Microprocessor engine management applied to hydrogen/petrol operation

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MICROPROCESSOR ENGINE MANAGEMENT APPLIED
TO HYDROGEN/PETROL OPERATION

by

ANDREW LAURENCE EMTAGE

A DOCTORAL THESIS

Submitted in partial fulfilment of the requirements
for the award of
Doctor of Philosophy of the Loughborough University of Technology
December 1987

Department of Transport Technology

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This thesis describes the application of microprocessor engine management techniques to the control of the hydrogen/petrol engine. A discussion of the world's fuel resources and the need for energy conservation is followed by a review of the literature related to the use of hydrogen as a fuel. The concept of hydrogen supplementation is introduced and then the work of other researchers in this field is studied in some detail in order to establish the control requirements for hydrogen/petrol operation. A survey of the literature relating to engine management techniques precedes a description of the microprocessor-based controller which was developed for this work. Following this is a description of the engine calibration process which involves the use of specially developed surface-fitting and contour-tracing software. Steady-state operation in the hydrogen/petrol mode resulted in significant energy savings but poor driveability was obtained when the control system was fitted into a Ford Transit Crew Bus. Transient operation during the ECE-15.04 test resulted in a small fuel economy gain but the exhaust emissions exceeded the legislative limits. It was concluded that, although the steady-state performance showed promise, further development of the control system was required to meet the demands of transient operation.
ACKNOWLEDGEMENTS

The author wishes to acknowledge the co-operation and assistance which he has received from many sources. In particular he is indebted to Dr G G Lucas who supervised his work on the hydrogen/petrol engine, to Dr P A Lawson and Dr C H Machin who introduced him to the application of microprocessors in control environments, and to E G Jenkins and Dr Q R Ali for their advice on control theory. The project would have been impossible without the assistance of Mr C D Hackett, Mr D M Harkis, Mr A Broster, Mr G A Knowles and the rest of the technical team in the Department of Transport Technology at Loughborough University.

The author wishes to acknowledge the financial support of the Science and Engineering Research Council who funded the entire project, and also the involvement of the Ford Motor Company and BL Technology Ltd in terms of enthusiasm, advice, engines and vehicles.

Finally the author would like to thank his wife for helping to proof-read this work and for the patience with which she has endured his necessarily long working hours and apparent obsession with computers. Last but by no means least the author wishes to acknowledge his Creator God who has given him the privilege of helping to make the best use of the resources given to mankind.

STATEMENT OF ORIGINALITY

The author declares that the work presented in this thesis is original except where express reference is made to other published material or to private communications.
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<th>Meaning</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td><em>f</em></td>
<td>Residual exhaust gas mass fraction</td>
<td>-</td>
</tr>
<tr>
<td><em>g</em></td>
<td>Gravitational acceleration</td>
<td>m s^{-2}</td>
</tr>
<tr>
<td><em>m</em></td>
<td>Vehicle mass</td>
<td>kg</td>
</tr>
<tr>
<td><em>\dot{m}_a</em></td>
<td>Air mass flow rate</td>
<td>g s^{-1}</td>
</tr>
<tr>
<td><em>\dot{m}_h</em></td>
<td>Hydrogen mass flow rate</td>
<td>mg s^{-1}</td>
</tr>
<tr>
<td><em>\dot{m}_p</em></td>
<td>Petrol mass flow rate</td>
<td>mg s^{-1}</td>
</tr>
<tr>
<td><em>n_a</em></td>
<td>Axle ratio</td>
<td>-</td>
</tr>
<tr>
<td><em>n_g</em></td>
<td>Gearbox ratio</td>
<td>-</td>
</tr>
<tr>
<td><em>n_r</em></td>
<td>Reverse gear ratio</td>
<td>-</td>
</tr>
<tr>
<td><em>n_{1-n_4}</em></td>
<td>Gearbox foward ratios</td>
<td>-</td>
</tr>
<tr>
<td><em>r_r</em></td>
<td>Tyre effective rolling radius</td>
<td>m</td>
</tr>
<tr>
<td><em>s</em></td>
<td>Vehicle speed</td>
<td>m s^{-1}</td>
</tr>
<tr>
<td><em>t</em></td>
<td>Time</td>
<td>s</td>
</tr>
<tr>
<td><em>u_L</em></td>
<td>Laminar flame speed</td>
<td>m s^{-1}</td>
</tr>
<tr>
<td><em>v</em></td>
<td>Voltage</td>
<td>volts</td>
</tr>
<tr>
<td><em>x</em></td>
<td>Constant for flame speed correlation</td>
<td>-</td>
</tr>
<tr>
<td><em>y</em></td>
<td>Constant for flame speed correlation</td>
<td>-</td>
</tr>
<tr>
<td><em>A</em></td>
<td>Projected frontal area of vehicle</td>
<td>m^2</td>
</tr>
<tr>
<td><em>A_D</em></td>
<td>Velocity independent rolling resistance term</td>
<td>-</td>
</tr>
<tr>
<td><em>B_D</em></td>
<td>Velocity dependent rolling resistance term</td>
<td>-</td>
</tr>
<tr>
<td><em>C_D</em></td>
<td>Aerodynamic drag coefficient</td>
<td>-</td>
</tr>
<tr>
<td><em>D</em></td>
<td>Driver demand</td>
<td>%</td>
</tr>
<tr>
<td><em>I_w</em></td>
<td>Rotating inertia of one wheel</td>
<td>kgm^2</td>
</tr>
<tr>
<td><em>I_p</em></td>
<td>Rotating inertia of prop-shaft</td>
<td>kgm^2</td>
</tr>
<tr>
<td><em>I_e</em></td>
<td>Effective rotating inertia of engine</td>
<td>kgm^2</td>
</tr>
<tr>
<td><em>N</em></td>
<td>Engine speed</td>
<td>r/min</td>
</tr>
<tr>
<td><em>P_a</em></td>
<td>Atmospheric pressure</td>
<td>kPa</td>
</tr>
<tr>
<td><em>P_h</em></td>
<td>Hydrogen pressure upstream of hydrogen injectors</td>
<td>kPa</td>
</tr>
<tr>
<td><em>P_p</em></td>
<td>Petrol line pressure upstream of petrol injectors</td>
<td>kPa</td>
</tr>
<tr>
<td><em>R_0</em></td>
<td>Universal gas constant</td>
<td>J kg^{-1} K^{-1}</td>
</tr>
<tr>
<td><em>R_a</em></td>
<td>Gas constant for air</td>
<td>J kg^{-1} K^{-1}</td>
</tr>
<tr>
<td><em>R_h</em></td>
<td>Gas constant for hydrogen</td>
<td>J kg^{-1} K^{-1}</td>
</tr>
<tr>
<td><em>T_a</em></td>
<td>Ambient temperature</td>
<td>K</td>
</tr>
<tr>
<td><em>T_h</em></td>
<td>Hydrogen temperature upstream of hydrogen injectors</td>
<td>K</td>
</tr>
<tr>
<td><em>V</em></td>
<td>Volume</td>
<td>m^3</td>
</tr>
</tbody>
</table>
### Symbol | Meaning | Units
---|---|---
$V_s$ | Swept volume of engine | m³
$\dot{V}_a$ | Air volume flow rate | m³ h⁻¹
$\dot{V}_h$ | Hydrogen volume flow rate | l min⁻¹
$\dot{V}_p$ | Petrol volume flow rate | l h⁻¹
$\alpha_{igt}$ | Ignition timing relative to crankshaft position | ° BTDC
$\alpha_{ijt}$ | Petrol injection timing relative to crankshaft timing | ° BTDC
$\beta$ | Hydrogen energy fraction of total fuel energy | -
$\varepsilon$ | Energy per unit mass ratio of hydrogen to that of petrol | -
$\gamma$ | Ratio of specific heat capacities | -
$\Delta t$ | Fuel injection open duration | ms
$\Delta m$ | Fuel injection fuel mass | mg
$\Delta P_h$ | Hydrogen pressure drop across regulator | kPa
$\eta_t$ | Brake thermal efficiency | %
$\eta_v$ | Volumetric efficiency | %
$\theta$ | Throttle angle (90° = fully open) | -
$\rho_a$ | Air density | kg m⁻³
$\rho_h$ | Hydrogen density | kg m⁻³
$\rho_p$ | Petrol density | kg m⁻³
$\tau$ | Digital sampling period | s
$\phi$ | Equivalence ratio | -
$\omega$ | Engine speed | rad s⁻¹

### LIST OF ABBREVIATIONS

<table>
<thead>
<tr>
<th>Abbrev.</th>
<th>Meaning</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>b.m.e.p.</td>
<td>Brake mean effective pressure</td>
<td>kPa</td>
</tr>
<tr>
<td>b.p.</td>
<td>Brake power</td>
<td>kW</td>
</tr>
<tr>
<td>b.t.</td>
<td>Brake torque</td>
<td>N m</td>
</tr>
<tr>
<td>b.s.f.c.</td>
<td>Brake specific fuel consumption</td>
<td>mg J⁻¹</td>
</tr>
<tr>
<td>e.d.</td>
<td>Net energy density</td>
<td>MJ m⁻³</td>
</tr>
<tr>
<td>i.m.e.p.</td>
<td>Indicated mean effective pressure</td>
<td>kPa</td>
</tr>
<tr>
<td>l.c.v.</td>
<td>Lower calorific value or net specific energy</td>
<td>MJ kg⁻¹</td>
</tr>
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</table>

<table>
<thead>
<tr>
<th>Abbrev.</th>
<th>Meaning</th>
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<tbody>
<tr>
<td>ABDC</td>
<td>After bottom dead centre</td>
</tr>
<tr>
<td>A/D</td>
<td>Analogue to digital</td>
</tr>
<tr>
<td>ATDC</td>
<td>After top dead centre</td>
</tr>
<tr>
<td>Abbrev.</td>
<td>Meaning</td>
</tr>
<tr>
<td>--------</td>
<td>---------------------------------------------</td>
</tr>
<tr>
<td>BBDC</td>
<td>Before bottom dead centre</td>
</tr>
<tr>
<td>BTDC</td>
<td>Before top dead centre</td>
</tr>
<tr>
<td>D/A</td>
<td>Digital to analogue</td>
</tr>
<tr>
<td>DBW</td>
<td>Drive-by-wire</td>
</tr>
<tr>
<td>DMA</td>
<td>Direct memory access</td>
</tr>
<tr>
<td>ECU</td>
<td>Electronic control unit</td>
</tr>
<tr>
<td>EMC</td>
<td>Electromagnetic Compatibility</td>
</tr>
<tr>
<td>EPROM</td>
<td>Erasable programmable read-only memory</td>
</tr>
<tr>
<td>LAN</td>
<td>Local area network</td>
</tr>
<tr>
<td>MBT</td>
<td>Minimum for best torque</td>
</tr>
<tr>
<td>NMI</td>
<td>Non-maskable interrupt</td>
</tr>
<tr>
<td>PCI</td>
<td>Programmable communications interface</td>
</tr>
<tr>
<td>PIC</td>
<td>Programmable interrupt controller</td>
</tr>
<tr>
<td>PIT</td>
<td>Programmable interval timer</td>
</tr>
<tr>
<td>PPI</td>
<td>Programmable peripheral interface</td>
</tr>
<tr>
<td>RAM</td>
<td>Random access memory</td>
</tr>
<tr>
<td>ROM</td>
<td>Read-only memory</td>
</tr>
<tr>
<td>TFVU</td>
<td>Time for volume used</td>
</tr>
<tr>
<td>VAF</td>
<td>Vane air flow (meter)</td>
</tr>
<tr>
<td>VDU</td>
<td>Visual display unit</td>
</tr>
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</table>
Chapter 1

INTRODUCTION
1.1 WORLD FUEL ENERGY RESERVES

The global energy crises of the 1970's highlighted the world's dependency on non-renewable fossil-based fuels. Shortfalls of these fuels in the short term may be caused by geopolitical disturbances and fluctuations in world markets but in the long term their supply will be limited by the depletion of economically viable reserves.

Predictions of fossil fuel reserves vary widely depending upon the models used and the estimates regarding both the discovery of new reserves and the advances in technology which aid economical extraction. Thring[1] suggests that, in the non-Communist world, there will be a shortfall between supply and demand for oil from approximately 1990. Whatever the variations between predictions, it is clear that alternative fuel systems are necessary for the long term and that conservation of fossil fuels in both the short and medium terms will help the transition to these new systems.

