained from tubes close to the distributor and vice-versa at high fluidisation velocities.

No effects of tube position on heat transfer coefficient were observed provided that the tubes were fully immersed in the bed. However, if some or all of the tubes were exposed due to reduction of bed expansion at low fluidising velocities, the heat transfer
coefficients were seen to fall rapidly.

Grewal et al (142) report some of the effects of distributor design on heat transfer coefficients measured on a single horizontal tube. They noted small, but reasonably consistent, changes in heat transfer coefficient when distributors having 7.7% and 37.5% open areas were used. At low fluidising velocities higher coefficients were measured when the larger open area distributor was used. This situation reversed as fluidising velocity increased and reversed yet again at yet higher fluidising velocities. They explained these effects in terms of bubble generation and solids motion. The larger open area distributor gave rise to more bubbles than the smaller open area distributor. At low velocities this aided solids motion and gave rise to high heat transfer coefficients. However, at moderate velocities bubble coalescence from the larger open area distributor reduced the solids movement and the heat transfer coefficient for the small open area distributor became the larger. At sufficiently high fluidising velocities, the heat transfer coefficient became dependent upon particle packing density, (1-E), which was somewhat higher for the large open area distributor, although why this should occur was not discussed.

3.10 Principal conclusions

3.10.1 Fluidisation characteristics

1) A number of correlations exist for $U_{mf}$ and it has been found that the correlation due to Wen and Yu gives adequate predictions for the RFB up to $Ga=7\times10^4$.

2) When the particle Reynolds number lies between 10 and 100, the
value of \( U_m \) is relatively insensitive to changes in bed temperature at constant pressure. This effect needs to be confirmed for the RFB.

iii) The characterisation of minimum operating condition may best be described by the fully fluidised condition rather than the incipient fluidised condition in both stationary and rotating beds.

3.10.2 Mixing characteristics

i) Satisfactory radial mixing will occur between inert and reactive particles in the RFB provided that the fluidisation index is equal to or greater than 2:1.

ii) There is conflicting data on crossflow mixing in the RFB. It is, therefore, prudent to distribute solid fuel particles across the entire bed surface in order to achieve adequate fuel distribution.

3.10.3 Elutriation

i) The effects on elutriation of combustor geometry need to be investigated in order to test theories put forward for elutriation mechanisms in the RFB.

3.10.4 Heat transfer

i) Heat transfer from bed to tubes in the RFB is proportional to the Galileo number and increases with increasing particle size and reducing bed depth.

ii) Heat transfer coefficients are, typically, in the range 50 to 250 W/m² in the RFB.
3.10.5 **Combustion of solid fuels**

i) Combustion efficiencies are normally high (ca. 90%) provided that there is adequate residence time for carbon burn-up.

ii) Fines recycle can be used to increase combustion efficiency to very high values (ca. 99%).

iii) The use of large solid fuel particles minimises carbon carry-over and may eliminate the need for fine recycle.

3.10.6 **Emissions**

i) Low emissions of NOx are normal with fluidised bed combustion, however, under pressurised conditions the excess air level can have a significant effect on NOx emission rate.

ii) Emissions of SO2 can be minimised by the addition of sulphur receptors (limestone or dolomite) to the fluidised bed and by controlling the combustion temperature to between 800 and 900°C. This latter aspect can be relaxed at elevated pressure as there appears to be little change of absorption efficiency between 800 and 975°C at high pressure.
Fig 3.2 Typical Fluidisation Characteristic
FIG 3.3 RFB PRESSURE DROP CHARACTERISTICS
(from Deinken (97))
FIG 3.4  TYPICAL RFB COMBUSTION ENVELOPE

USING PROPANE FUEL (from Metcalfes[41])
CHAPTER 4

COMBUSTOR DESIGN

4.1 INTRODUCTION

The mechanical design of the rotating fluidised bed combustor is as important to its success as, for example, internal fluid mechanics. In order to be a feasible alternative to other types, the combustor must achieve higher intensity of combustion, greater flexibility of operation and rapidity of response, with minimum extra complication and equal reliability and at a competitive cost. The principal mechanical difficulty for the RFB is the fact that a relatively large mass of metal and bed material, approximately 16,000 kg for a 1000 MW design (15), must be made to revolve at speeds in the region of 150 rpm in a high temperature environment. This problem is additional to those normally associated with controlled combustion within a stationary fluidised bed. Performance of the RFB therefore, must be shown to be significantly better than conventional designs of high performance stationary fluidised bed. In an exploratory project such as reported here, the combustor design should be simple, robust and easily modified whilst reflecting, as closely as possible, the specification for the envisaged the full scale combustor.

Metcalfe (41), whose work was the immediate forerunner of the current study, itemised the basic design choices open to the designer. Included in the options, are such features as particle material type, particle
mean diameter and combustor length to diameter ratio. The interaction of all the design features will govern the overall performance of the combustor and it would not be possible to produce a single design that would suit all applications. The aim of the design phase should be to produce a combustor design that will allow exploration of some of the design alternatives and their effects, in isolation, on the combustor performance. It should also lead to a combustor with which experiments can be readily performed. The results obtained from the experimental programme may produce empirical rules from which larger combustors might be designed with reasonable confidence. However, scale-up can be a major problem (80). The experimental data can also be compared with predictions from theoretical models and indicate how the latter need be modified to account for interactions.

The information required in the optimisation of the combustor geometry can be itemised as follows:-

1) Length to diameter ratio for the distributor

2) Exhaust port diameter to distributor diameter (or bed free surface diameter) ratio.

3) Freeboard shape

These features were examined as part of the experimental programme.

The logical starting point for the design was to use the same configuration as a combustor operated satisfactorily in the past and upon which an optimisation study could be readily undertaken. For the
current project the design developed by Metcalfe (41) seemed to be a reasonable starting point. This design had been shown to be capable of stable combustion of gas and coal, although poor combustion efficiencies were associated with coal firing. Metcalfe's design was simple and relatively easy to modify.

The important items involved in the current design are examined individually, bearing in mind that pressurised operation was the ultimate aim.

A general assembly engineering drawing can be found in Appendix B along with detail drawings of those items of particular importance to the combustor design. The latter include the exhaust port air-cooled chimney, the distributors and the distributor upper end plate.

Items such as the plenum, plenum end plates and combustor drive shaft and bearings follow the designs due to Metcalfe (41) and had similar dimensions, but were manufactured to specifications applicable to pressure vessels. The detail designs of these items are not important to scope of the research and thus are not included in this thesis.

4.2 Plenum Chamber

Two basic plenum designs are possible. Firstly, the plenum may be attached to the drive shaft and rotate with the drive shaft/distributor assembly, Fig. 4.1. This approach eases the problems of sealing the fluidising airflow paths but was thought best suited to small units, where the airflow is not restricted by the internal diameter of the drive.
shaft. The fixed plenum would act as the pressure vessel for elevated pressure operation and is a necessary safety feature at all times.

The second approach involves a plenum that is fixed to some support structure as shown in Fig. 4.2. The combustor rotates inside this fixed plenum and fluidising air is introduced through the plenum wall or via the drive shaft housing assembly.

This approach only involves pressure seals at the drive shaft, "A", and at the exit port, "B", and therefore simplifies the overall design. If reliable sealing can be achieved this design results in a more practical approach for a full size combustor. Metcalfe also showed that a more uniform air distribution was obtained with this design.

4.3 Distributor Design

There exists a large variety of designs for fluidised bed distributors. Descriptions and characteristics for the more common types can be found in texts (21, 80). Much effort has been expended investigating the performance of these distributors and the characteristics of the fluidisation that each produces. Whitehead (in 80) discusses the effects of scale-up on the performance of a fluidised bed reactor and points out the extreme uncertainty involved in extrapolating results obtained from small scale rigs, especially when these results have been obtained on a rig having a significantly different distributor design. Fig. 4.3 shows some of the common forms of large scale distributor design. Each has advantages and disadvantages and it is certainly not possible to say that there is a 'best' option for all applications.
The most common forms of distributor found in small experimental rigs are porous ceramic or sintered metal tiles and pierced metal sheet. These materials are often commercially available in convenient dimensions covering a variety of physical characteristics eg open area, pressure loss etc.

Porous ceramics have poor thermal shock resistance and may crack under the severe temperature gradient between plenum temperature and combustion temperature. However, the large internal area may allow considerable cooling by the fluidising gas and may act as a flame trap.

Sintered metal tiles offer the prospect of very good internal cooling because of the very large surface area of the interstices. Also, they are available in a wide variety of materials, stainless steel in particular, and may be fabricated and machined as a normal metal sheet. Welding, however, is a problem that requires special techniques in order to obtain satisfactory results.

The principal deficiency associated with ceramic and sintered materials is that the normal application for these products is filtration, often at modest pressures and temperatures, so that many products lack the physical strength desirable for a combustor.

Furthermore, in order to satisfy the size requirements of the planned RFBC the porous cylinder would have to be specially manufactured, as standard filter items available had a maximum diameter of about 6". The resulting, very expensive, distributor was considered inappropriate, especially as a full scale distributor was expected to follow one of the designs shown in Fig 4.3 b, c, or d.
Pierced sheet offers a method of distributor construction applicable to both small and large scale. Metcalfe (41) and Demican (40) both found that a pierced metal sheet distributor would give satisfactory fluidisation and long life provided that airflow rates are high enough to give adequate cooling from the relatively small surface area.

This type of distributor was adopted as the basic design for the current study. The actual material used was a commercial pierced stainless steel sheet manufactured by Hein-Lehmann A.G., Dusseldorf, F.D.R., under the trade name "Conidur". Details of the materials used can be found in Appendix B.

The distributor dimensions were similar to those used by Metcalfe, ie, 205mm top diameter and 80mm axial length, the diameter being limited by the plenum internal diameter.

The distributor area was chosen such that, with the available compressed air facilities, gas velocities in the bed of up to about 7 m/s would be possible at maximum available pressure and air mass flow rate. Information relating to the air compressor can be found in Appendix C.

Having determined the maximum distributor area it was possible to determine the distributor axial length since the diameter was determined by the plenum inside diameter. In fact, the dimensions of Metcalfe's distributors very nearly satisfied the requirements for gas velocities of about 7 m/s at maximum operating flow conditions and thus, for the sake of continuity very similar dimensions were used.
Work by Levy et al. (107) suggested that a distributor manufactured as a conic frustrum of small included angle, $4^\circ$ – $8^\circ$, aided initial bed distribution during start-up. Based on this information the initial distributor was fabricated as a conic frustrum having an included angle of $4^\circ$. This shape of distributor is more difficult to fabricate than a right cylinder and its adoption depends on it giving repeatable improvements of some aspect of combustor performance. For this reason, a cylindrical distributor of similar dimensions was manufactured from the same material as the conic distributor and was used in a series of comparative tests at atmospheric pressure and temperature. These tests showed that the conic distributor was not a significant improvement over the cylindrical distributor and, in fact, might be the source of detrimental operating characteristics. Thus right cylindrical distributors were designed and manufactured for the combustion experiments.

Initially, the distributors were manufactured from "Conidur" material, the overall dimensions being the same as those of the cylindrical distributor used in the atmospheric test work. The "Conidur" material used in the combustion trials had smaller perforations and smaller open areas than those used for the atmospheric work. Details of the materials used can be found in Appendix B. The smaller perforations allowed the use of smaller bed particles (and thus improved heat transfer properties) and the smaller open area would give a distributor pressure loss characteristic that would result in the distributor pressure loss being equal to or greater than the bed pressure loss over a reasonable range of operating conditions.

The pressure loss constraint is necessary in order to help maintain
gas flow distribution in the event of bed instabilities occurring and follows recommendations from earlier work (41).

Experience with these distributors showed that distributor cooling could be difficult during the start-up phase: this is discussed fully in Chapter 7. Following a series of tests using a conventional fluidised bed, a "standpipe" type of distributor was designed. Further details of this distributor are given in Chapter 7.

Operational problems during start-up prevented successful combustion trials with the stand-pipe distributor and it was found necessary to procure a porous stainless steel distributor. Details of the porous material used, porous stainless steel, PSS grade F, from Pall Filtration Limited, Portsmouth, can be found in Appendix B and a detail drawing of the distributor is included in Chapter 7.

4.4 Distributor Upper End Plate

This plate forms one of the axial boundaries of the combustion chamber and may be used for a number functions. Exhaust gases from the combustion zone discharge from the combustor via a port, or ports, cut in the plate. Removal of excess bed material may be achieved by weir action through orifices cut in the plate. Finally, a pressure seal may be incorporated at the outer edge of the plate to effect sealing of the plenum chamber from the exhaust ducting.

The design, shown in Appendix B, incorporates a central circular exhaust port whose diameter was determined from the results of the
atmospheric flow studies.

The outside diameter of the plate was such that a seal could be incorporated at the outer edge if the need arose, however this option was not utilised

Weirig orifices were not included as data on which to estimate rates of bed material build-up and optimum bed depth would not be available until after combustion trials has been undertaken. This data is essential for estimating the required size and position of weirs.

Due to the high temperatures, ca. 1200°C, that this plate was expected to encounter, the material chosen for its construction was heat resisting stainless steel grade 321. This material has the added virtue of maintaining a surface finish adequate for sealing purposes even when exposed to high temperatures and severe temperature gradients.

The actual metal temperature reached could only be predicted from rather uncertain modelling of the heat transfer. The model is described in Appendix A where estimates of metal temperature are shown for the cooled chimney described in section 4.5

4.5 Exhaust Port Chimney

The exhaust port chimney performs a vital function in the operation of the current combustor as it forms the 'hot end' pressure seal. Metcalfe (41) used a large water cooled seal which, whilst effective, was cumbersome and restricted access to the combustor.
The seal proposed for the current combustor consisted of a short metal cylinder machined at one end to attach to the distributor end plate shown in Fig. 4.4. A labyrinth was located on the outer surface of the opposite end. The labyrinth was sized to run inside a short stainless steel containment tube that was fixed to the exhaust ducting. The chimney and labyrinth seal were cooled by a film of cold air generated by a shrouded ring of holes located at the upstream end of the chimney tube. Air for this film could be bled from the plenum air flow or be supplied from a separate source.

A labyrinth seal is normally designed with a very small running clearance, typically 0.127 - 0.38mm (0.005 - 0.015 inches). This, however, can only be done when the running temperatures of the seal can be controlled within known limits. For the current design, running temperatures would depend upon combustor operating conditions, film cooling mass flow rate and the rate of change of film cooling effectiveness with respect to axial displacement from the film cooling ring.

Combustor operating conditions can be estimated moderately well. However, the required cooling air mass flow rates can only be estimated when the cooling effectiveness is known. The latter is critically dependent upon cooling ring design criteria, which are, generally speaking, patented and the subject of strict commercial secrecy. In designing the cooling ring, the author used his own experience of "order of magnitude" cooling flow rates and approximate cooling ring designs. It should be stressed that no proprietary information was used in the design.
As a starting point, cooling mass flow rates of between $0.17 \times 10^{-6}$ and $0.667 \times 10^{-6}$ kg/s/cm/cm$^2$ were assumed at an initial temperature of 20$^\circ$C. The pressure loss across the cooling ring was set at 0.4% at start-up and an operating maximum of 5.5% with a downstream pressure of 101325 N/m$^2$, these values resulted in an estimated flow area of $28.27 \times 10^{-6}$ cm$^2$, and with $C_D = 0.8$, this was incorporated in the cooling ring as a single row of 36 holes of 1mm diameter.

The running temperature of the chimney with an exhaust temperature of 1000$^\circ$C was estimated to be approximately 700$^\circ$C when cooled by air at 20$^\circ$C. The estimation procedure can be found in Appendix A. Making due allowance for thermal expansion the cold running clearance was set at 0.69 mm. The running clearance at temperature was expected to be about 0.005 mm.

The labyrinth itself is rather crude because accurate design of the labyrinth chambers was made secondary to strength and resistance to thermal distortion and abrasion. It was felt necessary to demonstrate the feasibility of using a labyrinth seal before incorporating a refined design. A unit of good mechanical integrity was necessary in order to withstand any rough treatment encountered during the initial hot trials where adjustment of cooling mass flow rates might be found to be necessary in order to maintain satisfactory running clearance.

The labyrinth was designed to have five annular chambers, as shown in Fig. 4.4 each 3mm square in section separated by walls 1mm thick. Grade 321 heat resisting stainless steel was chosen for the manufacture of both the chimney and the labyrinth outer containment. The material has
a low coefficient of expansion and its hardness gives good abrasion resistance.

The initial design of film-cooled chimney is shown in Fig. 4.5. The labyrinth seal and the ring of 1mm diameter cooling holes are clearly visible.

4.6 Coal Injector

The coal injector used in the combustor is shown in Fig 4.6. This, in common with other items expected to encounter high temperatures, was manufactured from grade 321 stainless steel. The development of the design is discussed later in the text.

For convenience the injector was mounted in the exhaust ducting and entered the combustor via the exhaust chimney. This approach would probably not be used in a practical combustor as it may severely limit the injector life. An alternative position for the injector is through the combustor drive shaft. In this way only the end section of the injector is subjected to high temperature. However, the added complication of this approach would only be justified if a significant improvement in component life was achieved.

4.7 Combustor Taper Rings

Early in the combustion trials it was found that start-up was difficult due to a lack of control of flame stability. Metcalfe (41)
and Demircan (12) had found that stabilisation of a flame of premixed propane/air mixture on the free surface of the bed could be achieved solely by control of the fuel: air ratio although slightly different start-up techniques were adopted in the two cases. Neither of the techniques developed were found to be satisfactory for the current combustor; the reasons are discussed fully in Chapter 7. Essentially, it was thought that the acceleration of the fluidising air flow, as it travelled from the distributor radially inwards towards the exhaust port, was such that the gas velocity in the freeboard was sufficient for the flame to be blown out of the combustor. In order to overcome this problem, two tapered rings were designed. These rings were fitted inside the distributor and produced a divergent cross sectional shape in the combustion zone as shown in Fig. 4.7. The divergence angle of 50° was chosen so that, 15mm downstream of the distributor, the effective gas flow area was approximately 20% greater than that just downstream of the distributor.

The effects of this area change were two-fold. First, it ensured that in the range of air flow rates used for start-up, the fuel:air mixture velocity at the bed surface was significantly less than the natural flame propagation speed for the applied fuel/air ratio and upstream temperature. This ensured that once the mixture ignited the flame would tend to move radially outward, rather than inward, and stabilise on the bed surface.

The second effect was that the reduction in flow velocity 'matched' to some extent, the reduction in centrifugal acceleration through the bed. This helped to minimise the tendency of the bed to fluidise in layers starting at the free surface and reduced the the transition from
packed to fully fluidised bed.
FIG 4.2 FIXED PLENUM RFB COMBUSTOR
ARROWS INDICATE GROSS AIRFLOW PATHS

FIG 4.3 SOME COMMON DISTRIBUTOR DESIGNS
FIG 4.4 FILM COOLED CHIMNEY AND LABYRINTH SEAL
FIG 4.6 PNEUMATIC COAL FEED HEAD
FIG 4.7 Combustor with shaper rings installed
CHAPTER 5

THE RIG FACILITY

5.1 Introduction

The experimental programme was conducted in two phases:

1. Operation at atmospheric temperature and,

2. Operation at elevated temperature.

Three different rotating fluidised bed units were utilised during the atmospheric temperature work:

(a) a perspex cold flow model,

(b) a combustor modified for cold flow work and,

(c) the combustor described in Chapter 4.

The two cold flow RFBs were designed and built by Metcalfe(41). Only the combustor described in Chapter 4 was used for high temperature operation. The two cold flow rigs are described in section 6.2 of Chapter 6 in which the cold flow modelling work is presented.
For the combustion evaluation trials, four major pieces of equipment were deemed necessary:

i) A gas analysis system capable of measuring the concentration in the exhaust gases of oxygen, carbon dioxide, carbon monoxide, unburnt hydrocarbons, oxides of nitrogen and oxides of sulphur.

ii) An exhaust gas cleaning system capable of extracting elutriated inert bed material, unburnt fuel and fly-ash and, perhaps, to remove some of the acidic gas components, $SO_2$ and $SO_3$.

iii) A pressure let-down and exhaust silencing system.

iv) A solids handling system capable of transferring solids from atmospheric pressure to the combustor, operating at elevated pressure, at a controlled rate.

Description of the four systems, and where appropriate, consideration of practical installations, follows under separate headings.

In addition to the aforementioned major items, facilities were needed for measuring gas temperatures, pressures and flow rates through the process. These are discussed briefly at the end of this chapter.

References sited in this chapter are included, separate to the main references, after the last section of this chapter.
5.2 Gas Analysis System

The major emissions from stationary combustion sources are particulates, oxides of sulphur, SOx, and oxides of nitrogen, NOx (1). Emissions of carbon monoxide, CO, and unburnt hydrocarbons, UHC, are, generally speaking, small in comparison with particulates, SOx and NOx and are not, to the author's knowledge, subject to any regulatory measures at the time of writing.

Low emissions of CO and UHC result from high combustion efficiency. Conversely, the levels of CO and UHC are a measure of the combustion inefficiency and, therefore, during the development of a new combustion system, measurement of these two components is vital.

Emissions of oxides of nitrogen and sulphur are both the subject of stringent regulations in the United States (2) and Japan and sulphurous emissions must become regulated more strictly in the United Kingdom and Europe as it becomes necessary to utilise fuels having increasingly high sulphur content.

In order to assess the performance of the rotating fluidised bed combustor completely, a comprehensive gas analysis facility was required. A number of gas analysers were required, as were a cooled gas sampling probe, a heated filter, and a heated transfer line. The gas analyser types and the units selected are discussed in Appendix C.

5.2.1 General Requirements

The general requirements of the gas analysis system are discussed in Appendix D. A gas analysis system, recommended by the Environmental
Protection Agency for gas turbine certification in the USA, is shown in Fig. 5.1. The system installed in the RFB rig facility was modelled on this system.

5.2.2. **Gas Sampling Probe Design**

Two water-cooled probe designs are shown in Fig. 5.2. The first shows individual quenching tubes taking the sampled gas from the sampling points to a mixing tube. In the second design the sampled gas passes through orifices into a combined quenching and mixing chamber. In both cases, the water flow rate and temperature may be adjusted independently so that the sampled gas is quenched rapidly to the required temperature.

The first design has the advantage of rapid sample quenching, but large numbers of joints make the design prone to leaks. Further, the hypodermic tubing used to make the quench tubes is easily blocked by particulate matter. The second design has the advantage of simplicity and maintainability; should any of the orifices become blocked, it is an easy matter to remove the blocking material. Quenching is not as effective and results in slightly less controllable sample temperatures. However, it was the author's experience that variations of sample temperature over quite a wide range, 120°C – 250°C, has no significant effect on the measured concentrations provided that gas temperatures do not fall below the acid dew point.

Both probe designs were manufactured and evaluated. The first design was found to leak at the brazed joints and was abandoned. The second design appeared to operate satisfactorily although it was never tested in conditions of high temperature and pressure.
Alternative designs may be found in the literature (4) but are intended for much larger scale apparatus. Scaling down the designs was not practicable as many critical dimensions and ratios could not be maintained.

5.2.3 Probe Cooling System

In order that the gas sample be quenched to the desired temperature within the probe body and also maintain satisfactory mechanical integrity of the probe itself, water cooling was necessary. The cooling system is discussed in Appendix D.

Variations of both water flow and temperature were possible so that optimum water conditions could be achieved. An electric heating element was included to preheat the cooling water before commencing a test; thus reducing the heat-up time. Cooling coils were incorporated to prevent the cooling water boiling.

5.2.4 Sample Transfer Line

Ideally, it is necessary to transfer the gas sample to the analyser quickly through an inert transfer line. In order that the line does not become blocked by particulate matter a filter may be incorporated in the sampling probe or at the probe outlet.

The heated sample filter, shown in Fig. 5.1, was manufactured and supplied by Analysis Automation Ltd. of Eynsham, Oxfordshire. This filter comprises a stainless steel body, electrically heated, housing a replaceable paper filter. The whole unit could be heated to 100°C which
was sufficient to prevent any significant temperature loss in the gas sample because of the very short residence time of the gas in the filter body.

The heated transfer line was a Thermo-Electron Model 212 unit capable of satisfactory operation up to 10 atm. pressure and 400°C. The line bore was 3/16" which meant that, in order to satisfy a 2 second residence time requirement (EPA (5)) in a line 25 feet long, the minimum gas sampling rate had to be 4.1 l/min. The flow rates recommended by the EPA (5) result in a residence time of approximately 0.20 seconds. Thus, in the temperature range desired for the gas sample, any changes in the gas composition between the sampling point and the analyses should be insignificant.

5.2.5 Sampling Station and Frequency

The gas analysis system was designed so that continuous analysis of exhaust gases drawn from a station close to the combustor exit could be performed. Choosing a station close to the combustor exit would allow inference to be drawn about the effects of operating conditions (ie. fluidisation index, combustion temperature, inerts and fuel properties) on the combustion characteristics. Sampling from a station well downstream allows post-combustion chamber reactions to alter gas concentrations and thus produce unrepresentative analysis results. A further point to bear in mind is that the use of continuous analysis allows time averaged results to be obtained.
5.3 Exhaust Gas Cleaning

Exhaust gas cleaning was installed for two reasons:- Firstly, in order that some assessment might be made of carried over solids which were expected to be in the size range 0.5mm and comprise a mixture of ash and unburnt carbon. In addition to the solids, the exhaust gas was expected to contain up to 3% SO₂ (at stoichioimetric conditions) when burning anthracite coal. Some aspects of gas cleaning are discussed in Appendix D.

It was found that the quoted costs of a high efficiency cyclone capable of operating at the high temperatures and pressures planned in the test programme were, in the main, greater than funds available. A venturi scrubber, offered by ESMIL Ltd., of St. Neots, Cambs., was within budget limits and was expected to satisfy, as reasonably as possible, the cleaning and safety requirements. The scrubber layout is shown in Fig. 5.3.

The collection efficiency of the scrubber was quoted as 95% for particles of diameter greater than 1 micron.

A scrubbing water treatment plant also had to be provided and is described in Appendix D.

5.4 Pressure Let-Down and Exhaust Silencing

5.4.1 Pressure Let-Down

The choice of pressure let down method is a relatively simple one
between a series of fixed nozzles or a valve. A nozzle is simple and robust but only allows operation at a fixed ratio of gas mass flow rate to gas pressure, at a given flow temperature (assuming that the flow is choked). Thus, in order to obtain experimental data for a range of mass flow rates and a range of gas pressures a number of nozzles of different diameters are required.

On the other hand, a valve that is required to operate at both elevated temperatures and pressures will be very expensive but allows operation over wide ranges of mass flow rates and pressures simultaneously. The choice then, based on priorities, is between cheapness and durability or experimental flexibility.

In the current work, much effort was put into devising a cheap but temperature resistant valve. Water cooled and aircooled designs were considered but rejected because of the complexity of the resulting designs.

Several uncooled butterfly valves, constructed from heat resistant alloys, were offered by Woodhall-Duckham Ltd., but satisfactory delivery dates could not be met. Consequently, a range of fixed interchangeable nozzles were designed. These were manufactured from EN6 steel and heat treated to give some degree of wear resistance. Erosion by slotted material was expected to occur at the nozzle inner surface if combustion experiments were conducted without the gas clean up system installed.

5.4.2 Exhaust Silencing

Apart from achieving satisfactory life from the exhaust nozzles, the
main problem associated with high pressure experiments is high efflux noise. When the pressure ratio across a nozzle is greater than a critical value, the flow velocity at the nozzle throat (plane of minimum area) is sonic. Downstream of the throat the flow diverges and decelerates. Thus if the exhaust flow is vented into still ambient air in an experimental test cell then, at the boundary between the exhaust flow and the atmosphere severe shear rates will occur. The high shear rates between the two gas phases cause turbulent pockets of gas to be formed and it is the rapid evolution and destruction of these "eddies" that gives rise to the high level of noise associated with very high speed flows (6). The energy dissipated by a jet in this way can be estimated from the equation due to Lighthill (7):

\[ W = kD^2 \frac{\rho_j^2 U_j^8}{\rho_a A^5} \text{ watts} \]

where \( k = \) constant (Lighthill quotes \( 10^4 \) for a turbulent jet)

\( D = \) orifice diameter
\( \rho_j = \) gas density in the jet
\( \rho_a = \) ambient air density
\( A = \) sonic velocity at the critical pressure ratio
\( U_j = \) velocity of the jet
\( \text{SNL} = 10 \log_{10} (W) \) (\( \text{SNL} = 10^{-12} \))

For a 10mm diameter nozzle exhausting air, at ambient temperature, the energy dissipated from choked (ie. sonic) flow is 0.51 W or 117dB (rel to \( 10^{-12} W \)).

Further investigations and the selected silencer are discussed in
Appendix D. The type of silencer finally chosen for the rig was an ejector detuner, built to design criteria supplied by Cullum Detuners Ltd., Derby.

5.5 Solids Handling System

The basic objective for this system, as stated in the introductory section, is to feed solid fuel and bed additives, at a controlled rate, into the pressurised combustor. A number of feeding mechanisms were considered. The choice of feed mechanism is dependent upon the method chosen for the final introduction of the fuel and additives to the bed. During the early stages of the project gravity feeding was thought to be a possible candidate for the final injection phase as Demircan (40) had utilised this method successfully. Metcalfe (41) had, however, found it necessary to apply pneumatic lean phase transport in order to ensure satisfactory fuel distribution. Devices like spreaders and stokers were not considered practical for application in this project: most are intended for operation at atmospheric pressure throughout the transport process. The principal types are described in Table 5.1. Some of these devices can be adapted for transporting materials against an adverse pressure gradient and most can perform a metering function as well the transport function. This dual capability greatly eases the problem of solids feeding.

Consideration of Table 5.1 leads to the following conclusions:-

1) Two strategies may be followed in transporting from a low pressure to a high pressure.
ii) One may attempt to feed directly against the pressure ie the feeder must develop an internal pressure gradient.

iii) One may employ an air lock as an intermediate stage between low and high pressure.

The feed system adopted in this particular project utilised the strategy (b) but some consideration was given to (a) as it offered the prospect of a simpler and, perhaps, more reliable system.

The simplest and most direct method of achieving a pressurised solids feed system is to use a fully enclosed and pressurised hopper charged with sufficient material to satisfy test requirements. The problems then reduce to accurate, repeatable metering of the solids. This does not satisfy the requirements of a practical system, of course, but is adequate for initial test purposes. It was considered that devising a system that would form the basis of a practical installation would be of more benefit to the project.

5.5.1 Metering and Transfer Methods

The various devices considered for metering coal of small particle size are shown in Table 5.2.

Generally, a wide operating range is usually possible if careful choice of the metering device and simple control is made. However, it is only the vibrator which has no moving parts in contact with the crushed coal. This is a very important feature as crushed coal is very abrasive, especially when wet.
The rotary valve offers the prospect of fulfilling both metering and transfer functions but the limit on pressure difference across valves available in the UK was about 30lbf/sq.in and so several valves would be necessary in order to meet the desired pressure difference of, approx., 85 lbf/sq.in. The resulting valve network would, inevitably, be very difficult to synchronise and have too high a probability of mechanical failure.

The simplest choice available for the LP to HP transfer function is that of a pair of gas tight valves in series. These may be operated sequentially, manually or automatically, in order to produce an airlock between a hopper at atmospheric pressure and one at the combustor operating pressure. Various linear action valves are available, designed specifically for use in powder handling. However, there did not appear to be any commercially available system of a size small enough for laboratory use.

Appendix D describes the considerations made before the final choice of feeding system was made and also describes the installed system in detail.

5.5.2 Installation and Operation

The final installation of the solids handling system is shown in Fig. 5.4. Building the various components on a common vertical axis made access to the atmospheric hopper a little difficult but this disadvantage was offset by the convenience of gravity feed through to the pressurised hopper. Further, this installation minimised the floor space requirement.
Operation of the automatic locking system was, normally, very reliable. However, there were occasions when one or other of the ball valves would remain open while the other cycled. This problem could have been due to the control circuit output relays sticking but was more likely due to alignment inaccuracies between the pneumatic cylinders and ball valves.

5.6 Rig Controls

In order to monitor the operation of the combustor a number of items were required. System variables to be monitored were temperatures, pressures, air and fuel gas flow rates, combustor rotational speed and solid fuel flow rate.

Temperatures throughout the system were measured using chromel-almunel thermocouples connected to electronic digital thermometers. Two thermometer were used, a Comark 3000 and a Comark 5000, the latter being connected to a temperature trip switch which interrupted fuel feed in the event of excessive bed temperature occurring.

Pressures were measured using both manometers and Budenberg 1000 kPa Standard Test Gauges. The manometers could be isolated using stop valves in the event of pressures exceeding the manometer upper limit of measurement. By the use of multi-position distribution valves eight pressure tappings in the system could be monitored by using just one pressure gauge and two manometers (one water and one mercury). Two other test gauges were used for continuous monitoring of the plenum pressure and the pressure upstream of the main airflow orifice. Pressure differences across the airflow measurement orifices were
measured using dedicated manometers (water and mercury). This allowed continuous checking of the airflows through a test.

Airflow rates were measured using orifices designed to BS1042 and also with various sizes of rotameter. The main combustion airflow was measured using a 2\" or 2 1/2\" orifice in a 3\" N.B. line. Bypass airflow for cooling of the labyrinth seal was measured using a 1/4\" orifice mounted in a 1\" N.B. line. The airflow used as pneumatic transport air for the coal fuel was measured using a GEC-ELLIOT CONTROLS 24 K rotameter and the airflow to the pilot burner used in the latter stages of the project was measured with an 18A rotameter.

All propane flows were measured using rotameters manufactured by G.E.C. Elliot Controls as follows:– pilot gas: type 7A; main combustion gas types 14A and 18A. All measurements were corrected for temperature, pressure and gas molecular weight.

Combustor rotational speed was monitored with a Comark 2101 Tachometer linked to an optical sensor mounted on the combustor support frame and viewing a striped disk attached to the drive shaft.

Solids flow rate was controlled by varying the frequency and amplitude of the electrical power supply to the coal feeder. A comprehensive calibration showed repeatable feed control of \( \pm 2\% \) on mass flow.
References


7. LIGHTHILL J. (in ref. 6).
### TABLE 5.1

**SOLIDS TRANSPORT DEVICES.**

<table>
<thead>
<tr>
<th>DEVICE</th>
<th>ADVANTAGES</th>
<th>DISADVANTAGES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mechanical Valves</td>
<td></td>
<td>At least two valves required to operate sequentially.</td>
</tr>
<tr>
<td>a) Slide/Gate/ Knife valves</td>
<td>Wide range of sizes in a variety of materials</td>
<td>Prone to clogging and erosion.</td>
</tr>
<tr>
<td>b) Ball valves</td>
<td>Wide range of sizes self cleaning. Cheap</td>
<td>Expensive when automated.</td>
</tr>
<tr>
<td>c) Pneumatic in-line valves</td>
<td>Completely free passage to fuel flow</td>
<td>Expensive (£100 per valve). Pressure range may be limited</td>
</tr>
<tr>
<td>Rotary Valve (Blowing Seal)</td>
<td>Combined transfer and metering functions</td>
<td>Expensive. Erosion of valve materials. Limited pressure ratio</td>
</tr>
<tr>
<td></td>
<td></td>
<td>capability ca. 30 lb psi max.</td>
</tr>
<tr>
<td>DEVICE</td>
<td>ADVANTAGES</td>
<td>DISADVANTAGES</td>
</tr>
<tr>
<td>-----------------</td>
<td>---------------------------------</td>
<td>-----------------------------------</td>
</tr>
<tr>
<td>Plastic Extrusion</td>
<td>Combined LP to HP transfer and metering function</td>
<td>No commercial units available.</td>
</tr>
<tr>
<td>Centrifugal feeding</td>
<td>May operate against very high pressure ratios</td>
<td>No commercial units available. Very complicated design. High rotational speed necessitates special safety procedures.</td>
</tr>
<tr>
<td>DEVICE</td>
<td>POWER SOURCE</td>
<td>ADVANTAGES</td>
</tr>
<tr>
<td>-----------------</td>
<td>------------------</td>
<td>-----------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Vibrator (Enclosed tray or tube)</td>
<td>Electricity or Compressed air</td>
<td>No moving parts. Fuel flow totally enclosed - no dust problem. Wide range of operation. Simple control of flow rate via frequency and amplitude modulation. cheap.</td>
</tr>
<tr>
<td>Screw Feeder</td>
<td>Electricity</td>
<td>Wide range of operation and simple control via control of screw speed. Fuel flow totally enclosed. Manufacturers claim repeatable results over very wide range of flow rates.</td>
</tr>
<tr>
<td>DEVICE</td>
<td>POWER SOURCE</td>
<td>ADVANTAGES</td>
</tr>
<tr>
<td>--------------</td>
<td>--------------</td>
<td>-----------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Rotary Value (Blowing Seal)</td>
<td>Electricity</td>
<td>Wide range of operation and simple control via control of valve speed. Fuel flow totally enclosed. May provide limited pressure seal - 30 p.s.i per unit.</td>
</tr>
</tbody>
</table>
FIG 5.1 GAS SAMPLING AND ANALYSIS SYSTEM

KEY
FID - Flame Ionisation Detector
NOX - Chemiluminescence Detector
DUAL IRGA - Infra-Red Gas Analyser
Z,S - Zero and Span Calibration gases
R - Chart recorder
E,T - Heating element and thermostat

- Unburnt Hydrocarbons
- Oxides of Nitrogen
- Oxides of Carbon
a) INITIAL DESIGN

COOLING WATER INLET

SAMPLE OUT

COOLING WATER OUTLET

1 mm BORE HYPODERMIC TUBES

b) IMPROVED DESIGN

COOLING WATER OUTLET

INLET

SAMPLE OUT

1 mm DIAMETER HOLES

FIG 5.2 EXHAUST-GAS SAMPLING PROBE DESIGNS
FIG 5.3 VENTURI SCRUBBER INSTALLATION
FIG 5.4 COAL FEED INSTALLATION
CHAPTER 6

FLUIDISATION AND RELATED EXPERIMENTS CONDUCTED AT

ATMOSPHERIC TEMPERATURE AND PRESSURE

6.1 Introduction

Early investigations, at atmospheric temperature and pressure, demonstrated that rotating fluidised beds display the same fluidisation characteristics as those found in conventional fluidised beds. This work (eg. 31) showed that the bed pressure loss curve has the same features as those found in conventional beds with the fluidiseo pressure loss similar to the bed weight per unit area. The small RFB rigs in which some of these early studies were conducted were not realistic representations of a combustor geometry but showed that centrifugal fluidisation was feasible. Recent work (eg. 34, 35) has been performed in more realistic, laboratory scale, combustor having mechanical designs that could be translated into full scale combustion hardware. The information obtained from these models is, therefore, much more useful during the design stage of a new RFB combustor.

