MIXTURE PREPARATION IN AUTOMOTIVE SPARK-IGNITION ENGINES

- with particular reference to multipoint fuel systems.

by

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ABSTRACT

This thesis commences by reviewing current methods of fuel mixture preparation in automotive spark-ignition engines, identifying both desirable and adverse characteristics. Experimental techniques for measurement of fuel droplet spray size distributions are considered, before describing design and operation of novel test rigs incorporating a laser-diffraction drop sizing instrument. Such measurements were made upstream and downstream of the intake valve on representative multipoint fuel injected engine components. Comparisons are drawn between the mixture quality delivered by this modern form of fuel system and more traditional carburetted systems. Interpretation of such Malvern Particle Sizer results is considered critically.

In-cylinder mixture preparation on one engine under test proved poor to the extent that a majority of the fuel entered the combustion chamber in liquid form. Excessive droplet size was postulated as a probable contributory factor. Motored rigs, based on two distinct types of cylinder head, were devised in order to quantify the wallfilm by measurement. A significant conclusion was that its prevalence (or otherwise) is indeed a direct function of the droplet size achieved upstream of the inlet valve.

A computer model was constructed to predict in-cylinder droplet trajectories, and confirmed that relatively smaller droplets introduced at the valve seat would travel further within the cylinder before wall impaction, increasing the time available for evaporation.

Rig experimentation and droplet trajectory theory is complemented by explanation of a cylinder pressure based technique for assessment of mixture preparation changes on the performance of a running engine. The benefits of a new type of port throttled fuel injection system, specifically designed to produce finer droplets at low load, are thereby demonstrated in terms of combustion stability. Reduced contamination of incoming charge by exhaust residuals, via a mechanism of port pressure recovery, is also held to be partially responsible for stability improvements and reduction in fuel consumption.
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>General nomenclature</td>
<td>11</td>
</tr>
<tr>
<td>Acknowledgements</td>
<td>14</td>
</tr>
<tr>
<td><strong>CHAPTER 1.0 LITERATURE SURVEY</strong></td>
<td></td>
</tr>
<tr>
<td>1.1 INTRODUCTION</td>
<td>16</td>
</tr>
<tr>
<td>1.2 FUEL METERING METHODS AND TRENDS</td>
<td></td>
</tr>
<tr>
<td>1.2.1 Carburettors</td>
<td>17</td>
</tr>
<tr>
<td>1.2.2 Fuel injection systems</td>
<td>18</td>
</tr>
<tr>
<td>1.3 SIGNIFICANCE OF MIXTURE PREPARATION</td>
<td></td>
</tr>
<tr>
<td>1.3.1 Effects on fuel economy</td>
<td>20</td>
</tr>
<tr>
<td>1.3.2 Effects on emissions</td>
<td>21</td>
</tr>
<tr>
<td>1.4 INVESTIGATING MIXTURE PREPARATION</td>
<td></td>
</tr>
<tr>
<td>1.4.1 Possible approaches</td>
<td>22</td>
</tr>
<tr>
<td>1.4.2 Fuel droplet sprays</td>
<td>22</td>
</tr>
<tr>
<td>1.4.3 Fuel droplet sizing techniques</td>
<td>24</td>
</tr>
<tr>
<td>1.4.4 Manifold investigations</td>
<td>28</td>
</tr>
<tr>
<td>1.4.5 Inlet valve and cylinder flows</td>
<td>30</td>
</tr>
<tr>
<td>1.4.6 Topics for continuing investigation</td>
<td>31</td>
</tr>
<tr>
<td><strong>CHAPTER 2.0 TEST ENVIRONMENT AND EQUIPMENT</strong></td>
<td></td>
</tr>
<tr>
<td>2.1 FUEL SYSTEMS TEST FACILITY</td>
<td>37</td>
</tr>
<tr>
<td>2.2 FUEL AND AIRFLOW MEASUREMENT</td>
<td>38</td>
</tr>
<tr>
<td>2.3 DROPLET SIZE MEASUREMENT</td>
<td></td>
</tr>
<tr>
<td>2.3.1 Malvern Droplet and Particle Size Analyser principles</td>
<td>38</td>
</tr>
<tr>
<td>2.3.2 Malvern Analyser calculation methods</td>
<td>40</td>
</tr>
<tr>
<td>2.3.3 Interpretation of a Malvern measurement</td>
<td>42</td>
</tr>
</tbody>
</table>
Chapter 3.0 EXPERIMENTAL WORK -
MIXTURE PREPARATION UPSTREAM OF INLET VALVE

3.1 MULTIPOINT INJECTOR SPRAYS INTO FREE AIR
3.1.1 Characteristics of injector mixture preparation 53
3.1.2 Arrangement of experiment 53
3.1.3 Typical free air spray results 55
3.1.4 Injector-to-injector variation 56
3.1.5 Spray photography 57

3.2 MULTIPOINT INJECTOR SPRAYS IN STEADY AIRFLOW
3.2.1 Experimental arrangement 58
3.2.2 Results at various air flow rates 59
3.2.3 Effect of fuel properties 61

3.3 INJECTOR AND CARBURETTOR SPRAY QUALITY COMPARED
3.3.1 Carburettor spray measurements 63
3.3.2 Comparison of carburettor and injector results 64

CHAPTER 4.0 EXPERIMENTAL WORK -
MIXTURE PREPARATION DOWNSTREAM OF INLET VALVE

4.1 STEADY FLOW SIMULATIONS
4.1.1 Steady flow 'blown' rig 80
4.1.2 Typical results with steady flow blown rig 81
4.1.3 Photographs and confirmation of evaporation effect 82
4.1.4 Importance of injector orientation 83
4.1.5 Steady flow 'suction' rig 85
4.1.6 Typical results with steady flow suction rig 86

4.2 CYLINDER WALLFILM MEASUREMENT
4.2.1 Requirement for a wallfilm measurement technique 87
4.2.2 Possible approaches to wallfilm measurement  

4.3 PULSATING FLOW RIGS FOR WALLFILM MEASUREMENT 
4.3.1 Rig based on Ford V6 engine  
4.3.2 Rig based on Ford Zeta four-valve engine  

4.4 WALLFILM RIG RESULTS 
4.4.1 Ford V6 engine results  
4.4.2 Ford Zeta engine results  
4.4.3 Inference from wallfilm rig results  

CHAPTER 5.0 THEORETICAL PREDICTION OF DROPLET TRAJECTORY 

5.1 INTRODUCTION  

5.2 EQUATIONS OF MOTION 
5.2.1 Radial direction  
5.2.2 Tangential direction ($\omega > \dot{\theta}$)  
5.2.3 Tangential direction ($\omega < \dot{\theta}$)  

5.3 DIFFERENTIAL EQUATION SOLUTIONS 
5.3.1 Radial direction  
5.3.2 Tangential direction ($\omega > \dot{\theta}$)  
5.3.3 Tangential direction ($\omega < \dot{\theta}$)  

5.4 DRAG COEFFICIENT  

5.5 EVAPORATION MODEL  
5.5.1 Fuel properties for evaporation model  

5.6 COMPUTER PROGRAM  
5.6.1 Program validation  

5.7 PARAMETRIC STUDIES  
5.7.1 Choice of input values  
5.7.2 Simulation results  

Page 88  
Page 89  
Page 90  
Page 91  
Page 93  
Page 95  
Page 112  
Page 113  
Page 113  
Page 113  
Page 114  
Page 116  
Page 116  
Page 117  
Page 118  
Page 118  
Page 119  
Page 120  
Page 120  
Page 122
CHAPTER 6.0 TESTS OF PORT THROTTLED ENGINE

6.1 INTRODUCTION

6.2 PORT THROTTLING
6.2.1 Theoretical benefits of port throttles
6.2.2 Design of port throttle unit for test bed engine
6.2.3 Test bed installation
6.2.4 Engine management system
6.2.5 Commissioning and operation of port throttled engine

6.3 DATA ACQUISITION AND ANALYSIS METHODS
6.3.1 Mixture quality assessment
6.3.2 Data acquisition with LabVIEW
6.3.3 Cylinder pressure transducer selection
6.3.4 Data acquisition triggering

6.4 ENGINE TESTING
6.4.1 Objectives
6.4.2 Notes on presentation of results
6.4.3 Baseline results - standard engine
6.4.4 Pressure drop across port throttles
6.4.5 Optimisation of fuel injection timing
6.4.6 Optimisation of ignition timing
6.4.7 50° valve overlap
6.4.8 Implications of port throttle tests

OVERALL CONCLUSIONS

FIGURES

Fig. 1.1 The simple carburettor: schematic to show basic principle
Fig. 1.2 Response of carburettor droplet size to manifold depression
Fig. 1.3 Equivalence ratio requirements at various conditions
<table>
<thead>
<tr>
<th>Fig.</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.4</td>
<td>Multipoint fuel injection system (Bosch LE-Jetronic)</td>
<td>33</td>
</tr>
<tr>
<td>1.5</td>
<td>Single-point fuel injection system (Bosch Mono-Jetronic)</td>
<td>34</td>
</tr>
<tr>
<td>1.6</td>
<td>Cross-section through Bosch multipoint fuel injector</td>
<td>34</td>
</tr>
<tr>
<td>1.7</td>
<td>Typical response of exhaust emissions to equivalence ratio ((\lambda))</td>
<td>35</td>
</tr>
<tr>
<td>2.1</td>
<td>General arrangement of Fuel Systems Test Facility</td>
<td>47</td>
</tr>
<tr>
<td>2.2</td>
<td>Malvern Analyser: schematic of optical arrangement</td>
<td>48</td>
</tr>
<tr>
<td>2.3</td>
<td>Fourier Transform lens</td>
<td>48</td>
</tr>
<tr>
<td>2.4</td>
<td>Geometry of semi-circular detector</td>
<td>48</td>
</tr>
<tr>
<td>2.5</td>
<td>Diffracted light energy distribution in the focal plane</td>
<td>49</td>
</tr>
<tr>
<td>2.6</td>
<td>Intersection between spray cone and laser beam (schematic)</td>
<td>49</td>
</tr>
<tr>
<td>2.7</td>
<td>Estimated number of droplets in sampling volume</td>
<td>49</td>
</tr>
<tr>
<td>2.8</td>
<td>Effect of fuel volatility on Malvern Analyser histograms</td>
<td>50</td>
</tr>
<tr>
<td>2.9</td>
<td>Sensitivity of Sauter mean diameter to small droplets</td>
<td>50</td>
</tr>
<tr>
<td>3.1</td>
<td>Multipoint injector sprays into free air: size distributions</td>
<td>66</td>
</tr>
<tr>
<td>3.2</td>
<td>Free air spray size distributions with involatile fuel</td>
<td>67</td>
</tr>
<tr>
<td>3.3</td>
<td>Injector-to-injector variation: 10 ms measurement delay</td>
<td>68</td>
</tr>
<tr>
<td>3.4</td>
<td>Injector-to-injector variation: 14 ms measurement delay</td>
<td>69</td>
</tr>
<tr>
<td>3.5</td>
<td>Viewing passage for Malvern Particle Size Analyser</td>
<td>70</td>
</tr>
<tr>
<td>3.6</td>
<td>Injector spray size distributions at various air flowrates</td>
<td>70</td>
</tr>
<tr>
<td>3.7</td>
<td>Effect of port vacuum on injector spray size distribution</td>
<td>71</td>
</tr>
<tr>
<td>3.8</td>
<td>Injector sprays with various fuels (free air)</td>
<td>71</td>
</tr>
<tr>
<td>3.9</td>
<td>Injector sprays with various fuels (900 rev/min idle)</td>
<td>72</td>
</tr>
<tr>
<td>3.10</td>
<td>Injector sprays with various fuels (1500 rev/min road load)</td>
<td>73</td>
</tr>
<tr>
<td>3.11</td>
<td>Injector sprays with various fuels (2000 rev/min WOT)</td>
<td>74</td>
</tr>
<tr>
<td>3.12</td>
<td>Carburettor droplet size distributions for Conditions 1 - 6</td>
<td>75</td>
</tr>
<tr>
<td>3.13</td>
<td>Comparison of carburettor and multipoint injector droplet sizes</td>
<td>76</td>
</tr>
<tr>
<td>4.1</td>
<td>Particle Sizer and flashgun triggering circuit</td>
<td>97</td>
</tr>
<tr>
<td>4.2</td>
<td>Laser beam path over valves in blowing simulation</td>
<td>98</td>
</tr>
<tr>
<td>4.3</td>
<td>Typical in-cylinder droplet sizes measured on 'blown' rig</td>
<td>98</td>
</tr>
<tr>
<td>4.4</td>
<td>Normalised air velocity around inlet valve periphery</td>
<td>98</td>
</tr>
<tr>
<td>4.5</td>
<td>Spurious Analyser indications due to refraction by fuel vapour</td>
<td>99</td>
</tr>
<tr>
<td>4.6</td>
<td>Importance of injector orientation - effect on spray in cylinder</td>
<td>99</td>
</tr>
<tr>
<td>4.7</td>
<td>Importance of injector orientation - effect on spray in port</td>
<td>100</td>
</tr>
<tr>
<td>4.8</td>
<td>Dummy cylinder for steady flow suction rig</td>
<td>100</td>
</tr>
</tbody>
</table>
Fig. 4.9  Steady flow suction rig: typical in-cylinder droplet sizes 101
Fig. 4.10  Cross-section of wallfilm skimming cylinder 101
Fig. 4.11  Section through endoscope 102
Fig. 4.12  Effect of skimming suction on wallfilm collected (V6 engine) 102
Fig. 4.13  Influence of injector fuel supply pressure on cylinder wall film 102
Fig. 4.14  Cylinder wall film as a function of carburettor depression 103
Fig. 4.15  Effect of skimming suction on wallfilm collected (Zeta engine) 103

Fig. 5.1  Notation for mathematical model of droplet trajectory 125
Fig. 5.2  Effect of initial droplet diameter on predicted trajectory 125
Fig. 5.3  Forced evaporation rate according to initial droplet diameter 126
Fig. 5.4  Effect of initial radial droplet position on predicted trajectory 126
Fig. 5.5  Effect of initial radial velocity of droplet on trajectory 127
Fig. 5.6  Effect of initial droplet angular velocity on trajectory 127
Fig. 5.7  Effect of air swirl velocity on predicted trajectory 128
Fig. 5.8  Trajectories of SBP 3 and n-heptane droplets compared 128
Fig. 5.9  Effect of increased initial droplet temperature 128

Fig. 6.1  Schematic of port throttle disc sealing method 150
Fig. 6.2  Incompatibility of port shape with 4 disc port throttle system 150
Fig. 6.3  Pressure loss characteristic for port throttle and de-act units 151
Fig. 6.4  Injector orientation relative to port throttle discs 151
Fig. 6.5  Workshop drawing for basic machining of port throttle body 152
Fig. 6.6  Diagram of a typical 'Sub-VI' as used in LabVIEW 153
Fig. 6.7  Labview 'front panel', showing one form of results presentation 153
Fig. 6.8  Logarithmic motored P-V plot for establishing trigger angle 154
Fig. 6.9  Optimisation of pressure drop across port throttles, 800 rpm 155
Fig. 6.10  Reduced area of pumping loop on port throttled Zeta engine 156
Fig. 6.11  Optimisation of port throttled fuel injection timing, 800 rpm 157
Fig. 6.12  Optimisation of port throttled fuel injection timing, 1200 rpm 158
Fig. 6.13  Optimisation of port throttled fuel injection timing, 1500 rpm 159
Fig. 6.14  Response to alteration of ignition timing, 800 rpm 160
Fig. 6.15  SDIMEP and LNV compared for each engine configuration 161
Fig. 6.16  Fuel pulsewidths compared for each engine configuration 161
Fig. 6.17  Response to alteration of ignition timing, 1200 rpm 162
Fig. 6.18  Response to alteration of ignition timing, 1500 rpm 163
Fig. 6.19  Comparison of engine speed profiles, with 50° valve overlap 164
Fig. 6.20 Optimisation of port throttled ignition timing, at 50° overlap 164
Fig. 6.21 Relative improvement of idle stability due to port throttles 165

PLATES

Plate 1 UCL 'Fuel Systems Test Facility' 51
Plate 2 Plenum chamber for vacuum supply 51
Plate 3 Equipment layout for rig tests of free air sprays 76
Plate 4 Bosch manual control unit and coil for injector triggering 77
Plate 5 Typical form of multipoint fuel injector spray (gasoline) 77
Plate 6 Poor quality spray during initial pulses (gasoline) 77
Plate 7 Perspex replica of injector mounting tube 78
Plate 8 Arrangement for measuring droplet sizes of Weber carburettor 78
Plate 9 Cut-down Ford V6 cylinder head mounted on Malvern Analyser 104
Plate 10 General view of V6 Blowing Rig, with perspex dummy cylinder 104
Plate 11 Injector mounting in manifold - blowing rig 105
Plate 12 General view of Malvern Analyser and blowing rig 105
Plate 13 Pitot tube for measuring flow velocity around inlet valve 106
Plate 14 General view of steady flow suction rig, based on Ford V6 engine 106
Plate 15 Detail of dummy cylinder on steady flow suction rig 107
Plate 16 Preliminary version of suction rig with motored inlet valves 107
Plate 17 Pulsating flow rig for wallfilm measurement, based on Ford V6 engine 108
Plate 18 Pulsating flow rig for wallfilm measurement, based on Ford V6 engine 108
Plate 19 Endoscopic view of Ford V6 engine combustion chamber 109
Plate 20 Pulsating flow rig for wallfilm measurement, based on Zeta engine 109
Plate 21 Endoscopic view of Ford Zeta engine combustion chamber 109
Plate 22 V6 wallfilm rig with Weber carburettor fitted 110
Plate 23 Zeta wallfilm rig with port throttles fitted 110
Plate 24 Port throttle unit: view on face adjoining cylinder head 166
Plate 25 Port throttle unit, showing transition to suit twinned manifold 166
Plate 26 Detail of port throttle disc and hand blending of wall contour 167
Plate 27 General layout of Ford Zeta engine test cell 167
Plate 28 Location of pressure transducer in No.3 cylinder 168
<table>
<thead>
<tr>
<th>Appendix I</th>
<th>References</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Appendix II</td>
<td>Problems experienced with Malvern Analyser software</td>
<td>175</td>
</tr>
<tr>
<td>Appendix III</td>
<td>Latex and alumina particle histograms</td>
<td>178</td>
</tr>
<tr>
<td>Appendix IV</td>
<td>Steady flow fuel injection test conditions</td>
<td>182</td>
</tr>
<tr>
<td>Appendix V</td>
<td>Surface tension and viscosity data for hydrocarbons</td>
<td>183</td>
</tr>
<tr>
<td>Appendix VI</td>
<td>Carburettor test conditions</td>
<td>183</td>
</tr>
<tr>
<td>Appendix VII</td>
<td>Hydrocarbon data as averaged to represent SBP3 properties</td>
<td>185</td>
</tr>
<tr>
<td>Appendix VIII</td>
<td>'DROPTRAJ' program flowchart, nomenclature and listing</td>
<td>189</td>
</tr>
<tr>
<td>Appendix IX</td>
<td>Test conditions and test results from Zeta engine tests</td>
<td>204</td>
</tr>
<tr>
<td>Symbol</td>
<td>Designation</td>
<td>Units</td>
</tr>
<tr>
<td>--------</td>
<td>-----------------------------------------------------------------</td>
<td>------------</td>
</tr>
<tr>
<td>A</td>
<td>droplet projected area</td>
<td>m²</td>
</tr>
<tr>
<td>C_{dr}</td>
<td>drag coefficient (radial direction)</td>
<td>-</td>
</tr>
<tr>
<td>C_{dt}</td>
<td>drag coefficient (tangential direction)</td>
<td>-</td>
</tr>
<tr>
<td>C_p</td>
<td>Specific heat capacity at constant pressure</td>
<td>kJ/kgK</td>
</tr>
<tr>
<td>D</td>
<td>diameter of droplet</td>
<td>μm</td>
</tr>
<tr>
<td>d</td>
<td>diameter of orifice</td>
<td>μm</td>
</tr>
<tr>
<td>D_{3,0}</td>
<td>mean volume droplet diameter</td>
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<td>D_{3,2}</td>
<td>Sauter mean diameter</td>
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</tr>
<tr>
<td>Γ</td>
<td>exchange coefficient of fuel vapour in mixture</td>
<td>kg/ms</td>
</tr>
<tr>
<td>G_{fs}</td>
<td>Rate of phase change per unit surface area</td>
<td>kg/m²s</td>
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<tr>
<td>k</td>
<td>thermal conductivity</td>
<td>kW/mK</td>
</tr>
<tr>
<td>m</td>
<td>droplet mass</td>
<td>kg</td>
</tr>
<tr>
<td>μ</td>
<td>viscosity</td>
<td>P</td>
</tr>
<tr>
<td>v</td>
<td>relative velocity between air and liquid streams</td>
<td>m/s</td>
</tr>
<tr>
<td>NNu'</td>
<td>mass transfer number (cf. ref No. 65)</td>
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<tr>
<td>Pr</td>
<td>Prandtl number (= μC_p/kvap)</td>
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<tr>
<td>Q</td>
<td>volume flow rate</td>
<td>m³/hr</td>
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<tr>
<td>θ</td>
<td>angular position of droplet</td>
<td>rad</td>
</tr>
<tr>
<td>θ'</td>
<td>tangential component of droplet velocity</td>
<td>rad/s</td>
</tr>
<tr>
<td>θ''</td>
<td>droplet acceleration in tangential direction</td>
<td>rad/s²</td>
</tr>
<tr>
<td>ρ</td>
<td>density</td>
<td>g/cm³</td>
</tr>
<tr>
<td>r</td>
<td>radius at which droplet introduced to cylinder</td>
<td>m</td>
</tr>
<tr>
<td>r</td>
<td>radius of droplet (alternative usage of r)</td>
<td>m</td>
</tr>
<tr>
<td>r'</td>
<td>radial component of droplet velocity</td>
<td>m/s</td>
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<tr>
<td>r''</td>
<td>droplet acceleration in radial direction</td>
<td>m/s²</td>
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<td>Re</td>
<td>Reynolds number</td>
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<td>Unit</td>
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<td>--------</td>
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<td>-----------------</td>
</tr>
<tr>
<td>σ</td>
<td>surface tension</td>
<td>dyne/cm</td>
</tr>
<tr>
<td>Sc</td>
<td>Schmidt number ((= \mu/pDvap))</td>
<td></td>
</tr>
<tr>
<td>(s_r)</td>
<td>swirl ratio (swirl velocity + crankshaft angular velocity)</td>
<td></td>
</tr>
<tr>
<td>t</td>
<td>time (also step width)</td>
<td>s</td>
</tr>
<tr>
<td>(\omega)</td>
<td>angular velocity of solid body air rotation</td>
<td>rad/s</td>
</tr>
<tr>
<td>We</td>
<td>Weber number ((= \sigma^2/pDv^2))</td>
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</tr>
<tr>
<td>Y</td>
<td>mass fraction</td>
<td></td>
</tr>
</tbody>
</table>

**Subscript**

<table>
<thead>
<tr>
<th>Subscript</th>
<th>Referring to...</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>air</td>
</tr>
<tr>
<td>fs</td>
<td>value at fuel surface</td>
</tr>
<tr>
<td>i</td>
<td>an initial condition</td>
</tr>
<tr>
<td>L</td>
<td>liquid (fuel)</td>
</tr>
<tr>
<td>max</td>
<td>a maximum value</td>
</tr>
<tr>
<td>p</td>
<td>a discrete pulse of spray</td>
</tr>
<tr>
<td>s</td>
<td>a sampling time</td>
</tr>
<tr>
<td>vap</td>
<td>fuel vapour</td>
</tr>
<tr>
<td>(\infty)</td>
<td>an ambient value</td>
</tr>
</tbody>
</table>

**Acronym**

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Meaning</th>
</tr>
</thead>
<tbody>
<tr>
<td>AFR</td>
<td>Air/fuel ratio</td>
</tr>
<tr>
<td>BMEP</td>
<td>Brake mean effective pressure</td>
</tr>
<tr>
<td>BSFC</td>
<td>Brake specific fuel consumption</td>
</tr>
<tr>
<td>CoV</td>
<td>Coefficient of variation (in engine stability analysis)</td>
</tr>
<tr>
<td>DOHC</td>
<td>Double overhead camshaft</td>
</tr>
<tr>
<td>DVM</td>
<td>Digital volt meter</td>
</tr>
<tr>
<td>ECE</td>
<td>United Nations Economic Commission for Europe</td>
</tr>
<tr>
<td>ECU</td>
<td>Electronic control unit</td>
</tr>
<tr>
<td>EDIS</td>
<td>Electronic distributorless ignition system</td>
</tr>
<tr>
<td>Acronym</td>
<td>Definition</td>
</tr>
<tr>
<td>---------</td>
<td>------------</td>
</tr>
<tr>
<td>EFI</td>
<td>Electronic fuel injection</td>
</tr>
<tr>
<td>EGO</td>
<td>Exhaust gas oxygen sensor</td>
</tr>
<tr>
<td>EGR</td>
<td>Exhaust gas recirculation</td>
</tr>
<tr>
<td>IMEP</td>
<td>Indicated mean effective pressure</td>
</tr>
<tr>
<td>ISC</td>
<td>Idle speed control</td>
</tr>
<tr>
<td>ISFC</td>
<td>Indicated specific fuel consumption</td>
</tr>
<tr>
<td>IVC</td>
<td>Inlet valve closed/closing</td>
</tr>
<tr>
<td>IVO</td>
<td>Inlet valve open/opening</td>
</tr>
<tr>
<td>LDA</td>
<td>Laser doppler anemometry</td>
</tr>
<tr>
<td>LNV</td>
<td>Lowest normalised value (in engine stability analysis)</td>
</tr>
<tr>
<td>MBT</td>
<td>Minimum spark advance for best torque</td>
</tr>
<tr>
<td>MPI</td>
<td>Multipoint injection</td>
</tr>
<tr>
<td>NVH</td>
<td>'Noise, vibration, and harshness' (in perception of engine operation)</td>
</tr>
<tr>
<td>SDIMEP</td>
<td>Standard deviation of IMEP</td>
</tr>
<tr>
<td>SERC</td>
<td>Science and Engineering Research Council</td>
</tr>
<tr>
<td>SI</td>
<td>Spark ignition</td>
</tr>
<tr>
<td>Smd</td>
<td>Sauter mean diameter (also referred to as D3,2)</td>
</tr>
<tr>
<td>SSA</td>
<td>Specific surface area (of droplet)</td>
</tr>
<tr>
<td>TBI</td>
<td>Throttle body injection</td>
</tr>
<tr>
<td>TDC</td>
<td>Top dead centre (of piston)</td>
</tr>
<tr>
<td>VI</td>
<td>Virtual Instrument in LabVIEW software</td>
</tr>
<tr>
<td>WOT</td>
<td>Wide open throttle</td>
</tr>
</tbody>
</table>
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CHAPTER 1.0

LITERATURE SURVEY
CHAPTER 1.0

LITERATURE SURVEY

1.1 INTRODUCTION

A fundamental requirement of all spark-ignition engines is that an air/fuel mixture of closely
controlled proportions and quality should reach each combustion chamber. The assembly of
components designed to satisfy this demand is known collectively as the 'fuel system', and
consists principally of some form of atomizing device, be it carburettor or fuel injector, which
discharges into a manifold or port for onward supply of the mixture to the various engine
cylinders. The term 'mixture preparation' describes the state of fuel and air after mixing,
with reference to how much fuel has been vaporized, how small the droplets are, and how
they are distributed in the air stream.

Since gasoline will burn only as a vapour, the processes of droplet formation by atomization,
and subsequent evaporation, are central to the development of fuel systems. Ignition can occur
typically at gravimetric air/fuel ratios between 12:1 and 18:1, but the substantial variations
in exhaust emissions, power and efficiency experienced within that range call for precision in
fuel metering to secure the best compromise. Between the extremes, dependent on the fuel
composition, is the stoichiometric AFR, at which just sufficient air is present to convert all the
fuel to completely oxidised products. Towards the lean end, the quantities of incomplete
combustion products are small, whereas under conditions of excess fueling these amounts rise
rapidly, degrading the combustion efficiency to 60% or lower from the approximate 90%
attainable under lean conditions. In a practical engine, the extent to which the weak limit can
be approached depends significantly on the uniformity of mixture distribution and quality
between cylinders, as the onset of misfire in any one cylinder defines the limiting case for the
whole power unit. Any manifold wall fuel film is liable to cause not only inter-cylinder
distribution problems but also adverse mixture preparation in the combustion chamber itself,
as characteristically larger fuel droplets, which are less able to premix with air to form a
readily combustible mixture, tend to persist past the inlet valve.

Economic and environmental pressures to investigate fuelling phenomena are of course long-
established, but in the 1990's are being added ever more stringent legislative requirements for
new vehicles to comply with prescribed emission standards\textsuperscript{25}. Attitudes in the marketplace,
responding mainly to high fuel prices, but possibly also to concern over suggested connections

\textsuperscript{†} References alphabetically in Appendix I
between 'global warming' and an imbalance of atmospheric CO$_2$, appear to be making fuel economy a prime consideration in choice of vehicle; indeed, in the United States, statutory minimum economy levels are now demanded. In direct conflict with such considerations, manufacturers have to recognise customer expectations of driveability - not least in terms of good performance during warm up and transients.

The importance of the fuel system in all of these areas is paramount, and the true criterion of that system must be the quality which it can actually deliver to the area where combustion is to take place, downstream of the intake valve, with freedom from cyclic variation.

1.2 FUEL METERING METHODS and TRENDS

1.2.1 Carburettors

Historically, the carburettor (Fig. 1.1) has been much the most common mixture preparation device for S.I. engines, performing the dual roles of spraying fuel into the air stream induced by the engine and metering the fuel quantity appropriate to engine load. Air passes through a venturi and entrains fuel from a discharge tube sited in the region of depression at the throat, so that fuel flow is directly referenced to the pressure drop, which in turn is consequent on the instantaneous engine air flow rate. The air/fuel mixture next encounters a throttle disc, permitting control of power at part load conditions by reduction of the quantity admitted to the engine, before entering a manifold for transport to individual cylinders.

The carburettor has probably owed much of its success to the fact that spray originating from the venturi experiences further atomisation at part throttle conditions, due to the local acceleration of flow around the leading and trailing edges of the throttle disc. As Finlay et al.$^{18,19}$ have shown from measurements on a running engine, the fuel droplet sizes emerging from an air-valve carburettor are a function of the manifold depression set up by throttling the air flow, and typically form a distribution (Fig. 1.2) with Rosin-Rammler characteristic diameters of 6 - 8 µm at the lightest loads, increasing to around 70 µm once the throttle is opened wide for maximum power. They recognised the existence of a critical pressure ratio across the throttle disc, beyond which further reduction in manifold pressure can achieve no increase in air velocity, and correspondingly no benefit to droplet size.

The conclusion is that at light loads the carburettor provides extremely fine droplets, maximising evaporation, but that under high-load conditions its performance deteriorates drastically. As Barnes$^6$ notes, however, the carburettor remains attractive on grounds of first

$^\dagger$ Figures at end of Chapter
cost, and subject to emissions legislation it is therefore likely to continue in widespread use, perhaps more particularly in cheaper vehicles and with greater use of electronic trims to control idle speed, fuelling during overrun and cold starting enrichment.

1.2.2 Fuel injection systems

There is an optimum air/fuel ratio for every speed and load point on the engine map, dependent on the aspect of performance it is desired to optimise, whether this be full load power output, part load economy, or minimum emissions. Lenz\textsuperscript{43} of the Technical University of Vienna plotted this optimal AFR as a three-dimensional representation corresponding to speed and load (Fig. 1.3), pointing out that the plot would vary according to engine temperature, mechanical condition and atmospheric conditions, and that it is extremely difficult to compensate for all these variables in a carburettor design. Further, if such exhaust gas after-treatment techniques as the three-way catalyst are to be applied, and optimum conversion efficiency realised, then the acceptable AFR window either side of stoichiometric is extremely narrow.

This problem has stimulated the development of fuel-injection techniques which better lend themselves to electronic control, enabling the amount of fuel injected per cycle to be varied in response to sensors which define actual engine operating conditions\textsuperscript{10,43}. The electronic control element also facilitates introduction of deliberate constraints during calibration of the engine map, in order that operating conditions which would produce unacceptable emissions levels may be avoided. It is thought-provoking to contrast this ease of alteration with the methods available to pioneers of fuel injection in the pre-electronic era - adjustments of AFR for cold-starting, warm-up, cruising, coasting, sudden acceleration or air density change then had to be made through ingenious mechanical systems, typically involving a spherical cam on which each surface point defined a particular set of operating circumstances.

In contrast to a carburetted system, multipoint injection systems (Fig. 1.4) inject fuel into the individual intake ports relatively close to the cylinder where it is needed, rather than into the intake system generally. Throttle response ought therefore to reflect the reduced mixture transport time, and inter-cylinder distribution should be optimised, subject only to uniform injector performance and equal air flow into each cylinder. It is also believed that port-injection increases power, by allowing increase of compression ratio beyond the knock limit which would constrain the same engine if fitted with a carburettor, and torque output is enhanced through improved volumetric efficiency and more accurate fuel metering, which helps to avoid 'flat spots'. Tighter control over air/fuel ratio during cold starting and warm-up is enabled, and having circumvented the manifold wall wetting problems often inherent in carburetted systems, a freer hand in tuned intake manifold design is possible.
Single-point throttle body injection systems (Fig. 1.5), described by Heywood, by definition lose this advantage, but offer a less costly alternative by using one or two injectors to meter fuel into the air flow directly above a throttle body. However, the introduction of fuel in pulses, rather than in direct response to air flow as with the carburettor, can lead to inter-cylinder distribution and mixture preparation problems. This is particularly so as large droplets tend to be characteristic of the low pressure injectors normally used, and in passing through the manifold these may not be able to follow directional changes without appreciable wall impaction occurring. There have nonetheless been some dramatic claims made for TBI systems: in 1988 Chrysler reported 20% increase in maximum power as a result of discarding the carburettor on a 5.2 litre V8 truck engine in favour of fuel injection.