Currently, the 'balance' of engine development effort is targeted at the reduction of NO\textsubscript{x} and hydrocarbon exhaust emissions but, due to a growing worldwide concern over the destruction of the ozone layer in the upper atmosphere, this balance may shift towards fuel consumption reduction in order to control the production of carbon dioxide which has been identified as a major factor contributing to the ozone problem.

For these reasons, in an effort to conserve the available petroleum reserves, it is necessary in the short term to develop more energy efficient vehicles and, for the world's medium and long term energy requirements, to develop alternative fuel systems.

1.2 REDUCTION OF AUTOMOTIVE FUEL CONSUMPTION

As a result of tyre rolling resistance (which is proportional to vehicle weight) and the force required during acceleration, vehicle mass has a significant influence on fuel consumption. For this reason, mass reduction is an important factor in the quest for fuel economy and, in an effort to achieve this goal, not only have new materials been developed but the use of finite element analysis techniques has enabled the design of more mass-efficient structures.

As has been mentioned above, the rolling resistance of vehicle tyres has an important effect on fuel economy. This is especially true at slow average speeds such as those encountered in urban driving and efforts are being made by tyre manufacturers to reduce rolling resistance while maintaining acceptable road holding and wear characteristics.
The aerodynamic drag force on a vehicle is proportional both to its drag coefficient and to the square of the head-on relative air velocity. The value of this coefficient is therefore very important for rural and motorway driving. Recent years have seen an increase in the public's awareness of the importance of the aerodynamic drag coefficient and this has been reflected in the styling of modern cars. Efforts in this direction are continuing with attention being paid to details such as flush fitting glazing, the careful entrainment of cooling air and the "cleaning up" of the vehicle underside.

An engine usually has a distinct area in its torque-speed map where thermal efficiency is at a maximum. It would therefore be advantageous to operate the engine as close as possible to this region during normal vehicle operation and, in order to achieve this, the engine must be matched to the vehicle by a suitable 'power train'. Recent years have seen the widespread use of overdrive fifth gears which sacrifice acceleration for fuel economy gains during high speed cruising. Refined designs of Constantly Variable Transmissions (CVTs), with the support of microprocessor based power train controllers, promise to make substantial improvements in fuel economy.

The spark-ignition piston engine is currently the most widely used automotive power unit but there are a number of alternatives having various advantages and disadvantages. The Wankel and Rankine (steam) designs offer some benefits but are unlikely to be widely adopted because of their poor thermal efficiencies. The Stirling engine has a higher thermal efficiency than the spark-ignition engine but not only is it very heavy and bulky but it also has a poor transient response thus making it an unlikely contender. Electric vehicles are unlikely to replace the conventional piston engine, except perhaps in some urban driving, because of their limited range of operation coupled with the relatively high cost of electricity from the national grid. There may be some advantage in using a hybrid power plant in which a small piston engine runs continuously at its peak efficiency while recharging the battery pack. The 'free power turbine' gas turbine unit, with regenerative heat exchanger, may be a serious contender for automotive use because its torque characteristics imply a simple and light transmission system which could maintain a good engine-vehicle match over most of the operating range and, although the thermal efficiency of this design is low when compared to the spark ignition piston engine, the unit's ability to use a wide range of fuels suggests that the automotive gas turbine could help to conserve premium fuels. Furthermore, the development of high temperature ceramics for turbine blades promises to improve the thermal efficiency by raising the maximum cycle temperature. The compression-ignition piston engine has a better thermal efficiency than the spark-ignition engine but its market is limited by the split of fuel products at the refinery. Lucas\[2\] suggests that the spark ignition piston engine is likely to remain the main contender for automobile power units until at least the year 2000.
Spark-ignition piston engine development for fuel economy, within the sometimes incompatible constraints of exhaust emission legislation, has proceeded in several directions. Electronic engine management systems have contributed to the accurate control of engine parameters relating to fuel economy. Microprocessor-based ignition control allows the use of higher compression ratios without the problem of knock. Combustion chamber designs having "fast burn" characteristics and "knock tolerance" are being developed. Cylinder disablement and variable valve-timing promise torque control without the need for throttling of the air intake on spark ignition engines. The use of new materials and finite element analysis can result in lighter and more compact engines. The introduction of ceramic materials has prompted research into the possibility of engines which can operate close to the adiabatic thermal efficiency. In addition, the use of ceramics can substantially reduce the heat loss from the combustion chamber while turbocharging could be used to reclaim energy from the exhaust gases. 'Lean burn' engines may use fuel stratification or some other means to provide combustion in which there is excess air and, provided that sufficiently high flame speeds are maintained, the excess air can result in higher thermal efficiency and lower exhaust emissions.

1.3 ALTERNATIVE FUELS

The use of alternative fuels provides a means of conserving and eventually replacing the use of petroleum based fuels. Liquid fuels are to be preferred because of their compatibility with the existing distribution infrastructure but gaseous fuels are not to be ruled out.

Lucas and Richards[3] suggest that the abundant reserves of coal could be used to synthesize, via the Fischer-Tropsch or other processes, a range of transport fuels similar to those available from petroleum. A high quality grade of petrol can also be obtained by partial oxidation of methane (natural gas or bio-gas) with water.

Methane (natural gas) can be obtained from biomass by anaerobic digestion and is suitable as a fuel but has to be stored either in high pressure cylinders or as a cryogenic. Alternatively, methane could be converted into methanol which is a liquid fuel.

Liquid Petroleum Gas (LPG) consists of fuels such as propane and butane and can be stored as a liquid under pressure. LPG engines can run at higher compression ratios than petrol engines and thereby obtain better thermal efficiencies but displacement of air by the gaseous fuel results in poor volumetric efficiency and hence poor specific power. Although LPG is linked to the availability of petroleum its use may help to conserve the higher quality petroleum based fuels in the short and medium terms.
Spinks\cite{4} points out that some countries, such as Brazil, are suitable for energy farming on a large scale and can produce sugar or starch for conversion to ethanol. However, energy farming results in an inefficient use of land and would therefore be unsuitable for densely populated countries unless waste agricultural produce was used.

Duxbury\cite{5} reports that a West German study of alternative fuels concluded that methanol was the best alternative to petroleum based fuels in the short and medium terms. Methanol can be obtained from wood, coal or methane and its anti-knock characteristics enable high compression ratios to be used. Low flame temperatures lead to low $\text{NO}_x$ emissions while wide flammability limits allow stable operation at low equivalence ratios. The main disadvantages of methanol are its low energy density (about half that of petrol), poor volatility (resulting in cold-starting difficulties), its corrosive effect on some materials used in current engines, and its affinity for water which can cause the engine to stall in extreme cases. Nevertheless, methanol/petrol mixtures are currently being used in West Germany and in the USA and such use represents a way to conserve petroleum based fuels.

Hydrogen is the one important alternative fuel which is yet to be discussed. The raw material for this fuel is in abundant supply and there are no environmental problems associated with its combustion products other than the oxides of nitrogen when it is burned in air.

1.4 HYDROGEN AS A FUEL

Hydrogen should be considered as an energy distribution medium rather than as a conventional fuel. The basic raw material for hydrogen is water which is simply the product of the combustion of hydrogen and oxygen. Any energy available from burning hydrogen must therefore have been derived from some external source such as coal, or nuclear or solar energy. The main advantages of hydrogen are its non-toxic exhaust emissions and the fact that the basic raw material is recycled but the high flame temperature of hydrogen at some equivalence ratios results in significant $\text{NO}_x$ levels in the exhaust products.

1.4.1 Properties

Table 1.1 compares the combustion related properties of hydrogen to those of petrol, Liquid Petroleum Gas (LPG) and methanol. The majority of data for this comparison were obtained from Goodger\cite{6}.

The low lean-limit equivalence ratio of hydrogen permits the use of very lean operating mixtures thus paving the way for high thermal efficiencies, while the resulting low peak cycle
temperatures help to reduce NOx emissions substantially. The high thermal efficiency arises from an increase in the value of $\gamma$, lower heat loss to the engine coolant and reduced dissociation.

The higher flame speed of hydrogen helps to maintain as near constant volume combustion as possible when operating at lean mixtures, while the low minimum ignition energy of hydrogen helps in the rapid initiation of stable, self-sustaining flame kernels thereby reducing cyclic variations during lean combustion. In addition, the low quenching distance of hydrogen minimizes flame quenching in proximity to the combustion chamber walls.

The high spontaneous ignition temperature of hydrogen helps to reduce the risk of fire following vehicle collision when escaping fuel may come into contact with hot engine parts.

Although the high specific energy of hydrogen suggests that the storage tank might be compact and light, this is not the case. Section 1.4.4 indicates that the storage of hydrogen is the biggest problem hindering its adoption as an automotive fuel.

The low energy density of gaseous hydrogen is detrimental in that the consequentially low engine volumetric efficiency leads to poor maximum specific power. However, at extra cost, this could be overcome by adopting direct injection of the hydrogen into the cylinder after inlet valve closure.

<table>
<thead>
<tr>
<th>Property</th>
<th>Petrol (g)</th>
<th>LPG (g)</th>
<th>MeOH (g)</th>
<th>H2 (g)</th>
</tr>
</thead>
<tbody>
<tr>
<td>lean limit equivalence ratio in air</td>
<td>0.58</td>
<td>0.57</td>
<td>0.51</td>
<td>0.1</td>
</tr>
<tr>
<td>max. flame speed in air (m/s)</td>
<td>0.5</td>
<td>0.47</td>
<td>0.55</td>
<td>3.5</td>
</tr>
<tr>
<td>Spontaneous Ign. Temp. (°C)</td>
<td>280-400</td>
<td>405-520</td>
<td>385</td>
<td>574</td>
</tr>
<tr>
<td>min. ignition energy (mJ)</td>
<td>0.55</td>
<td>0.25</td>
<td>0.19</td>
<td>0.019</td>
</tr>
<tr>
<td>quenching distance (mm)</td>
<td>2.84</td>
<td>1.8</td>
<td>1.5</td>
<td>0.6</td>
</tr>
<tr>
<td>net specific energy (MJ/kg)</td>
<td>43.0</td>
<td>46.47</td>
<td>19.94</td>
<td>120.24</td>
</tr>
<tr>
<td>net energy density (MJ/m3)</td>
<td>202</td>
<td>86.4</td>
<td>27.0</td>
<td>10.3</td>
</tr>
</tbody>
</table>

(at 15 °C and 101.3 kPa)

1.4.2 Production and Distribution

Kukkonen goes into some detail concerning viable methods of hydrogen production during the next 15 to 25 years. Basically hydrogen can either be commercially produced by coal gasification or by electrolysis of water using electricity generated from coal, nuclear fission, solar energy or possibly nuclear fusion. The report goes on to point out that the coal
gasification process would be the most economical but a comparison of total coal energy per mile for current technology vehicles puts hydrogen at a disadvantage compared to other coal derived fuels such as methanol, reformed methanol and synthesized petrol.

Long distance distribution of hydrogen would be by pipe-line but the conversion of existing facilities would entail substantial modifications to cater for the problems of hydrogen embrittlement. Vehicle filling stations could offer hydride pack hire, or alternatively, the adoption of chilled water cooling of the hydride containers could significantly reduce the filling time for fixed units.

1.4.3 Engine Adaptation
The utilization of hydrogen for fuelling piston engines has been widely studied[8-32]. One of the first attempts to burn hydrogen in internal combustion engines resulted from the development of transatlantic hydrogen filled dirigibles. It was hoped that by using some of the lift-hydrogen as propulsion fuel there would be no need to re-ballast for fuel usage. Subsequent research by Ricardo concluded that this approach was not feasible because of severe engine knock and backfiring.

The principal causes of backfire are thought to be the high residual gas temperature resulting from slow combustion at very low equivalence ratios and combustion chamber hot spots such as the exhaust valve, spark plug and carbon particles derived from the cylinder wall oil film. The very low ignition energy of hydrogen is thought to be the main cause of this phenomenon but there is some evidence[25] that a chemical effect within the residual gas progressively inhibits the normal combustion reaction, thus causing excessive combustion times so that the residual gas is still burning when the inlet valve opens. This evidence consisted of cylinder pressure records of successive engine cycles leading up to the 'backfiring' cycle. It was observed that the peak pressure progressively decreased for the cycles just preceding the backfire.

Knock is similarly thought to arise from high cycle temperatures at near stoichiometric equivalence ratios as well as from the presence of combustion chamber hot spots. Lucas and Morris[25] have shown that the knock-limited maximum compression ratio of the hydrogen engine varies from 18:1 at $\phi = 0.35$ to 8.5:1 at $\phi = 1.0$. The high cycle temperatures occurring at stoichiometric conditions also give rise to high $\text{NO}_x$ emissions.