Metcalfe (41), using combustor models having no cooling tubes within the bed, studied fluidisation and mixing characteristics of very thin beds (typically 10 – 15mm thick) of small silica sand particles, 90 – 180 microns and 180 – 250 microns diameter. Demircan (40) studied fluidisation and mixing of fairly thick beds (typically 3 – 5 cm thick)
of moderately large silica sand particles, approximately 1mm diameter, in model combustors having simulated cooling tubes mounted within the bed. Shakespeare et al (36) and Kroger et al (112), on the other hand, studied fluidisation and mixing in thick beds (5cm thick) having no simulated cooling tubes. They fluidised particles of either round sand or spherical glass beads, having mean sizes ranging from approx. 0.5mm up to 1.5mm.

It was evident that a "gap" existed in knowledge of centrifugal fluidisation relating to the fluidisation characteristics of moderately large particles, ca. 250 - 1000 microns, within radially 'thin' beds, at ambient temperature; information of behaviour of such sized sand particles at elevated temperature was also lacking.

The objectives of the initial stages of the experimental programme were, thus, to investigate the fluidisation characteristics of silica sand particles within this size range, to study the mixing of inert and reactive particles and to identify some of the physical and geometrical constraints on RFB performance.

6.2 Apparatus

Two experimental rigs were employed in this phase of the project, both originating from the work of Metcalfe (41). The rig used to investigate fluidisation characteristics and performance limitations was Metcalfe's "No. 2 combustor" modified to accept a perspex plate at the upper end of the rotating assembly, Fig. 6.1, so that visual observations and photographic records of bed phenomena could be made. The physical dimensions of the distributor were kept as close as possible to those
used by Metcalfe so that reasonable comparisons could be made between the results obtained and those obtained from previous work.

Two distributors manufactured from 'Condiur' material were employed, the first having a conic frustum shape and the second being cylindrical; pore size and open area distribution were the same in both cases. The former allowed some comparison with Levy's work whilst the latter allowed comparison with Metcalfe's work. The combined results allowed assessment of the relative merits of the two types of distributor design. Details of these two distributors, including supplier details, can be found in Table 6.1.

Comparison of the flow characteristics of the two distributors, Fig. 6.2, shows that their resistance to fluid flow was identical. It is, therefore, reasonable to conclude that any significant differences in the performance of the RFB, when using the conical and cylindrical distributors, were due to distributor shape.

Fig. 6.3 shows a schematic layout of the rig assembly. The air supply to this rig was obtained from a "Secomak" centrifugal fan, Model 492/2, supplied by Secomak Air Products, Stanmore, Middx. Air mass flow rates were measured using orifices with upstream and downstream pressure tappings spaced D and D/2 as specified in BS1042, mounted in a D=3" nominal bore pipe. With all relevant corrections applied, mass flows could be calculated with an accuracy of about ± 1%. All air pressures were measured using water filled U-tube manometers giving accuracies of ± 1mm water gauge and all air temperatures were measured using chromel-alumel thermocouples connected to "Comark 3000" digital thermometer giving a rated accuracy of ± 1°C.
Rotational speed was measured with a "Comark 2101" electronic tachometer using a light sensitive trigger sensor. Magnetic sensing was found to be impossible due to interference, presumably from the fan motor. Comparison of rotational speeds measured by the tachometer and those deduced by "freezing" the motion with a stroboscope usually agreed very well; the difference was rarely greater than 20 r.p.m. and normally about 10 r.p.m.

The second rig used in this section of the work was a small perspex unit designed and used by Metcalfe. The distributor in this unit was manufactured from a sintered bronze cylinder 112.5mm inside diameter by 50mm long. This type of distributor which falls in the general class of 'porous media' produces very uniform fluidisation. For this reason, this particular rig was considered ideal for establishing fundamental criteria for inert/reactive particle mixing, but the small overall size of the rig made it unsuitable for investigating performance characteristics.

Air to this rig was supplied from a "Secomak" centrifugal fan, Model 492/1, but this time, for convenience, flow rates were measured using a G.E.C.-ELLIOT 47K rotameter. This device measured air flow rates with an accuracy of ± 2% (manufacturer's specification). Air pressures were measured using a water manometer and temperatures using chromel-alumel thermocouples connected to a Comark 3000 digital thermometer. A schematic of this rig is shown in Fig. 6.4.
6.3 Experimental Procedure

6.3.1 Leakage and Distributor Flow Characteristics

This section applies to the modified "No. 2 combustor" only. The procedure for determining both characteristics was the same with the exception that, in the former case, the exhaust port was sealed so that all the airflow vented through the leakage pathways. In this case, the pressure within the combustion space was always equal to that in the plenum chamber.

The procedure was then as follows:-

1. Set rotational speed to some nominal figure (say 400 rpm)
2. Start airflow at a low rate and record orifice, plenum chamber and combustor space (in the case of the distributor characteristic) pressures and temperatures.
3. Repeat (2) until a sufficiently wide range of flow rates was covered. The characteristics were then described using the parameters shown in Appendix E.

6.3.2 General Start-up Procedure

The procedure adopted was the same for both rigs with, necessary minor variations for different types of experiment. A mass of bed material, silica sand, was introduced into the stationary combustor. (The actual mass employed was dependent upon the required mass distribution or bed depth). Table 6.2 shows the particle sizes and bed masses employed for the mass distributions and bed depths investigated.
The combustor was then rotated at the speed necessary for the required radial acceleration at the distributor plane and the airflow started. After the bed had been distributed as evenly as possible, judging from visual observation under stroboscopic illumination, the airflow rate was increased until bubbles could be seen erupting at the bed surface. Where possible, the airflow rate was increased until bubbles were observed to burst over the whole bed surface. Fig. 6.5 is an example of this condition. Here, a bed of sand particles, \( dp = 0.598 \) mm is fluidised at approximately \( 2 \) m/s under an applied acceleration of 20 times gravity, in the No. 2 combustor.

6.3.3 Pressure Loss and Fluidisation Quality

After the bed had been thoroughly agitated for about 30 seconds, the airflow was increased slowly until particles of bed material could be observed escaping in the exhaust stream when the latter was illuminated using a stroboscope. In many cases, this condition was not reached due to the limited airflow available from the fan at the required pressure. Once the maximum condition had been attained note was made of all the relevant operating pressures and temperatures and then the airflow rate was reduced by a small amount. The reduction imposed was determined by the drop in plenum chamber pressure, where a change of 20 mm water gauge gave a change in overall pressure loss (distributor plus bed) of approximately 0.2% of the plenum pressure. Once again, the relevant operating parameters were noted and the process repeated down to zero flow. This procedure was repeated for a range of applied radial accelerations between 10 times and 40 times normal gravity in steps of 10g.
6.3.4 Coal Particle Mixing in Silica Sand Beds

Both the modified No. 2 combustor and Metcalfe's perspex rig were used in this section of the work. It was not necessary to re-calibrate the perspex model since no modifications were made, thus Metcalfe's flow characteristic for the distributor could be utilised.

a) Perspex Rig

The initial work on mixing was carried out in the perspex rig. The object of this work was to identify gross mixing phenomena and so the coal particles were mixed with the sand particles before introduction to the model. The start-up and fluidisation procedure was identical to that used in the fluidisation experiments. By using a coarse calibration of the rotameter, fluidisation velocities could be set at pre-determined values with reasonable accuracy (+5%) thereby allowing comparative experiments to carried out easily. Table 6.3 shows the range of sand particle sizes and coal particle sizes investigated in this work.

b) Modified No. 2 Combustor

The main body of mixing work was carried out in this apparatus as it was more representative of the proposed pressurised combustor. The types of experiment performed can be broken down into the following sets:

i) Sand and coal mixed prior to introduction to combustor
ii) Coal introduced to distributed, fluidised bed.
iii) Development of coal injector device.
The initial set of experiments were performed to confirm that the trends observed on the small perspex rig would be reproduced in a larger unit with a somewhat different type of distributor.

The second set of experiments showed the mixing history of coal particles introduced into specific areas of the fluidised bed. These established criteria for the design of a coal injection system.

The third set of experiments investigated methods of achieving the goals established from part (ii).

6.3.5 Distributor Evaluation

As shown in Table 6.1 two types of distributor were evaluated in the modified No. 2 combustor. The reasons for making the comparison was to investigate whether the advantages offered by conic distributors, as suggested by work done by Shakespeare et al (36) and by Kroger et al (112), were significant. The evaluation was made on fluidisation characteristics and operating range of given bed particle sizes and bed mass distributions for tapered and cylindrical distributors. Comparison of mixing effectiveness of the two types, as judged by visual observation, was also made.

6.3.6 Exhaust Port Diameter

The effects of changing the exhaust port diameter were investigated as it seemed reasonable that the conditions for elutriation with different particle sizes might be affected by this dimension. Two exhaust port
diameters were evaluated viz 120mm and 90mm giving ratios of exhaust port
diameter: distributor diameter of 0.6 and 0.45 respectively.

6.4 Results and Discussion

6.4.1 Bed Spreading During Start-up

The spreading of the bed material over the distributor during run-up
is the first stage of the start up sequence for the RFB combustor. In
the ideal situation, the bed will spread evenly across the distributor
surface, during the run-up to the desired operating rotational speed for
start-up, using little or no fluidising airflow. Shakespeare et al (36)
have suggested the use of a conic frustrum distributor to aid this
initial spreading of bed material. The range of included angles tested
by these workers was 4° – 8°.

The suitability of the conic distributor for thin beds (Levy used beds
of about 5cm radial thickness) was tested in the modified No. 2
combustor. Table 6.1 gives details of the 4° "Conic distributor" used
and the "Cylindrical distributor" against which it was compared.

A variety of bed particles sizes were used in the evaluation as well
as three bed masses, see Table 6.2, at a nominal start-up radial
acceleration of 10g.

Complete spreading of the bed material was not achieved with, any of
the combinations of distributor and particle size and bed mass, without
the use of some fluidising airflow. The amount of airflow required does
not appear to have any clear correlation with particle size,
see Fig. 6.6. It might be noted here that all the particle sizes used were Geldhart type "B" and thus may be expected to be "well-behaved". It is, therefore, suggested that the airflow required probably depended upon the initial distribution of bed material on the base plate of the combustor. There did not appear to be any significant difference between the airflows required when the conic distributor and when the cylindrical distributor were used and generally, the airflow required was about 40% of the fully supported airflow rate. The 4° included angle was chosen for the conic distributor as this would lead to the minimum maldistribution of bed material over a range of rotational speeds (see Levy (36)). Whilst a larger angle might improve the run-up bed spreading characteristics, it would inevitably lead to maldistribution of bed material at high rotational speeds and thus cause maldistribution of airflow and non-uniform fluidisation.

Therefore, it may be concluded that conical distributors of small included angle offer no advantages to bed distribution during the run-up of an RFB.

6.4.2 Fluidisation Characteristics - Pressure Loss

6.4.2.1 Presentation of Data

The fluidisation characteristics of a variety of bed and distributor combinations are presented in Fig. 6.7 to 6.22. Perhaps the first feature the reader will notice is that the parameters used to describe the flow characteristics are unconventional in the field of fluidised bed combustion technology where the historical influence of boiler design and chemical reactor design is strong. The parameters used in this work
reflect the intention to relate the results of the project, in the first instance, to gas turbine applications. For this reason, the author chose to use parameters conventional in gas turbine design. Thus, instead of absolute pressure losses, the dimensionless quantities pressure ratio and percentage pressure loss are quoted while, instead of fluidising velocity, the gas dynamic flow function, \( Q \), is used. The flow function is not a dimensionless group but is derived from dimensionless groups, as shown in Appendix E.

Much fuller information on relationships derivable from consideration of the dynamics of gas flows can be found in standard texts (eg.146) and Appendix E of this thesis. For comparison and completeness, however, some of the results are presented as dimensional quantities. The great advantage of the flow function is that it describes the flow conditions completely and, since tables of isentropic flow parameters are widely available, they can be used directly in design calculations generally. The performance characteristics of both compressors and turbines are normally expressed in terms of the flow function and it is therefore, a very useful tool in matching compressors, combustors and turbines together.

Figures 6.7 to 6.22 show "carpet plots" of the percentage pressure loss measured across combinations of bed and distributor plotted as a function of applied radial acceleration and applied flow function, \( Q \). The plots include several features that will be discussed in sections 6.4.3, 6.4.4 and 6.4.5.
6.4.2.2 Fluidisation Results

Historically, the point of minimum fluidisation has been identified by plotting the absolute pressure loss across the bed against the absolute gas velocity through the bed. When this method is applied to the RFB it is necessary to calculate the bed pressure loss by subtracting, from the measured bed plus distributor pressure loss, the pressure loss of the distributor alone when measured at the same flow condition.

\[ \Delta P_b = \Delta P_{(d+b)} - \Delta P_d \] 6.1

However, Trivedi and Rice (148) have shown that the presence of a bed can modify the pressure loss characteristic of the distributor. The extent of the modification will depend upon the fluid dynamic properties of the bed since, at subsonic velocities, downstream disturbances, such as the pressure loss due to the bed, propagate upstream. As a result, for a given mass flow rate of gas, the distributor pressure loss can be expected to be greater, when a bed is present than when a one is not. Beds of different particle sizes or densities will have different effects on the distributor pressure loss characteristic, especially at the higher velocities encountered in the RFB. Thus the subtraction method may not be particularly accurate, but the errors probably do not alter the prediction of the flow rate at which the bed becomes fully fluidised as this should be nearly independent of the distributor. The minimum fluidising condition, established from the intersection of packed bed pressure loss curve and the fluidised bed pressure loss line as previously seen in Fig. 3.2 may be affected by modification to the distributor characteristic since the point of intersection on Fig. 3.2 depends upon the slope of the packed bed characteristic. This slope
depends upon the distributor characteristic via equation 6.1 in which $\Delta P_d$ is a function of flow velocity. $\Delta P_b$ is independent of flow velocity only after the bed is fully fluidised.

Fig. 6.23 shows an example of the bed pressure loss characteristics derived from equation 6.1. (The characteristics are shown as percentage pressure loss vs flow function and actual derived pressure loss vs flow velocity). A point of minimum fluidisation and a point of full support can be estimated on both graphs. These can then be plotted on the combined pressure loss characteristics or used for correlation purposes.

6.4.2.3 Discussion of Fluidisation Results

An important feature to notice on the bed pressure loss characteristics is, as Metcalfe showed, the gradual transition from packed to fluidised bed conditions, the transition being more pronounced at low rotational speeds, where the relative change in radial acceleration through the bed is significant, and also when a relatively wide range of particle sizes is used in the bed. These aspects lend further support to Gel'perin's idea of fluidisation by layers (86).

When the bed pressure loss, calculated from the relationship:

$$\Delta P_b = \frac{M_b n g}{A_d} = \frac{M_b w^2}{2\pi H_d}$$

6.2

was compared with the bed pressure derived from equ. 6.1 it was found that good agreement existed for the results obtained from the experiments conducted with the conic frustum distributor but significant discrepancies were found between predicted and derived pressure losses.
for the case of the cylindrical distributor.

One possible reason why the predictions agree in one case and not the other is that when a conic distributor is used there is a greater likelihood of an even distribution of bed material and hence a near-constant mean bed thickness over a range of rotational speeds. When a cylindrical distributor is utilised, at low rotational speeds the bed is likely to have a pronounced parabolic shaped surface which only becomes cylindrical when the rotational speed results in a radial acceleration much greater than the axial acceleration 'g'. This problem could be overcome by using the calculation method adopted by Shakespeare et al (36) who split the bed into a number of axial elements and applied equ. 6.2 to each element, summing the calculated pressure drops to give the overall loss.

Two assumptions are made in the application of these relationships:—

(i) that the time average thickness of the bed is constant everywhere within the bed.

(ii) that the bed is rotating at a known angular velocity, \( w \), usually taken to be equal to that of the distributor. This assumption requires the bed to be in solid body rotation. If the bed is actually rotating at a velocity different to that of the distributor, the gas flow may then be in free vortex motion and the pressure loss equation becomes:

\[
\Delta P = \frac{M_d w^2 (r_0)^2}{2\pi H_d (r_1)}
\]  

6.3
Shakespeare et al (36) who measured pressure losses across a bed fluidised on a conic distributor found that equation 6.2 could be used to predict the pressure losses with a high degree of accuracy.

Metcalfe (41), on the other hand, measured pressure loss across a bed fluidised on a cylindrical distributor and found that equ. 6.2 over-predicted the pressure loss, when the rotational speed of the bed was assumed to be equal to that of the distributor.

Table 6.4 shows that ratio of derived : predicted bed pressure loss in the experiments conducted in this project. It is clear that equation 6.2 predicts the bed pressure loss accurately when a conic distributor is used but that an over-prediction occurs when applied to the cylindrical distributor data.

These findings are in accordance with previous work by Shakespeare et al (36) and Metcalfe (41). A high speed cine film taken to study an apparent bed instability showed that the surface layers of the bed appeared to be rotating more slowly than the distributor. This film was taken when the conic distributor was installed. The implication of this is that the bed is not in true solid body rotation and that equation 6.3 may hold true in the surface layers of the bed. Now, equation 6.3 will, for a given angular speed, \( w \), predict a higher pressure drop than equation 6.2, however, from the one cine film available it is not possible to draw any firm conclusions on the actual angular speed of the bed or the variation of angular speed through the bed.

It is clear, however, that when thin beds are to be used in an RFB, care must be exercised in predicting the bed pressure loss and it is
always advisable to measure the combined distributor and bed loss. This combined data should then be used in subsequent performance specification.

6.4.3 Minimum Operating Condition

6.4.3.1 Alternative Definitions

It is conventional to relate the operation of fluidised bed devices to the minimum fluidisation velocity, \( U_{mf} \), but this parameter does not define the minimum operating condition since it is the gas velocity at which the 'average' particle within the bed is just supported. The practical minimum operating condition is defined by the gas velocity at which the whole bed is supported. Metcalfe (41) calls this the "fully supported velocity" \( U_{fs} \). Richardson (in 80) argues that \( U_{fs} \) will be less repeatable than \( U_{mf} \) because the former will be dependent upon containment geometry and initial bed packing. The author believes that the same argument holds for \( U_{mf} \) since the determination of this value depends upon the accurate measurement of the rate of increase of pressure drop through a packed bed with increase in gas velocity. This rate of change will be dependent upon bed packing and thus assessment of \( U_{mf} \) is not likely to be more repeatable than \( U_{fs} \) on Richardson's criterion.

The fully fluidised velocity can be determined in two ways. On the combined distributor and bed characteristic, Fig. 6.7 to Fig. 6.22, an inflexion indicates the condition of full support. It is not always easy to identify the inflexion point and thus it is sometimes useful to use the alternative method, via, the derived bed pressure loss characteristic, to identify the gas velocity or flow function at which full support condition occurs. The values thus obtained may be plotted
on the combined pressure loss characteristics and then 'smoothed'. Smoothing is possible since it is reasonable to assume that the value of $U_{fs}$ for any given bed will be described by a 'well behaved' function of particle size, gas conditions and applied radial acceleration. The author believes that the use of the carpet plot technique is the most reliable way of presenting and utilising RFB operating characteristics.

6.4.3.2 Prediction of Minimum Operating Condition

Fig. 6.24 shows a plot of various correlations proposed for $Re_{mf}$ and $Re_{fs}$ mentioned in the literature survey, section 3.2.2. These are due to Wen and Yu (83), Richardson (80) and Metcalfe (41). It is evident that there is little spread in the results for values of Galileo Number up to $Ga = 10^4$ and that, even at relatively high values of $Ga$, the deviation between correlations lies within the uncertainty limits of any given correlation. These correlations, were derived for a limited range of Galileo No, with the exception of Richardson's which is derived from theoretical considerations. Extrapolations outside the prescribed range of values of the parameters is likely to give misleading predictions.

In the current work, values of Galileo Number are generally higher than those used in these previous studies, being $4 \times 10^4$ to $10^6$ compared with Wen and Yu's correlation limits of $10^{-3}$ to $10^3$ and Metcalfe's correlation limits of $10^3$ to $7 \times 10^4$.

Fig. 6.25 shows values of $Re_{mf}$ plotted against Galileo Number. The Galileo Number is calculated as if the whole bed felt the applied radial acceleration measured at the distributor surface. Similarly Fig. 6.26 shows values of $Re_{fs}$ plotted against Galileo Number. It is evident
that the majority of the measured values of \(Re_{mf}\) and \(Re_{fs}\) fall well below the extrapolated predictions, bearing out the comments made about the extrapolation of correlations.

6.4.3.3 Proposal for New Correlations

On the basis of the results presented in Fig. 6.25 and 6.26, new correlations are proposed, applicable at atmospheric temperature and pressure (\(T_{gas}\) up to about 40° C, \(P_{gas}\) up to about 1.05 atm). These correlations are as follows:

\[
Re_{mf} = (13.873^2 + .01724 \cdot Ga)^{1/2} - 13.873
\]

\[
Re_{fs} = (4.3665^2 + .01868 \cdot Ga)^{1/2} - 4.3665
\]

These correlations were generated by using 117 values each of \(Re_{mf}\), \(Re_{fs}\) and \(Ga\), measured in the current work, although the sparcity of data for \(Ga < 10^5\) means that the correlations cannot be applied with the same confidence as for \(Ga > 10^5\). Both correlations have a maximum deviation of about 30%.

It is worth noting that there is marginally less scatter in the values of \(Re_{mf}\) than in those of \(Re_{fs}\) although no importance is attached to this observation.

Comparison with the well known correlations shown on Figs. 6.24 to 6.26 Fig. 6.25 shows that the correlations derived from the data presented here fall within the ± 34% standard deviation of that due to Wen and Yu, for small values of \(Ga\).
Thus it can be argued strongly that these correlations may have wider applicability than that of Wen and Yu.

Certainly, within the RFB application there does not appear to be any need to use different correlations for 'small' and 'large' values of $Ga$.

6.4.3.4 Effect of changing the Exhaust Port Diameter on Minimum Operating Condition

During the fluidisation experiments the exhaust port diameter was reduced from 120mm to 90mm as this was found to delay the onset of elutriation, see section 6.4.4. It was also found that reducing the exhaust port diameter also had an affect on the minimum operating condition. This is illustrated in Fig. 6.27. It is evident that, when the exhaust port diameter is reduced, the value of plenum flow function $Q_p$ at which the bed becomes fully fluidised is increased. It can be shown that this effect can be predicted via consideration of the gas dynamics of the system; this is done so in Appendix F. The main consequence of this geometrical effect is considered in section 6.4.4 which presents information gathered regarding the elutriation characteristics of this RFB.

6.4.4 Aspects of Elutriation

Elutriation, the loss of particles from the combustor, will dictate the maximum operating condition if no other limitations apply. In conventional combustors elutriation is thought to be strongly linked to bubble bursting at the bed free surface since the bursting of bubbles
imparts considerable velocities to particles in the bubble vicinity. In the rotating fluidised bed, complex particle dynamics produced by the interaction of the applied radial acceleration, the rotational velocity and aerodynamic drag on the particle can result in particles spending significant time in the freeboard zone before either returning to the bed or being swept away in the exhaust gas stream.

Theoretical studies (39) have suggested that all but very small particles will return to the bed surface at moderate fluidising velocities (up to 5x Umf).

Whenever possible during the fluidisation experiments, the airflow rate at which particles of bed material were observed to be carried out from the combustor was determined. The range of applied accelerations over which this was possible for any given bed was limited by the capacity of the fan and thus for the larger sizes of particles it was only possible to determine elutriation conditions for one or two imposed radial accelerations.

The conditions at which elutriation occurred were determined by visual observation of the exhaust flow under stroboscopic illumination. When particles of size comparable to those in the bed were observed, the relevant operating conditions were recorded. The results are plotted as "Elutriation Lines" on Figs. 6.7 to 6.22. The conditions thus indicated show the maximum continuous operating condition at which the rate of bed material will be small at atmospheric temperature and pressure.

The maximum operating envelope of the RFB at atmospheric conditions is, therefore, defined by the Fully Fluidised Boundary and the
Elutriation Boundary as illustrated previously in Fig. 2.l. These boundaries will be different for different bed temperatures, at given nominal pressures, and thus do not fully characterise the operating regime of the RFB, but do provide a useful starting point.

The information gathered on elutriation is summarised in Fig. 6.28 to Fig. 6.31. Fig. 6.28 shows the effect of changing the exhaust port diameter on the value of plenum chamber flow function, \((Q)_e\), at which elutriation was first observed, for different bed thicknesses and different applied accelerations at the bed surface. It is evident that reducing the exhaust port diameter has a greater effect on \((Q)_e\) for thick beds than for thin beds. This phenomenon is probably a combination of two factors. Elutriation from an RFB is dependent, first, upon the ejection of particles into the freeboard space by bubble splashing and, second, by the dynamics of the particle motion thereafter. It would appear that, in the RFB tested, the reduction in exhaust port diameter from 120 mm to 90 mm increased to distance from the bed surface to the exhaust port to a value greater than that equivalent to the transport disengagement height for a given \(Q_p\), this effect being marginal for a thin bed (10 kg/cm²) and more significant for a thick bed (15 kg/m²).

Figs. 6.29 to 6.31 show the limits of operation available with the distributor and bed combinations tested. The operating limit is shown as the ratio \((Q)_e: (Q)_f\) and is plotted against applied acceleration at the bed surface. The information shown in these figures allows comparison of the operating characteristics of thin and thick beds and the effects of changing one aspect of the combustor geometry, viz, exhaust port diameter. The first aspect to notice is the, apparently, small range of operating conditions available at atmospheric conditions;
the ratio \((Q)_e:(Q)_{fs}\) is always less than 2. Larger ratios may be obtained in actual combustion conditions where \((Q)_{fs}\) will be markedly reduced by the increased viscosity of the hot fluidising gases. However, since the effects of temperature on \((Q)_e\) cannot be predicted with any great confidence, the suggestion of an increase in operating range may be misleading.

The second aspect to notice, see Fig. 6.29 and Fig. 6.30, is that, for a given combustor geometry and bed thickness, the operating range is principally controlled by the size of the smallest particle in the bed; this would be expected. Changing the mean particle size by increasing the upper particle size has little influence on achievable operating range. Fig. 6.30 shows that when the lower particle size boundary is increased to .6mm from .5mm an increase in operating range is achieved. The magnitude of the effect is, obviously, dependent upon the magnitude of the change in lower particle size.

Fig. 6.31 shows the effect on \((Q)_e:(Q)_{fs}\) of changing the exhaust port diameter. We see in the three cases of bed loading, viz., 10 kg/m², 12.5 kg/m², 15 kg/m², that \((Q)_e:(Q)_{fs}\) for .5 - .71 mm sand, is changed by changing the exhaust port diameter. The change can be explained by examining the information presented in Fig. 6.27 and 6.28 mentioned earlier. In Fig. 6.27 it is shown that reducing the exit port diameter, and hence the flow area, increases the value of \(Q\) required to obtain full support conditions at any given applied acceleration. In Fig. 6.28 it is shown that reducing the exit port diameter also increases the value of \(Q\) at which elutriation begins. However, the ratio of \((Q)_e:(Q)_{fs}\) shows that the rate of change of \((Q)_e\) is not always as great as that of \((Q)_{fs}\). Hence, for the
thinnest bed, the ratio \( Q_e : (Q)_{fs} \) is actually reduced by the reduction in exit port diameter; for the intermediate thickness bed, there is little change in the ratio, and for the thickest bed, the ratio is increased.

The significance of these results is that limited test work suggests that, for a given bed particle size range, the optimum exit port diameter when a 'thick' bed is utilised is smaller than when a 'thin' bed is utilised, from the point of view of maximising the range of input airlow conditions over which the bed will be fluidised without elutriation occurring.

The penalty incurred by reducing the exit port area is an increase in the total pressure loss of the combustor. Now, boilers operate with pressure losses of a few inches of water gauge, while gas turbines operate with pressure losses that are, typically, 5 - 8% of the compressor outlet pressure. It is very difficult for an RFB to satisfy the former but the latter can be accommodated relatively easily provided that applied accelerations are not large, ie up to 30 gravities, and the distributor effective flow area is also relatively large 2 - 6%( ie low pressure loss). When the forgoing is true, a bed mass distribution of 12.5 kg/m² would give rise to combustor total pressure losses of less than 10% over the whole operating range.

From the point of view of pressure loss, the proceeding evidence on the effects of exhaust port diameter on operating range would support the use of beds whose radial thickness was the minimum allowable by other considerations, eg. combustion efficiency. As a result, the pressure loss through the bed would be minimised and also, because a larger
exhaust port could be used, the combustor total pressure loss would be minimised.

The author considers that there is insufficient data available on which to perform a correlation analysis for elutriation conditions that would yield a meaningful result. Indeed, in view of the complex nature of elutriation from an RFB, it is unlikely that a correlation would be of any value. A simple theoretical analysis of elutriation is given Chapter 8. This analysis, based on the work of Chevray et al (39) and Subzwar (42), does not satisfactorily describe the observed elutriation phenomena and illustrates the difficulty in studying this aspect of RFB performance.

6.4.5 Bed Instabilities

During the early fluidisation experiments a phenomenon was encountered that had not been reported in recent works on RFB technology but which had been mentioned by Deinken (97). This phenomenon was bed instability or "bounce" (the term used by Deinken). The occurrence of "bounce" was first noticed in this project because of the audible sound produced by the induced pulses in the exhaust airflow and because of the mechanical vibration induced in the whole rig assembly. The condition at which exhaust pulsation occurred were noted and are shown in Figs. 6.7 to 6.19 as a "Pulsation Line" on the flow characteristics. This description was chosen because of the most obvious effects, viz, pulsation of the exhaust flow and mechanical vibration.

In order to investigate the pulsation phenomenon more fully, a high speed cine camera was mounted above the rig and focussed on the free
surface of the bed. The resulting film showed that, under pulsating conditions, the eruption of swarms of small bubbles, evenly distributed over the entire free surface alternated with a period of apparent quiescence. The appearance seemed to correspond closely to the description of "bounce" given by Deinken (97). Sections of the film demonstrating the two effects can be seen in Fig. 6.32. The operating conditions are shown in Table 6.5. Using a Bruel and Kjaer 2209 sound level meter, an estimate of the exhaust flow pulsation frequency was obtained and this was compared with the frequency of bubble swarm eruption derived from analysis of the high speed film. The frequency of bubble swarm eruption on the film was found to be approximately 140Hz, whilst the maximum amplitude of pulsation frequency measured by the sound meter occurred in the range 125 - 250 Hz. This strongly suggests that the exhaust flow pulsation was connected with the bubble swarm eruptions.

The pulsation phenomenon was encountered when the tapered distributor, described in Table 6.1, was mounted in the rig. It can be seen in Figs. 6.7 to 6.19 that pulsation occurred with all the bed particle sizes and bed mass distributions tested and that, as the mass distribution of bed material was increased, the difference in fluidising velocity between the occurrence of pulsation and the onset of elutriation gradually reduced. At any given air mass flow rate, the pulsation could be suppressed by increasing the applied acceleration but could subsequently be restored by increasing the air mass flow rate, provided that elutriation did not occur first.

The characteristics of the pulsation did not appear to be affected when the exhaust port diameter of the combustor was reduced.
When the RFB was operated with the cylindrical distributor, described in Table 6.1, no pulsing of the exhaust flow or mechanical vibration of the rig was encountered. A more limited range of beds was used for this section of the test work but the bed particle size ranges and the bed mass distributions utilised were common to both tapered and cylindrical distributor configurations.

The measured pulsation frequencies were compared with a number of models described by Verloop and Hertjees (147). Further, organ-pipe resonance of the inlet ducting and the airflow relaxation frequency of the combustor space were estimated. The results are shown in Table 6.6. None of these models satisfactorily estimated the measured pulsation frequency and so other mechanisms for the pulsation must be sought.

Chevray et al (39) have modelled bubble dynamics in the RFB. Under certain conditions, they deduced that the presence of large volumes of fluidising gas in the surface layers of the bed could lead to spontaneous bubble generation in these layers. These bubbles might appear as swarms if the fluidising gas was evenly distributed throughout the bed surface layers. The model does not suggest that bubble generation would be periodic, however.

6.4.5.1 Suggested Mechanism of Pulsation

Based on the measurements and calculations of frequency and the possible generation sources, it is thought that "organ pipe" pressure fluctuations generated in the inlet ducting alternately starved and flooded the combustor with fluidising air thus giving rise to the periods
of bed quiescence and bubble swarm eruption, the latter being generated by
the mechanism proposed by Chevray et al (39). When a cylindrical
distributor was substituted for the tapered distributor, it is thought
that the less uniform distribution of bed material allowed gas bypass to
occur in the thinner portions of the bed and this would have acted as a
"pressure relief".

As this phenomenon may cause operating limitation for practical
combustors further investigation may be justified.

6.5 Particle Mixing

This feature of fluidised bed operation is recognised as being
extremely important to successful combustion of solid fuels (eg. 2) and
much effort has been put into developing f.b.c. designs that can be fired
on a range of solid fuels (eg. 3). Particle mixing is of equal
importance in the RFBC but there is only limited information available to
help determine the requirements for effective mixing (41,42).

The work undertaken in this section had three objectives:-

1) To determine the range of inert and reactive particle size that
could be made to mix effectively.

2) To determine the fluidising gas flow requirements for
effective mixing.
3) To determine a design of reactive particle feeder which would produce the desired initial conditions for subsequent effective mixing.

Objectives 1) and 2) were studied in the small perspex rig described in section 6.2 and the results confirmed in the modified combustor rig. Objective 3) was studied on the modified combustor rig only.

6.5.1 Particle Size and Fluidising Velocity Criteria

The literature survey had revealed that there was insufficient information on the effects of inert and reactive particle sizes on mixing effectiveness for the purposes of combustor and feeder design. The work of Metcalfe (41) seemed to suggest that mixing effectiveness might be sensitive to the choice of particle size ranges and the concomitant minimum fluidising velocities, whilst that of Kroger et al (127) suggested the possibility of relative insensitivity under certain circumstances, viz, round particles of equal density.

In order to explore mixing further two sets of experiments were conducted. The first experiments were conducted with the inert and reactive particles premixed. In this way it was hoped that the effects of initial particle distribution would be minimised by starting with a near homogeneous mixture. It would then be possible to determine:

1. If effective radial mixing could be achieved at all, and
2. the minimum fluidising gas flow required to achieve this radial mixing.
The second set of trials were conducted by introducing a mass of reactive particles on to the surface of a distributed, but defluidised, bed. In this way the effect of initial particle distribution could be studied during subsequent fluidisation:

6.5.1.1 Effects of Particle Size Ranges and Fluidising Velocity

The results of fluidising homogeneous mixtures of the particle size ranges, shown in Table 6.3, are shown in Fig. 6.33. The experiments showed that, if the fluidising velocity was equal to or greater than twice the minimum fluiding velocity of the inert particles, effective radial mixing of inert and reactive particles could be achieved with all of the mixtures studied. It should be noted, however, that elutriation of small coal particles occurred when the smallest coal size ranges (.250 - .355 mm and .355 - .500 mm) were fluidised in the larger size range (.355 - .500 mm) sand beds.

Within the range of coal particle and sand particle sizes tested there did not appear to be an upper limit on coal : sand particle size ratio for good radial mixing. Thus it would appear that, provided a near-homogeneous initial mixture can be obtained, the target initial coal particle size should be as large as can be transported reliably from storage to combustion zone.

6.5.1.2 Effect of Non-Homogeneous Initial Mixture

The effects of an initially non-uniform distribution of coal in the bed was investigated by introducing a mass of coal on to a distributed, but de-fluidised, bed. It was found that the subsequent mixing depended
upon where the coal particles had been introduced on to the bed and there
did not appear to be any apparent difference when different sand or coal
size ranges were used.

Fig. 6.34 is a sketch showing the result of dropping the coal on to
the combustor baseplate such that the coal was spun out, by centrifugal
forces, on to the lowest region of the bed. The coal would form a band,
usually about lcm wide, rising up from the base plate. Upon fluidisation
of the bed, the coal was seen to mix vigorously with the bed material in
a radial direction in the lower region of the bed but no upwards axial
movement of the coal particles could be detected, even after several
minutes fluidisation.

Fig. 6.35 shows the result of introducing coal particles on to the
centre section of the defluidised bed. Once again, an initial band of
ccoal was formed which would mix vigorously in a radial direction but
which did not show any upwards axial movement. Some downwards axial
movement was seen but the effect seemed to be small.

A similar result was found when the coal was introduced into the
upper region of the bed surface.

Thus, it can be seen that transverse mixing, ie. that perpendicular to
the fluidising gas flow, is poor in the RFB in the same way as is found
in conventional fluidised bed combustors. The obvious solution, as used
in conventional FBC is to use a design of coal feeder such that the fuel
is introduced into the bed to form, as nearly as possible, a uniform
distribution over the whole bed surface.
These findings contrast markedly with the 'tumbling' mixing reported by Demircan et al (35) in which an RFB, operated with its axis at an angle to the vertical, exhibits the churning mixing action similar to the common cement mixer providing an induced transverse mixing component.