In an undoubtedly partial review, Greiner et al of Robert Bosch GmbH, predicted that by 1990 electronic fuel-injection would be universal in new cars for the American market, and that a large percentage of European cars also would be so equipped. There are as yet only tentative signs in the UK of either form of fuel-injection being adopted on a large scale for the smaller engine capacities; however, the progressively more stringent ECE emissions standards applicable above 1400cc displacement have led to a shift towards multipoint injection for performance and prestige cars, encouraged also by technical and marketing considerations. The mixture preparation of fuel injected automotive engines is consequently a highly topical subject for investigation.

Multipoint injection systems are classified according to the method by which air flow into the engine is deduced. On the basis of engine sensor indications of speed, manifold pressure and air temperature, a speed-density system calculates the air flow, although accurate implementation of this method can be problematical for engines operating with ram-effect induction or supercharging. Alternatively, the air flow can be measured by a meter sited upstream of the throttle, as on the Bosch L-Jetronic system employed on Ford EFI V-6 engines destined for the British market. This offers numerous advantages, such as inherent compensation for differences in cylinder filling arising from manufacturing tolerances, wear, or alterations in valve adjustment, as well as improved idle stability and a reduced requirement for acceleration enrichment; the system reacts so immediately to change of mass airflow rate that sudden throttle openings do not cause lean cutting problems. It also compensates automatically for differences in exhaust system back pressure, such as might occur over the service life of a catalytic converter.

The fuel injectors in Bosch systems are electromagnetically actuated and supplied with fuel under pressure from an external pump:- when the electronic control unit (ECU)
provides an excitation pulse to the solenoid coil, an armature is pulled in, lifting a valve needle off its seat and admitting fuel via a precise annular metering section and atomizing pintle to the inlet port. Thus the fuel mass injected is a function of the pulse duration, which lies normally in the range 1.5 to 10 ms. Heywood observes that these durations can occupy between 10° and 300° of crank angle rotation, thereby raising the question of when best to trigger the injector in the cylinder cycle, and how that event should be phased with the inlet valve opening and closing events.

Primary inputs to the ECU in the L-Jetronic system comprise air flow and engine speed signals, with further information from an engine block temperature sensor, starter switch and throttle potentiometer being interpreted to optimise fuelling for starting, warm-up, idle and transients. If the engine is operating with closed loop feedback control, to ensure precise fuelling at close to stoichiometric and permit use of a three-way catalyst, an oxygen sensor in the exhaust would take over the task of monitoring air/fuel ratio once warm-up is complete.

1.3 SIGNIFICANCE OF MIXTURE PREPARATION

1.3.1 Effects on fuel economy

The implications for fuel economy of mixture preparation have long been recognized, and a large body of literature has been generated. Nonetheless, Harrow warns of the difficulty in ascertaining that benefits claimed for any particular mixture preparation system are present under all operating conditions and are uniquely due to the better mixture quality. Exploitable fuel economy benefits, he emphasizes, will stem from mixture quality changes only if preparation can be improved for all engine conditions and if the carburettor or fuel injection system is retuned to exploit the leaner mixtures then permissible. Nightingale's view is that different engines show different sensitivities to mixture quality, some requiring a well-prepared mixture prior to the inlet valve, perhaps by arranging that a large proportion of the fuel be already evaporated, whereas others will be less critical and improve the mixture preparation at a late stage, by means such as ensuring a high-velocity flow through the inlet valve, as described by Beale and Hodgetts. It is apparent that good mixture preparation can enhance economy by improving either geometric fuel distribution or combustion, or both. Lenz shows that refined mixture preparation leads to considerably faster combustion, tolerant of higher amounts of exhaust gas recirculation (improving emissions), while improvement of fuel consumption due to lower pumping work, reduced heat transfer (as gas temperatures are lower), and less dissociation in the burned gases are cited by Heywood as further reasons why good mixture preparation can be beneficial. Others have noticed that
less spark advance is required when running with improved mixture preparation, implying that combustion is indeed more rapid.

Many workers have investigated the route of ensuring a well-prepared mixture early on in a central fuel preparation system, an exhaust heated 'hot spot' being in any case common practice with manifolds to help evaporate excessive wall film. It is well established\textsuperscript{9,33,43} that a homogeneous mixture of fuel and air in the engine inlet manifold, such as is produced by vapourizing, is better able to divide equally between branches for even cylinder-to-cylinder ('geometric') distribution; Lenz\textsuperscript{43} considers that droplets below about 10 µm diameter will follow practically every flow deflection in normal manifolds, and Robison and Brebob\textsuperscript{67} found that vaporization tank carburetion reduced the air/fuel ratio spread to 0.6 or less. This is a prerequisite for lean burn strategies, in order to prevent the weak limit being reached noticeably earlier in one cylinder than in others. Shayler and Armstrong\textsuperscript{69} noted approximately 2% deterioration in brake specific fuel consumption arising from a shift of one air/fuel ratio in one cylinder. Dodd and Wisdom\textsuperscript{13}, performed comparative tests, at steady speeds only, on a single-cylinder engine fuelled first by a port-mounted injector and then by a heated fuel vapourising device capable of producing dry mixtures downstream of the throttle. With the vaporised mixture the lean limit was extended by over two air/fuel ratios, however, the full throttle power loss which would presumably result from the increased mixture temperature, and the increased volume necessary for the fuel in its vapour phase, was not specified. Later, after a similar exercise, Hughes and Goulburn\textsuperscript{33} reported an extension of over three air/fuel ratios in lean limit but a 30% derating of maximum brake power, amply demonstrating the conflict between power output and economy, although an 8-10% reduction in specific fuel consumption was still realised.

In comparing the air/fuel ratio between cylinders for warmed-up engines, carburetted and injected respectively, Lenz saw that the carburettor performed well at low loads, but that fuel injection appeared preferable at high loads. This reflects the characteristically different droplet size distributions of the two systems, as will be shown, and possibly also better uniformity in flowrate between injectors at longer pulsewidths.

1.3.2 Effects on emissions
In the developed countries, regulations governing maximum pollutant levels from vehicles have been steadily tightened since the pollution problem was first perceived in the 1940's and '50's. The introduction\textsuperscript{6,25} of a further European standard, ECE 15.05, continues the trend, requiring carbon monoxide (CO), unburned hydrocarbon (HC) and nitric oxide (NOx) emission levels to be below re-defined limits over a test cycle which includes idle, accelerations, cruises and overruns.
Local charge inhomogeneity at the time of combustion greatly affects these emissions, as Shayler has reported. Output of NOx increases in rate with the combustion temperature, while excessive carbon monoxide formation is characteristic of rich fuel air mixtures, there then being insufficient oxygen available to burn all the carbon in the fuel to carbon dioxide; some CO will, however, always be present from dissociation. Unburned HC emissions originate particularly from any elements of fuel/air mixture forced into crevices remote from the main combustion area by compression and combustion, and which therefore escape burning. The classic variation of HC, CO, and NOx concentration without EGR (Fig. 1.7) shows that HC's and CO's reduce progressively as the mixture is weakened (although a rise in HC occurs beyond about 18:1), and that NOx peaks at just lean of stoichiometric but reduces steeply either side. Thus, to meet emissions targets there is a need for uniform mixture preparation, to avoid stratified rich and lean areas in the combustion chamber, and to control the overall air/fuel ratio with precision.

In their investigation of vaporized lean mixtures, Hughes and Goulburn concluded that a 19:1 AFR was near to optimum for their particular 2 litre engine. By operating with such weak mixtures it was demonstrated that all three pollutant emissions could be brought simultaneously to a low level, and it was claimed that under driving conditions transient emissions were less than on the conventionally carburetted version. However, as the vaporizer could not operate until the engine was warm, the cold-starting emissions (a substantial part of the ECE 15 requirements) would probably not have shown such improvement.

1.4 INVESTIGATING MIXTURE PREPARATION

1.4.1 Possible approaches

Investigation of the actual mixture preparation in an engine situation, rather than straightforward analysis of the effects of changes, is no easy task. The areas involved tend not to be readily visible and must be made accessible without disturbing existing flow phenomena. Engine-based studies have provided useful information on the transport of mixture in manifolds, but some researchers have preferred to devise air-flow simulation rigs not involving a running engine, or to develop mathematical models. In the present work it is the two latter approaches which have been mainly employed, with final validation taking place on a test bed engine.

1.4.2 Fuel droplet sprays

Before considering mixture preparation in the engine context, a review of the main parameters affecting droplet size is appropriate. Investigators have tended to adopt either
empirical methods, proposing equations for droplet size on the basis of experimental results, or to deduce from theoretical considerations the maximum droplet size which should be able to exist under particular conditions. As monodisperse sprays are seldom encountered, sprays are often characterised by means of a Sauter mean diameter, \( D_{3,2} \), which represents the droplet diameter of a notional spray of a uniform size such that it has the same overall liquid volume as the actual spray, and also the same total surface area\(^5\).

After work with a water jet atomised by a high speed air stream, Nukiyama and Tanasawa\(^5\) put forward an equation of the empirical type in 1937, which appears\(^5\) to read across moderately well to the carburettor situation.

\[
D_{3,2} = \frac{585}{v} \frac{\sqrt{\sigma_L}}{\sqrt{\rho_L}} + 597 \left( \frac{\mu_L}{\sqrt{\sigma_L \rho_L}} \right)^{0.45} \left( \frac{1000 \dot{Q}_L}{\dot{Q}_A} \right)^{1.5}
\]  

(1)

From this equation, smaller SMD would appear to follow from increased relative velocities between air and liquid streams, reduced surface tensions and increased densities.

In the 1950's Ingebo and Foster\(^3\) similarly produced an empirical equation to describe an injection of fuel from a plain orifice at right angles to an air stream, which suggested that mean volume droplet diameter falls with the diameter of discharge orifice, as would be expected, and that increasing Reynolds number and/or reducing Weber number were beneficial:-

\[
D_{3,0} = 3.9 \left( \frac{1}{Re} \right)^{0.25} \left( \frac{\sigma_L}{\rho_A \dot{d} v_i^2} \right)^{0.25}
\]  

(2)

Likewise, Lorenzetto and Lefebvre\(^4\) derived a dimensionally correct expression for plain jet atomiser SMD from an analysis of experimental data, suggesting that surface tension, viscosity, orifice diameter, and fuel/air ratio were all important, and claiming an accuracy of 8\%. Richards et al\(^6\) have reported on similarly empirical investigations into air-blast atomisers, coming to the same conclusions, and noting, as did Ingebo and Foster, that atomisation quality improves with increases in ambient air density. This latter point is of some concern to multipoint injection engines, where at part load conditions the injector is spraying into a region of manifold depression.
The theoretical route has been followed by Hinze\textsuperscript{31} and Prandtl\textsuperscript{63}. By combining a theoretical analysis of aerodynamic forces on a droplet with experimental determination of the constant in his equation, Prandtl showed once again that maximising air density, minimising fuel surface tension and maximising the average velocity between air and fuel should lead to the evolution of the finest droplets:

\[ 15.4 \approx \frac{D_{\text{max}} \rho_A u^2}{\sigma_L} \]  \quad (3)

This work was further studied by Hinze, with an emphasis on the effect of relative velocity variations with time, and the existence was proposed of a dimensionless viscosity group affecting the critical Weber number for shatter; the effect was a 6\% reduction of predicted droplet diameter as compared with the Prandtl equation.

### 1.4.3 Fuel droplet sizing techniques

The importance of mixture preparation is certain, and the droplet size equations above offer some guidance as to how it might be optimised, but for fundamental research and development work it is obvious that reliable methods of fuel droplet size measurement are required, to detect subtle changes in distribution of fast moving and intermittent sprays which would not be apparent visually. In a comprehensive review of drop size measurement techniques Jones\textsuperscript{36} categorizes the numerous available methods into three areas - mechanical, electrical, and optical - but assesses them with the objective of application to dense diesel sprays, whereas spark-ignition fuel system sprays are likely to be relatively rarified.

(i) Mechanical drop sizing methods

Techniques under this heading rely variously either on forms of direct measurement or provide for drops of different sizes to accumulate in specific collectors, enabling a distribution to be deduced. A long-established method, used by Nukiyama and Tanasawa, is to allow droplets to impact on a plate coated with an oil, magnesium oxide or vaseline\textsuperscript{36}, so that the resultant impression can subsequently be sized under a microscope. In the context of S.I. engine research this would be extremely time consuming, as many slides would have to be exposed to ensure any kind of accuracy, and in addition a coefficient would have to be applied to each impression to account for droplet deformation on impact. Also, there would be a tendency with well prepared mixtures for the droplets simply to bypass the plate and pass around either side, so that only the larger and faster droplets might impact. Increasing the plate proportions would only make the method still more intrusive on the engine environment.
Cascade impactors also use coated plates, but exploit the ability of smaller droplets to follow the airflow, through arranging several levels of slides so that each collects droplets of a specific size which have passed around the previous slide. However, the usual method of measuring mass for derivation of a distribution would be impractical with a volatile gasoline spray, and the alternative of counting particles through a microscope too tedious.

Other mechanical possibilities reviewed by Jones, such as replacement of the fuel by molten paraffin wax and measurement of the solidified particles, or the sedimentation technique, where use is made of the different terminal velocities of different sized particles, are even less suited to the present requirements.

(ii) Electrical droplet sizing methods

Electrical methods rely variously on use of droplets as momentary resistors, such as in the Wicks-Dukler technique, as influences on the local resistance of a wire (hot wire anemometry) or utilise the electrostatic capacity of drops to draw a size-dependent charge from an electrically charged wire. An attraction of such methods is that the probes are thin and therefore have little effect on the airflow.

The Wicks-Dukler technique, as described by Jones, arranges for momentary bridging by drops of a gap between needles across which a potential difference exists. The size of a droplet can supposedly then be derived from varying the needle separation and analysing frequency counts. In practice it has been found that drop velocity, and the penetration of the needles into the drop, are factors which greatly affect the apparent resistance and that the technique is prone to error. A further disadvantage is that an electrically conducting fluid is required, whereas for engine fuel system work it is desirable to have capability to test representative fuels.

Similar reservations apply as to droplet velocity and impingement position effects in the charged wire method, which Jones records has been developed by Gardiner and Bailey. A charge transfer to impinging droplets is apparent from voltage pulses across the wire, but liquid drops with low electrical conductivities have been found to give rise times long enough that successive impingements can become superimposed. Jones concluded that for all these reasons the charged wire technique suffered inherent problems which were insurmountable.

Hot wire anemometry has its basis in the local cooling and resistance increase caused by droplets impacting on a resistively heated wire. However, Jones quotes Moore and Bragg as discovering that the technique was once again susceptible to variable indications according to
the impact velocity of the drops. It was found that Weber numbers have to be less that about 2.5 in order that a droplet be retained on the wire. Medecki and Magnus, Jones reports, later suggested that the upper limit of droplet velocity would be of the order of 3 m/s. The fuel velocity at exit from a multipoint injector is typically over 20 m/s, and air valve carburettors can produce mixture velocities in excess of 60 m/s just downstream of the throttle disc, so this method again appears ill-suited to S.I. engine fuel spray measurements. A general observation on 'electrical' droplet sizing techniques as reported in the literature is that, per measurement, none would sample more than only a very small area of spray. In an intake manifold, or downstream of an inlet valve, it is preferable to extract size distributions over a larger measurement volume.

(iii) Optical drop sizing methods

Optical approaches to particle size measurement are numerous. Fraidl lists photography and holography, light scattering methods, and sundry others such as obscuration and extinction. A major attraction of these methods for engine work is that they are non-intrusive to the spray, except to the extent that provision of suitable optical access may necessitate modifications to the surrounding manifold or cylinder.

Photographic methods require illumination of extremely short duration in order to minimise image blur, and subsequent sizing of film images is time consuming. Plimon et al of AVL-LIST illustrate gasoline injector sprays into ambient atmosphere by means of single shot photography, but the resolution appears adequate only for a first order impression of the spray structure. Jones performed trials of a pulsed ruby laser as an intense light source of around 25 ns duration with which to freeze droplet motion on film; however, background 'noise' on the photographs from the coherent laser proved to be a limitation. Also mentioned in his review was use of a Quantimet Image Analyser on spray photographs, the problem being stressed of reliance on operator judgement as to which droplets are in focus. Another constraint in attempting to size droplets off photographs is avoidance of situations where spray density is such that droplets on the negative overlap, or merge into one another. Image analysis of shadow photographs, the incident light extinction method of Kamimoto et al would probably be unsuited to S.I. fuel systems, as it has been devised with non-evaporating diesel sprays in mind.

More recently, Lo et al reported on their development of a laser imaging technique, building on earlier wet hologram work by Thompson. This utilises a typical laser repetition rate of 30 pulses per second and a ten second recording time for each selected position in the spray. A focus selector devised by Hotham passes only the in-focus images from a vidicon to a TV
monitor, at magnifications of up to 400x. Droplet sizing is performed either visually by comparing a still frame with a screen scale or by Image Analyzer for automatic particle counts and production of size distributions. In addition to particle sizing capability, Lo et al also state that the system can deduce particle velocity by measuring the separation between double images gained at high laser repetition rates. The technique has been used successfully to investigate irregularities in fuel injector performance at opening and closing, evaluation of design changes aimed at improving atomisation, and to show the pattern of fuel breakup on exit from an injector. A valuable aspect of the method is that the spray can be inspected on screen before deciding to collect data, in contrast to most other methods which do not readily permit the investigator to see directly what is being measured.

Another approach of value in fuel spray studies is that of laser diffraction, much of the initial development being due to Swithenbank et al at the University of Sheffield. In passing through the spray to be measured, a parallel incident laser beam is diffracted by angles dependent on particle size, producing overlapping Fraunhofer diffraction patterns characteristic of each group of sizes. A photo-electric detector is therefore able to measure light intensity at different radii and transmit light energy data to computer for deconvolution to a particle size distribution. As the light energy distribution depends only on the droplet diameter, and not on the spatial position of the droplet in the light beam, the measurements are independent of droplet velocity and position for all practical fuel system mixture velocities (q.v.). It is advantageous for fuel system work that the 9mm diameter beam produces a measurement volume of sufficient size to contain a representative sample of mixture - this would be in contrast to optical particle counters and phase doppler particle analysers where the control volume is formed at the intersection of thin beams. Multiple scattering would be a drawback with dense sprays, such as in diesel injection, but the limiting concentrations are high enough that Fraidl reports no difficulties of this kind with S.I. engine mixtures.

A proprietary instrument arising from this concept is that marketed as the Malvern Droplet and Particle Sizer (see 2.3.1). Azzopardi, working first with Yeoman and later with Negus, tested the system against suspensions of glass spheres in a stirred cell, cross checking both by photography and use of a Coulter counter. In general the agreement of Sauter mean diameters was considered very good, discrepancies all falling in the range 0.7 - 11%, but the 1976 model was unable to produce meaningful results if the particle sizes differed significantly from a Rosin-Rammler distribution. Since that time a 'model-independent' facility has been added, allowing analysis of distributions other than monomodal. SU Fuel Systems reported in 1978 on initial testing with a Malvern instrument during which, in sizing Latex particles, a correlation with Coulter Counter readings of +/- 2.5 μm was obtained.
The mixture preparation of a variety of fuel systems - conventional carburettor, sonic throat carburettor, air blast atomiser and sonicore nozzles - was also measured on a steady-flow blowing rig with SBP3 (a hydrocarbon solvent) as test fluid. A number of other independent assessments of the accuracy of this instrument have also indicated generally satisfactory operation, for example those by Hirleman\textsuperscript{32} and Dodge\textsuperscript{14}.

While the Malvern Particle Sizer appears admirable for droplet sizing work, and is rapid in operation, it tells nothing of vapour concentrations in the spray. Indeed, although not reported in the literature, it would seem plausible that an evaporating spray could produce unwanted refraction effects on the beam. In a laser light extinction innovation by Plimon et al\textsuperscript{62}, probably originating from Chraplyvyy\textsuperscript{11}, it is suggested that hydrocarbon molecules exhibit absorption bands at certain wavelengths in the infrared, and hence that irradiation of a fuel spray may result, respectively, in scattering by the liquid drops and absorption by the vapourized components. Interpretation of the extinction signals would then provide a first indication of the presence of vapour: for example, areas of high concentration would show up as pronounced extinction of the infrared radiation.

1.4.4 Manifold investigations

Past research seems only to have generated a modest amount of literature concerning drop sizes immediately downstream of single-point fuel systems, nor does transport of prepared mixtures during passage through manifolds appear to be covered comprehensively.

There appear to be few if any such references from throttle-body injection systems. However, droplet sizes leaving the throttle plate of an air-valve carburettor have been measured by Finlay et al\textsuperscript{18}, who applied a Malvern particle sizer to a running 1.8 litre test bed engine. Optical access for the laser beam was accomplished via glass windows in a special viewing passage through which the mixture passed on leaving the carburettor, this being sufficiently insulated from engine vibration by flexible connection to the intake manifold as to leave Sizer background readings practically unaffected. However, droplet impaction on the windows did prove troublesome and some useful palliatives were evolved, such as a telescopic 'tuning limb' at the air intake to alter the pressure-time history at the viewing section, and air circulation to deflect droplets away from the windows. Rosin-Rammler characteristic diameter variation with throttle angle was seen to adopt the typical behaviour as mentioned in 1.2.1. These findings are complementary to those of Yeoman and Lightfoot\textsuperscript{81}, who investigated mixture preparation in the manifold of a similar capacity four-cylinder carburetted engine, also using an early version of the Malvern analyser. Modifications to site the detector remotely, providing an optical link through a fibre bundle, were made for reasons of space. Optical access was gained by fitting quartz windows to allow the laser beam to penetrate the
depth of the manifold in the plenum section. At constant speed the results depict a characteristic diameter reduction as manifold depression is increased - at 4000 rev/min this ranges from approximately 80 microns to 11 microns. Alternatively, at constant manifold depression, the characteristic diameter was seen to increase with engine speed and air flowrate, conceivably because the higher air flows tend to re-entrain more wall film liquid as large droplets. However, some of Yeoman and Lightfoot’s distributions appear to have been bi-modal and the Malvern analyser, at its then stage of development, may not have been capable of interpreting these accurately.

One of the more major programs of investigation into manifold processes was carried out by Trayser et al76 for the Environmental Protection Agency in the United States. The theoretical part of their work indicated, not surprisingly, that the most promising approach for improving distribution is to minimise or eliminate the presence of liquid fuel film on inlet manifold walls. However, while the desirability of ultrafine atomization, long passage bend radii and low air velocity were established experimentally, it was concluded that even with droplets as small as 14 microns, significant impaction still occurs due to flow induced turbulence. At 35 μm total impaction would occur at all flowrates during a 90° bend; the recommendations were consequently, and not for the first time, in favour of intake air and manifold surface heating. Demel and Lenz12 were more optimistic, and felt that while improved fuel atomization for engines with central mixture formation might not cure all wall film problems, it would reduce their extent and diminish the heat requirement at the hot spot, to the benefit of volumetric efficiency. Indeed, a reduction of wall film flowrate under cold starting conditions was obtained by Nightingale and Tsatsami56 using an experimental sonic throat carburettor and 127mm long extension tube. The latter, it was believed, allowed sufficient extra time for deceleration of the air stream that fewer droplets impacted on the manifold wall directly opposite the carburettor.

A summary of inlet manifold investigations would be incomplete without brief mention of the mathematical modelling approach, as typified by the joint work of Low47 and Winterbone80. A theoretical model for the flow of droplets entrained in a non-steady air flow has been produced, taking into account heat transfer, evaporation and drag, and using two sets of wave characteristics, with one set each of droplet path and gas path lines. The model has been used to simulate the flow in the intake of a single cylinder four stroke S.I. engine running at 3000 rev/min, yielding the respective velocities of droplets and gas approaching the intake valve, and showing also evaporative cooling of the gas during the period of inlet valve closure. Such a simulation, it was considered, would be a valuable aid in understanding the transportation and evaporation of fuel in intake systems.
1.4.5 Inlet valve and cylinder flows

In both single and multipoint engines, the mixture is invariably conveyed from the manifold to the combustion chamber in a pulsating flow via an intake port and one or more poppet valves. As the raison d'etre of any fuel system must be to present the engine combustion chamber with the most advantageously prepared mixture, fuel droplet sizes in the mixture emerging from the valve curtain area are of great importance. In a running engine this would pose severe accessibility and durability problems for any measurement system, and such an approach appears not to have been attempted; the alternative, of creating a rig simulation in either steady or pulsating flow, is more practical.

Existing literature concentrates largely on flows of air through inlet valves, rather than flows of mixture. Arcoumanis and Whitelaw discuss the two interrelated flow parameters of significance to engine performance - flow capacity and annular velocity distribution - but scarcely mention the importance of mixture preparation. Typical of the published works is the report of Kastner et al, who have studied the influence of valve and valve seat geometry on discharge coefficient with the aid of a steady flow rig. It was found that the discharge coefficient based on valve curtain area was a discontinuous function of the valve lift/diameter ratio, and that the three different flow regimes recognized arose probably from a detachment of flow from the valve head at intermediate lifts, and then, as the valve lift was increased still further, from the seat as well. In a more recent study of steady flow through an axisymmetric port, by Tindal, Cheung and Yianneskis, four different modes of flow were noted over the range of valve lift.

The applicability of steady flow rig testing to the pulsating situation prevailing in the engine has been considered by various investigators, including Kastner, Gerber and their respective co-workers. It is concluded that slight limitations do apply - the changeover between different modes of flow may for instance occur at slightly different valve lifts under running conditions - but that over a normal range of engine speeds the discharge coefficient is comparable in both situations. Gerber et al made a direct comparison between a steady flow rig and a motored test bed diesel engine, analysing turbulence and the swirl and axial mean velocities, and concluded that rig testing did indeed provide a fair prediction of the swirl flow, although it was not quite so satisfactory on axial flow. A qualification added by Vafidis and Whitelaw, was that steady flow tests can only provide an accurate simulation of the valve exit flow under normal engine conditions when the piston is at least half a valve diameter away from the cylinder head. This observation arose from their velocity field measurements on a motored model engine, when it was noted that the velocity distribution at the exit plane of the intake valve was influenced by the pressure field in the cylinder during the early part of induction.
The velocity component measurements of Vafidis and Whitelaw were obtained using laser Doppler anemometry, a technique in wide use for such purposes. Suen and Yianneskis described a typical arrangement for unobtrusive LDA measurements on a motored high speed diesel engine, in which an acrylic cylinder extension was added to give optical access to the swept and clearance volumes. The intersection of two beams from a 2W Argon Ion laser, split and frequency shifted by a rotating diffraction grating, was traversed across these regions to produce reflected bursts of light from seeding particles, which were in turn modulated by a Doppler frequency corresponding to particle velocity. A purpose-built frequency counter processed the Doppler signals and provided an accurate velocity measurement with both good spatial and temporal resolution. A dedicated single-cylinder engine for 'flying' LDA work, in which the measurement point is moved through the flow to freeze the turbulence structure, has been developed by Ricardo Consulting Engineers plc and is reported by Glover. A software package displays the results as a computer animation, looking down the cylinder towards the valves, and this appears to allow easy interpretation of velocity magnitude and direction, and also of turbulence in the two measurement directions.

1.4.6 Topics for continuing investigation

It emerges during this survey that the desirability of good mixture preparation in S.I. engines is universally acknowledged, and that the pace of research is increasingly being forced by demanding fuel economy and emissions considerations. There does, however, appear to be a paucity of information on certain fairly basic aspects of multipoint injection, and the increased usage of such fuel systems suggests that these areas could prove worthwhile for further investigation.

Initially, some independent drop sizing studies on multipoint injector sprays were felt to be of interest, as existing data originates mainly from manufacturers and reports such as those of Greiner et al, and even from these it is apparent that the typical standard of preparation leaves much to be desired. The major concern is then the quality of multipoint mixture preparation actually achieved downstream of the inlet valve, and how this may be improved. Relevant considerations would be little understood parameters such as injector location in the port, injection timing relative to valve position and injector spray pattern. It would also be important to validate on a representative running engine any conclusions reached from rig testing or mathematical modelling.
Fig 1.1  The simple carburettor: schematic to show basic principle

Fig 1.2  Response of carburettor droplet size to manifold depression (cf Ref. 18)
Fig 1.3  Equivalence ratio requirements at various conditions (cf Ref. 43)

Fig 1.4  Multipoint fuel injection system (Bosch LE-Jetronic)
Fig 1.5 Single-point fuel injection system (Bosch Mono-Jetronic)

Fig 1.6 Cross-section through Bosch multipoint fuel injector
Fig 1.7 Typical response of exhaust emissions to equivalence ratio ($\lambda$)
CHAPTER 2.0

TEST ENVIRONMENT AND EQUIPMENT
CHAPTER 2.0

TEST ENVIRONMENT AND EQUIPMENT

2.1 FUEL SYSTEMS TEST FACILITY

This investigation has been possible as a direct result of the setting up in 1987 of an SERC funded 'Fuel Systems Test Facility' (Fig. 2.1 & Plate 1†) in the Department of Mechanical Engineering at University College London. The concept was to provide a controlled environment in which fuel systems could safely be rig tested on representative fuels. Advantages of the air flow rig approach include the absence of engine maintenance and warm-up, and reduced day-to-day performance variations. Measurement apparatus is also likely to function more accurately in these surroundings.

Suction is applied to fuel systems under test by a Hick Hargreaves CHR 1200 liquid ring vacuum pump in which water is the service liquid. A large plenum (Plate 2) serves as a mixing chamber for the dilution of the air/fuel mixture with additional air immediately after it leaves the fuel system, ensuring that the overall mixture is then well below the lower explosive limit, as well as smoothing the vacuum supply. It also incorporates a bursting disk of rubberised fabric in one end, facing away from personnel, and an adjustable safety valve normally set to limit the vacuum to approximately 50 cmHg (20" Hg), which has proved ample for simulation of all throttled conditions. Air supply to the fuel system is controlled by an air-spill valve in the plenum chamber, in conjunction with a second throttle in the supply pipe.

The internal dimensions of the test cell itself are 4.25m x 2.1m x 2.1m. An air-conditioning system allows a consistent ambient temperature to be maintained within. If desired, this can be at 5°C below the outside laboratory temperature of 20°C. Various safety precautions have been embodied:- for fire resistance the room is lined throughout with stainless steel sheet, three hinged explosion vents are fitted in the roof, the ventilation system has dampers on its inlet and outlet ducts with fusible links for automatic closure in the event of fire, all mains electrical switches and connections are sited on an external wall, and a Chubb flammable gas detection system is fitted with two HC level sensors in the test cell and another in the exhaust duct. The system is calibrated for a mixture of air and butane such that an alarm bell will ring if the mixture in the cell reaches 15% (or in the case of the exhaust 30%) of the flammability limit. An extraction fan ensures that the exhaust system is always below atmospheric

† Plates at end of Chapter
pressure. Two separation chambers are included in a ventilated plant room beneath the test cell, the first to separate liquids from gases and the second a settling tank to separate any liquid hydrocarbons from the water.

2.2 FUEL AND AIRFLOW MEASUREMENT

An AVL Dynamic Fuel Balance and Fuel Calculator are provided in the Test Facility for fuel flowrate measurement. Deduction of instantaneous gravimetric fuel flow rate is made from the rate of change in fuel mass \((\frac{dm}{dt})\) measured by a capacitive displacement transmitter, fuel being consumed from an integral supply tank supported frictionless on a blade spring. Measurements can alternatively be averages over a particular time interval, which may range from 5s to 250s as selected. Conversion to volumetric values is elementary, given knowledge of fuel density.

When fuel injection systems are under test, it is practical to transfer across to a rig situation the values of fuel pulsewidth previously interrogated from the ECU of a running test bed engine at the condition of interest. Subject to the pressure drop across the injectors being the same in both cases, and to identical fuel calibration slopes, this method is the ideal; in such circumstances the AVL system serves mainly to provide a cross-check.

Available options for measuring airflow rate were a standard flow nozzle to BS1042, a hot wire anemometer of the type used as an air meter on a fuel injected engine, or Flowrator meters of various ranges. Check-calibrations against the latter (taken to be standard instruments) were made by flow-testing the others in series with it, and arranging for the depression at entry or exit of the nozzle, if applicable, to be the same as in the specific test situation. The airflow rate associated with some of the work involving single cylinders was so low as to preclude use of the standard hot wire anemometer. In these cases the main passage through the meter was sealed in order to direct the entire flow through the bypass section containing the resistive element. By this means sensitivity was regained, and the anemometer output recalibrated against a Flowrator with low range float.

2.3 DROPLET SIZE MEASUREMENT

2.3.1 Malvern Droplet and Particle Size Analyser principles

In its physical construction the Malvern Analyser consists of two main units mounted on a rigid base, one a transmitter, containing a 2 mW helium-neon laser of wavelength 632.8 nm,
the other a receiver; both units contain lens systems. In the transmitter this takes the form of a beam expander and 35 μm pinhole spatial filter which together produce a collimated 9 mm diameter beam into which the spray is directed during a measurement. The introduction of droplets into the beam causes diffraction through angles dependent on droplet diameter (Fig. 2.2), the scattering occurring mainly in the forward direction due to the intense diffraction lobe close to the axis of the beam direction. A Fourier transform lens in the receiver unit forms the far field diffraction pattern of this scattered light at its focal plane, meaning that independent of the position of a droplet in the beam, and regardless of whether it is moving, the diffraction pattern is stationary and centred on the optical axis. Parallel light always focusses on the axis and any given conical diffraction angle always results in the same radial displacement in the focal plane (Fig. 2.3). It is therefore at the focal plane that a purpose-built, multi-element detector is mounted, this being carried on rails with detents to allow exact positioning at the focal length of the particular lens in use.

The output of each of the thirty one concentric detector elements hence reflects the summation of the intensity from all particles within a given size band, and after mathematical transformation the sum of all such results describes the droplet size distribution. Perhaps surprisingly, the detector array need in fact only be semi-circular (Fig. 2.4), as spherical droplets yield conical diffraction patterns which extend both above and below the beam axis. The structure of one of the cylinder head test rigs unavoidably masked a portion of the scattered light, but using knowledge of the form of the detector it was straightforward to configure the experiment so that this blocked light would not in any case have contributed to the measurement.