Woolley and Hendriksen[17] have shown that the use of sodium cooled exhaust valves, together with water induction (to lower the cycle temperature), help to reduce the problems of backfire, knock and $\text{NO}_x$ emissions.
Homan et al\textsuperscript{[29]} describe the use of direct cylinder hydrogen injection in a spark ignition engine. This precludes the possibility of backfiring and, since the rate of pressure rise can be controlled by the rate of hydrogen injection, it is possible to eliminate knock. High specific power is obtainable since the volumetric efficiency is not impaired by the inclusion of hydrogen in the inlet stream. The only disadvantage is the added cost and complexity of the high pressure injection system.

1.4.4 In-Vehicle Storage
Hydrogen may be stored in a vehicle in three ways. Storage as a compressed gas (17 MPa) is inexpensive and provides for ease of operation but Lucas and Richards\textsuperscript{[3]} have shown that for the equivalent of 16 imperial gallons of petrol, hydrogen storage would weigh about one ton and occupy nearly two cubic meters of space. In addition, the storage of any high pressure gas presents a safety hazard in the event of vehicle collision.

The cryogenic storage of liquid hydrogen for automotive use has been demonstrated by several researchers\textsuperscript{[12,14,21,27,32]}. This type of storage is both complex and expensive and, in addition, loss rates of 1\% per day have to be countered by catalytic oxidation or pressurization to avoid the dangerous escape of hydrogen. The cost and energy associated with the liquification process must also be considered.

The interstitial storage of hydrogen in some metal alloys promises to provide the best solution to the storage problem. Such metal hydride systems have been widely developed\textsuperscript{[14,18,21,33-36]} and provide improved safety over the other two methods of storage. Hydrogen is absorbed into the metal under pressure and is accompanied by the release of heat. Engine coolant water is circulated around the hydride in order to desorb the hydrogen. The heat exchanger used for the desorption process may also be used to cool the metal during the absorption process, thus decreasing the recharging time. Lucas and Richards\textsuperscript{[36]} have modelled the absorption and desorption processes in order to predict the performance of various hydride tank designs. Contamination of the hydride material can be a problem if air is admitted to the system and, should this occur, the only solution is to heat the hydride to a high temperature under a strong vacuum.

Table 1.2 is derived from work by Goodger\textsuperscript{[37]} and compares the three hydrogen storage systems described above with that for an energy equivalent petrol storage system. From this data the cryogenic method of hydrogen storage is the clear winner in terms of both volume and weight.
<table>
<thead>
<tr>
<th></th>
<th>pressurized</th>
<th>Cryogenic</th>
<th>Hydride</th>
<th>Petrol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight</td>
<td>17</td>
<td>3.3</td>
<td>4.8</td>
<td>1.0</td>
</tr>
<tr>
<td>Volume</td>
<td>24</td>
<td>3.8</td>
<td>4.1</td>
<td>1.0</td>
</tr>
</tbody>
</table>

1.4.5 On-Board Generation

One solution to the storage problem is to generate hydrogen from some suitable high energy density parent fuel such as petrol or methanol. Steam reforming of petrol to give hydrogen and carbon dioxide has been demonstrated\cite{10,38} as has the thermal or catalytic decomposition of petrol to give hydrogen and carbon monoxide\cite{39-42}.

The main disadvantage of using petrol as the parent fuel is that the overall conversion is exothermic thereby resulting in a fuel mixture with a lower specific energy than the parent fuel and there is some doubt as to whether the improvements in combustion efficiency can more than offset the converter losses. If methanol is used as the parent fuel the dissociation process is endothermic and uses exhaust energy to increase the specific energy of the resulting fuel mixture. In this way the overall engine efficiency benefits from both the recycling of exhaust energy and the hydrogen derived improvements in combustion efficiency. Work on this type of catalytic converter has been carried out by Adams\cite{43} and Lucas et al\cite{44}.

1.4.6 Safety

Karim\cite{45} compares the safety of compressed methane as an automotive fuel with petrol, propane and hydrogen. This report concludes that hydrogen is not as safe as compressed methane but its author does not consider the use of hydride storage or the danger of accidental shearing of the valve on a high pressure methane container in the event of vehicle collision.

Kukkonen\cite{7} points out that the hydride storage system is extraordinarily safe and releases hydrogen only on the application of heat and, even if the hydride pack was punctured and a flame played on the hydride material, the evolution of hydrogen would be limited by the desorption reactions.

In the event that hydrogen did escape, it would disperse rapidly unlike heavier fuels. It should also be noted that there is little radiation from a hydrogen flame as compared with a hydrocarbon flame and so radiation burns are less likely.
1.5 HYDROGEN SUPPLEMENTATION

From the preceding sections it has been shown that it is desirable to conserve petroleum based fuels in the short and medium terms while development continues on production, distribution and utilization of alternative fuels. Hydrogen has been shown to be a very promising alternative fuel but there are significant problems associated with the operation of hydrogen fuelled vehicles; not the least of which is the storage problem.

The concept of hydrogen supplementation is the improvement of the thermal efficiency of engines using conventional hydrocarbon fuels by supplementing them with relatively small quantities of hydrogen. The addition of hydrogen can extend the lean-limit equivalence ratio while maintaining a sufficiently high flame speed. In this way the main fuel is used more efficiently and only a small quantity of hydrogen needs to be stored. Further improvements can be made by generating the hydrogen on the vehicle, as described in Section 1.4.5, thus eliminating the necessity for separate hydrogen storage.

Considerable research effort[38-44,46-63] has already been spent on the concept of hydrogen supplementation and this work is reviewed in Chapter 2. The optimal control of the two fuelling rates, ignition timing and throttle position necessitates the use of a microprocessor based controller and Chapter 2 therefore also reviews the literature relating to electronic engine management systems. Chapter 3 describes the engine management system developed by the author for this work while Chapter 4 describes the calibration procedure and presents performance data. The main fuel used for this work was chosen to be petrol so that a standard engine design could be adapted with the least number of modifications.
Chapter 2

LITERATURE SURVEY
2.1 HYDROGEN SUPPLEMENTATION

This section gives brief descriptions of papers relating to hydrogen supplementation in respect of engine performance and is presented in chronological order. Two of the papers dealing with the flammability limits and flame speeds of hydrogen + petrol + air mixtures are dealt with in some detail because, although this thesis is not concerned with the thermodynamic modelling of hydrogen-supplemented combustion, an understanding of the influence of these parameters on the performance of the hydrogen/petrol engine is important in relation to the optimization of equivalence ratio and hydrogen energy fraction.

Coward and Jones\cite{46} developed a model to predict the lean-limit equivalence ratio of a mixture of two flammable gasses in air. The model is based on the theory that a lean-limit mixture of one flammable gas in air, when mixed in any proportion with a lean-limit mixture of another flammable gas in air, will also result in a lean-limit mixture. The following equation describes, in the notation of the paper, the requirements for a lean-limit mixture of hydrogen and petrol in air:

\[
\frac{n_h}{N_h} + \frac{n_p}{N_p} = 1
\]  

Equation 2.1 may be rewritten to give the following:

\[
n_p = N_p (1 - \frac{n_h}{N_h})
\]  

(2.2)

It can be shown that the overall equivalence ratio and hydrogen energy fraction of the mixture are given by the following equations:

\[
\phi = \frac{(2.38n_h + 55.62n_p)}{(1 - n_h - n_p)}
\]  

(2.3)

\[
\beta = \frac{n_h}{n_h + \frac{202}{10.3n_p}}
\]  

(2.4)
From Goodger\[6\] $N_h = 0.04$ and $N_p = 0.01$. Having chosen a value of $n_h$, the corresponding values of $n_p$, $\phi$ and $\beta$ for lean-limit operation may be found by evaluating Equations 2.2, 2.3 and 2.4. Figure 2.1 describes the function of lean-limit equivalence ratio versus hydrogen energy fraction as determined by this model. It is clear that the benefits from increasing hydrogen supplementation follow the law of diminishing returns and, because of the costs associated with both hydrogen production and on-board storage, it is likely that only limited supplementation would be cost effective.

Eccleston and Fleming\[47\] compared the practical lean-limit equivalence ratio of a CFR engine when using natural gas and natural gas supplemented with 10 mole$\%$ and 20 mole$\%$ hydrogen ($\beta = 3.25\%$ and $\beta = 7.02\%$ respectively). Supplementing natural gas with hydrogen was found to significantly extend the lean-limit equivalence ratio. The compression ratio was 8:1, engine speed was 1000 r/min and the ignition timing was constant at 15° BTDC.

Breshears, Cotrill and Rupe\[38\] found that the thermal efficiency of a CFR engine could be improved from a value of approximately 25$\%$ to nearly 35$\%$ by supplementing petrol with hydrogen. It was found that the EPA 1977 standard for NO$\textsubscript{x}$ emissions could be met by operating at an equivalence ratio of 0.6 and supplementing the petrol with 5$\%$ by weight of hydrogen ($\beta = 12.8\%$). Hydrocarbon emissions were 4 to 10 times above the limit specified by the EPA 1977 standard. The compression ratio was 8:1.

Stebar and Parks\[48\] carried out tests on a CFR engine and found that supplementing petrol with a 10$\%$ mass fraction of hydrogen ($\beta = 23.7\%$) extended the lean-limit equivalence ratio from 0.89 to 0.55 and reduced NO$\textsubscript{x}$ emissions from 20 g/kWh (5.6 $\mu$gJ$^{-1}$) to 0.27 g/kWh (0.075 $\mu$gJ$^{-1}$). Engine power decreased 30$\%$ but thermal efficiency increased from 33$\%$ to 37$\%$. Hydrocarbon emissions were adversely affected. The compression ratio was 8:1, engine speed was 1200 r/min and the ignition timing was MBT.

Houseman and Hoehn\[40\] carried out tests on a 1973 Chevrolet 350 CID V-8 engine which was fitted into a vehicle. Hydrogen-rich product-gas was supplied from a generator which converted petrol to hydrogen and carbon monoxide. The addition of hydrogen extended the lean-limit equivalence ratio to 0.54 resulting in a decrease in specific fuel consumption of approximately 10$\%$ over the entire speed range. A spark advance of 40° to 50° BTDC was necessary. NO$\textsubscript{x}$ emissions were below the EPA 1977 standard but hydrocarbon and carbon monoxide levels were not.

Parks\[49\] carried out experiments on a CFR engine at 1200 r/min and operating with a compression ratio of 8:1. MBT ignition timing was employed. The equivalence ratio was decreased from 0.9 in
steps of 0.1 and β values of 13%, 23%, 48% and 100% were selected to study the effects of hydrogen enrichment on hydrocarbon and NOX emissions. For all β values other than 100%, the minimum HC emission level occurred at approximately \( \phi = 0.8 \) and then increased until the lean-limit for that β was reached. At constant \( \phi \), the HC emissions decreased as β increased. For all values of β, NOX emissions peaked at approximately \( \phi = 0.85 \) and then fell sharply as \( \phi \) decreased towards the lean-limit. At constant \( \phi \), NOX actually increased slightly as β increased. Figures 2.2 & 2.3 were obtained from this paper and describe these characteristics for HC and NOX emissions. Figure 2.4, also taken from this paper, describes a graphical technique for finding the values of \( \phi \) and β corresponding to maximum allowable values for HC and NOX emissions. The intersection of the line representing the maximum allowable NOX emissions, with the horizontal line representing the maximum allowable HC emissions, defines the optimum operating point with respect to exhaust pollutants. However, this point does not necessarily correspond to the point of maximum thermal efficiency.

MacDonald[50] tested a 400 CID V-8 engine with a compression ratio of 9.6:1. The engine operating point was chosen to be 2.62 bar b.m.e.p. at 1600 r/min and MBT ignition timing was employed. With optimal hydrogen supplementation (β = 31.3%) at \( \phi = 0.52 \), the thermal efficiency was 8% better than the petrol-only value which occurred at \( \phi = 0.76 \). This choice of equivalence ratio for hydrogen supplementation resulted in the NOX emission target of 0.57 g/kWh (0.16 μJ/g·l) just being met. Figure 2.5 was obtained from this paper and describes β as a function of \( \phi \) corresponding to the maximum obtainable thermal efficiency at every value of \( \phi \).