6.5.2 Coal Particle Feeder Development

This short programme was carried out using the modified No. 2 combustor. A few test runs were carried out to check that the criteria determined for particle size and fluidising velocity on the perspex rig held true on this larger model which had a pierced steel distributor. Experiments were conducted with both the conic and cylindrical distributors and with sand particles in the ranges .500 - .600 mm and .500 - .710 mm. In all cases, the results confirmed the conclusions of the initial work, viz, that large coal particles would mix well with a bed operated as twice $U_{mf}$ (ie bubbling) and that a uniform distribution of coal particles was necessary as axial movement of the coal particles in the bed was very small. Axial motion did not appear to be aided by the divergence of the conic distributor as had been anticipated.

6.5.2.1 Spreader Type Feeder

This feeder shown in Fig. 6.36 consisted of a round metal plate, 100mm in diameter, supported by a 12mm diameter post which was fixed to the combustor baseplate. Fuel particles, dropped on to the centre of the plate, which rotated at the same speed as the combustor, would be thrown outwards on to the bed surface with axial distribution being caused by both random particle motion (due to random sizes and shapes of the fuel particles) and also by the various trajectories of the particles after
leaving the edge of plate.

Four configurations of spreader were tested:

1. Flat plate mounted at 1/4 combustor height above baseplate.

2. Flat plate mounted at 1/2 combustor height above baseplate.

3. Flat plate mounted at 3/4 combustor height above baseplate.

4. Dished plate mounted at 1/2 combustor height above baseplate.

All four configurations showed varying degrees of success depending upon combustor rotational speed and fluidising air velocity.

**Configuration (1)** - At all rotational speeds and low gas velocities the fuel particle were seen to be concentrated in a narrow band near the combustor baseplate. Very little axial distribution was achieved. As fluidising gas velocity was increased from $2 \text{U}_{mf}$ to about $2.5 \text{U}_{mf}$ axial distribution improved a little but elutriation of the smaller fuel particles became noticeable with the obvious implications on combustion efficiency.

**Configuration (2)** - Fuel distribution with this spreader was similar to that encountered in (1) with the exception that the initial band of fuel particles was positioned closer to mid combustor height with gradual thinning towards the baseplate. Elutriation seemed more significant at velocities similar to those used in (1) due to the reduced distance between the plate and the combustor exit port.
Configuration (3) - In the same way as (1) and (2), an initial band of fuel was formed which gradually thinned towards the base plate. This configuration produced the most even distribution of fuel of the three flat plate spreaders tested but two features showed that it could not be used in practice. First, elutriation rates were very high because of the proximity of the plate to the exit and, second, the choking effect of the plate on the exhaust flow appeared to affect the fluidisation of the bed, encouraging gas bypass in the upper regions and effectively eliminating visible bubbling near the baseplate.

Configuration (4) - This was tried as a compromise between (2) and (3). It was thought that a dished plate might produce enough upwards velocity in the fuel particles to cause spreading to the upper regions of the bed. The 100mm diameter plate was dished to give a 5mm displacement at the edge giving an effective radius of curvature of the dish of about 240mm. Upon testing, it was found that the dishing did not have a strong effect on axial distribution. Elutriation, however, was as significant as observed in (2). Thus it was decided that the spreader type of feeder should not be employed since the criteria for fuel distribution could not be satisfied.

6.5.2.2 Pneumatic Feeder

Metcalfe (41) had found that a pneumatic feed pipe located at about 3/4 combustor height above the baseplate could achieve satisfactory fuel distribution over the surface of the bed. However, when it was necessary to use high flow rates of transport air the possibility arose of severe disturbance of the bed by the jet efflux from the transport tube. For this reason, pneumatic transport had been discounted initially but, after
the failure of the spreader feeder it was decided to attempt development of a pneumatic feeder that did not cause the problem encountered by Metcalfe.

The criteria for satisfactory operation of the pneumatic feeder were determined to be:–

i) high transport velocity in delivery pipe; air velocity to be greater than 6m/s (determined by a simple test which showed that 12mm pipe blocked with 3mm coal particles at lower air velocities.

ii) low transport air velocity but high fuel particle velocity at the feeder head; air velocity to be ca. 25% of delivery pipe velocity ie. 1.5m/s, and particle velocity to be as near as possible to delivery pipe velocity.

Criteria (i) is an operational requirement and not open to mainpulation.

Criterion (ii) suggested that a feeder head shape needed to be developed which would allow rapid deceleration of the airflow but not allow sufficient time for significant deceleration of the fuel particles. It is common practice in conventional coal fired fluidised bed boilers to use bifurcated heads on pneumatic feeders (eg.3) to reduce bed disturbance by jet efflux.

Two feeder heads were designed, both having exit areas of about four times that of the main delivery pipe. These heads, shown in Fig. 6.37,
were

a) a radially expanding channel formed by two flat discs and
b) a cruciform head formed from two pipes, of the same diameter as the main delivery pipe, attached at right angles to the delivery pipe.

Both types gave satisfactory performance in the spreading of coal particles over the bed surface but the cruciform head showed little sensitivity to axial position and more consistent performance with change in fluidising velocity. Elutriation rates with both feeders appeared to be low, even with fairly high fluidising velocities (up to 3m/s) and interference with bed fluidisation was minimal, suggesting that criteria (i) was being fulfilled to a reasonable degree.

Particle distribution at the bed surface seemed to be more random and particle velocities higher when the cruciform feeder head was used and it was, therefore, decided to manufacture a stainless steel cruciform feeder head and delivery pipe for use in the pressurised combustor.

6.5.3 Combustor Fuel Feeder

A drawing of this feeder can be found in Appendix B. The feeder was designed such that the cruciform head would be located at about mid-combustor height. The lengths of the feeder head arms were made as large as could be conveniently passed through the combustor exit port, resulting in a cruciform span of 80mm. The feeder tube and head tube diameters were 13mm and the material chosen was stainless steel, grade 316.
6.6 Elutriated Particle Capture

During the fluidisation experiments it was noted that, when bed particles were elutriated, the particles' angular momentum was translated into a strong component of radial velocity downstream of the exit port. As a result, the particles were separated from the exhaust flow and collected on the top surface of the combustor top plate. It had been recognised, early in the project, that elutriation might be a problem for two reasons: the first being that elutriated fuel would result in loss of combustion efficiency and a possible fire hazard in the exhaust stack and the second being that elutriated ash particles could result in erosion of exhaust valves or nozzles and also pose an explosion hazard in the exhaust stack.

Capture of elutriated particles close to the exhaust port had been described in the patent awarded to Elliot (149). The intention of the devices shown in the patent were to recycle elutriated particles to allow combustion of 'lost' fuel or to transport hot particles to a heat exchanger for steam raising. It was felt that both these strategies would involve excessive mechanical complexity and thus it was decided that efforts should be made only to capture the elutriated particles for later examination.

The device tested on the cold flow rig is shown, diagrammatically, in Fig. 6.38. Elutriated particles would be thrown into the inner collection chamber by the radial component of velocity. By imposing a suction on the outer collection chamber, it was intended to induce the captured particles to be sucked in to this region and thus prevent over-loading of the inner collection chamber.
A very limited test series showed that the device would capture particles as large as .500 mm, provided that a pressure difference of about 10 cm water gauge (98.1 N/m²) was generated between the two chambers. The actual flow rate of gas through the chambers was not measured but was estimated to be about .0354 m³/s through the twelve 1/4" diameter holes joining the inner and outer chambers. This was equivalent to about 2.75% of the fluidising gas flow rate at the maximum fluidising velocity tested of about 2.5 m/s.

Despite the encouraging results obtained with this very simple model it was decided that inclusion of such a device in the combustor would introduce unnecessary additional complexity to the rig operation. Furthermore, since the venturi scrubber installed in the rig was expected to remove all particulars of a size larger than .001 mm, additional particle removal was unnecessary.

6.7 Summary

Cold fluidisation work was carried out to extend the knowledge of fluidisation characteristics of medium size particles in a rotating fluidised bed.

It was found that correlations derived in earlier work and recognised 'standard' correlations for $U_m$ and $U_f$ over-estimated the measured values at high Galileo number and new correlations have been proposed that better fit the experimental results and agree well with previous work.

The effects of combustor geometry were studied. It was found that, when using thin beds of up to 15 mm radial thickness, there is no
advantage in using a conic frustrum distributor. Varying the size of the exhaust port was found to affect both the flow rate at which elutriation started and the ratio of the flow rate at the fully supported condition to that at initiation of elutriation. It would appear that there may be an optimum size of exhaust port for a given distributor diameter and bed radial thickness.

A fluidisation instability occurred which appears to be linked to the shape of the distributor. The instability occurred when a conic distributor was used to support the bed material but did not occur when a cylindrical distributor was used.

It has been demonstrated that particle mixing perpendicular to the gas flow direction is poor and that, in order to achieve relatively even fuel distribution within the bed, fuel must be spread over the entire bed surface. To this end, a fuel injector for use in the rotating combustor was developed.

It has been shown that it is possible to capture elutriated particles just downstream of the exhaust port by making use of the particles' angular momentum. However, development of a recycling system was considered to be outside the scope of this project and thus the idea was not taken further than a simple demonstration.
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Conic Distributor</th>
<th>Cylindrical Distributor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pore Size</td>
<td>400 microns</td>
<td>400 microns</td>
</tr>
<tr>
<td>Taper Angle (included)</td>
<td>4°</td>
<td>0°</td>
</tr>
<tr>
<td>Diameter (at combustor base)</td>
<td>200 mm</td>
<td>200 mm</td>
</tr>
<tr>
<td>Diameter (at combustor base)</td>
<td>205 mm</td>
<td>205 mm</td>
</tr>
<tr>
<td>Length</td>
<td>80 mm</td>
<td>80 mm</td>
</tr>
<tr>
<td>Open Area (including Cd)</td>
<td>7.6 %</td>
<td>6.3 %</td>
</tr>
<tr>
<td>Material</td>
<td>&quot;CONIDUR&quot;</td>
<td>&quot;CONIDUR&quot;</td>
</tr>
<tr>
<td>Manufacturer</td>
<td>HEIN LEHMANN A.G.</td>
<td>HEIN - LEHMANN A.G.</td>
</tr>
<tr>
<td></td>
<td>Postfach 4000</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Düsseldorf</td>
<td></td>
</tr>
<tr>
<td>Designation</td>
<td>Thickness 0.5 mm</td>
<td>Thickness: 0.5 mm</td>
</tr>
</tbody>
</table>
TABLE 6.2  BED PARAMETERS

<table>
<thead>
<tr>
<th>PARTICLE SIZE RANGE</th>
<th>SIZE RANGE mm</th>
<th>MEAN PARTICLE SIZE mm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>.500 - .600</td>
<td>.548</td>
</tr>
<tr>
<td></td>
<td>.500 - .710</td>
<td>.596</td>
</tr>
<tr>
<td></td>
<td>.600 - .710</td>
<td>.653</td>
</tr>
<tr>
<td></td>
<td>.600 - .850</td>
<td>.714</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>BED MASS DISTRIBUTIONS kg/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thin Bed</td>
</tr>
<tr>
<td>Median Bed</td>
</tr>
<tr>
<td>Thick Bed</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>--------------------------</td>
</tr>
<tr>
<td><strong>Silica Sand Particle Sizes</strong></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td><strong>Coal Particle Sizes</strong></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>dp mm</td>
</tr>
<tr>
<td>-------</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>M/A</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>dp mm</td>
</tr>
<tr>
<td>-------</td>
</tr>
<tr>
<td>M/A</td>
</tr>
<tr>
<td>10</td>
</tr>
<tr>
<td>10</td>
</tr>
<tr>
<td>20</td>
</tr>
<tr>
<td>30</td>
</tr>
<tr>
<td></td>
</tr>
</tbody>
</table>
### TABLE 6.4(c) EXHAUST PORT DIAMETER = 90 mm CYLINDRICAL DISTRIBUTOR

<table>
<thead>
<tr>
<th>dp mm</th>
<th>.500 -.600</th>
<th>.500 -.710</th>
<th>.600 -.850</th>
</tr>
</thead>
<tbody>
<tr>
<td>M/A</td>
<td>12.5</td>
<td>12.5</td>
<td>12.5</td>
</tr>
<tr>
<td>ng</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>0.88</td>
<td>0.82</td>
<td>0.83</td>
</tr>
<tr>
<td>20</td>
<td>0.89</td>
<td>0.84</td>
<td>0.89</td>
</tr>
<tr>
<td>30</td>
<td>0.94</td>
<td>0.90</td>
<td>0.90</td>
</tr>
</tbody>
</table>
**TABLE 6.5** Operating conditions for Fig. 6.32

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Distributor type</td>
<td>Tapered Conidur</td>
</tr>
<tr>
<td>Bed material</td>
<td>Sand</td>
</tr>
<tr>
<td>Size range</td>
<td>.5 -.6 mm</td>
</tr>
<tr>
<td>Mass distribution</td>
<td>10 kg/m²</td>
</tr>
<tr>
<td>Applied acceleration at distributor</td>
<td>20g</td>
</tr>
<tr>
<td>Applied acceleration at bed surface</td>
<td>18g</td>
</tr>
<tr>
<td>Plenum flow function</td>
<td>7.3x10^{-4}</td>
</tr>
<tr>
<td>$Q_p : Q_{fs}$</td>
<td>1.78</td>
</tr>
<tr>
<td>Fluidising velocity</td>
<td>3.64 m/s</td>
</tr>
<tr>
<td>Bed pressure loss</td>
<td>1.77 %</td>
</tr>
<tr>
<td>Combustor pressure loss</td>
<td>3.28 %</td>
</tr>
</tbody>
</table>
### TABLE 6.6
MEASURED, DEDUCED AND CALCULATED FLUCTUATION FREQUENCIES

<table>
<thead>
<tr>
<th>SOURCE</th>
<th>FREQUENCY Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured exhaust flow pressure fluctuation,</td>
<td>124 - 250</td>
</tr>
<tr>
<td>Bruel and Kjaer Noise Meter (Max. Response)</td>
<td></td>
</tr>
<tr>
<td>Deduced bubble swarm eruption frequency from high speed film</td>
<td>approx. 140</td>
</tr>
<tr>
<td>Models from Verloop and Heertjes (147)</td>
<td></td>
</tr>
<tr>
<td>a) Simple Harmonic Motion (Bubbling Bed)</td>
<td>27 - 50</td>
</tr>
<tr>
<td>b) Simple Harmonic Motion (Turbulent Bed)</td>
<td>17 - 31</td>
</tr>
<tr>
<td>c) Slugging</td>
<td>60 - 155</td>
</tr>
<tr>
<td>Organ pipe generation = velocity of sound in inlet ducting / inlet pipe length</td>
<td>approx. 64</td>
</tr>
<tr>
<td>Airflow relaxation = (air mass flow rate) (air mass in combustor + plenum)</td>
<td>approx. 27.5</td>
</tr>
</tbody>
</table>
FIG 6.1 Modified "No.2 Combustor"

GLAND PACKING

SEALING RING

PERSPEX TOP

DISTRIBUTOR

AIR INLET (4 OFF)

BEARING AND SEAL
FIG 6.2 Pressure loss characteristics of Conidur distributors

\[ \frac{\Delta P}{P_p} \%
\]

KEY

+ - tapered distributor (4° included angle)

x - cylindrical distributor

\[ Q_p = \frac{m_a \sqrt{T_p}}{C_d A_h P_p} \]
FIG 6.4 Schematic of perspex model cold flow rig
FIG 6.5 Fluidisation in the RFB
FIG 6.6 Airflow required to achieve bed distribution at 10 times gravity

MEAN PARTICLE SIZE - MICRONS

AIR FLOW REQUIRED TO ACHIEVE BED DISTRIBUTION

AIR FLOW AT FULLY SUPPORTED CONDITION

KEY

BED MASS kg/m²

○ 10

□ 12.5

△ 15
FIG 6.7 Fluidisation characteristics of 500-600 micron sand

Bed mass 10 kg/m², Particle density 2630 kg/m³

Condur distributor. Taper angle=2°. L/D=0.4.

Combustor Ω_e/Ω_d = 0.6
FIG 6.8 Fluidisation characteristics of 500-600 micron sand

Bed mass 12.5 kg/m². Particle density 2630 kg/m³

Conidur distributor. Taper angle=2. L/D=0.4

Combustor $\frac{\varnothing_c}{\varnothing_d} = 0.6$
FIG 6.9 Fluidisation characteristics of 500-600 micron sand

Bed mass 15 kg/m², Particle density 2630 kg/m³

Condur distributor, Taper angle = 2°, L/D = 0.4

Combustor $\vartheta_e/\vartheta_d = 0.6$
FIG 6.10  Fluidisation characteristics of 500-710 micron sand

Bed mass 10 kg/m², Particle density 2630 kg/m³

Conidur distributor, Taper angle=2°, L/D=0.4

Combustor $\phi_e/\phi_d=0.6$

![Graph showing fluidisation characteristics](graph.png)

- Applied Acceleration n x g

\[
\left( \frac{\Delta P}{P} \right) \% = \frac{10^5 X(\Delta V/T)}{(A_d P)} d + b
\]

- Elutriation Limit
- 'Pulsation' Fully supported Condition
FIG 6.11 Fluidisation characteristics of 500-710 micron sand

Bed mass 12.5 kg/m². Particle density 2630 kg/m³

Conidur distributor. Taper angle=2°. L/D=0.4

Combustor ø_e/ø_d=0.6
Fluidisation characteristics of 500-710 micron sand

Bed mass 15 kg/m². Particle density 2630 kg/m³

Conidur distributor. Taper angle=2°, L/D=0.4

Combustor Øe/Ød=0.6
FIG 6.13  Fluidisation characteristics of 500-710 micron sand

Bed mass 10 kg/m$^2$. Particle density 2630 kg/m$^3$

Conidur distributor. Taper angle=2. L/D=0.4

Combustor $\phi_e/\phi_d=0.45$

\[
\left(\Delta P/P\right)_p \times \frac{10^5 X(m/\sqrt{T})}{(A_d P)}
\]

Applied Acceleration $nxg$

Elutriation Limit

'Pulsation' Fully supported Condition
FIG 6.14 Fluidisation characteristics of 500-710 micron sand

Bed mass 12.5 kg/m². Particle density 2630 kg/m³

Conidur distributor. Taper angle=2. L/D=0.4

Combustor $\bar{\Omega}_e/\bar{\Omega}_d=0.45$
Fluidisation characteristics of 500-710 micron sand

Bed mass 15 kg/m², Particle density 2630 kg/m³

Conidur distributor, Taper angle=2, L/D=0.4

Combustor Ø₁/Øₐ=0.45

\[
\left(\frac{\Delta P}{P}\right)\% = \frac{10^5 X(\Delta \sqrt{T})}{(A_d P)}
\]
Fluidisation characteristics of 600-710 micron sand

Bed mass 10 kg/m². Particle density 2630 kg/m³

Conidur distributor. Taper angle=2. L/D=0.4

Combustor $\varnothing_e/\varnothing_d=0.45$
FIG 6.17 Fluidisation characteristics of 600-710 micron sand

Bed mass 12.5 kg/m². Particle density 2630 kg/m³

Conidur distributor. Taper angle=2°. L/D=0.4

Combustor ø_e/ø_d=0.45

---

Diagram:

Applied Acceleration nxg

\[ \left( \frac{\Delta P}{P} \right) \%
\]

\[ \frac{\left( \frac{X}{m \sqrt{T}} \right)}{\left( \frac{A_d P}{P} \right)} \times 10^5 \]

- Elutriation Limit
- 'Pulsation' Fully supported Condition
FIG 6.18 Fluidisation characteristics of 600-740 micron sand

Bed mass 15 kg/m². Particle density 2630 kg/m³

Conidur distributor. Taper angle=2°, L/D=0.4

Combustor ø_e/ø_d=0.45
FIG 6.19 Fluidisation characteristics of 600-850 micron sand
Bed mass 10 kg/m². Particle density 2630 kg/m³
Conidur distributor. Taper angle=2. L/D=0.4
Combustor Ø_e/Ø_d=0.45

\[ \left( \frac{\Delta P}{P} \right) \frac{10^5 \times \left( \frac{\varphi \sqrt{T}}{A_d P} \right)}{p/d+b} \]

Applied Acceleration nxg

Elutriation Limit
'Pulsation'
Fully supported Condition
Fluidisation characteristics of 500-600 micron sand
Bed mass 12.5 kg/m², Particle density 2630 kg/m³
Condur distributor. Taper angle=0, L/D=0.4
Combustor $\varnothing_e/\varnothing_d=0.45$
FIG 6.21 Fluidisation characteristics of 500-710 micron sand
Bed mass 12.5 kg/m². Particle density 2630 kg/m³
Conidur distributor. Taper angle = 0. L/D = 0.4
Combustor d_e/d_d = 0.45

![Graph showing fluidisation characteristics](image-url)

- Applied Acceleration nxg
- Flutration Limit
- Fully supported Condition

\[
\left( \frac{\Delta P}{P} \right)_{\%} = \frac{10^5 X(m \sqrt{T})}{(A_d P_p)}
\]
FIG 6.22 Fluidisation characteristics of 600-850 micron sand
Bed mass 12.5 kg/m². Particle density 2630 kg/m³
Conidur distributor. Taper angle=0, L/D=0.4
Combustor θ /θ₀ = 0.45

Applied Acceleration nxg

\( \left( \Delta P \right) / P \) % = 10⁻⁵ \( x(\Delta \sqrt{T}) \) 

\( \frac{A_d P}{P} \)

Fully supported Condition
FIG 6.23  Comparison of bed pressure loss curves.
500-600 micron sand 12.5 kg/m². Corresponds to FIG 6.8
Comparison of various correlations for $Re_{mf}$ and $Re_{fs}$

**Curve 1:** Wen & Yu Ref:63) $Re_{mf}=(33.7^2+0.040G_a^{*})^{-0.5}-33.7$

**Curve 2:** Richardson Ref:68) $Re_{mf}=(25.7^2+0.0365G_a^{*})^{-0.5}-25.7$

**Curve 3:** Metcalfe Ref:43) $Re_{mf}=(3.369+0.0014G_a^{*})^{0.752}$

**Curve 4:** Metcalfe Ref:43) $Re_{mf}=4.1854+0.0004G_a^{*}$

**Curve 5:** Metcalfe Ref:43) $Re_{fs}=(1.708+0.0024G_a^{*})^{0.689}$

**Curve 6:** Metcalfe Ref:43) $Re_{fs}=9.7579+0.0003G_a^{*}$
Measured particle Reynolds Number at $U_{mf}$

<table>
<thead>
<tr>
<th>SYMBOL</th>
<th>PARTICLE SIZE $\mu$m</th>
<th>EXIT DIA. mm</th>
<th>REMARKS</th>
</tr>
</thead>
<tbody>
<tr>
<td>•</td>
<td>500 - 600</td>
<td>120</td>
<td>TAPERED DISTRIBUTOR.</td>
</tr>
<tr>
<td>*</td>
<td>500 - 710</td>
<td>120</td>
<td></td>
</tr>
<tr>
<td>Δ</td>
<td>500 - 710</td>
<td>90</td>
<td>CYLINDRICAL DISTRIBUTOR</td>
</tr>
<tr>
<td>▽</td>
<td>600 - 710</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>○</td>
<td>600 - 850</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>▲</td>
<td>500 - 600</td>
<td>90</td>
<td>PRESSURED RIG</td>
</tr>
<tr>
<td>▼</td>
<td>500 - 710</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>●</td>
<td>600 - 850</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>•</td>
<td>355 - 500</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>□</td>
<td>425 - 500</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>□</td>
<td>425 - 600</td>
<td>90</td>
<td></td>
</tr>
</tbody>
</table>

**CURVE 1** - THIS WORK $Re_m = (13.873 \times 2 + 0.01724Ga)^{0.5} - 13.873$

**CURVE 2** - THIS WORK $Re_m = (4.3665 \times 2 + 0.01866Ga)^{0.5} - 4.3665$

**CURVE 3** - METCALFE REF:43 $Re_m = (3.369 + 0.0014Ga)^{0.752}$

**CURVE 4** - METCALFE REF:43 $Re_m = (4.165 + 0.0004Ga)^{0.752}$

**CURVE 5** - METCALFE REF:43 $Re_m = (1.708 + 0.0024Ga)^{0.689}$

**CURVE 6** - METCALFE REF:43 $Re_m = (9.757 + 0.0003Ga)^{0.752}$
FIG 6.26 Measured particle Reynolds Number at $U_{fs}$

![Graph showing particle Reynolds number vs. Galileo number with data points and curves.]

**Curve 1:** This work $Re_m = (13.873^2 + 0.01724Ga^2)^{0.5} - 13.873$

**Curve 2:** This work $Re_m = (4.3665^2 + 0.01868Ga^2)^{0.5} - 4.3665$

**Curve 3:** METCALFE REF(43) $Re_m = (3.339 + 0.0014Ga^2)^{0.752}$

**Curve 4:** METCALFE REF(43) $Re_m = 4.1854 + 0.0004Ga$

**Curve 5:** METCALFE REF(43) $Re_m = (1.708 + 0.0024Ga)^2 - 0.689$

**Curve 6:** METCALFE REF(43) $Re_m = 9.7579 + 0.0003Ga$

<table>
<thead>
<tr>
<th>SYMBOL</th>
<th>PARTICLE SIZE $\mu m$</th>
<th>EXIT DIA. mm</th>
<th>REMARKS</th>
</tr>
</thead>
<tbody>
<tr>
<td>+</td>
<td>500 - 600</td>
<td>120</td>
<td></td>
</tr>
<tr>
<td>o</td>
<td>500 - 710</td>
<td>120</td>
<td></td>
</tr>
<tr>
<td>△</td>
<td>500 - 710</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>□</td>
<td>600 - 710</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>▽</td>
<td>600 - 850</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>o</td>
<td>500 - 600</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>△</td>
<td>500 - 710</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>▽</td>
<td>600 - 850</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>o</td>
<td>355 - 500</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>△</td>
<td>425 - 500</td>
<td>90</td>
<td></td>
</tr>
<tr>
<td>▽</td>
<td>425 - 600</td>
<td>90</td>
<td></td>
</tr>
</tbody>
</table>

TAPERED DISTRIBUTOR

CYLINDRICAL DISTRIBUTOR

PRESSURED RIG
FIG 5.27 Effect on fully supported flow function of reducing the exhaust port diameter
Sand bed 500-710 microns

\[ 10^5 \times Q_p \]

\( C E N T R I P E T A L \ A C C E L E R A T I O N \ A T \ B E D \ S U R F A C E \ n \times g \)

<table>
<thead>
<tr>
<th>KEY</th>
<th>BED MASS kg/m²</th>
<th>BED DEPTH mm</th>
<th>( \frac{\varphi_e}{\varphi_d} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>⊗</td>
<td>10</td>
<td>10</td>
<td>0.45</td>
</tr>
<tr>
<td>•</td>
<td>12.5</td>
<td>15</td>
<td>0.6</td>
</tr>
<tr>
<td>•</td>
<td>12.5</td>
<td>15</td>
<td>0.45</td>
</tr>
<tr>
<td>□</td>
<td>15</td>
<td>20</td>
<td>0.6</td>
</tr>
<tr>
<td>□</td>
<td>15</td>
<td>20</td>
<td>0.45</td>
</tr>
</tbody>
</table>
FIG. 6.28 Effect on flow function at elutriation limit of reducing the exhaust port diameter

Sand bed 500-710 microns

<table>
<thead>
<tr>
<th>KEY</th>
<th>BED MASS kg/m²</th>
<th>BED DEPTH mm</th>
<th>$\phi_e/\phi_d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>+</td>
<td>10</td>
<td>10</td>
<td>0.6</td>
</tr>
<tr>
<td>@</td>
<td>10</td>
<td>10</td>
<td>0.45</td>
</tr>
<tr>
<td>■</td>
<td>12.5</td>
<td>15</td>
<td>0.6</td>
</tr>
<tr>
<td>□</td>
<td>12.5</td>
<td>15</td>
<td>0.45</td>
</tr>
<tr>
<td>▲</td>
<td>15</td>
<td>20</td>
<td>0.6</td>
</tr>
<tr>
<td>△</td>
<td>15</td>
<td>20</td>
<td>0.45</td>
</tr>
</tbody>
</table>
FIG 6.29 Effect on operating range of changing the particle size range and the bed thickness

CENTRIFUGAL ACCELERATION AT BED SURFACE $n \times g$

<table>
<thead>
<tr>
<th>KEY</th>
<th>BED MASS $kg/m^2$</th>
<th>BED DEPTH $mm$</th>
<th>SAND SIZE RANGE $\mu m$</th>
</tr>
</thead>
<tbody>
<tr>
<td>+</td>
<td>10</td>
<td>10</td>
<td>500 - 600</td>
</tr>
<tr>
<td>Ω</td>
<td>10</td>
<td>10</td>
<td>500 - 710</td>
</tr>
<tr>
<td>⋄</td>
<td>12.5</td>
<td>15</td>
<td>500 - 600</td>
</tr>
<tr>
<td>⋄</td>
<td>12.5</td>
<td>15</td>
<td>500 - 710</td>
</tr>
<tr>
<td>■</td>
<td>15</td>
<td>20</td>
<td>500 - 600</td>
</tr>
<tr>
<td>□</td>
<td>15</td>
<td>20</td>
<td>500 - 710</td>
</tr>
</tbody>
</table>
FIG 6.30  Effect on operating range of changing the minimum particle size

CENTRIPETAL ACCELERATION AT BED SURFACE.

<table>
<thead>
<tr>
<th>KEY</th>
<th>BED MASS kg/m²</th>
<th>BED DEPTH mm</th>
<th>SAND SIZE RANGE μm</th>
</tr>
</thead>
<tbody>
<tr>
<td>+</td>
<td>10</td>
<td>10</td>
<td>500 - 710</td>
</tr>
<tr>
<td>⊕</td>
<td>10</td>
<td>10</td>
<td>600 - 710</td>
</tr>
<tr>
<td>●</td>
<td>12.5</td>
<td>15</td>
<td>500 - 710</td>
</tr>
<tr>
<td>◆</td>
<td>12.5</td>
<td>15</td>
<td>600 - 710</td>
</tr>
<tr>
<td>■</td>
<td>15</td>
<td>20</td>
<td>500 - 710</td>
</tr>
<tr>
<td>●</td>
<td>15</td>
<td>20</td>
<td>600 - 710</td>
</tr>
</tbody>
</table>
FIG 6.31  Effect on operating range of reducing the exhaust port diameter

CENTRIPETAL ACCELERATION AT BED SURFACE $n \times g$

<table>
<thead>
<tr>
<th>KEY</th>
<th>BED MASS $\text{kg/m}^2$</th>
<th>BED DEPTH $\text{mm}$</th>
<th>$\varphi_e/\varphi_d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>+</td>
<td>10</td>
<td>10</td>
<td>0.6</td>
</tr>
<tr>
<td>$\Phi$</td>
<td>10</td>
<td>10</td>
<td>0.45</td>
</tr>
<tr>
<td>$\bullet$</td>
<td>12.5</td>
<td>15</td>
<td>0.6</td>
</tr>
<tr>
<td>$\Phi$</td>
<td>12.5</td>
<td>15</td>
<td>0.45</td>
</tr>
<tr>
<td>■</td>
<td>15</td>
<td>20</td>
<td>0.6</td>
</tr>
<tr>
<td>$\bullet$</td>
<td>15</td>
<td>20</td>
<td>0.45</td>
</tr>
</tbody>
</table>
FIG 6.32 Pulsation phenomenon in the RFB

Quiescent bed

Bubbling bed

see text for operating conditions
FIG 6.33 Fluidisation index required for good radial mixing

![Graph showing fluidisation index requirements for good radial mixing. The graph includes a scale for applied acceleration at the distributor (n x g) and a key for sand particle size range (μm). The key indicates different symbols for different size ranges: + for 180 - 250 μm, O for 250 - 355 μm, and Δ for 355 - 500 μm. The graph also indicates satisfactory and inadequate radial mixing.]
FIG 6.34 Axial coal distribution when fuel particles were dropped on to the combustor baseplate
FIG 6.35 Axial coal distribution when fuel particles were introduced on to the middle of the bed
FIG 6.36  Spreader feeder

DIMENSIONS - mm

h = 20, 40, 60

d = 100

r = 100
FIG 6.37 Coal feeder head designs
FIG 6.38 Vortex capture of elutriated material

INDUCED VORTEX

ELUTRIATED PARTICLES

INNER CHAMBER

OUTER CHAMBER

SUCTION

COMBUSTOR
TOP PLATE
Chapter 7

Experiments at Elevated Temperature

7.1 Introduction

During the period of time in which the investigations at atmospheric temperature were taking place, the combustor described in Chapter 4 was manufactured. Combustion trials commenced as soon as a minimum rig facility was available. The initial rig did not include a coal feeding system or the venturi gas scrubber and thus trials were limited to combustion of propane at atmospheric pressure. Nevertheless, these trials allowed evaluation and development of the rotating labyrinth seal, the specification for which no proprietary seal manufacturer could meet. Further, an investigation of possible start-up techniques was made because known methods of start-up did not give the necessary degree of control. The latter study was crucial to subsequent work on coal combustion. These tests were conducted with a duct fitted, that exhausted directly into the main exhaust chimney. No valves or nozzles were installed that would cause pressurisation of the combustor.

In due course the full rig facility was constructed and further combustion trials were conducted. These trials were therefore under conditions nearer to those expected in a practical pressurised installation.
7.1.1 Test Objectives

The objectives of the combustion trials were as follows:-

i) To determine a reliable start-up technique for gas combustion in a confined RFB, and to check the operation of the air cooled labyrinth seal.

ii) To determine the criteria governing transition from propane combustion to self sustaining coal firing and establish a satisfactory changeover sequence.

iii) To investigate the combustion performance when using coal as fuel, at atmospheric and elevated pressures.

iv) To measure exhaust gas composition during coal combustion and attempt to relate these measurements to operating parameters.

7.1.2 Overview of Combustion Trials

One method of raising the bed temperature from cold is to burn premixed propane and air in it. Broughton (52) showed that the following sequence gave reliable start-up in a stationary bed.

i) Adjust the fluidising air velocity to $U_{mf}$

ii) Start igniter or pilot flame

iii) Increase propane flow to give an air to fuel ratio that is approximately stoichiometric.
This stabilised the flame on the surface of the bed and the particles thereby heated.

This method was adapted by Metcalfe (41) for successful start up of RFB when the exhaust port was open to the atmosphere. However, when tried with the present rig two main problems arose:

1) Flashback or burning back into the plenum.

ii) The flame stabilised on the coal feeder head instead of on the bed surface.

The former led to overheating and damage to the distributor whilst the latter prevented the bed reaching an adequately high temperature for coal combustion. A re-appraisal of distributor design was therefore initiated to overcome the integrity problems.

Combustion trials were undertaken in a 6" diameter, conventional, shallow fluidised bed in order to evaluate the feasibility of using a "stand pipe" type distributor, shown previously in Fig 4.3. When a design having satisfactory mechanical integrity had been found, a cylindrical distributor was designed and constructed. Concurrently, cylindrical distributors fabricated from porous stainless steel sheet were also manufactured by Pall Filtration UK Limited, Portsmouth. These distributors were completed only a short time before the end of the project and so the scope of experiments using these distributors was limited.

It was found that sufficient heating of the bed using gas firing,
when using the stand pipe distributor, could not be achieved due to the
high gas velocities required to achieve fluidisation of the bed material
composed of 1 mm particles. A premixed flame would not stabilise in
the combustor. Further, relatively high fluidising velocities were
expected to be necessary for distributor cooling due to the small
internal surface area of the air passages.

Heating of the bed was, therefore, attempted using a gas torch
mounted in the freeboard space. Although a heat release of 25 KW was
available from the propane supply it was found that insufficiently high
bed temperatures could be achieved for self sustaining coal firing to be
initiated. Tests with the stand pipe distributor were, thus, discontinued.

A very small number of combustion trials were undertaken with the
porous distributor installed. During these trials it was found
necessary to operate with all of the exhaust ducting removed. It was
found that stable propane flames could be attached to the bed surface
and that control of bed temperatures was relatively easy. However,
when transition to coal firing was attempted repeated failures
occurred. The bed temperature either fell rapidly and could not be
retrieved without the use of premixed propane or rose uncontrollably
resulting in extensive bed fusion.

7.2 Initial Rig Set-Up

The initial rig configuration is shown diagrammatically in Fig.
7.1. The only physical differences between this rig and that used for
the cold flow work was that all the ducting was constructed from steel
piping rather than plastic tube and that the combustion space was enclosed by metal end plates rather than perspex. A short length of water-cooled ducting, mounted above the exhaust port, provided a means of holding igniters, pressure probes, thermocouples and the gas sampling probe. This ducting was connected to the main exhaust chimney by a length of flexible steel tubing, similar to that used for central heating flues.

Early tests were carried out using the Secomak 492/2 fan as an air supply. When ducting work was completed, all tests were carried out using the Howdenair AW 250 compressor as the air supply. The latter gave a considerably wider operating envelope for the air supply and thus the limits for test work were no longer set by the limits of the air supply.

All operating pressures during this phase of the work were measured using water tube manometers and combustor operating temperatures were measured using Chromel-Alumel thermocouples, mounted at 30° displacements on the axial centre line of the distributor, and connected, via a slip ring assembly (show in Fig. 7.2) to a Comark Series 5000. Temperatures upstream and downstream of the combustor were measured using Cr-Al thermocouples connected to a Comark Series 3000 digital thermometer.

The ignition source chosen for the experiments was a high tension spark produced between two copper electrodes, in the first few runs, and subsequently between two spark plugs having 12" long central electrodes. These are shown in Fig. 7.3. High tension supply came from an encapsulated transformer giving a rated output of 10 KV at
.025 ma. This was sufficient to spark across a gap of approximately 20 mm between the electrodes with sufficient energy release to ensure ignition of a stoichiometric propane/air mixture.