All elements of the display are scanned simultaneously during a measurement to capture a 10 μs 'snapshot' of the light energy data, these voltages being retained in a bank of 'sample-and-hold' amplifiers before conversion to machine readable form by an 11 bit AD convertor. In practice, a number of measurement sweeps are made of the detector readings in order to build up a representative average, and accumulated background data is subtracted from this sample data to give the data set used as the basis of calculations in the associated Olivetti M24 microcomputer.

The reliability of the results obtained depends on the quantity of droplets present in the measuring volume. If there are too few, statistical and signal-to-noise ratio related problems may occur. Conversely, multiple scattering due to an excessive spray density would result in a systematic bias towards smaller average drop sizes, because the net scattering angle is increased if a ray encounters a further droplet having previously been scattered.
At the centre of the detector is a small hole through which the centre of the laser beam will pass if unscattered. An auxiliary detector on the optical axis is thus able to indicate the attenuation caused by placing a sample in the beam, expressed as an obscuration:

\[
\text{Obscuration} = 1 - \left( \frac{\text{Light intensity with droplets present}}{\text{Light intensity in the absence of droplets}} \right)
\]  

(4)

Obscurations in the range 5 - 50% are generally indicative of acceptable spray densities, although the UCL Analyser is equipped with an optional high gain mode which permits measurements with only 1%, if circumstances make this unavoidable.

Some optical considerations specific to the instrument were of relevance in test rig design. In particular, the range lens focal length creates a constraint on measurement location: samples must be placed within one focal length of this lens to avoid the phenomenon of 'vignetting', in which the scattering angle of some of the diffracted light (maximum 11°, for 1.2 μm particles) would fall outside the lens aperture. The overall measurement range for the Analyser with standard bed is 1.2 - 564 μm, achieved in three stages by use of a set of lenses with overlapping individual ranges to cover the complete dynamic range. For the droplet sizes encountered in gasoline fuel systems research it has been found that the 100 mm and 300 mm focal length lenses are most appropriate, with their respective ranges of 1.9 - 188 μm and 5.8 - 564 μm. A software enhancement issued since the work under review now enables 'blending' of like results taken with different focal length lenses, allowing presentation of droplet sizes outside the span of a single lens. Because diffraction patterns are only formed when the droplet diameter exceeds the He-Ne laser wavelength (0.6328 μm), droplets of 1 μm diameter are realistically the smallest measurable, although for S.I. engine fuel systems this limitation is unlikely to be a concern.

One further requirement is for a stable ambient lighting condition, in order that previous background readings can be meaningful. Overhead fluorescent lighting had to be excluded by a cover whilst working in the open laboratory area, but within the Fuel Systems Test Facility an alternative low power lamp was situated remote from the measurement area, and proved to have no effect on background readings.

2.3.2 Malvern Analyser calculation methods

At the detector plane the diffracted light appears as a series of concentric alternating light and dark rings, which for a group of droplets is uniquely related to their size distribution. The

- 40 -
diameter of the diffraction patterns is inversely proportional to the droplet diameter. For small forward angles, the intensity distribution \( I(P) \) in the focal plane, is given by:

\[
I(P) = I_0 \left[ \frac{2 J_1(ka\omega)}{ka\omega} \right]^2
\]

(5)

where:

- \( I_0 \) = intensity at centre of diffraction pattern (=ED/\( \lambda^2 \))
- \( k = 2\pi/\lambda \)
- \( a \) = droplet radius
- \( \omega = s/f \) = radial displacement in focal plane + focal length
- \( E \) = incident energy on droplet
- \( D = \pi a^2 \) [projected area of droplet]
- \( \{ J_n(x) \} \) notation for Bessel function of order \( n \)

A central maximum and oscillations of rapidly decreasing amplitude with increasing radius are indicated by this expression, associated maxima and minima producing light and dark rings respectively. The developers of the Malvern instrument have found it easier in practice to infer droplet size from the light energy distribution, working from an expression for the light energy \( L \) within any ring in the focal plane:

\[
L_{s_1, s_2} = E \left[ J_0^2 \left( \frac{kas_1}{f} \right) + J_1^2 \left( \frac{kas_1}{f} \right) - J_0^2 \left( \frac{kas_2}{f} \right) - J_1^2 \left( \frac{kas_2}{f} \right) \right]
\]

(6)

\( ... s_1 \) and \( s_2 \) being radii defining the ring.

Alternatively, as \( E \) is the energy incident on a droplet, and if \( W \) be the weight of all such droplets, then the light energy at any ring is the summation of contributions from all individual droplets of whatever size:

\[
L_{s_1, s_2} = C \sum_{i=0}^{M} \frac{W_i}{a_i^2} \left[ \left( \frac{J_0^2 + J_1^2}{s_1} \right) - \left( \frac{J_0^2 + J_1^2}{s_2} \right) \right]
\]

(7)

\( ... \) where the summation applies over \( M \) size groupings, and the constant \( C \) depends on laser power and optical efficiency.

Notwithstanding the fact that a droplet diffracts light to all radii, for each droplet size the energy distribution curve peaks at one particular radius. Each ring is thus most sensitive to
droplet sizes which yield peaks falling within its defining radii. For light of He/Ne wavelength it can be shown that the first maximum of the energy distribution occurs at:

\[
\left( \frac{\sigma}{f} \right)_m = \frac{0.1385}{a}
\]

This can be seen clearly in the dimensionless energy distribution graph of Fig. 2.5.

As there are 31 rings, and each is most sensitive to one particular droplet size, the derived size distribution is classified into corresponding groups. Constant resolution across the measurement distribution has been achieved by increasing the size bands in geometric progression, such that the division of band width by mean size of band remains constant; however in the upper size ranges this does imply extremely generous band widths, of which the 87.2 - 188 μm top bin (when using the 100 mm range lens) is typical. This would yield poor size resolution if the particular spray happened to have a high proportion in that bin.

The computer deduces the volume size distribution that gives rise to the observed scattering characteristics by a process of least squares fitting of the theoretical scattering characteristics to the observed data. In this work the 'model independent' mode was invariably used, allowing the best fit result to be obtained without prior assumption of a form of size distribution. This best allows the characterisation of multi-modal distributions with high resolution; the precise inversion algorithm used to obtain the particle size distribution is, however, known only to the manufacturers. Reference should be made to Appendix II for details of some calculation-related queries arising during the work. Indeed, discussions between Malvern Instruments and UCL resulted directly in software and documentation revisions of general benefit to all users.

A result is conveyed in four main ways by the standard Malvern presentation:

1) A histogram of percentage of spray by volume in a range of droplet size bands
2) A cumulative plot of percentage of spray by volume against droplet size.
3) A tabulation of percentage of droplets by volume in each size band.
4) A series of characteristic derived diameters describing the spray.

2.3.3 Interpretation of a Malvern measurement

A line-of-sight average is analysed by this instrument, the field of view consisting of a volume defined by the intersection of 9 mm diameter laser beam and spray. For the case of a
pintle-type injector spraying across the optical axis into free air, the content of this volume would depend on the distance of the laser beam from the pintle. At very near distances the beam could be intersecting a conical liquid sheet rather than resolving separate drops (a situation which would probably reflect in a high obscuration) whereas in the breakup region and beyond, the theoretical sampling volume comprises the two intersections between the circular beam and a droplet spray conforming essentially to the surface of a hollow cone (Fig. 2.6). In practice the inside of the spray cone could well contain some spray, due to the collective effects of air motion, imperfect pintle centreing, gravity and random fluctuations in the breakup process, but it is instructive to make use of the hollow cone model as a conservative guide to the number of droplets actually sampled:

If \( t_p \) is the time that the entire pulse of spray takes to cross the optical axis \( x-x \) (sensibly approximating to the electrical pulsewidth), and \( t_s \) be the detector sampling time, then the proportion of the injected fuel involved in a Malvern measurement sweep is:

\[
\frac{2 \times \left( \pi r^2 \right)}{\left[ 2 \pi \left( 1 + \frac{\alpha}{\tan \alpha} \right) \right] \times \left( 2r \right)} \times \frac{t_s}{t_p}
\]

(9)

Of these terms, \( t_s \) and \( r \) are fixed (10 \( \mu s \) and 4.5 mm), while the variables may be substituted with representative values. The appearance of \( t_p \) in the denominator indicates that electrical pulsewidth does not in fact affect the number of droplets sampled: were it to be increased (reducing the value of Eq. 9) then the reduced proportion applied to the greater quantity of fuel injected during the longer pulse would still give the same result. In the example of Fig. 2.7, based on data from an idling 1800 cc Ford Zeta DOHC engine with 15° spray cone injectors (\( \alpha = 7.5^\circ \)), the number of drops is derived for two representative sampling distances \( l \) between pintle and beam.

Malvern Instruments Ltd. state somewhat vaguely that measurement of between 100 and 10,000 droplets is necessary to characterise a spray. From Fig. 2.7, the number of droplets actually sampled in one sweep of a conical spray would appear to be dependent strongly on Smd, but rather less so on distance of the measurement volume from the injector pintle. Even the pessimistic case of the hollow cone spray easily exceeds the minimum required number of droplets in just the one sweep of the detectors, given an Smd of around 40 \( \mu m \) or less, but for an Smd of 80 \( \mu m \) only 17 droplets might be sampled when \( l=150 \) mm, implying a need for at least 6 sweeps. However, for a scan in the manifold or port it is expected that the number of droplets sampled by the beam would be higher, as the more dispersed spray is likely to encounter a greater length of laser beam. As many sweeps as possible were generally taken, consistent
with the time involved in pausing to allow a previous spray to clear, and in some cases the necessity to avoid excessive wetting of windows or lenses by spray.

The instrument response is proportional to the spatial frequency (drops per unit distance) in the measuring volume, and hence produces a spatial average. Should droplet velocity be size dependent, this might be thought likely to cause a difference between the temporal and spatial averages, and thus between measured and true droplet size distributions. Fortunately, the 10 μs per sweep sampling time of the Malvern Analyser is so fast that a 'velocity biassing' effect seems highly improbable, given that for the near absurd case of two droplets, one stationary and the other moving with sonic velocity, the distance between them would change by only 3.3 mm in 10 μs.

2.3.4 Analyser validation

It is claimed by the manufacturers that the instrument calibration is inherent by virtue of the basic diffraction principles involved. Resort to a reference standard should not be necessary, and nor are any adjustments possible, although it is noted that a reference reticule is offered as an accessory, presumably to satisfy the user of continued accuracy should components be disturbed then reassembled. Malvern quote an accuracy of ±4% on volume mean diameter, as checked by such a reticule technique. Hirleman and Dodge have shown that six different Malvern instruments can measure Smd of a distribution of dots on a reference reticule with a maximum error (20 sweeps) of 2.7% and a total spread of 5.4%.

Some limited cross-checks were carried out on instrument accuracy, involving 30,000 sweeps of reference particle suspensions contained within a PS1 small volume stirred cell. As in an earlier SU Fuel Systems investigation, DVB latex particles previously sized by Coulter Counter were employed, being nominally of 9.5 microns and 40.8 microns diameter. An alumina catalyst was also sieved to produce standard cuts of 45-53 micron and 75-90 micron. Reasonable agreement was found between the nominal latex particle sizes and the Analyser readings, as may be judged from the results in Appendix III. The displayed Smd for the alumina particles is also consistent, to the extent that Smd values correspond to the upper sieve size, although the reason for broadening of the result histogram beyond the sieve limits was not established.

2.3.5 Effect of evaporation on laser diffraction droplet sizing

A possible difficulty in sizing volatile fuel sprays with the Malvern instrument has received only a passing reference in the literature. This concerns the effect of evaporation; it has been found during the present work that the refractive index gradient existing in a rapidly evaporating spray can cause refraction of the laser beam, which in turn registers a signal on the innermost rings of the Malvern detector plate. The instrument is designed to look for
diffracted light signals, so it naturally interprets the refracted light as diffracted light, which leads to an indication of large droplets being present whether or not they are actually there. An illustration of this effect is provided by the results in Fig. 2.8, which depict free air sprays (q.v.) from a Bosch injector with fuels of differing volatility. The three fuels involved, in descending order of volatility, are: winter and summer gasolines (to BS 4040), and white spirit. The group of three plots to the top of the figure depicts the droplet size distributions at the beginning of a spray, while those beneath show the corresponding distributions towards the end. It may be observed that the tail end of spray distributions for both gasolines indicate the presence of droplets with diameters of the order of 400-500 μm, most conspicuously so for the volatile winter grade gasoline, while there is no indication of large droplets for the relatively involatile white spirit. Although there is some difference between the relevant physical properties of the 'fuels', it seems unlikely that large droplets were actually present. Subsequent photography, and scaling off the projected transparencies, showed that such large droplets were not to be found. Thus, the Malvern Analyser has to be used with care in sizing volatile sprays, and it was the practice during this work to check results by photography, or by observing the effect of using a less volatile fuel in the same circumstances, whenever significant light was detected by the inner two rings of the detector. In some circumstances it was also found advantageous to use the air conditioning system to reduce the ambient Test Facility temperature to the minimum attainable.

2.3.6 Use of Sauter mean diameter

As fuel sprays invariably consist of droplets with a range of diameters, various forms of derived mean diameter are often used to summarise sprays for convenient comparison. The most commonly used mean diameter in the i.e. engine context - and one quoted in the Malvern presentation of results - is the Sauter mean diameter (Smd), which represents the droplet diameter of a notional monosized spray with the same overall volume as the actual spray, and also with the same overall surface area.

In mathematical terms:-

\[
\text{Smd} = \frac{\sum_{j=1}^{i} N_j D_j^3}{\sum_{j=1}^{i} N_j D_j^2}
\]

(10)

where the droplets are divided into i size bands, and there are \( N_j \) drops of mean diameter \( D_j \) in the \( j \)th size band. The rate of evaporation of a spray is of prime interest to i.e. engine researchers, and Smd would therefore appear especially relevant because rate of evaporation is a function of surface area.
However, for a spray where the majority (by volume) of droplets are quite large, it has become apparent that the presence of a minority (by volume) of very small droplets will greatly reduce the Smd of the spray. It is a physical fact that small droplets will have a large surface area and the value of Smd would correctly reflect this. This is illustrated in Fig. 2.9, where data relating to the inset fuel spray distribution has been modified by introducing into each size band in turn a spurious signal representing the presence of 1% (by volume) of fuel droplets. The plot shows that the effect is almost negligible when introduced into the higher bands but dramatic when introduced into the smaller size bands. Certain of the sprays encountered during this work have had indications of very small droplets whose presence can neither be confirmed or discounted as they are so difficult to check by photography. However, supposing an instrument were to be used to measure the size distribution of a spray in terms of proportions by volume, and further supposing that this instrument was susceptible to some form of spurious signal, then if the spurious signal appeared in a small size band it would have a very powerful effect on the indicated value of Smd. For this reason, work subsequent to that presently described has concentrated on a new approach to specifying mean spray size which is both relevant to evaporation and insensitive to spurious signals.
Fig 2.1 General arrangement of Fuel Systems Test Facility
Particles A and B are same size, thus their diffraction angles $\phi$ are the same.

Light diffracted by particles of a given size at any position in space always falls on the detector at the same radius.
Fig 2.5
Diffracted light energy distribution in the focal plane (cf Ref. 44)

Fig 2.6 Schematic showing intersection between 'hollow cone' injector spray and Analyser laser beam

Fig 2.7
Estimated number of droplets in sampling volume
Fig 2.8
Effect of fuel volatility on Malvern Analyser histograms

Fig 2.9
Sensitivity of Sauter mean diameter to small droplets
Plate 1  UCL 'Fuel Systems Test Facility'

Plate 2  Plenum chamber for vacuum supply
CHAPTER 3.0

EXPERIMENTAL WORK -
MIXTURE PREPARATION
UPSTREAM OF VALVE
CHAPTER 3.0
EXPERIMENTAL WORK -
MIXTURE PREPARATION UPSTREAM OF VALVE

3.1 MULTIPOINT INJECTOR SPRAYS INTO FREE AIR

3.1.1 Characteristics of injector mixture preparation

Several distinct shapes of fuel spray are available within the Bosch range of multipoint fuel injectors: conical spray with needle valve, pencil beam, conical spray with multihole orifice plate, and a special two spray type designed specifically to target both inlet valves of a 4-valve engine. In this initial work the first mentioned was tested, being the type used on Ford V6 engines fitted to the Scorpio and Granada models. Bosch claim that the sharp outlet edge at the end of the machined conical shape in the valve body promotes good fuel atomisation, so assisting the basic objective of distributing the fuel evenly into the air stream.

Fuel issuing from such an injector encounters the surrounding air with a relative velocity which tears the ligaments of fuel apart to form droplets. These are then encouraged by surface tension to become spheres, as such a form has the minimum surface energy. Continuing break-up into smaller droplets is likely to occur so long as any remaining relative velocity exists to exert aerodynamic forces on the droplets, provided that this aerodynamic destabilizing force sufficiently exceeds the consolidating influence of surface tension. Due to this relatively random method of droplet formation, a considerable spread of droplet size is normal, although repetitive measurement would be expected to yield a repeatable mean distribution.

Discussions with a representative of Robert Bosch GmbH have suggested that differences in mixture preparation across the duration of the spray pulse are to be expected for two main reasons. Larger droplets during injector opening may be due to a pressure drop inside the nozzle during acceleration of the fuel, and characteristically smaller droplets at the end of the pulse can sensibly be attributed to the smaller gap as the needle closes.

3.1.2 Arrangement of experiment

Initially it was considered logical to examine the mixture preparation produced by multipoint fuel injectors spraying into free air, in order to provide a baseline for later assessments of in-cylinder mixture preparation and a check on manufacturer's data. A first order appreciation was also gained of certain other relevant factors, such as fuel supply pressure, injector-to-injector variation, and, significantly, fuel volatility, as has been
mentioned (Para 2.3.5) in connection with unwanted refraction effects and consequent spurious components at the large end of the droplet size distribution.

Measurements of the spray size distribution produced by a Bosch EV1.3A (type 0 280 150 727) multipoint injector were accomplished by arranging for the injector to spray horizontally into the laser beam from a distance of 125 mm, corresponding to the distance in Ford V6 engines from injector tip to the inlet valve. Mounting of the injector was such that its axis and that of the laser beam were perpendicular, with the centreline of the injector at the same level as the optical axis and within the lens focal length. The sample measurement volume can therefore be defined as the intersection of the 9 mm diameter beam and the spray; with only 10μs required for each measurement sweep the results obtained are the mean of 'snapshots' of the droplet size distribution present in this volume at the time of measurement. To ensure coverage of the anticipated large range of droplet size, the 300 mm range lens was used on the Malvern Analyser. Plate 3 indicates the general layout.

Injector triggering signals came either from a Bosch manual control unit (Plate 4) or directly from the switched 12V/2.5A output terminals of the Malvern Analyser Spray Synchroniser unit, which, coincidentally, provided a similar square wave pulse. The manual control unit is designed to supply representative square wave trigger signals for the experimental operation of electronically controlled injectors if triggering from an ECU is not practicable. An end-stage appropriate to the particular type of injector under test must be selected and fitted into the basic control stage. Even when using the Bosch unit, the Spray Synchroniser was retained in its designed role for predetermining a time delay between spray triggering and the instant of measurement. Having established by trial the delay time necessary for the fuel spray to reach the laser beam, as inferred by first occurrence of a positive obscuration, any temporal variations in the spray quality can then be followed by re-testing with progressively longer millisecond delays. In all cases where the Bosch unit was employed the basic combined time for electrical delays and spray travel was around 17 ms; when triggering direct from the spray synchroniser the figure was 8 ms. Injection pulsewidth was set at 8.2 ms, known to be a representative light load value from test bed experience of a running V6 engine.

The fuel system was pressurised by a standard electric roller-cell fuel pump as removed from the Scorpio vehicle, operating through a filter and 3.45 bar (50 psi) pressure regulator, and supplied from a header tank in a continuously circulating system. Fuels used were variously BS4040 winter and summer gasolines, a BP 'additive free base super petrol', white spirit, and SBP3, the latter being a hydrocarbon solvent less volatile than gasoline but which corresponds closely in those physical properties relevant to drop formation and initial droplet diameter.
3.1.3 Typical free air spray results

For these early measurements it was customary to record only ten sweeps, given that some seconds were required for the previous spray to clear and that no fume extraction had been provided for the fuel vapour. Retrospectively, and having undertaken the calculations described in Para 2.3.3, this number is felt to have been on the marginal side. Even so, monitoring of the screen display of detector output for each sweep generally showed fair uniformity between successive sweeps, although the first was invariably different. While it was quite plausible that this genuinely reflected some peculiarity of the first spray of the injector, the matter was raised with Malvern Instruments and proved to be simply the effect of the initial screen display not being scaled to fit the available space. A more serious penalty of performing this free air spray work at an early stage of UCL's acquaintance with the Malvern Analyser was that the version M4.1 software irregularities described in Appendix II had yet to be discovered. Interpretation of the results must therefore rely on comparison of histograms, rather than reported Sauter mean diameters.

Results for a spray of 'additive free base super petrol' are shown in Fig. 3.1, with delay times ranging from 17.5 ms (leading edge of spray) to 25 ms, which should more than cover the lifetime of an 8.2 ms duration pulse. At the front edge the distribution spans 150 - 550 μm and is centred on 300 μm. Only 0.5 ms later the distribution has shifted noticeably towards smaller sizes to centre on 200 μm and cover the range 100 - 425 μm, behaviour which is consistent with the Bosch hypothesis of larger droplets during pintle opening. By 20 ms delay the primary peak is situated at the further reduced value of 100 μm, but a secondary tendency towards apparent droplets of the order of 500 μm has developed. This latter is symptomatic of the unwanted refraction effects described in Para 2.3.5, and later confirmed (q.v.) with simultaneous photography. At 25 ms (trailing edge of spray) the situation has worsened sufficiently that the evaporative peak dominates the distribution, while the genuine data is predominantly around 80 μm. Evidently some kind of stratification effect could contribute to the tendency to finer droplets as 10% obscuration still prevailed even at 30 ms. Droplets from earlier in the spray pulse which have slowed down and continue to be resident in the measurement volume would obviously be likely to reduce in diameter by evaporation, although such an effect could equally reflect in spurious indications as mentioned.

For comparison with the above, white spirit was substituted as a test fuel in an attempt to eliminate evaporation effects from the measurement. The volatility of white spirit is considerably lower even than that of a summer grade gasoline, while it remains similar in the essential characteristics of relative density, surface tension and viscosity. At the leading edge of the white spirit spray (Fig. 3.2), again with a delay time of 17.5 ms, the obscuration of
6.03% is remarkably comparable to that previously for gasoline (7.04%). The droplet size distribution itself is also little changed, although that for the white spirit has a slight bias towards relatively smaller droplets. At 18 ms there are some distinctly smaller droplets (~ 80 μm) and the distribution between size bands is slightly more even than with gasoline. By 20 ms there are more sub 100 μm droplets but the distribution has become curiously bi-modal, with peaks at around 150 μm and 350 μm; this is not thought indicative of an evaporation effect as subsequent longer delays reveal return to a monomodal distribution peaking at around 90 μm and spread widely from 50 μm to 564 μm. An insignificant minority of droplets also show in the range 25 - 40 μm.

The inescapable conclusion of these and numerous supporting tests is that distinctly large fuel droplets are produced by this type of injector. While a diameter in the region of 90 μm tends to predominate, all higher bin sizes invariably have significant content. These observations are consistent with droplet size spectra shown by Fraidl\(^{20}\) for the case of a shorter (2.2 ms) pulselength and closer measuring distance of 50 mm. In this multipoint case a small droplet size might at first appear less crucial than for a single-point system, since MPI assures good geometric distribution of fuel between manifold branches and reduces the path lengths susceptible to wall-wetting. However, droplets of this size arriving in the combustion chamber cannot lend themselves to rapid evaporation and fast burning, even assuming they are able to negotiate the port and inlet valve without impaction.

3.1.4 Injector-to-injector variation

Two more injectors of nominally the same type as the ex-Scorpio example were also tested in further work with the same experimental arrangement. One of these injectors, identified as 'Scorpio 2', originated from the same vehicle whereas the other had been removed from a similar engine installed in a Ford Sierra Estate; unfortunately no more detailed history of previous running time or conditions was available.

Comparative tests were carried out using summer grade pump petrol and with the injector triggered directly by the Spray Synchroniser, thus altering the timescale of the required measuring delay as compared with results discussed previously. Two different delay times were used: 10 ms, corresponding to a 'snapshot' approximately one quarter way into the spray pulse, and 14 ms, three quarters of the way through the lifetime of the pulse. In so far as the 10 ms distributions (Fig. 3.3) centre on 200 μm, and are monomodal, all the injectors behave similarly at the lower delay. However the Sierra Estate injector produces a distribution which is spread considerably more evenly, and over the relatively larger range of 70 - 500 μm. Little distinction can be drawn between the spray quality of the three injectors at 14 ms delay (Fig. 3.4). Although all measurements suffer similarly from a false peak at large droplet sizes
they also indicate genuine droplets in the band 55 - 200 μm, within which range the dominant size is of the order of 125 μm. For the 'Scorpio 2' injector a tail end of fine sizes down to 30 μm is just evident, but accounts for only a negligible proportion of the total fuel volume.

On the basis of this admittedly limited comparison it is felt that the original ex-Scorpio injector produces droplet size characteristics adequately representative of its type. This particular example was therefore identified and retained in all subsequent rig testing involving Ford V6 engine hardware.

3.1.5 Spray photography

While the Malvern Analyser undoubtedly samples a larger and more representative volume of spray in any one measurement than do many rival instruments, it does not readily provide the user with an overall picture of the form of the spray. It can be of value to complement laser diffraction sizing with photography, in order to gain an impression of spray cone angle and areas of droplet concentration. This in turn helps identify areas where detail particle size analysis would be of particular interest.

Some photography of free air multipoint injector sprays of gasoline was undertaken in conjunction with a final year project by Qureshi and serves to illustrate the nature of an injector spray uninfluenced by air flow: results from an 8.2 ms pulsewidth spray appear as Plates 5 and 6. The exposure was controlled by the duration of two electronic flashguns mounted side by side 35 cm from the spray and aimed 45° off the spray axis, operating in otherwise complete darkness with the Minolta X-700 camera shutter held open on 'B' setting. A matt black paper background was positioned behind the spray. Kodachrome P1600D or alternatively Ektachrome P800/1600 slide film were used, processed at 1600 ASA. Injector and flashguns alike were switched from the Spray Synchroniser via an interface unit devised at UCL by Orchard, enabling the flash to be triggered at a preset delay after the start of the injector drive signal.

Unfortunately, even the flash duration of approximately 1/3000th of a second was evidently insufficiently short to capture the motion with a sharp image, preventing meaningful scaling of droplet sizes. Paradoxically, the slight 'streaking' of droplets has some virtue as it helps to reveal the instantaneous direction of droplet travel. If it were essential to 'freeze' the droplets a pulsed laser visible light source would no doubt be a more effective means of illumination; according to recent information flash durations with such apparatus can now be as short as 30 ns.

- 57 -
Plate 5 was exposed 6 ms into the pulse and shows an entirely typical injector spray, which takes the form of a narrow cone of around 15° included angle. That considerable momentum is imparted to the injected fuel seems evident from the continued high velocity - as inferred from elongated images - of droplets which have already penetrated significant distances into the air. An implication could be that the spray will tend to travel in very much the direction that it is aimed, perhaps notwithstanding the surrounding airflow, with the obvious danger of spray deposition on port walls, particularly at changes of direction or section. The transition into rather coarse droplets from the dense sheets of liquid fuel leaving the injector is also apparent. However, it is only fair to add that droplets of more desirable smaller sizes could be invisible to an elementary photographic technique with limited resolution.

An abnormality which occurred several times in testing is highlighted by Plate 6. It was found that the initial injection pulses at the beginning of a test session occasionally produced a spray of extremely poor shape, and (on the original projected transparencies) visibly excessive droplet sizes. After approximately five pulses the spray would revert to conical form, presumably as whatever gum or deposit affecting the pintle was flushed out. This lesson was read across to the concurrent Malvern Analyser tests of free air sprays, during which it became the practice to 'clear' the injector by allowing it to pulse repetitively before commencing measurements. Such a problem as witnessed could have adverse effects on the cold start case for which a well distributed spray of fine droplets is arguably most necessary, as the mixture may become over-rich before the spray quality has become conducive to first fire.

3.2 MULTIPOINT INJECTOR SPRAYS IN STEADY AIR FLOW

3.2.1 Experimental arrangement

Following on from investigation of free air sprays, it was decided to examine the characteristics of the same EV1.3A injector when discharging into moving air. A steady airflow was produced using suction from the Test Facility damping chamber. An upstream throttle was provided, in order that the pressure of the airstream could be reduced below atmospheric for certain tests simulating part load operation. Such conditions of reduced air density are relevant because theoretical and empirical studies (Para 1.4.2) suggest that this will result in evolution of relatively larger droplets.

The injector was wire-mounted concentrically within a 45 mm diameter brass tube, which was in turn sited vertically over a viewing passage designed to allow the Malvern laser beam optical access to the fuel spray. A spacing of 125 mm was once again established between the injector tip and the centre of the measuring volume. The viewing passage (Fig. 3.5) consists essentially of a large aluminium block containing two perpendicular passages, respectively for
the mixture and laser beam, with the latter passage sealed at either end by an adjustable glass window. Correct alignment and cleanliness of the latter are prerequisites for sound measurements. Normal procedure was to make final alignment adjustments with the intended test condition vacuum applied to the windows, in order to ensure firm seating of the window carriers in their housings. Shutters incorporated in the viewing passage served to prevent fuel contamination of the windows during set-up periods when laser access was not required.

No attempt was made to devise an injector mounting in any way representative of the V6 engine port: at this stage the objective was to study the injector spray alone, and it was feared that to simulate the port could introduce wall-wetting problems which would disrupt measurement of the basic spray. Indeed, to confirm that this was not the case even with the simple tubular passage, a transparent replica (Plate 7) was produced to enable visual observation of sprays at the anticipated test conditions. For measurement purposes a more robust metal version of identical geometry was used throughout.

Fuel was supplied to the injector from a closed circuit system which included the AVL Fuel Balance and a pressure regulator, the latter to be adjusted as a function of the simulated manifold condition in order to maintain a differential of 3.45 bar across the injector. The high pressure pump was switched on only when necessary, as it was found that the temperature of the relatively small quantity of fuel in circulation otherwise rose appreciably over an extended period. Once again, the Bosch manual control unit in conjunction with the Malvern Spray Synchroniser provided the injector triggering signals, enabling a delay period to be set between injector triggering and Particle Sizer reading. Ten 'sweeps' of the detector elements were made for each reading, involving ten separate injector pulses. At conditions involving low air flow, at least 15s was allowed between 'sweeps' to allow the previous spray to disperse. For ready comparison between different test points, the aim was to take the droplet size distribution measurement at a consistent point in the lifetime of the spray. The delay which gave an obscuration of .01 was determined in each case by trial (with the Analyser operating in High Gain mode) and taken to be the delay necessary to sample the leading edge of the droplet spray. By adding on half of the spray duration the actual test delay was arrived at. Thus, all readings to be presented relate to the middle part of the injector spray.

### 3.2.2 Results at various air flow rates

Three basic conditions were simulated - idle, 1500 rev/min road load and 2000 rev/min with throttle wide open. Some readings were also taken without the vacuum pump in operation, amounting to essentially the same experimental situation as for the initial free air sprays albeit now in the vertical plane. For this test the spray duration was set arbitrarily to the same value as for the case of 1500 rev/min road load. Appendix IV explains the rationale
behind the selections of airflow and depression, for which the premise has been that steady flow tests intended to simulate the induction stroke must utilise an air flow rate four times the actual mean air consumption of the engine, as the induction process occurs in only one quarter of the total time required for a complete cycle.

Some representative histograms for these varied airflows are shown in Fig. 3.6. By now a revised version M6.10 of Malvern software was available, enabling some credence to be given to the displayed Smd values. Although gasoline was used, the combined precautions of a relatively low volatility fuel blend, and reduction of rig intake temperature to around 16.5°C using the Test Facility air conditioning system, appear to have eliminated evaporation problems from the Malvern measurements. The nominally 'free air' droplet size distribution equates well to previous results, tending to confirm that the passage between injector and measurement volume has not altered the spray as compared with the uncontained case.

Once again, the main indications from the 'free air' spray distribution are of droplets in the region of 100 μm, consistent with the reported Smd of 113.8 μm. Surprisingly, the 900 rev/min 'idle' simulation shows a positive improvement in favour of smaller droplet sizes, yielding an Smd of 80.1 μm despite the reduced air density implied by the high depression of 45.7 cmHg. Using an air flow representative of induction at a 1500 rev/min road load condition, and 30 cmHg depression, the trend towards smaller droplets continues, with Smd now down to 55.0 μm and the majority of droplets concentrated in the size range 60 - 80 μm. Contrarily, with the upstream throttle fully open and the airflow over three times higher than at road load the results becomes more akin to the 'free air' case, only with the addition of a minor but not insignificant presence of smaller droplets ranging down to 20 μm. These droplets conceivably arise from some limited breakup in the air flow of larger droplets formed at the injector. It seems probable that the wide open throttle results are worse than those at part load because injection takes place into a much faster moving airstream, reducing the relative velocity between air and fuel; in comparing the two situations, this factor perhaps more than offsets the negative effect of injecting into air of reduced density. Indeed, a calculation of respective fuel and air velocities (based on known flow rates and pulsewidth for the 2000 rev/min wide open throttle condition, and taking internal diameter of the injector mounting port as 44 mm and annular outlet area of the injector to be 0.0908 mm2), yields a fuel velocity of 18.6 m/s and an air velocity of 18.0 m/s, implying an almost negligible relative velocity.