Finegold[42] suggested that thermal efficiency was a strong function of equivalence ratio and, to a large extent, independent of fuel type. This appears to hold true for any fuel up until the point where its lean flammability limit is approached. Figure 2.6 was obtained from this paper and describes the performance of a CFR engine during petrol-only and hydrogen-only operation. Work carried out on a 1973 Chevrolet 250 CID V-8 engine gave rise to the following conclusions: 1) Lean combustion with any fuel can improve fuel economy without the penalty of increased NOX emissions. 2) Lean-burn engines using petrol and hydrogen-supplemented petrol require exhaust after-treatment for hydrocarbon control. Figure 2.7 was obtained from this paper and indicates the spark timing tradeoff between thermal efficiency and NOX emissions. Spark retardation from MBT timing reduces NOX emissions significantly and thermal efficiency is only slightly penalized.

Rauckis and McLean[51] carried out work on a CFR engine at 1000 r/min with ignition timing set at 10° BTDC and running with a compression ratio of 8:1. Equivalence ratio was varied between 1.12 and 0.57 and β was varied from 0% to 28%. Cylinder pressure data were collected and used in a zero-dimensional combustion model to calculate 0-2%, 2-10% and 10-90% mass fraction burn durations. Indicated thermal efficiency and cycle-to-cycle cylinder pressure variation data were also
calculated. It was found that the ignition delay period, as characterized by the 0-2% mass fraction burn duration, was significantly affected by the addition of hydrogen, especially at lean equivalence ratios. The other two combustion periods were not affected to the same extent by hydrogen addition. This confirms the theory that the delay period is dominated by chemical dynamic effects while the main combustion process is dominated by turbulent transport. Cycle-to-cycle cylinder pressure variations were significantly reduced by hydrogen addition. It was found that for constant flame development duration, the cycle-to-cycle variations increased as equivalence ratio decreased. Therefore chemical dynamics as well as flame development time affect cycle-to-cycle cylinder pressure variations.

Jordan\[52\] carried out work on a single-cylinder engine of 298 cm$^3$ and compression ratio of 7.8:1. Comparisons were made between conventional throttled operation, using petrol and un-throttled hydrogen/petrol operation, with three flow-rates of hydrogen including zero. It was reported that very lean combustion was possible because of the increase in combustion rate due to hydrogen addition. This resulted in part-load efficiency improvements and reductions in HC, CO and NO$_X$ emissions. It was noted that lean operation required sufficient hydrogen addition in order to avoid increases in combustion duration but no thought was given to the possibility that, for a given hydrogen flow-rate, there might be an optimum equivalence ratio for each engine load point. In other words, un-throttled operation may not be the optimum method of operation, even if pure hydrogen were used.

Varde\[54\] carried out experiments on a small single-cylinder engine with a compression ratio of 5.9:1. Two flow-rates of hydrogen were used at operating points of 380 kPa at 1700 r/min and 415 kPa at 2000 r/min. It was found that hydrogen supplementation increased engine efficiency and reduced cyclic cylinder pressure variations. Flame speed measurements indicated an increase in flame speed as hydrogen supplementation was increased. The equivalence ratio for peak flame speed decreased as hydrogen flow-rate increased. This was due to the fact that for a constant hydrogen flow-rate, the hydrogen energy fraction increased as equivalence ratio decreased.

Schafer\[SS\] conducted tests on a single-cylinder engine of 298 cm$^3$ and compression ratio of 7.8:1. Methanol was used as the main fuel with hydrogen supplementation at part-load operation. Electronic fuel injection was employed to control the flow-rates of methanol and hydrogen. Full-throttle testing was carried out at 2000 and 4000 r/min. Hydrogen supplementation reduced HC and NO$_X$ emissions, shortened the combustion delay period and increased both the thermal efficiency and combustion rate.

May and Gwinner\[56\] experimented with a 2.8 litre 6-cylinder fuel injected Daimler-Benz engine. A modified D-Jetronic fuel injection system was used to control the flow-rates of hydrogen and
petrol. Un-throttled operation was employed throughout the engine’s operation range during dual-fuel operation. A microprocessor-based controller was developed for the engine so that it could be used in a vehicle. Equivalence ratios between 1.43 and 0.17 and hydrogen energy fractions between 0% and 100% were programmed into the controller to give optimum thermal efficiency and exhaust emissions. The 1982 West German emission limits (ECE) were met but the 1982 USA requirements (CVS) could not be met. It was suggested that partial throttling might help to reduce the total mass of emissions by simply reducing the mass flow through the engine. No attention was paid to the possibility that partial throttling might also help to increase the part-load thermal efficiency.

Richards[57] and Lucas & Richards[58] conducted tests on a BL 1275 cm³ 4-cylinder engine at compression ratios of 8.9:1 and 11.7:1. The hydrogen flow-rate was set to 69.5 mg/s so that the engine would just idle with full throttle and no petrol. Full-throttle operation with hydrogen supplementation was found to increase part-load thermal efficiency. This was attributed to lean operation and higher flame speeds. Lean operation resulted in decreased combustion temperatures, which in turn reduced the heat losses to the coolant and raised the ratio of the specific heats ($\gamma$) and hence increased the air standard efficiency of the engine. Maximum power was found to be slightly penalized during hydrogen supplementation because of the drop in volumetric efficiency due to the displacement of air by the hydrogen. This could be overcome by direct cylinder injection of the hydrogen but would be costly. When the hydrogen flow-rate was increased to 89 mg/s, the part-load thermal efficiency was increased but the maximum power was further reduced. It was suggested that partial throttling might improve the thermal efficiency by increasing the equivalence ratio which would result in higher flame speeds. No attempt was made to study the joint effects of equivalence ratio and hydrogen energy fraction on the performance of the engine. Hydrogen supplementation resulted in a reduction in NOx and CO emissions but HC emissions were found to be high at very low loads where the equivalence ratio approached the lean flammability limit.

May et al[59] carried out work on a Daimler-Benz 2.8 litre fuel injected 6-cylinder engine. Hydrogen was admitted by means of six solenoid-operated injectors situated in the inlet ports. The injectors were timed according to the ignition firing order so that backfiring could be controlled by the opening time of the injectors. A mapped electronic ignition system was employed to provide accurate calibration of ignition timing. Partial throttling was used to limit equivalence ratios to values greater that 0.2, and the hydrogen energy fraction was varied from 0% near full load to 100% at very low loads. Dual-fuel operation resulted in fuel consumption improvements of up to 18% compared with petrol-only operation. HC specific emissions were only slightly improved compared with petrol-only operation but these might be further reduced by increasing the throttling to increase the equivalence ratio. It was suggested that a minimum equivalence ratio of 0.3 instead
of 0.2 should be used. NO\textsubscript{x} and CO specific emissions were considerably reduced compared with petrol-only operation.

Milton and Keck\cite{160} carried out combustion bomb experiments to evaluate flame speeds of hydrogen, acetylene, propane and methane, and mixtures of hydrogen with each of the other gases. The experiments were conducted over a range of temperatures and pressures from 0.5 to 7 atm and 300 to 550 K under stoichiometric conditions. The hydrogen/acetylene burning velocities approximated to the values indicated by proportional averaging of the values for the individual gases. Hydrogen/methane and hydrogen/propane mixtures exhibited a peak in burning velocity shortly after flame initiation, followed by a reduction, and then a second increase in flame speed. No explanation is offered for the double-peak behaviour but the author of this thesis believes that this may be due to the propagation of two flames. Since the hydrogen/oxygen combustion reactions are independent of the combustion reactions for the other fuel, it is possible that the hydrogen combustion will also proceed independently. The first peak in combustion rate may be due to the propagation of the hydrogen/oxygen flame, while the second peak could be due to the propagation of the flame for the second fuel.

Yu et al\cite{61} have developed a model to predict the laminar flame speeds of various hydrocarbon + hydrogen + air mixtures. Tests indicated that the flame speed of hydrocarbon/air mixtures was substantially increased by the addition of stoichiometrically small quantities of hydrogen, and that the resulting speed could be correlated with that for the hydrocarbon fuel and a parameter indicating the extent of hydrogen addition. The theory assumes that the small quantity of hydrogen in the mixture oxidizes completely, thus leaving the hydrocarbon fuel to burn with an effective equivalence ratio ($\phi_p$) as defined, in the notation of the paper, by the following equation:

\[
\phi_p = \frac{C_p / \left( C_a - \frac{C_h}{C_a} \right)_{st}}{C_p / C_a}_{st}
\]  

(2.5)

Where $C_p$, $C_h$ and $C_a$ are the mole fractions of petrol, hydrogen and air respectively, and the sum of $C_p$, $C_h$ and $C_a$ is unity. The subscript 'st' stands for 'stoichiometric'. The extent of hydrogen addition is defined by the following equation which describes the ratio of the amount of hydrogen plus the amount of air necessary to completely oxidize it, to the amount of hydrocarbon fuel plus the net amount of air after hydrogen oxidation:
\[ R_h = \frac{C_h + \frac{C_h}{[C_h \div C_a]_{st}}}{C_p + \frac{C_h}{[C_h \div C_a]_{st}}} \]  

(2.6)

From Goodger[6] \((C_h/C_a)_{st} = 0.4202\) and \((C_p/C_a)_{st} = 0.01798\). With these values Equations 2.5 and 2.6 may be rewritten as follows:

\[ \phi_p = \frac{55.62C_p}{C_a - 2.38C_h} \]  

(2.7)

\[ R_h = \frac{3.38C_h}{C_p + C_a - 2.38C_h} \]  

(2.8)

By using \(C_a = 1 - C_p - C_h\) Equations 2.7 and 2.8 can be further simplified:

\[ \phi_p = \frac{55.62C_p}{1 - C_p - 3.38C_h} \]  

(2.9)

\[ R_h = \frac{C_h}{0.2959 - C_h} \]  

(2.10)

The authors of this paper found that for \(R_h \leq 1\) the laminar burning velocity \((u_L)\) could be correlated as follows:

\[ u_L(\phi_p, R_h) = u_L(\phi_p, 0) + \alpha_p(\phi_p)R_h \]  

(2.11)

The variation in \(\alpha_p\) with \(\phi_p\) is very small and can be taken as 0.83 m s\(^{-1}\) for methane and propane. Similar work has not been carried out with petrol-based mixtures but Sher and Hacohen[64-66] have suggested that this value is also useful for simulation studies of hydrogen-supplemented petrol/air combustion.

The equations for overall equivalence ratio and hydrogen energy fraction, given by Equation 2.3 and Equation 2.4 respectively, are rewritten here in the notation of this paper:
\[
\phi = \frac{2.38C_h + 55.62C_p}{1 - C_h - C_p} \quad (2.12)
\]
\[
\beta = \frac{C_h}{C_h + 19.61C_p} \quad (2.13)
\]

Equation 2.13 may be re-arranged to make \( C_h \) the object:

\[
C_h = \frac{19.61C_p\beta}{1 - \beta} \quad (2.14)
\]

Substituting for \( C_h \) from Equation 2.14 into Equation 2.12, and re-arranging to make \( C_p \) the subject, yields the following equation:

\[
C_p = \frac{\phi(1 - \beta)}{19.61\beta(2.38 + \phi) + (1 - \beta)(55.62 + \phi)} \quad (2.15)
\]

Equation 2.12 may be re-arranged to make \( C_p \) the object:

\[
C_p = \frac{\phi - C_h(2.38 + \phi)}{55.62 + \phi} \quad (2.16)
\]

Substituting for \( C_p \) from Equation 2.16 into Equation 2.14 gives \( C_h \) as a function of \( \phi \) and \( \beta \):

\[
C_h = \frac{19.61\beta\phi}{\beta\phi(19.61 - 1) + (19.61 \times 2.38) - 55.62] + 55.62 + \phi} \quad (2.17)
\]

Since the authors have only claimed that this theory is valid for \( R_h \leq 1 \) it would be useful to transform this to a relationship between \( \phi \) and \( \beta \). From Equations 2.12 and 2.13 \( \beta \) can be derived as a function of \( \phi \) and \( C_h \):
When \( R_h = 1 \), then from Equation 2.10 \( C_h = 0.14795 \), thus giving the limiting value of \( \beta \) as a function of \( \phi \). Alternatively Equation 2.18 may be re-arranged to make \( \phi \) the object, thus giving the following relationship:

\[
\phi = \frac{C_h (55.62 - \beta (55.62 - (19.61 \times 2.38)))}{\beta (19.61 - C_h (19.61 - 1)) - C_h}
\] (2.19)