Pressure tappings in the inlet ducting and in the RFB combustor were connected to water and mercury manometers and to Budenberg test gauges giving the capability of measuring pressures between a few millimetres of water and 1000 kPa. The long metal tube shown in Fig. 7.3 is the freeboard pressure tapping.

7.3 Initial Combustion Trials

7.3.1 Scope

This work was designed with two objectives:

i) To test, and develop if necessary, the air cooled labyrinth seal on the combustor exit port described in Chapter 4.

ii) To establish the operating criteria for controlled heating of the bed from ambient to a satisfactory operating temperature using the combustion of a premixed propane/air mixture in the combustor.

These objectives were necessarily linked and investigated concurrently in the first series of combustion trials. The first stages of the experimental method were very similar to those used in the cold flow work. The required mass of sand bed was placed in the combustor and rotation started. The bed was then spread across the distributor using a combination of centrifugal action and a low airflow. The bed was
fluidised to ensure even distribution of material and then the air flow was reduced to a predetermined level dependent upon the desired start-up conditions. This was, as determined by Metcalfe (41), that the initial superficial gas velocity should be approximately \( \frac{1}{4} U_{mf} \) (where \( U_{mf} \) is the minimum fluidising velocity of the particles under cold conditions).

When the required airflow was achieved, the ignition source was switched on, a visual check made on the spark and propane flow started at approximately stoichiometric conditions for the applied airflow. When ignition occurred, as indicated by a steady rise in the temperature of the exhaust gas stream, the air and gas flows were adjusted in an attempt to stabilise the flame front on the bed surface. If this was achieved the bed temperatures would start to rise rapidly and gas and air flows would be adjusted to control the rate of temperature rise.

During the first few runs problems were encountered with maintaining constant rotational speed of the distributor as the exhaust gas temperature rose. It was soon realised that the labyrinth seal was overheating and rubbing on its containment.

Various modifications were investigated as a means of relieving the problem. Finally it was found necessary to adopt a separate air supply for cooling purposes.

During this development period a number of problems were encountered with flame front control, flashback into the plenum chamber, overheating and fusion of the bed and overheating of the distributor. A number of parameters were varied in an attempt to obtain better control of the gas
combustion. These were: sand particle size range, initial superficial air velocity, gas:air ratio and position of igniters.

Details of the trials and findings can be found in succeeding sections.

7.3.2 Fluidisation Experiments

These trials were performed in order that the fluidisation characteristics of the selected bed materials and Conidur distributors could be identified. The results are shown in Fig.s 7.4 to 7.6. Tables 7.1 and 7.2 show details of the Conidur distributors and bed materials respectively.

The principal difference between the distributors used in this part of the project and those used in the earlier work was that of hole size. In the cold flow trials the distributor hole size was 400 microns whereas in the combustion trials the hole size was 250 microns. Further, the open areas were approximately 7% and 2% respectively. The smaller hole size allowed smaller bed particles to be used whilst the smaller open area was expected to prevent flash back of flame into the plenum chamber.

The information gathered on minimum and fully fluidised flow conditions were included in the data used to generate the correlations for $Re_{mf}$ and $Re_{fs}$ in Section 6.4.3.3.

7.3.3 Air-Cooled Labyrinth Seal Development

This development work was carried out in conjunction with the initial
start-up trials. A mass of sand, sufficient to give a bed mass distribution of 12.5 kg/m², was introduced into the combustor and rotation started to give an applied acceleration of 10g. Distribution of the bed was achieved by fluidisation. The high voltage spark was switched on and a visual check made on its operation. Airflow was then started at a rate to give a velocity approximately equal to 1/4 \times U_{mf} and propane fed at a rate to give fuel-lean conditions (the actual volumetric air to propane ratio used was about 50:1). Propane flow rate was then increased until ignition occurred. This method was found to be more controllable than starting with a rich mixture and tended to give more gentle "ignition pops" which put little stress on the manometer etc.

In the first few runs the spark gap was positioned approximately 2-3 cm from the bed surface. With ignition occurring in this position there was a tendency for the flame front to move downstream and stabilise in the air-cooled chimney and not upstream to stabilise on the bed surface. As a result, the air-cooled chimney was seen to glow red hot and start to rub in its containment due to over-expansion.

Re-positioning the spark gap to approximately 1 cm above the bed surface and operating with slightly lower airflow rates reduced the tendency for the flame to move downstream but did not always prevent the flame stabilising downstream of the combustor. There seemed to be a very delicate balance necessary between airflow rate, fuel/air ratio and igniter position which could not be controlled with sufficient reliability.

In order to prevent serious damage to the labyrinth seal on the
outside surface of the air-cooled chimney two modifications were tried in an attempt to improve cooling performance.

The first modification involved fitting circular rings machined from high carbon steel into the labyrinth seal channels as shown in Fig. 7.7. It was hoped that the inclusion of these rings would reduce air leakage through the seal and increase the cooling flow through the film. However, it was found that there appeared to be insufficient change in the airflow balance to improve cooling rates significantly.

The second modification tried was an increase in the film cooling hole area. Each of the original 1 mm diameter holes was drilled out to 1.8 mm diameter increasing the cooling flow by about 150% from .00084 kg/s to ca. .0021 kg/s.

Some improvement was found in the ability of the chimney to operate in elevated temperatures without binding in its containment. Exhaust temperatures up to about 1000°C did not cause interference but when temperatures rose above this level, as happened frequently during the initial start-up sequence, some interference could still occur.

It was evident that an independent control of cooling airflow would be necessary and thus separate cooling flow ducting was installed as shown in Fig. 7.8.

It was found that, when using controlled cooling flow rates of approximately .0026 kg/s, temperatures as high as 1300°C in the exhaust stream would not cause over expansion of the chimney. Fig. 7.9 shows the air-cooled chimney after a number of combustion runs. The regions
where the cooling film and exhaust products began to mix is evident and emphasises the need for careful cooling film design.

7.3.4 Start-Up Trials

Problems with flame stabilisation were described in Section 7.3.3. These problems occurred, initially, when using beds of sand having particle sizes in the range 355–500 microns fluidised on the cylindrical Conidur distributor and attempting start-up at fluidisation velocities equal to $1/4 \times U_{mf}$ (cold).

Adjustment of the spark gap position alleviated the stabilisation problem to some extent but was not totally reliable as the gap position could not be controlled satisfactorily during installation due to the long electrode length.

It was evident that the gas/air mixture velocity in the region of the spark gap at the chosen start-up conditions was close to the natural flame propagation speed and had to be reduced in some manner. The easiest option was to reduce the gas/air mixture volume flow rate. This was tried with the flow rate reduced to about 20% of $U_{mf}$ (cold). The flame front position, as indicated by thermocouples in the combustor, was now found to stabilise near the bed. However, persistent problems were found with flash back into the plenum chamber.

The actual flash back path could not be determined. It is possible that the flame flashed through the labyrinth seal or cooling film holes; however, the velocities in these regions were relatively high and the flow should have been propane free (although there would, inevitably,
have been some leakage from the plenum). This path is, therefore, discounted. The more likely flash back path was through the bed and distributor even though the mixture velocity through the holes in the distributor should have been about 1.3 m/s. The reason that this is believed to have been the flash back path is that very high bed temperatures were achieved, locally; often in the region of 1200°C. Local fluidisation in these regions would have lead to high heat transfer rates from bed to distributor. High (measured) distributor temperatures, ca 600°C could have ignited the propane/air mixture in the plenum. It was, therefore, concluded that the problem was one of pre-ignition rather than true flash back.

It was obvious that flow rates had to be increased in order to restore adequate distributor cooling. In order to do this, but still satisfy the need for low gas/air velocities in the free-board, two free-board taper rings were designed and manufactured, Fig. 7.10.

These simple devices were fitted inside the distributor to give 50° diverging bed walls and free-board and thus, at constant gas temperature, a radial flow velocity that reduced with reducing radius.

When fitted with these rings the effective distributor area was reduced by 50%. This meant that beds of much smaller mass could be accommodated. It was anticipated that temperature control would be more difficult due to the reduced thermal capacity of the bed and thus very fine control of fuel feed rates would be necessary.

Start-up was attempted with the gas:air mixture velocity again at about 1/4 * \( U_{mf} \) (cold) at the bed surface. It was then found that
stabilisation of the flame front near the bed surface was much more easily achieved and the tendency for the flame to travel downstream was considerably reduced. Nevertheless, control of bed temperature was still found to be extremely difficult. The tendency for local high temperatures and resulting bed fusion did not reduce and the original distributor, fabricated from Conidur sheet having a thickness of 0.50 mm, became severely buckled. A second distributor, fabricated from 2 mm thick Conidur sheet, was used in the next series of trials in which attempts were made to achieve coal combustion in the RFB.

Table 7.3 shows the operating conditions for which reasonable flame front control was achieved in the initial start-up trials. These data do not represent conditions for controlled gas combustion within the bed.

Uniform bed temperatures were found to be impossible to attain; it was not unusual to find one part of the bed at a temperature of about 150°C whilst other parts were near fusion temperature. Restoring a uniform bed temperature from these conditions by variation of air or propane flow rate was never achieved. The "best" conditions attained, with the 2mm thick distributor, gave bed temperatures between about 915°C and 1030°C. The non uniformity can be seen clearly in the polar plot shown in Fig. 7.11. These temperatures were not steady but it was felt that coal combustion might be possible with these starting conditions. Table 7.4 shows the operating conditions for which the above bed temperatures applied.

These temperatures were very sensitive to variations in air and gas flow rate. Attempts to increase fluidising velocity at constant bed temperature by increasing the air flow rate and gas flow rate
simultaneously resulted in rapid quenching of the bed temperature whilst attempts to first raise the bed temperature to an even 1050°C by increasing the gas flow rate usually caused significant portions of the bed to fuse. Coal combustion was, therefore, attempted at the low fluidising velocity and low initial bed temperature.

7.3.5 Coal Combustion

A number of trials were conducted to determine the criteria for transition from gas-only combustion to coal combustion using anthracite fuel of 1-3 mm particle size range. These trials were conducted in the combustor fitted with the 2 mm thick Conidur distributor and the combustor taper rings. The following is a description of the final trial which came closest to achieving steady state coal combustion.

Initial bed heating was achieved using propane combustion (air flow rate: - 304 l/m, propane flow rate: - 19 l/m; AFR: - 16:1; bed particle size range: - .355 - .500 mm). The bed stabilised at a temperature of about 750°C ± 50°C. Coal feeding was started at 84 g/m and maintained for about 10 seconds. The bed temperature was seen to rise rapidly for about 15 seconds and then become steady at 830°C after about 2 minutes. Coal feeding was restarted at 84 gm/m in an attempt to further increase the bed temperature, however it remained steady at 840°C. The coal feed rate was reduced to 51.6 gm/minute and bed temperature rose to 870°C in about 30 seconds.

Propane flow was now shut off and after a delay of about 5 seconds the bed temperature was seen to fall rapidly. Gas flow was restarted at 14.6 l/m (AFR = 26:1) when the bed temperature had fallen to 850°C.
The bed temperature rose to 940°C in 2 1/2 minutes and the coal feed rate was then increased to 66 gm/m.

The propane flow was slowly reduced to 11.4 l/m and the bed temperature stabilised at 960°C. Coal feed rate was increased to 84 gm/m and when the bed temperature reached 980°C the propane flow was shut off. Once again the bed temperature was seen to fall rapidly.

It became obvious that true steady-state coal combustion would not be possible. It appeared that the bed temperature could only be maintained at the desired level by use of the propane as support.

Two possibilities present themselves as explanations for the observed phenomena. The first possibility is that when the propane flow was shut off the sudden drop in bed temperature led to ce-fluidisation of the bed, reduction in solids mixing and reduced gaseous diffusion to and from the hot coal particles with the result that steady combustion of the coal was lost. This dynamic problem requires separate and detailed investigation; identification of this problem is believed to be an important finding.

The second possibility is much more simple. Coal feed rates during the trials were, in general, higher than those used by Metcalfe under similar operating conditions in similar beds. It is therefore possible that failure to attain steady-state combustion of coal was due to excessive masses of coal in the bed which did not reach sufficiently high temperatures for self-sustaining combustion to occur.

There is no clear evidence to support either of the possibilities and
the problem could only be resolved by further investigations using more comprehensive monitoring of combustion conditions including direct observations of the bed.

7.3.5.1 Bed Fusion

Inspection of the combustor at the end of the final trial revealed large masses of fused sand, ash and anthracite particles. Figs. 7.12, 7.13 and 7.14 show two of these fused masses. The "edge-on" view shown in Fig. 7.12 shows the fused solid material to be the same shape as the bed, one side being "moulded" by the distributor and the other by the bed free surface. Both Fig. 7.12 and 7.13 (another "edge-on" view from a different part of the bed) show that fusion occurred through the entire bed depth. The thickness of the fused mass is approximately 2 cm which is roughly equal to the, fluidised, bed thickness. It is also clear in Fig. 7.13 that radial mixing of the coal particles had occurred before fusion took place, because of the presence of partly burnt coal particles visible in close contact with the distributor.

7.3.5.2 Distributor Mechanical Integrity

During the final trial, the distributor temperatures were seen to rise above 700°C. Peak temperatures recorded were 816°C, 946°C, 783°C and 738°C at equispaced intervals around the distributor circumference. It was suspected that these high temperatures had caused damage to the distributor, so the rig was dismantled and the distributor inspected.

The inspection of the distributor showed that severe buckling had
occurred, as shown in Fig. 7.15(a). Air flow checks showed that the open area of the distributor had increased by 40% indicating that burning of the perforations had also occurred. It was therefore decided that a different design of distributor should be used in the RFB.

Fig. 7.15(b) shows a distributor used by Metcalfe (41), photographed by the present author. It is evident that local hot areas had also occurred with this distributor. Metcalfe reports distributor temperatures in the region of 400°C although his thermocouples were not secured tightly against the distributor. It would appear that more extensive sensing is desirable for satisfactory monitoring of distributor temperatures.

There are limits to the number of thermocouples which could be installed and it might be better to design for greater uniformity of combustion by accepting a higher pressure loss through the distributor and a more robust distributor which would not distort significantly if it became hot.

Accordingly, a "stand pipe" distributor and a distributor made from porous stainless steel filter material were tried.

7.4 Development of the Stand Pipe Distributor

7.4.1 Mechanical Design

During the literature survey it had been discovered that two popular types of distributor used in various large-scale fluidised bed rigs and in commercial package boilers were the stand pipe and the tuyere
distributors. These designs have the advantage of leaving a layer of unfluidised bed between the air entry holes and the distributor plate.

Ancillary tests were conducted using a simple flat plate stand pipe distributor which was designed and manufactured from 1/4" mild steel plate and 5/16" BSF screws. Initially the stand pipes were made using design 'a' shown in Fig. 7.16. Each screw had a central passage 1/8" (3mm) drilled through the shank up to the centre of the head. Intersecting with this latter position, three 1/16" (1mm) radial passages were drilled. The effect was to produce radial jets of air effluxing into the bed about 2 mm above the distributor plate. In this way a thin insulating layer of sand was produced between the hot bed and the plate.

This distributor was fitted into a small conventional fluidised bed, shown in Fig. 7.17, and the containment filled to a depth of 2 cm with .180 - .250 mm sand. Air was fed, via a rotameter (type 24K), and control valve, from the shop air supply and propane fuel gas fed, via a rotameter (type 7K), from a regulated bottle supply. Chromel-Alumel thermocouples connected to a Comark 3000 digital thermometer were used to measure the temperatures of the inlet air and gas flows, the bed temperature and the distributor top surface temperature.

7.4.2 Combustion of Gas and Coal

Steady-state combustion of propane at 1100°C was achieved using the start-up technique described by Broughton (52). The bed was fed with air until the bed surface was seen to be just fluidised. Propane gas flow was started and ignited just above the bed surface. The propane
flow rate was then adjusted until the flame stabilised on the bed surface. A bed temperature increased, the propane flow rate was gradually reduced to prevent excessive temperatures.

The various operating parameters at start-up and steady-state are shown in Tables 7.5 and 7.6.

Coal combustion was achieved simply by bringing the bed temperature to about 750°C and introducing a small mass of coal particles (ca 15 grams). When the bed temperature started to rise, after an initial fall of ca 35°C during the coal heating period, the propane flow was stopped quickly. The bed temperature was maintained at a roughly steady value by feeding 5 or 10 gram batches of coal every 30 seconds depending upon the chosen bed temperature. Feeding coal in 5 gram batches produced a bed temperature of ca 890°C ± 15°C whilst feeding 10 gram batches allowed the bed temperature to rise to ca 935°C. These two features are illustrated in Fig.s 7.18 and 7.19 respectively. Also shown in these figures are the respective distributor plate temperatures. In Fig. 7.18 the distributor plate temperature is seen to be between 210°C and 250°C below the bed temperature whilst in 7.19 the distributor plate temperature is seen to be about 300°C below the bed temperature at very similar fluidising velocities. The difference between the two distributor temperatures is attributed to an extra layer of insulation, in the latter case, provided by a ceramic wool blanket, about 3mm thick, laid on the top surface of the distributor plate, shown in Fig. 7.16.

The small difference between the distributor plate temperatures is due to the fact that the plate is heated by conduction through the stand
pipe studs and thus extra layers of insulation afford little advantage. The indirect heating of the distributor was confirmed by a simple heat transfer analysis which is described in Appendix A.

Upon inspection, it was found that severe damage had been sustained by some of the stand pipe studs and the whole assembly was scaled over by a layer of black oxide. This is shown in Fig. 7.20. The scaling was so extensive that many of the air efflux holes were nearly totally blocked. It was evident that stainless steel studs would be necessary.

Four stainless steel studs, design "b" in Fig. 7.17, were manufactured and fitted to the distributor plate. The bed was fired as before and run through a number of heating and cooling cycles during a period of thirty minutes. Subsequent inspection of the distributor showed that no damage or scaling had been suffered by the stainless steel studs.

The results of these tests suggested that a stud standpipe distributor might work in the RFB. The design adopted is shown in Fig. 7.21. A "free hanging" support was adopted to avoid any possibility of buckling due to overheating. It was decided that the stand pipes would be manufactured from stainless steel screws and the distributor cylinder, carbon steel. The length of the stand pipes was chosen such that a 6 mm layer of ceramic insulation, either wool blanket or cement, could be installed next to the distributor cylinder. After consultation with various manufacturers, a ceramic cement manufactured by Mackenzie Ceramics (available through trade outlets) was chosen. This cement was chosen because its maximum operating temperature was quoted at 1260°C and it could be applied by brushing or trowelling.
Difficulty was experienced in the manufacture of the stainless steel stand pipes due to the hardness of the material and thus installation of the finished distributor, shown in Fig. 7.22, was delayed for some months. During this period theoretical studies were undertaken to investigate some of the phenomena experienced in earlier parts of the project. This work is described in Chapter 8.

Air flow checks were carried out determine leakage rates and the effective open area of the distributor. It was found that the pressure loss across the distributor could be described accurately, (.9999 correlation coefficient up to 8.4% pressure loss), by the equation:-

\[ \Delta P_{\text{d}} \times 100 = 231024339.8 \times \frac{Q_{\text{p}}^{2.328}}{P_{\text{p}}} \quad \text{7.1} \]

This corresponds to an effective open area of approximately 0.5%.

7.4.3 RFB Stand-Pipe Distributor Tests

7.4.3.1 Fluidisation

Limited fluidisation data was gathered for a bed with particles in the size range 0.5-0.71 mm. These are shown in Figure 7.23. It is evident that the overall pressure loss characteristic is similar to those seen in Chapter 6 but that the condition when the bed becomes fully fluidised is not easily identified. This is due to the stand-pipe distributor having an effective open area of about 0.5% compared with between 6% and 7% open area of the Conidur distributors used in the cold flow work and 1.6% and 6.3% of the Conidur used in the
initial combustion trials.

One problem noted with the stand-pipe distributor was that fines in the sand quickly blocked the air jet holes thus increasing the pressure loss for a given flow rate and disturbing the fluidisation. It was decided therefore that much larger particles should be used in the bed for combustion trials. Particles in the range 1.18 - 1.4 mm were chosen as a convenient size for the combustion trials.

7.4.3.2 Combustion Trials

It was obvious that, with the very large particles chosen for the bed, start-up would be better attempted by using a pilot torch directed at the bed surface rather than use premixed propane and air as this might allow the bed to be fluidised during the heating procedure. The restriction of keeping average gas velocities through the bed below the flame propagation speed was felt to have too high a risk of causing flash back and damaging the distributor.

A suitable propane torch was supplied, free of charge, by Wellman-Selas Limited, Manchester. The 1/2" diameter premix gas torch head had a rated output of 5KW at 10" WG gas pressure but was found to give about 20-25KW output when driven by higher pressures (ca 100" WG) from a propane bottle.

The torch was mounted in the combustor with the flame directed at about 45° to the bed surface. It was hoped that this would ensure adequate flame impingement on the bed surface whilst preventing significant interference with the fluidisation.
The first trial using this rig showed that higher heat outputs than available from the torch would be necessary for heating the bed to temperatures in the region of 900°C. Table 7.7 shows the operating conditions attained with maximum heat output from the torch.

Visual observation of the torch flame during the tests showed that the gas flame was being swept away from the bed surface by the strong vortex flows in the freeboard. This prevented direct contact heating of the bed particles by the flame.

Now, the expected maximum heat output from the bed at atmospheric pressure was expected to be of the order of 30kW. The lack of success in heating the bed with a torch having a heat output similar to maximum expected from steady-state combustion in the bed suggests that, in order for this approach to show greater success, some development of the torch may be necessary. Since the torch was gas fuelled the flame was not very radiant and it is thought that a much greater possibility of success would be gained by utilising a liquid fuelled torch. This approach was beyond the scope of the current work.

Attempts to boost bed heating by injecting propane into the fluidisation air in the plenum were unsuccessful. Whilst bed temperatures increased generally by about 350°C - 450°C it was found that there was still a significant difference between the bed temperature at the top of the combustor (ca 850°C) and the bed temperature at the bottom (ca 600°C).

No further testing was done with this distributor as the component thought necessary for successful start-up performance, viz a liquid
fuelled pilot torch, was not available.

7.5 *Porous Distributor*

7.5.1 *Mechanical Design*

In Chapter 4 the arguments against using a porous distributor were discussed in detail. However, as the proposed alternatives, i.e. pierced sheet and stand-pipe distributors had been found to have operational problems, it was decided that a porous distributor should be tried.

The design of the distributor is shown in Figure 7.24. As with the stand-pipe distributor, a hanging basket type of installation was used. Sintered stainless steel, manufactured by Pall Filtration (UK) Limited, Portsmouth, was selected as a suitable porous material. The type of stainless steel chosen was grade 316 and the distributor was manufactured from rolled porous sheet, 3 mm thick. The designation of the sheet used was Grade "F". This grade has a relatively high pressure loss characteristic but the density of the sintered material was felt to be an aid to mechanical integrity. The pressure loss characteristic is shown in Figure 7.25. It is clear that the relationship between pressure loss and flow function is not a simple power law and it was necessary to employ a quadratic representation as follows:

\[
\Delta P \% = 0.5797 + 315310_p + 90448040Q_p^2
\]

\[\frac{\Delta P}{P}\]

7.2

This model is a further example of the use of the flow function
described in Appendix E. Over the range of flow function tested, the model represented the pressure loss characteristic with a correlation coefficient of 0.999.

The rig used for the testing of the porous distributor initially included all the items described in Chapter 5 and is shown schematically in Figure 7.26. As fully pressurised operation was not attempted the exhaust nozzle in the silencer section was not fitted. During the limited test period all features of the rig performed satisfactorily.

However, it was found that with the fully enclosed system, start-up with premixed propane and air was not possible as flames tended to stabilise in the exhaust ducting rather than on the bed surface in much the same way as encountered when the pierced steel distributors were installed. As a result start-up was attempted with the exhaust ducting removed and the exhaust port of the combustor open to the atmosphere.

7.5.2 Fluidisation Experiment

As the time available for the conclusion of the project was limited, fluidisation of only one sand bed, having particles in the size range 0.18-0.25 mm, was investigated. The results of this experiment are shown in Figure 7.27. The effect of the high pressure drop of the distributor is clear; fluidisation is seen to occur at relatively large values of plenum flow function when compared with the results obtained for larger particles fluidised on pierced steel distributors and the overall pressure losses at and above the fully supported condition are two to three times the corresponding values for the larger particles. Clearly, the penalty for using a sintered steel distributor would be
high pumping power and reduced overall thermal efficiency of the plant. It would be necessary, in a full scale RFB combustor, to employ a much lower pressure loss material that the one employed here and obtain adequate mechanical strength through careful mechanical design.

7.5.3 **Combustion Experiments**

Both gas and coal combustion were attempted in the RFB fitted with the porous distributor. Gas combustion was found to be relatively successful and remarkably controllable. However, satisfactory coal combustion was not achieved.

7.5.3.1 **Gas Combustion**

The initial combustion trials on the RFB fitted with the porous distributor were done with the complete ducting fitted. However, it was found to be impossible to achieve a stable flame front on the bed surface when attempting to burn a premixed propane/air mixture. It was decided that a trial should be conducted with the exhaust ducting removed in order to determine whether recirculation of atmospheric air through the exhaust port was important to flame stabilisation.

The propane torch used in the tests on the stand-pipe distributor was used as the ignition source. This was positioned close to the exhaust port and directed into the freeboard zone. The pressure in the freeboard was measured in a long stainless steel probe. Bed temperatures, freeboard temperatures and plenum temperatures were monitored continuously using chromel-alumel thermocouples connected to a multi-pen chart recorder.

247.
Combustion experiments were carried out using a bed of sand, 0.18 -
.25 mm particle size, spread at 12.5 kg/m².

Start up was found to be extremely easy using the technique suggested
by Demircan (40). This required that the propane first be lit as a
diffusion flame at the exhaust port with very little fluidising air.
The airflow was then increased until the flame retreated inside the
combustor and became attached to the bed surface.

Rates of temperature rise were found to be quite modest but the
temperature distribution was non-uniform. Typically, in the upper
portion of the bed, the temperature would rise to about 900°C in about 3
minutes. The temperatures in the middle and lower portions of the bed
would follow slowly rising to similar temperatures after 3-5 minutes.
It is interesting to note that when the temperature in these regions
started to rise, the rate of increase was of the same order of magnitude
as that of the upper portion. The reason for the delay between the
temperature rise in the different portions of the bed is thought to be
linked to the heating time of the metal containment. The total time
from ignition to steady-state was, typically, 20 minutes.

Typical start-up and steady-state operating conditions are shown in
Tables 7.8 and 7.9. Operating the combustor over a wide range of
fluidising velocity and air to fuel ratio was not achieved as it was
found that the combustion performance was found to be very sensitive to
fluidising airflow rate and rotational speed. It was found that, if
the airflow rate was increased to a value greater than the maximum shown
in Tables 7.8 and 7.9, bed temperature could not be maintained even when
more fuel gas was introduced. Rapid cooling of the bed resulted and it
was necessary to employ the full start-up procedure in order to regain satisfactory bed temperatures.

Similarly, increasing the rotational speed above that to give a radial acceleration of 10 gravities was found to cause loss of combustion (probably due to loss of fluidisation). Increasing the fluidising air flow, to maintain fluidisation, and propane flow, to maintain fuel/air ratio, did not solve this problem.

There does not appear to be any fundamental reason for the limited combustion capability and it is believed to have been due to excessive proportions of the measured air and propane flows being lost via the labyrinth seal, because the plenum pressure required to overcome the distributor pressure loss was so high. A comparison of Figures 6.2 and 7.25 shows the difference in pressure loss characteristics of the Conidur and porous distributors.

The fact that substoichiometric air to fuel ratios were usually necessary to maintain bed temperatures can be explained in terms of the combustor design. It is believed that the propane was not fully premixed with the fluidising air on entry to the plenum chamber. The method of propane injection, viz four plain pipes 3" upstream of the plenum, was far from ideal but was very simple to install. It is possible that the propane/air mixture at the entry to the plenum was stratified to the extent that the majority of the propane passed through the upper portion of the bed or was lost via leakage through the labyrinth seal.
Clearly, the start-up and steady-state combustion performance was makedly different to that observed by Metcalfe (41) and Demircan (40). The differences may have been due to the geometry and mechanical design of the combustor but the controlling factors have not been identified.

7.5.3.2 Transient Response

Although the experiments were not designed for the investigation of transient combustion some observations are worth noting. During the start-up sequence the bed temperature was observed to rise slowly just after ignition reaching ca 200°C after about a minute. Temperatures in the upper bed would then start to rise fairly rapidly reaching ca 900°C after a further two minutes. The bed temperature would then need ca 15 minutes to settle to a steady value.

This behaviour is significantly different to the transient behaviour noted by Metcalfe (41) and by Demircan (40) although the start-up technique employed was essentially the same as used by these workers. This suggests, most strongly, that the transient response of the RFB is influenced by the combustor geometry.

Another aspect of the transient response was that, when the propane flow rate was increased during steady-state combustion, in order to increase the bed temperature, the latter did so quite slowly. Typically, after increasing the propane flow rate, the bed temperature would rise from 870°C to 1000°C in about 30 seconds to 1 minute. This is rather slower than the heating rates reported by Metcalfe who found that 15-20 seconds was more a typical heating time.
No firm conclusions can be drawn from this limited information and, furthermore, there is insufficient data from which to suggest possible mechanisms.

7.5.3.3 Coal Combustion

The combustion of anthracite was attempted after some experience had been gained in propane combustion. Three different techniques for transferring from propane to anthracite combustion were investigated.

a) Preheating the bed to 950°C using propane and then attempting gradual changeover to anthracite firing.

b) Preheating the bed to 950°C and then quickly shutting off the gas flow whilst simultaneously starting the coal flow at 50 g/m for 30 seconds and then reducing the flow rate to 30 g/m.

c) Heating a bed, with included a mass of coal, determined from data published by Chakraborty (118), premixed with the sand particles. Preheating was attempted using premixed propane, to a temperature at which the anthracite should have produced self-sustaining combustion, viz 750-800°C.

The results of these trials are summarised in Table 7.10. In the two series of trials described in (a) and (b) the propane combustion conditions were the same as those described in Tables 7.8 and 7.9. In all three series of trials the sand particle size range used was 0.18 to 0.25 mm whilst the coal particle size range used was 1-3.35 mm. An
analysis of the coal is shown in Table 7.11. This coal was chosen as
the low ash content would minimise problems of build-up in the bed and
also dust elutiation. However, the low volatile content produces a
fuel that reacts slowly and this aspect increases the problems of
control during changeover from gas to coal firing or during change from
one steady-state condition to another.

No attempts were made to install fully automatic control as it was
found that a time delay occurred between changes in fuel feed rate
(propane or coal) and observed changes in bed temperature. It was,
therefore, evident that some form of anticipatory control system would
be necessary but time constraints did not allow the development of such
a system.

7.6 Summary of Findings

The findings of the combustion test work fall into two groups,
namely, mechanical design aspects and combustion aspects.

7.6.1 Mechanical Design Aspects

a) The air-cooled exhaust port chimney with its associated labyrinth
seal has been found to operate satisfactorily but required a larger
cooling air flow rate, 0.0026 kg/s, than had been originally expected.
This suggests that the optimum film cooling ring dimensions had not been
used thus preventing the generation of a strong coherent cooling film.
Clearly, further development is necessary to determine cooling
ring design specifications.

b) It has been found that pierced steel distributors are prone to damage by burning and local distortion due to temperatures in the region of 900°C. The choice of air flow rates for cooling purposes and air to fuel ratios to prevent burning back are critical in achieving acceptable life for the combustor.

c) A stand-pipe distributor was tested in a conventional fluidised bed and found to have excellent mechanical integrity provided that the stand-pipes were made from heat resisting stainless steel.

d) Manufacturing a stand-pipe distributor for the RFB was found to be difficult due to the requirement to drill small holes in hard stainless steel. The stand-pipe distributor temperature was noted to be high during combustion trials despite the insulating layer of sand.

e) A porous stainless steel distributor was tested and found to suffer no damage after several hours of exposure to temperatures in the region of 900-1000°C. Further, the distributor design used was structurally strong and did not require external stiffening as did the pierced steel distributors.

f) The encastrate distributor mounting used by Metcalfe has been found to result in severe distributor buckling when the distributor was heated to temperatures above 700°C.

g) A "hanging basket" mounting used for the installation of the
stand-pipe and porous distributors was found to give no mechanical problems and, in fact, avoided some assembly problems associated with the encastré installation.