In terms of droplet formation theory such an apparently favourable response of droplet size solely to reduced air density would not be readily explicable. Two special test conditions were arranged, in the hope of clarifying the situation by separating the two influences of port depression and air motion. The injector was operated with the port air maintained
respectively at 30 cmHg and 45.7 cmHg depression, and the upstream throttle sealed to prevent airflow. In both cases the spray duration relating to the 1500 rev/min road load test condition was retained. These distributions (Fig. 3.7) are practically identical both to each other and to those for the quasi 'free air' sprays conducted on the same rig at ambient pressure. The implication may be that the injector is in fact insensitive to manifold vacuum, and that the relatively good spray performance achieved at part load conditions should be attributed instead to the airflow conditions. Admittedly, the displayed values of Smd for the tests into quiescent air at below atmospheric pressure are lower than those for the earlier 'free air' sprays, but this is regarded more as an apt demonstration of the misleading sensitivity of Smd in cases where a mere trace of the total fuel volume exists as small droplets.

The hypothesis that spray characteristics on this rig change between test conditions due to variations in air flow, rather than air density, is perhaps consistent with the fact that control of the former is (in part) by a throttle disc situated upstream of the injector. At the simulated low load conditions this throttle is in an almost closed position, limiting the flow area to two small crescent shaped areas through which air is accelerated to near sonic velocity under the influence of the high pressure drop. A turbulent wake - such as is likely to be created downstream of the disc by interaction of vortices originating from either side - would have obvious potential for redistributing and mixing the droplet spray into the airstream, and perhaps also for breaking down larger droplets, assuming it persists sufficiently far downstream. This is felt to be the likely mechanism by which the 900 rev/min idle simulation has yielded significantly smaller droplets than did injection into quiescent air. At 1500 rev/min road load the performance is further improved, arguably because - with air flow now in excess of 2.5 times greater - the turbulating effect of the still partially closed throttle disc is increased.

3.2.3 Effect of fuel properties

A general comment on spray size distributions presented thus far is that droplet diameters are predominantly of the order of 100 μm, whether the surrounding air be quiescent or flowing. Occasional reductions of Smd in some cases are certainly of interest, although these may not be unrelated to the throttle location on the specific rig, but in no case do they represent a dramatic improvement towards a finer spray. It was decided to make use of the steady flow injector test rig to investigate the sensitivity of droplet size to differing fuel properties (such as might occur within the tolerance and seasonal adjustments provided for by the BS4040 specification) in case initial tests had unwittingly been utilising a gasoline blend with characteristics unsuited to evolution of the finest spray. Various theoretical and empirical equations (Para 1.4.2) are all agreed that fuel surface tension and viscosity are fuel properties directly affecting the process of droplet formation.
Four hydrocarbons were selected on the basis of their relative surface tensions and viscosities (see Appendix V) and used in successive tests at the same test conditions as for the gasoline previously. Although the variation of these properties exceeded that to be expected of proprietary pump gasolines, it was hoped that any trends would be more clearly visible as a result. In every case the modified surface tension and viscosity of the economically available test fluids exceeded that of gasoline, and therefore would be expected to give more adverse results, but it was hoped to gain a feel for the strength of any effect, and possibly to extrapolate an opinion on whether worthwhile smaller droplet sizes might be attainable given more suitable modification of fuel properties. The relevant droplet size histograms, ranked in order of increasing surface tension, appear in Figs. 3.8 - 3.11 and should be compared also with the comparable gasoline results in Fig. 3.6.

It seems reasonable to expect that the 'free air' sprays would provide the most stable indications of any influence on droplet size of the various fuel properties. Ironically, in the case of these sprays no correlation whatsoever would appear to exist between the properties under investigation and the Smd values, indeed the droplet size distributions are virtually indistinguishable from one another. At the simulated 900 rev/min idle condition, however, there are positive indications that increasing surface tension leads to larger droplet sizes, a trend that is only interrupted by the result for toluene. If evaporation rate be directly proportional to surface area then (on this basis alone) a gasoline spray would evaporate about 1.2 times faster than one composed of the same volume of o-xylene. The respective vapour pressures of the fuels should of course be considered in any true comparison of evaporation rates, as volatility also exerts a powerful influence. In terms of viscosity it is possible to pick out individual results suggesting that fuels of relatively high viscosity, such as o-xylene, produce larger droplets, but there seems to be no universal pattern.

With the simulation of 1500 rev/min road load it is evident that the Smd for toluene is much reduced compared with the idle simulation (as was the case earlier for gasoline), n-octane and o-xylene Smd's are little changed, but the value for n-decane is greatly increased; a clear indication is that gasoline performs best with respect to minimum Smd. The worst is n-decane, which has a medium surface tension but high viscosity, and returns a specific surface area around 50% of that for the gasoline. A few minor discrepancies aside, the Smd values increase with surface tension and viscosity for the case of 2000 rev/min wide open throttle. Around 20% improvement in evaporation rate would be expected to result purely from the surface area increase between o-xylene (Smd 100.0 μm) and gasoline (Smd 83.9 μm).
Although some significant variations in droplet size arose as an apparent result of employing fuels of differing surface tension and viscosity, the reliability of the exercise is open to question, bearing in mind that the 'free air' sprays disclosed no such tendencies. Nor was it possible to separate the two effects of surface tension and viscosity in any convincing manner. It can however now be concluded that the typically large droplet diameters recorded with gasoline are a true reflection of the injector performance, not merely a consequence of unsuitable fuel properties. An ideal next step would perhaps have been to test fuels of lower surface tension and viscosity than current gasolines, but it was suspected that any benefits were unlikely to be positive enough to repay such an effort. Alteration of the fuel system itself would appear to be the most pragmatic approach to improvement of spray quality.

3.3 INJECTOR AND CARBURETTOR SPRAY QUALITY COMPARED

3.3.1 Carburettor spray measurements

For comparison with the multipoint injector results, some Malvern Analyser measurements of a typical carburettor spray in steady flow were taken at a similar distance downstream. The Weber twin-choke downdraught carburettor (type 26/28TLDM 15A) was prepared for testing by addition of a spring to bias the primary throttle disc end float in a consistent direction, and also by fitment of an angle indicator to the same throttle spindle; a 14V supply was connected to hold the idle cut-off solenoid valve at running position. As this work sought to record atomisation behaviour of a typical throttle disc, rather than to perform a full assessment of this particular carburettor, it was considered necessary only to look at the primary choke. The carburettor (Plate 8) was therefore bolted via an adaptor plate to the same viewing passage as before, with the primary centred over the passage bore; the distance between throttle disc and measurement volume was then approximately 140 mm. A tapping into the viewing passage was connected to a vertical mercury manometer for readings of "manifold" depression.

Malvern Analyser measurements were made at six different test points covering a range of throttle angles and simulated manifold depressions, the derivations of which were based on a 1300cc engine capacity (for which the carburettor is designed) and are explained fully in Appendix VI. Essentially these comprise conditions of idle, low load, full throttle, some further intermediate combinations, and finally a test with an excessive air flow. The results presented are in general an average of 25 'sweeps' of the Malvern detector elements, the exception being Condition 4 with which persistent window-wetting difficulties were encountered, presumably resulting from spray deflection by the throttle disc. In this case the number of 'sweeps' was limited to ten, and at times five, to keep the shutters open period short enough to prevent the detector background reading becoming invalidated by contaminated viewing windows. With this precaution, test-to-test repeatability became as good for
Condition 4 as for the other conditions, judged by repeat tests yielding Sauter mean diameters within 5% of the first values.

The results histograms [Fig. 3.12] clearly illustrate classic carburettor droplet size behaviour as a function of engine load. At the 750 rev/min idling condition 1, during which the throttle disc was only 1.2° open, the spray quality is excellent with a majority of the droplets being well below 10 μm diameter. The very low reported Smd of 3.5 μm is in this case a fair reflection of the destiny of much of the fuel. Opening the throttle to 14.5° for Condition 2, a simulation of 1500 rev/min road load, produces a noticeable bias towards slightly larger droplets although the overall range of sizes is comparable to idle and the mixture preparation continues to be very much better than for a multipoint injector. For Condition 3 the throttle is only 4.6° further open but the distribution has broadened to span from below 2 μm to above 100 μm, with the majority of the fuel volume existing as droplets of around 15 μm. Compared with preceding conditions, a dramatic increase in Smd occurs at Condition 4 as a result of opening the throttle to 46°, revealing the inherent carburettor problem of worsening mixture preparation as the scale of throttle disc atomisation diminishes. Once this sharp deterioration in mixture preparation has occurred, further throttle opening makes little difference, as may be judged by the similarity of results for the 2000 rev/min wide open throttle test (Condition 5). This distribution peaks at 100 μm in a manner strongly reminiscent of a typical injector result, but unlike the injector there is a significant presence of smaller droplets in the region of 40 μm, which no doubt account for the calculated Smd of 73.0 μm. It should be understood from Appendix VI that the final test (Condition 6) does not attempt to simulate a realistic engine state but was included simply to see whether doubling the air flow from the preceding condition would have any favourable influence on the downstream droplet size. Notably finer droplets are indeed produced: although the basic form of the 100 μm peak is little changed some additional droplet shattering has evidently occurred, yielding a sizeable tail end of smaller droplets reaching towards 10 μm.

3.3.2 Comparison of carburettor and injector results

The conclusion from this work is that at light loads the carburettor is an economic generator of extremely fine droplets, but that its performance declines rapidly as the throttle is opened, particularly as the level of manifold vacuum falls below around 15 cmHg. These characteristics are summarised in Fig. 3.13, and alongside are repeated some multipoint injector droplet size distributions from comparable conditions.

With peaks at nearly 100 μm in each case the injector performance is fairly uniform throughout the range of airflows and port depressions, and is by no means dissimilar to that of the carburettor at wide open throttle. It is, however, markedly inferior to the carburettor at
part load - at which condition many vehicles of necessity spend much time operating. This is not to suggest that the long established single-point form of fuel system is to be preferred, on the contrary such a system has many problems of its own, not least mixture maldistribution between cylinders and an inherent incompatibility with the modern forms of closed loop electronic control which are essential to maintain the stoichiometric operation required by exhaust catalysts. However, it is salutary to note the high quality of atomisation which such a simple mechanism as high velocity airflow past a throttle disc can produce.
Fig 3.1 Multipoint injector sprays into free air: droplet size histograms at various measurement delays ('Additive free base super petrol' throughout)
Fig 3.2 Multipoint injector sprays into free air: droplet size histograms at various measurement delays (White spirit 'fuel' throughout)
Fig 3.3  Injector-to-injector variation: 10 ms measurement delay
Fig 3.4 Injector-to-injector variation: 14 ms measurement delay
Fig 3.5 Viewing passage for Malvern Particle Size Analyser

W88/76 Gasoline
D(3,2) = 113.8 µm SSA = 0.0527 m²/c.c. 'Free air'

900 rev/min (Idle)

W88/76 Gasoline
D(3,2) = 80.1 µm SSA = 0.0749 m²/c.c.

1500 rev/min (road load)

W88/76 Gasoline
D(3,2) = 55.0 µm SSA = 0.1091 m²/c.c.

2000 rev/min (WOT)

Fig 3.6 Injector spray size distributions at various air flowrates
Fig 3.7 Effect of port vacuum on injector spray size distribution

Fig 3.8 Injector sprays into free air. Fuels of varying surface tension and viscosity
Fig 3.9 Injector sprays with various fuels
(900 rev/min idle simulation)
Fig 3.10 Injector sprays with various fuels
(1500 rev/min road load)
V. n-Octane
D(3,2) = 93.3 μm  SSA = 0.0643 m²/c.c

V. n-Decane
D(3,2) = 99.8 μm  SSA = 0.0601 m²/c.c

V. Toluene
D(3,2) = 97.5 μm  SSA = 0.0615 m²/c.c

V. o-Xylene
D(3,2) = 100.0 μm  SSA = 0.0600 m²/c.c

Fig 3.11 Injector sprays with various fuels
(2000 rev/min WOT)
Fig 3.12 Carburettor droplet size distributions for Conditions 1 - 6 (cf Appendix VII)
Winter grade gasoline fuel
Fig 3.13 Comparison of carburettor and multipoint injector droplet sizes

Plate 3 Equipment layout for rig tests of free air sprays
- injector positioned to fire across Analyser beam
Plate 4  Bosch manual control unit and coil, for injector triggering

Plate 5  Typical form of multipoint fuel injector spray (gasoline)

Plate 6  Poor quality spray during initial pulses (gasoline)
Plate 7
Perspex replica of injector mounting tube (bolted to top of viewing passage)

Plate 8  Arrangement for measuring Weber carburettor droplet sizes
CHAPTER 4.0

EXPERIMENTAL WORK - MIXTURE PREPARATION DOWNSTREAM OF VALVE
CHAPTER 4.0
EXPERIMENTAL WORK -
MIXTURE PREPARATION DOWNSTREAM OF VALVE

4.1 STEADY FLOW SIMULATIONS

4.1.1 Steady flow 'blown' rig

Preliminary investigations of in-cylinder mixture preparation centred around an elementary 'blowing rig', with which droplet sizes immediately downstream of an inlet valve could be measured in steady flow. The arrangement comprised a portion of cylinder head from the European version of the 2.9 litre Ford V6 engine, together with corresponding sections of intake manifold and plenum chamber, all cut down to the length of one combustion chamber to enable mounting of the assembly between the Malvern Analyser transmitter and receiver units. This permitted the laser beam to pass across the inlet valve about 12mm above the face of the cylinder head, through the area where most droplets were likely to emerge (Plate 9). Considerations of focal length dictated use of the 300mm range lens, with the implied inability to detect droplets smaller than 5.8 μm, but from knowledge of injector sprays both in free air and steady airflow this was not thought likely to be a serious deficiency. To simulate the wall shape of the absent engine cylinder a perspex replica was fitted (Plate 10), and provided with a small circular aperture and larger glass window for the incident and diffracted laser beam respectively. Air supplied by a Howden screw-type compressor, at rates calculated to be representative of typical mean engine air consumptions at various engine speeds and loads, was then blown into the induction system and emerged via the inlet valve, which although fixed for any one test could be set to whatever lift was desired, with the aid of slip gauges and an adjustable collar. Control and supply of the fuel injector were set up as in the initial free air spray work, with a hose connection to the solitary injector. Rather than use the unwieldy fuel rail for locating and retaining the injector, the hose was clamped securely [Plate 11] to ensure the same injector orientation as with the normal method. Only later did experience prove this alignment to be significantly more critical than ever anticipated.

It is to be noted also that in this preliminary rig the cylinder head was mounted inverted relative to its orientation on the installed engine, both for convenience in applying the Malvern Analyser and to permit easy camera access towards the intake valve from the open end of the dummy cylinder. Exclusion of ambient light (by draping a dark cloth over the frame visible in Plate 12) allowed the exposure to be controlled purely on flash duration, enabling common connection of Particle Sizer and flashgun triggering [Fig. 4.1] to obtain concurrent
photographic and laser diffraction records of the spray during a sweep. In combination these methods provided a reasonable overview of the spray, and, coincidentally, gave objective proof of evaporation problems with Malvern Analyser measurements.

4.1.2 Typical results with steady flow blown rig

First attempts at readings with the described equipment resulted only in the Malvern Analyser indicating near zero obscurations, despite trial and error experimentation with a miscellany of values for air flow rate, valve lift, injector trigger pulsewidth, and measurement delay period. As the problem was visibly an unsuitable choice of laser beam position, the cylinder head was rotated on the Analyser bed so that the beam scanned across a different diameter of the combustion chamber. Both alignments are shown in Fig. 4.2.

Analyser readings now returned modest but positive obscurations in the range 2 - 5%, which are nonetheless acceptable provided the high gain facility is selected during analysis. The main concern proved to be that spray deposition on the cylinder window at times restricted measurements to only five sweeps, in the interests of ensuring the initial background data remained valid. Tests were carried out with airflows set in turn to simulate road load running at 1500 rev/min, the case of 2000 rev/min wide open throttle, and the arbitrary condition of the latter airflow doubled. Valve lift was held constant at 5.0 mm throughout, this being approximately one half of the maximum. Early tests with gasoline disclosed such extreme signs of evaporation in the raw data that SBP3 was substituted as fuel during the tests described. Even so, about the only certain development when measurement delays were increased successively (in an effort to identify any temporal variations in the spray) was the growth of prominent evaporative peaks in the distributions.

Compared with free air tests previously it is noteworthy that the typical in-cylinder droplet sizes of around 50 -75 µm indicated by the Analyser were on average smaller, suggesting that the presence of even a cold inlet valve may cause some secondary break-up of droplets formed at the injector. Such a theory may be consistent with the invariably chaotic form of the bottom end of the distributions (Fig. 4.3), just as might be the anticipated result of random impact, breakup and re-entrainment processes. As the air flowrate was increased so the biasing towards smaller droplets continued, although in absolute terms the majority of the fuel continued to exist in the form of droplets which are subsequently shown to be over large.

It was noticed while setting up for Malvern scans that around 30 mm of cylinder wall circumference was particularly prone to being wetted, this being the area on the far side of combustion chamber from the injector in the general direction of the spark plug. Indeed, at no condition did fuel spray emerge from more than a small proportion of the valve periphery -
typically all fuel visible and photographed was within an arc subtending an angle of less than 30° at the centre of the inlet valve. Arrangements were therefore made for a brief investigation of the flow velocity around the valve, with a view to establishing whether such a localised spray is simply the inevitable effect of imparting momentum to fuel during injection, or is related partially to some significant non-uniformity of airflow. A dummy inlet valve was made with a drilling through the stem and head, the latter tapped to mount a curved Pitot tube specially formed to align parallel with the valve seat [Plate 13]. An inclined water manometer measured pressure at the hollow stem as the valve was rotated in its guide through eight equally spaced increments of rotation. Various steady air flowrates and valve lifts were set up and the Pitot tube was kept centred throughout between the valve and seat. From the dynamic pressure at each test point the local air velocity was first calculated then normalised by a mean velocity deduced from the valve curtain area and overall mass flow rate. Polar plots of normalised values confirm that reduction of valve lift and/or increase of flowrate make the radial velocity distribution progressively more uniform. Referring to the 1500 rev/min road load airflow results as a typical example (Fig. 4.4), the air velocity around the valve periphery can be seen to be essentially constant at lifts of approximately one quarter and one half of maximum. For the case of maximum lift the relatively lower velocity on one side probably reflects a feature in the porting of the cylinder head. However, the variation of airflow around the valve periphery is insufficiently marked to account for such localised sprays as were seen: these no doubt follow from the direction in which the injector discharges through the open valve.

4.1.3 Photographs and confirmation of evaporation effect

The majority of Analyser readings on the blown rig were accompanied by at least one simultaneous exposure of a transparency. When projected to a large size (on an untextured screen of white paper or melamine board) the definition of droplets was sufficient to give a fair impression of the location and disposition of the spray. Unfortunately, the degree of enlargement preferred, enough typically to give a valve diameter of around 1 m, results in such photographs being ill-suited for print reproduction, other than in certain cases where particularly large drops are prominent. Scaling of droplet sizes from the projection screen was a straightforward matter using the inlet valve diameter (42 mm) as a length reference. Estimation of the likely inaccuracy in the large scale measurement also provided a guide as to the probable error implied in the scaled diameter. In retrospect a useful control test would have been to ‘seed’ the airflow with particles of known size, as a check on scaling accuracy.

It was found effective to site the flashgun externally and to one side of the dummy cylinder, the surface of the machined perspex wall serving as an excellent inbuilt diffuser. To reduce confusion with the bright casting of the cylinder head the droplets were illuminated from an
angle chosen to place the background partially in the shadow of the open valve. Good depth of field was obtained through use of an 800 ASA film, making practical a lens aperture of f11.

The main value of the simultaneous photographs proved to be in their justification of existing doubts (cf para 2.3.5) over Malvern Analyser indications of isolated peaks at very large droplet diameters. One representative example is included as Fig. 4.5, showing the Analyser histogram from a 1500 rev/min road load airflow test with gasoline fuel, and alongside the corresponding photograph. The very largest drops estimated by scaling were put at 200 \( \mu m \pm 44 \mu m \), and it might even be argued that this is most likely to be on the high side, given the possible effects of image burning and blur. However, while 200 \( \mu m \) would be in reasonable agreement with the primary peak of the bi-modal histogram there is certainly no photographic evidence of droplet diameters as large as 550 \( \mu m \), as reported in the Analyser data and histogram. Since the image on the film chanced to be almost exactly half full size, one would expect any 500 \( \mu m \) drops recorded on the slide to be discernible - even to the naked eye - as bright spots of around 0.010" (0.25 mm) diameter. With this experience, and earlier evidence from free air tests on involatile fuels, it became somewhat more acceptable to ignore or suppress evaporative peaks in the droplet size histograms. However, a less subjective method of addressing this problem was confirmed as a requirement deserving future attention.

4.1.4 Importance of injector orientation

Droplet photography also provided graphic evidence of one of the main results from this phase of work - that injector orientation is of the greatest significance to the mixture preparation realised in the cylinder. Ford V6 injectors are customarily aimed towards the back of the inlet valve, so in any simulation at ambient temperature it is to be expected that some fuel would enter the combustion chamber from the rim of the valve. However, it was found quite accidentally that alterations of less than 4° in the injector alignment could result in nearly all the fuel becoming attached to the port wall at an early stage, yielding an exceptionally coarse-looking mixture downstream of the inlet valve.

Results taken with white spirit, once again at a 1500 rev/min road load airflow, are shown in Fig 4.6 to demonstrate the point. The spray in the photograph with the valve labelled '103' is as good as can be obtained with this particular hardware: fine enough, indeed, to be barely visible unless projected, when droplets may be seen moving out from the valve seat in a 'north-easterly' direction. A dramatic contrast is provided by the second illustration in the figure. During this exposure all conditions and settings were as for the first, save for the fuel hose end of the injector (Plate 11) being displaced vertically upwards by 3 mm. In this ultimate example of bad mixture preparation the fuel droplets are all too obvious, and the 20 to 400 \( \mu m \)
droplet size range indicated by laser diffraction seems feasible, especially given that a test 'fuel' of the lowest readily obtainable volatility was employed.

Incidentally, during one prolonged period of steady flow blown rig operation, with poor injector orientation, it became evident that some fuel was collecting in the port rather than reaching the combustion chamber above. It was then appreciated that cylinder heads and ports should in future be tested only in their normal installed attitudes, regardless of any extra difficulties in applying the Malvern Analyser which such a requirement might bring. Conversion to a suction rig could be arranged at the same time, in order to approach more nearly the practical case of mixture induction by downward motion of a piston in cylinder.

The opportunity was taken meanwhile to explore the subject of injector orientation a little further, by running several tests on an existing airflow rig with motored valves. Utilising components from an identical Ford V6 engine, this rig comprised a cylinder head mounted on an angle iron frame, the normal camshaft (in a non-standard overhead position), and a powerful electric motor to drive the latter via a toothed belt. The same perspex dummy cylinder as used on the basic blown rig was attached to an end position on the cylinder head, and a compressed air supply was similarly connected to the inlet of the induction system. Importantly, the inlet manifold had been modified to accept an endoscope and separate light guide, giving scope for limited observation and photography of fuel spray behaviour inside the manifold.

Using white spirit 'fuel', and with airflow, pulsewidth and camshaft speed set to represent a 1500 rev/min road load condition, photographs were taken with the free end of the injector displaced in various directions by the same amount as earlier produced the dramatic result of Fig. 4.6. Synchronisation was so arranged that the start of injection coincided with inlet valve opening, and exposures were timed to capture the developed spray two thirds of the way through a pulse. Severe reduction of flash intensity, inseparable from transmission by fibre optic cable between the Balcar Minibloc unit and interior of the port, necessitated push-processing at 3200 ASA of the Fujichrome P1600D Professional film; illumination in the photographs even so leaves much to be desired. However, in both photographs of Fig. 4.7 the shiny valve stem is visible as a clear point of reference where it emerges from the valvecap, the head of the valve being lowermost and the injected fuel approaching from top left. Fair targeting of the spray onto the back of the inlet valve (left hand picture) contrasts with signs of impingement on the port wall in the other photograph, a degree of change which once again seemed disproportionate to the relatively small modification of injector orientation.

Such sensitivity might well translate into running problems with the Ford V6 engine, having regard to the somewhat imprecise method of locating and retaining all the injectors by one
pressed steel fuel rail. It is doubted whether this arrangement can hope to guarantee precisely the same injector orientation for all cylinders. The likelihood of reproducing the same injector positions during reassembly after maintenance is also questionable. Such concerns were expressed to the manufacturers, and it is noted that the fuel rail has been re-engineered to advantage on more recent engines, with machining of the areas affecting injector location.

4.1.5 Steady flow suction rig

The main complication encountered in reconfiguring the rig to mount the cylinder in its normal installed attitude was that the Malvern Analyser had also to be rotated to a new semi-inverted position. For the portion of diffraction pattern on the cylinder side of the optical axis it could otherwise happen that the line of sight to the range lens and detector would be blocked physically by the presence of the cylinder head. As the detector 'rings' are in fact merely semicircular the simple solution is to locate the Analyser so that the detector is placed to receive only the uninterrupted half of the diffraction pattern.

An internal examination of Analyser transmitter and detector revealing no obvious reasons to prevent inversion, the units were removed from their normal base, and bolted at the same spacing to the underside of a substantial frame constructed from aluminium I-beam. As may be seen in Plate 14, this was inclined at 30° to reflect the 60° vee angle of the Ford V6 engine. Correct optical alignment was easily regained and maintained with the re-mounted Analyser, suggesting that the rigidity of the new bed was quite adequate.

Bearing in mind that suction was now to be applied to the cylinder, the former method of providing a small hole in the cylinder wall for entry of the incident laser beam became inadmissible. A dummy cylinder with provision for two 'Utility Grade Crown Glass' windows was designed (Fig. 4.8 and Plate 15), the larger square window sealed permanently in position on the receiver unit side of the cylinder, the small circular window for the incident beam being housed in a carrier which could be rotated within an eccentric sleeve to adjust the window angle in swash-plate fashion. By this means any signs of internal reflection could be spilled from the alignment display. A window thickness of 6.35 mm thickness was chosen to prevent loss of optical alignment through window distortion; however, with rigs of this type it is advisable in any case always to make a point of re-checking alignment with the intended test vacuum applied. The mounting flange on the cylinder head was drilled for possible location of the cylinder in 15° steps of angular position, to allow the laser beam to sample different areas of the combustion chamber if required. Initially, the measurement volume was arranged as shown for the blown rig.
Amongst other relevant features of the suction rig is the large chamber included immediately downstream of the dummy cylinder, which hopefully provides a more uniform flow in cylinder than would result from a direct connection to the 3” bore suction hose. Incidentally, experience has shown it to be essential that large hoses subjected to high vacuum be of the reinforced type incorporating stiffening rings. At the inlet end of the rig a slide throttle is mounted on the cut down end of the plenum chamber, and a mercury manometer connected to an existing pressure tapping. The fuel rail was also cut down to the length of one cylinder and blanking plates silver soldered over open ends, enabling the injector to be located and supplied as in the normal way, and the usual pressure regulator to be fitted. This regulator automatically references the pressure of the fuel supplied to the injector to the prevailing manifold depression, so that the injector calibration remains valid regardless of pressure changes in the port.

4.1.6 Typical results with steady flow suction rig

The objective originally envisaged for these tests was to gain an understanding of the respective implications for droplet size of valve lift, air flow rate, manifold pressure, and fuel type. At an early stage the more difficult variable of cylinder head temperature was even considered, which could probably have been assessed to a limited extent by filling existing coolant passages with hot water or oil.

However, no sooner had testing at ambient temperatures begun than it was immediately obvious that the quantity of fuel spray coming through to the cylinder was so minimal as practically to rule out use of the Malvern Analyser, even with high gain mode. The majority of the fuel instead appeared in liquid form, streaming past the valve and down the cylinder wall, creating window wetting problems as a further obstacle to accurate droplet size measurements. It was just possible, albeit at a borderline obscuration, and by setting a limit of five measurement sweeps, to obtain some sample droplet size measurements at steady airflow conditions corresponding to 1500 rev/min road load and 2000 rev/min wide open throttle, with valve lift set at 5 mm. Typical Smd values for these respective conditions were 54 µm and 46 µm, possibly suggesting that a higher airflow round the valve assists somewhat with breakup of the droplets, but both distributions [Fig. 4.9] nonetheless extended to droplet diameters as large as 200 µm, and peaked at approximately 100 µm. No evaporation was evident in the raw data despite the use of gasoline as test fuel, conceivably because the ambient temperature of 16°C in the Test Facility was very much cooler than the outlet of the Howden compressor used formerly. However, the possibility of exploring different laser beam paths across the combustion chamber was not proceeded with, as conditions in cylinder clearly did not lend themselves to useful characterisation by droplet size measurement.
Instead, the Malvern receiver unit was demounted from the rig to permit easier visual access through the larger window to the combustion chamber, illumination being provided externally by a hand held light guide. Various airflow conditions (900 rev/min idle, 1500 rev/min road load, 2000 rev/min wide open throttle) were set up in turn, and the injector pulsed at the correct repetition rate for the simulated crankshaft speed. For each airflow the valve lift was stepped through the full range of lift in 1mm increments while observing the mixture preparation visually. This exercise confirmed that the only circumstance in which wallfilm flow did not predominate was at very small valve lifts, where the relatively high air velocity in the valve curtain area is presumably sufficient to shear fuel liquid fuel into some semblance of a spray. Nonetheless, the full circumference of the cylinder wall is visibly wetted. Conversely, at high valve lifts and low air flowrates the mixture preparation degenerated into a succession of drips from the rim of the valve, and a stream down the low side of the angled cylinder wall. In this case the spark plug became extremely wet with fuel, a phenomenon which may correlate with some internal findings by Ford during an investigation of reportedly excessive cranking times to achieve a cold start.

The essential message coming from this unfortunate rig, after extensive visual observation as described, is that at ambient temperature and in steady flow the nature of in-cylinder 'mixture preparation' for the hardware under study is definitely wallfilm rather than droplet spray.

4.2 CYLINDER WALLFILM MEASUREMENTS

4.2.1 Requirement for a wallfilm measurement technique
Having encountered such poor mixture preparation, the character of experimental measurements obviously had to undergo a radical change from precision droplet sizing to in some way quantifying the wallfilm. Previous reasons for avoiding pulsating flow rigs - principally vibration and ease of optical access - became less compelling as a result.

To establish whether equally adverse mixture occurred under pulsating flow conditions, some abbreviated visual tests were run using the hybrid rig shown in Plate 16. This was constructed from the Ford V6 cylinder head with motored valves and the dummy cylinder head with windows recently used on the steady flow suction rig. Timed triggering of the injector was provided by the proprietary Malvern rotation sensor operating from a mark on the cam belt pulley. Although observations were complicated by extreme fuel obscuration of both windows, the tests entirely confirmed the continued existence of largely wallfilm flow. It might be added as a rider that some reassurance was also gained as to the veracity of predictions from steady flow tests.
A wallfilm measurement technique was therefore necessary to gain objective data on the effects of air flow rate, injection duration and injection timing. With such a method in place it was hoped to investigate ways of improving the downstream preparation, during which minimisation of the wallfilm would be one criterion. Ultimately, this process might be expected to allow reversion to droplet sizing techniques, once the mixture preparation has been sufficiently improved.

4.2.2 Possible approaches to wallfilm measurement

A variety of possible wallfilm measurement techniques were reviewed before finalising the choice. One consideration relevant to most methods is at what axial position along the cylinder collection should be attempted - too far downstream and some of the film may have been lost to evaporation and re-entrainment. Equally, attempts to collect too far upstream might involve undue disruption of the cylinder shape, and hence affect the flow itself. For this reason the obvious option of creating sharp-edged 'pockets' in the cylinder wall to gather the fuel was not favoured. This approach might also have been ineffective in the case of a wallfilm on the 'overhead' side of the inclined cylinder bore.

An entirely non-intrusive system capable of monitoring the entire cylinder length would have been to line the cylinder with some form of replaceable card and use a dyed fuel. This should provide an exact record of the areas affected by wallfilm, with some possibility of inferring magnitude from the density of colouring. The idea was rejected, however, as a physical measurement was sought, be it gravimetric or volumetric, of the fuel quantity ending up on the cylinder wall. It did seem that this might be achievable by embedding an absorbent material in the cylinder wall, the weight increase of which could be determined after a given number of injection pulses, but fluid loss through evaporation and draining downstream would be just two of the probable drawbacks.

Still another proposal, and unquestionably one with novelty on its side, was to divide the cylinder wall into discrete areas thermally insulated from one another, then to supply just sufficient heat to maintain a dry surface condition on each during continuous running. PTC capsules removed from electrical manifold pre-heaters are sufficiently compact and powerful for this suggestion to border on practicality, unfortunately the problem would be automating judgement of the degree of wetness/dryness of the heated surface. The fuel flowrate over each area would be calculable from the power supplied to the relevant heater, although losses to the air alone would have had to be estimated and subtracted. As a rough guide to the order of heat input required overall it was calculated that about 500W would be necessary to evaporate all the fuel at a 2000 rev/min wide open throttle condition. Reluctantly, it was concluded that while the concept may have possibilities this was a far from ideal situation in
which to develop it, not least because the area where measurements are required is relatively small and inaccessible.

The last option considered, and the one finally adopted, was incorporation of a length of porous bronze filter element into the base of the dummy cylinder as a skimming section. This 'Porosint' material was sourced from GKN Sheepbridge Ltd. in an almost exactly correct inside diameter for representation of the 93mm bore of the Ford V6 engine. Approximate calculations using catalogue data for water flow confirmed that the available 1/8" wall thickness and Grade F (45 μm) porosity would pass the anticipated fuel flow rate many times over. It was envisaged that a suction marginally in excess of cylinder depression would be applied to an annular space on the outside of the porous ring, entraining any wallfilm into external collection vessels. The natural concern was that the skimming suction might bias the main airflow towards the cylinder wall, and even result in erroneous collection of airborne droplets; fortunately, preliminary tests (q.v.) suggested that this was not the case.