Metghalchi and Keck\[62\] have developed a correlation for laminar flame speeds of various fuels, including iso-octane, with air at high temperatures and pressures. Assuming similar characteristics for petrol and iso-octane permits the formulation of the following equation for the laminar flame speed of petrol/air flames:

\[
u_L = u_{L0} \left( \frac{T}{298} \right)^x \left( \frac{P}{100} \right)^y (1 - 2.1f)
\] (2.20)

Where,

\[
u_{L0} = 0.2632 - 0.8472(\phi_p - 1.13)^2
\] (2.21)

and,

\[x = 2.18 - 0.8(\phi_p - 1)
\] (2.22)

and,

\[y = 0.22(\phi_p - 1) - 0.16
\] (2.23)

Where \( T \) is in K, \( P \) is in kPa and \( f \) is the residual exhaust gas mass-fraction. \( \phi_p \) was chosen as the equivalence ratio to be compatible with the model of Yu et al\[61\]. The value of \( u_L \) from Equation...
2.20 (assuming $P = 3000$ kPa, $T = 600$ K and $f = 0.15$) can be substituted into Equation 2.11 for the term $u_L(\phi_p,0)$. This forms the basis for calculating the laminar flame speed of hydrogen/petrol/air mixtures in terms of $\phi$ and $\beta$ and the result of this analysis is given in Figure 2.8. The top, right boundary of the flame speed map represents the $R_h \leq 1$ limit within which Yu et al.\cite{Yu61} claim that their model is valid. The lower boundary of the map represents the theoretical lean-limit equivalence ratio for hydrogen/petrol/air mixtures. It should be noted that the authors only tested the model with propane and methane based mixtures and that they did not use values of $\phi_p$ leaner than the lean-limit of the hydrocarbon fuel. Figure 2.8 presents flame speed data over a range of equivalence ratios which extended well past the lean-limit for petrol. During the calculation, in order to ensure that the flame speed of the mixture remained positive, the first term on the right-hand side of Equation 2.11 was prevented from becoming negative. The result of this is indicated across the middle of the map by the sharp turns in the contour lines. These occur at the points where $\phi_p$ equals the lean-limit equivalence ratio for petrol-air mixtures. The kinks at the top-right of the map occur when there is just sufficient air for the hydrogen to fully oxidize. Obviously the model is not valid for petrol-based mixtures at all values of equivalence ratio within the limit $R_h \leq 1$ thus indicating the need for further research.

Lucas and Emtage\cite{Lucas63} have developed a microprocessor-based control system for a 2 litre hydrogen/petrol engine, and this is described in a paper which was presented at the I.Mech.E. conference "Computers in Engine Technology" at Cambridge in March 1987. This control system will be described in detail in Chapter 3. Preliminary results indicated a significant saving in fuel consumption for lean operation.

Sher and Hacohen\cite{Sher64-66} have carried out work on a 2310 cm$^3$ 4-cylinder engine with petrol and hydrogen-supplemented petrol. Hydrogen supplementation of up to 15% hydrogen energy fraction resulted in part-load fuel consumption improvements of 10-20%. The improvements were relative to petrol-only operation at the same engine speed, b.m.e.p. and inlet manifold pressure. Consequently, the equivalence ratios were similar. Further improvements should have been possible if the speed and b.m.e.p. values had been maintained while the throttle was opened up to decrease the equivalence ratio. This would, of course, have increased the hydrogen energy fraction requirement. As a result of maintaining a near stoichiometric equivalence ratio, the HC and CO emissions were low but the NO$_X$ emissions were high because of the high combustion temperatures.

To sum up the requirements for hydrogen/petrol operation, hydrogen energy fraction ($\beta$), equivalence ratio ($\phi$), and ignition timing ($\alpha_{ig}$) need to be optimized to give the maximum thermal efficiency within the constraints of the legislative requirements for HC and NO$_X$ exhaust emissions for a given speed/torque engine operating point. In order to achieve this, the engine will require a
management system capable of independently controlling hydrogen mass-flow-rate, petrol mass-flow-rate, throttle position and ignition timing.

2.2 ELECTRONIC ENGINE MANAGEMENT

2.2.1 Historical Review

It appears that the Bendix Corporation invented the concept of electronic fuel injection in the 1950's, but Robert Bosch GmbH have been chosen as an example of a fuel injection manufacturer because of the company's continued development in this field. Bosch's first commercial system, the "D-Jetronic"[67] fuel injection system, was produced as early as 1967. This system appears to have followed the earlier development of a system designated the ECGI[68] and featuring sequentially timed multi-point fuel injection. The "D-Jetronic" was a multi-point injection system utilizing speed/density calculations for air mass-flow-rate. In the mid 1970's Bosch introduced the "L-Jetronic"[69,70] system which uses direct air-flow measurement as a means of improving the accuracy of air/fuel metering. At about the same time, Bosch upgraded its mechanical fuel injection system with the addition of simple electronic controls. This was the "K-Jetronic"[69] system which was in turn superceded by the "KE-Jetronic"[69-71]. In the early 80's the "LH-Jetronic"[69,70,72] system was introduced, featuring a fast response hot-wire air mass-flow meter[73]. By now the electronic control units featured digital rather than analogue techniques, while the introduction of special purpose microcontrollers allowed more complex functions to be performed, such as Lambda closed-loop air/fuel ratio control, based on exhaust oxygen sensors. More recently Bosch have introduced the "Mono-Jetronic"[69,70] single point injection system and the "Motronic"[69,74] fuel injection system which includes ignition timing control in addition to fuel metering. The latest developments at Bosch[75,76] include the development of a new low cost, low pressure, single point injection system and the use of hybrid technology for ignition and fuel injection controllers. The latter technique allows further miniturization and improvements in the robustness necessary to house these systems within the engine compartment. The ECU of the "LE-Jetronic" system, for example, is situated within the air flow meter housing.

Other manufacturers such as Ford[77-81], Bendix[82-84], Mitsubishi[85,86], Toyota[87-94], Nippon Electric[95,96], Motorola[97,98], Shell[99], National Semiconductor[100], Alfa Romeo[101], Nissan[102], Hitachi[103-105], Cummins[106], Lucas[107,108], Renault[109], Texas Instruments[110] and other research organizations[111-119] have also contributed to further development of engine management technology.
Most current engine management systems gain their engine timing information either from the distributor (the trend is for the introduction of distributorless ignition systems), or from the flywheel teeth. One problem with distributor-based sensors is that of dither in angle wrap-up between crankshaft and distributor. Wolber and Ebaugh\cite{106} discuss the relative merits of several types of position sensor in current use. Usually one sensor is used to detect TDC crankshaft position while another is used to provide a frequency proportional to engine speed. The TDC sensor can be eliminated if a ‘missing’ pulse is provided at TDC for the speed sensor. This missing pulse can be regenerated electronically so that speed sensing is not affected, and at the same time, a TDC signal is made available to the microcontroller. The number of pulses per engine revolution is important for the response rate of the speed signal. For timing purposes, the crank angle variation between pulses has to be assumed to be linear with time, and because of cyclic speed fluctuations, the resulting angular error increases as the number of pulses per revolution decreases.

Air mass-flow-rate sensing gives an indication of engine load and is essential for the accurate control of equivalence ratio unless Lean Exhaust Gas Oxygen (LEGO) sensing is employed. One technique is to map the mass-flow-rate through the engine against inputs of engine speed and throttle position. This results in a very fast response system and, for this reason, has been used on some high performance cars\cite{101}. The main drawback is that the initial calibration becomes invalid as the engine wears and as dirt builds up on the throttle plate.

Manger\cite{70} has developed a Vane Airflow Sensor (VAF) which responds to the dynamic pressure of the air stream by allowing the deflection of a moving vane in opposition to a return spring. The deflection of the vane is measured by a conducting plastic potentiometer which, by the geometric design of the wiper track and the laser trimming of Cermet resistors, can be designed to give a non-linear characteristic. The use of a compensation vane acting in a damping chamber counteracts the effects of pressure waves on the down-stream face of the main vane. Although the steady state accuracy of this type of air flow meter is very good (±1%), the transient response is poor due to the inertia of the moving vane. This necessitates the implementation of a transient strategy to adjust the apparent air/fuel ratio during rapid throttle angle perturbations. The VAF sensor housing includes an air temperature sensor on the inlet side so that density correction can be applied by the ECU.

Sasaki et al\cite{86} have developed a Karman Vortex airflow sensor. A bluff body is used to generate Karman vorticies in the air stream and these are detected ultrasonically. The frequency
with which the vorticies are detected is proportional to the air volume-flow-rate. A temperature sensor is used along with an assumed value of barometric pressure in order to calculate the air mass-flow-rate. The response rate of the sensor is fast but flow reversals are not taken into account.

Tsuruoka et al[93] have developed a Karman Vortex airflow sensor using an optical technique for detecting the vorticies. The small pressure fluctuations associated with the vorticies are ducted to an ultra-lightweight mirror suspended by torsion wires. The mirror deflections caused by the pressure fluctuations are detected by a light-emitting diode and a photo-transistor arrangement. A fast response and a wide dynamic range are achieved but corrections must still be made for temperature and pressure effects.

Barriol et al[109] have helped to develop an Ionic air flow sensor. The sensor generates a periodic 140 µs high tension pulse, with a peak potential of 6kv, between a series of stainless steel needle points and a pair of grids. The reciprocal of the transit time of the resulting ionic cloud between the grids is proportional to the air volume-flow-rate. A temperature sensor is required along with an assumed value of barometric pressure in order to calculate air mass-flow-rate. The response of the sensor is fast but flow reversals are not taken into account.

Mauer[115] has developed an air volume-flow-rate sensor based on a capacitive drag force measurement technique. A light-weight membrane is deflected by the dynamic pressure of the air stream and forms one of the capacitor plates. The second capacitor plate is fixed in position. The capacitance changes are detected as frequency changes in an oscillator circuit. The frequency changes are demodulated by a phase-locked-loop to a voltage signal which is inversely proportional to the square of the air velocity. A microprocessor corrects this signal for non-linearities such as the laminar/turbulent transition and fluid density variations. The sensor operates with less than 1% uncertainty over a 100:1 range. The unit has a fast response rate but is not capable of measuring flow reversals.

Sumal and Sauer[73] and Sasayama et al[103] have developed true air mass flow sensors based on the hot-wire-anemometer principle. Two platinum resistors, forming two arms of a resistance bridge circuit, are placed in the air stream. One of the wires operates at ambient temperature while the other is heated by electrical current to maintain its temperature at 100 °C above the other 'compensation' resistor. Since the rate of heat loss from the heated wire is a function of temperature and forced convection, the magnitude of the heating current is then a function of the mass-flow-rate.
Cops and Moore[107] have developed a true air mass-flow sensor based on the corona discharge principle. A metal disk is mounted on a cylindrical insulator situated centrally within a flow-tube, the inner surface of which is coated with a conducting material. A high potential is maintained between the disk and the flow-tube such that a continuous stream of ions travel from the edge of the disk to the conducting layer on the flow-tube surface. It can be shown that the axial deflection of the ion stream is proportional to the mass-flow-rate of air through the flow-tube. The axial deflection of the ion stream is detected by treating it as a wiper on a potentiometer represented by the conducting material on the flow-tube. The difference between the currents flowing from either end of the flow-tube indicates the axial deflection. It is possible therefore to detect flow reversals and this feature, along with a very fast response time (<1ms), makes this a very attractive contender for automotive use. Unfortunately the unit is prone to dirt build-up and is sensitive to variations in atmospheric humidity. Current versions are very expensive and are not as rugged as might be necessary for automotive use.

The most common method of electronically controlling petrol flow-rate to an engine is by means of solenoid-operated fuel injectors. A 'single point' fuel injection system uses one high flow injector, usually mounted in the throttle body, to control the flow for all cylinders. A more expensive system is to use one injector in each inlet port, thus eliminating manifold mal-distribution effects. The most common design of multi-point fuel injector is produced by Bosch[120] and uses a pintle valve to meter the fuel flow. The Bendix[83] DEKATM multi-point fuel injector is designed to be externally interchangeable with other designs but uses a thin critical orifice to meter the fuel instead of a pintle nozzle. This approach, also used by the Lucas 'disc' injector, overcomes many of the calibration drift problems associated with clogging and wear. The needle used to close the measuring orifice is also lighter than a typical pintle arrangement and consequently better linearity results. In order to achieve good linearity from any injector it is necessary to drive the armature with an initially high current and, after 1-1.5ms, the current should be reduced to a 'hold-on' level. In this way the opening and closing delays are minimized. Injection sequencing is also important. Using one injection period per engine cycle, instead of two, increases the minimum duration required thus avoiding the non-linear region of the injector.
Chapter 3

THE ENGINE MANAGEMENT SYSTEM
3.1 SYSTEM SPECIFICATION

Section 2.1 concluded, on the basis of work by other researchers, that an engine management system for the hydrogen/petrol engine would be required to control petrol mass-flow-rate, hydrogen mass-flow-rate, throttle position and ignition timing. This chapter describes the engine management system which was developed to meet these requirements.