7.6.2 Combustion Aspects

a) Start-up using premixed propane and air as fluidising gas and ignition above the bed surface has been found to have a number of constraints which are difficult to reconcile. It was found very difficult to match the superficial fluidisation velocity with the requirement to be below the flame propagation speed whilst retaining sufficient cooling capacity, even with very fine control of air and gas flow rates. If the air/gas velocity was slightly high at the bed surface, it was found that the flame tended to travel downstream and leave the combustor. However, if the gas/air velocity was too low, there was a tendency for flash-back into the plenum chamber which resulted in distributor damage.

b) The problems of flame stabilisation were alleviated, to a large extent, by the installation of tapered rings which produced a divergent freeboard of 50° included angle.

c) It was found that, even with controllable flame stabilisation, uniform heating of the bed was very difficult to obtain. There was a tendency for localised rapid heating of the bed which often resulted in infusion of the bed material in these zones. This behaviour was observed with beds composed of sand particles of 355-500 microns and larger.
d) When the porous distributor was installed, sand particles of 180-250 microns could be fluidised and start-up with premixed propane and air was successful when the exhaust port was open to the atmosphere. However, when the combustor was enclosed by exhaust ducting bed heating was again found to be very difficult. It is concluded that successful bed preheating of the bed using premixed gas and air is more dependent upon a steady supply of oxygen to the freeboard in order to give good flame stability and uniform flame distribution rather than the size of bed particle which governs the matching of fluidisation velocity and flame propagation speed.

e) When the stand-pipe distributor was installed, bed heating was attempted by using a gas torch. A maximum gross heat output of 25 KW was found to be sufficient to raise the bed temperature to 600°C. The radiant heat input to the bed was estimated to about 11.5KW. It is concluded that a more radiant heat source is necessary to raise the bed temperature to that required for coal combustion.

f) Steady-state gas combustion was found to be stable and relatively easily controlled whilst using the porous distributor provided that the fluidisation velocity was low and the fuel to air mixture was rich.

g) Control of gas combustion was found to be difficult if the applied acceleration was greater than 10g. It is thought that this is due to loss of fluidisation as the rotational speed was increased and insensitivity of gas to air ratio control as the air flow rate was increased to compensate for loss of fluidisation.
h) Steady, self sustaining coal combustion was not achieved. Several transition methods from gas to coal combustion were attempted but it was found that either the bed temperature fell rapidly or local fusion of sand, coal and ash particles occurred.
TABLE 7.1(a) Cylindrical Conidur Distributors Fitted to RFB Combustor

<table>
<thead>
<tr>
<th>Feature</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter</td>
<td>200 mm</td>
</tr>
<tr>
<td>Length</td>
<td>80 mm</td>
</tr>
<tr>
<td>Pore size</td>
<td>250 microns</td>
</tr>
<tr>
<td>Thickness</td>
<td>0.5, 2.0 mm</td>
</tr>
<tr>
<td>Open area ( \text{inc} C_d )</td>
<td>6.3, 1.6 %</td>
</tr>
</tbody>
</table>

Distributor material: 316 grade Stainless Steel

TABLE 7.1(b)

<table>
<thead>
<tr>
<th>Distributor thickness mm</th>
<th>Pressure loss equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>289932.3 ( q_p^{1.694} )</td>
</tr>
<tr>
<td>2.0</td>
<td>1172576.8 ( q_p^{1.707} )</td>
</tr>
</tbody>
</table>

TABLE 7.2 Silica Sand Beds

<table>
<thead>
<tr>
<th>Sand Particle Sizes mm</th>
<th>Bed Mass Distribution kg/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>.355 - .500</td>
<td>12.5</td>
</tr>
<tr>
<td>.425 - .500</td>
<td>12.5</td>
</tr>
<tr>
<td>.425 - .600</td>
<td>12.5</td>
</tr>
</tbody>
</table>

257.
### TABLE 7.3 Gas combustion Start-Up Conditions

**Conidur distributors**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bed particle size range</td>
<td>0.355 - 0.500 mm</td>
</tr>
<tr>
<td>Applied radial acceleration</td>
<td>10g (at distributor)</td>
</tr>
<tr>
<td>Fluidising airflow rate</td>
<td>0.307 kg/s/m²</td>
</tr>
<tr>
<td>Fluidising velocity (STP)</td>
<td>0.255 m/s</td>
</tr>
<tr>
<td>Propane flow rate (STP)</td>
<td>69 l/m</td>
</tr>
</tbody>
</table>

### TABLE 7.4 Steady State Gas Combustion Operating Conditions

**Conidur distributors**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bed particle size range</td>
<td>0.355 - 0.500 mm</td>
</tr>
<tr>
<td>Applied acceleration</td>
<td>10g at distributor</td>
</tr>
<tr>
<td>Fluidising airflow rate</td>
<td>0.313 kg/s</td>
</tr>
<tr>
<td>Propane flow rate (STP)</td>
<td>17.2 l/m</td>
</tr>
<tr>
<td>Volume AFR</td>
<td>11.8:1</td>
</tr>
<tr>
<td>Fluidising velocity (STP)</td>
<td>0.255 m/s</td>
</tr>
<tr>
<td>Average bed temperature</td>
<td>975°C</td>
</tr>
</tbody>
</table>
**TABLE 7.5**  Gas Combustion Start-Up Conditions

**Stand Pipe Distributor**

<table>
<thead>
<tr>
<th>Bed particle size range</th>
<th>0.18 - 0.25 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air flow rate (STP)</td>
<td>132 l/m</td>
</tr>
<tr>
<td>Fluidising velocity (STP)</td>
<td>0.132 m/s</td>
</tr>
<tr>
<td>Propane flow rate (STP)</td>
<td>7.5 l/m</td>
</tr>
<tr>
<td>Vol' AFR</td>
<td>17.7:1</td>
</tr>
</tbody>
</table>

**TABLE 7.6**  Gas Combustion Steady-State Conditions

**Stand-Pipe Distributor**

<table>
<thead>
<tr>
<th>Bed particle size range</th>
<th>0.18 - 0.25 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air flow rate (STP)</td>
<td>132 l/m</td>
</tr>
<tr>
<td>Fluidising velocity</td>
<td>0.130 m/s</td>
</tr>
<tr>
<td>Propane flow rate</td>
<td>6.1 l/m</td>
</tr>
<tr>
<td>Vol AFR</td>
<td>21.4:1</td>
</tr>
<tr>
<td>Average bed temperature</td>
<td>1100°C</td>
</tr>
<tr>
<td>Parameter</td>
<td>Value</td>
</tr>
<tr>
<td>-----------------------------------------------</td>
<td>----------------------------</td>
</tr>
<tr>
<td>Bed particle size range</td>
<td>1.18 - 1.4 mm</td>
</tr>
<tr>
<td>Bed mass</td>
<td>0.39 kg</td>
</tr>
<tr>
<td>Bed mass distribution</td>
<td>10 kg/m²</td>
</tr>
<tr>
<td>Nominal radial acceleration</td>
<td>5 g</td>
</tr>
<tr>
<td>Fluidising airflow rate</td>
<td>0.01649 kg/s</td>
</tr>
<tr>
<td>Cold fluidising velocity</td>
<td>0.37 m/s</td>
</tr>
<tr>
<td>Torch heat output</td>
<td>25.45 kW</td>
</tr>
<tr>
<td>Radiated heat output (estimated)</td>
<td>11.5 kW</td>
</tr>
<tr>
<td><strong>Bed Temperatures</strong></td>
<td></td>
</tr>
<tr>
<td>Top</td>
<td>550°C</td>
</tr>
<tr>
<td>Middle</td>
<td>300°C</td>
</tr>
<tr>
<td>Bottom</td>
<td>150°C</td>
</tr>
<tr>
<td></td>
<td>At ignition</td>
</tr>
<tr>
<td>--------------------------------</td>
<td>-------------</td>
</tr>
<tr>
<td>Bed Temperature °C</td>
<td>15</td>
</tr>
<tr>
<td>Airflow rate l/s NTP</td>
<td>seal cooling air only</td>
</tr>
<tr>
<td>Propane flow l/s NTP</td>
<td>0.36</td>
</tr>
<tr>
<td>Superficial fluidising velocity m/s</td>
<td>0.0082</td>
</tr>
<tr>
<td>Excess air %</td>
<td>sub stoichiometric</td>
</tr>
<tr>
<td>Bed temperature °C</td>
<td>850</td>
</tr>
<tr>
<td>-------------------</td>
<td>--------</td>
</tr>
<tr>
<td></td>
<td>Min</td>
</tr>
<tr>
<td>Air flow rate l/s (NTP)</td>
<td>3.55</td>
</tr>
<tr>
<td>Propane flow rate l/s (NPT)</td>
<td>0.2</td>
</tr>
<tr>
<td>Superficial fluidising velocity m/s</td>
<td>0.3</td>
</tr>
<tr>
<td>Actual air</td>
<td>0.7</td>
</tr>
<tr>
<td>Stoichiometric air</td>
<td></td>
</tr>
<tr>
<td>Start-Up Method</td>
<td>Observations</td>
</tr>
<tr>
<td>--------------------------------------------------------------------------------</td>
<td>------------------------------------------------------------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Gradual change from propane to coal firing</td>
<td>Extinction of combustion and, eventually, excessive concentration of coal in the bed.</td>
</tr>
<tr>
<td>Sudden change from propane to coal firing</td>
<td>Either a rapid fall in bed temperature and extinction of combustion or bed fusion or loss of bed material due to coal nozzle transport air impinging on the bed surface</td>
</tr>
<tr>
<td>Preheating the bed with propane after precharging the bed with a known mass of coal</td>
<td>Unstable combustion. Equal incidence of either extinction of bed combustion or bed fusion. In the latter, nonhomogeneous mixtures of coal and sand were found in the fused zones.</td>
</tr>
</tbody>
</table>
### TABLE 7.11 Anthracite Composition (Dry Basis)

<table>
<thead>
<tr>
<th>Component</th>
<th>Mass Content %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fixed carbon</td>
<td>85.58</td>
</tr>
<tr>
<td>Volatile matter</td>
<td>7.5</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>3.42</td>
</tr>
<tr>
<td>Ash</td>
<td>6.4</td>
</tr>
<tr>
<td>Sulphur</td>
<td>0.99</td>
</tr>
<tr>
<td>Chlorine</td>
<td>0.02</td>
</tr>
</tbody>
</table>

Calorific value 33460 kJ/kg

The moisture content of "as received" coal was 0.87%.
FIG 7.1 INITIAL COMBUSTOR RIG INSTALLATION
FIG 7.2 IGNITERS AND DOWNSTREAM PRESSURE TAPINGS

266.
FIG 7.4 FLUIDISATION CHARACTERISTICS OF 355-500 MICRON SILICA SAND PARTICLES, BED MASS 12.5 kg/m². CONIDUR DISTRIBUTOR WITH .25mm PORES
FIG 7.5  FLUIDISATION CHARACTERISTICS OF 425-500 MICRON SILICA SAND PARTICLES. BED MASS 12.5 kg/m².
CONDUR DISTRIBUTOR WITH .25mm Pores
FIG 7.6  FLUIDISATION CHARACTERISTICS OF 425-600 MICRON SILICA SAND PARTICLES. BED MASS 12.5 kg/m².
CONIDUR DISTRIBUTOR WITH .25 mm PORES
FIG 7.7 Air cooled chimney modifications
FIG 7.8 Independent cooling air supply for rotating chimney and seal.
FIG 7.9 Rotating assembly after initial combustion trials

Note burn marks and mixing patterns inside

air cooled chimney
FIG 7.10  Combustor with shaper rings installed
All temperatures in °C

- Freesboard temperatures

FIG7.11  Polar plot of temperature in the RFB combustor
during propane combustion
Distributor used in current project

Distributor used by Metcalf(41)

FIG 7.15 Buckled Canidur distributors
Stud installation showing optional insulation

FIG 7.16 Flat plate standpipe distributor
Perspex RFB used in initial mixing trials

Shallow fluidised bed used in combustion trials on different standpipe distributors

FIG 7.17 Equipment used in ancillary fluidisation and combustion trials
FIG 7.18 Temperature history of shallow fluidised bed fitted with standpipe distributor without ceramic insulation. 0.18-0.25 mm sand bed, 1-3 mm coal.
FIG 7.19  Temperature history of shallow fluidised bed fitted with standpipe distributor and ceramic insulation. 0.18-0.25 mm sand bed, 1-3 mm coal.
FIG 7.20  Standpipe distributor after combustion of coal

Note heavy scaling of mild steel tuyeres
FIG 7.21 Standpipe distributor general assembly. Mounting dimensions to suit

RFB
FIG 7.22  Standpipe distributor for RFB
FIG 7.23 Fluidisation characteristics of 500-710 micron silica sand. Bed mass 12.5 kg/m$^2$. Particle density 2630 kg/m$^3$. Standpipe distributor.
FIG 7.24 Porous distributor general assembly. Mounting dimensions to suit RFB
FIG 7.25 Pressure loss Characteristic of the porous stainless steel distributor

Note high pressure loss at modest flow function
FIG 7.27 Fluidisation characteristics of 180-250 micron silica sand. Bed mass 12.5 kg/m$^3$.

Porous stainless steel distributor

$\frac{\Delta P_{d+b}}{P_p} \times 10^5 \times \frac{m}{\sqrt{T}}$

Interpolated characteristics

Fully supported condition

Elutriation
CHAPTER 8

THEORETICAL STUDIES OF OBSERVED PHENOMENA

8.1 Introduction

Although the course of this project was mainly one of design and experimental development it was necessary and useful to make some theoretical assessment of the phenomena and difficulties as they arose.

Three aspects of the operation of the RFB were investigated, viz:–

i) Particle dynamics within the combustor downstream of the bed free surface.

ii) Matching of fluidisation velocity and flame speed at the free surface of the bed.

iii) Fuel-gas bypass in the bed.

8.1.1 Particle Dynamics

In this instance, an understanding of particle dynamics in the freeboard can give an approximation to the likely gas flow conditions at which certain sizes of particles will be entrained in the exhaust gas (and thereby swept out of the combustor) at given rotational speeds. This is of great importance as it limits the combustor operating flow
range. Estimation of such an elutiation limit and the amount of solid fuel fines burnt in the freeboard before they escape from the combustor is therefore of considerable value. The analysis is described in Section 8.2.

8.1.2 Flame Stability and Fuel Gas Bypass

These aspects were investigated in an attempt to explain difficulties, encountered during the initial combustion work, related to bed pre-heating.

The start-up technique adopted for the initial combustion experiments in this project was based upon that developed by Metcalfe (41). Here the bed is fluidised vigorously to distribute the sand evenly; the airflow is then reduced to about 1/4 of that required to just fluidise the bed. Fuel-gas to give a roughly stoichiometric mixture is then supplied to the plenum chamber and, after passing through the bed, is ignited in the freeboard space by a spark or pilot burner. Air and fuel-gas flow rates are then adjusted until the flame is stabilised on the bed surface. This last stage requires that the bed surface be clearly visible or remotely monitored in some precise manner. It was found, in this project, that attempting to stabilise the flame in the desired position by observing the temperatures recorded by thermocouples in the freeboard and the bed (thus if the flame was positioned between the two thermocouples the upstream thermocouple would indicate a significantly lower temperature than the downstream one) was insufficiently sensitive to flame position. As this method of flame front detection was inadequate it was impossible to avoid bed overheating on the one hand or movement of the flame front out of the
combustor on the other.

The work performed on fluidising velocity/flame speed matching was an attempt to overcome this problem. The aim of the analysis was to calculate the range of gas velocities at which the bed would be fluidised and, at the same time, the flame front would be stabilised on the bed surface. This analysis is given in Section 8.3.

Fuel-gas bypass, namely the transport of fuel gas mixture through the fluidised bed in bubbles which do not burn, has been investigated by Broughton (108). Not all the bubbles of gas/air mixture fail to burn. However, Broughton showed that a simple model based on the two phase theory of fluidisation (80) resulted in a means of estimating the expected bypass fraction for a given bed material particle size. The maximum allowable bypass fraction is the difference between the injected fuel gas flow rate and the minimum required to maintain a given bed temperature. From this analysis it is possible to estimate the minimum mean particle size that will allow self sustaining combustion in a gas fired fluidised bed. This method is shown in Section 8.4.

8.2 Particle Dynamics in the Combustor Freeboard

The subject of particle dynamics has been studied by Chevray et al (39) and Subzware (42). The analysis shown here draws heavily on the work of the latter.

The gas flow in the freeboard is complex. Chevray and his co workers and Subzware have assumed that, close to the nominal surface of the bed, the magnitude and direction of the flow is predominantly
determined by the motion of the bed and is, essentially, in solid body rotation ie a forced vortex. Here the angular velocity is constant and the tangential velocity inversely proportional to the radius, ie

\[ U_0(r) = w \cdot r \]  

8.2.1

Close to the exhaust port and at smaller radial positions the gas flow is assumed to be in free vortex motion so that

\[ U_0(r) = w \cdot r_1^2 / r \]  

8.2.2

between the two is a transition zone of indeterminate size.

For the purposes of the analysis here it is assumed that:

a) The transition occurs instantaneously.

b) The transition boundary is co-axial with the exhaust port and of the same radius.

Chevray et al (39) suggest that the motion of particles projected into the freeboard is described by the following equations.
Acceleration

\[ \ddot{r} = (\dddot{r} - \dot{\theta}^2) \hat{i} + (\ddot{r} + 2\dot{\theta}) \hat{j} + \ddot{z} \hat{k} \] 8.2.3

where \( \hat{i}, \hat{j} \) and \( \hat{k} \) are orthogonal unit vectors.

Forces

Centripetal

\[ F_{ce} = -\omega^2 r \hat{i} \] 8.2.4

Coriolis

\[ F_{co} = -2mr\dot{\theta} \hat{i} + 2mr \dot{\theta} \hat{j} \] 8.2.5

Gravity

\[ F_g = -mg \hat{k} \] 8.2.6

Aerodynamic drag

\[ F_d = \frac{1}{2} \rho A U^2 C_d \left[ U_x \hat{i} + U_y \hat{j} + U_z \hat{k} \right] \]
\[ \quad \left( \left| U_x \right| \quad \left| U_y \right| \quad \left| U_z \right| \right) \] 8.2.7

In order to determine the trajectory of a particle projected into the freeboard it is necessary to solve the acceleration equation imposing the force equations as descriptions of the component accelerations.

Thus, combining these equations and substituting, \( B \), where

\[ B = \frac{3}{16} \int_0^\infty \rho \, dp \] 8.2.8

286.
it is found that:

$$\ddot{r} - r\dot{\theta}^2 = w^2 r + 2 w r \dot{\theta} - BC_d |U| U \dot{U} \quad 8.2.9$$

$$r\ddot{\theta} + 2 r \dot{\theta} = -2 w r - BC_d |U| U \dot{U} \quad 8.2.10$$

$$z = -g - BC_d |U| U \dot{U} \quad 8.2.11$$

If the gas flow up to the exhaust port radius follows assumptions (a) and (b) then for $r_s < r < r_e$

$$\dot{r} = U_\hat{r} - \frac{Q}{2\pi r h} \quad 8.2.12$$

$$\dot{\theta} = \frac{U_\theta}{r} + \frac{w r^2 - w}{r^2} \quad 8.2.13$$

and

$$\dot{z} = U_z \quad 8.2.14$$

differentiating with respect to time

$$\ddot{r} = \frac{dU_\hat{r}}{dt} + \frac{Q}{2\pi r^2 h} \dot{r} \quad 8.2.15$$

$$\ddot{\theta} = \frac{1}{r} \frac{dU_\theta}{dt} - \frac{U_\theta \dot{r} - 2 w r^2 \dot{\theta}}{r^2} \quad 8.2.16$$

and

$$\ddot{z} = \frac{dU_z}{dt} \quad 8.2.17$$

upon substitution we find that

$$\frac{dU_\hat{r}}{dt} = w^2 r + (2 w r + r \dot{\theta}) \ddot{\theta} - \frac{Q}{2\pi r^2 h} (U_\hat{r} - \frac{Q}{2\pi r h}) - BC_d |U| U \dot{U} \quad 8.2.18$$

297.
\[
\frac{dU_t}{dt} = \left( U_t - \frac{Q}{r} \right) \left( -\frac{U_r}{r} - 4\pi r^2 \right) - \frac{\rho C_d L U_t U_r}{r^2}
\]

8.2.19

\[
\frac{dU_r}{dt} = -\frac{g}{2\rho r} - \frac{\rho C_d L U_t U_r}{r^2}
\]

8.2.20

Stepwise integration of these equations gives instantaneous values of \( U_t, U_r \) and \( U_z \) and hence corresponding values of \( r, \theta \) and \( z \) may be determined.

The principal drawback in this technique is that the equations are "stiff" and thus very small time increments \( O(10^{-5}) \) sec must be employed in the interactive procedure if unstable solutions are to be avoided. The computer programme listing can be found in Appendix C. The computer used was an HP 9845.

Incorporation in the model of the transition from forced to free vortex gas flow at the exhaust port boundary was found to be impossible due to the inherent instability of the solution method. It may be necessary to model the transition as a gradual change, which is closer to reality, or to use a more specific solution technique for stiff differential equations such as that used by Chevray.

For the purposes of the current work, it is sufficient to assume that particles whose trajectories intersect with the exhaust port boundary will be lost from the combustor in the exhaust stream since the average gas velocities in this region are very large in comparison with those encountered closer to the bed surface and also have a strong axial component.
The initial values of radial, tangential and axial velocity from which the solution technique proceeds are important in determining the fate of a given particle. Particles are projected into the freeboard from the cap of a bursting bubble. Chevray et al, in their analysis, assumed that the particle is projected with velocity components equal to the bubble rise velocity

\[ U_b = 0.711(n g D_b)^{0.5} \]  

8.2.21

from a bed operating at a fluidisation index of 2.

As a result, their analysis predicts that only small particles, 
\( d < 20 \) microns would be lost from the combustor. Larger particles, 
\( d_p > 20 \) microns, either enter stable orbits within the freeboard or return to the bed.

Subzwari believed that the initial conditions used by Chevray et al to be unrepresentative because measurements made of elutriated particle sizes from an atmospheric temperature rotating fluidised bed (42) showed that particles are large as 1.3 mm could be elutriated at a fluidisation index of 2.

As an alternative, Subzwari suggests that the initial radial and tangential velocity component should be made equal to

\[ U_r = U_b + 3U_{mf} \]  

8.2.22

and \[ U_t = w r_s \]  

8.2.23
The equivalent of this radial velocity has been used as an explanation for elutiation from a conventional fluidised bed (150).

Subzwari reports that the effect of these initial conditions is to raise the estimation of the diameter of the largest elutriated particle to 150 microns but this, still, is too low.

Using Subzwari's initial conditions applied to the geometry of the combustor used in this project particle trajectories for a number of conditions of gas flow, rotational speed, particle density, particle diameter and initial velocity were determined. These predicted trajectories can be found in Figures 8.1 to 8.15 and a summary of the basic data can be found in Table 8.1.

Figures 8.1 to 8.11 (initial conditions for which are given in Table 8.1) show trajectories of particles at atmospheric temperature and pressure in initial gas velocities of 1.25 and 2.5 times \( U_{mf} \) (as estimated from Richardson (in 80) equation 3.17).

It is evident that when the initial radial velocity of the particle is equal to the bubble rise velocity or to the minimum fluidisation velocity, only small particles \( d_p < 25 \) microns are lost from the combustor. However, when the initial radial velocity is increased to \( (U_{df} + 3U_{mf}) \) particles as large as and larger than 500 microns can be lost the combustor when the exit port radius is 0.06 m \( (r_e/r_s = 0.67) \). However, when the exit port radius is reduced to 0.045 m \( (r_e/r_s = 0.5) \), see Figures 8.7 to 8.11, the largest particles lost, even at \( U/U_{mf} = 5 \), are only about 50 microns, whereas particles of 500 microns were elutriated from the cold flow rig at \( U/U_{mf} = 2.7 \). The
predictions from this model thus seem increasingly unreliable as the exit port diameter is reduced and it is clear that further examination of the model and experimental observations are needed. One reason for the apparent inconsistency may be that, in reality, if the diameter of the exit port is altered the actual gas flow patterns within the combustor may be modified, whilst the model assumes that no modification occurs. The model thus predicts trajectories that are independent of combustor geometry. This part of the investigation had to be terminated here.

Further investigations are warranted because the analysis of particle trajectories in swirling flow is a matter of wider interest than in the RFB alone; eg cyclone separators and cyclone and swirl burners.

The trajectories of particles of coal ejected from bed surface are shown in Figures 8.12 to 8.15. Whilst these predictions may not be particularly accurate a preliminary estimate of the carbon carryover at various operating conditions can be made.

At atmospheric pressure and a gas temperature of 850°C when the applied radial acceleration is 10 times gravity and the surface gas velocity equal 5 $U_m^p$ it is predicted that particles smaller than 100 microns will be elutriated, Figure 8.13. If the coal feed mean size is 1.732 mm (1-3 mm range) then the predicted carbon loss is $< 0.02\%$.

At a gas pressure of 5 atm. and gas temperature of 850°C when the applied radial acceleration is 30 times gravity and the surface gas velocity equal to 10 m/s (4.45 $U_m^p$), again it is predicted that particles below 100 microns will be elutriated, Figure 8.15. Thus the
expected carbon loss is again $< 0.02\%$.

Even if these predictions are gross underestimates and coal particles as large as 500 microns are lost from the combustor the carryover loss is still only 2.4% and, whilst not desirable, may be acceptable.

8.2.1 Summary of section 8.2

Whilst the prediction of particle loss using the simple model described here is not in good agreement with measured particle loss, more representative techniques for describing and analysing the gas flow in the freeboard are required for reliable predictions. The value of predicting particle trajectories in swirling flows cannot be denied and warrants further investigation. The simple model shown here provides insight into the complex interactions possible within the RFB and describes, qualitatively, the observed effects of geometrical changes.

8.3 Fluidising Velocity/Flame Speed Matching

It has already been stated that the aim of this analysis was to determine the range of gas velocities at which the bed would be fluidised and, at the same time the flame of a stoichiometric mixture of propane and air would be stable on the bed surface during the start-up sequence. In these circumstances, the air and gas flow through the bed will be at a temperature that is set by the air feed temperature up until the time when the surface layers of the bed become heated by the flame. Thus we can assume that the initial velocity balance must be achieved at near-ambient temperatures.
Minimum and fully supported fluidising velocities, predicted from the correlations of Wen and Yu (83) and Richardson (80) and from the correlations proposed in Chapter 6, were compared with the natural flame velocities for a stoichiometric mixture of propane and air at various temperatures given by Rose and Cooper (145). The latter velocity was found to be satisfactorily described by the equation

\[ U_s = 3.39 \times 10^{-5} T_g^{1.65} \]  

where \( T_g \) = fuel/air mixture temperature, °K.

This has a correlation coefficient of 0.99 for the range \( T_g = 0°C \) to \( T_g = 250°C \).

The flame speed and the minimum or fully supported fluidising velocities for particles of mean size 100, 200 and 300 microns are shown in Figures 8.16, 8.17, 8.18 and 8.19. The range of applied acceleration to the particles is from 5x gravity to 15x gravity.

It is evident that initial flame stabilisation is only possible, at 10g, for particles whose diameter is less than 200 microns. Metcalfe found it possible to stabilise flames on a bed of particles 180-250 microns, \( d_p = 212 \) microns. This is mainly due to the fact that Metcalfe did not try to stabilise a flame on a cold fluidised bed but instead on a cold packed bed - a technique found to be unsatisfactory where the combustor was fully enclosed.

At an applied acceleration of 5g it should be possible to just fluidise a bed of 300 micron particles and also stabilise a flame on the...
surface. Metcalfe, however, found that unstable combustion resulted when a bed of particles 250-355 microns, $d_p = 298$ microns, was used and, assuming that stability of the initial flame was not a problem (Metcalfe does not comment on this) then some other explanation must be sought.

In summary, then, we can see that it should be possible to stabilise a propane flame on a cold fluidised bed provided that the applied acceleration is not greater than $5 \times g$ for particles up to 300 microns and not greater than $10 \times g$ for particles up to 200 microns. If larger particles are to be used, the flame can only be stabilised on a packed bed with the attendant risk of excessive bed temperatures.

8.4 Fuel-Gas Bypass in Bubbles

Unburnt fuel-gas passing through the bed as bubbles is an important phenomenon because of the operating limits that it might impose. Gas which bypasses the bed to burn in the freeboard imparts only a fraction of its heat to the body of the bed and thus contributes little to the maintenance of bed temperature. The amount of gas which actually bypasses the bed in mainly dependent upon particle size (as $U_{mf}$), bubble diameter, bed depth and superficial gas velocity. To a lesser extent the bypass is dependent upon average bed temperature in terms of gas density and viscosity. It should be possible to predict, for a given bed temperature, depth and superficial gas velocity the mean particle size at which no bypass occurs. Further, from a knowledge of the fuel properties it should be possible to predict the air:fuel ratios that a given bed will be able to burn stably at different superficial velocities.
Broughton (108) has suggested, as a first approximation, that the fraction of gas to bypass a bed in the bubbles may be predicted using the following equation:

\[ f = \frac{(U - U_{mf})e^{-X}}{U} \]  

\[ X = \int_{0}^{L} \frac{QdL}{U_bV_b} \]  

\[ Q = \text{volumetric exchange rate and } U_b, V_b \text{ are bubble velocity and volume respectively.} \]

The principal assumption used in the derivation of this equation was that the Two Phase Theory of Fluidisation (Davidson and Harrison (80)) described the fluid mechanics of the fluidised bed and that the fraction of bypass was determined by the gas phase exchange rate between the bubble phase and emulsion phase.

Broughton assumed a fixed bubble velocity of 0.35 m/s and obtained the curves shown in Figure 8.20. When compared with experimental results presented in the same paper, it is evident that the simple equation 8.4.1 can be used to approximate the particle size at which combustion becomes unstable, in a conventional fluidised bed, due to only just enough gas burning to maintain a given bed temperature.

In order to extend this model to the rotating fluidised bed some minor modifications are made to Broughton's model. Rather than assume a fixed bubble velocity, the following relationship used by
Chevray et al (39), in their analysis of bubble motion in the RFB is used.

\[ U_b = 0.711(n_g D_b)^{0.5} \text{ m/s} \quad \text{(8.4.3)} \]

Using the same relationship for \( Q \) as used by Broughton ie

\[ Q = 0.75\pi U_m L_{mf} D_b^2 \text{ m}^3/\text{s} \quad \text{(8.4.4)} \]

we find

\[ X = \frac{9U_{mf}L_{mf}}{1.422(n_g D_b)^{0.5}D_b(1-E_b)} \quad \text{(8.4.5)} \]

Using Broughton's assumption that \( D_b = 0(L_{mf}) \), and bubble hold-up, \( E_b \), reaches 0.4 when \( U - U_{mf} = 0.2 \)

\[ X = \frac{3.368U_{mf}}{(n L_{mf})^{0.5}} \quad \text{(8.4.6)} \]

This may be substituted in equation 8.4.2 to give an approximate model for fuel gas bypass in a rotating fluidised bed.

Figures 8.21 to 8.43 show the results of evaluating this model with the following parameters:-

- \( n \): 5, 10, 15, 20, 25, 30 times gravity
- \( U \): 0.5, 0.6, 1.5 m/s
- \( L_{mf} \): 10, 20 mm
- \( T_B \): 15°C and 850°C
- \( P \): 1 atm
- \( Re_{mf} = (33.7^2 + 0.0408 \text{ Ga}^2)^{1/2} - 33.7 \)
Several interesting features of the RFB combustor, and also some potential problems can be examined in light of the analysis. Metcalfe (41) found that the RFB gas combustor will operate satisfactorily with smaller particles than a conventional fluidised bed gas combustor. Comparing Broughton’s analysis with the modified version we can see that it is the effect of applied radial acceleration on \((U - U_{mf})\) that produces this effect. The current analysis shows that, for a fixed applied acceleration, superficial velocity and bed temperature the effect of bed depth on gas bypass is not strong. However, if bed thickness is fixed and the applied acceleration varied from 5 to 30 times gravity a measurable reduction in the particle size for a given bypass fraction is predicted. Thus in moving from a conventional to a rotating fluidised bed, smaller particles are applicable.

Furthermore, fuel bypass becomes small when particle sizes greater than 250 microns are used.

The consequence of this is that very fine adjustment of the fuel:air ratio is required in order to balance the heat release of the fuel by the heat absorbed by the cold air. This may go some way towards explaining why Metcalfe was unable to attain stable combustion temperatures with 250-355 microns sand and why unstable combustion and bed overheating was encountered with a bed of 355-500 microns sand in the work reported here.

During start-up conditions the sets of curves Figures 8.21 to 8.38 apply. Here the initial bulk temperature of the bed and fluidising gas is between 15°C and 850°C and gas velocities are between 0.5-1.5 m/s. The model predicts that little or no fuel gas will bypass the bed.
if average particle sizes greater than about 250 microns are employed. Now, using Metcalfe's method of bed heating it is necessary to pass a stoichiometric mixture through the bed in order to achieve fuel gas ignition in the freeboard. If the bed particles are larger than 250 microns, the flame will move quickly into the bed and, because the heat release rate is considerably greater than the heat removal rate, bed overheating and distributor damage can occur very quickly if rapid adjustments are not made to the fuel gas flow rate. This sequence of events closely resembles the events that occurred when combustion in a bed of sand 355-500 microns was attempted in the current work.

Metcalfe does not report difficulties in start-up with a bed of 250-355 microns sand but does mention that stable operating temperatures were impossible to achieve. Examining the sets of curves in Figs. 8.2l to 8.32, we can see that at bed temperatures of ca. 850°C the predicted fuel gas bypass in beds of particles of 300 micron diameter is very small ca. 15% at 5x g and 0% at 30x g. Thus, even at low rotational speeds there is very little adjustment tolerance available on air to fuel ratio rendering the attainment of stable temperatures very difficult.

It would appear, then, that to achieve stable combustion within an adiabatic rotating fluidised bed the bed particle size should be less than about 250 micron, depending upon particle properties.

The problem resulting from this, admittedly tentative, conclusion is that a distributor is required that will fluidise sand particles of less than 250 micron diameter and prevent seepage of the small fractions into the plenum space.
The obvious solution is a sintered metal distributor but, as stated in Chapter 4 (which discussed some features of distributor design), the strength of sintered materials is lower than that of solid metal and sintered materials are also prone to blocking. Nevertheless, when tried in the small scale RFB, a porous distributor was quite successful. A different solution may need to be sought for a full scale RFB.

8.4.1 Prediction of Air:Fuel Ratio

The prediction of operating air to fuel ratio, (AFR), in an adiabatic bed should be possible by equating heat input rate to heat removal rate if radiant and conductive losses are small enough to be ignored. The heat input rate is given by

\[ H_I = (\text{Vol gas})_{\text{STP}} \times \text{calorific value} \quad 8.4.6 \]

whilst the heat removal rate is given by

\[ H_R = (\text{Vol air})_{\text{STP}} \times \rho_a \times C_p \times (T_b - T_i) \quad 8.4.7 \]

If all of the supplied fuel gas burns ie \( f = 0 \) then

\[ H_I = H_R \]

and

\[ (\text{AFR})_{\text{STP}} = \frac{\text{calorific value}}{\int a \cdot C_p \cdot \Delta T} \quad 8.4.8 \]
If fuel gas bypass occurs such that \( f = x \) then

\[
X \cdot H_I = H_R
\]

and

\[
(AFRI)_{STP} = \frac{X \cdot \text{calorific value}}{p_a \cdot c_p \cdot \Delta T}
\]

Thus it should be possible to predict the AFR that will just support a specified bed temperature (when \( f = 0 \)) and the AFR necessary when a certain level of bypass occurs (when \( f = X \)).

Figures 8.39 to 8.41 show the effects on gas bypass at a bed temperature of 1150°C and a bed depth of 0.01m of changing rotational speed and particle size for a range of superficial velocity of 1 m/s - 5 m/s. Metcalfe's experimental work indicated the likely range of AFR possible with 180-250 micron sand under the above conditions and it is instructive to examine measured and predicted AFR. Table 8.2 shows this comparison.

Clearly the model does not describe the process at all well. Not only is there considerable disparity between measured and predicted values of AFR especially at low velocities but also the trends of measured and predicted values are different. In the case of the measured AFR, there is a definite increase with increasing fluidising velocity with a tendency to asymptote at the higher applied accelerations. In the case of the predicted AFR, the values are essentially constant with respect to velocity for a given applied acceleration and increase slightly at fixed velocity and increased acceleration.
There are two distinct weaknesses in the model. First, the value for \( U_{mf} \) is estimated by using a modified correlation which is derived from packed bed information in order to be consistent with Broughton's analysis. Secondly, the bubble rise velocity is also estimated from a modified correlation and there is no experimental justification for its applicability. Inaccuracies resulting from either will have a significant effect on the predicted fuel bypass fraction because both \( U_{mf} \) and the bubble rise velocity occur in the exponential.

The effect of changing the correlation for \( U_{mf} \) in the calculation procedure from Wen and Yu's (83) to that determined in Chapter 6 is illustrated by Figures 8.42 and 8.43. Figure 8.42 shows the predicted fuel bypass at atmospheric temperatures at an applied acceleration of 5 gravities. Comparing this with Figure 8.21 we can see that, because of exponentiation, the slightly higher values of \( U_{mf} \) result in a much steeper predicted fall-off of bypass fraction with increasing particle size. This would make the constraints on the upper limit of particle size even more severe than predicted by the original analysis.

Figure 8.43 shows the predicted fuel bypass at a bed temperature of 1150°C and an applied acceleration of 30 gravities. Fuel bypass is predicted to be absent, even for gas velocities of 5 m/s, if the bed particle size is larger than 100 micron. Thus if the mean particle size were 212 micron, from a bed of 180-250 microns, the predicted operating AFR would be 67.6:1 irrespective of gas velocity. Comparing this with the results presented in Table 8.2 we see that the prediction is further from the measured values than the original analysis.
8.4.2 Summary of section 8.4

Whilst the model for fuel gas bypass can to some extent help to explain the problems of start-up and temperature control in gas fired fluidised beds there is a need for investigation and quantisation of $U_{mf}$ and $U_b$ at elevated temperatures in a rotating fluidised bed before the model of fuel bypass can be utilised with confidence throughout the operating regime of the RFB. Furthermore, it is probable that conductive and radiant losses must be accounted for.
### TABLE 8.1 Initial Conditions for Particle Trajectory Predictions

<table>
<thead>
<tr>
<th>Fig. No.</th>
<th>Rotational Speed Rad/sec</th>
<th>Equivalent Applied Acc $^n$ n*g</th>
<th>Initial Radial Velocity</th>
<th>Initial Tangential Velocity</th>
<th>Initial Axial Velocity</th>
<th>(Surface) gas $(n \cdot U_{mf})$</th>
<th>Exit Port Radius m</th>
</tr>
</thead>
<tbody>
<tr>
<td>8.1</td>
<td>31.32</td>
<td>10</td>
<td>$-U_b$</td>
<td>$U_b$</td>
<td>$U_{br}$</td>
<td>1.25</td>
<td>0.06</td>
</tr>
<tr>
<td>8.2</td>
<td>31.32</td>
<td>10</td>
<td>$-U_{mf}$</td>
<td>$U_{mf}$</td>
<td>$U_{mf}$</td>
<td>1.25</td>
<td>0.06</td>
</tr>
<tr>
<td>8.3</td>
<td>31.32</td>
<td>10</td>
<td>$U^*$</td>
<td>$W_{rS}$</td>
<td>0</td>
<td>1.25</td>
<td>0.06</td>
</tr>
<tr>
<td>8.4</td>
<td>31.32</td>
<td>10</td>
<td>$-U_b$</td>
<td>$W_{rS}$</td>
<td>0</td>
<td>2.5</td>
<td>0.06</td>
</tr>
<tr>
<td>8.5</td>
<td>31.32</td>
<td>10</td>
<td>$-U_{mf}$</td>
<td>$W_{rS}$</td>
<td>0</td>
<td>2.5</td>
<td>0.06</td>
</tr>
<tr>
<td>8.6</td>
<td>31.32</td>
<td>10</td>
<td>$U^*$</td>
<td>$W_{rS}$</td>
<td>0</td>
<td>2.5</td>
<td>0.06</td>
</tr>
<tr>
<td>8.7</td>
<td>31.32</td>
<td>10</td>
<td>$-U_b$</td>
<td>$U_b$</td>
<td>$U_{br}$</td>
<td>1.25</td>
<td>0.045</td>
</tr>
<tr>
<td>8.8</td>
<td>31.32</td>
<td>10</td>
<td>$U^*$</td>
<td>$W_{rS}$</td>
<td>0</td>
<td>1.25</td>
<td>0.045</td>
</tr>
<tr>
<td>8.9</td>
<td>31.32</td>
<td>10</td>
<td>$-U_b$</td>
<td>$W_{rS}$</td>
<td>0</td>
<td>2.5</td>
<td>0.045</td>
</tr>
<tr>
<td>8.10</td>
<td>31.32</td>
<td>10</td>
<td>$-U_{mf}$</td>
<td>$W_{rS}$</td>
<td>0</td>
<td>2.5</td>
<td>0.045</td>
</tr>
<tr>
<td>8.11</td>
<td>31.32</td>
<td>10</td>
<td>$U^*$</td>
<td>$W_{rS}$</td>
<td>0</td>
<td>2.5</td>
<td>0.045</td>
</tr>
<tr>
<td>8.12</td>
<td>31.32</td>
<td>10</td>
<td>$U^*$</td>
<td>$W_{rS}$</td>
<td>0</td>
<td>2.5</td>
<td>0.06</td>
</tr>
<tr>
<td>8.13</td>
<td>31.32</td>
<td>10</td>
<td>$U^*$</td>
<td>$W_{rS}$</td>
<td>0</td>
<td>5</td>
<td>0.045</td>
</tr>
<tr>
<td>8.14</td>
<td>31.32</td>
<td>10</td>
<td>$U^*$</td>
<td>$W_{rS}$</td>
<td>0</td>
<td>5</td>
<td>0.045</td>
</tr>
<tr>
<td>8.15</td>
<td>57.18</td>
<td>30</td>
<td>$U^*$</td>
<td>$W_{rS}$</td>
<td>0</td>
<td>4.45</td>
<td>0.045</td>
</tr>
</tbody>
</table>

$U^* = (U_b + 3U_{mf})$
TABLE 8.2 Comparison of AFR Measured by Metcalfe (41) and Predicted
Operating AFR for 180-250 Micron Sand at 1150° C

<table>
<thead>
<tr>
<th>No of gravities</th>
<th>U_s m/s</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>meas. predict</td>
<td>meas. predict</td>
<td>meas. predict</td>
<td>meas. predict</td>
<td>meas. predict</td>
</tr>
<tr>
<td>10</td>
<td>43:1 55.3:1</td>
<td>45:1 54.4:1</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>20</td>
<td>34:1* 62.4:1</td>
<td>47:1 60:1</td>
<td>54:1 60:1</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>30</td>
<td>28:1* 65:1</td>
<td>43:1 63.9:1</td>
<td>50:1 63.7:1</td>
<td>54:1 63.3:1</td>
<td></td>
</tr>
</tbody>
</table>