4.3 PULSATING FLOW RIGS FOR WALLFILM MEASUREMENT

4.3.1 Rig based on Ford V6 engine
The apparatus for measuring wallfilm on the Ford V6 engine was built around the aforementioned pulsating rig, and consisted in the main of one cylinder bank complete with its standard manifold, plenum chamber, upstream throttles, fuel rail and repositioned camshaft (Plates 17 & 18). Only the three inlet valves are operated, the camshaft being driven at up to 1015 rev/min by a 3.0 kW Lenze Simplabelt variable speed motor. Three 'dummy' cylinders 200 mm long connect with a large chamber to which suction is applied from the main Test Facility supply, so the air consumption of the rig (as indicated by a DVM from a hot wire anemometer upstream of the plenum throttles) is one half of that measurable off the test bed V6 engine at the desired condition. For wallfilm flowrate measurement a cylindrical section of sintered porous material was included in the base of the central cylinder (Fig. 4.10). This is fitted with eight suction tappings around its circumference to enable wallfilm to be skimmed off over a measured time period and collected for quantity measurement. Thin metal dividers spaced every 45° around the exterior of the skimming section are intended as a deterrent to flow around the annular collection area, hopefully allowing the fuel collected at each suction tapping to reflect the wallfilm deposited in its particular region. Failing this, it is in any case the sum total of all wallfilm gathered which is of prime interest.

Pipes from all the suction tappings are directed into individual graduated glass tubes of 20ml capacity. These are in turn wholly contained within two large dessicators, to which the
Requisite skimming suction is applied from a vacuum pipe connected to the main test facility damping chamber via control valves. The rig was instrumented with all necessary manometers and suction gauges to allow monitoring of cylinder and plenum depressions, as well as of the skimming suction differential being applied.

An endoscope (Fig. 4.11) looking along the bore allows a visual check that wallfilm is indeed collected in full, and permits viewing of the mixture preparation generally. Plate 19 gives some idea of the field of view through the endoscope, although in practice the clarity is considerably better than photographs would suggest. An air bleed tube venting to ambient (giving a leak into the plenum) was sited beside the endoscope lens to give a flow of air past the window and blow off any fuel deposits. The leak does not affect the airflow through the engine as it occurs downstream of the airflow meter, but a compressed air source would be necessary to maintain the flow during wide open throttle tests.

Illumination is provided by an Endolux continuous light source through an Ultralux liquid light guide mounted vertically in the floor of the plenum, beneath the central wallfilm measurement cylinder. The light guide faces upwards towards the valves and is so located as to prevent the endoscope tube casting shadows onto the areas under examination.

Injector triggering once again used the Malvern PS58 rotation trigger sensor to provide a timed signal capable of variation to give any desired phasing relative to inlet valve events. In conjunction with the Spray Synchroniser display this sensor also doubles indirectly as a tachometer, by providing the time required for each camshaft rotation. With 1:1 gearing on the camshaft drive simulated engine speeds cannot exceed 2030 rev/min, but in the present work this is no disadvantage. As usual, the Bosch Manual Control Unit was included in the triggering circuit to provide a pulse of representative shape.

4.3.2 Rig based on Ford Zeta four-valve engine

An ex-development 1.8 litre Ford Zeta 2B engine provided an ideal basis for a further wallfilm measurement rig, incorporating as it does design features typical of modern four-cylinder engines in this class and providing a contrast with the same manufacturer's time-honoured V6 design. The most significant features of the engine are that it has four valves per cylinder, double overhead camshafts, and multipoint injection. Incidentally, it should be emphasised that at the time of constructing of this rig there was no prior knowledge of mixture preparation quality downstream of the valve in this engine, so the exercise was approached in the knowledge that there would not necessarily be wallfilms to measure, but that results of any kind would be valuable for comparison with those from the V6.
In devising the wallfilm rig it was possible to leave complete and untouched the vast majority of the top end of the engine (Plate 20). The inlet camshaft pulley was substituted for one compatible with the electric motor drive belt and various cylinder head bolt holes were plugged to enable the cam cover to be filled with oil for valve train lubrication. In the absence of an oil pressure supply the hydraulic tappets were modified with solid distance pieces to ensure the correct valve clearance.

Although the middle cylinder was selected for wallfilm measurement the remaining combustion chambers were also connected to the suction supply, via balance pipes, in order to allow the same overall engine air consumption to be set as for the complete engine. Design of the skimming cylinder was similar to the V6 rig except that the suction tappings were set lower down, towards the bottom edge of the porous liner. Surprisingly, surplus 'Porosint' material from the first rig was found to be sufficiently malleable to be formed to the smaller 80 mm bore diameter of the Zeta engine, after cutting out and discarding the excess length of wall. Once again endoscopic access was provided, and Plate 21 gives an impression of the cylinder head and valve geometry then visible.

4.4 WALLFILM RIG RESULTS

4.4.1 Ford V6 engine results

A preliminary task was to deduce representative air and fuel flows from practical measurements with the test bed V6 engine. Because low load running forms the most critical part of current emissions cycles, tests at idle and 1500 rev/min (set to 30 cmHg manifold depression) were of main interest, but a 2000 rev/min wide open throttle condition was also included. A hot wire air meter of the type used on the Zeta engine was connected upstream on the V6 engine, together with a DVM to indicate the voltage output, and the AVL Fuel Balance was temporarily transferred to the Test Cell and plumbed into the fuel system. Measured air meter voltages relating to the whole six cylinder engine were then substituted into a calibration polynomial as used by the Zeta ECU to calculate flowrate from the meter output. The resulting air flowrate was then halved (to reflect the three cylinder test rig) and reverse-substituted into the DOC file polynomial to deduce the required voltage to set on the rig. This was actually made a very simple operation by the expedient of plotting the calibration on reversed axes to find the equation in the other term, and using the CricketGraph facility to generate a curve-fit and describing equation.

Once the rig was satisfactorily commissioned the immediate requirement was to establish the minimum vacuum differential necessary to remove the wallfilm, and to check for the
existence of any tendency to entrain airborne fuel. Tests were carried out over a range of 0 - 26.6 kPa vacuum differential and the total quantity of fuel collected by the porous wall was measured at each increment; to reduce possible error through evaporative losses the 'fuel' used was SBP3. Air and fuel flows during these tests reflected a 2000 rev/min wide open throttle condition, on the basis that this would provide the maximum wallfilm flowrate that the skimming section would be expected to collect during the planned tests. The timing of the rotation sensor was such that the start of injection coincided with IVO. A characteristic curve of wallfilm, expressed as a percentage of the total fuel flowrate delivered by the injector, against skimming suction, is plotted in Fig. 4.12, and demonstrates that the percentage skimmed is practically constant once the differential is past a low threshold value: consequently, 6 kPa was standardised upon. This curve is of a most fortunate shape, as it provides reassurance that even if were the skimming suction to be quadrupled from the standard value, the quantity of wallfilm collected would stay unaffected. A reasonable inference is that the skimming section must only be gathering wallfilm, otherwise increasing suction differential might be expected to entrain more and more fuel droplets from the air, which would not result in a plateau of constant wallfilm percentage.

Another important indication from this same result is the sheer prevalence of wallfilm flow at this test condition. Around 70% of the total fuel injected ends up in this form, leaving only a minority to exist as spray. From observations through the endoscope during tests even the latter comment may be optimistic, as the only positive signs of fuel other than wallfilm were sizeable drips from the rim of the intake valve.

Subsequent measurements of in-cylinder wallfilm, again with SBP3, as a function of injector fuel supply pressure produced the significant result in Fig. 4.13. Bearing in mind that an increase in fuel pressure reduces the droplet sizes produced by pintle-type injectors, this graph can be interpreted as indicating that the quantity of wallfilm is heavily dependent on the standard of mixture preparation upstream of the inlet valve. While the wallfilm percentages are lower in this case of 1500 rev/min road load fuelling than when the vacuum differential was being investigated with the settings of 2000 rev/min wide open throttle, they are still very high - 37% at the lowest pressure employed and never less than 22%, the latter from a test at the correct injector pressure of 3.45 bar. It was not possible to take readings for airflows lower than 1500 rev/min road load because the poor mixture preparation manifested itself entirely as a series of drips off the inlet valve and seat, rather than as a stream on the cylinder wall. It is therefore suspected that at the cold-cranking condition, of which this ambient temperature rig would be a fair simulator, mixture preparation would be equally poor.
A logical question arising from the work was whether finer droplets than produced by the injector at its standard pressure would be of significant benefit to downstream preparation. The fuel injection system was therefore disabled and a Weber carburettor fitted directly to the appropriate manifold branch (Plate 22), being a known source of fine droplets at low load conditions (cf 3.3.1). Another sequence of wallfilm flow measurements was carried out with SBP3 at the same 1500 rev/min fuel flowrate as with the injector. Manifold vacuum was varied from zero to the high depression of an idle in order to ensure that the carburettor developed its full range of droplet sizes. At wide open throttle the wallfilm was 28% (comparable with the percentage under injection), but it fell progressively to zero as higher depressions produced smaller droplets (Fig. 4.14). The spray emanates from a greater proportion of the curtain area than was formerly the case and appears of excellent quality, being so fine that it is evident mainly as a slight 'flickering' of the illumination level. These results are despite the inlet tract being far from optimum for operation with a fuel system sited well upstream.

Another aspect of in-cylinder mixture preparation concerns the possible effects of the phasing of injection timing relative to valve opening events. The EEC IV Ford control system triggers the V6 injectors in two batches of three, meaning that each injector is triggered at a different time in the cycle of its respective cylinder. Measurements have been made hitherto with the injector trigger commencing at TVO, but a limited number of wallfilm readings were taken also for the cases of triggering at maximum valve lift, IVC, and at 180° camshaft rotation before maximum valve lift. Only a very tentative picture has emerged, and as the possibility that 'trends' may be no more than experimental scatter cannot be discounted the relevant graph is withheld from inclusion. It can only be said that there is possibly an advantage in triggering at IVO (20-22% wallfilm) rather than 180° camshaft rotation before maximum lift (25-26% wallfilm). Although these timings should not be treated as general they may be indicate the existence of an effect, even with the components at ambient temperature.

By way of a footnote, it can be added that repeat trials with BS7070 gasoline confirmed the existence of a severe wallfilm problem even with this realistically volatile fuel. Percentages were typically lower by a tenth than those recorded with SBP3; the specific gravity of the wallfilm was found to be 0.831 compared with 0.740 for the fuel supplied.

4.4.2 Ford Zeta engine results

A concurrent series of tests on a running Zeta engine (to be described in Chapter 6.0) effectively predefined the test conditions of main interest for the Zeta wallfilm rig. These were simulations of idling at 800 rev/min, and road load running at 1200 and 1500 rev/min
respectively. Once again, SBP3 'fuel' was used and rig settings of airflow and fuel pulsewidth settings were easily arrived at by direct measurement on the operating engine.

Initial running concentrated on establishing the skimming suction requirements for the new rig. As with the V6, and for the same reason, air and fuel flows for this work were set to represent 2000 rev/min wide open throttle running. Figures for this condition were deduced by applying an assumed 85% volumetric efficiency to the 1.8 litre capacity of the Zeta and calculating on a 14.5:1 AFR. Timing of injection was arranged to simulate the setting specified in the calibration provided with the UCL test bed Zeta engine - which provides for injection to be completed 272 crank degrees before piston TDC. It was at once apparent, both from wallfilm measurements and observation through the endoscope, that cylinder wall impaction occurred on an almost negligible scale, relative to the standard V6 engine. A probable explanation may be that centrifugal effects of the Zeta in-cylinder air motion affect the droplet trajectory rather less than does the suspected strong axial swirl in the V6. Although the wallfilm quantities collected were relatively minimal, requiring long test durations (10-15 minutes) for a fair size sample, a similar correspondence between wallfilm percentage and skimming suction was replicated (Fig. 4.15). It is interesting to note that a little fuel is collected even with no suction applied, perhaps because of the repositioning of suction tappings to the lowermost edge of the skimming section. In all cases, wallfilm fuel was gathered only from the two tappings immediately adjoining the valves. The overall conclusion was that a vacuum differential of 5 cmHg (comparing closely to 4.5 cmHg with the V6) would be appropriate for tests on this rig.

On attempting to establish the 800 rev/min idle test condition it was found that available pump capacity was just insufficient to maintain both the correct 50 cmHg manifold depression and the required skimming differential: 44 cmHg had therefore to be accepted as the nearest alternative. The measured wallfilms were 12.8% and 12.3%, the latter figure from a re-test on a different occasion. Given that these values are around two and a half times greater than those encountered at 2000 rev/min wide open throttle (at which condition a fuel quantity over eleven times greater is injected) the result perhaps hints at a problem characteristic of four valve engines - that the intentionally large port areas yield such low gas velocities at idle that mixture preparation and distribution can be compromised. During 1200 and 1500 rev/min road load simulations a good deal of the pent roof of the combustion chamber around the inlet valves was visibly wetted, yet the fuel concerned appeared to be re-entrained before running down the cylinder wall. The absence of any measurable percentage of wallfilm at these conditions must not therefore be taken as indicating significantly better mixture preparation than at idle. Indeed, re-entrainment seems to produce drops of a rather coarse nature, which go on to wet the wall very much further down the dummy cylinder.
Learning from evidence of the beneficial effects to be had from small initial droplet sizes, as gained from the V6 engine rig with carburettor attached, an entirely new fuel system had by now been developed for the Zeta engine, with formation of small droplets as a prime objective. Since this is explained fully in Chapter 6.0 suffice it for now to note that it provides for an individual port throttle downstream of each standard fuel injector, so that load control is shared between these throttles and the single upstream throttle. Brief tests of this fuel system were carried out on the wallfilm measurement rig [Plate 23]. With the port throttles fully open, results and observations were identical to those obtained with load control similarly by upstream throttle under the normal injection system. However, the port throttles were next adjusted progressively to carry an increasing proportion of the total pressure drop, at the same time observing in-cylinder mixture preparation through the endoscope. As the drop across the port throttle approached 20 cmHg, so the visible fuel changed decisively from existing as wallfilm and coarse re-entrained droplets to being a 'cloudy' mixture filling the whole cylinder. This change was further borne out by the fact that even a considerably extended trial did not result in any trace of wallfilm being collected. Tests at the 1200 and 1500 rev/min road load conditions gave similar behaviour, again with the same clearly defined threshold of port throttle depression beyond which mixture preparation improved.

4.4.3 Inference from wallfilm rig results

The dominant mode of mixture preparation observed in the Ford V6 cylinder with standard injection was undoubtedly wallfilm flow. While the test rig operated at ambient temperature it did by so doing simulate the cold start regime, which constitutes an element of the emissions cycle which can prove particularly demanding. In the warmed up engine, where combustion chamber temperatures may exceed 300°C, it is probably true that hot surfaces would mask poor mixture preparation by flash evaporating much of the liquid fuel. However, with current and impending emissions legislation to minimise unburnt HC's, such measured percentages of wallfilm as 70% at 2000 rev/min wide open throttle should be of great concern. It has, for example, been reported$^{25}$ that 40 to 50% of the unburnt HC emissions from a spark-ignition engine are attributable to the effect of incompletely atomised fuel entering the crevice above the top piston ring during compression, then returning unburnt into the cylinder during the expansion stroke.

Fortunately, a promising solution appears to be at hand if very small droplets can be provided upstream of the valve. Such a technique entirely transformed in-cylinder preparation on the V6, and noticeably improved an already fair situation on the Zeta. There are several aspects to the advantage gained - firstly that smaller droplets are evidently able to negotiate flow deflections around the valve without impacting (just as Lenz$^{43}$ concluded in
relation to manifolds), and can then remain airborne in the cylinder airflow so that a wallfilm problem does not arise. A spray of finer droplets is of course advantageous in itself for fuel evaporation and combustion, although practical engine tests are clearly necessary to quantify the anticipated benefits.

A final observation is that the wallfilm rig work has shown how improved mixture preparation might be expected to enhance engine operation. It was unclear formerly whether large droplets would be prone to impact as they approached and tried to pass the inlet valve, whereas small droplets would avoid impaction, or whether large droplets would pass the valve but be significantly less suitable for fast and complete combustion due to slower evaporation and mixing. Results on the present rigs have indicated that all such mechanisms are likely to be relevant, and that large droplets are unlikely to pass the valve at all without impacting and creating a wallfilm.
Fig 4.1  Particle Sizer and flashgun triggering
Fig. 4.2 Laser beam path over valves in blowing simulation (Ford V6 engine)

Fig. 4.3 Typical in-cylinder droplet sizes measured on 'blown' rig (800 rev/min idle airflow, 5 mm valve lift, SBP3 'fuel')

Fig. 4.4 Normalised air velocity around inlet valve periphery (1500 rev/min airflow rate)
Maximum droplet size

\[ \text{Maximum droplet size} \leq 200 \, \mu m \pm 44 \, \mu m \]
(from scaling)

Secondary peak due to refraction of laser beam by fuel vapour

---

Fig. 4.5 Confirmation of spurious Analyser indications with volatile gasoline fuel (due to refraction of laser beam by fuel vapour)

---

Fig. 4.6 Importance of injector orientation - effect on spray in cylinder. Optimised (top) vs. misaligned from optimum (bottom)
Fig. 4.7  Importance of injector orientation - effect on spray in port.
Fair targeting of valve (LH) vs. wall wetting (RH)

Fig. 4.8  Dummy cylinder for steady flow suction rig
1500 rev/min ROAD LOAD SIMULATION
fuel: pump petrol
valve lift: max
D(3,2) = 53.76 µm
obs. = 0.0318

2000 rev/min WOT SIMULATION
fuel: pump petrol
valve lift: max
D(3,2) = 46.01 µm
obs. = 0.0126

Fig. 4.9  Steady flow suction rig: representative in-cylinder droplet sizes

Fig. 4.10  Cross-section of wallfilm skimming cylinder
Fig. 4.11  Section through Vinten endoscope

Fig. 4.12  Effect of skimming suction (ΔP) on quantity of wallfilm collected (V6 engine)

Fig. 4.13  Influence of injector fuel supply pressure on quantity of cylinder wall film (V6)
Fig. 4.14 Cylinder wall film as a function of carburettor depression

Fig. 4.15 Effect of skimming suction ($\Delta P$) on quantity of wallfilm collected (Zeta engine)
Plate 9  Cut-down Ford V6 cylinder head mounted on Malvern Analyser

Plate 10  General view of V6 Blowing Rig, with perspex dummy cylinder
Plate 11 Injector mounting in manifold of blowing rig

Plate 12 General view of Malvern Analyser and blowing rig
Plate 13 Pitot tube for measuring flow velocity around inlet valve

Plate 14 General view of steady flow suction rig, based on Ford V6 engine
Plate 15
Detail of dummy cylinder on steady flow suction rig

Plate 16
Preliminary version of suction rig with motored valves
Plates 17 and 18
Pulsating flow rig for wallfilm measurement, based on Ford V6 engine
Endoscopic views of Ford V6 (19) and Ford Zeta (21) combustion chambers - porous cylinder wall inserts visible in both cases

Plate 20
Pulsating flow rig for wallfilm measurement, based on Zeta engine
Plate 22
V6 wallfilm rig with Weber carburettor in place of fuel injection system

Plate 23
Zeta wallfilm rig with port throttles fitted
CHAPTER 5.0

THEORETICAL PREDICTION OF DROPLET TRAJECTORY
5.1 INTRODUCTION

In parallel with rig-based investigations of fuel droplet behaviour in cylinders, a mathematical model was developed to predict droplet trajectories in the combustion chamber. It was envisaged from the start that the model would be run as a computer program, facilitating parametric studies which would most relevantly involve initial droplet diameter but also include the effects of swirl ratio and the location in cylinder at which droplets are introduced. Of paramount interest was comparison with the main experimental conclusion that reduction of droplet size greatly reduces or eliminates cylinder wallfilm. In the interests of simplicity, the model was based on an elementary two-dimensional circular cylinder head shape, but it was considered important that drop evaporation be taken into account, and for this aspect an evaporation analysis routine pre-existing at UCL was found to lend itself to adaptation.

Air motion was taken to be a solid body rotation of angular velocity $\omega$ about the central axis of the circular cylinder, into which a single droplet is introduced at some radius $r$ and angular location $\theta$ with specified initial components of radial and tangential velocity, as indicated in Fig. 5.1. Account must obviously be taken of the inlet valve seat location when specifying the radius, but beyond this the model makes no further reference to the presence or characteristics of the valve. Further assumptions are that gravitational effects are negligible, that in practice no momentum transfer would occur between droplets to alter their trajectory, and that no significant momentum would be lost by the air whilst interacting with the droplets.

5.2 EQUATIONS OF MOTION

Resolving separately in the radial and tangential directions yields the three equations which together define the motion of a droplet of constant mass, for the possible input cases:

5.2.1 Radial direction

$\left( \ddot{r} - \dot{r}^2 \right) m = -\frac{r}{2} \rho A C_{D_r} (r)^2$

$\Rightarrow \ddot{r} - \dot{r}^2 + kr^2 = 0$

where $k = \frac{\rho A C_{D_r}}{2m}$

(11)
5.2.2 Tangential direction \((\omega > \dot{\omega})\)

\[
(r\ddot{\theta} + 2r\dot{\theta})m = \frac{1}{2}\rho AC_{D, t}(r\omega - r\dot{\theta})^2
\]

\[
\Rightarrow \frac{1}{r}\frac{d}{dt}(r^2\dot{\theta}) = kr^2(\omega - \dot{\theta})^2
\]

where \(k = \frac{\rho AC_{D, t}}{2m}\) \hfill (12)

5.2.3 Tangential direction \((\omega < \dot{\omega})\)

\[
(r\ddot{\theta} + 2r\dot{\theta})m = -\frac{1}{2}\rho AC_{D, t}(r\dot{\theta} - r\omega)^2
\]

\[
\Rightarrow \frac{1}{r}\frac{d}{dt}(r^2\dot{\theta}) = -kr^2(\dot{\theta} - \omega)^2
\]

where \(k = \frac{\rho AC_{D, t}}{2m}\) \hfill (13)

5.3 DIFFERENTIAL EQUATION SOLUTIONS

5.3.1 Radial direction

In the apparent absence of general analytical solutions for \(r\) and \(\dot{r}\) in equation 11, a numerical approach involving Runge-Kutta and Predictor-corrector formulae was adopted\(^{37}\). The variable \(\dot{\theta}\) can reasonably be treated as constant over a small step, an acceptable approach for this purpose since a method of incrementing through small steps in time is used when producing the locus of the droplet trajectory. This simplification allows equation 11 to become:-

\[
\ddot{r} = \Omega^2 r - kr^2
\]

(substituting \(\Omega = \dot{\theta}\)) \hfill (14)

For this form solutions for the first time step of duration \(t\) are given as:-

\[
r_1 = r_0 + t\left[\ddot{r}_0 + \frac{1}{6}(k_1 + k_2 + k_3)\right]
\]

\hfill (15)

\[
\dot{r}_1 = \dot{r}_0 + \frac{1}{6}(k_1 + 2k_2 + 2k_3 + k_4)
\]

\hfill (16)

and in this case:-

\[
k_1 = t\left(\Omega^2 r_0 - k\dot{r}_0^2\right)
\]

\[
k_2 = t\left(\Omega^2[r_0 + \frac{1}{8}t\dot{r}_0 + \frac{1}{8}t^2k]\right) - k\left[\dot{r}_0 + \frac{1}{2}k\dot{r}_0^2\right]
\]
Expressing equations 15 and 16 as solutions for the I th time step enables straightforward calculation for the first three time steps. At subsequent time steps 'r' and 'r′' can more accurately be determined by a published Predictor-corrector technique:

\[
\dot{r}_1^p = \dot{r}_{1-4} + \frac{4}{3} t (2 f_{1-1} - f_{1-2} + 2 f_{1-3})
\]

where

\[
f_{1-1} = \Omega^2 r_{1-1} - k r_{1-1}^2
\]

\[
f_{1-2} = \Omega^2 r_{1-2} - k r_{1-2}^2
\]

An iteration starting from the predicted \( \dot{r}_1^p \) produces corrected values:

\[
r_1^{c(1)} = r_{1-2} + \frac{1}{3} t \left[ \dot{r}_1^p + 4 \dot{r}_{1-1} + \dot{r}_{1-2} \right]
\]

\[
\dot{r}_1^{c(1)} = \dot{r}_{1-2} + \frac{1}{3} t \left[ \Omega^2 (r_1^{c(1)}) - k (\dot{r}_1^{c(1)})^2 + 4 f_{1-1} + f_{1-2} \right]
\]

\[
r_1^{c(2)} = r_{1-2} + \frac{1}{3} t \left[ \dot{r}_1^{c(1)} + 4 \dot{r}_{1-1} + \dot{r}_{1-2} \right]
\]

\[
\dot{r}_1^{c(2)} = \dot{r}_{1-2} + \frac{1}{3} t \left[ \Omega^2 (r_1^{c(2)}) - k (\dot{r}_1^{c(1)})^2 + 4 f_{1-1} + f_{1-2} \right]
\]

The program continues this process until convergence results in successive solutions which differ by less than an arbitrary one per cent.

5.3.2 Tangential direction (\( \omega > \dot{\theta} \))

To aid in solving Equation 12, the variable 'r' can sensibly be considered constant over a small step, applying the same reasoning as for 'r' previously.

The equation becomes:

\[
\ddot{\theta} = \Omega^2 (\omega - \dot{\theta})^2
\]

\[
\frac{d\Omega}{dt} = \Omega^2 (\omega - \Omega)^2
\]

(substituting \( \Omega = \dot{\theta} \))
\[ \int \frac{1}{(\omega - \Omega)^2} \, d\Omega = \int r k \, dt \]

\[ \frac{1}{\omega - \Omega} = r k t + d \quad \text{(where } d \text{ is a constant)} \]

\[ t = \frac{1}{r k (\omega - \Omega)} - \frac{d}{r k} \]

at \( t = 0 \), \( \Omega = \dot{\theta}_0 \) \quad \therefore \quad \frac{d}{r k} = \frac{1}{r k (\omega - \dot{\theta}_0)}

\[ \Rightarrow d = \frac{1}{(\omega - \dot{\theta}_0)} \]

\[ t = \frac{1}{r k} \left\{ \frac{1}{(\omega - \Omega)} - \frac{1}{(\omega - \dot{\theta}_0)} \right\} \]

\[ \Omega = \dot{\theta} = \omega - \frac{1}{r k t + \frac{1}{\omega - \dot{\theta}_0}} \]

(17)

and further substituting \( c = \frac{1}{\omega - \dot{\theta}_0} \) gives:

\[ \Omega = \frac{d\theta}{dt} = \omega - \left\{ \frac{1}{r k t + c} \right\} \]

\[ \theta = \omega t - \frac{1}{r k} \log_e(r k t + c) + f \quad \text{(where } f \text{ is a constant)} \]

when \( t = 0 \), \( \theta = \theta_0 \) \quad \therefore \quad f = \theta_0 + \frac{1}{r k} \log_e c

\[ \Rightarrow \theta = \omega t + \theta_0 - \frac{1}{r k} \left\{ \log_e(r k t + c) - \log_e(c) \right\} \]

(18)

Equations 17 and 18 hence provide analytical solutions for \( \dot{\theta} \) and \( \theta \) respectively.
5.3.3 Tangential direction \((\omega < \dot{\theta})\)

Similar working to that in 5.3.2, but from the starting point of Equation 13, results in two further analytical solutions:

\[
\Omega = \dot{\theta} = \omega - \frac{1}{rkt + \frac{1}{\omega - \theta_0}}
\]

(19)

\[
\Rightarrow \theta = \omega t + \theta_0 + \frac{1}{r_k} \{ \log_e(rkt - c) - \log_e(-c) \}
\]

(20)

5.4 DRAG COEFFICIENT

An empirical method of estimating droplet drag coefficient was located in literature by Ingebo\(^{34}\), the expression originating from NACA research concerning droplet and solid sphere behaviour in clouds. Subject to a Reynolds number within the range 6 - 400, and a droplet diameter range of 20 - 120 \(\mu m\), the drag coefficient is approximated simply by:

\[
C_D = 27 \ Re^{-0.84}
\]

(21)

However, the drag coefficients \(C_{Dr}\) and \(C_{Dt}\) (referred to in Equations 11, 12 and 13) cannot be obtained solely by direct insertion of the radial and tangential components of velocity into Equation 21, as the relationship between drag force and velocity is not one of simple proportionality. Appropriate modifying factors are therefore introduced in the program.

In this model the Reynolds numbers are updated for every time step by utilising the radial and tangential velocities calculated at the end of the previous, and a check is made that they fall in the permissible range. Previous workers\(^{17,82}\) appear simply to have disregarded the lower constraint placed on Reynolds number by Ingebo, and by so doing imply the occurrence of very high drag coefficients as \(Re\) becomes small, followed by a step change to zero drag coefficient at \(Re=0\). In this case, when \(Re\) proves to be less than 6 the drag coefficient is instead arbitrarily frozen at the value pertaining to \(Re=6\), in the apparent absence of a correlation for drag coefficient which holds true at these low Reynolds numbers.

5.5 EVAPORATION MODEL

The step-by-step trajectory calculation has to be modified to reflect the rate of change of mass caused by evaporation. It is assumed that the change in velocity of the vaporized mass during the time interval is small and that the effects of mass variation can be approximated by updating the droplet mass at the start of each new iteration. A routine by Williams\(^{78}\) was
adapted for this purpose by embodying a velocity correction factor to compensate for forced convection, his original work having considered the droplet to be stationary within quiescent air. A limitation which has remained is that the spray is considered to be composed of a single component fuel (the fuel data originally embedded in the program applied to n-heptane), whereas any normal petrol has numerous components of significance. It should be noted that the current program treats in-cylinder pressure as atmospheric, simply because it is for this condition that the incorporated fuel data is valid. Likewise, initial fuel temperatures other than 20°C will require the fuel density value incorporated in the trajectory element of the routine to be adjusted accordingly.

In obtaining his equations for droplet behaviour, Williams made the following assumptions:-

a) Evaporating droplet has spherical symmetry
b) Quasi-steady state
c) Large distance between droplets (hence no influence from surrounding droplets)
d) Uniform droplet temperature
e) Constant surrounding air temperature
f) No chemical reaction
g) No radiation
h) Shear work and kinetic energy negligible.

He then adopted an established theoretical approach as used by Spalding\textsuperscript{70}, involving application of Fick's Law of Diffusion to yield an expression for $G_{fs}$, the rate of phase change of liquid leaving unit area of the fuel surface.

$$G_{fs} = \frac{\Gamma_{vap}}{r_{fs}} \ln \left[ 1 + \frac{Y_{vap}}{1-Y_{vap}} \right]$$

(kg/m\textsuperscript{2} s) \hspace{1cm} (22)

For the present case of relative motion exists between the droplet and surrounding air, a corrective multiplier must be applied to Equation 22. Ranz and Marshall\textsuperscript{65} describe an empirical mass-transfer number derived from analysis of experimental data on mass-transfer rates for spheres:-

$$N_{Nu} = 2.0 + K_1 (Sc)^m (Re)^n$$

(23)

Where: $K_1 = 0.6$, $m=1/3$ and $n=1/2$
In arriving at this number, comparable to the Nusselt number for heat transfer, use has been implied of Reynold's heat-mass transfer analogy, which allows the thermal and concentration gradients within the boundary layers (controlling heat and mass transfer respectively) to be taken as identical, hence allowing the mass transfer rate to be calculated using the equation for heat transfer rate; Aquino\textsuperscript{1} found such an analogy to be acceptable. Ranz and Marshall restricted their main study to a Reynolds number range of 0 to 200, but demonstrated also that the range could be extrapolated as far as Re = 1000 with safety. When Re approaches zero, $N_{Nu}$ in Equation 23 tends correctly to 2.0. However, any multiplier applied to $G_f$ must have unity value at zero Reynolds number, in order to be without effect at quiescent conditions where no compensation for forced evaporation is required. The whole term is therefore divided out by two to give a consistent modifier.

5.5.1 Fuel properties for evaporation model

Fuel-dependent aspects of the evaporation routine were revised in order to extend simulation capabilities to include SBP3 at atmospheric pressure. The data was obtained through the kind assistance of the BP Research Centre, where it was determined by sample analysis that a fair 'single component' approximation to the characteristics of this multi-component liquid would be produced by averaging the individual properties of normal-octane, iso-octane and cyclo-octane. For three of the temperature-dependent items (vapour pressure, specific heat and saturated vapour density), the technique adopted was to graph each set of data then utilise the curve-fitting facility of the plotting package to deduce a defining equation. An incidental point concerning such methods, particularly where higher order polynomial fits are involved, is that application of the equation must be restricted to the domain employed during derivation, unless the satisfactory behaviour of the curve fit can be established for other inputs. Incorporation of the resulting equations as program lines then provides automatic calculation of the relevant fuel properties for the prevailing reference or fuel surface temperature. These graphs, together with the supplied data used in devising a representation of SBP3, are included for reference as Appendix VII.

5.6 COMPUTER PROGRAM

Droplet trajectory program "DROPTRAJ" was written in HP-BASIC for an HP-87 minicomputer, with Williams' existing EM-V1-5 Evaporation Program modified as necessary from MS GWBASIC to run as a subroutine. The program uses the solutions described to solve the fundamental equations of motion (11,12 & 13), and prints out the droplet position before incrementing a small step in time and repeating the process. Into this calculation is fed the current droplet diameter, as updated by the evaporation routine.
Instead of reporting droplet diameter at successive time increments until the droplet has, for practical purposes, evaporated entirely, this subroutine now adopts the timebase of the main program steps, and loops once only per step to return the current droplet diameter for use in deriving the next droplet position. Essentially, the calculation is taking the most recently calculated value for rate of change of drop size and multiplying by the time step to deduce the reduction in drop size since the previous time step. The new estimate of droplet diameter follows by subtracting this reduction from the droplet size pertaining to the previous step.

Calculation of the timestep itself is by arbitrary division into 100 equal parts of the time required for one third of the induction stroke to occur, on the basis that it is during this phase of the cycle that the droplet trajectory is of most significance; a screen prompt therefore requests keyboard input of the engine speed which it is desired to simulate. The cylinder bore is embedded in the program (currently 93 mm, representing the 2.9 litre Ford V6 engine), enabling execution to terminate automatically once the droplet has impacted the notional cylinder wall.

Full details of program nomenclature, an outline flowchart, and a program listing are included in Appendix VIII.