Flexibility in the design allowed the system to function either as a development tool, with interactive control of various parameters, calibrations and strategies, or as a dedicated stand-alone controller in a vehicle.

Most of the techniques and strategies used were gleaned from the literature survey in Section 2.2, but some novel ideas were also tested. In order to concentrate on the particular problems of hydrogen/petrol operation, transient strategies such as cold and hot starts were not implemented and update cycle times were not minimized as rigorously as might be expected in a production engine management system.

3.1.1 The Dedicated System
With reference to Figure 3.1, the dedicated system comprised the Electronic Control Unit (ECU), the engine and the accelerator pedal. The ECU (comprising the microprocessor and the interface electronics) monitored driver-demand, engine speed, air mass-flow-rate, crankshaft position, cycle position, throttle limits and hydrogen pressure. The control strategy then determined and set the ignition timing, throttle angle, petrol injection timing, petrol injection duration and hydrogen injection duration. In the vehicle, the only control the driver had was via the ignition switch, accelerator pedal and a biased toggle switch on the interface electronics rack which provided a 30 s fuel enrichment period for starting.

3.1.2 The Development System
Figure 3.1 indicates the system components which were used exclusively for development work. In development mode, the ECU monitored petrol mass-flow-rate, hydrogen mass-flow-rate and brake torque from the engine test-bed. From this information, the ECU calculated and displayed values of brake thermal efficiency, equivalence ratio, hydrogen energy fraction and other variables which were important for the optimization of the engine's calibration. The Visual Display Unit (VDU) was used to display this information and also provided the means to interactively alter various parameters and strategies.

In addition to the VDU there was a printer for hard-copy recording of test-bed data, a communications link via a Local Area Network (LAN) to a variety of main-frame computers and
a pair of 8" floppy-disk drives. The disk drives formed the hardware foundation for the implementation of a disk-based software operating system called CP/M. CP/M is a well proven, if outdated, operating system for microcomputers, which supports the necessary tools for computer program development, namely a filing system, editors, assemblers, linkers and various software debugging tools. Most of the calibration software was run on a Honeywell (Multics) mainframe computer because of the high level of numerical computation required. The transfer of the data for the look-up-tables to the CP/M system was made via the LAN.

3.2 HARDWARE DESCRIPTION

3.2.1 Microcomputer
The microcomputer used was an Intel Multibus iSBC 80/24 single board computer with an 8085A-2 microprocessor operating at 4.8 MHz. The basic random access memory (RAM) was extended from 4K bytes to 8K bytes by the addition of an Intel iSBC 301 Multimodule board. The read-only memory (ROM) was configured for four Intel 2764 eraseable programmable read-only memory (EPROM) devices which gave a capacity of 32K bytes. This on-board memory operated without any processor wait-states and may be used in the future for the dedicated control system in order to reduce the update cycle time. However in order for the disk drive controller board to operate in direct memory access (DMA) mode, it was necessary to use a separate memory board for the development system.

Included on the iSBC 80/24 board were the following peripheral devices: An Intel 8259A Programmable Interrupt Controller (PIC) was used to arbitrate and administer various interrupt requests. An Intel 8253 Programmable Interval Timer (PIT) was used for baud-rate generation, simulated crankshaft encoder pulses and ignition timing control. An Intel 8251A Programmable Communications Interface (PCI) device was used to control RS232C serial communication to the VDU. Two Intel 8255A Programmable Peripheral Interface (PPI) devices were used to provide up to 48 digital signals to and from the interface electronics.

One of the two iSBX Multimodule expansion sockets on the iSBC 80/24 board was fitted with an Intel iSBX 311 analogue input Multimodule board providing 16 channels of analogue to digital (A/D) conversion. The second iSBX socket was fitted with an Advanced Micro Devices Am94/1541 Stepper Motor Controller SBX module used for throttle actuation.

A modification was added to the iSBC 80/24 board in the form of a small logic board which provided the basis for a hardware TRACE facility. Once enabled, this board generated a Non-Maskable Interrupt (NMI) after three instructions. A system monitor program, booted
from EPROM after POWER-ON, used this hardware facility to trace through interrupt driven software.

The iSBC 80/24 board was positioned in the Multibus backplane to function as a second priority sub-master. The highest priority was given to an Advanced Micro Devices Am95/6120 Dual-Density Floppy Disk Controller board. This arrangement was chosen so that DMA disk reads were not impeded. This board was used to control a Quandon Electronics QMS TFDU2 dual 8” floppy disk drive unit with a formatted capacity of 0.97 Mbytes (DS/DD).

The third Multibus board in the system was an Advanced Micro Devices Am96/5232 Programmable RAM/EPROM And I/O Board. This board provided all of the system memory as well as four additional peripheral devices. An Intel 8251A PCI provided the system with an Auxiliary RS232C serial communications port which was used for communication with other computers via a LAN. An Intel 8255A PPI was used to provide up to 24 digital signals to and from the interface electronics. An Advanced Micro Devices Am9513 System Timing Controller provided five dual-register programmable timers which were used to control the petrol and hydrogen injectors. The first register controlled the timing of injection while the second controlled the duration of injection. The fourth peripheral device was a SBX Maths Co-Processor Multimodule fitted to the iSBX socket on the Am96/5232 board. This Multimodule was specially designed for this work by the author, and uses an Advanced Micro Devices Am9511A Arithmetic Processor running at 4 MHz. The module combined an 8-bit I/O interface with 16/32-bit integer or 32-bit floating-point arithmetic, and its use greatly reduced the control system’s update cycle time.

The memory on the Am96/6120 board was configured into four 'quads' of four devices each, as described in Figure 3.2. Quad-0 was mapped to location 0000H during POWER-ON or RESET and contained the complete dedicated control system in four 2764 EPROM's (32K bytes). Depending on the state of a switch on the interface electronics rack, control was either transferred directly to the dedicated control program or Quad-1 was enabled and the system monitor program initiated. Quad-1 was mapped to location 8000H and contained the system monitor program, the disk operating system booter and the Basic Input/Output System (BIOS) for the disk operating system in EPROMs. Quad-2 and Quad-3 each contained four 6264 RAM devices so that Quad-2 provided 32K bytes of RAM starting at 0000H and Quad-3 started at 8000H to complete 64K bytes of contiguous RAM. The latter two quads were swapped-in during the booting of the disk operating system. The memory decoding was effected by four Programmable Array Logic (PAL) devices; the boolean equations for which are given in Appendix A along with those for the I/O decoder PAL for this board. All of the system I/O is accessed via the I/O bus and Appendix B contains the description of the I/O address map.
A comprehensive list of the modifications made to, and options selected for, the various commercial boards is given in Appendix C.

3.2.2 Sensors

Engine Timing & Speed: The most important sensor in the control system was the crankshaft encoder. A Gaebridge 180SM/30HD/5/10 was chosen because of its rugged construction and reliability. The unit was an optical system with TTL outputs on one channel (GATE) timed to give a 1° pulse at 90° BTDC and 180° later at 90° ATDC, and 180 pulses per revolution (1:1 mark/space ratio) on a second channel. The GATE pulses were used as reference points for ignition and injection timing and the crankshaft angle from these references was counted in 1° steps (CLK) obtained by multiplying the signal from the second encoder channel by two. Engine speed was obtained as an analogue voltage by means of a Frequency-to-Voltage (F/V) converter acting on the CLK signal. The microcomputer sampled this voltage by means of an A/D converter. Table D.1 in Appendix D gives the calibration data for engine speed and Figure 3.3 describes this information graphically. The resulting calibration for engine speed in r/min is as follows:

\[ N = 16.87 \times ATDSPD - 11.34 \]  

Another timing sensor detected the firing of the sparkplug for cylinder #1 and outputs a TTL pulse (MARK). This signal gave essential engine cycle timing information so that sequential petrol injection could be used. Several different types of sensor have been tried but to date none have been found to be entirely reliable. Inductive sensors are prone to interference from neighbouring sparkplug leads and the difference between the true signal and those induced was masked by the change in signal magnitude with engine load. Another design used a Neon bulb in series with the high-tension current and an optical sensor to detect the ionization light emission. It is not clear why this technique still suffers from interference but it may be due to radiated noise affecting the amplifier stages. Perhaps the best solution would be to use a Hall effect transducer to detect the passing of one of the camshaft lobes but this has been left for future work.

Driver Control Demand: The second most important sensor was required to monitor the accelerator pedal position. A linear potentiometer was chosen for this purpose because of its low cost and the simplicity of the associated instrumentation. The steps taken to ensure safety of operation are described in Section 3.2.4 but essentially tests were carried out for an open circuit and detached wiper. In either case the system failed 'safe'. Table D.2 in Appendix D gives the calibration data for driver control demand and Figure 3.4 describes this
information graphically. The resulting calibration for demand as a function of analogue input number is as follows:

\[ D = 0.2755x_{ATDCIN} - 41.05 \]  

(3.2)

**Air Mass-Flow-Rate:** Air mass-flow-rate was the third most important input and any sensor had to provide an accurate fast response signal. A Hitachi type 4AM hot-wire anemometer was used at first because of its fast response, direct mass flow reading and simple interfacing requirements. The unit accepted an un-regulated 12V supply and gave an output of 0-5V. The meter was calibrated at the factory by laser trimming of special resistors and the calibration (in g/s) was expressed as the following 4th order polynomial which is described graphically in Figure 3.5:

\[ m = 0.1357v^4 - 0.9640v^3 + 8.062v^2 - 14.87v + 9.817 \]  

(3.3)

The data for the analogue input voltage calibration is given in Table D.3 and described graphically in Figure 3.6. The resulting calibration is as follows:

\[ v = 9.523x10^{-3}x_{ATDHIT} - 7.686x10^{-2} \]  

(3.4)

This unit's fast response was considered to be an asset because it largely overcame the problems of equivalence ratio excursions during throttling transients. At some operating conditions however, the unit appeared to give too high a reading and this was attributed to the occurrence of flow reversals. This has not been conclusively proved but in the interim an alternative, well-proven, air volume-flow-rate meter was adopted.

The alternative air-flow meter used was a Bosch Vane Air Flow (VAF) meter (Bosch Part No. B-280-150-620, Ford Part No. 0-280-202-063, Ford Type No. 85GB-12B529-BA) as used on the Ford T88 2 litre fuel injected OHC engine. This type of meter measured air volume-flow-rate by the deflection of a spring loaded vane situated in the main air flow. An embeded thermistor monitored the inlet air temperature which was used to calculate the mass-flow-rate based on the assumption that deviations from standard ambient pressure were small. Should altitude compensation be required then a separate absolute pressure sensor would be necessary. Laser trimming of calibration resistors was carried out during production to give the volume-flow-rate calibration (in m³/hr) described in the following equation and depicted in
Figure 3.7. The data for this calibration is given in Table D.4 and may be found in Appendix D.

\[ \hat{V}_a = 2.103v^4 - 12.36v^3 + 35.48v^2 - 32.09v + 17.64 \]  

(3.5)

The data for the analogue input voltage calibration is given in Table D.5 and described graphically in Figure 3.8. The resulting calibration is as follows:

\[ v = 9.0153 \times 10^{-3} \times \text{ATDVAF} - 4.1705 \times 10^{-2} \]  

(3.6)

The inertia of the vane in this type of flow meter introduced a significant lag in its response to throttling transients and so an equivalence ratio compensation strategy was necessary for good driveability.