* Extrapolated from measured data(41)
APPLIED G = 10 *Gravity
BED PARTICLE DP = 600 MICRONS
GAS DENSITY = 1.177 KG/M^3
GAS VISCOSITY = .0000189 NS/MT
SURFACE GAS VEL. = 2.0076 M/S (1.2500622665 *Um/s)
GAS MASS FLOW = .1069 KG/S
PARTICLE DENSITY = 2630 KG/M^3
BED SURF. RADIUS = .09 M
EXHAUST RADIUS = .06 M
PART. RELEASE HT = .04 M
ROTATIONAL SPEED = 31.32 RAD/S
BUBBLE VELOCITY = .704 M/S
MIN. FLUID/N VEL = 1.606 M/S
INIT. RADIAL VEL = -.704 M/S
INIT. ANG. VEL = .704 M/S
INIT. AXIAL VEL = .704 M/S
STEP SIZE = .0001 sec
TIME OF FLIGHT 25 MICRON = .0199 sec
TIME OF FLIGHT 50 MICRON = .1 sec
TIME OF FLIGHT 75 MICRON = .0249 sec
TIME OF FLIGHT 100 MICRON = .019 sec
TIME OF FLIGHT 250 MICRON = .0169 sec
TIME OF FLIGHT 500 MICRON = .0161 sec

FIG 8.1 Particle trajectories at atmospheric conditions
APPLIED G = 10 *Gravity
BED PARTICLE DP = 600 MICRONS
GAS DENSITY = 1.177 KG/M^3
GAS VISCOSITY = 0.0000189 NS/⁴
SURFACE GAS VEL. = 2.0076 M/S < 1.2500622665 *Umf)
GAS MASS FLOW = 0.1069 KG/S
PARTICLE DENSITY = 2630 KG/M^3
BED SURF. RADIUS = 0.09 M
EXHAUST RADIUS = 0.06 M
PART. RELEASE HT = 0.04 M
ROTATIONAL SPEED = 31.32 RAD/S
BUQUE VELOCITY = 0.704 M/S
MIN. FLUID N VEL = 1.606 M/S
INIT. RADIAL VEL = -1.606 M/S
INIT. ANG. VEL = 1.606 M/S
INIT. AXIAL VEL = 1.606 M/S
STEP SIZE = 0.0001 sec
TIME OF FLIGHT 25 MICRON = 0.0194 sec
TIME OF FLIGHT 50 MICRON = 0.1 sec
TIME OF FLIGHT 75 MICRON = 0.0184 sec
TIME OF FLIGHT 100 MICRON = 0.0169 sec
TIME OF FLIGHT 250 MICRON = 0.0141 sec
TIME OF FLIGHT 500 MICRON = 0.0136 sec

FIG 8.2 Particle trajectories at atmospheric conditions
APPLIED G = 10 *Gravity
BED PARTICLE DP = 600 MICRONS
GAS DENSITY = 1.177 KG/M^3
GAS VISCOSITY = .0000189 NS/FT
SURFACE GAS VEL. = 2.0076 M/S (1.2500622665 * Umf)
GAS MASS FLOW = .1069 KG/S
PARTICLE DENSITY = 2630 KG/M^3
BED SURF. RADIUS = .09 M
EXHAUST RADIUS = .06 M
PART. RELEASE HT = .04 M
ROTATIONAL SPEED = 31.32 RAD/S
BUBBLE VELOCITY = .704 M/S
MIN. FLUID’N VEL = 1.606 M/S
INIT. RADIAL VEL = -5.522 M/S
INIT. AXIAL VEL = 2.919 M/S
STEP SIZE = .0001 sec
TIME OF FLIGHT 25 MICRON = .0129 sec
TIME OF FLIGHT 50 MICRON = .1 sec
TIME OF FLIGHT 75 MICRON = .0136 sec
TIME OF FLIGHT 100 MICRON = .0121 sec
TIME OF FLIGHT 250 MICRON = .0105 sec
TIME OF FLIGHT 500 MICRON = .0104 sec

FIG 8.3 Particle trajectories at atmospheric conditions
APPLIED G = 10 *Gravity
BED PARTICLE DP = 600 MICRONS
GAS DENSITY = 1.177 KG/M^3
GAS VISCOSITY = .0000189 NS/M
SURFACE GAS VEL. = 4.0152 M/S (~2.500124533 *Umf)
GAS MASS FLOW = .2138 KG/S
PARTICLE DENSITY= 2630 KG/M^3
BED SURF.RADIUS = .09 M
EXHAUST RADIUS = .06 M
PART. RELEASE HT= .04 M
ROTATIONAL SPEED= 31.32 RAD/S
BUBBLE VELOCITY = .704 M/S
MIN. FLUID`N VEL= 1.606 M/S
INIT.RADIAL VEL =-.704 M/S
INIT.ANG. VEL = 2.819 M/S
INIT.AXIAL VEL = 0 M/S
STEP SIZE = .0001 sec
TIME OF FLIGHT 25 MICRON = .0085007 sec
TIME OF FLIGHT 50 MICRON = .028 sec
TIME OF FLIGHT 75 MICRON = .0198 sec
TIME OF FLIGHT 100 MICRON = .0143 sec
TIME OF FLIGHT 250 MICRON = .0117 sec
TIME OF FLIGHT 500 MICRON = .0114 sec

FIG 8.4 Particle trajectories at atmospheric conditions
APPLIED G  = 10 *Gravity
BED PARTICLE DP = 600 MICRONS
GAS DENSITY  = 1.177 KG/M^3
GAS VISCOSITY = 0.000189 HS/M
SURFACE GAS VEL. = 4.0152 M/S < 2.500124533 *Umf
GAS MASS FLOW = 0.2138 KG/S
PARTICLE DENSITY = 2630 KG/M^3
BED SURF. RADIUS = 0.09 M
EXHAUST RADIUS = 0.06 M
PART. RELEASE HT = 0.04 M
ROTATIONAL SPEED = 31.32 RAD/S
BUBBLE VELOCITY = 0.704 M/S
MIN. FLUID'N VEL = 1.606 M/S
INIT. RADIAL VEL = -1.606 M/S
INIT. ANG. VEL = 2.819 M/S
INIT. AXIAL VEL = 0 M/S
STEP SIZE = 0.0001 sec
TIME OF FLIGHT 25 MICRON = 0.0076007 sec
TIME OF FLIGHT 50 MICRON = 0.026 sec
TIME OF FLIGHT 75 MICRON = 0.0196 sec
TIME OF FLIGHT 100 MICRON = 0.0144 sec
TIME OF FLIGHT 250 MICRON = 0.0114 sec
TIME OF FLIGHT 500 MICRON = 0.0112 sec

FIG 8.5 Particle trajectories at atmospheric conditions
APPLIED G = 10 *Gravity
BED PARTICLE DP = 600 MICRONS
GAS DENSITY = 1.177 KG/M3
GAS VISCOSITY = .0000189 Ns/m
SURFACE GAS VEL = 4.0152 M/S (2.500124533 *UmF)
GAS MASS FLOW = .2138 KG/S
PARTICLE DENSITY = 2630 KG/M3
BED SURF.RADIUS = .09 M
EXHAUST RADIUS = .06 M
PART. RELEASE HT = .04 M
ROTATIONAL SPEED = 31.32 RAD/S
BUBBLE VELOCITY = .704 M/S
MIN. FLUID'N VEL = 1.606 M/S
INIT.RADIAL VEL = -5.522 M/S
INIT.ANG. VEL = 2.819 M/S
INIT.AXIAL VEL = 0 M/S
STEP SIZE = .0001 sec
TIME OF FLIGHT 25 MICRON = .0052006 sec
TIME OF FLIGHT 50 MICRON = .0047 sec
TIME OF FLIGHT 75 MICRON = .0046006 sec
TIME OF FLIGHT 100 MICRON = .0046006 sec
TIME OF FLIGHT 250 MICRON = .0046 sec
TIME OF FLIGHT 500 MICRON = .0045 sec

FIG 8.6 Particle trajectories at atmospheric conditions
APPLIED G = 10 *Gravity
BED PARTICLE DP = 600 MICRONS
GAS DENSITY = 1.177 KG/M^3
GAS VISCOSITY = .0000187 NS/M
SURFACE GAS VEL. = 2.0075 M/S < 1.25 *Umf>
GAS MASS FLOW = .1069 KG/S
PARTICLE DENSITY = 2630 KG/M^3
BED SURF. RADIUS = .09 M
EXHAUST RADIUS = .045 M
PART. RELEASE HT = .04 M
ROTATIONAL SPEED = 31.32 RAD/S
BUBBLE VELOCITY = .704 M/S
MIN. FLUID VEL = 1.606 M/S
INIT. RADIAL VEL = -.704 M/S
INIT. ANG. VEL = .704 M/S
INIT. AXIAL VEL = .704 M/S
STEP SIZE = .0001 sec
TIME OF FLIGHT 25 MICRON = .0352 sec
TIME OF FLIGHT 50 MICRON = .1 sec
TIME OF FLIGHT 75 MICRON = .0249 sec
TIME OF FLIGHT 100 MICRON = .019 sec
TIME OF FLIGHT 250 MICRON = .0169 sec
TIME OF FLIGHT 500 MICRON = .0161 sec

FIG 8.7 Particle trajectories at atmospheric conditions
APPLIED G = 10 * Gravity
BED PARTICLE DP = 600 MICRONS
GAS DENSITY = 1.177 KG/M^3
GAS VISCOSITY = .0000107 NS/M
SURFACE GAS VEL. = 2.0075 M/S < 1.25 * Umf
GAS MASS FLOW = 1.869 KG/S
PARTICLE DENSITY = 2630 KG/M^3
BED SURF. RADIUS = .09 M
EXHAUST RADIUS = .045 M
PART. RELEASE HT = .04 M
ROTATIONAL SPEED = 31.32 RAD/S
BUBBLE VELOCITY = .704 M/S
MIN. FLUID'N VEL = 1.606 M/S
INIT. RADIAL VEL = 5.522 M/S
INIT. ANG. VEL = 2.819 M/S
INIT. AXIAL VEL = 0 M/S
STEP SIZE = .0001 sec
TIME OF FLIGHT 25 MICRON = .0278 sec
TIME OF FLIGHT 50 MICRON = .1 sec
TIME OF FLIGHT 75 MICRON = .0136 sec
TIME OF FLIGHT 100 MICRON = .0121 sec
TIME OF FLIGHT 250 MICRON = .0105 sec
TIME OF FLIGHT 500 MICRON = .0104 sec

FIG 8.8 Particle trajectories at atmospheric conditions
APPLIED G = 10 * Gravity
BED PARTICLE DP = 600 MICRONs
GAS DENSITY = 1.177 KG/M^3
GAS VISCOSITY = .0000187 NS/M
SURFACE GAS VEL. = 4.015 M/S (2.5 * Umf)
GAS MASS FLOW = .2136 KG/S
PARTICLE DENSITY = 2630 KG/M^3
BED SURF. RADIUS = .09 M
EXHAUST RADIUS = .045 M
PART. RELEASE HT = .04 M
ROTATIONAL SPEED = 31.32 RAD/S
BUBBLE VELOCITY = .704 M/S
MIN. FLUID/N VEL = 1.686 M/S
INIT. RADIAL VEL = -.704 M/S
INIT. ANG. VEL = 2.819 M/S
INIT. AXIAL VEL = 0 M/S
STEP SIZE = .0001 sec
TIME OF FLIGHT 25 MICRON = .0135 sec
TIME OF FLIGHT 50 MICRON = .1 sec
TIME OF FLIGHT 75 MICRON = .0196 sec
TIME OF FLIGHT 100 MICRON = .0143 sec
TIME OF FLIGHT 250 MICRON = .0116 sec
TIME OF FLIGHT 500 MICRON = .0114 sec

FIG 8.9 Particle trajectories at atmospheric conditions
APPLIED G = 10 *Gravity
BED PARTICLE DP = 600 MICRONS
GAS DENSITY = 1.177 KG/M^3
GAS VISCOSITY = .0000187 NS/M
SURFACE GAS VEL. = 4.015 M/S (2.5 *Umf)
GAS MASS FLOW = .2136 KG/S
PARTICLE DENSITY = 2630 KG/M^3
BED SURF. RADIUS = .09 M
EXHAUST RADIUS = .045 M
PART. RELEASE HT = .04 M
ROTATIONAL SPEED = 31.32 RAD/S
BUBBLE VELOCITY = .794 M/S
MIN. FLUID/N VEL = 1.606 M/S
INIT. RADIAL VEL = -1.606 M/S
INIT. ANG. VEL = 2.019 M/S
INIT. AXIAL VEL = 0 M/S
STEP SIZE = .0001 sec
TIME OF FLIGHT 25 MICRON = .0126 sec
TIME OF FLIGHT 50 MICRON = .01 sec
TIME OF FLIGHT 75 MICRON = .0194 sec
TIME OF FLIGHT 100 MICRON = .0144 sec
TIME OF FLIGHT 250 MICRON = .0114 sec
TIME OF FLIGHT 500 MICRON = .0112 sec

FIG 8.10 Particle trajectories at atmospheric conditions
APPLIED G = 10 *Gravity
BED PARTICLE DP = 600 MICRONS
GAS DENSITY = 1.177 KG/M^3
GAS VISCOSITY = .0008187 NS/M
SURFACE GAS VEL. = 4.015 M/S (2.5 * Umf)
GAS MASS FLOW = .2138 KG/S
PARTICLE DENSITY = 2630 KG/M^3
BED SURF. RADIUS = .09 M
EXHAUST RADIUS = .045 M
PART. RELEASE HT = .04 M
ROTATIONAL SPEED = 31.32 RAD/S
BUBBLE VELOCITY = .704 M/S
MIN. FLUID/N VEL = 1.606 M/S
INIT. RADIAL VEL = -5.522 M/S
INIT. ANG. VEL = 2.019 M/S
INIT. AXIAL VEL = 0 M/S
STEP SIZE = .0001 sec
TIME OF FLIGHT 25 MICRON = .0099 sec
TIME OF FLIGHT 50 MICRON = .1 sec
TIME OF FLIGHT 75 MICRON = .0137 sec
TIME OF FLIGHT 100 MICRON = .0114 sec
TIME OF FLIGHT 250 MICRON = .0096 sec
TIME OF FLIGHT 500 MICRON = .0095 sec

FIG 8.11 Particle trajectories at atmospheric conditions
APPLIED G = 10 *Gravity
BED PARTICLE DP = 600 MICRONS
GAS DENSITY = .314 KG/M^3
GAS VISCOSITY = .0000467 NS/M
SURFACE GAS VEL. = 3.207 M/S (2.49961028839 *Um)
GAS MASS FLOW = .0456 KG/S
PARTICLE DENSITY = 1340 KG/M^3
BED SURF. RADIUS = .09 M
EXHAUST RADIUS = .06 M
PART. RELEASE HT = .04 M
ROTATIONAL SPEED = 31.32 RAD/S
BUBBLE VELOCITY = .704 M/S
MIN. FLUID/N VEL. = 1.283 M/S
INIT. RADIAL VEL. = -4.553 M/S
INIT. ANG. VEL. = 2.819 M/S
INIT. AXIAL VEL. = 0 M/S
STEP SIZE = .0001 sec
TIME OF FLIGHT 500 MICRON = .0104 sec
TIME OF FLIGHT 600 MICRON = .0104 sec
TIME OF FLIGHT 700 MICRON = .0103 sec
TIME OF FLIGHT 800 MICRON = .0103 sec
TIME OF FLIGHT 900 MICRON = .0102 sec
TIME OF FLIGHT 1000 MICRON = .0102 sec

FIG 8.12 Coal particle trajectories at atmospheric pressure and 850 C
FIG 8.13Coal particle trajectories at atmospheric pressure and 850°C.
APPLIED G = 10 *Gravity
BED PARTICLE DP = 600 MICRONS
GAS DENSITY = 1.572 KG/M^3
GAS VISCOSITY = .0000467 NS/M
SURFACE GAS VEL. = 5.055 M/S (5 *Um)
GAS MASS FLOW = .3594 KG/S
PARTICLE DENSITY = 1340 KG/M^3
BED SURF. RADIUS = .09 M
EXHAUST RADIUS = .045 M
PART. RELEASE HT = .04 M
ROTATIONAL SPEED = 31.32 RAD/S
BUBBLE VELOCITY = .704 M/S
MIN. FLUID'H VEL = 1.011 M/S
INIT. RADIAL VEL = -3.737 M/S
INIT. ANG. VEL = 2.819 M/S
INIT. AXIAL VEL = 0 M/S
STEP SIZE = .0001 sec
TIME OF FLIGHT 50 MICRON = .0079001 sec
TIME OF FLIGHT 75 MICRON = .011 sec
TIME OF FLIGHT 100 MICRON = .0244 sec
TIME OF FLIGHT 250 MICRON = .0109 sec
TIME OF FLIGHT 500 MICRON = .0181 sec
TIME OF FLIGHT 750 MICRON = .0099 sec

FIG 8.14 Coal particle trajectories at 5atm. pressure and 850 C
FIG 8.15  Coal particle trajectories at 5 atm, pressure and 850 C
$U_{mf}$ estimated from:

$$Re_{mf} = (33.7^2 + 0.0408Ga)^{0.5} - 33.7$$

FIG 8.16 Minimum fluidising velocity and flame propagation speed of propane
FIG 8.17 Minimum fluidising velocity and flame propagation speed of propane

\[ U_{mf} \, \text{estimated from:} \]

\[ \text{Re}_{mf} \cdot (25.7^2 + 0.365G_a)^{0.5} - 25.7 \]
FIG 8.18 Minimum fluidising velocity and flame propagation speed of propane
FIG 8.19 Fully supported fluidising velocity and flame propagation speed of propane
FIG 8.20 Fuel bypass in a shallow fluidised bed
(from ref. 108)
Particle density = 2630 kg/m$^3$
Air density = 1.225 kg/m$^3$
Air temperature = 15°C
Bed thickness = 0.01 m
Applied acc$^n$ = 5 g

FIG 8.21 Gas bypass at ignition conditions
Particle density = 2630 kg/m$^3$
Air density = 1.225 kg/m$^3$
Air temperature = 15°C
Bed thickness = 0.01 m
Applied acc$^n$ = 10 g

FIG 8.22 Gas bypass at ignition conditions
Particle density = 2630 kg/m³
Air density = 1.225 kg/m³
Air temperature = 15°C
Bed thickness = 0.01 m
Applied acc = 15 g

FIG 8.23 Gas bypass at ignition conditions
FIG 8.74 Gas bypass at ignition conditions
Particle density = 2630 kg/m³
Air density = 1.225 kg/m³
Air temperature = 15°C
Bed thickness = 0.02 m
Applied acc^n = 10 g

FIG. 8.25  Gas bypass at ignition conditions
FIG 8.26  Gas bypass at ignition conditions

- Particle density = 2630 kg/m³
- Air density = 1.225 kg/m³
- Air temperature = 15°C
- Bed thickness = 0.02 m
- Applied acc^n = 15 g
Particle density = 2630 kg/m$^3$
Air density = 0.318 kg/m$^3$
Air temperature = 850°C
Bed thickness = 0.01 m
Applied acc$^n$ = 5 g

FIG 8.27 Gas bypass at start-up conditions
Particle density = 2630 kg/m$^3$
Air density = .318 kg/m$^3$
Air temperature = 850$^\circ$C
Bed thickness = .01 m
Applied acc$^n$ = 10 g

FIG 8.26 Gas bypass at start-up conditions
Particle density = 2630 kg/m³
Air density = .318 kg/m³
Air temperature = 850°C
Bed thickness = .01 m
Applied acc°n = 15 g

FIG 8.29  Gas bypass at start-up conditions
Particle density = 2630 kg/m³
Air density = .318 kg/m³
Air temperature = 850°C
Bed thickness = .01 m
Applied acc^n = 20 g

FIG 8.30 Gas bypass at start-up conditions
Particle density = 2630 kg/m³
Air density = 0.318 kg/m³
Air temperature = 850°C
Bed thickness = 0.01 m
Applied acc^n = 25 g

FIG 8.31 Gas bypass at start-up conditions
Particle density = 2630 kg/m³
Air density = .318 kg/m³
Air temperature = 850°C
Bed thickness = .01 m
Applied acc = 30 g

FIG 8.32 Gas bypass at start-up conditions
Particle density = 2630 kg/m³
Air density = 316 kg/m³
Air temperature = 850°C
Bed thickness = 0.02 m
Applied acc = 5 g

FIG 8.33 Gas bypass at start-up conditions
Particle density = 2630 kg/m$^3$
Air density = 0.316 kg/m$^3$
Air temperature = 850°C
Bed thickness = 0.02 m
Applied acc$^n$ = 10 g

FIG 8.34 Gas bypass at start-up conditions
Particle density = 2630 kg/m$^3$
Air density = 0.318 kg/m$^3$
Air temperature = 850$^\circ$C
Bed thickness = 0.02 m
Applied acc$^n$ = 15 g

FIG 8.35 Gas bypass at start-up conditions
Particle diameter - microns

Particle density = 2630 kg/m³
Air density = .318 kg/m³
Air temperature = 850°C
Bed thickness = .02 m
Applied acc^n = 20 g

FIG 8.36 Gas bypass at start-up conditions
Particle density = 2630 kg/m$^3$
Air density = 0.318 kg/m$^3$
Air temperature = 850°C
Bed thickness = 0.02 m
Applied acc$^N$ = 25 g

FIG 8.37 Gas bypass at start-up conditions
FIG 8.38  Gas bypass at start-up conditions

Particle density = 2630 kg/m³
Air density = 0.318 kg/m³
Air temperature = 850°C
Bed thickness = 0.02 m
Applied acc = 30 g
Particle diameter = 400 microns

Particle density = 2630 kg/m³
Gas density = 0.251 kg/m³
Bed temperature = 1150 °C
Bed thickness = 0.01 m
Applied acc^n = 10 g

FIG 8.39 Gas bypass at steady state conditions
Particle density = 2630 kg/m³
Gas density = .251 kg/m³
Bed temperature = 1150°C
Bed thickness = .01 m
Applied acc^n = 20 g

FIG 8.40 Gas bypass at steady state conditions
FRACTION OF FUEL UNBURNED

PARTICLE DIAMETER - MICRONS

Particle density = 2630 kg/m$^3$
Gas density      = 0.251 kg/m$^3$
Bed temperature  = 1150°C
Bed thickness    = 0.01 m
Applied acc$^n$  = 30 g

FIG 8.41 Gas bypass at steady state conditions
Particle density = 2630 kg/m³
Air density = 1.225 kg/m³
Air temperature = 15°C
Bed thickness = 0.01 m
Applied accn = 5 g

FIG 8.42 Gas bypass at ignition conditions
Particle density = 2630 kg/m$^3$
Gas density = 0.251 kg/m$^3$
Bed temperature = 1150° C
Bed thickness = 0.01 m
Applied acc$^N$ = 30 g

FIG 8.43  Gas bypass at steady state conditions
CHAPTER 9

CONCLUSIONS AND RECOMMENDATIONS

9.1 Review of Results

The findings of the work presented in this thesis fall naturally into three categories which are reviewed in the following sections. These are:

i) combustor design

ii) fluid dynamics, and

iii) combustion.

9.1.1 Combustor Design

9.1.1.1 Distributor Design

a) It has been demonstrated that the use of a distributor, shaped as a conic frustum of small included angle, does not, significantly, aid the initial spreading of bed material during run-up. Fluidisation air is required to achieve complete spreading.

b) It has been found that the use of a conic frustum distributor may be the cause of undesirable fluid dynamics in the bed in the form of unstable gas flows. It should be noted that the mechanism of the instability was not definitely identified but the instabilities did not occur when a cylindrical distributor was used.

c) Pierced sheet distributors were found to have poor mechanical integrity in adverse combustion conditions. Flash-back into the plenum...
chamber or local hot spots, sometimes due to bed fusion, resulted in irreparable damage to the perforations and, in some cases, severe buckling of the sheet.

d) A stand-pipe distributor was designed and tested in a conventional shallow fluidised bed and found to have good mechanical integrity when the stand-pipes were manufactured from stainless steel. Mild steel stand-pipes were found to corrode quickly when coal was burnt in the bed.

e) A cylindrical stand-pipe distributor was manufactured. The manufacture was very time consuming and it is possible that this type of distributor may be excessively expensive to manufacture in a full scale design.

f) A porous stainless steel distributor was fabricated from commercial filter material. Although the fabrication costs were high the resulting distributor had good mechanical strength and showed excellent resistance to high temperatures. It is recommended that future small scale RFB combustion work be carried out using this type of distributor.

g) It was found that the distributor installation method should be such that the distributor is free to expand in an axial direction. This avoids the possibility of serious buckling which can occur with the encastre installation method.

9.1.1.2 Exhaust Port Seal

a) A short, 80 mm long, chimney was manufactured with an integral labyrinth seal and a gas turbine type cooling ring.
b) It was shown that, when operating with exhaust gas temperatures in
the region for 1000 - 1200°C cooling flow rates of the order of 0.0021
kg/cm²/cm were required in order to prevent overheating of the chimney.

c) Provided that satisfactory cooling rates are utilised and the
cooling ring pressure loss is matched to that of the distributor the
cooled labyrinth seal appears to be a good alternative to water-cooled
seals.

9.1.1.3 Exhaust Port Diameter

a) Two exhaust port sizes were tested, under atmospheric conditions,
with a fixed distributor diameter of 200 mm; the ratios of exhaust port
diameter:distributor diameter being 0.6:1 and 0.45:1.

b) It was found that, when the port diameter was changed, both the flow
at which elutriation occurred and the range of fluidisation of the bed
also changed. There appeared to be an "optimum" bed depth at which the
gas flow, at initiation of elutriation, was maximised and the
fluidisation range unchanged. Further work is required in order that
the effect of exhaust port diameter on operating limits can be fully
characterised.

9.1.1.4 Solids Feeding

a) In order that satisfactory axial mixing of solid fuel particles and
bed particles would occur it was found necessary to employ a feeder
design that spread the fuel particles evenly across the bed surface.
b) A cruciform pneumatic feeder head was designed and tested and found to give excellent distribution of fuel. A single feeder head was found to be sufficient to feed fuel to the surface of a bed of 180 mm (surface) diameter by 80 mm axial length.

9.1.2 Fluid Dynamics

9.1.2.1 Operating Regime - Descriptive Parameters

a) It has been shown that it is possible to present fluidisation data in a form compatible with that used in the performance analysis of gas turbine engines.

b) The use of combined distributor and bed pressure loss as a percentage of the plenum pressure was presented as a function of the plenum flow function, $Q_p$.

c) It was shown how the use of these parameters can be used in the analysis of combustor fluid dynamics.

9.1.2.2 Initial Run-up

a) It has been found that it is necessary to employ fluidisation gas flow in order to achieve even distribution of bed material across the distributor during initial run-up. This finding holds for both conic frustrum and cylindrical distributors when the mass of local material will produce a radially thin bed ($\leq 20\%$ distributor radius).

b) Approximately 40\% of the minimum fluidisation flow was required to
achieve satisfactory bed distribution with bed particles having mean sizes in the range .212 mm to .714 mm.

9.1.2.3 Minimum Fluidisation and Fully Supported Fluidisation

a) The minimum fluidisation flow function and fully supported flow function were determined for a number of particle size ranges between .180 mm to .25 mm up to .710 mm to .85 mm. Bed depths employed were equivalent to mass distributions of 10, 12.5 and 15 kg/m² and these beds were subjected to applied radial accelerations in the range 5 to 40 times normal gravity.

b) A new correlation is proposed for the minimum fluidisation condition for Galileo number in the range $4 \times 10^4$ to $10^6$ and is as follows:-

$$Re_{mf} = (13.873^2 + 0.01724 \text{ Ga}^3)^{1/2} - 13.873$$

c) A new correlation for the fully supported condition is proposed for Galileo numbers in the range $4 \times 10^4$ to $10^6$, viz:-

$$Re_{fs} = (4.3665^2 + 0.01868 \text{ Ga}^3)^{1/2} - 4.3665$$

d) The scatter of data around these correlations suggests an accuracy of about $\pm 30\%$ which is comparable with the correlation proposed by Wen and Yu.

9.1.2.4 Elutriation

a) It has been confirmed that the flow rate at which elutriation begins
is determined by the size of smallest particle in the bed (assuming that all the particles in the bed are the same density).

b) It has been shown that, with the combustor design employed in this project, the operating range, \((Q_p \text{ (elutriation)}) : Q_p \text{ (full support)})\), would be limited to 2:1 unless high pressure losses, \((> 5\% \text{ plenum pressure})\), were accepted.

c) It was found that a reduction in exit port diameter would increase the plenum flow function, and hence fluidisation velocity, at which elutriation would begin.

d) Further, a reduction in exit port diameter was found to change the operating range. The change was dependent upon bed thickness. For beds 10 mm thick, the operating range reduced, for thicker beds \((t = 15 \text{ mm})\) the operating range was essentially unchanged and for beds of 20 mm thickness the operating range was increased.

e) A theoretical model of elutriation, proposed by Chevray et al (39), was tested and found to overestimate the flow rate at which given sizes of particles would elutriate.

9.1.2.5 Instabilities

a) When operating the RFB with a conic frustum distributor installed, unstable operating conditions were encountered, with all bed particle sizes and bed thicknesses tested at atmospheric temperature and pressure. The instability took the form of pressure and flow fluctuations.
b) The frequency of the fluctuations appeared to be independent of the bed mass and particle size range suggesting a mechanism linked to the combustor design rather than the bed or particle properties.

c) The instability was not encountered when a cylindrical distributor was installed and thus it is concluded that the mechanism is linked to the shape of the distributor.

d) When fluidising a radially thick bed the instability appeared to initiate elutriation under some operating conditions.

e) The actual mechanism of the fluctuations could not be definitely identified but may be akin to the "bounce" condition described by Deinken (97).

9.1.2.6 Solids Mixing

a) The mixing of coal particles in a bed composed of sand particles was found to be satisfactory in the radial direction, i.e., between the bed surface and distributor, provided the fluidising velocity was greater than about twice minimum fluidising velocity.

b) Mixing in the axial direction, i.e., perpendicular to the average gas flow, was found to be very poor although there was a tendency for coal particles to migrate towards the combustor baseplate.

c) Satisfactory mixing of coal particles in both radial and axial directions was achieved by using a coal feeder that spread the coal particles over the whole bed surface.
9.1.3 Combustion

9.1.3.1 Start-Up

a) Start-up was attempted in a fully enclosed combustor using premixed propane and air following the criteria established by Metcalfe (41) who experimented on a combustor which had a exhaust port open to the atmosphere.

b) It was found that flame front stabilisation was very difficult under the operating conditions described by Metcalfe ie $1/4 \times U_{mf}$ (cold).

c) The use of lower fluidising velocities resulted in flash back into the plenum chamber which caused severe damage to the pierced steel distributors.

d) The use of metal inserts in the combustor which produced diverging bed and freeboard walls and a decelerating gas velocity aided flame front stabilisation and allowed partially successful preheating of the bed. The success was marred by damage to the distributor due to overheating.

e) Prior to the ring inserts being installed it was often observed that, after ignition, downstream temperatures were considerably higher than temperatures recorded in the bed. These temperatures were very stable and it was concluded that the flame front had stabilised either on the coal feed pipe or the igniter electrodes.

f) Air used to cool the coal feed pipe and exhausted into the combustor.
freeboard was found to improve the chances of stabilising a flame front on the bed surface. The effects were not repeatable and hence were not quantified.

9.1.3.2 Gas Combustion

a) Satisfactory steady-state combustion was not achieved when using a pierced steel distributor. The expected even temperature distribution in the bed was not achieved and variations of ± 50°C about a mean of 750°C were not uncommon.

b) Control of bed temperatures when burning propane gas in a bed fluidised on a pierced steel distributor was very difficult. Small changes in fluidising air flow rate or propane flow rate could cause rapid bed cooling or bed fusion.

c) Local high temperatures in the bed caused severe buckling and burning of the pierced steel distributors.

d) Combustion was attempted in the RFB with a "stand-pipe" distributor installed. A premixed gas:air torch mounted in the freeboard was used for preheating the bed; the large bed particles (dp = 1 mm) precluded the use of premixed propane:air combustion in the freeboard due to the high gas mixture velocity required for fluidisation.

e) A heat output of 25 kW from the torch was found to be insufficient for bed heating in this configuration. It was discovered that the flame from the torch was being swept away from the bed surface by the swirling gas flow in the freeboard and thus the bed was being heated by
radiant heat transfer alone. The low emissivity of a premixed propane/air flame was insufficient to give adequate bed heating. The use of the standpipe distributor was discontinued as there was insufficient space to install a larger gas torch and no liquid fuelled torch was available.

f) A porous stainless steel distributor was designed and a limited series of trials were conducted with the combustor exhaust open to the atmosphere.

g) Preheating of a bed of silica sand particles, .18 mm - .25 mm, with premixed propane and air was found to be relatively easy and none of the previous problems associated with flame stability and mechanical integrity were encountered.

h) Steady state combustion of premixed propane and air mixtures was also successfully achieved with the porous distributor installed. However, the range of fluidising velocities in which the combustion could be controlled was severely limited. Equally, the fuel to air ratio and applied acceleration were also limited to a relatively narrow range. The reasons for this behaviour were not established but may have been due to large losses of fuel gas through the labyrinth seal which had been designed to operate with a pierced distributor of much smaller pressure loss.

1) Within the range of conditions determined for stable combustion it was found that the response of the combustor to changes in fuel to air ratio was slow; a behaviour quite unlike that found by other workers. This slow response, however, did give the advantage of very easy
control. Again, this behaviour could be linked to the imbalance in gas flows caused by the mismatch between the distributor pressure loss and labyrinth seal pressure loss characteristics.

j) Bed temperatures in the range 850-1000°C could be achieved after a preheating time of between 10-20 minutes and as stated in (i) were easily controllable.

k) It is thought that the ease of control of start-up and steady-state combustion was due to induction of atmospheric air into the freeboard space providing a supply of oxygen extra to that in the fluidising air flow. This suggestion is supported to some extent by the observation that, during earlier trials, the air used to cool the coal feed pipe during bed preheating appeared to aid flame stabilisation in the freeboard. However, this behaviour did not occur consistently and, at the time, was thought to be coincidental.

9.1.3.3 Coal Combustion

a) Transition from propane to coal combustion was never satisfactorily achieved although it was found that some degree of coal combustion could be achieved with the support of a constant supply of propane.

b) Several transition techniques were assessed and each appeared to give rise to characteristic behaviour.

c) The first technique evaluated was to reduce, gradually, propane flow rate and, simultaneously, gradually increase coal flow rate. This technique usually resulted in loss of bed temperature but, if the coal
flow rate was increased too rapidly, bed fusion would occur very quickly.

d) The second technique studied was that developed by Metcalfe in which the bed was heated to a steady 950-1000°C by propane combustion and then coal flow rate started at a predetermined rate and propane flow stopped simultaneously. This technique always resulted in rapid bed cooling. It appeared that there was insufficient heat capacity in the bed to initiate coal combustion. The coal used, a hard anthracite, had a relatively low volatile content and thus is slow to start burning. If the coal had been high in volatiles the technique for transition described by Metcalfe may have met with greater success due to the rapid release of heat from volatiles combustion.

e) The final technique attempted was to precharge the bed with a mass of coal that should have been sufficient to maintain bed temperature after the propane had been shut off. This technique seemed very sensitive to the mass of the initial charge of coal. Too small a mass resulted in rapid bed cooling when the propane was shut off whilst, too large a mass caused bed fusion. It was very difficult to estimate the required mass of coal since the dynamics of combustion of the charge during preheat is determined by the initial distribution of particle sizes which could only be determined approximately and so it was virtually impossible to use similar size distributions in the various charge masses. Therefore consistent combustion dynamics could not be guaranteed.

9.1.4 Important findings

a) Porous distributors made from sintered stainless steel have been
found to have better mechanical integrity than pierced sheet distributors.

b) Tapered distributors have been found to give rise to instabilities in the fluidising air flow and do not aid the initial distribution of bed material. It is concluded that cylindrical distributors should be used henceforth.

c) An aircooled exhaust seal has been developed that allows operation at combustion temperatures up to 1200°C without the use of external cooling media and heat exchanges.

d) Two correlations are proposed for the estimation of $U_{mf}$ and $U_{fs}$; both have an expected average error of 30% in the range of Galileo number between $4 \times 10^4$ to $10^6$.

e) It has been shown that the flow rate at which elutriation begins and the range of fluidisation flows between full support and elutriation are effected by the diameter of the exhaust port. Further more, for a given exhaust port to distributor size ratio there appears to be an optimum bed depth.

f) A model for the prediction of elutriation conditions proposed by Chevray et al has been shown to overpredict the flowrate at which given particles start to elutriate.

g) Adequate radial mixing of solids in the bed has been shown to occur when the fluidisation index ($U/U_{mf}$) is greater than 2:1. Further, it has been shown that in order to achieve good axial solids mixing it is
necessary to utilise a fuel feeder that produces an even initial
distribution over the bed surface.

h) It has been found that preheating of the bed using premixed fuel gas
and air is very difficult with a fully enclosed combustor. When the
combustor exhaust was open to the atmosphere, allowing the induction of
air into the freeboard, preheating the bed was found to be relatively
easy.

i) Steady-state gaseous combustion was found to be unstable with a
fully enclosed combustor leading to distributor overheating and bed
fusion. Using an exhaust port open to the atmosphere resulted in
considerably improved combustion stability.

j) Transition to coal combustion from gas combustion was found to be
impossible using manual control. It is concluded that an automatic
control system is required if this transition is to be achieved
satisfactorily.

k) Due to the difficulties encountered with both preheating and
steady-state combustion and with combustor integrity using premixed gas
fuel and air it is believed that alternative methods of bed heating
should be developed.

9.2 Conclusions

The potential advantages of a rotating fluidised bed combustor ie.
high intensity combustion, increased range of fluidising velocity and
turndown, and rapid response, were explored but they were found not to
be achievable due to three categories of problem.