5.6.1 Program validation

The trajectory routine was validated individually before consideration of evaporation was added, using temporary program lines to calculate velocities and accelerations respectively from the changes in position and velocity over one timestep. These values were then compared satisfactorily with those computed from the equations of motion, and found to be consistent. Although limited by a paucity of independent experimental data with which to draw comparisons, Williams had previously compared typical EM-V1-5 results with those of El-Wakil et al\textsuperscript{15} and attained reasonable confidence. As compensation for forced convection in no way affects program logic, further checks on the modified evaporation subroutine were deemed unnecessary.

Correct operation of the combined programs was established in two ways. Firstly the air and fuel temperatures were set deliberately low to suppress evaporation, and it was established that trajectory results were as for those attained initially with constant droplet diameter. Secondly, the program was run with no initial droplet velocity and infinitesimally small swirl (to avoid a division by zero in the calculations), yielding the same progress in reduction of droplet diameter as for the evaporation routine working alone from the same initial temperatures. Additionally, back-substitution of results into the original droplet equations continued to give agreement.
5.7 PARAMETRIC STUDIES

The developed program was used to investigate the significance of numerous variables in relation to droplet trajectory during the induction stroke. Particular attention was paid to the effect of initial droplet diameter, in view of the evident benefits to mixture preparation of small droplets during wallfilm measurement work. With the current "DROPTRAJ" program there is scope for varying the following initial inputs:

- Droplet diameter
- Radial position
- Angular position
- Radial velocity
- Angular velocity
- Swirl velocity
- Fuel type
- Air temperature
- Fuel temperature

5.7.1 Choice of input values

To constrain the exercise to manageable proportions it was decided to adopt one realistic set of inputs as a standard, then to vary certain of the variables in turn. The rationale behind these standard values and variations (bracketed) is given below:

a) **Droplet diameter = 50 μm (5 μm/10 μm/25 μm/120 μm)**

These diameters form a representative selection, chosen with a view to allowing the implications of model results to be compared with the experimental measurements of wallfilm (Chapter 4.0), which were undoubtedly a function of droplet diameter. At present 120 μm is the largest droplet diameter which may be simulated, a limitation arising solely from restrictions on the validity of the Ingebo drag coefficient expression.

b) **Radial position = 20 mm (10 mm/30 mm)**

Realistic radii at which a droplet may enter the V6 engine cylinder were easily determined, by direct measurement of the distance between the combustion chamber centre and the rim of the inlet valve.

c) **Angular position = 0°**

This variable was included in anticipation of plotting the trajectory directly from the HP87 using the standard dot matrix printer. In practice, to obtain report quality figures, the polar coordinates of droplet position were entered manually into an Apple Macintosh plotting package (CricketGraph) and interpolated between to indicate the trajectory. For this process it was most convenient to take as a zero datum the angular position at which the droplet first enters the cylinder.
d) **Radial velocity = 0 m/s (10 m/s / 30 m/s)**

The disposition of the Ford V6 inlet tract in relation to the inlet valve appears unlikely to impart a definite radial component of velocity to the incoming fuel droplet. However, to determine the likely effect for cases where the inlet flow is not of a wholly tangential nature, two random velocity values were also tested. Some appreciation of an order of magnitude for these was gained from Heywood30, in which a correlation of ten times mean piston speed is quoted as typifying the radial and axial gas velocities to be expected from a conical jet issuing into the cylinder. At respective crankshaft speeds of 900 rev/min and 1500 rev/min, this would indicate gas velocities of 21 m/s and 36 m/s.

e) **Angular velocity = 180 rad/s (60 rad/s / 1800 rad/s)**

Reading Heywood's gas velocities across to treatment of the present case as purely tangential motion, and taking account of the initial radial position, angular velocities of the droplet about the cylinder centre would be:

<table>
<thead>
<tr>
<th>Radius</th>
<th>10 mm</th>
<th>20 mm</th>
<th>30 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>900 rev/min</td>
<td>2100 rad/s</td>
<td>1050 rad/s</td>
<td>700 rad/s</td>
</tr>
<tr>
<td>1500 rev/min</td>
<td>3600 rad/s</td>
<td>1800 rad/s</td>
<td>1200 rad/s</td>
</tr>
</tbody>
</table>

Although one of the variations of angular velocity is broadly in line with the above suggestions, a much lower 'standard' value of 180 rad/s was chosen, as was a still lower second variation. These are intended to represent the possible situations of injection onto a closed valve, or deposition of a majority of the fuel onto the port walls or valve (as appears to occur in the V6 engine). Either eventuality may deny droplets an angular velocity component, until they re-accelerate again in the combustion chamber to something of the order of swirl velocity.

f) **Swirl velocity = 188 rad/s @ 900 rev/min, swirl ratio 2**

(377 rad/s @ 900 rev/min / 565 rad/s @ 900 rev/min)

A swirl ratio of 2 was taken as standard for the V6 engine, following advice from Ford, and in conjunction with crankshaft speed this defines the swirl velocity. Respective swirl ratios of 4 and 6 are implied in the two variations.

g) **Fuel type = SBP3 (n-Heptane)**

SBP3 'fuel' was simulated throughout, for parity with previous wall film test measurements. One run was performed with n-heptane data, mainly for reasons of curiosity, all other variables being set to the standard case.
h) Air temperature = 20°C (60°C)

For reasons of time the air temperature was left at 20°C throughout, except for one excursion to 60°C. This was intended to simulate a hot start after a period of heat soak, during which underbonnet or in-plenum temperatures inevitably rise considerably above normal stable operating values.

i) Fuel temperature = 20°C

Although potentially a variable, fuel temperature was held constant at what is a moderately high ambient value for the UK, but the effect of variations could easily be explored in future. However, it is commonly held that evaporation rate is influenced far less by initial fuel temperature than by the temperature of the surrounding air, so no dramatic effects on trajectory are to be envisaged.

5.7.2 Simulation results

The program was run for all the described combinations of input in turn. In some simulations the time span of the trajectory prediction (scaled in all cases to one third of the induction stroke duration) was time enough for the droplet to reach the cylinder wall. In other more favourable cases it was not and program line 320 would require very minor amendment if it were desired to know the trajectory during the remainder of the induction stroke. Trajectory results are presented pictorially as Figs. 5.2 - 5.9. In each case the droplet path resulting from the 'standard' combination of inputs is superimposed, and the alteration of one particular variable only is shown within each figure.

Changes of droplet diameter [Fig. 5.2] are shown by the simulation to influence very strongly the droplet trajectory. The inference of these theoretical results is identical to that gained from the wallfilm measurements made using the Ford V6 rig with varying droplet sizes: large droplets rapidly impact the cylinder wall whereas smaller droplets (25 μm or less) readily conform to the swirling flow and remain in the air. Perhaps more significantly still, execution of the 5 μm drop trajectory calculation actually terminated because the imposed 99.5% evaporated condition was reached, with the droplet diameter falling to 0.05 μm at step 66. In contrast, and as shown in Fig. 5.3, a similar evaporation rate per unit surface area applied instead to a 25 μm droplet (5 times the diameter, and 125 times the volume), results in a percentage rate of diameter reduction which is almost negligible in the context of the available cycle time. Evaporation rates of all droplet sizes would naturally be very much higher with gasoline fuel, but the fact remains that small initial sizes will continue not only to have a relative advantage in terms of evaporation, but are also less likely to impact cylinder walls. Indeed, the movement of the 5 μm droplet radially outward during the 7.3 ms
period of simulation before complete evaporation occurred was only 3.1 mm. In a similar total
time (8.9 ms), the 120 μm droplet actually reached the wall and impacted.

As might be expected, introduction of the fuel droplet at progressively larger radii from the
cylinder centre (Fig. 5.4) results in earlier impact with the cylinder wall: after about 80° arc of
travel (taking 7.3 ms) around the cylinder for a droplet introduced at 30 mm radius, as against
120° (over 11 ms) for a droplet entering at the 20 mm 'standard' radius. A start point only 10
mm from the centre entirely prevents wall wetting, by virtue of the fact that the drop
trajectory establishes in a tighter curve. It has been noticed during steady flow tests that the
orientation of port injection on the actual V6 engine most typically yields emergent droplets at
positions corresponding to a radius of 25 - 30 mm, and the simulation confirms this as least
desirable.

The initial radial velocity of the droplet (Fig. 5.5) proved to have dramatic implications for
trajectory, although the observation should be qualified by noting that where an input radial
velocity is high - relative to the droplet angular velocity - so the former will inevitably exert
a powerful influence on the direction taken by the droplet. It is evident from the trajectories
shown on page 127 that any element of radial velocity will be detrimental to the cause of
retaining droplets in the airstream; in particular, the starting radial velocity of 30 ms causes
impact on the nearest wall surface in only 1.1 ms, a mere 3.3% of the time available for
induction.

As might be expected from inspection of the equations defining droplet motion, increase of the
initial droplet angular velocity from the low 'standard' value brings about earlier impact on
the wall, with the droplet diverging only modestly from its initial tangential direction (Fig.
5.6). Although the clear indication from the point of view of wall wetting is that high initial
droplet angular velocities are best avoided, it would appear that this variable has not quite
so sensitive a role in the matter of determining droplet trajectory as droplet diameter and
radial velocity, notwithstanding its direct effect on droplet radial acceleration.

Equally, the swirl velocity of the air rotating in the cylinder may not perhaps be of prime
significance in altering the actual droplet trajectory, but it does markedly affect the time in
which a droplet reaches the wall. The general effect on trajectory of increasing the swirl
velocity (Fig. 5.7, p128) can be gauged from a comparison of the trajectories for 188 rad/s and
565 rad/s swirl velocity, the latter showing a small but positive bias in the direction of the
cylinder centre. With the lowest swirl velocity of 188 rad/s the droplet does not quite reach
the cylinder wall in the overall timescale of the simulation, whereas 377 rad/s produces
impact with the wall at step 60 (20% of induction stroke), and 565 rad/s likewise at step 43 (14.3 % of induction stroke).

In comparing the trajectories obtained with SBP3 and n-heptane fuels (Fig. 5.8) practically no difference is to be seen, as might be expected given that the properties of the two liquids are quite similar; some reassurance was thus gained that no gross errors had occurred in embedding the necessary fuel data. Examination of values on the raw printouts confirms that the n-heptane evaporated marginally more than SBP3, explaining why the trajectory for the former is fractionally more aligned with the main swirling flow.

Essentially the same comment applies to the one variation to 60°C of ambient air temperature, for which it is unexpectedly difficult to resolve any difference from the standard trajectory (Fig. 5.9). On reflection this might have been foreseen, given the low volatility of the fuel and the fact that any influence on trajectory would be second order, arising as a further effect of diameter reduction by evaporation. Operating a similar comparison from a starting point of much smaller initial droplet size would doubtless reveal a divergence in paths, with that for 60°C air temperature tending relatively in the direction of the cylinder centre.

Just as indicated by experimental measurements of wall film, the simulations have shown reduction of droplet diameter to be a most promising approach to minimisation of in-cylinder wallfilm. The model also demonstrates the attainability of orders of magnitude increase in droplet evaporation rate, through modest droplet size reductions such as from 25 μm to 10 μm. As a logical progression from this work, and to form a final phase for the present study, it was decided to devise a mixture preparation system capable of producing very much finer droplets than are associated with current conventional multipoint systems, and to assess the practical benefits on a fully-instrumented test bed engine.
Fig. 5.1
Notation for mathematical model of droplet trajectory

Fig. 5.2
Effect of initial droplet diameter on predicted trajectory
Fig. 5.3    Rate of forced evaporation for SBP3 droplets - effect of different initial droplet diameters compared

* time step 100 corresponds to 11.11 ms

Fig. 5.4    Effect of initial radial position of droplet on predicted trajectory

A = 10 mm
B = 20 mm
C = 30 mm
Fig. 5.5
Effect of initial radial velocity of droplet on predicted trajectory

A = 0 m/s
B = 10 m/s
C = 30 m/s

Fig. 5.6
Effect of initial droplet angular velocity on predicted trajectory

A = 60 rad/s
B = 180 rad/s
C = 1800 rad/s
Fig. 5.7
Effect of air swirl velocity on predicted trajectory

A = 188 rad/s
B = 377 rad/s
C = 565 rad/s

Fig. 5.8
Trajectories of SBP3 and n-heptane droplets compared

A = SBP3
B = n-heptane

Fig. 5.9
Effect of increased initial droplet temperature (SBP3)

A = 20°C
B = 60°C
CHAPTER 6.0
ENGINE TESTS
CHAPTER 6.0

TESTS OF PORT THROTTLED ENGINE

6.1 INTRODUCTION

The consistent message from wallfilm measurement rigs and computer simulations alike has been the desirability of very much smaller droplets than are associated with current multipoint fuel injection systems. A prime question, therefore, is how such an improvement may be obtained, having regard to the severe economic constraints inseparable from passenger car production in a highly competitive industry.

One method of improving the atomisation of a fuel injector would undoubtedly be by straightforward increase of fuel supply pressure, but as well as being to the probable detriment of dynamic range and small-quantity linearity this would certainly require a more costly pump. Alternatively, compressed air might be introduced via a shroud at the injector tip, the high velocity air causing the droplets to break up further as they emerge from the fuel nozzle. In a comparison of different injector types and positions, Bandel et al\textsuperscript{5} confirmed that air assist gave most stable combustion, highest lean limits, and lowest ISFC. Emmanthal et al\textsuperscript{16} also devised a novel air-forced injection system to replace the normal injectors. While air shrouding options are now available from both Bosch and Lucas, a possible problem with this technique is that the additional air supplied to assist droplet break-up may itself be more than sufficient to run the engine at idle and low load conditions. In his undergraduate project at University College, Fry\textsuperscript{21} worked on a solution which modulated the air supply so that it was provided only for the duration of injection.

However, recalling the very favourable droplet size analysis of a carburettor at part load (3.3.1), and the evident improvements conferred by this fuel system on rig tests of in-cylinder mixture preparation (4.4.1), it was decided to design and test on the Ford Zeta engine a fuel system which would capitalise instead on the simple, effective and reliable action of throttle disc atomisation. It was hoped that locating conventional multipoint injectors upstream of individual port throttles would give improved droplet sizes at part load (as with single point injection systems) but retain all the cylinder-to-cylinder distribution benefits of multipoint injection. At high loads there would clearly be no droplet size reduction, but this was thought tolerable as the high load area of the operating map is arguably least sensitive\textsuperscript{5} to mixture quality. Some work already completed at UCL by Williams\textsuperscript{79} certainly provided grounds for optimism, but only later was it discovered that the Ford Motor Company in the USA\textsuperscript{53}, BMW
in Germany, and Toyota in Japan were already working quite independently along similar lines, although perhaps for different reasons.

6.2 PORT THROTTLING

6.2.1 Theoretical benefits of port throttles

While the initial stimulus for investigating port throttling was purely minimisation of low load droplet sizes, further consideration disclosed several other possible benefits from this intake system configuration.

Firstly, the port throttle discs effectively double as swirl control valves at precisely the operating conditions where increasing the swirl of the mixture may be most advantageous, particularly for a four valve engine such as the Zeta in which homogeneity of the charge, and mixing of the air, fuel and residual gases may otherwise be suffering through modest incoming gas velocities at low load. If an engine were to be designed from the outset to embody port throttling it may be feasible to optimise the inlet tract for full load volumetric efficiency with less compromise for swirl generation, and instead rely on the port throttles to create sufficient turbulence. The correspondence between turbulence and flame propagation is well established: Kyriakides and Glover\textsuperscript{40} have for example looked at the relationship between in-cylinder air motion and combustion, and found strong correlation between turbulence intensity and 10 - 90% mass fraction burned crank angle. Nissan\textsuperscript{22} obtained an extension of lean limit by almost two air-fuel ratios, and gave photographic evidence of more rapid combustion to support their claim that deactivation of one port at low load by an 'inlet control valve' gave both improved mixture formation and increased gas motion inside the cylinder.

Another aspect of operation where port throttles would appear likely to show promise is in relation to deterring exhaust backflow (during the valve overlap period) into the low pressure areas of port and plenum at idle and low load. In the valve closed period of a port throttled engine it is to be anticipated that the intake port pressure would recover towards ambient as air flows in from the main plenum past the port throttle. Assuming the volume between inlet valve and port throttle reaches ambient pressure before the valve overlap period, there should be no pressure differential to cause reverse flows into the intake system from the cylinder. This should eradicate any tendency during the subsequent induction to imbalanced cylinder filling at conditions of high manifold vacuum. A necessity is that the volume between the valve and throttle disc must be small enough for full pressure recovery to take place within the limited time available. Inlet valve throttling\textsuperscript{71} would no doubt be the logical continuation of this line of thought, but it may be that a port throttle system in fact represents the closest viable alternative for production.
One further interesting possibility, aired by Newman and co-authors after practical testing and a modelling exercise using the Ford GESIM simulation software, is the likelihood of a significant reduction of pumping losses with port throttling, the suggested mechanism being that the intake of fresh air into the cylinder becomes predominantly an expansion rather than a throttling process. Indeed, a modified pumping loop of reduced proportions was a persistent characteristic in the Zeta engine results to be described.

6.2.2 Design of port throttle unit for test bed engine

One disc per cylinder was decided upon, for the purposes of initial trials. Although the elongated geometry of the ports in the Zeta 2B head would best lend itself to an eight throttle layout, it was considered that the problems of sealing and obtaining equal air flows would be more tractable with four discs only. During Williams' preliminary investigation, which for expediency used an eight throttle adaptation of spare port deactivation units, the collective effect of small air leaks had made it impossible to bring the engine to idle speed wholly by use of port throttles.

An original method of improving disc sealing was devised, and proved sufficiently effective that the engine could readily be stopped solely by shutting the port throttles. When a conventional circular throttle disc is firmly closed, the periphery furthest from the axis of rotation presses hard against the throttle body wall, but diminishing pressure is exerted at the edges nearer the spindle. In the design which was adopted, two flats are cut on opposite sides of the disc, and sliding bushes on the spindle are maintained in positive contact with these under an end-load provided by conical springs. Sealing between spindle and sliding bush, and between bush and throttle body is provided by nitrile 'O' rings. Slight leakage paths do remain, via the lands on the low pressure side of the 'O' rings, but these should be minimal and in any case consistent. Fig. 6.1 indicates the general layout.

From considerations of port pressure recovery the throttle discs in the present design had to be placed as close as possible to the cylinder head, especially given that the cast-in port volume already corresponded to 24.5% of the engine swept volume. A slide throttle system held attractions for minimising volume, but potential difficulties in implementation (and debatable relevance to quantity production) favoured conventional throttle discs with modification as described. The total port to swept volume ratio finally achieved was 31.4%, including the further volume enclosed by the standard heatshield gasket. A computer prediction from the Ford engine simulation model, available after the design was finalised, suggested that for an engine geometry comparable with the Zeta, at 1 bar BMEP and 800 rev/min idle, full port pressure recovery will only occur if this ratio is no greater than around 22%.
With the throttle discs being of necessity so close to the engine there was little scope for a good transformation from the racetrack-shaped port at the cylinder head face to a circular throttle body section, and then back into the standard twinned manifold. In the vertical plane only modest extra divergence is required to accommodate the throttle disc, but in planview a considerable local contraction in port width is unavoidable (Fig. 6.2). As the work was to concentrate on proving the concept at low loads it was considered unimportant that this poor shape seemed likely to impair volumetric efficiency at high load. However, steady flow comparison of the port throttle unit with the standard 2B deactivation unit (throttles and inlet valve fixed wide open in both cases) later disclosed the surprising fact that the former requires less manifold pressure to produce a given air flowrate (Fig. 6.3).

The injectors are oriented such that for all likely part throttle angles the majority of the 15° spray cone will be directed centrally at the lower (downstream) half of the disc; during their work on TBI systems Takeda et al. have previously identified such a geometry as beneficial for minimising throttle body wall wetting. At wide open throttle most of the fuel is expected to impact on the top surface of the disc, as shown in Fig. 6.4, although it is possible that the flow velocities associated with open throttle angles may be sufficient to re-entrain a proportion of the liquid fuel as fine droplets.

In its construction the port throttle unit consists of two major components, each machined from solid aluminium. The part mounting directly on the cylinder head (Fig. 6.5 and Plate 24) has four cylindrical bores for the throttles, and a through hole for their common spindle. Bolted to this, as seen in Plate 25, is a further section which provides a transition into the twinned manifold shape, being locally faced and bored to act also as an upstream mounting for the standard fuel rail and injectors. Downstream of the port throttles the walls were blended by hand (Plate 26) to eliminate abrupt edges from which any liquid fuel might otherwise be sheared, possibly conferring mixture preparation benefits indistinguishable from the throttle disc atomisation advantages mainly under study.

The 43.00 mm diameter throttle discs were sized to give a flow area equal to the total port cross-section at the manifold flange, and used a closing angle of 15°. Couplings join the four individual sections of throttle spindle to allow relative adjustment for initial balancing. Throttles are operated via a micrometer thread adjuster acting on a lever halfway along the spindle.

6.2.3 Test bed installation

The subject Zeta 2B (Plate 27) was an ex-development engine kindly supplied to UCL by Ford, having recently been removed in running order from a test vehicle. Connection to a Heenan &
Froude DPX2 hydraulic dynamometer was made via a gearbox casing containing a clutch. Exhaust gases were piped into a large bore extraction system, with a water manometer connected to a tapping near the EGO sensor to check consistent back pressure in the exhaust throughout. An alternator was fitted, but to ensure a consistent accessory load the battery was given a freshening charge before each session of testing. A standard fuel supply system was arranged, with continuous circulation through the fuel rail from and to the remote tank. Reference air pressure for the regulator was supplied from the plenum as normal. One batch only of proprietary unleaded gasoline to BS7070 was used for all reported tests.

Pressure tappings downstream of each port throttle were connected to individual mercury manometers for ready comparison during set-up. A shut-off valve for use during measurements was fitted at each tapping, as it was found that the tube and manometer volume otherwise appeared to de-stabilise the idle. A further mercury manometer was also connected at the standard plenum tapping. All readings other than pressures were taken directly from the digital display on the 'Cal Consol', essentially a sophisticated form of 'breakout box' which can interrogate the ECU and make temporary changes for test purposes.

6.2.4 Engine management system

The 16-bit EECIV microprocessor unit, the heart of the engine management system, was mounted adjacent to the test bed and connected using the vehicle wiring loom. Only one minor change was required: the throttle position sensor wiring was taken to a spare unit in order that the desired Cal Consol flag could be set independently of actual upstream throttle angle.

Crankcase ventilation was in all cases piped to upstream of the air meter and the Cal Consol address ARCHLK was accordingly set to zero throughout. In general the ISC valve was left disconnected and spark feedback was disabled by setting KSPKNO and KSPKNU to zero. All tests used closed loop fuelling, but except during ignition sweeps the spark timing was almost invariably fixed at the calibrated value for the condition concerned. Cal Consol fuel slope settings were altered to match the calibration of the yellow-capped injectors fitted to this particular engine. All fuel injection timings to be quoted are relative to the end of injection, and are expressed in degrees before TDC at start of induction stroke.

6.2.5 Commissioning and operation of port throttled engine

The normal functional checks on engine and EECIV operation were made before starting each test session. Stoichiometry was confirmed using LAMBSE1, with observation of EGO sensor switching as additional verification of closed loop fuelling. The behaviour of spark timing was checked via address C2. Appreciable r.p.m. drop was noted on ISC disconnection, as would be expected as air supply to the engine is reduced. After fitting the port throttle assembly the throttles were checked for equivalence of air flow by comparison of individual pressure drops.
Slight spindle adjustment brought all within 1.2 cm Hg (1.6 kPa) of the mean (at idle with load control entirely by port throttles), but regular checks on balance were made throughout. Fuel was syringed around injector seals and throttle spindles, but as no detectable speed increase resulted the system was assumed to be free of air leaks.

In all cases the engine coolant temperature was allowed to stabilise at 186°F before attempting to set up a condition. Idle was obtained by manipulation of the standard upstream throttle and/or port throttles to obtain 800 rev/min, the sole external loads on the engine being the driveshaft into the gearbox and the alternator. In standard configuration, a plenum vacuum of 50 cmHg (66.5 kPa) was typical. For tests using port throttles at standard valve overlap, the port depression was deduced by subtracting from this value the prevailing steady plenum vacuum. However, when valve overlap was increased from 20° to 50°, in an attempt to determine whether the port pressure recovery effect might preserve acceptable combustion stability, the reference plenum vacuum for this deduction also changed. The engine was therefore run initially under the sole control of the upstream throttles to determine the new plenum vacuum at idle. However, the plenum pressure pulsations were such that the reading of 29 cm Hg (38.6 kPa) must be regarded only as an estimate.

Dynamometer settings for 1200 and 1500 rev/min road loads on the baseline engine derived from a 60 kPa (45.1 cm Hg) inlet vacuum recommended by Ford as representative for both cases. However, the pressure drop created by the port throttles was too pulsatory to relate to the steady baseline value at the plenum, so for port throttled tests the engine was set instead to the same load as had equated to the specified vacuum in baseline tests.

6.3 DATA ACQUISITION AND ANALYSIS METHODS

6.3.1 Mixture quality assessment

Improvements in mixture preparation, such as it is hoped port throttling will confer, can affect the performance of a fully-warm test bed engine in various ways. While extension of the lean limit would undoubtedly be one probable indicator, this is of little practical value to a catalyst equipped stoichiometric engine such as the Zeta. Cyclic variations in the cylinder pressure and alteration of the composition of the exhaust gas, both of which relate directly to the quality of the combustion process, are considered to be more useful parameters for judging changes in the consistency of idle and low load running.

Fluctuations in the cylinder pressure from one cycle to the next can arise for various reasons. The extent of fresh air and residual gas mixing in the cylinder may differ, most significantly in
the vicinity of the spark plug, while variations in the amount, proportions and quality of air and fuel mixture also have a particularly direct effect on the repeatability of pressure-volume diagrams. Substantial cyclic variations in the combustion process are undesirable because they constrain the designer to adopt a more conservative compression ratio and ignition timing than would be dictated by the 'average' cycle. Allowance must be made for the possibility of an isolated fast burning cycle (which would effectively advance the predetermined ignition timing and increase susceptibility to knock), with some consequent loss of potential performance. A more subjective but equally important aspect of excessive cyclic variation is that driveability and driver satisfaction with NVH levels is likely to be adversely affected.

The most readily obtainable parameter from cylinder pressure measurements is the peak pressure, which over a number of cycles can be processed to give a mean value or standard deviation. Alternatively, these parameters can be divided to give a coefficient of variation (CoV) of standard deviation over the mean, expressed as a percentage.

However, if the pressure data has been accurately phased with the crankshaft position, and certain features of engine geometry such as crank radius and connecting-rod length are known, a great deal more information about each cycle can be extracted. The crank angle at which peak pressure occurs provides graphic evidence of early or late burning and misfires, and the indicated mean effective pressure (IMEP) may be calculated. As for peak pressure, these values are best analysed statistically over many cycles of recorded data. A convenient index for comparison, and one favoured by the Ford Motor Company, is the lowest normalised value of IMEP, being the lowest single IMEP present in the data, expressed as a percentage of the mean IMEP (and here referred to simply as 'LNV'). By definition only one poor cycle in an otherwise excellent series is enough to return an unacceptable LNV, so it should be recognised that this is a most critical judge of engine performance. In certain circumstances the somewhat less sensitive behaviour of standard deviation of IMEP (SDIMEP) may be preferable.

6.3.2 Data acquisition with LabVIEW

The system developed at UCL both to acquire and subsequently analyse cylinder pressure data is based on an Apple Macintosh II desktop computer, using National Instruments LabVIEW 2.06 Instrumentation Software. Instead of conventional programming in text lines a set of "Virtual Instruments" (VI's) is used, each of which is a software routine in itself. Such VI's may also be built up from a number of sub-VI's graphically wired together on the screen, just as text programs might be constructed from subprograms, or real components hard-wired together to create a physical instrument. The software controls interface boards, handles all data manipulation, and provides a means for presenting the data in tabular or graphical form, using a combination of prewritten and user-defined VI's. Each program consists of two levels, the diagram and the panel. The diagram [Fig. 6.6] is the arrangement of VI's which is
compiled to form the executable code, whilst the panel (Fig. 6.7) is the user interface which allows input and output of information in the required form.

The raw output from the piezoelectric pressure transducer was connected first to a Kistler 566 charge amplifier, then the resulting analogue voltage signal was sent 'in parallel' to a digital storage oscilloscope (for convenient observation of the pressure trace) and to the Macintosh computer for data acquisition and post-processing. Associated input/output (I/O) interface cards, one of which incorporates a 12-bit on-board A/D converter, are supplied by National Instruments. The data is written to a sequential file on 80 MB hard disk prior to analysis using a separate routine; with computer memory upgraded to 8 MB the recording capability was 500 cycles on a single-cylinder, at a sample interval of around 1° crank angle, but memory constraints affecting the separate analysis program limit the effective number to 240 cycles. To remain safely within these, 225 cycles was adopted as standard during this work. A major benefit of separating acquisition and analysis is that a large amount of data can be gathered from the engine in any one test session, with the more lengthy analysis taking place at some convenient later time. Elimination of the requirement to set a pressure trace offset before acquiring data also eased the test routine immeasurably. Following some preliminary tests modifications were made to input the offset at the analysis stage, so that re-runs are possible if the best average value is not selected first time. In allocating this offset, the criterion is that the late exhaust part of the cylinder pressure trace be referenced to absolute exhaust backpressure.

One drawback with this otherwise cost-effective data acquisition system is the length of time required to carry out a full analysis on a large number of cycles: around 20 minutes is necessary for the extensive computations involved in processing the data from 225 cycles. As the processing time has an approximately linear relationship with the number of cycles, there is obviously a strong incentive to use the minimum number of cycles that will provide acceptable accuracy for the purpose in hand. Lancaster et al considered this to be strongly dependent on the stability of the engine condition under investigation. They contrasted a highly stable stoichiometric running condition - which needed only 40 cycles to give 99.9% confidence that the population mean differed from the sample mean by no more than 3% - with unstable operation near the lean misfire limit which required 300 cycles for the same confidence. The present use of 225 cycles on a stoichiometric engine seems reasonable alongside these figures, but in any case the preoccupation here is not so much absolute accuracy of values as meaningful comparison of relative results.
6.3.3 Cylinder pressure transducer selection

Established practice at UCL had been to use a Kistler type 6001 piezoelectric transducer, but in addition to the latter being prone to drift, a sudden local dip was sometimes seen to occur towards the end of the expansion trace, often producing such anomalous results as BMEP exceeding IMEP. Advice from Kistler, particularly for accurate work where parameters such as IMEP are to be derived, was instead to use a 6121A1 transducer. With its integral ceramic protection shield, 'transient temperature error' for this transducer is reduced by orders of magnitude relative to the 6001. Muller et al. compared six different transducers by fitting them simultaneously into the same cylinder head. A water cooled transducer of known stable characteristics was used for reference measurements, then engine operating conditions were varied whilst noting deviations of the other transducers from the reference. The best results were obtained with transducers having both ceramic in front of the diaphragm, and water cooled adaptors extending partially in front of the protection shields. However, extreme limitations of access on the Ford Zeta engine cylinder head (in which a majority of the combustion chamber is valve area) dictated use of the 6121A1 as the next best solution, and one compatible with the tapping already existing on this engine in No.3 cylinder (Plate 28).

6.3.4 Data acquisition triggering

The standard cam phase sensor provided the data acquisition system with a crank angle reference, and the motoring method was employed to determine the corresponding trigger angle to set in the LabVIEW analysis package. This technique assumes motored compression curves to be of a polytropic form \(pv^n = k\), implying that with correct pressure-to-volume phasing a logarithmic \(P - V\) plot appears as a straight line (Fig. 6.8). Common practice was to record 20 motored cycles with the throttle closed and find the mean position of peak pressure resulting from an analysis with 0° set as the trigger angle. From this it was simple to calculate the trigger required to give peak pressure at a nominal 0.5° before TDC, which was in turn applied to a log-log plot as a sensitive trial of accuracy. In the event of non-linearity or crossover the trigger angle was finely adjusted and checked again.

6.4 ENGINE TESTING

6.4.1 Objectives

Two main objectives were identified for consideration when acquiring test data from the described Zeta 2B engine:

1. To determine whether the improved mixture preparation produced by atomisation at the port throttles (as observed on the Zeta wallfilm measurement rig) is reflected in
improved combustion stability, relative to the conventionally throttled engine.

2. To establish the lowest pressure drop across the port throttles which would still be of worthwhile benefit to combustion stability, and to identify fuel injection and ignition timings giving maximum stability and fuel economy.

6.4.2 Notes on presentation of results

A prefix was allocated to denote each engine configuration:-

A Base engine
B Port throttles fitted, only upstream throttle used
C Port throttles fitted, both throttles in use
D Injection timing sweep, both throttles in use
E Ignition sweep, both throttles in use
F 50° valve overlap, upstream throttle only
G 50° valve overlap, both throttles in use

Engine condition denoted by digits after prefix:-

8 800 rev/min idle
12 1200 rev/min road load
15 1500 rev/min road load

Test number appears after hyphen:-

A8-26.0 Base engine configuration, 800 rev/min idle, Test No.26 of this type, (valve overlap as standard).

Rare instances of one or two abnormal cycles in a test being plainly attributable to spurious triggering were encountered during post-processing of the data. In order to salvage such results a 'conditioning package' was devised in LabVIEW, which allowed the spurious cycles to be deleted and replaced as required with some of the 'spare' cycles normally available in excess of the 225 required. In such a case the conditioned data file would become A8-26.1.

Salient figures for each test are summarised in the tables of 'Test Conditions' and 'Test Results' included as Appendix IX. Incidentally, all references to IMEP are to be taken as implying the net IMEP, defined as gross IMEP less Pumping MEP.
6.4.3 Baseline results - standard engine

Before disturbing the engine to fit port throttles, designation 'A' tests were first carried out to establish baselines at the three basic test conditions of 800 rev/min idle, 1200 and 1500 rev/min road load. The assessment can only be that idle stability of the engine is less than desired, bearing in mind that the results apply to a single cylinder, rather than to the whole engine, for which Ford's target 400 cycle LNV is at least 75%. It is believed, however, that the stability of the standard engine was considerably improved before the developed version of the Zeta was signed off for production. Scatter was quite prevalent in the results, but a representative figure for LNV would be of the order of 50%, with SDIMEP at around 0.13 bar. Peak pressure variance was typically around 7% and IMEP variance around 20%.