Engine Brake Torque: A Heenan-Froude Dynamatic MkII dynamometer was used on the development test-bed and the associated 'Test Automation' controller provided an analogue output voltage proportional to brake torque. The analogue input calibration data for brake torque is given in Table D.6 in Appendix D, and is described graphically in Figure 3.9. The calibration for brake torque in Nm may be expressed as follows:

\[ b.t. = 0.4950 \times \text{ATDBTK} + 0.3333 \]  

(3.7)

Hydrogen Flow-Rate: Hydrogen flow-rate sensing was performed by an electronic gap-meter operating at 350 kPa (gauge) upstream of the injectors and plenum. This unit used a float operating inside a vertical tapered glass tube. The height of the float was proportional to the flow-rate of hydrogen which was detected by an optical array of Light Emitting Diodes (LEDs) and sensors. A 4-20mA current-loop output was provided so that the microcontroller could monitor the use of hydrogen. The unit was calibrated in litres/min ATP (101.3 kPa, 293 °C) when operated at 501.3 kPa (absolute) and 20 °C. The correction factor \( f_h \) for operating the unit at temperatures and pressures other than these is given by the following equation:

\[ f_h = \sqrt{\frac{P_h}{501.3}} \times \frac{293}{T_h} \]  

(3.8)
The analogue input calibration data for indicated litres/min (ATP) is given in Table D.7 in Appendix D and presented graphically in Figure 3.10. The calibration function is as follows:

\[
\dot{v}_h = 5.5514 \times 10^{-1} \times \text{ATDHFL} - 4.8984 \times 10^{1}
\]

(3.9)

**Petrol Flow-Rate:** The petrol fuel injection system on the engine consisted of a high pressure pump supplying fuel to one end of the fuel rail on which the four injectors were mounted. A pressure regulating valve was mounted on the other end of the fuel rail and maintained a fuel pressure of 250kPa, relative to manifold pressure, by dumping excess fuel back to the storage tank. In order to measure the net fuel used by the engine, two turbine flow-meters were fitted to the supply and return lines. These transducers provided frequency outputs proportional to flow-rate and a special microprocessor flow-monitor calculated the net flow-rate and transmitted the information to the microcontroller. Electromagnetic interference from the engine's ignition system was dealt with by completely screening the transducers. When the engine had warmed up, the fuel passing through the fuel rail was heated sufficiently so that, on release to atmospheric pressure through the pressure regulating valve, some vapourization of the lighter fractions occurred. The presence of the resultant gas bubbles, in addition to the pulsatile nature of the return flow, served to make the measurement of the return flow very erratic. A conventional 'Time For Volume Used' (TFVU) arrangement, with the return flow piped back to the measurement volume at atmospheric pressure, overcame the problems of pulsatile flow. Chilling of the supply fuel line helped to reduce the generation of vapour bubbles which would otherwise have affected the effective volume of the measurement chamber. Unfortunately the nature of this measurement system precluded it from being used to give a continuous fuel flow-rate signal to the microcontroller. In the end, the injectors were accurately calibrated so that the fuel flow-rate could be inferred from the petrol injection duration.

**Exhaust Emissions:** Standard laboratory exhaust emissions equipment was made available to measure carbon monoxide, carbon dioxide, hydrocarbons and oxides of nitrogen. A 'Signal Instruments' hydrocarbon analyser model 3000 was used to measure parts per million of propane (C\textsubscript{3}H\textsubscript{8}). An 'Analysis Automation' series 440 chemiluminescent NO\textsubscript{X} analyser was used to measure parts per million of nitric oxides. The carbon monoxide and carbon dioxide analysers were infra-red units by 'ADC'. The hydrocarbon analyser was fed with 'wet' exhaust gases, via a heated line running at 190 °C, directly from the engine 'down pipe'. The exhaust gases for the remaining analysers were fed through a second heated line, running at 190 °C, to a permeation drying unit which removed the water content of the sample. This equipment was
never intended to be interfaced directly to the microcontroller and so measurements were recorded manually.

3.2.3 Actuators

Ignition: The microcontroller provided a timing signal to the interface electronics unit which drove a conventional ignition coil. The high tension voltage was directed to the correct spark plug by means of the distributor.

Petrol Injection: The petrol flow-rate was controlled by four solenoid-operated injectors (Bosch Part No. 0-280-150-219) mounted one per cylinder in the inlet ports just above the valves. This type of injector has a linear flow versus open-time characteristic over most of its operating range but there is an effective delay time which is the difference between the opening and closing delays. The non-linearity at the top of the range can usually be ignored by choosing an injector with a sufficiently high static flow-rate. The non-linearity at the bottom of the range can not be overcome in the same way and so the opening delay has to be minimized or taken into account during calibration. For this control system the opening delay was minimized by using a high solenoid current during the first 1.5 ms and then reducing it to a low hold-on level for the rest of the injection duration. In this way the opening duration was reduced by increasing the magnetic opening force. The closing duration was also reduced because the lower hold-on current resulted in a lower back e.m.f. voltage when the current was switched off.

The petrol injectors were calibrated as a set of four by conducting a TFVU test. The computer generated crankshaft encoder signal was used to simulate engine operation at various speeds while the injection duration was varied. The fuel from all four injectors was directed into a 500 cm³ graduated cylinder and the time was recorded with an electronic stopwatch. Injector operation was enabled and disabled by a switch with one hand while the stopwatch was operated with the other. The data for this calibration is given in Table D.8 in Appendix D and described graphically in Figure 3.11. The effective calibration function in ms is as follows:

$$\Delta t = 1.0266 \times 10^{-1} \Delta m_p + 9.4281 \times 10^{-1} \quad (3.10)$$

Hydrogen Injection: The control of the hydrogen flow-rate presented a challenging problem. One solution considered was that of using an electronically controlled regulating valve to control the upstream pressure at a critical-flow orifice. By sensing upstream temperature it would have been possible to directly control the mass-flow-rate of hydrogen. Unfortunately the
response times for available regulators were of the order of tens of seconds. Some commercial mass flow controllers using the hot-wire-anemometer principle were also rejected on the basis of poor response times.

May et al.[56,59] describe the use of 'modified' petrol injectors for the control of hydrogen flow-rate. The detail design of these injectors has not been published but the use of standard injectors could possibly give rise to two problems. Firstly, the petrol injector is designed so that the fuel promotes the conduction of heat away from the solenoid coil. The effectiveness of hydrogen to provide the same level of cooling is an unknown factor. Secondly, the viscosity of the fuel helps to dampen the closing impact of the pintle nozzle, thus minimizing mechanical distortion and wear which would affect the injector’s calibration. The first of these problems was alleviated by the use of the 'stepped current' injector driving technique described in the last paragraph of Section 2.2. The low average current reduced the need for cooling. Unfortunately, during the closing phase, the lower back e.m.f. allowed the pintle to close quickly, thus increasing the severity of the seat impact. Despite this potential problem, four injectors (Bosch Part No. 0-280-150-355) were mounted in a manifold as described in Figure 3.12 and the output from the manifold was piped to the engine’s throttle body just downstream of the throttle plate as shown in Figure 3.13. A 100 cm$^3$ reservoir was situated immediately upstream of the injector manifold. This was done in the interest of linearity and in an effort to prevent pressure pulsations from being fed back to the gap-meter which was used for hydrogen flow-rate measurement.

The calibration data for the hydrogen injectors is presented in Table D.9 in Appendix D and is described graphically in Figure 3.14. The scatter on this graph is an indication of the reliability of this calibration and points to the need for a better means of controlling the hydrogen mass-flow-rate. The effective calibration function for $\Delta t$ in ms, given $\Delta m_h$ in mg, is as follows:

$$\Delta t = 4.2374\Delta m_h + 0.54 \quad (3.11)$$

The calibration data for the pressure drop through the hydrogen pressure regulator is presented in Table D.10 in Appendix D and is described graphically in Figure 3.15. The effective calibration function for $\Delta P_h$ in kPa, given $\dot{m}_h$ in mgs$^{-1}$, is as follows:

$$\Delta P_h = 0.1544\dot{m}_h \quad (3.12)$$
Throttle Actuation: Section 2.1 concluded that driver independent throttle control was a requirement of the control system and two types of actuator are discussed here. DC servo-motors offer high torque, fast response and high resolution throttle actuation in a compact unit but the main disadvantage is their high cost. The other alternative, stepper motors, are cheaper but have reduced resolution, lower torque and slower response times. Faster throttle response times than those possible by normal driver actuation may appear to be unnecessary but in a Drive-By Wire (DBW) system such as this, there are occasions, such as the change from hydrogen-petrol to petrol-only operation, when very high response rates may be required. Another problem with stepper motors is that their low cost assumes that no position feedback is required. Instead they rely on their controllers to keep track of the number of steps made in each direction. The controller therefore has no way of knowing if a step has been missed as the result of trying to accelerate the motor too fast.

A Superior Electric Slo-Syn (Type M092-FD-8109) stepper motor was used in conjunction with a McLennan TMI64C translator module and the following actions were taken to overcome some of the problems associated with this method of actuation.

In half-step mode the motor increments were in 0.9° steps and this was improved with a 4:1 toothed belt and pulley reduction to give 0.225° steps at the throttle plate. This degree of resolution was found to be necessary at low throttle angles because of increased engine performance sensitivity. The reduction in response time could have been alleviated by using a smaller, lower torque motor and even better response could possibly have been achieved if a non-linear linkage had been used between the motor and throttle plate thus resulting in throttle increments which are proportional to engine performance.

In this system, the lack of position feedback information was overcome by using limit detecting microswitches at fully open and fully closed throttle positions. During initialization the throttle was operated to these limit positions in order to initialize the step count and operating travel. Software limits were set to two steps before the microswitch limits so that any overshoot which occurred near the limits during normal operation would not activate them. If for any reason the motor missed two steps, or overshot and gained two steps, the error was detected on the next occasion that the throttle was commanded to the fully open or fully closed position depending on the sign of the error.

The Am94/1541 stepper motor controller module's facility for speed ramping was utilized. Ramping allowed the motor to attain higher speeds than was otherwise possible by limiting the acceleration to a level beyond which steps might have been missed. Once programmed with an
upwards ramping pattern, the controller used this to accelerate to, and to decelerate from, the maximum speed selected. If the number of steps to be moved was less than the number of steps in the ramping pattern, the maximum speed achieved was lower.

3.2.4 Interface Electronics

In a production engine control system, the interface electronics are housed in the same enclosure as the microcomputer. This is done for compactness and simplicity but care needs to be taken to ensure that the processor is not affected by electrical interference. Interference can take three forms. Firstly, high current switching can induce 'spikes' on the power supply rails. Secondly, electrostatic discharges can severely damage integrated circuits. Thirdly, radiated electromagnetic interference (EMI) can be caused by the ignition system, telecommunication transmission and radar. Electromagnetic radiation can induce resonance effects in printed circuit lands leading to problems which are very difficult to isolate. Considerable research effort is being directed at the prediction of ElectroMagnetic Compatability (EMC) for various system components by, among other means, finite element techniques. Adequate shielding of the enclosure can be the single most important precaution and many production ECUs are housed in the passenger compartment well away from the ignition system. The generous use of capacitive decoupling for the power rails and input signals helps to overcome the effects of spikes and the stringent avoidance of 'ground loops' contributes to the elimination of self-induced 'noise'.

It was not feasible to produce this engine controller in one enclosure and so four separate enclosures were used. The first enclosure contained the microcomputer and its mains power supply. The second, consisted of a 19" rack with various inserted modules. This unit contained most of the low power interface electronics. The ignition and injection driver stages were mounted in an air tight housing located adjacent to the engine thus allowing the use of short, high current connections and so, the resulting radiated interference was kept away from the microcomputer. The fourth enclosure contained the relays for the petrol pump and hydrogen solenoid valve relays.

Most of the interface logic consisted of CMOS devices but full advantage was not taken of the noise immunity by using 12v or 15v. The reason for this was that the computer ports were TTL compatible and in addition some of the interface ICs were only available in TTL technology. All of the logic therefore operated at 5v. Schmitt input devices were used wherever possible to combat the signal integration effects caused by long interconnections. These inputs use 'level' sensitive detection (rather than edge sensitive detection), with built-in hysteresis, to 'square up' the waveform.
Power Supply: A 'clean', stable power supply is essential for this type of work and special attention must be paid to the 'ground' (GND) power rail. This control system used a single GND rail for both digital and analogue circuits but the author's subsequent experience has indicated that Digital GND and Analogue GND should only be connected at one point, preferably at the power supply, with an inductance and resistance in series as shown in Figure 3.16. The inductance blocks any high frequency noise while the resistor limits the d.c. current. In this way the two GNDs are held at the same nominal potential without the digital noise being transmitted to the analogue circuits.

The power supply used for this work accepted an unregulated 12 volt input and provided regulated outputs at +5 volts, ±12 volts and ±15 volts. The 5 volt supply catered for the majority of the load while the remaining 2-rail supplies catered for a few low power analogue circuits.