1) **Difficulties of control of combustion both in transient and steady-state**

a) Preheating of the bed using premixed propane and air was very difficult in a fully enclosed combustor. The behaviour of the combustor was very different when the exhaust port was open to the atmosphere. It appeared that induction of air into the freeboard occurred with an open combustor and this greatly assisted flame stabilisation.

b) Steady-state combustion of propane was also found to be unstable when the combustor was fully enclosed. This led to distributor overheating, distortion and bed fusion. An open ended combustor exhibited considerably improved combustion stability.

c) Transition from steady-state combustion of propane to coal combustion was found to be impossible using manual control. It is concluded that the development of an automatic control system is essential if this transition is to be achieved satisfactorily.

d) In view of the above difficulties with the use of premixed propane and air and additional problems with mechanical integrity it is recommended that alternative methods of bed heating should be explored.

2. **Geometric Scale**

Significant development difficulties were encountered by having to

372.
work with a combustor of small geometric scale. The difficulties encompassed installation of ignition and fuel feeding components and in process control. In particular, the freeboard space was insufficient to allow installation of high output pilot burners which would have obviated the start-up heating difficulties.

3. **Design for Satisfactory Mechanical Integrity**

Pierced sheet distributors were found to have poor mechanical integrity in adverse combustion conditions. Stand-pipe and porous distributors were found to have significantly better mechanical integrity but it must be recognised that it will probably be necessary to make a trade off between strength and pressure loss. Further, it is essential to ensure proper integration of the functions of all the items of the combustor eg. distributor and aircooled seal pressure loss characteristics.

Fluidisation and solids mixing were found to be satisfactory. The most serious problem that arose was the bed instability which was remedied by replacing the conic frustum distributor, originally installed, with cylindrical distributors.

Fuel feeding from an atmospheric pressure hopper to a hopper designed for pressurised operation was achieved using an air lock system controlled by a timing circuit. This system was essentially trouble-free.
9.3 Recommendations

9.3.1 Combustion Applications

The principal recommendations for future work on RFB combustion are three-fold, and are as follows:

a) It is evident that control of the combustion process by present manual techniques requires a great deal of skill in the adjustment of air and fuel feed rates. It is recommended that a project be undertaken to develop an automatic control system suitable for application to the RFB. This system would require some degree of anticipatory or algorithmic control. This would be likely to require development of a microprocessor-based system. Experiments which will generate information from which a realistic model for combustion can be formulated are urgently required.

b) Since the control system is interconnected with combustion process behaviour; development of a comprehensive model of combustion in the RFB is an important requirement. This model could be tested against experimental results reported by Dermircan (40) and Metcalfe (41). If adequate, the model could then be used to establish an effective control strategy.

c) Perhaps the most important future work that should be done on the RFB is a realistic assessment of its potential applications both from a technical and economic standpoint. The questions that must be answered
are:-

1) Can the RFBC give the desired performance more reliably and economically than alternative combustion concepts in each particular application?

ii) Does the application justify the complexity of the RFBC?

iii) Will the RFBC have low long-term operating costs that would outweigh the expected high capital cost of the RFB installation?

The decision to be made on future RFB combustion development should rest on such an assessment.

9.3.2 Other Applications

Since there are applications of the RFB outside the combustion field in which the ability of the RFB to handle large gas throughputs at moderate temperatures would be an advantage, eg drying, gas-solid reactions, gas clean-up, it is felt that some further low temperature studies may be worth mounting.

In this connection further work is required on the effects of RFB geometry on mixing and elutriation, and development of techniques for solids injection and removal would be of considerable general value.
APPENDIX A

HEAT TRANSFER MODELS FOR AIR-COOLED
CHIMNEY AND AIR COOLED DISTRIBUTORS.

A.1 Air-Cooled Chimney

A.1.1 Background

The insulating properties of a film of cool air are used in a gas turbine to maintain the temperature of combustion chambers and turbine blades at acceptable levels. Modelling of the heat transfer from hot gases through the cool gas film into the metal is a complex subject due to the mixing of the hot gases and the cool film. The mixing limits the distance over which the cooling film is effective. Measurements of the cooling characteristics of film cooling devices lead to the derivation of a 'cooling effectiveness' which is different for different designs. This cooling effectiveness cannot be derived analytically and must be measured by experiment. If it is assumed that no mixing takes place an estimate of the heat transfer from hot gas to cooled metal can be made.
A.1.2 Heat Transfer Model

Suppose the temperature of the hot gas, the metal chimney and cold feed air are $T_g$, $T_m$ and $T_c$ respectively. Suppose also that the initial temperature of the cooling film is $T_f1$ ($=T_c$), the mass flow of cooling film air is $\dot{m}$, and the final temperature of the film is $T_f2$.

The heat flow from the hot gas to the film can be estimated by

$$Q = -h.A. \ (\text{Log Mean Temperature Difference})$$

Thus

$$Q_1 = -h \cdot 2\Pi r.l \cdot \frac{T_f2 - T_f1}{\ln (T_g - T_f2)}$$

\[ (T_g - T_f1) \]
Evaluation of the heat transfer coefficient, $h$, is not straightforward but may be estimated from the Nusselt relationship.

$$Nu = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4}$$

for most gases $Pr=1$ (approx.)

Expanding $Nu$

$$\frac{h \cdot 2r}{k} = 0.023 \cdot Re^{0.8}$$

For interface 1

$$h_{1,2r} = 0.023 \cdot k_{1} \cdot Re^{0.8}$$

thus

$$Q_{1} = 0.0723 \cdot k_{1} \cdot Re^{0.8} \frac{(T_{f2} - T_{f1})}{\ln (T_{g} - T_{f2})} \ln (T_{g} - T_{f1})$$

A.2

The heat flow from the gas film to the metal chimney is given by

$$Q_{2} = h_{2} \cdot A \cdot LMTD$$

$$= h_{2} \cdot 2 \cdot \Pi \cdot r \cdot l \cdot (T_{f2} - T_{f1})$$

$$\ln (T_{m} - T_{f2})$$

$$\ln (T_{m} - T_{f1})$$

378.
The heat transfer coefficient again may be estimated from the Nusselt equation with the result that

\[ Q_2 = -0.0723 \cdot k_2 \cdot l \cdot Re^{0.8} \frac{(T_f 2 - T_f 1)}{\ln(T_m - T_f 2) - (T_m - T_f 1)} \]  

\[ \text{A.3} \]

Further \[ |Q_2| = |Q_1| = \dot{m} \cdot C_p \cdot (T_f 2 - T_f 1) \]  

\[ \text{A.4} \]

Heat conduction through the chimney wall gives:

\[ Q_2 = -k_2 \cdot 2 \cdot \pi \cdot r \cdot l \cdot \frac{(T - T_0)}{t_2} \]  

\[ \text{A.5} \]

The heat transfer from the metal chimney to the cold gas is given by:

\[ Q = -h \cdot A \cdot (T_0 - T_c) \]

whence \[ Q_3 = -h_3 \cdot 2 \cdot \pi \cdot r \cdot l \cdot (T_0 - T_c) \]

\[ = -0.0723 \cdot l \cdot k_4 \cdot Re^{0.8} \cdot (T_0 - T_c) \]  

\[ \text{A.6} \]

In equilibrium

\[ Q_3 = Q_2 \]

Thus using the initial conditions:

\[ T_g = 1000^\circ \text{C} \]
\[ T_{f 1} = T_c = 20^\circ \text{C} \]
\[ t_1 = 3 \text{mm} \]
\[ r = 50\text{mm} \]
\[ t_2 = 2\text{mm} \]
\[ l = 40\text{mm} \]
\[ \dot{m} = 0.000084\text{ kg/s} \]
\[ C_p = 1.03\text{ KJ/kg K} \]
\[ \omega = 31.32\text{ rad/s (10 x gravity)} \]
\[ k_1 = 81 \times 10^{-3}\text{ W/MK} \]
\[ \mu_1 = 50.9 \times 10^{-6}\text{ Ns/m}^2 \text{ (exhaust gas taken to be air equivalent)} \]
\[ k_3 = 21.8\text{ W/MK @500°C} \]
\[ k_2 = k_4 = 26.1 \times 10^{-3}\text{ W/MK} \]
\[ \mu_2 = \mu_3 = 18 \times 10^{-6}\text{ Ns/m}^2 \text{ @20°C} \]

It was estimated that the metal temperature, \( T_m \), at equilibrium would be approximately 700°C.

The errors resulting from the assumed heat transfer models and heat transfer coefficients are not easily quantifiable but the impact of these errors can be minimised by making allowances for increasing cooling flows in the mechanical design of the film cooling device.

A.2 Air-Cooled Distributors

A.2.1 Background

Modelling of the heat transfer to and from distributors is rather simpler than the problem set out in the previous section. A great deal of work has been done on bed-to-wall heat transfer and it is a relatively simple matter to choose a heat transfer coefficient of the
correct order of magnitude. Equally, estimation by experience gives a good approximation of the distributor to fluidising gas heat transfer coefficient.

A.2.2. Heat Transfer Model

Heat transfer from the bed to the distributor is described by the equation.

$$Q_1 = -U_1 A_d (T_b - T_i)$$  \hspace{1cm} A.6

Heat transfer through the distributor is given by

$$Q_2 = -k A_d \frac{(T_i - T_o)}{t}$$  \hspace{1cm} A.7

Heat transfer from the distributor to the fluidising gas is given by

$$Q_3 = -U_2 A_d (T_o - T_g)$$  \hspace{1cm} A.8

(if it is assumed that the heat flux to the fluidising gas as it passes through the distributor is small. This is adequately true for all but porous metal distributors.)
In equilibrium:

\[ Q_1 = Q_2 = Q_3. \]

If the initial values of temperature are

\[ T_b = 1000^\circ C \]
\[ T_g = 100^\circ C \]
\[ T_i \text{ and } T_o \text{ = unknown} \]

and heat transfer coefficients are given by

\[ U_1 = 75 \text{ W/m}^2\text{K.} \text{ (Broughton and Elliot(32))} \]
\[ U_2 = 30 \text{ W/m}^2\text{K.} \text{ (estimate)} \]

with a distributor thickness

\[ t = 1\text{mm} \]

and thermal conductivity, typical for mild steel, of

\[ K = 40 \text{ W/mK.} \]

The equations can be solved simultaneously to give:

\[ Q = 969.55 \text{ Watts} \]
\[ T_i = 742.99^\circ C. \]
\[ T_o = 742.51^\circ C. \]

These results were found to agree well with temperatures measured in combustion trials using the Conidur distributor, see Chapter 7.

Changing the value of \( k \) in equation A.7 to a value typical for stainless steels of 34 W/mK resulted in little change in the solutions for \( Q_1, T_o \) and \( T_i \). The solutions of the equations gave

\[ Q = 969.46 \text{ Watts} \]
\[ T_o = 743.02^\circ C \]
\[ T_i = 742.45^\circ C. \]
APPENDIX B

RFBC Engineering Drawings

The drawings included in this appendix show general assemblies of; the Pressurised Rotating Fluidised Combustor, the Gas Sampling Section of the exhaust ducting and the Ejector Detuner installation designed by Cullum Detuners Ltd. Detail drawings of the important items of the combustor are also included. For brevity, drawings of items whose dimensions were not important to the fluid dynamics and combustion processes have not been included. Dimensions for these items can be scaled from the PRFBC General Assembly with adequate accuracy to allow their reproduction.

Material specifications

Conidur distributors                  ANSI 321
Porous distributor                   ANSI 316
Other stainless steels               ANSI 321

Footnote

Due to the size of the general assembly drawings, it has been necessary to insert them in a document wallet attached to the rear cover.
12 OFF HOLES Ø4 EQUISPACED.
EACH ROW

206 Ø

10

80

2°

200 Ø

DIM. m.m.: 1'U.O.S.
NO. OFF 1
MAT: CONIDUR
DUG. NO. PRC. 01.11

SCALE 1:2

PROJECTION

TITLE CONIC DISTRIBUTOR
12 OFF HOLES Ø4 EQUISPACED EACH ROW

DIMS. mm: 1 U.O.S. SCALE 1:2

NO. OFF 1

MATL: CONIDUR □ PROJECTION

DRC. NO. PRC.01.11 TITLE CYLINDRICAL DIST.
12 HOLES EQUISPACED D&T M4

\[ \phi 180^\circ \pm 0.1 \]
\[ \phi 200^\circ \pm 0.3 \]

**SCALE 1:2**

<table>
<thead>
<tr>
<th>DIMS.</th>
<th>m.m.</th>
<th>I.U.O.S.</th>
</tr>
</thead>
<tbody>
<tr>
<td>NO. OFF</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>MAT.</td>
<td>SS</td>
<td></td>
</tr>
<tr>
<td>DRG. N#</td>
<td>PRC.01.12</td>
<td>TITLE DIST MOUNTING RING</td>
</tr>
</tbody>
</table>

386.
### Dimensions

<table>
<thead>
<tr>
<th>DIMS.</th>
<th>m.m. ± 1 U.O.S.</th>
<th>SCALE 1:4</th>
</tr>
</thead>
<tbody>
<tr>
<td>NO. OFF</td>
<td>1</td>
<td>PROJECTION</td>
</tr>
<tr>
<td>MAT.</td>
<td>SS</td>
<td>PROJECT</td>
</tr>
<tr>
<td>DRG. NO.</td>
<td>PRC. 01.13</td>
<td>TITLE COMB R TOP PLATE</td>
</tr>
</tbody>
</table>
VIEW AT 'A'
5 x FULL SIZE

36 HOLES @ 10°, ø1

5 SLOTS IDENTICAL DIMS

4 HOLES @ 90°, ø6

<table>
<thead>
<tr>
<th>DIMS.</th>
<th>m.m.±1 U.O.S.</th>
<th>SCALE 1:2</th>
</tr>
</thead>
<tbody>
<tr>
<td>NO.OFF</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>MAT.</td>
<td>SS</td>
<td>☒ PROJECTION</td>
</tr>
<tr>
<td>DRG.Nº</td>
<td>PRC.01.21/1</td>
<td>TITLE AIRCOOLED SEAL</td>
</tr>
</tbody>
</table>
8 Holes Ø 19 @ 45°
PCD 216

Scale 1:2

DIMS. | m.m. | 1U.O.S.
--- | --- | ---
NO.OFF | 1
MAT. | SS
DRG.N° | PRC.01.21/2

PROJECTION

TITLE SEAL CYLINDER

388
<table>
<thead>
<tr>
<th>DIMS.</th>
<th>m.m.±1 U.O.S.</th>
<th>SCALE 1:2</th>
</tr>
</thead>
<tbody>
<tr>
<td>NO. OFF</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>MATL.</td>
<td>S.S.</td>
<td>PROJECTION</td>
</tr>
<tr>
<td>DRG. N°</td>
<td>GW0.1</td>
<td>TITLE COMBUSTOR RING</td>
</tr>
</tbody>
</table>

390.
1/2 BSP UNION

80

R80

13 mm OD TUBE 0.5 mm WALL

DRILL TO SUIT EXHAUST DUCTING

DIMENSIONS: mm

<table>
<thead>
<tr>
<th>Dim.</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>NO. OFF</td>
<td>1</td>
</tr>
<tr>
<td>Mat.</td>
<td>SS</td>
</tr>
<tr>
<td>DRG. NO.</td>
<td>GWO. 2</td>
</tr>
</tbody>
</table>

SCALE 1:4

PROJECTION

TITLE COAL FEEDER
APPENDIX C

EQUIPMENT SPECIFICATION

C.1. H.P. Compressor

C.2. Gas Analysers

C.2.1 Oxygen

C.2.2 Oxides of Carbon: CO and CO2

C.2.3 Unburned Hydrocarbons UHC

C.2.4 Oxides of Nitrogen NO and NO2 (NOx)

C.2.5 Gas Sample Transfer Line

C.2.6 Oxides of Sulphur SO2 and SO3

C.2.7 Particulates

C.2.8 Other Constituents

C.2.9 New Techniques
C.1. H.P. Compressor

Make and Model: Howdenair AW 250

Type: Double Screw Compressor

Drive: 250 kW Electric Motor

Rated Output: 1000 scfm @ 7 bar absolute.
C.2 GAS ANALYSERS

C.2.1 Oxygen

Measurements of oxygen concentration in exhaust gases may be used as a way of monitoring and controlling fuel:air ratio, FAR, in boilers (1). Oxygen analysers are simple and rugged and depend on the paramagnetic properties of oxygen molecules for the measurement of O₂ concentration.

Determination of fuel:air ratio from oxygen analysis (2) provides a useful cross-check on the values of FAR determined from other sources (direct measurements, carbon balance and heat balance).

The analyser utilised in the gas analysis system was a Taylor Servomex model OA273, a specification of which appears at the end of the section. Although this analyser was found to be rather sensitive to sample flow rate where, at high flow rates the analyser readings varied widely about a mean value, it was simple to use and held its calibration for long periods of time despite warnings to the contrary made by the manufacturer. Stable readings were achieved by careful control of the flow of sample gas to the analyser.
Oxygen Analyser

Make and Model : Taylor Servomex DA273

Power Supply : 240V 50Hz

Ranges : 0 - 5%
         0 - 25%
         0 - 100%

Accuracy : ± 1% FSD (+ abs. error of zero and span gases)

Operating Range : 0 - 50°C

Ambient Temp. Effect : ± 0.05% O₂ per ± 1°C
                      for ± 5°C change

Stability : recalibrate hourly

Response : 90% in less than 7 sec.
           with sample cell flow of 100 ml./min.

Sample Requirements

Flow : 0 - 150 ml./min
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure</td>
<td>atmospheric</td>
</tr>
<tr>
<td>Temperature</td>
<td>100°C max.</td>
</tr>
<tr>
<td>Gas Condition</td>
<td>dry</td>
</tr>
</tbody>
</table>
C.2.2 Oxides of Carbon: CO and CO\textsubscript{2}

Both carbon monoxide and dioxide are expected to be detected in the flue gas. It is desirable that the concentration of the former should be small, whilst that of the latter should be significant, actual levels depending upon the fuel to air ratio.

The usual method of measuring concentrations of carbon oxides is to make use of the absorption of infra-red radiation by the gas molecules. Infra-Red Gas Analysis, IRGA, is widely used for detecting particular gases because their molecules absorb infra red radiation in particular frequency bands. Unfortunately, these absorption bands tend to overlap and, as good resolution is required from the detection equipment when gas mixtures are to be analysed, it may be necessary to use other techniques for the analysis. In some cases, however, such as the automotive industry, very high resolution and sensitivity are not required.

Existing IRGA analysers in the Department of Mechanical Engineering were manufactured by the Horiba Company, Japan, for the automotive industry. These have poor sensitivity and resolution. The analysers are particularly poor for detecting the low concentrations of CO expected from the R.F.B. combustor, the operating accuracy of detection being approximately $\pm$ 2000 ppm. Although no regulations appear to exist for emissions of CO a quote from the SRC publication "Combustion Generated Pollution" (1) indicates accepted emission levels:— "The levels of CO emission from large boiler furnaces in low. It is normally kept below 0.01%, at all loads, in the flue gas." Therefore an analyser capable of measuring CO concentrations between 0 and 1000ppm with high
accuracy, better than 10ppm, is required. To complement this, an
analyser for CO₂ with a range of 0-20% with an accuracy of, say ± 0.5%
FSD ie. ± 1000ppm, is desirable.
Oxides of Carbon Analyser

Make and Model : Horiba Mexa 310

Power Supply : 240V 50Hz

Ranges

\begin{itemize}
  \item \textbf{CO} : 0 - 6\%
  \item \textbf{CO}_2 : 0 - 20\%
  \item \textbf{Accuracy} : ± 1\% FSD
  \item \textbf{Operating Range} : 0 - 30°C
  \item \textbf{Ambient Temp. Effect} : nil
  \item \textbf{Stability} : not specified
  \item \textbf{Response} : not specified
\end{itemize}

Sample Requirements

\begin{itemize}
  \item \textbf{Flow} : not specified
  \item \textbf{Pressure} : atmospheric
\end{itemize}
Temperature : atmospheric
Gas Condition : dry
C.2.3 Unburnt Hydrocarbons, UHC

During the combustion of a hydrocarbon fuel very complex chemical reaction chains may occur (1,3) that can give rise to a vast number of transitory and stable species. Among these species, carbon - hydrogen molecules, rings and chains seem to have great significance. If quenching to temperatures below 500°C should occur, due perhaps to adverse mixing conditions, these carbon - hydrogen species may escape from the combustion zone. The escape of these species represents a loss of heat output from the combustion device, i.e. combustion inefficiency, and their detection is obviously necessary.

Several methods are available for the detection of UHC. If specific species are known to be present then Infra red absorption or gas chromatography may be employed for their detection. If, as is usual, a range of species may be present the recommended method of detection (5) is that of Flame Ionisation Detection. Here, a sample of exhaust gas is fed to a small hydrogen flame which is positioned in a short metal tube to which is applied a high electrical potential. Hydrocarbon molecules entering the hydrogen flame ionise and the carbon ions released are attracted to the metal tube by the electrical field. On reaching the metal wall the carbon ions change the potential of the tube and this change is detected and related to the number of ions; output is then a measure of the carbon number.

The analyser chosen for the gas analysis system was an Analysis Automation Model 520 Series II. This analyser is a new design based on proven U.S. Technology. The analyser is simple to use and capable of good accuracy over a wide range of concentration.
Unburned Hydrocarbons Analyser

Make and Model : Analysis Automation
520 II

Power Supply : 240V 50Hz

Ranges :
0 - 10 ppm
0 - 25 ppm
0 - 100 ppm
0 - 250 ppm
0 - 1000 ppm
0 - 2500 ppm
0 - 10,000 ppm
Auto

Accuracy : 0.1% FSD

Operating Range : 0.50°C

Ambient Temp. Effect : nil

Stability : Better than ±1% FS in
24 hours

Response : 90% in 2 sec
Sample Requirements

Flow : 0.5 – 3 l/min
Pressure : atmospheric
Temperature : up to 200°C
Gas Condition : wet or dry
Supplier Address : Analysis Automation
                 Eynsham
                 Oxon.
C.2.4 Oxides of Nitrogen NO and NO2 (NOx)

Oxides of nitrogen are produced from fuel bound nitrogen and from atmospheric nitrogen. It has been found that NOx emissions are lower from fluidised bed combustors than from conventional combustors, largely due to lower combustion temperatures. The vast majority of the NOx produced appears to come from the fuel bound nitrogen (6). Measurements of NOx emitted from an FBC may be compared with expected levels based on the fuel bound nitrogen only and it may be possible to link excess levels of NOx with free board combustion, however, the chemistry of nitrogen fixation is extremely complex and subject to considerable uncertainty at the current time. It has been reported (6,7) that levels of NOx emission from FBC's can be less than those expected based on fuel bound nitrogen, an effect due, possibly, to complex reactions between NO and carbonaceous compounds. It is, therefore, important to measure emissions of NOx from any development combustion system in order to establish the effects of combustor design, and fuel and additive properties.

The analyser chosen for the system was the Thermo-Electron 10A. This is a chemiluminescence analyser capable of fast response with high resolution.

The basic analyser is capable of satisfying full EPA specifications.
Oxides of Nitrogen Analyser

Make and Model : Thermo Electron 10A

Power Supply : 250V 50Hz

Ranges : 0 – 2.5 ppm
         0 – 10 ppm
         0 – 25 ppm
         0 – 100 ppm
         0 – 250 ppm
         0 – 1000 ppm
         0 – 2500 ppm
         0 – 10,000 ppm

Accuracy : ± 0.5%FS

Operating Range : 0 – 40°C

Stability : ± 1% in 24 hours

Response : 90% in 0.7 sec (NO)
           90% in 1.0 sec (NOx)

Sample Requirements

Flow : 2 scfh
Pressure : atmospheric

Temperature : up to 150° C

Gas Condition : preferably dry (will accept wet gas with suitable modifications)

Supplier : Analysis Automation Oxon
C.2.5 Gas Transfer Line

Make and Model : Thermo Electron Model 212

Power Supply : 250 V 50Hz

Gas Contact Surface : Teflon

Max. Temperature : 232°C thermostat controlled

Max. Pressure : 1000 lb p.s.i @ 204°C

Core Diameter : 1/4" nominal

Supplier : Analysis Automation Oxon.
C.2.6 Oxides of Sulphur, SO₂ and SO₃ (SOₓ)

Oxides of sulphur are produced from fuel bound sulphur. SOₓ production can be avoided by removing fuel-bound sulphur before combustion occurs. SOₓ can also be removed from exhaust gases before emission into the environment by chemical scrubbing. Both of these processes are expensive in hardware and energy and reduce the cost effectiveness of the overall process.

In FBC systems, fuel-bound sulphur may be removed by the addition of calcium and magnesium compounds to the bed (eg. 8,9). Even with these measures some SOₓ is emitted and its concentration may be assessed via gas chromatography or ultra violet absorption.

A gas chromatograph already existing in the Mechanical Engineering Department had been found to be unreliable and required very expensive calibration gas mixtures. Analysis equipment utilising U-V absorption is also very expensive and funds were not available for the purchase of a suitable unit. For these reasons analysis of exhaust gases for SOₓ emissions was not available.
C.2.7 Particulates

In recent years much effort has been expended on the development of methods of measuring the solids concentration in a flowing gas stream, (eg. 10, 11; 12, 13). There are two reasons that measurement of solids concentration is so important. Firstly, particulate solids pose a considerable technological problem in the design of heat exchanges and turbines. Secondly, the particulates, especially the sub-micron sizes with irregular shapes, may pose a health hazard similar to asbestosis and silicosis (14, 15).

Two generic types of measurement technique are proposed in the literature, viz, intrusive and non-intrusive. The intrusive type may be a simple in-line filter but is normally a wide-mouthed probe sampling at iso-kinetic flow rates, the gas sample being ducted to any of a variety of collection devices: filters or impactors (10) cyclone trains (11). These result in direct measurements of particle size and concentration. The non-intrusive method is based on the diffraction of a laser beam eg. (12, 13). Here, a beam of coherent light is passed through the flue gas and the light diffracted by particles in the flow is collected by photo-diodes or photo-multipliers as a continuous or stepwise function. The intensity of the diffracted light and the corresponding angle of the diffraction can be analysed to give a measure of the particulates present in the gas stream. Both methods are under development in establishments throughout the world.

BS3405 (16) describes a technique for measuring solids burden using a cyclone train. This technique has, however, been found to result in erroneous results (17). It was found that when particulates were
present in a sampled gas stream the pressure loss across the cyclone train was equivalent to that of a smaller gas flow rate (the difference depending on the material used, particle size and concentration). The standard technique, however, requires calibration of the cyclone train for gas flow rate versus pressure loss. Therefore, in order to use the specified technique successfully, a comprehensive set of data and calibration curves for a given cyclone train is required. The obvious way to avoid the problem is to measure the gas flow rate after the solids have been removed. This method is used by at least two other systems, one which uses a cyclone train (11), the second employing a combination of impactors and filters (10). Both of these systems are very complicated and were beyond the scope of current project to reproduce.

Several non-intrusive particulate measurement systems have been developed (eg. 12,13) and various claims are made for the devices. Sub-micron measurement is limited by the wavelength of the laser light whilst multiple scattering limits the maximum concentration, $0 \left(10^3\right) - 0\left(10^4\right)$ particles/cc. The Spectron Development Laboratories Inc. unit (13) uses an Argon Ion laser source which has a wavelength of 0.488 micron. This analyser uses scattered light-intensity for sizing particles between 0.5 and 10 micron and interferometry for sizing particles in the range 2 - 100 micron. No comparison is available in the reference with other measurement techniques. Another system, developed by the Leeds and Northrup Co., based on a Helium-Cadmium laser, may be used to assess particulates from submission to 20 um in diameter (12). The report included comparison with two other measurement techniques, viz. Anderson Impactor and Gelmain membrane filters. Reasonable agreement is claimed between the laser derived
particle sizes and the impactor derived sizes, but agreement between measured dust loadings was not achieved. This lack of agreement was attributed to the sampling methods used by the impactor and filter techniques in which the total loading in the exhaust duct flow is derived from samples taken at one or a number of diameters across the flow. Assuming that circumferential variations are small, the derived loadings depend upon the accuracy of the data reduction alone, however there is no justification for the assumption of small variations.

A system similar to the L and N Co. unit is commercially available in the U.K. from Malvern Instruments Ltd., Malvern, Worcs, (18). Initial development of this unit was carried out at Sheffield University (19,20). Unfortunately, the high cost of this system ( £12,000) prevented it being employed in the current project.

From the foregoing limited comments it can be seen that there is no definitive method for the measurement of particulates size, size range and concentration in a gas flow. However, well known techniques (cyclones, impactors and filters) are being refined and new techniques that do not suffer the limitations of the former are being developed. The problems of particulate measurement should soon be overcome, it will then be for the various interested parties-regulatory bodies and equipment manufactures - to decide acceptable particulate sizes and loadings.

C.2.8 Other Constituents

The principal remaining gaseous constituents of the exhaust flow are H₂, N₂ and H₂O. The amount of the N₂ in the exhaust gases is
either assumed to be equal to that of the incoming air to the combustion chamber or determined by a nitrogen balance from the measured NOx concentration and the (known) concentration of fuel bound nitrogen.

The hydrogen concentration should be very small, perhaps only a few ppm, because most of the fuel bound hydrogen will either form H2O or hydrocarbons. Measurement of H2 concentration was, therefore, not considered.

Measurement of the water vapour present in the exhaust gases is extremely difficult (5). Further, the hydrometers available had intermittent operation capability only and thus of little use where fluctuating concentrations might be expected. Also, it would appear that the presence of sulphur compounds (SO2, SO3, H2S) causes errors in the readings obtained due to the modification of dew-point temperature (5). Measurements of water vapour concentration was not attempted and for this reason alone wet analysis capability of all the operating analysers was essential.

C.2.9 New Techniques

Although there are many gas-specific analysers on the market there is no single comprehensive system available for the continuous total analysis of an exhaust stream. One technique that is being developed by both academic establishments and industry is that of Laser Raman Spectroscopy (21). Here, a pulse of laser light is directed into the flow under observation. The molecules in the flow absorb and then radiate the energy in specific frequency bands which is then collected by suitable apparatus for spectroscopic analysis. The technique has
been shown to be capable of identifying a useful range of chemical species via broad band analysis as well as the species temperature via narrow band analysis. It is not difficult to foresee a laser based analysis systems that would provide much of the information (chemical and quantitative analysis and physical conditions) that might be required of a process stream. This very powerful tool could be applied to process control as well as providing a very comprehensive research analysis facility.
References

2. MONROE E.S., Combustion, September, 1972.
9. WRIGHT S.J., 3rd Int. Conf. on Fluidised Bed Combustion.
16. BS3405 1961 (Revised 1971).

18. Leaflet ref. 792DST. Malvern Instruments, Spring Lane, Malvern, Worcs.


APPENDIX D

COMBUSTION RIG REQUIREMENTS

D.1 Gas Analysis System Requirements

The way in which gas samples are withdrawn, conditioned and transferred to the analysis equipment has a very significant effect on the accuracy of the analysis. All too often, reported results must be treated with caution due to poor probe design or inadequate transfer facilities.

When designing gas sampling probes and transfer lines several factors must be considered:-

1. The expected condition of the gas stream from which the sample is to drawn:– are temperatures and pressures high or low?

2. Do conditions of temperature or pressure occur at which the expected constituents begin to degrade, change phase, react with other constituents or the probe/transfer line materials?

3. Does the presence of any given constituent affect the operating characteristics of analysers specific to other constituents?

4. Are there specified or suggested procedures or equipment applicable to the process in question?
5. Are the gas samples to be taken from a specified position in the gas stream?

6. Is the analysis required to be intermittent or continuous?

Failure to consider any of these questions fully can result in invalid analysis results.

For example, it has been found that sampling gas streams for particulate loadings is subject to uncertainty due to the detrimental effects of the particulates on the characteristics of some of the measurement systems proposed. (2).

In the proposed combustion test programme the following considerations were appropriate:-

1. The probe would have to withstand high temperatures and pressures. It would need, therefore, to be robust and, probably, water cooled.

2. Exhaust Stream Constituents - The gas stream to be sampled was expected to contain $N_2$, $O_2$, $CO_2$, $H_2O$, $CO$, NOx, SOx and UHC along with traces of alkali metals and halogen compounds.

a. NOx, SOx, alkali metals and halogens are all reactive with the common metals, aluminium, copper and iron and NO is known to be reactive even with some "inert" polymers such as nylon.
b. The effects of temperature on the sampled constituents is significant. At temperatures below 150°C, the hydrocarbons, in particular the larger molecules, will start to condense on the probe and transfer tube walls. At similar temperatures the \( \text{SO}_2 \) and \( \text{SO}_3 \) will start to react with the \( \text{H}_2\text{O} \) to form sulphurous acid (\( \text{H}_2\text{SO}_3 \)) and sulphuric acid (\( \text{H}_2\text{SO}_4 \)) and these, too, will start to condense on the probe and transfer tube walls and in the analysis equipment. The temperature at which this condensation occurs is called the "acid dew point" and, at atmospheric pressures, is approximately 150°C (1). The need to avoid these condensation conditions is obvious.

At temperatures above about 300°C cracking and oxidation of the hydrocarbons can occur and further oxidation of the CO and NO might also occur. Even though the oxidation rates will be low at this level of temperature, sufficient reaction may occur that the accuracy of the analysis could be affected (1).

The range of temperature in which the sampled gases should be during the sampling transfer and analysis stages may therefore be set as 150°C - 300°C.

3. It has already been noted that the presence of particulates modified the performance of some types of apparatus intended to measure dust loadings. In dealing with gaseous components care must be taken because some analysers react to components other than the one the analyser is specifically designed to detect.

The technique used for measuring concentrations of CO and
CO₂ is based on infra-red absorption. Some chemical species absorb infra-red radiation at wavelengths similar to those absorbed by CO and CO₂. These include water vapour and some hydrocarbons. If the analysis instruments used lack sensitivity and resolution it is necessary to remove the H₂O and UHC before analysis is performed. This practice is forbidden by the EPA for some registration purposes unless accurate corrections can be made (3) and is not recommended in any event as the gas concentrations are then determined at different conditions to those pertaining to the other analysis points.

The technique used for measuring the concentration of UHC is Flame Ionisation Detection. This method can be affected by the presence of oxygen but the effect can be reduced or eliminated by the design of the analyser.

4. To the author's knowledge, there are no specified sampling or measurement techniques applicable to exhaust gas analysis in experimental projects at the time of writing. There are, however, regulations in the United States covering the emissions from "New Sources" (4). The emission levels quoted in these regulations were used for guidance in this project and it was logical to use the sampling and measurement techniques suggested by the governing body, the EPA. The main requirements can be summarised as follows (3):

1. Probe and transfer lines must be made from and lined with, where appropriate, inert materials.
2. The probe design must be such that about 80% of the principal pressure loss must occur at the probe inlet.

3. The gas sampling rates and distance from sampling point to analysis instruments must be such that gas residence time is less than 2 seconds.

4. The sample temperature must be reduced rapidly to approx. 150°C within the gas sampling probe and maintained at this level through the transfer lines.

5. The gas analysis should be conducted under "wet" conditions i.e. no dessicants, separators or drying columns may be used.

6. The gas analysis equipment should be set-up as shown previously in Fig. 5.1.

These recommendations were followed as closely as practicable. Sample probe designs and sampling techniques were used that were found satisfactory in previous combustion work (5) carried out by the author.

D.2 Exhaust Gas Cleaning

Removal of solids having sizes greater than about 25 microns could be achieved with very high efficiencies, better than 99%, by a high efficiency cyclone (8). However, the effective operating range of a cyclone is only about 4:1, beyond which performance deteriorates
further, the removal effectiveness reported may be inadequate for the following reasons:-

1. The rig installation was sited in a test cell beneath the main University building, and used an exhaust chimney designed for the safe operation of an industrial gas turbine, fuelled with gas-oil, which produces particulate and acid free exhaust. The exhaust chimney was lined with a soft mortar-like lining capable of withstanding temperatures of about 350°C; temperatures higher than this had to be avoided whilst minimising the risk of acid deposition on the chimney walls.

2. It was feared that if substantial quantities of particulates 20 microns were vented into the chimney a fire or explosion hazard might result.

Taking the above comments into account two gas cleaning methods were considered, viz. high efficiency cyclones and a venturi scrubber. Advantages and disadvantages of the two methods are given in Table D.1. Other methods such as grit arrestors; barrier filters; bag filters; packed, moving and fluidised bed filters were considered but discounted as being either inappropriate or unlikely to operate with guaranteeable characteristics. Only commercial gas cleaning equipment was considered appropriate for this application since safety was such a high priority.

The high efficiency cyclone was discounted due to the relatively limited operating range, typically 4:1 and thus a venturi scrubber was selected. The chosen unit was manufactured by Esmil Ltd., St. Neots, Cambridge. This unit had a rated of efficiency of 95% for particles of
D.3 **Scrubbing Water Supply and Treatment Plant**

The effective and safe operation of a venturi scrubber depends upon a continuous supply of clean water to the quench sections and to the venturi scrubber. Figure D.1 shows a flow diagram of the water plant. The water for quenching of the exhaust gas and eventual scrubbing was supplied from the University sump system using the surplus capacity of the Howden Compressor cooling water system.

The water so supplied, approximately 15g.p.m at about 20 psig, was then pressurised to 120 psig using a Worthington Simpson 5MV4 multi line centrifugal pump and split in to the three flows indicated in Fig. D.1. Control of the three flow rates was effected via calibrated nozzles supplied with the scrubber.

After separation from the exhaust gases, the spent water was ducted via a Spirax Sparco 907 steam trap, which effected pressure reduction, to a surge tank of about 1350 gallons capacity. It was expected that the larger particles captured by the scrubbing water would settle out in this tank. In order to capture particles escaping from the surge tank in the water flow return to the sumps a multi element filter supplied by Doulton Industrial Products was incorporated in the return line. This filter would capture all particles greater than 15 um with clean filter elements.