At 1200 rev/min and 1500 rev/min road load the combustion stability was more than acceptable, subjectively as well as via the derived indices of performance. Both conditions attained LNV's of 92% and correspondingly low SDIMEP's of 0.06 bar and 0.05 bar.

6.4.4 Pressure drop across port throttles

The feasibility or otherwise of a port throttle system for mass production quite probably depends on the proportion of pressure drop which the port throttles are required to carry. Obviously the lower this pressure drop, the less rigorous need be the manufacturing tolerances required to ensure an acceptable balance between the port throttles. Two compromises are unfortunately implied in any move to share load control between the four port throttles and the one standard upstream throttle. Droplet sizes are likely to be larger than if control were wholly by port throttles, and port pressure recovery cannot now be complete as it tends towards plenum depression rather than ambient pressure. The first task, after fitting and successfully commissioning the port throttle unit, was therefore to investigate this question experimentally.

Contributing some data points to this process are the 'B' designation tests, during which the port throttle assembly was fitted but carried no pressure drop, load control remaining solely the prerogative of the standard upstream throttle. The unexpected effect in some cases was very slight improvement in stability and economy over the base engine, perhaps in response to the altered injector orientation or modified patterns of turbulence. However, the tests confirmed that the majority of any improvement which might be achieved with port throttles in use would be genuinely attributable to this system, and not simply to some coincidental benefit of the new port shape and injector location. Spark feedback had an adverse effect on LNV and SDIMEP, contrary to the result A8.27.1 for the base engine.
Continuing tests, with the port throttles brought into use, started under the false premise that the best results would be achieved with the maximum port throttle pressure drop. Characteristics of carburettor droplet size as a function of manifold depression after all indicate excellent mixture preparation at the highest pressure differentials across the throttle plate. However, preliminary tests at 800 rev/min idle, controlled solely by port throttles, gave unexpectedly mediocre results. A test during which 25\% of the pressure drop was carried by the upstream throttles suggested a marginal improvement, although the result remained in any case much inferior to base engine performance. A full 'depression sweep' was carried out at an 800 rev/min idle ("C" designation tests), showing behaviour as depicted in Fig. 6.9. In this case the LNV improves gradually as port throttle pressure drop is increased from 0 to 8" Hg, but then falls sharply at the next higher pressure drop increment. It was concluded that a safe value of pressure drop to adopt would be 4.72 " Hg (= 12 cm Hg = 16 kPa), and this was made the standard in all subsequent tests.

Other areas of interest on Fig. 6.9 are the response of peak pressure variance to port throttle depression, and also the effect of the latter on the mean angle at which peak pressure occurs. In both cases the behaviour seems paradoxical when considered alongside the apparent adverse effect of high port throttle depression on stability: peak pressure variance decreases steadily with increasing depression, as does the angle after TDC of peak pressure. The last mentioned effect implies that the combustion event takes place more rapidly, which might indeed be consistent with the increasingly good mixture preparation expected from throttle disc atomisation as depression is increased. However, it may be that faster combustion is inherently less consistent on a cycle-to-cycle basis, or that another effect is at work which negates any advantages of better mixture preparation.

One hypothesis for the second case might involve the concept of port pressure recovery. Quite probably, full port pressure recovery will only occur if the throttle disc is open beyond some critical angle, and this may not be achieved if the port throttle is well closed to produce all or most of the pressure drop. Backflow of exhaust into the intake system would not then be deterred and the consequence might well be a loss of stability just as seen. A second possibility is that with the throttle disc nearly closed, to achieve a high depression, the present scheme of one throttle disc per pair of ports could be prone to delivering unequal mixtures to each port. A more open throttle angle may make the geometry less critical.

With 16 kPa differential across the port throttles at idle, corresponding to only one quarter of the total pressure drop required for this condition, it was encouraging to discover that the pumping loop nonetheless proved to be of modified proportions, just as conjectured in 6.2.1. Typical pressure-volume diagrams for standard and port throttled engines are shown in Fig.
6.10; attention is however drawn to the effect of LabVIEW autoscaling on pressure axes, which must be allowed for in drawing comparisons. It is evident that the port throttled pumping work will be significantly less, by virtue of the air required for induction now reaching the inlet valves in an expansion rather than a purely throttled process. This effectively cuts a corner at the minimum volume end of the pumping loop, reducing the area enclosed and hence the integral $p \, dV$.

6.4.5 Optimisation of fuel injection timing

a) 800 rev/min

With port throttle depression set to the standard deduced in 6.4.4, and spark fixed as per calibration, a succession of "D" designation tests was carried out at 800 rev/min idle. This data is presented in Fig. 6.11, where it may be seen that there existed a clear optimum injection timing of 210° which raised the LNV to 70%, compared with around 60% at the calibrated timing of 272°; SDIMEP fell to less than 0.09 bar. While the comparison may not be entirely fair, since Ford's criteria in selecting the calibrated value are unknown, it is clear that injection timing has a strong effect. From Fig. 6.11, it seems likely that this effect is related to the inlet valve opening and closing events. The optimum stability is achieved with injection being complete immediately before valve closing, a situation in which high flow velocities at the valve may refine the mixture preparation. Injection timings between 238° and 710° imply a fuel residence time on the valve, and indeed the calibrated 272° gives near maximum time. There is no indication of any preference for a short or long residence time, and while the performance is fairly predictable, free from any drastic deteriorations such as occur at 50°, it was concluded that the 210° timing (which avoids residence) was optimum.

b) 1200 rev/min

Another series of "D" tests at the 1200 rev/min road load condition, again with spark remaining as calibrated, produced the results depicted in Fig. 6.12, which are rather different from those at idle. Stability as indicated by LNV and SDIMEP is once again relatively poor at 50° injection timing, but phasing of injection just before IVC no longer produces any sudden improvements as indicated at idle. The best stability is now returned with the moderate residence time implied by injection delays of 500-600°. Another difference is that fuel economy now shows a definite response to injection timing, actually favouring short to moderate residence time, which fortunately is compatible with the decision to settle on 550° as an optimum injection timing at this condition. At this timing it is also notable that the mean peak pressure reaches a minimum about 10% lower than is obtained at some shorter injection delays, although this serves only to give a result comparable to the base engine, rather than an overall reduction.
c) 1500 rev/min

It is questionable whether the engine response to change of fuel injection timing at 1500 rev/min road shows any real trends. Fig. 6.13 suggests that the situation is one of experimental scatter which can be made to look dramatic by greatly expanded scales. Although the BSFC might at first appear to be showing some dependency, a check with the relevant table of Test Conditions (Appendix IX) shows the recorded differences in FUELPW1 to be smaller than can realistically be resolved from the flickering digital display of the Cal Consol. In the absence of a positive optimum the injection timing at 1500 rev/min was therefore left at the calibration value of 272° in subsequent tests.

6.4.6 Optimisation of ignition timing

a) 800 rev/min

Optimisation of ignition timing at idle took place using the 210° fuel injection timing, as determined in part (a) of 6.4.5 above. In the resulting Fig. 6.14 there is some inconsistency between LNV and SDIMEP plots, perhaps because the former is as mentioned an extremely sensitive indicator of stability. One interpretation might be that a timing as advanced as 34° is desirable, giving as it does lowest SDIMEP and fuel pulsewidth, but a price would be paid in worse peak pressure variance and higher peak cylinder pressures. It is understood that Ford do not in any case seek to use MBT timings for idle, so as to leave latitude for spark feedback control. Results were therefore considered satisfactory at the calibrated 20° spark test point, these being usefully more stable and economic than without port throttles, as is revealed by the comparative Figures 6.15 and 6.16.

b) 1200 rev/min

Perhaps because of the intrinsically greater stability at this faster condition, the results in Fig. 6.17 yield very much clearer trends than at idle. An optimum ignition timing appears to be 18°, with which stability is maximised and BSFC simultaneously minimised. The only reservation might be that this timing is sufficiently close to a sudden turn-up in fuel consumption that a value nearer 20° might be more prudent. Even so, this would still represent a significant retard from calibration, with probable attendant reductions in NOx formation rates through lower peak gas temperatures, and implies that the port throttles are encouraging faster combustion. The port throttled stability at this condition is distinctly better than at other configurations, including that of the base engine, although it is perhaps unwise to derive too much encouragement from improvements to LNV's which may well already be highly acceptable to a driver. At this condition there is in fact a penalty
associated with the gain of stability, being the unexplained slight increase of BSFC from 0.6494 kg/kWh (base engine) to around 0.705 kg/kWh.

c) 1500 rev/min

Essentially the same behaviour with changing spark timing was experienced at this condition as for tests at 1200 rev/min. Fig. 6.18 shows that retarding as far as 18° would again be in order, although to be more conservative much the same performance could still be produced at 20° or 22°. This retard has the same effect of reducing mean peak pressures as in b) above, which is probably why the peak pressure variance once again registers less favourably than in standard configuration. No BSFC penalty is apparent at this speed, indeed quite the opposite effect can be claimed, values being around 7% less than the baseline value.

6.4.7 50° Valve Overlap

Having confirmed that improved combustion stability and fuel economy result from a port throttled fuel injection system, it was decided to set up a particularly demanding engine condition to demonstrate the efficacy (or otherwise) of the port pressure recovery effect in deterring exhaust backflow into the intake. In this DOHC engine the inlet and exhaust cams are on separate belt-driven camshafts, permitting straightforward adjustment of valve timing to simulate the characteristics of camshafts designed to give a longer valve overlap period. In changing the latter from 20° to 50° the camshaft adjustments were made symmetrically.

It was immediately evident that the increase in valve overlap caused spectacular degradation of engine idle stability. This was as anticipated, given that power output considerations are likely to induce the designer to opt for the maximum practical overlap as standard. The most pragmatic treatment of these results is therefore to assess the port throttle in a 'damage limitation' role, by contrasting port throttled and un-port throttled results, instead of focussing closely on absolute values of LNV and SDIMEP.

It was found that with port throttles wide open, and control entirely by upstream throttle, the engine was difficult to start and most reluctant to run below 900 rev/min without stopping, even with the ISC valve connected. Engine speed profile 'A' in Fig. 6.19 (from test F8-2.0) is representative of this situation. In contrast 'B' (test G8-1.0) shows the very much lower minimum speed attainable when port throttles were employed using the same pressure drop, ignition and injection timings as were found optimum at standard overlap.

Even with port throttling, however, the stability at normal idle speed as indicated by LNV and SDIMEP was poor enough that it was felt that injection timing optimisation would be of little significance. Testing thus concentrated on the ignition sweep shown in Fig. 6.20.
Reference to a 'peak' might be misleading, since the LNV values are without exception negative, but it is apparent that stability reaches some sort of maximum at 20° spark, as was concluded for the standard overlap case. Unexpectedly, peak pressure variance and mean peak pressures are distinctly lower than for the base engine, but fuel pulsewidths with the increased overlap are somewhat higher. Fig. 6.21 contrasts the essential 50° overlap behaviour for the cases with and without port throttles, showing the important result that port throttling has improved stability by an order of magnitude. Fuel pulsewidths with and without port throttling were respectively 2.26 ms and 2.79 ms.

One-off port throttled tests were performed at 1200 rev/min and 1500 rev/min (G12-1.0 and G15-1.0) to give some appreciation of stability improvement with load. Although in both cases stability was greatly inferior to standard results (LNV's ~79% vs 90%), the deterioration was perhaps contained within acceptable levels. There are also unexpected bonuses of much lower mean peak pressure, and comparable or improved BSFC values, as compared to those recorded at standard overlap.

6.4.8 Implications of port throttle tests

It is felt that experience with the port throttled Zeta engine has borne out the inferences of both previous wallfilm rig and mathematical modelling work, namely that improvement in mixture preparation is likely to be a powerful enough effect to translate directly into measurable benefits on the test bed. Although the revised fuel injection system was designed principally to achieve better mixture preparation, certain real or potential advantages are associated with its configuration.

Engine stability and fuel economy improvements were especially evident in the idle case. With suitable optimisation of injection timing and spark the LNV for idle at 800 rev/min was raised from approximately 50% to 70%, at the same time gaining a small fuel consumption reduction (no doubt partly due to lower pumping losses) of the order of 3%. At 1200 and 1500 rev/min road loads the desirability of port throttling is less because base engine stability is already at acceptable levels, but at 1200 rev/min small benefits, particularly to SDIMEP, were seen, and at 1500 rev/min a fall in BSFC of 6% compared to the base engine was noted. However, if port throttles have a role at these road load conditions it may be less for stability and economy than for possible NOx reductions: more rapid combustion typically allows retard of calibrated spark by 7°, giving lower peak cylinder pressures.

These advantages would be to little avail if such a system could not be implemented at a realistic cost. In this regard the surprise discovery has been that only a modest proportion (around 25%) of the manifold pressure drop need be created by the port throttles. The throttle
disc sealing measures taken during this work can now be seen to have been over elaborate: a conventional throttle body and disc made to normal production standards should give adequate sealing. Balance problems between individual throttles are also greatly de-sensitised by sharing the overall pressure drop in favour of the standard upstream throttle. If a vacuum source is needed for other vehicle services it may still be possible to provide this by tapping the main plenum chamber. The only slight theoretical reservation about biasing the depression in these proportions is that the prevention of exhaust backflow by port pressure recovery may be compromised, as the port will effectively recover only to the plenum pressure which remains below that of the exhaust.

Although it is recognised that exhaust emissions are perhaps the most critical consideration in current engine design, a conscious decision was made not to collect port throttled emission data at this stage, out of concern that the limited measuring equipment on hand might be inadequate for authoritative results. However, trends which have been seen, such as the feasibility of retarding spark timing, and improved idle fuel economy, hopefully would imply reduced emission levels. Improvements accruing from better mixture preparation would be particularly desirable, as they are likely to apply also at conditions of cold starting and initial running, where catalysts will not have reached their operating temperature and levels of emissions tend to be very high.

Finally, an interesting and coincidental possibility for the future is that port throttle mixture preparation and port pressure recovery effects could perhaps be directed towards regaining acceptable stability on high performance engines with increased valve overlap periods. At the present time it can only be said that the effect of not using port throttles at 50° overlap was all too apparent. Undoubtedly the most telling demonstration at 50° overlap was the subjective reduction of vibration to near normal idle levels when the port throttles were gradually closed from being wide open.

OVERALL CONCLUSIONS

Work towards better understanding and improvement of mixture preparation processes is of course no new activity for SI engine researchers, but there has been an increase in interest since the advent of emissions legislation, accompanied also by a shift in emphasis of which the present work is symptomatic. Formerly, effort tended to concentrate on the preparation and transportation processes occurring within manifolds and ports, where a central fuel system (usually a carburettor) supplied a number of cylinders. Designers aimed above all to obtain an equal and consistent division of fuel between cylinders, in which process a spray of fine
droplets, or even a vaporised mixture, was found to be highly advantageous. The fuel preparation of multipoint fuel injection systems received less attention as it was thought that there was little to be gained, accurate geometric distribution no longer depending on spray quality.

However, it has gradually become apparent in the more recent literature that improving the mixture preparation of multipoint systems is likely to aid cold starting and be of value to emissions in the critical warm-up period before the catalyst becomes fully active. Engine testing during the present work has given definite indications that improved mixture preparation from a port throttled multipoint injector system will even show benefits on a fully warm engine, in the form of reduced cyclic variations in cylinder pressure at idle and low load running conditions. Continuation of these developments, now under way, promises further improvements in engine performance.

The significant conclusions of this study are itemised below for convenient reference:-

(a) Extensive use of the described facilities for rig testing of fuel systems has confirmed that the use of representative fuels, which are by definition highly flammable, need give no undue grounds for safety concern, providing appropriate precautions are taken in design and operation of the test area. The successful Fuel Systems Test Facility at UCL shows that the cost of establishing the latter need not be prohibitive, contrary to some industrial opinion.

(b) The Malvern Particle Sizer has shown itself to be of value in fuel systems research, subject to awareness by the user that misleading indications of large droplets can arise in measurements of evaporating sprays. Caution should also be exercised in interpreting the Smd values for sprays where extremely small droplets are indicated.

(c) A comparison of droplet size distributions immediately downstream of the the two most universal types of fuel system, carburettor and multipoint injector, shows the injector to be producing droplets of a larger order than desirable for the combustion process. At light loads the carburettor provides extremely fine droplets, maximising evaporation, but its performance deteriorates drastically under high-load conditions to give droplet sizes comparable with those of the injector.

(d) Injector orientation was found to have a very significant and potentially adverse effect on mixture preparation quality in the cylinder, for the specific case of the Ford V6 engine hardware tested.
(e) Although air velocity in the Ford V6 valve curtain area was reasonably uniform around the valve, the mixture was seen to emerge from only a limited area under fuel injection operation.

(f) The dominant mode of mixture preparation observed in the Ford V6 cylinder with standard fuel system during rig tests was cylinder wallfilm. A computer simulation of anticipated droplet trajectory suggested that reduction in fuel droplet size to the order of 25 μm, or less, would go some way towards rectifying this problem. In-cylinder preparation was indeed found to be drastically improved when smaller droplets (Smd < 10 μm) were provided upstream of the valve on the motored V6 rig.

(g) In further rig tests, based on the Ford Zeta four valve engine, the proportion of fuel existing as wallfilm was found to be relatively insignificant. All traces of cylinder wallfilm were eliminated when an entirely new port throttled fuel injection system was substituted for the standard system, and the mixture quality in cylinder was visibly improved.

(h) Trials of the port throttle system on a running Zeta engine established that the improved mixture preparation at low load gave quantifiable benefits to combustion stability and fuel consumption at idle, whilst also allowing retard of spark, with a presumed reduction in NOx.

Further to the above conclusions, the following five points are to be recalled in any assessment of the original contributions of this work towards the general knowledge of mixture preparation in automotive spark-ignition engines:-

1. A good deal of work in the literature appears to have concentrated on analysis and modelling of air flow behaviour in cylinders, frequently with little or no reference to the presence of fuel. Alternatively, researchers have simply experimented on a running engine with different methods of mixture preparation and observed the consequences, then attempted to rationalize them. A comprehensive investigation would be correct in drawing on both such approaches, but in this work the extra dimensions of direct observation, photography and droplet size measurement have been added, both upstream and downstream of the inlet valve. The numerous test rigs involved in applying these methods are all original designs.

2. In particular, the porous cylinder wall rig for wallfilm measurement appears to have been regarded as novel, and as such this method and the results from it have attracted the attention of Ford, Ricardo and Hitachi, amongst other commercial concerns.
3. The computer program "DROPTRAJ" is entirely original, excepting the evaporation subroutine for which all credit is due to Williams.

4. Various problems with Malvern Analyser software and result interpretation came to light as a direct result of exercising a critical approach during the reported work. As certain of these defects appear to have escaped even long established users of the instrument, their rectification (via a widespread free issue revision of software and manual) can justly be claimed as a form of original contribution.

5. The idea of port throttles as a means of load control evolved through a number of investigations involving TBI units at UCL, and came to be regarded there as an original configuration. Later receipt of a paper written by Ford personnel in the USA, and the recent implementation by other companies of similar tactics on production vehicles, makes plain the fact that others, probably equally independently, had been working on the same concept.
Fig. 6.1 Schematic of port throttle disc sealing method

Fig. 6.2 Schematic to show incompatibility of Zeta 2B port shape with four-disc port throttle system
Fig. 6.3  Pressure loss characteristics of de-act and port throttle units (comparative steady flow tests using the same Zeta 2B cylinder head and manifold, inlet valve at maximum lift)

![Graph showing pressure loss characteristics](image)

Fig. 6.4  Injector orientation relative to port throttle discs

![Diagram showing injector orientation](image)
Fig. 6.5 Workshop drawing for basic machining of port throttle body
Fig. 6.6
Typical diagram for a LabVIEW 'Sub-VI', in this case serving to produce graphical and numerical outputs from an input array.

Fig. 6.7
Part of Labview 'front panel', showing typical presentation of cylinder pressure analysis.

**FILE REFERENCE**
800 r/min, idle

**MEAN PEAK PRESSURE**
7.1062

**SD PEAK PRESSURE**
0.6468

**MEAN PEAK PRESSURE C.A.**
11.9

**SD PEAK PRESSURE C.A.**
5.05

**MEAN IMEP**
0.7768

**SD.IMEP**
0.1179
Fig. 6.8
Motored pressure-volume diagram plotted on logarithmic axes: approximately straight line compression trace (bracketed) results only when correct value of trigger angle is set in cylinder pressure data analysis routine.
Fig. 6.9 Optimisation of pressure drop ($\Delta P$) across port throttles
(800 rev/min idle, injection timing = 272°, spark timing = 20°)
Fig. 6.10 Area of pumping loops compared for base and port throttled Ford Zeta engine at 800 rev/min idle (note different scales on pressure axes)
Fig. 6.11 Optimisation of port throttled fuel injection timing (800 rev/min idle, $\Delta P = 4.7$ Hg, spark timing $= 20^\circ$)
Fig. 6.12 Optimisation of port throttled fuel injection timing
(1200 rev/min, ΔP = 4.7" Hg, spark timing = 27°)
Fig. 6.13 Optimisation of port throttled fuel injection timing
(1500 rev/min, ΔP = 4.7″ Hg, spark timing = 28°)
Fig. 6.14 Response to alteration of ignition timing (800 rev/min idle, AP = 4.7 Hg, injection timing = 210°)
Fig. 6.15 SDIMEP and LNV compared for each engine configuration, 800 rev/min idle

Fig. 6.16 Fuel pulsewidths compared for each engine configuration, 800 rev/min idle
Fig. 6.17 Response to alteration of ignition timing
(1200 rev/min, ΔP = 4.7 Hg, injection timing = 550°)
Fig. 6.18 Response to alteration of ignition timing
(1500 rev/min, ΔP = 4.7" Hg, injection timing = 272°)
Fig. 6.19 Comparison of engine speed profiles, with 50° valve overlap

Fig. 6.20 Optimisation of port throttled ignition timing, at 50° overlap
(nominal 800 rpm idle, injection timing = 210°, ΔP = 4.7” Hg)
Fig. 6.21  Relative improvement of idle stability due to port throttles (nominal 800 rpm idle, injection timing = 210°, ΔP = 4.7" Hg)

KEY:
A = fixed spark, no ISC, port throttled
B = ISC and spark feedback, port throttled
C = fixed spark, no ISC, upstream throttle
Plate 24 Port throttle unit: view on face adjoining cylinder head

Plate 25 Port throttle unit, showing transition to suit twinned manifold
Plate 26 Detail of port throttle disc and hand blending of wall contour

Plate 27 General layout of Ford Zeta engine test cell
Plate 28 Location of pressure transducer in No.3 cylinder, also showing micrometer thread adjustment of port throttle angle
APPENDICES
APPENDIX I : REFERENCES


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78. Williams, P.A. A computer program to simulate the evaporation trends of fuel droplets from SI engine mixture preparation devices. UCL internal report, January 1991.

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Several further causes of discrepancy in Malvern Analyser readings were encountered during this research, additional to the concerns previously expressed regarding beam refraction by an evaporating spray, and also those relating to the extreme sensitivity of Smd to small droplets. These discrepancies resulted both from software problems and also signals (again probably spurious) which indicated the presence of very small droplets at conditions where the spray sizes were known to be large.

1. It was first noticed that the values of Sauter Mean Diameter (Smd) and Specific Surface Area (SSA) as tabulated in the M4.1 results presentation did not reliably obey the simple relationship implied by the definition of the two parameters, namely that SSA = 6/Smd. Although the manufacturers responded to UCL's report of this problem by issuing a revised version of software (6.1), the consistency of Smd and SSA continued to be unacceptably poor.

2. Further discussions between UCL and Malvern Instruments, concerning the mismatch between Smd and SSA, disclosed that each was being calculated independently, rather than by straightforward division of one parameter to find the other. The fundamental methods used for calculating the derived diameters and areas were therefore considered in some detail.

3. It was determined that the approach of the Malvern software is first to calculate the values of droplet percentages in each size band of the results table, then solve an equation for each band to give directly the percentage of droplets by volume in each. SSA is then calculated from the data in the table, taking the geometric mean of the band boundaries as the characteristic diameter for each size band. It was established by hand calculation, and also with the aid of a BASIC routine run on the HP87 computer, that the Malvern calculation of SSA was in fact correct, thereby isolating Smd as the doubtful parameter.

4. Smd is said to be calculated in the software by a process of interpolating and integrating the tabulated data to produce the cumulative plot, and thence the series of characteristic dimensions describing the spray. Subsequent to version M4 series software issues the manufacturers have revised the interpolation procedure to improve accuracy, resulting in Smd and SSA now being largely consistent.

5. In checking the performance of the improved Smd calculation at UCL (M6.10 software) a different kind of problem became apparent. Occasionally the tabulated results will indicate a
very small percentage of droplets in the bottom bin, with still smaller droplets below the lower cut-off size of the bottom bin (at which small diameters the diffraction theory may be invalid). An example of such a case is shown below, where it is also clear that the histogram presentation does not reflect these table entries.

In practical terms the lifetime before complete evaporation of such small droplets would be very short indeed, and as they represent an infinitesimal proportion of the fuel in the spray they might be considered of no significance. However, two problems are raised: firstly, whether such droplets in fact exist at all, secondly the powerful effect which they might be expected to exert on the Smd calculation if included in it, bearing in mind the points made on p.46 and in Fig. 2.9.

6. Referring to the general form of the histogram above, it is thought to be highly unlikely that droplets of sub 2.4 μm diameter are being formed when the main atomising process is producing the majority of the fuel in the form of droplets with diameters well above 50 μm, thus their physical existence is strongly doubted. Whether the indications arise because of some sort of anomalous diffraction/refraction effect, reflections of reflected light from the viewing passage windows, or an extraneous light effect is presently unknown.
7. What has now been confirmed, however, is that Malvern Instruments selected 2% as a threshold of volume response below which the volume was considered spurious and was ignored in the integration process of early software releases, explaining the apparent inconsistency referred to in 5. and 6. above. The present software (version SB.09) now transforms the whole of the volume distribution, without restraint by any threshold value.

8. It may be concluded that the Malvern Analyser does generally give accurate measurements of droplet and particle size distributions, but that some superseded versions of software were liable to produce errors in certain characteristic dimensions even though the basic histograms were in themselves substantially correct. Users of current software would be advised to exercise vigilance in order to ensure that there are no spurious large-droplet signals (caused by evaporation effects) or spurious small-droplet signals (caused by as yet unidentified effects).
APPENDIX III: MALVERN ANALYSER READINGS ON LATEX AND ALUMINA PARTICLES OF KNOWN SIZE

Instruments M6.10 Date 23-02-1989 Time 15:37

DVB Latex (9.50 micron) : 1 drop 30,000 sweeps.

System number 2196 Diode es965

Malvern Instruments MASTER Particle Sizer M6.10 Date 23-02-89 Time 15-37

<table>
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<th>% under in band</th>
<th>Size (um)</th>
<th>% under in band</th>
<th>Result source: MCHECK</th>
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<td>100.0</td>
<td>17.7:</td>
<td>38.9:</td>
<td>Focal length = 100 mm.</td>
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<td>100.0</td>
<td>15.2:</td>
<td>37.3:</td>
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<td>35.0:</td>
<td>Volume distribution</td>
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<td>33.6:</td>
<td>Beam length = 14.3 mm.</td>
</tr>
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<td>30.8:</td>
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<td>28.5:</td>
<td>Volume Conc. = 0.0019 %</td>
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<td>Log. Diff. = 4.75</td>
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<td>6.3:</td>
<td>24.3:</td>
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<tr>
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<td>8.5:</td>
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</tbody>
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- 178 -
**DVB Latex 40.8 micron (old stock) 30,000 sweeps**

<table>
<thead>
<tr>
<th>Size (um)</th>
<th>% under in band</th>
<th>Size (um)</th>
<th>% under in band</th>
<th>Result source</th>
<th>Sample</th>
<th>Visc.</th>
<th>Volume distribution</th>
<th>Beam length</th>
<th>Coll.</th>
<th>Obscuration</th>
<th>Volume Cone.</th>
<th>Log. Diff.</th>
<th>Model</th>
<th>Focal length</th>
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<td>0.1338 sq.m./cc</td>
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<td>99.9</td>
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<td>99.9</td>
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<td>0.0</td>
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<td>99.9</td>
<td>0.1338 sq.m./cc</td>
</tr>
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<td>99.9</td>
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<td>0.00</td>
<td>99.9</td>
<td>99.9</td>
<td>0.1338 sq.m./cc</td>
</tr>
</tbody>
</table>

**System number 2196 Diode es965**

**Malvern Instruments MASTER Particle Sizer**

**Date 23-02-89 Time 17:16**

**Focal length = 100 mm.**

**Experiment type oil**

**Beam length = 14.3 mm.**

**Volume distribution**

**Volume Cone. = 0.0127 %**

**Log. Diff. = 5.77**

**Model indp**

**D(v,0.5) = 44.3 um**

**D(v,0.9) = 50.2 um**

**D(4,3) = 45.1 um**

**D(3,2) = 44.8 um**

**Span = 0.2**

**Spec. surf. area = 0.1338 sq.m./cc.**
-53/45 micron sieved alumina catalyst, 30,000 sweeps.
-90/+75 micron sieved alumina catalyst, 30,000 sweeps.
APPENDIX IV: STEADY FLOW FUEL INJECTION TEST CONDITIONS

Air and fuel flow data are based on the Ford Scorpio 2.9 litre V6 engine, this being the source of the EV1.3A injector employed.

Port intake temperature $16.5^\circ C \pm 1^\circ C$

<table>
<thead>
<tr>
<th>Simulation</th>
<th>Air Flow</th>
<th>Depression</th>
<th>Spray Duration</th>
</tr>
</thead>
<tbody>
<tr>
<td>A 900 rev/min idle</td>
<td>6.2 cfm</td>
<td>45.7 cm Hg</td>
<td>5.94 ms</td>
</tr>
<tr>
<td>B 1500 rev/min road load</td>
<td>16 cfm</td>
<td>30 cm Hg</td>
<td>6.64 ms</td>
</tr>
<tr>
<td>C 2000 rev/min WOT</td>
<td>58 cfm</td>
<td>~ 0</td>
<td>13.5 ms</td>
</tr>
</tbody>
</table>

Calculation of conditions:

A Air consumption of the 2.9 litre V6 engine at idle was quoted by Ford as 9.34 cfm. Dividing by six therefore gives the mean air flow per cylinder, but as induction is only one quarter of the engine cycle this figure is in turn multiplied by four to approximate the air flow rate during induction. In the absence of other data, the port depression was set to 45.7 cm Hg, as in the idle case for the carburettor work.

The spray duration has been calculated to give a realistic air-fuel ratio of 14:1 if the mean air flow were to be used; however, the fourfold increase in air flow rate causes the actual AFR in the port to be four times leaner than this.

B As for A, except that Ford provided a measured manifold depression. The spray duration was calculated (again using a suitable flow calibration graph) on the basis of the mean air flow rate, but with an AFR of 16.5:1.

C An overall engine air consumption of 87 cfm was deduced from an assumed volumetric efficiency of 85%. Thereafter the calculation was for A, but based on an AFR of 15:1.
### APPENDIX V: SURFACE TENSION AND VISCOSITY DATA FOR HYDROCARBONS

<table>
<thead>
<tr>
<th>&quot;Fuel&quot;</th>
<th>Density @ 20°C kg/l</th>
<th>Surface Tension @ 20°C dyn/cm</th>
<th>Dynamic Viscosity @ 20°C cP</th>
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</thead>
<tbody>
<tr>
<td>n-decane</td>
<td>0.7283</td>
<td>24.9</td>
<td>0.918</td>
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<tr>
<td>n-octane</td>
<td>0.7018</td>
<td>23.2</td>
<td>0.540</td>
</tr>
<tr>
<td>o-xylene</td>
<td>0.8749</td>
<td>31.8</td>
<td>0.805</td>
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<tr>
<td>Toluene</td>
<td>0.8665</td>
<td>28.9</td>
<td>0.593</td>
</tr>
<tr>
<td>Gasoline</td>
<td>0.7278</td>
<td>21.0</td>
<td>0.368</td>
</tr>
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</table>

### APPENDIX VI: CARBURETTOR TEST CONDITIONS

Intake temperature 16.5°C ± 1°C

<table>
<thead>
<tr>
<th>Simulation</th>
<th>Throttle</th>
<th>Air Flow</th>
<th>Manifold Vacuum</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. 750 idle</td>
<td>1.2° open</td>
<td>4.21 cfm</td>
<td>45.7 cm Hg</td>
</tr>
<tr>
<td>2. 1500 road load</td>
<td>14.5° open</td>
<td>13.0 cfm</td>
<td>45.0 cm Hg</td>
</tr>
<tr>
<td>3. -</td>
<td>19.1° open</td>
<td>14.5 cfm</td>
<td>16 cm Hg</td>
</tr>
<tr>
<td>4. -</td>
<td>46° open</td>
<td>16.0 cfm</td>
<td>1.1 cm Hg</td>
</tr>
<tr>
<td>5. 2000 WOT</td>
<td>WOT</td>
<td>17.6 cfm</td>
<td>~0</td>
</tr>
<tr>
<td>6. -</td>
<td>WOT</td>
<td>39.0 cfm</td>
<td>~0</td>
</tr>
</tbody>
</table>


Sources of carburettor test conditions

1. Air flow and manifold vacuum values from previous measurements on an idling 1300 cc engine. Throttle opening of 1.2° was found compatible with this flow and depression.

2. Air flow scaled down by 1300/1700 from existing 1500 rev/min road load data for a 1700 cc engine-vehicle combination. Manifold vacuum as for the 1700cc engine. Throttle angle as necessary to yield 13.0 cfm/45.0 cm Hg.

3. This test point was created as an intermediate between 2 and 4, when a significant change in droplet size distribution was seen to occur between the latter. It does not attempt to simulate a specific engine operating condition.

4. Condition 4 sets the primary throttle 46° open (i.e. secondary just opening) and air flow at 16 cfm - Manifold vacuum follows from these. No specified operating condition is being reproduced, but 16 cfm at this throttle angle is a realistic proportion of a 1300cc engine’s wide open throttle air consumption at 2000 rev/min - which would be around 20 cfm.

5. At this condition, with both primary and secondary throttle discs wide open, an engine’s air consumption would be divided between the two. Applying an assumed volumetric efficiency of 85% to the 1300cc swept volume at 2000 rev/min produces a total air consumption of 39 cfm. This has been crudely apportioned in the ratio of throttle disc areas (1.2S:1P) to arrive at 17.6 cfm for the primary tested alone.

6. For comparison with 4 above, and to see the effect of a much increased airflow, this condition departs from realism and passes the full 39 cfm air consumption through the primary passage only.
## APPENDIX VII: HYDROCARBON DATA AS AVERAGED TO SIMULATE SBP3 PROPERTIES

<table>
<thead>
<tr>
<th></th>
<th>n-octane</th>
<th>iso-octane</th>
<th>cyclo-octane</th>
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<tr>
<td><strong>MOLECULAR WEIGHT</strong></td>
<td>114.2</td>
<td>114.2</td>
<td>112.2</td>
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<td><strong>CRITICAL TEMPERATURE</strong></td>
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<td>543.9</td>
<td>647.2</td>
</tr>
<tr>
<td><strong>CRITICAL PRESSURE</strong></td>
<td>2.488</td>
<td>2.563</td>
<td>3.560</td>
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<tr>
<td><strong>BOILING POINT</strong></td>
<td>398.8</td>
<td>372.4</td>
<td>422.0</td>
</tr>
<tr>
<td><strong>MELTING POINT</strong></td>
<td>216.0</td>
<td>166.0</td>
<td>287.4</td>
</tr>
<tr>
<td><strong>15°C</strong></td>
<td>15°C</td>
<td>15°C</td>
<td>15°C</td>
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<tr>
<td><strong>25°C</strong></td>
<td>25°C</td>
<td>25°C</td>
<td>25°C</td>
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<tr>
<td><strong>B.Pt</strong></td>
<td>B.Pt</td>
<td>B.Pt</td>
<td>B.Pt</td>
</tr>
<tr>
<td><strong>LATENT HEAT OF VAPORISATION</strong></td>
<td>kJ/kg</td>
<td>kJ/kg</td>
<td>kJ/kg</td>
</tr>
<tr>
<td></td>
<td>362.3</td>
<td>307.1</td>
<td>361.7</td>
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<tr>
<td><strong>DENSITY (liquid)</strong></td>
<td>0.7044</td>
<td>0.6970</td>
<td>0.8398</td>
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<tr>
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<td>0.6961</td>
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<td><strong>Cex</strong></td>
<td>1.173E-3</td>
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<td>1.198E-3</td>
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<tr>
<td><strong>VAPOUR DENSITY (ideal gas)</strong></td>
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<td>kg/m³</td>
<td>kg/m³</td>
</tr>
<tr>
<td></td>
<td>4.831</td>
<td>4.831</td>
<td>4.746</td>
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<td><strong>LIQUID HEAT CAPACITY</strong></td>
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<td>2.026</td>
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<tr>
<td></td>
<td>2.216</td>
<td>2.069</td>
<td>1.914</td>
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<tr>
<td></td>
<td>2.582</td>
<td>2.434</td>
<td>2.504</td>
</tr>
</tbody>
</table>
Partial pressure of SBP3 vapour

\[ y = 1093.7 \cdot 16.99x + 9.88 \cdot 2x^2 - 2.554 \cdot 4x^3 + 2.494 \cdot 7x^4 \]
Variation of $C_p$ with temperature for SBP3

$$y = 0.80003 + 4.2436e^{-3x}$$

$C_p$ (kJ/kgK)

TEMPERATURE (K)
Variation in Saturated Vapour Density for SBP3

\[ y = -1.8022 - 3.5829e^{-2}x + 5.7957e^{-4}x^2 - 2.4562e^{-6}x^3 + 3.3136e^{-9}x^4 \]
APPENDIX VIII : COMPUTER PROGRAM FLOWCHART

START

READ AIR, FUEL & ENGINE CONSTANTS

INPUT INITIAL DROP SIZE & POSITION

INPUT INITIAL DROP VELOCITY

INPUT SWIRL VELOCITY (Ω)

THETADOTO = Ω

INPUT ENGINE RPM

CALCULATE TIMESTEP

SET INITIAL CONDNS FOR EVAPORATION

PRINT INITIAL CONDITIONS

CALCULATE REYNOLDS No.s

RENORAD <= 400 & RENOTAN <= 400

YES

CALC. DRAG COEFFICIENTS

CALCULATE PFS, YFVS

CHECK VALUES ARE IN RANGE

CALC. CURRENT PROPERTIES

CALCULATE GAMFV, GFS

SUBR INITI

SUBR INITI

SUBR FUDAT

SUBR FUDAT

STAGE 1

STAGE 2

STAGE 3

STAGE 4

STAGE 5

SUBR PROPS

SUBR PROPS
PRINT CURRENT POSITION

R >= CYLBORE/2

PRINT 'DROP HAS IMPACTED CYLINDER WALL'

END
## APPENDIX VIII: COMPUTER PROGRAM NOMENCLATURE

### VARIABLES

<table>
<thead>
<tr>
<th>Name</th>
<th>Description</th>
<th>Units</th>
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<td>AIRK</td>
<td>Thermal conductivity of air</td>
<td>kJ/m s K</td>
</tr>
<tr>
<td>BM</td>
<td>Mass diffusion transfer number</td>
<td>-</td>
</tr>
<tr>
<td>CDR</td>
<td>Drag coefficient in radial direction</td>
<td>-</td>
</tr>
<tr>
<td>CDT</td>
<td>Drag coefficient in tangential direction</td>
<td>-</td>
</tr>
<tr>
<td>CPFV</td>
<td>Specific heat of fuel vapour at constant pressure</td>
<td>kJ/kgK</td>
</tr>
<tr>
<td>CPLF</td>
<td>Specific heat of liquid fuel at constant pressure</td>
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</tr>
<tr>
<td>DIAM</td>
<td>Current droplet diameter</td>
<td>μm</td>
</tr>
<tr>
<td>DIFF</td>
<td>Molecular diffusion coefficient</td>
<td>m²/s</td>
</tr>
<tr>
<td>DMIC</td>
<td>Specified initial droplet diameter</td>
<td>μm</td>
</tr>
<tr>
<td>DRFSDT</td>
<td>Rate of change of fuel surface radius</td>
<td>m/s</td>
</tr>
<tr>
<td>DROPAREA</td>
<td>Projected area of fuel droplet</td>
<td>m²</td>
</tr>
<tr>
<td>DROPDIA</td>
<td>Current droplet diameter</td>
<td>m</td>
</tr>
<tr>
<td>DROPMASS</td>
<td>Current droplet mass</td>
<td>kg</td>
</tr>
<tr>
<td>DT</td>
<td>Time step</td>
<td>s</td>
</tr>
<tr>
<td>DTFSDT</td>
<td>Rate of change of fuel surface temperature</td>
<td>K/s</td>
</tr>
<tr>
<td>DUMM2</td>
<td>Specified initial distance of drop from centre of cylinder</td>
<td>mm</td>
</tr>
<tr>
<td>DUMM3</td>
<td>Specified initial angular position of droplet</td>
<td>°</td>
</tr>
<tr>
<td>DUMM4</td>
<td>Temporary (dummy) variable equal to RDOTCORR1</td>
<td>m/s</td>
</tr>
<tr>
<td>DURATION</td>
<td>Time for one third of induction stroke to take place</td>
<td>s</td>
</tr>
<tr>
<td>F1</td>
<td>Temporary variable in Predictor-Corrector routine</td>
<td>-</td>
</tr>
<tr>
<td>F2</td>
<td>Temporary variable in Predictor-Corrector routine</td>
<td>-</td>
</tr>
<tr>
<td>F3</td>
<td>Temporary variable in Predictor-Corrector routine</td>
<td>-</td>
</tr>
<tr>
<td>FRACT</td>
<td>Volume fraction of fuel evaporated</td>
<td>-</td>
</tr>
<tr>
<td>FVK</td>
<td>Thermal conductivity of fuel vapour</td>
<td>kJ/m s K</td>
</tr>
<tr>
<td>GAMFV</td>
<td>Exchange coefficient of fuel vapour in mixture</td>
<td>kg/m s</td>
</tr>
<tr>
<td>GASK</td>
<td>Thermal conductivity of fuel/air gas mixture</td>
<td>kJ/m s K</td>
</tr>
<tr>
<td>GFS</td>
<td>Rate of phase change of liquid per unit surface area</td>
<td>kg/m²s</td>
</tr>
<tr>
<td>GFSQUI</td>
<td>As GFS, but applicable to quiescent conditions only</td>
<td>kg/m²s</td>
</tr>
<tr>
<td>HL</td>
<td>Latent heat of fuel vaporization</td>
<td>kJ/kg</td>
</tr>
<tr>
<td>INITDIA</td>
<td>Initial droplet diameter</td>
<td>m</td>
</tr>
<tr>
<td>K1</td>
<td>Temporary variable in Runge-Kutta working</td>
<td>-</td>
</tr>
<tr>
<td>K1DUMM</td>
<td>Dummy variable to simplify calculation of K1</td>
<td>-</td>
</tr>
</tbody>
</table>
K2  Temporary variable in Runge-Kutta working
K2DUMM Dummy variable to simplify calculation of K2
K3  Temporary variable in Runge-Kutta working
K3DUMM Dummy variable to simplify calculation of K3
K4  Temporary variable in Runge-Kutta working
K4DUMM Dummy variable to simplify calculation of K4
KCONV Coefficient to correct for forced convection
KRAD Equation of motion constant in radial direction
KTAN Equation of motion constant in tangential direction
OMEGA Angular velocity of in-cylinder air swirl rad/s
PFS Vapour pressure at fuel surface kPa
PWRN An exponent used in calculation of FVK
QFS Heat flux through gas phase close to liquid surface kJ/m²s
R Radial distance of droplet from centre of cylinder m
RCHK R calculated directly from equation of motion m
RCORR Current R during Predictor-Corrector iterations m
RCORR1 First approximation to R in Predictor-Corrector m
RDDOTCHK Radial acceleration calc. from successive positions m/s²
RDOT Velocity of droplet in radial direction m/s
RDOTCHK Radial velocity calc. from successive positions m/s
RDOTCORR Current RDOT during Predictor-Corrector iterations m/s
RDOTCORR1 First revision of RDOT in Predictor-Corrector m/s
RDOTO Initial radial velocity of droplet m/s
RDOTPRED Initial estimate of RDOT in Predictor-Corrector m/s
RENORAD Re for radial component of droplet velocity -
RENOTAN Re for tangential component of droplet velocity -
RERR Percent error between R and RCHK %
REY Re for resultant of droplet velocity components -
RFS Fuel surface radius m
RHOAIR Ambient air density kg/m³
RHOFV Fuel vapour density kg/m³
RHOG Gas mixture density kg/m³
RHOLF Liquid fuel density kg/m³
RO Initial value of R m
RPM Crankshaft speed (4-stroke assumed) rev/min
S1 Temporary variable -
S2 Temporary variable -
S3 Temporary variable -
SCH Schmidt number -

- 193 -
T  Time elapsed from start of simulation   s
TD  Difference between fuel and ambient temperatures   K
TFSC  Temperature of fuel surface   °C
TFSK  Temperature of fuel surface   K
THETA  Angular position of droplet   rad
THETADOT  Angular velocity of droplet   rad/s
THETADOTO  Specified initial angular velocity of droplet   rad/s
THETADUM1  Temporary variable to simplify calculation of THETA(I)   -
THETADUM2  Temporary variable to simplify calculation of THETA(I)   -
THETAO  Initial angular position of droplet   rad
TINFC  Air temperature at infinity (ambient)   °C
TINFK  Air temperature at infinity (ambient)   K
TREF  Reference temperature (from 1/3 rule)   K
TS  Timestep between updates of droplet position   s
TYME  Time elapsed since start of evaporation   ms
VOL1  Initial volume of drop x constant   m³
VOL2  Current volume of drop x constant   m³
YAREF  Mass fraction of air @ reference composition   -
YFVREF  Mass fraction of fuel vapour @ reference composition   -
YFVS  Mass fraction of fuel vapour at fuel surface   -

CONSTANTS

A  Constant for fuel, used to calculate PFS   -
AIRDENS  Ambient air density   kg/m³
AIRVISC  Ambient air viscosity   kg/ms
AM  Molecular mass of air   kg/kmol
B  Constant for fuel, used to calculate PFS   -
CEX  Coefficient of thermal expansion for fuel   K⁻¹
CYLBORE  Bore of simulated engine   m
DK  Fuel constant used in calculating variable DIFF   -
FM  Molecular mass of fuel   kg/kmol
FUELDENS  Density of simulated fuel   kg/m³
HLFBN  Latent heat of vaporization @ TFBN   kJ/kg
PATM  Atmospheric pressure   kPa
RHOF28  Liquid fuel density @ 288.6 K   kg/m³
TFBN  Boiling temperature of fuel   K
TFCR  Critical temperature of fuel   K

- 194 -
APPENDIX VIII : COMPUTER PROGRAM LISTING

10 REM *************************************************
20 DISP "** PROGRAM 'DROPTRAJ' by M.J.Miller, Mechanical Engineering, UCL **"
30 DISP " Incorporating Evaporation Model EM-VI-5 (Mod) by P.A.Williams - "
40 REM *************************************************

45 REM **** VERSION FOR SBP 3 ****
46 REM *************************************************

50 DIM R(101), RDOT(101), THETA(101), THETADOT(101), KRAD(101), KTAN(101), RENORAD(101),
       RENOTAN(101), RDOTPRED(101), RCORR1(101), RDOTCORR1(101), RCORR(101), RDOTCORR(101)

60 DIM RFS(101), DROPDIA(101), DROPAREA(101), DROPMASS(101)
70 CLEAR
80 BEEP @ DISP "Ensure correct AIRDENS and AIRVISC for simulated ambient temp."
90 DISP
100 AIRDENS=1.177 ! [300K/1 atm]
110 AIRVISC=18.46*10^-6 ! [300K/1 atm (dynamic)]
120 CYLBORE=.093 ! [V6/m]
130 FUELDENS=721 ! [SBP3 @ 20 degrees]
140 DISP "INPUT INITIAL DROPLET DIAMETER (microns)"
150 INPUT DMIC
160 INITDIA=DMIC*10^-6 ! [microns to meters]
170 DISP "INPUT INITIAL RADIAL POSITION OF DROP (mm)"
180 INPUT DUMM2
190 RO=DUMM2/1000 ! [initial radial position in m]
200 DISP "INPUT INITIAL ANGULAR POSITION OF DROP (degrees)"
210 INPUT DUMM3
220 THETA0=DUMM3*PI /180 ! [degrees to radians]
230 DISP "INPUT INITIAL RADIAL VELOCITY OF DROP (m/s)"
240 INPUT RDOTO
250 DISP "INPUT INITIAL ANGULAR VELOCITY OF DROP (rad/sec)"
260 INPUT THETADOTO
270 DISP "INPUT SWIRL VELOCITY (rad/sec)"
280 INPUT OMEGA
290 IF THETADOTO=OMEGA THEN DISP "ANGULAR VELOCITY MUST NOT EQUAL SWIRL VELOCITY"
300 GOTO 1230 ELSE 300
310 DISP "INPUT ENGINE RPM"
320 DURATION=10/RPM ! [one third of induction stroke]
330 TS=DURATION/100 ! [one timestep]
340 RENORAD=AIRDENS*RDOTO*INITDIA/AIRVISC ! [initial check on Re]
350 RENOTAN=AIRDENS*RDOTO*(OMEGA-THETADOTO)*INITDIA/AIRVISC ! ["""]
360 IF RENORAD>= 400 THEN DISP "RENORAD EXCEEDS 400" ELSE 370
370 IF ABS(RENOTAN)>= 400 THEN DISP "RENOTAN EXCEEDS 400" ELSE 375
375 PRINT "SBP3 USED FOR THE FOLLOWING PROGRAM RUN........"
380 PRINT "INITIAL DROPDIA (microns)=";DMIC,"RPM=";RPM,"TS=";TS,"OMEGA (rad/s)="
390 GO SUB 1880 ! [set initial conditions for evaporation]
400 PRINT
410 PRINT "R(0) (mm)=";DUMM2,"THETA(0) (degrees)=";DUMM3
420 PRINT "R(0) (m)=";RO,"THETA(0) (radians)=";THETAD
430 PRINT
440 C=(OMEGA-THETADOTO)^-1
450 REM * Now commencing step-by-step calculation *
460 FOR I=1 TO 3 ! [initial Runge-Kutta steps]
470 T=I*TS ! [T is time elapsed]
480 IF I=1 THEN 490 ELSE 500
490 RDOTO(I-1)=RDOTO @ THETADOT(I-1)=THETADOTO @ R(I-1)=RO @ DROPDIA(I-1)=INITDIA
500 GO SUB 520
510 GOTO 650
520 RENORAD(I)=AIRDENS*RDOTO(I-1)*DROPDIA(I-1)/AIRVISC
522 IF THETADOTO>OMEGA THEN 524 ELSE 530
524 RENOTAN(I)=-(AIRDENS*RDOTO(I-1)*OMEGA-THETADOTO(I-1))*DROPDIA(I-1)/AIRVISC
526 GOTO 540
530 RENOTAN(I)=AIRDENS*RDOTO(I-1)*OMEGA-THETADOTO(I-1)*DROPDIA(I-1)/AIRVISC
540 IF RENORAD(I)>= 400 THEN DISP "RENORAD EXCEEDS 400" ELSE 550
550 IF RENOTAN(I)>= 400 THEN DISP "RENOTAN EXCEEDS 400" ELSE 560
560 DROPAREA(I)=PI * DROPDIA(I-1)^2/4
570 DROPMMASS(I)=FUELDENS*PI * DROPDIA(I-1)^3/6
580 IF RENORAD(I)<= 6 THEN CDR=K*:27*RENORAD(I)^-.84 ! [K*:radial drag coeff. corr. factor]
600 KRAD(I)=AIRDENS*DROPAREA(I)*CDR*.5*DROPMASS(I)^-1
610 IF RENOTAN(I)<=6 THEN CDT=6 @ GOTO 630
620 CDT=K1*27*RENO.XAN(I)^-.84 ! [K1: tangential drag coeff. corr. factor]
630 KTAN(I)=AIRDENS*DROPAREA(I)*CDT*.5*DROPMASS(I)^-1
640 RETURN
650 GOSUB 1320 ! [calling evaporation subroutine]
660 REM * Now calculating R(I) and RDOT(I) *
670 IF I=1 THEN 680 ELSE 690
680 RDOT(I-1)=RDOTO @ THETA(I-1)=THETADOT0 @ R(I-1)=R0
690 K1DUMM=THETADOT(I-1)^2*R(I-1)-KRAD(I)*RDOT(I-1)^2 ! [K for R-K]
700 K1=K1DUMM*TS
710 K2DUMM=THETADOT(I-1)^2*(R(I-1)+TS*RDOT(I-1)^2+TS*K1*1.25)-KRAD(I)*(RDOT(I-1)
720 K2=K2DUMM*TS
730 K3DUMM=THETADOT(I-1)^2*(R(I-1)+TS*RDOT(I-1)^2+TS*K1*1.25)-KRAD(I)*(RDOT(I-1)
740 K3=K3DUMM*TS
750 K4DUMM=THETADOT(I-1)^2*(R(I-1)+TS*RDOT(I-1)^2+TS*K3*1.5)-KRAD(I)*(RDOT(I-1)+K3)
760 K4=K4DUMM*TS
770 R(I)=R(I-1)+TS*(RDOT(I-1)+(K1+K2+K3)/6)
780 RDOT(I)=RDOT(I-1)+(K1*2*2*K2+2*K3+K4)/6
790 REM * Now calculating THETA(I) AND THETADOT(I) *
800 GOSUB 815
810 GOTO 870
815 IF THETADOT0>OMEGA THEN 816 ELSE 820
816 THETADUM1=LOG (R(I)*KTAN(I)*T-C)-LOG (-C)
817 GOTO 830
820 THETADUM1=LOG (R(I)*KTAN(I)*T+C)-LOG (C)
830 THETADUM2=THETADUM1/(R(I)*KTAN(I))
835 IF THETADOT0>OMEGA THEN 836 ELSE 840
836 THETA(I)=OMEGA*T+THETAO+THETADUM2
837 GOTO 845
840 THETA(I)=OMEGA*T+THETAO-THETADUM2
845 IF THETADOT0>OMEGA THEN 846 ELSE 850
846 THETADOT(I)=OMEGA-((-R(I)*KTAN(I)*T+C)^-1)
847 GOTO 860
850 THETADOT(I)=OMEGA-(R(I)*KTAN(I)*T+C)^-1
860 RETURN
870 PRINT "R(";I;")=";R(I),"THETA(";I;")=";THETA(I) @ PRINT @ PRINT
890 NEXT I
900 FOR I=4 TO 100
910 T=I*TS
920 GOSUB 520 ! [Subroutine to calc Cd and K]
930 GOTO 940
940 GOSUB 1320 ! [calling evaporation subroutine]
950 REM * Now applying Predictor-Corrector for step 4 onwards *
960 IF I=4 THEN RDOT(I-4)=RDOTO ELSE 970
970 F1=THETADOT(I-1)*2*R(I-1)-KRAD(I-1)*RDOT(I-1)^2
980 F2=THETADOT(I-2)*2*R(I-2)-KRAD(I-2)*RDOT(I-2)^2
990 F3=THETADOT(I-3)*2*R(I-3)-KRAD(I-3)*RDOT(I-3)^2
1000 RDOTPRED(I)=RDOT(I-4)+4*TS*(2*F1-F2+2*F3)/3
1010 RCORR1(I)=R(I-2)+TS*(RDOTPRED(I)+4*RDOT(I-1)+RDOT(I-2))/3
1020 RDOTCORR1(I)=RDOT(I-2)+TS*(THETADOT(I-1)*2*RCORR1(I)-KRAD(I)*RDOTPRED(I)^2+4*F1+F2)/3
1030 DUMM4=RDOTCORR1(I)
1040 FOR J=2 TO 50
1050 IF J=2 THEN RDOTCORR(J-1)=DUMM4 ELSE 1060
1060 RCORR(J)=R(I-2)+TS*(RDOTCORR(J-1)+4*RDOT(I-1)+RDOT(I-2))/3
1070 RDOTCORR(J)=RDOT(I-2)+TS*(THETADOT(I-1)*2*RCORR(J)-KRAD(I)*RDOTCORR(J-1)^2+4*F1+F2)/3
1080 IF J=2 THEN 1100 ELSE 1090
1090 IF ABS (RCORR(J)-RCORR(J-1))/RCORR(J)<.001 THEN GOTO 1110 ELSE 1100
1100 NEXT J
1110 R(I)=RCORR(J)
1120 RDOT(I)=RDOTCORR(J)
1130 GOSUB 815 ! [Calling subr to calc THETA(I) and THETADOT(I)]
1140 PRINT "R(";I;")=";R(I),"THETA(";I;")=";THETA(I) @ PRINT @ PRINT
1150 IF R(I)>=CYLBORE/2 THEN GOTO 1190
1170 NEXT I
1180 PRINT
1190 PRINT "Drop has impacted cylinder wall"
1200 DISP "Drop has impacted cylinder wall"
1210 DUMMS=T1*100 @ DUMM6=DURATION*3 @ DUMM7=DUMMS/DUMM6
1220 PRINT "Percentage of induction stroke before droplet impacted=";DUMM7
1230 END
1260 REM  *******************************
1270 REM  DROPLET EVAPORATION SUBROUTINE
1280 REM  *******************************
1290 REM  SET INITIAL VALUES
1300 IF I=1 THEN RFS(I-1)=INITIA/2 ELSE 1330
1310 VOL=RFS(I-1)^3
1320 IF I=1 THEN FRACT=0 ELSE 1350
1330 TYME=(I-1)*TS*1000 ! [time elapsed since start of evap /ms]
1340 REM SET TIME INCREMENT TO MATCH MAIN PROGRAM STEP
1350 DT=TS
1360 REM STAGE 1
1370 PFS=1099.7-16.99*TFSK+.09864*TFSK^2-.00025545*TFSK^3+.00000024946*TFSK^4
1380 YFVS=(1+(PATM/PFS-1)*(AM/FL))^-1
1390 REM STOP EXECUTION IF YFVS>=1 (TFSK>=TFBN) ELSE PROGRAM WILL CRASH
1400 IF YFVS>1 THEN PRINT @ PRINT "RANGE EXCEEDED, YFVS>=1" @ GOTO 1800
1410 BM=YFVS/(1-YFVS)
1420 TREF=TFSK+.3333*(TINF-TFSK)
1430 REM CHECK TREF AND TFSK ARE IN PERMITTED RANGE FOR SBP3 DATA
1440 IF TREF>353 THEN PRINT @ PRINT "RANGE EXCEEDED, TREF>353K" @ BEEP
1450 IF TREF<263 THEN PRINT @ PRINT "RANGE EXCEEDED, TREF<263K" @ BEEP
1460 IF TREF<263 THEN PRINT @ PRINT "RANGE EXCEEDED, TFSK<263K" @ BEEP
1470 REM STAGE 2
1480 REM CALCULATE PROPERTIES
1490 REM STAGE 3
1500 GOSUB 2180
1510 REM STAGE 4
1520 GFSQUI=GAMFV/RFS*LOG (1+BM) ! [GFS for quiescent case]
1530 SCH=AIRVISC/(RHOAIR*DIFF) ! [Schmidt No.]
1540 REY=(RENORAD(I)^2+RENOTAN(I)^2)^.5 ! [Re for resultant velocity]
1550 KCONV=1+3*SCH^.3333*REY^.5 ! [factor for forced convection]
1560 GFS=GFSQUI*KCONV ! [GFS compensated for forced convection]
1570 REM STAGE 5
1580 DRSFSD=-(GFS/RHOLF)
1590 REM
1600 TD=TINF-TFSK
1630 IF TD=0 THEN QFS=0 @ GOTO 1650
1640 QFS=-(CPFV*GFS*TD/(EXP(GFS*CPFV*RFS/GASK)-1)) ! GFS as at 1550 iff Le=1
1650 REM STAGE 6
1660 DTFSDT=3/(RFS*RHOLF*CPLF)*(-QFS-GFS*HL)
1670 REM END OF CALCULATION STAGES
1690 REM INCREMENT RFS,TFSK,FRACT AND TYME TO NEW VALUES
1700 RFS(I)=RFS(I-1)+DRFSDT*DT ! [RFS=fuel surface radius in m]
1710 DROPDIA(I)=RFS(I)*2 ! [meters]
1720 DIAM=DROPDIA(I)*1000000
1730 PRINT "DROPDIA =";DIAM
1740 VOL2=RFS(I)^3
1750 FRACT=1-VOL2/VOL1
1760 TFSK=TFSK+DTFSDT*DT
1770 TYME=TYME+DT*1000 ! [milliseconds]
1780 REM CHECK FOR ZERO DROP SIZE OR OVER 99.5% EVAPORATED
1790 IF RFS(I)<= 0 OR FRACT>= .995 THEN GOTO 1810 ELSE 1830
1800 BEEP @ BEEP @ BEEP
1810 DISP "EXECUTION TERMINATED - evaporation complete"
1820 BEEP @ BEEP
1830 RETURN ! [back to main program]
1840 REM *******************************************************
2725 REM
1850 REM *******************************************************
2735 REM SUBROUTINE INITI
1860 REM SUBROUTINE TO SET INITIAL CONDITIONS
1870 REM
1880 DISP "INPUT AMBIENT AIR TEMP IN DEG C (20)" @ INPUT TINFC
1890 DISP
1900 DISP "INPUT INITIAL DROP SURFACE TEMP IN DEG C (20)" @ INPUT TFSC
1910 DISP
1920 REM CONVERT TO UNITS OF KELVIN, METERS AND KPA
1930 TINFK=TINFC+273 @ TFSK=TFSC+273 @ RFS=DMIC/2000000
1940 PATM=101.325 @ AM = 28.96
1950 REM INPUT FUEL CONSTANTS
1960 GOSUB 2080
1970 REM PRINT OUT INITIAL VALUES
1980 DISP "INITIAL CONDITIONS ARE:" @ DISP
1990 DISP "AIR TEMP (DEG C)=";TINFC
2000 DISP "FUEL TEMP (DEG C)=";TFSC
2010 DISP @ DISP
2020 REM PRINT OUT INITIAL VALUES
2030 PRINT "AIR TEMP (DEG C)=";TINFC
2040 PRINT "FUEL TEMP (DEG C)=";TFSC
2050 RETURN
2060 REM *******
3175 REM
2070 REM *******

2080 REM SUBROUTINE FUDAT
2090 REM SUBROUTINE FOR DATA SPECIFIC FOR FUEL
2100 REM SBP3
2110 FM=113.5 @ DK=.00000002215
2120 HLFBN=292.7 @ TFBN=397.7 @ RHOF28=721
2130 CEX=.001093 @ TFCR=586.7
2140 RETURN
2150 REM *******

2160 REM
2170 REM *******

2180 REM SUBROUTINE PROPS
2190 REM SUBROUTINE TO CALCULATE PROPERTIES AT REFERENCE TEMPERATURE
2200 REM RESTRICTED VALIDITY EQUATIONS
2210 REM SBP3
2220 CRLF=.80803+.0042456*TFSK
2230 RHOFV=-1.8022-.035829*TREF+.00057957*TREF^2-.0000024562*TREF^3+3.3136E-9*TREF^4
2240 REM AIR
2250 AIRK=.0000045353+.000000071971*TREF
2260 RHOAIR=353.48*TREF^1.0002
2270 REM GENERALLY VALID EQUATIONS
CPFV=(.363+.000467*TREF)*(5-.001*RHDF2B)
S1=1.8*CEX*(TFSK-288.6)
S2=(TFSK-288.6)^2
S3=(TFCR-288.6)^2
RHDF=RHDF2B*(1-S1-.09000001*(S2/S3))
PWRN=2-.0372*(TREF/TBFB)^2
FVK=.000001*(13.2-.0313*(TFBN-273))*(TREF/273)^PWRN
YFVREF=.6667*YFVS
YAREF=1-YFVREF
GASK=YAREF*AIRK+YFVREF*FK
RHOG=YAREF*RHOAIR+YFVREF*RHOFV
HL=HBFBN*((TFCR-TFSK)/(TFCR-TFBN))^0.38
DIFF=DK*TREF^1.823/PATM
RETURN
REM ****************************************************************************
REM last line

Note: For simulation of n-Heptane fuel, program lines must be altered as follows:-

FUELDENS=683 ![n-HEPTANE @ 20 degrees]

REM N-HEPTANE
A=14.3896 @ B=3209.45 @ DM=100.16 @ DK=.0000000234
HLFBN=317.8 @ TFBN=371.4 @ RHDF2B=687.8
CEX=.00124 @ TFCR=540.17

REM N-HEPTANE
CPLF=1.9617-.001699B*TFSK+.0000008902001*TFSK^2
RHDFV=-.29.54+.35215*TREF-.0013982*TREF^2+.0000018519*TREF^3
As a check on calculated values, the following subroutine may be added to either of the simulations:

2430 REM SUBROUTINE TO CHECK RESULTS vs FUNDAMENTAL EQUATION
2453 RDDOT(I)=R(I)*THETADOT(I)^2-KRAD(I)*RDOT(I)^2
2455 THETADDOT(I)=KTHAN(I)*R(I)*THETA-THETADOT(I))^2
2460 RDOTCHK=(R(I)-R(I-1))/TS
2470 RDOTCHK=(RDOT(I)-RDOT(I-1))/TS
2471 THETADOTCHK=(THETA(I)-THETA(I-1))/TS
2472 THETADDOTCHK=(THETADOT(I)-THETADOT(I-1))/TS
2474 RDOTERR(I)=100*((RDOT(I)-RDOTCHK)/RDOT(I))
2478 RDOTERR(I)=100*((THETADOT(I)-THETADOTCHK)/THETADOT(I))
2480 RDOTERR(I)=100*((THETADOT(I)-THETADOTCHK)/THETADOT(I))
2500 PRINT "PERCENT ERROR IN RDOT(";I;")"=";RDOTERR(I)
2510 PRINT "PERCENT ERROR IN RDOT(";I;")"=";RDOTERR(I)
2520 PRINT "PERCENT ERROR IN THETADOT(";I;")")";THETADOTERR(I)
2530 PRINT "PERCENT ERROR IN THETADOT(";I;")")";THETADOTERR(I)
2540 PRINT @ PRINT
2550 RETURN

... and should be called by two additional main program lines 880 and 1160:

880 GOSUB 2430 ! [calling checking subroutine]
1160 GOSUB 2430 ! [calling checking subroutine]
APPENDIX IX : TEST CONDITIONS AND TEST RESULTS FOR ZETA ENGINE TESTING

<table>
<thead>
<tr>
<th>Data File Designation</th>
<th>Date of Test and (time)</th>
<th>Values read from Cal Consol</th>
<th>Exhaust back pressure (in WG)</th>
<th>Manifold vacuum At Plenum (in Hg)</th>
<th>Manifold vacuum At Port (in Hg)</th>
</tr>
</thead>
<tbody>
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Test Conditions - 1200 rev/min road load
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|---------|-----|------------------|-----------------|---------------------|---------------------|-----------------|----------------|-----------------|-----------------|-------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|
| B12-3.0 | 27  | 3.05             | 0.044            | 20.339              | 1.332               | 2.01            | 0.0914         | 0.50           | 3.24            | 92.08       | 2.005           | 13.25           | 0.6444          | 0.2799          | 1.35            | 0.2799          |
| G12-2.0 | 19  | 3.05             | 0.058            | 20.448              | 1.332               | 2.01            | 0.0914         | 0.50           | 3.24            | 92.08       | 2.005           | 13.25           | 0.6444          | 0.2799          | 1.35            | 0.2799          |
| G12-2.0 | 10  | 3.05             | 0.058            | 20.448              | 1.332               | 2.01            | 0.0914         | 0.50           | 3.24            | 92.08       | 2.005           | 13.25           | 0.6444          | 0.2799          | 1.35            | 0.2799          |</p>
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Test Results - 1500 rev/min road load