Water flow, from the settling tank, through the filter and to the sumps was achieved using a Worthington and Simpson centrifugal pump.
Model (2DM2) which was linked to a float switch in the surge tank in order to maintain a constant water level therein.

As the quench chamber and separator, see Fig. 5.3 had been designed to operate at 6 bar @ 600°C it was necessary to include a combustor shutdown system in the control network in order to minimise the time during which the above mentioned sections would be exposed to excessive temperature in the event of water flow failure. The emergency shutdown circuit is shown in Fig. D.2. This shutdown system was operated by a flow switch (a) in the outlet line from the hp pump. The normally open contacts of the flow switch were utilised to maintain a power supply to a three-pole change over relay. The three circuits controlled by this relay were:

a) power supply to the coal feed vibrator
b) fuel gas isolation solenoid valve
c) emergency mains water supply solenoid valve and warning lamp.

The third item was not, in fact, fitted due to the lack of a suitable solenoid valve. This, however, did not jeopardise the operation of the rig since a manually operated cooling water supply was available and circuit (b) was already fitted with a warning lamp.

During the limited period in which the water plant was in operation, three combustion runs, no problems were encountered.

D.4 Exhaust Silencing

In order to assess the potential noise problem, a 10mm plain orifice
was attached to the end of a pipe and compressed air source. Measurements of sound pressure level were made up to a distance of 1 metre on a hemisphere centred on the orifice. A BRUEL and KJAER type 2209 noise meter was used to measure the sound pressure level (SPL) in dB. No weighting filters were employed. The results of venting compressed air to atmosphere from an upstream pressure of 30lb/sq in are shown in Table D.2.

The values of sound pressure level shown in Table D.2 are those recorded at 8kHz as these appeared to be the highest in the range 2kHz - 32 kHz. The generated noise was of a broad band type with a low peak somewhere between 8 and 16 kHz. Figure D.3 shows the distribution of sound pressure level around the exhaust jet. A correlation derived from graphical data in "Woods Practical Guide to Noise Control, (8) shows that:

\[(SPL)_{100} = 59.62 \log_{10} \left( \frac{U}{100} \right) + 100 \quad \text{D.1}\]

Substituting the appropriate values for the simple test rig, the correlation predicts that, for a 100mm diameter orifice, the choked sound pressure level should be approximately 132 dB. The correlation may be extended to other diameter orifices so that:

\[(SPL)_{\varnothing} = 59.62 \log_{10} \left( \frac{U}{100} \right) + K\varnothing \quad \text{D.2}\]
from ref (8)

<table>
<thead>
<tr>
<th>ø</th>
<th>400</th>
<th>200</th>
<th>100</th>
<th>50</th>
<th>26</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kø</td>
<td>112</td>
<td>105</td>
<td>100</td>
<td>94</td>
<td>88</td>
</tr>
</tbody>
</table>

Substituting appropriate values into this adjusted correlation for a 50mm orifice, operating choked at 1200°K the estimated sound pressure level was 143 dB (at 1 metre).

The estimated SPL is not unreasonable when compared with those associated with gas turbine exhausts which may be of the order of 150 dB.

The need for an efficient silencer, even for cold, pressurised operation, was obvious. The upper limit on SPL at the exhaust termination-flange was set at 100 dB as the exhaust chimney is fixed to the wall of the main University building!

After extensive consultation with industry it became evident that commercial expansion silencers would probably not be able to achieve the desired noise attenuation. Furthermore ejector-detuners were found to be beyond the budget of the project. With much help from Cullum Detuners Ltd., Derby, the author was able to design an ejector-detuner and this was manufactured in the University. A general layout of the detuner appears in Appendix B. For commercial reasons, details of the internal design of the detuner cannot be revealed. For further information, the reader is referred to Cullum Detuners Ltd.

The ejector type of detuner silencer entrains air to cool and decelerate a jet flow. This feature was an advantage in the early stages of the combustion trials which were carried out without the gas
scrubber installed. During these trials the temperature of the exhaust products was considerably higher than the temperature limit set on the main exhaust chimney (350°C) and cooling of the exhaust gases was vital. For this reason the detuner was installed before any combustion work was undertaken. The detuner appeared to operate satisfactorily through the experimental test programme although no measurements of noise attenuation were made.

D.5 Solids Feed System - Valve Selection and Sequence Control

Ball valves are used in solids handling equipment (9) as they have the advantage over linear action valves of being self-cleaning; the action of the valve being such that the sealing surfaces are wiped clean by the orifice edges within the valve body, Fig D.4(a). This feature helps to prolong the life of the valve unit.

Automation of both linear action valves and ball valves can be achieved using electric or pneumatic actuators. The latter tend to be cheaper and produce very large forces whilst the former are more convenient for inclusion in a control system.

Pneumatic in-line valves, Fig. D.4 (b), seal via a pinch effect. They are very fast acting but material trapped by the flexible liner will reduce the effectiveness of the pressure seal and also may cause rapid wear of the flexible liner.

Rotary valves Fig. D.4 (c) offer the prospect of metering and pressure transfer in one unit but available valves were limited in operating pressure difference.
Other, less conventional, means of transferring solids from low to high pressure zones were investigated. Plastic extension has been studied by Ryason and England (10). Here, bituminous coal was heated to 350 - 400°C at which it became plastic and was then extruded using a conventional screw press. The resulting 'sausage' of coal could easily be fed through a suitable tube into a conventional PFBC and it may be possible to do likewise with a PRFBC. Metering of the fuel flow rate is effected by control of the screw press speed. A centrifugal feeding device has been investigated by Bonin et al (11) and modelled by Meyer (12). Fig. D.5 shows a simplified representation of the device. The solids are fed into a carousel of vertical tubes (A) which are mounted in a rotating housing (B). The solids drop down the tubes under the action of gravity and are forced out of the orifices (c) against the pressure difference by the action of centrifugal forces. Meyer (12) has shown that the design of the passage of the exit orifice is critical to the effective operation of the device.

D.5.1 Selected Coal Feeder

Plastic extrusion and centrifugal feeding both offer the means of achieving transport of solids against very large pressure gradients but both need much development before reliable units can be produced.

The choice of transfer means then reduces to a number of conventional or rotary valves in series. The former option was considered to be more reliable and was considerably cheaper. Ball valves were chosen because of their self-cleaning properties and very low price (about £5 per valve). Pneumatic actuation was found to be cheaper than electric actuators and provided, easily, the large forces required to move the
rather stiff ball valves. Sequencing of valve operation was controlled by a solid state clock circuit. This circuit is described in the following section.

It was decided that metering of the fuel flow should be done on the pressurised side of the locking valve to prevent intermittent interruption of the fuel flow. A vibratory feed was chosen for its simplicity and reliability. The final arrangement of the fuel feeding system is shown in Fig. D.6.

Transfer of the fuel from the metering station to the combustion zone was studied as part of the atmospheric pressure test programme is discussed in chapter 6.

D.5.2 **Airlock Valve Sequence Control**

Refering to Fig. D.6, the valve sequence control is activated when the level of fuel in hopper B falls below a specified height 'h'. The sequence of operation is then as follows-

1. Open valve A to charge the air lock with fresh fuel.
2. Close valve A.
3. Open valve B to tranfer the fuel to hopper B.
4. Close valve B.
5. Repeat actions (1) - (4) until the fuel level in hopper B
attains, at least, the specified height 'h'.

Valve actuation was achieved using "Enots" double-acting pneumatic cylinders (no. 605250000080) which were switched using "Enots" solenoid valves (no. 03601100000). Using shop airline pressure of ca. 80 lb/sq.in., actuation was positive and rapid.

Fuel level sensing in the pressurised hopper was achieved using a light beam and photo sensitive resistor (P.S.R.). Two perspex windows, of sufficient thickness to retain the operating pressure, were mounted on a diameter, Fig. D.6. Whilst the level of coal in the hopper was above the window level, no light reached the photosensitive resistor and the resistance maintains at a high value. When the fuel level fell below that of the windows, the increased light intensity falling on the p.s.r. reduced the resistance and this fall in resistance activated the first stage of the main control circuit.

A block diagram of the main control circuit is shown on Fig. D.7 and the circuit in Fig. D.8. The circuit may be divided into five sections:

i. The light operated switch

ii. An oscillator to provide clocking pulses

iii. A logic control

iv. A sequential timer

v. Output relays
D.5.2.1 Sequential Timer

The heart of the valve control circuit is a 556 dual timer Integrated Circuit made up from two precision bi-polar timers. Sequential timing is effected by connecting the output of one timer, via a differentiating circuit, to the input of the second timer. Thus when the input of timer T1 receives a negative trigger pulse its output goes high for a period determined by C10, R11 and PR2 in Fig. D8. At the end of this period the output from T1 falls to zero. This change of state is differentiated to produce a negative pulse which is fed to the input of T2 which then starts its timing period, the length of which is determined by C11, R12, and PR3. In order to prevent both outputs becoming high simultaneously due to continuous triggering of T1, the output of timer T2 is fed back to a control gate with prevents clock pulses reaching T1 when T2 is high. The time-relative diagram for the whole circuit is shown in Fig. D.9. Note that both timers T1 and T2 respond only to the first negative trigger pulse; subsequent pulses positive and negative are blocked internally until the full timing sequence is completed. The outputs from the sequential timer are fed, via two transistor switches TR4 and TR5, to two 12v relays which activate the solenoid valves.

D.5.2.2 Light Operated Switch

The operation of the photosensitive resistor, R2, has already been explained. When the resistance of the p.s.r falls the base bias of TR1 is increased. This changes the input to gate A from 1 to 0 allowing pulses from the oscillator to be fed to the logic control and then to the sequential timer via inverting transistor TR2.
D.5.2.3 Oscillator Circuit

This is a standard circuit utilising two NOR gates, in this case C and D, of a 4001 CMOS NOR chip. The clocking frequency is determined by the values of C13, R17 and PRA, the latter being used for adjustment of the time interval.

D.5.2.4 Logic Control Circuit

This circuit acts as an interface between the switching and clocking sections and the timing section and utilises the fourth gate in the 4001 NOR chip. The purpose of the logic control is to prevent trigger pulses being fed to the input of timer T1, which would cause the input of T1 to go high, when the output of timer T2 is high. To do this the output of T2 is fed back to the one input of the NOR gate, the other input being the output of the oscillator circuit. With a logic 1 on the T2 output the gate is disabled and no oscillator clocking pulses are fed to the timers. With a logic 0 on the T2 output the gate is enabled and clocking pulses are fed to the differentiating circuit at the input of T1 via transistor TR3. The latter ensures that clear, well defined square pulses are fed to the differentiating circuit (C7 and R10). Without these conditioned square pulses triggering of the timer became erratic. Differentiation of the square wave produced a 'spiked' input to the timer – see time - relative diagram Fig. D.9. The first negative spike triggers the sequence when its value is greater than 1/3 VCC.
References

5. BROCK N. Rolls Royce Ltd. Filton, Bristol, Private Communication.
9. MACAUBER DENSEVEYDOR, MACAUBER ENGINEERING LTD., Oyden Road, Doncaster, S. Yorkshire.
<table>
<thead>
<tr>
<th>METHOD</th>
<th>ADVANTAGES</th>
<th>DISADVANTAGES</th>
</tr>
</thead>
<tbody>
<tr>
<td>High Efficiency</td>
<td>i) Representative of real installation.</td>
<td>i) Removal efficiency falls with increasing temp.</td>
</tr>
<tr>
<td>Cyclone</td>
<td>ii) No operator action required.</td>
<td></td>
</tr>
<tr>
<td></td>
<td>iii) No ancillary equipment necessary.</td>
<td>ii) High pressure high temperature unit expensive.</td>
</tr>
<tr>
<td></td>
<td>iv) Dry products</td>
<td></td>
</tr>
<tr>
<td></td>
<td>v) Small floor area required.</td>
<td>iii) Does not remove acid constituents of gas stream.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>iv) Modest operating range.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>v) High gas stream outlet temperature</td>
</tr>
<tr>
<td>METHOD</td>
<td>ADVANTAGES</td>
<td>DISADVANTAGES</td>
</tr>
<tr>
<td>--------------</td>
<td>-----------------------------------------------------------------------------</td>
<td>---------------------------------------------------</td>
</tr>
<tr>
<td>Venturi Scrubber</td>
<td>i) Removal efficiency good at all temperatures.</td>
<td>i) Not representative of real installation.</td>
</tr>
<tr>
<td></td>
<td>ii) Removes some acid constituents of gas stream.</td>
<td>ii) Ancilliary equip. required-water feed, pumps, filters, tanks.</td>
</tr>
<tr>
<td></td>
<td>iii) Low pressure loss.</td>
<td>iii) Operator action required.</td>
</tr>
<tr>
<td></td>
<td>iv) Low gas stream outlet temperature.</td>
<td>iv) Inlet products.</td>
</tr>
<tr>
<td></td>
<td>v) Good operating range.</td>
<td>v) Large floor area required.</td>
</tr>
<tr>
<td></td>
<td>vi) Modest price (includes ancilliary equipment).</td>
<td></td>
</tr>
</tbody>
</table>

434.
<table>
<thead>
<tr>
<th>ANGLE FROM JET AXIS</th>
<th>DISTANCE FROM ORIFICE CENTRE (m)</th>
<th>PEAK RECORDED SPL</th>
</tr>
</thead>
<tbody>
<tr>
<td>0°</td>
<td>.05</td>
<td>135</td>
</tr>
<tr>
<td>0°</td>
<td>0.5</td>
<td>132</td>
</tr>
<tr>
<td>0°</td>
<td>1.0</td>
<td>127.5</td>
</tr>
<tr>
<td>45°</td>
<td>0.5</td>
<td>120</td>
</tr>
<tr>
<td>45°</td>
<td>1.0</td>
<td>112</td>
</tr>
<tr>
<td>90°</td>
<td>0.5</td>
<td>124</td>
</tr>
<tr>
<td>90°</td>
<td>1.0</td>
<td>109</td>
</tr>
</tbody>
</table>
DIRTY EXHAUST GAS
@ 1000°C FROM
RFB COMBUSTOR

TO MAIN
AIR
COMPRESSOR

FLOW SWITCH

1st STAGE
QUENCH
10 gpm

SEPARATION

2nd STAGE
QUENCH
3 gpm

VENTURI
SCRUBBING 2 gpm

FILTER

SURGE TANK

UNIVERSITY SUMP

CUT-IN

EFFLUENT
STEAM TRAP

FLOAT SWITCH

SHUTDOWN CIRCUITS

CLEAN EXHAUST
@ 100°C TO
SILENCER

FIG D.1 SCRUBBING WATER FLOW SHEET
FLOW SWITCH

MULTI-CONTACT RELAY

CF
FGS
ECWS

KEY
CF  COAL FEEDER
FGS  FUEL GAS SOLENOID VALVE
ECWS  EMERGENCY COOLING WATER SOLENOID VALVE
L  WARNING LAMP
NC  NORMALLY CLOSED CONTACT
NO  NORMALLY OPEN CONTACT

FIG D.2 EMERGENCY SHUTDOWN CIRCUITS
UPSTREAM AIR PRESSURE = 15 lbf/psig
UPSTREAM AIR TEMP. = 15°C

FIG D.3 SOUND PRESSURE LEVEL DISTRIBUTION
AROUND A 10mm PLAIN ORIFICE (8kHz band)
FIG D.4 SOME VALVE TYPES SUITABLE FOR SOLIDS FEEDING SYSTEMS
FIG D.5  CENTRIFUGAL SOLIDS FEEDER
FIG D.7 COAL FEEDER CONTROL CIRCUIT BLOCK DIAGRAM
### CIRCUIT KEY

#### TRANSISTORS

<table>
<thead>
<tr>
<th>TR1</th>
<th>2N3053</th>
</tr>
</thead>
<tbody>
<tr>
<td>TR2</td>
<td>2N3053</td>
</tr>
<tr>
<td>TR3</td>
<td>BC107</td>
</tr>
<tr>
<td>TR4</td>
<td>2N3053</td>
</tr>
<tr>
<td>TR5</td>
<td>2N3053</td>
</tr>
</tbody>
</table>

#### RESISTORS

<table>
<thead>
<tr>
<th>R1</th>
<th>2700</th>
</tr>
</thead>
<tbody>
<tr>
<td>R2</td>
<td>0R12</td>
</tr>
<tr>
<td>R3</td>
<td>150</td>
</tr>
<tr>
<td>R4</td>
<td>330</td>
</tr>
<tr>
<td>R5</td>
<td>1.5 K</td>
</tr>
<tr>
<td>R6</td>
<td>330</td>
</tr>
<tr>
<td>R7</td>
<td>100 K</td>
</tr>
<tr>
<td>R8</td>
<td>22 K</td>
</tr>
<tr>
<td>R9</td>
<td>68 K</td>
</tr>
<tr>
<td>R10</td>
<td>10 K</td>
</tr>
<tr>
<td>R11</td>
<td>22 M</td>
</tr>
<tr>
<td>R12</td>
<td>22 M</td>
</tr>
<tr>
<td>R13</td>
<td>10 K</td>
</tr>
<tr>
<td>R14</td>
<td>1.5 K</td>
</tr>
<tr>
<td>R15</td>
<td>1.5 K</td>
</tr>
<tr>
<td>R16</td>
<td>100 K</td>
</tr>
<tr>
<td>R17</td>
<td>100 K</td>
</tr>
</tbody>
</table>

#### CAPACITORS

<table>
<thead>
<tr>
<th>C1</th>
<th>1000 pf</th>
</tr>
</thead>
<tbody>
<tr>
<td>C2</td>
<td>100 pf</td>
</tr>
<tr>
<td>C3</td>
<td>100 pf</td>
</tr>
<tr>
<td>C4</td>
<td>1000 pf</td>
</tr>
<tr>
<td>C5</td>
<td>10,000 pf</td>
</tr>
<tr>
<td>C6</td>
<td>10,000 pf</td>
</tr>
<tr>
<td>C7</td>
<td>0.1 f</td>
</tr>
<tr>
<td>C8</td>
<td>0.1 f</td>
</tr>
<tr>
<td>C9</td>
<td>1000 pf</td>
</tr>
<tr>
<td>C10</td>
<td>1000 pf</td>
</tr>
</tbody>
</table>

#### POTENTIOMETERS

<table>
<thead>
<tr>
<th>PR1</th>
<th>4.7 K</th>
</tr>
</thead>
<tbody>
<tr>
<td>PR2</td>
<td>4.7 K</td>
</tr>
<tr>
<td>PR3</td>
<td>220 K</td>
</tr>
<tr>
<td>PR4</td>
<td>22 K</td>
</tr>
</tbody>
</table>

#### DIODES

<table>
<thead>
<tr>
<th>D21</th>
<th>BZX85 5V</th>
</tr>
</thead>
<tbody>
<tr>
<td>D1</td>
<td>IN4148</td>
</tr>
<tr>
<td>D2</td>
<td>IN4148</td>
</tr>
</tbody>
</table>
GATES
G1, 2, 3, 4 NOR 4001

RELAYS
REl & RE2 -D.P.C.O

TIMER
T1 556 dual timer
a) CLOCK PULSES

b) LOGIC OUTPUT FROM LIGHT SWITCH

c) LOGIC FEEDBACK FROM TIMER 2

d) DIFFERENTIATED TRIGGER PULSES TO TIMER 1

e) SEQUENTIAL TIMER OUTPUTS

FIG D.9 CONTROL CIRCUIT TIME-RELATIVE DIAGRAM
APPENDIX E

CHARACTERISATION OF PRESSURE LOSS

The characteristics of pressure loss in terms of operating parameters can be derived using dimensional analysis. The pressure loss of any flow system will be determined by the geometry of the system and the flow conditions. Thus the pressure loss, \( \Delta P \), will be some function of the flow area, \( A \), the gas mass flow rate, \( m \), and the gas conditions which can be described by any two of \( P, \rho \) or \( T \). To avoid the problem of deciding dimension for \( T \) we shall make use of \( P \) and \( \rho \).

Thus \[ \Delta P = f(m, A, P, \rho) \] \hspace{1cm} \text{(E1)}

This may be written as:

\[ \Delta P = (m^a \times A^b \times P^c \times \rho^d) \] \hspace{1cm} \text{(E2)}

by expressing each factor in terms of its dimensional equivalent, composed of mass, \( M \), length, \( L \), and time, \( T \), we find that:

\[ M L T^{-2} = \sum \text{constants} \left( (M T^{-1})^a (L^2)^b (M L^{-1} T^{-2})^c (M L^{-3})^d \right) \]

Dimensional homogeneity demands that:

for \( M \):
\[ l = a + c + d \]

for \( L \):
\[ -1 = 2b - c - 3d \]
and \( T = -2 = -a - 2c \)

thus

\[
a' = 2 - 2c = 2(1-c)
\]

\[
d = 1 - a - c
\]

\[
= 1 - (2-2c) - c
\]

\[
= -1 + c
\]

\[
d = -(1-c)
\]

\[
2b = -1 + c + 3d
\]

\[
= -1 + c - 3 + 3c
\]

\[
= -4 + 4c
\]

\[
b = -2 + 2c = -2(1-c)
\]

Thus

\[
\Delta P = \Sigma \text{constants} \ (m \ 2(1-c) \ A \ -2(1-c) \ p_c \ s^{-(1-c)}) \ E.3
\]

The "Pi Theorem" allows the parameters to be grouped together according to the relationship;

\[
n_v - n_d = n_g
\]

where \( n_v = \) number of independent variables

\( n_d = \) number of dimensions

and \( n_g = \) number of groups

448.
Now, \( n_v = 5 \) and \( n_d = 3 \), thus \( n_g = 2 \)

Division by \( P \) makes the LHS of E5 dimensionless giving

\[
\Delta P = \sum \text{constants} \left( m^{2(1-c)} A^{-2(1-c)} P^{-(1-c)} g^-(1-c) \right)
\]

or

\[
\Delta P = \sum \text{constants} \left( m \frac{\sqrt{T}}{P} \right)^{2(1-c)}
\]

writing \( g = \frac{P}{RT} \)

and since \( R \) = constant for a given gas

\[
\Delta P = \sum \text{constant} \left( \frac{m}{\sqrt{T}} \right)^{2(1-c)}
\]

E4

\[
\frac{AP}{P} \quad \left( \begin{array}{c} A \ P \end{array} \right)
\]

The group \( \frac{m}{\sqrt{T}} \) is the gas dynamic flow function, \( Q \), which

\[
\frac{AP}{P}
\]

appears in relationships which describe gas flows and can be used to characterise the performance of a variety of flow machines. In the current work the performance of a combustion device is described. \( Q \) may also be used to describe the performance of compressors and turbines and it is, therefore, a powerful tool in gas turbine design.

Now, in order to describe the pressure loss ratio, \( \frac{\Delta P}{P} \),
completely, in terms of \( Q \), would require the use of all the terms in expression E4. However, experience shows that, provided the pressure loss ratio is not too large, \( \Delta P < 15\% \), we can write:

\[
\Delta P = k \left( m \sqrt{T} \right)^n
\]

\[
P \quad \left( A \quad P \right)
\]

The magnitude of \( n \) is, typically, 2 but varies from system to system.

For larger pressure loss ratios it is necessary to employ more terms of expression E4.
APPENDIX F

DEPENDENCE OF Q_p ON A

The way that Q_p responds to a change in the exhaust port area, A, can be shown qualitatively by a consideration of the gas flow conditions. The only assumption that must be made is that the pressure loss across a given bed, fluidised at a gas velocity, U_b, will not be affected by the change in exhaust port area and will be equal to ΔP_b.

Consider an RFB where the exhaust port area can be A or A', where A < A', and with a fixed distributor area A_d.

At gas velocity, U_b the gas volume flow rate is given by

\[ V_g = A_d \times U_b \]  \hspace{1cm} \text{Fl}

In order that this same volume flow rate should pass through both A and A' the corresponding combustor static pressures p_c and p'_c must satisfy

\[ p_c > p'_c \]

As a consequence, the gas density, \( \rho \), satisfies

\[ \rho_g > \rho'_g \quad (\text{from} \quad \frac{p}{RT}) \]  \hspace{1cm} \text{F.2}

451.
and thus the gas mass flow rate, \( m \), satisfies,

\[
m > m' \quad \text{(from } m = \int g \, \text{AU)}
\]

Applying Bernoulli's Equation to the combustor pressure:

\[
P_c + \frac{1}{2} \int g \, U_b^2 = P_c
\]

where \( P_c \) = combustor total pressure with exhaust area \( A \).

and

\[
p'_c + \frac{1}{2} \int' g' \, U_b^2 = P'_c
\]

where \( P'_c \) = combustor total pressure with exhaust area \( A' \).

now

\[
P_c > P'_c
\]

and

\[
\int g > \int' g'
\]

thus

\[
P_c + \frac{1}{2} \int g \, U_g^2 > P'_c + \frac{1}{2} \int' g' \, U_b^2
\]

ie.

\[
P_c > P'_c
\]

Now if \( P_2 \) and \( P'_2 \) and the corresponding total pressures just downstream of the distributor

then

\[
P_2 = P_c + \Delta P_b
\]
and \[ P'_{2} = P'_{c} + \Delta P'_{b} \] \hspace{1cm} F.8

if \[ \Delta P_{b} = \Delta P'_{b} \] (as assumed)

then \[ P_{2} > P'_{2} \] \hspace{1cm} F.9

It follows, for a given distributor, that

\[ P_{1} > P'_{1} \]

where \( P_{1} \) is the plenum total pressure for a given gas volume flow rate.

Now, it is shown in the literature (146) that:

\[ \frac{dP_{c}}{dA} > \frac{dP_{c}}{dA} \]

Thus a change in the exhaust port area produces a larger change in \( m \) (which is reflected by \( dpc/dA \)) than in \( P_{1} \) (which is reflected by \( dPc/dA \)) and thus by reducing the area we can expect to measure a larger \( Qp \) for a given \( U_{b} \) and vice versa.
APPENDIX G

Particle Trajectory computer program

This appendix contains the computer program used to predict particle trajectories in the freeboard. It was developed from a program written by M. Subzwarl of the Mechanical Engineering Laboratory at G.E.C. Power Engineering, Whetstone, Leics.

The theory on which the program is based is described in Chapter 8. The program uses the method of finite differences with time steps of $1 \times 10^{-5}$ seconds. Larger time steps were found to produce unstable results at the inner radii.

The program was written for use on the HP9845 series computers and includes a number of graphics routines which generated Figs. 8.1 to 8.19.
103 REM PARTICLE TRAJECTORY PROGRAMME-TRAJE
113 DIM P(2,1000),Q(2,1000),S(2,1000),W(2,1000)
115 DIM X(2,1000),Y(2,1000),A(2,1000),T(6),D(6)
115 DATA 10
117 READ G
123 DATA 600
121 DATA 10,25,50,100,250,500
124 READ D8
125 FOR I=1 TO 6
127 NEXT I
130 PRINT PAGE
133 Count=1
133 D9=D(Count)
163 R9=D9/(2*10^-6)
193 REM B1=GASDEN,B2=GASVIS,B3=SURFGASVEL,B4=GASMASSFLO,B5=PARTDEN
193 REM B6=GASVOLFLO,R0=BEDSURRED,W=ROTSPEED,H=INITBEDHEIGHT
203 REM U1=SUBVEL,U=RELPARTVEL,C1=PARTCD
213 REM U2,U3,U4=RAI,TANG,AXIALPARTICLE VEL'S
211 B1=1.177
212 B2=.0000189
213 B4=.1069
214 B5=2630
215 R0=.090
215 W=31.32
217 U1=.704
213 U2=.704
213 U3=.704
223 U4=.704
221 H=.08
222 A=0
223 R7=.06
224 T8=.7854
225 Z0=.04
225 F1=1.606
227 F2=U2
223 F3=U3
229 F4=U4
255 B6=B4/B1
257 F5=B6/(2*PI*R0*H)
261 U=(U2^2+U3^2+U4^2)^.5
503 DEF FNACX)=INT(X*10^-4+.5)*10^-4
501 F5=FNAC(F5)
533 PRINT "PARTICLE SIZE =";D9;"MICRONS"
532 PRINT
533 PRINT "NO. RADIUS ANGLE AXIAL POS. TIME"
563 PRINT
573 T=0
575 R=R0
575 Z1=Z0
583 L=0
581 M=0
603 T2=T8*100/PI
613 IMAGE DD.DDD
623 PRINT USING 610;L,R,T2,Z1,T
621 Px0=R*COS(T8)
622 Py0=R*SIN(T8)
623 $sum=0
533 FOR L=1 TO 1000
531 M=L
553 B=3*B1/(B*B5*R9)
563 N1=2*R9*U*B1/B2
573 IF N1<.1 THEN 690
583 IF N1<1 THEN 692
584 IF N1<10 THEN 694
585 IF N1<100 THEN 696
593 D2=18.0
591 GOTO 699
592 D2=.75*(22.73+.0903/N1+3.69*N1)
593 GOTO 699
594 D2=.75*(29.17-3.689/N1+1.222*N1)
595 GOTO 699
596 D2=.75*(46.5-116.7/N1+.6167*N1)
599 C1=4+D2/3/N1
703 IF C1<.44 THEN 702
701 GOTO 703
702 C1=.44
703 IF R>R7 THEN 730
704 REM CALC. OF NEW RAD. VEL.
705 S1=W^2*R
706 S2=U2-B6*R/(2*PI*R7^2*R)
707 S3=U3/R+100*B6^2/(PI^2*W*R7^4*R^2)-W
708 S4=B6/(2*PI*R7^2*R)
709 S6=B*C1*U*U2
713 Y2=S1+(2*W*R*R*S3)*S3+S2*S4-S6
711 REM CALC. OF NEW TANG. VEL.
712 Y3=S2*(U3/R-2*S3)-2*W*S2-B*C1*U*U3
713 Y4=-9.81+B6/(PI*R7^2*R)*(U4-B6*Z1/(PI*R7^2*R))-B*C1*U*U4
714 GOTO 845
733 REM CALC. OF NEW RADIAL VEL.
743 S1=W^2*R
753 S2=U2-B6/(2*PI*R*R)
763 S3=U3/R+W*R0^2/R^2-W
773 S4=B6/(2*PI*H*R*R)
783 S6=B*C1*U*U2
793 Y2=S1+(2*W*R*R*S3)*S3+S2*S4-S6
303 REM CALC. OF NEW TANG. VEL.
323 X2=B*C1*U*U3
333 Y3=S2*(U3/R-4*W*R0^2/R^2)-X2
343 Y4=-9.81-B*C1*U*U4
345 IF R>R7 THEN 848
345 T=.0010
347 GOTO 850
343 T=.000001
353 A2=Y2*T
363 U2=U2+A2
373 A3=Y3*T
383 U3=U3+A3
393 A4=Y4*T
403 U4=U4+A4
413 U=(U2^2+U3^2+U4^2)^.5
323 Y5=U2-B6/(2*PI*R*R)
333 A6=Y5*T
343 Y6=U3-R+W*R0^2/R^2
353 A7=Y6*T
363 T8=T8+A7
373 T2=T8*180/PI
383 R=R+A6
393 A8=U4*T
1030 Z1=Z1+A8
1010 U2=FNAC(U2)
1020 U3=FNAC(U3)
1030 U4=FNAC(U4)
1040 R=FNAC(R)
1050 T2=FNAC(T2)
1034 Tsum=Tsum+T
1035 X=R*COS(T8)
1036 Y=R*SIN(T8)
1130 ON Count GOTO 1210, 1220, 1230, 1260, 1270, 1280
1210 P^{(1, M)}=X
1211 P^{(2, M)}=Y
1212 P0=M
1213 T^{(1)}=Tsum
1214 GOTO 1300 456,
Q(1,M)=X
Q(2,M)=Y
Q0=M
T(2)=Tsum
GOTO 1300
S(1,M)=X
S(2,M)=Y
S0=M
T(3)=Tsum
GOTO 1300
W(1,M)=X
W(2,M)=Y
W0=M
T(4)=Tsum
GOTO 1300
X(1,M)=X
X(2,M)=Y
X0=M
T(5)=Tsum
GOTO 1300
Y(1,M)=X
Y(2,M)=Y
Y0=M
T(6)=Tsum
GOTO 1300
PRINT USING 610;L,R,T2,Z1,Tsum
IF R>R0 THEN 1320
IF R<R0 THEN 1320
IF Z<X0 THEN 1320
IF Z>Y0 THEN 1320
NEXT L
Count=Count+1
0=Count
IF Count<7 THEN 138
PRINT PAGE
PLOTTER IS 13,"GRAPHICS"
GRAPHICS
LOCATE 20,110,10,100
SCALE -R0,R0,-R0,R0
AXES R0/5,R0/5,0,0
MAT A=P
A0=P0
GOSUB Plot
MAT A=Q
A0=Q0
GOSUB Plot
MAT A=S
A0=S0
GOSUB Plot
MAT A=W
A0=W0
GOSUB Plot
MAT A=X
A0=X0
GOSUB Plot
MAT A=Y
A0=Y0
GOSUB Plot
GOTO 1544
Plot:LINE TYPE 1
PLOT Px0,Py0,-2
FOR L=1 TO R0-1
PLOT A(1,L),A(2,L),-1
NEXT L
RETURN
PENUP
MOVE R0,0
FOR A=0 TO 2*PI STEP PI/200
DRAW R0*COS(A),R0*SIN(A)
NEXT A
PENUP
MOVE R7, 0
FOR A=0 TO 2*PI STEP PI/200
DRAW R7*COS(A), R7*SIN(A)
NEXT A
PENUP
DUMP GRAPHICS
EXIT GRAPHICS
PRINTER IS 0
PRINT SPA(30), "APPLIED G =" ; G ; "Gravity"
PRINT SPA(30), "BED PARTICLE DP =" ; D8 ; "MICRONS"
PRINT SPA(30), "GAS DENSITY =" ; E1 ; "KG/M^3"
PRINT SPA(30), "GAS VISCOITY =" ; E2 ; "NS/M"
PRINT SPA(30), "SURFACE GAS VEL. =" ; F5 ; "M/S"
PRINT SPA(30), "GAS MASS FLOW =" ; F4 ; "KG/S"
PRINT SPA(30), "PARTICLE DENSITY =" ; B5 ; "KG/M^3"
PRINT SPA(30), "BED SURF. RADIUS =" ; R0 ; "M"
PRINT SPA(30), "EXHAUST RADIUS =" ; R7 ; "M"
PRINT SPA(30), "PART. RELEASE HT =" ; Z0 ; "M"
PRINT SPA(30), "ROTATIONAL SPEED =" ; W ; "RAD/S"
PRINT SPA(30), "BUBBLE VELOCITY =" ; U1 ; "M/S"
PRINT SPA(30), "MIN. FLUID'N VEL =" ; F1 ; "M/S"
PRINT SPA(30), "INIT. RADIAL VEL =" ; F2 ; "M/S"
PRINT SPA(30), "INIT. ANG. VEL =" ; F3 ; "M/S"
PRINT SPA(30), "INIT. AXIAL VEL =" ; F4 ; "M/S"
PRINT SPA(30), "STEP SIZE =" ; T
FOR I=1 TO 6
PRINT SPA(20), "TIME OF FLIGHT =" ; D(I) ; "MICRON =" ; T(I) ; "sec"
NEXT I
STOP
END
PRINCIPAL REFERENCES

1. KEILY J., C.M.E. May 1980
2. ELLIOTT D.E. and VIRR M.J., Proc. 3rd Int. Conf. on FBC, Hueston Woods, 1972
7. SANDERSON P.R. and HOWARD J.R., Applied Energy 3, 977
9. WINKLER F. DRP 437,970
12. BADISCHEN ANILIN-U-SODA FABRIK A.G., BP 768656, 1957
13. BADISCHEN ANILIN-U-SODA FABRIK A.G., BP 776791, 1957
14. UNION CARBIDE CORP., BP 956 659, 1963
15. COMBUSTION ENGINEERING INC., BP 784545, 1955
16. COMBUSTION ENGINEERING INC., BP 785 399, 1957
19. ANON, USERDA Report FE-1514-T-9
HOKE R.C. et al, Combustion, Jan., 1975
HATCH L.P., REGAN W.H. and POWEL J.R., Nucleonics 18, 12, Dec., 1960
SHAKESPEARE W.J., LEVY E.K., and CHEN J.C., Powder Technology, 24, 1979
CHALCHAL S., SHEENAN T.V. and STEINBERG M., Brookhaven National Laboratory, BNL 19308, Oct., 1974
LEVY E.K., DODGE C., and CHEN J., ASME Paper 76 HT 68
DEMIRCAN, N., Ph.D. Thesis, University of Sheffield, 1978

460.

ELLIOTT D.E., Third Int. Conf on Fluidized Bed Combustion, Hueston Woods, 1972

SCIENCE RESEARCH COUNCIL "Combustion Generated Pollution" July 1976


461.
60 ROBERTSSON H., Proc. 3rd Int. Conf. FBC, December 1975.
72 BUTLER P., The Engineer, 19 April, 1979.
73 ANON Electrical Times p 24, April, 1979.
74 FISHLOCK D. The Times 20 June, 1980.

463.


122 RICKMAN W.S., Proc. Sixth Int. Conf. on Fluidized Combustion, Atlanta, Georgia, April, 1980.


129 ABEEL W.T. et al, Proc. Sixth Int. Conf. on Fluidized Combustion, Atlanta, Georgia, April, 1980.


131 HENSCHEL D.B., Proc. Sixth Int. Conf. on Fluidized Combustion, Atlanta, Georgia, April, 1980.


PETERSEN V. et al, Proc. Sixth Int. Conf. on Fluidized Combustion, Atlanta, Georgia, April, 1980.


ELLIOTT D.E., BP 1471598.

Additional references

OSTKAM G.W. & HOWARD J.K. Some effects of geometrical factors on the performance of a rotating fluidized bed.

OSTKAM G.W. & HOWARD J.K. A coal burning Rotating Fluidized bed combustor operating at atmospheric pressure.