Status Signals: All 'status' input signal lines and all signal lines which could have been disconnected during normal operation were terminated with pull-up resistors to the 5 volt rail. The signal was then assumed to fail to the 'high' state, thus facilitating error detection and handling. As an example of this technique Figure 3.17 describes the status circuit associated with the hydrogen pressure switches. Both the high and low pressure switches were normally closed (pressure within range) giving a '0' logic signal. If either low or high pressure occurred, the circuit to GND was broken and the pull-up resistor forced a '1' logic signal. This was detected by the microcomputer and interpreted as a hydrogen supply failure and petrol-only operation was instituted. The pull-up resistor was located within the interface rack. Thus, should the external connection be accidentally broken, the system failed safe.

Figure 3.17 also describes the circuit which was used to monitor the driver-demand potentiometer. One end of the potentiometer was connected to the 5 volt rail, while the other end was grounded via the LED of an optical coupler. During normal operation, the coupler detected the flow of current through the potentiometer and gave a 'potentiometer OK' TRUE signal (logic '0'). If either end of the potentiometer became disconnected, the pull-up resistor forced a FALSE signal (logic 1), thus indicating to the microcomputer that a fault had occurred. During normal operation, the potentiometer output signal could not reach ZERO because of the voltage drop across the LED and so the resistor between the signal and GND indicated a detached wiper by forcing the signal to ZERO. A high value resistor was used so as not to load the potentiometer.
Figure 3.17 also indicates how the two front panel status LEDs were driven. The LEDs were steadily lit during normal operation and flashed at 2 Hz if a fault occurred.

No attempt was made to build automatic cold start enrichment into the control system, so a biased toggle switch was provided on the front panel of the interface rack to request a 30 s enrichment period. The 'safe' mode for the ENRICH signal was FALSE, hence the pull-up resistor defined the FALSE state as a logic '1'. Figure 3.18 describes the circuit used to provide the 30 s signal. A front panel LED was lit to indicate the ENRICH TRUE condition.

Figure 3.18 also describes the use of a front panel switch with which the operator could select the development or dedicated mode prior to POWER-ON or RESET. The pull-up resistor was situated in the microcomputer so that the system defaulted to the development system if the interface rack was disconnected.

**Crankshaft Timing Signals:** The TTL crankshaft timing signals from the encoder were conditioned by Schmitt triggers in order to 'square up' the leading and trailing edges of the waveform. Figure 3.19 describes the way in which simulated crankshaft encoder signals could be selected in place of the real ones. The COMPUTER CLK signal was generated on the microcomputer by a spare timer and the frequency was software selectable. This signal was divided by 90 to produce a simulated GATE signal, thereby simulating a 180 pulse/rev encoder with two GATE pulses 180° apart. The real encoder was timed to the engine so that the two GATE pulses occurred at 90° BTDC and 90° ATDC. This provided suitable timing reference points for each of the two pairs of cylinders.

Figure 3.20 describes how the CLK* signal (see Figure 3.19) is further conditioned depending on the resolution of encoder used. The second stage of this conditioning multiplied the signal by two and required an input with 1:1 mark-space ratio in order to give a steady output. The first stage divided the signal by two and was selected by a front panel switch for 360 pulse/rev encoders. This stage was bypassed for 180 pulse/rev encoders. One output CLK pulse per degree of crankshaft rotation was then obtained by correct selection of encoder and switch position. A front panel LED indicated the presence of the CLK signal. The CLK+2 signal was used for the frequency-to-voltage conversion to provide an analogue output voltage proportional to engine speed.

**Fuel Supply Relays:** The petrol supply relay controlled the high pressure pump and was itself energized from the interface rack as shown in Figure 3.21. The GATE signal was conditioned by a 1 s retriggerable monostable multivibrator in order to provide a SAFETY
ENABLE signal as long as the crankshaft was rotating. The petrol supply relay was energized and a front panel LED was lit when the the SAFETY ENABLE signal was TRUE (logic 0).

The hydrogen supply relay controlled a solenoid valve in the hydrogen supply line and was itself energized by the simultaneous occurrence of both a HYDROGEN ENABLE signal from the microcomputer and a SAFETY ENABLE TRUE signal. The engine therefore had to be running before the hydrogen supply solenoid valve could be operated. A front panel LED indicated when the hydrogen supply relay was energized.

**Injector Drivers:** Figure 3.22 describes the technique used to generate the stepped current drive for the fuel injectors. The 1.5 ms monostable multivibrator defined the maximum duration of the high current injection period, but the total injection period was controlled by the duration of the INJ signal from the microcomputer. The value of the resistance in series with the protection diode was chosen to limit the back e.m.f. to within the maximum permissible voltage for the driver transistors. The 30 ms monostable multivibrator was used to 'stretch' the injection pulse to make it visible via the front panel LED.

**Ignition Driver:** Figure 3.23 describes the ignition driver used to trigger the ignition coil primary circuit when commanded by the IGN signal from the microcomputer. The 1 ms monostable multivibrator defined the anti-dwell duration for the primary current. The value of 1 ms was chosen to provide an adequate dwell duration at an engine speed of 6000 r/min. At low engine speeds, the dwell time was excessive but the use of some modern high energy coils would have resulted in overheating and permanent damage of the coil. Thus, in order to use one of these coils, it would have been necessary to include a current-limiting circuit. A conventional coil was used and, of course, the long dwell time at cranking speeds provided a strong, stable spark for starting. The diode across the collector and emitter of the driver transistor limited the reverse voltage to 0.6 volts, while the pair of Zener diodes limited the maximum positive voltage across the transistor to 360 volts. The 30 ms monostable multivibrator was used to 'stretch' the ignition pulse to make it visible via the front panel LED.

### 3.2.5 Engine And Testbed

The engine type used for this work was a Ford T88 2-litre OHC fuel injected unit with specifications as described in Appendix E. The engine used the standard inlet arrangement, including the air filter and drove an electrically disconnected alternator but not a fan. An encoder was driven by a toothed belt from the crankshaft. In retrospect it may appear that it would have been advantageous to adapt the standard engine management system for hydrogen-petrol operation, as did May et al\[56,59\], but access was not available to the necessary development facilities at the time that the project was commenced.
The test-bed dynamometer used initially was a Hennan-Froude DPX hydraulic unit with remote electric control of the sluice gate. This was found to be most unsuitable for an engine calibration exercise because of the absence of any kind of automatic feedback speed control. It was found necessary to build a digital speed control mode into the development system software so that the throttle automatically adjusted the engine torque to maintain constant speed. In this way, other control variables, such as equivalence ratio and ignition timing, could be optimized relative to thermal efficiency without the engine operating point drifting. For the final results described in this thesis, a 'Hennan-Froude' Dynamatic eddy-current dynamometer was used in conjunction with a 'Test Automation' control system which provided speed control.

Petrol flow-rate was inferred by the control system on the basis of an accurate calibration of the injectors and the hydrogen flow-rate was measured with an electronic 'gap meter' as described in Section 3.2.3. Air flow-rate was measured by the control system's air flow meter as described in Section 3.2.2.

Engine coolant temperature was measured by a thermistor in the engine block and displayed on the test-bed control cabinet.

Ambient temperature was measured next to the engine with a thermometer and barometric pressure was measured using a Fortin barometer in an adjoining laboratory.

A commercial grade of petrol was supplied to the injector fuel rail via the standard high pressure pump and filter. The fuel pressure was maintained at 2.5 bar relative to inlet manifold pressure by means of a standard regulator. The set point of the pressure regulator was found to be sensitive to the back pressure applied to its return relief port and so a large bore line was fitted to return the unused fuel to the storage tank.

Commercial grade hydrogen (99.99%) was supplied in gas bottles via a pressure regulator as shown in Figure 3.24.

3.3 SOFTWARE DESCRIPTION

3.3.1 Development Environment
When the computer hardware was first assembled there was no software available to run on it, and so another microcomputer, running under the CP/M environment, was used to develop the operating environment for the hardware. The first piece of software written was an EPROM
based system monitor program which provided access to memory and I/O locations. With this facility it was possible to test all of the various system components, making sure that addresses and other options were correctly selected. This monitor program was then further developed to edit, execute and debug simple assembler programs. Many of the software drivers for the hardware components were initially developed with this facility.

In the final version of the control system, the monitor program was executed at POWER-ON or RESET unless a switch on the interface electronics rack was selected for the dedicated system. The disk based operating system was booted from the monitor program by typing ‘B (control-B) on the VDU keyboard.

In order to use the CP/M operating system, it was first necessary to configure the software to suit the particular hardware environment which therefore required the development of a Basic Input/Output System (BIOS) which interfaced between CP/M and the hardware. This involved the writing of software drivers for the two serial ports and the parallel printer port but the main task was that of writing the driver for the 8” ‘floppy’ disk drives. When fully developed, the BIOS was stored in EPROM rather than on disk. At boot time, the BIOS was loaded into RAM and then used to load the rest of the CP/M operating system from disk.

Once the operating environment was established, it was used to develop a suite of system maintenance utilities. A disk FORMAT program was written so that new media could be initialized. A disk COPY program was developed so that security copies of software and data could be made. A HEADCLN utility was written to aid the use of a disk head cleaning kit. A MONITOR utility for loading code from disk to RAM and then passing control to the system monitor program provided the means to use the monitor’s hardware trace facility to debug programs. A TPGEN utility was developed to write the operating system onto the system tracks of disks. A TPMOV utility was written to copy the system image generated by the CP/M utility, MOVECPM, to a buffer accessible by TPGEN. It was sometimes possible to create CP/M files with lower-case characters in the filename but it was impossible to delete such files with the CP/M ERA command and a DELETE utility was written to cope with this eventuality. In order to facilitate communication between the microcomputer and the mainframe which was used for some of the engine mapping work, it was necessary to have a utility which could handle file transfers and provide an ‘intelligent terminal’ facility. A copy of Columbia University’s CP/M-80 implementation of their KERMIT file transfer protocol was configured for the microcomputer and was used in conjunction with a version of KERMIT running on Loughborough University’s Honeywell mainframe.
The software development tools available on disk consisted of a 'screen' editor for creating the programs, a 8085 'macro' assembler for compiling the program code, a linker for combining compiled code for one or more program segments into a single executable code file and a symbolic program debugger for tracing program faults. No hardware facility existed with the microcomputer for programming EPROMS and therefore this task was carried out on another microcomputer development facility.

3.3.2 Control Software

The software for the control system was split into four main parts. The first part contained the executable code and a small amount of data storage. The second part contained the data tables which defined the mathematics and A/D interrupt driven processes while the third and fourth parts contained the hydrogen-petrol and petrol-only look-up-tables respectively. All of the software was written in 8085 assembler and conditional assembly directives were used to differentiate between the development and dedicated control systems.

The executable code for the control system may be sub-divided into the interrupt driven and free running categories. The structure of the free running software is described in Figure 3.25 and consisted of non-critical code associated with status checking and mode selection such as detecting that the hydrogen pressure was low and deciding that the petrol-only look-up-table should be used instead of the hydrogen-petrol version. In the development system this code also updated the VDU display and executed keyboard commands.

For a time-critical application such as engine control, the use of interrupt driven software is desirable because it allows the processor to be continuously doing useful work. For example, if two tasks were executed sequentially and the first had to waste time in a loop while waiting for some hardware response, then the whole code would be delayed by that time. If however the two tasks were interrupt driven so that they executed in parallel, say twenty instructions at a time for each task, then if task A were to be held up for any reason, task B could have a greater share of the processor's attention until the delay was cleared, thus making the most efficient use of the processor and completing both tasks in the shortest possible time. Of course there would be some software overhead associated with switching between tasks and this must be balanced with the delays saved.

The main update cycle was performed by two interrupt routines which were synchronized with each other. With reference to Figure 3.26, the cycle commenced with the initiation of the A/D interrupt process. This process sampled engine speed, driver control demand and air flow before initiating the mathematics interrupt process. Any remaining analogue channels were sampled concurrently with the mathematics process and then the A/D process terminated.
The mathematics process was sub-divided into three sections. The first section scaled the speed and driver-demand inputs to construct a pointer into the main look-up-table. The interrupt process then halted and called a procedure to extract the four values from the table for each of the five control variables corresponding to the closest points to the pointer. The interrupt process then recommenced with the second section.

The second section of the mathematics process linearly interpolated between the four selected points to calculate for the five control variables. The values of air mass flow, equivalence ratio and hydrogen energy fraction were used to calculate the injection durations for hydrogen and petrol. The interrupt process then halted and called a procedure to set the five physical outputs before recommencing with the third section.

The third section of the mathematics process was used in the development system to calculate the various scaled VDU display values. Just before this section terminated the process, it initiated the A/D interrupt process, thus establishing the update cycle.

Software listings for the control software have not been included in this thesis because of their size but copies are available upon request from The Department of Transport Technology, Loughborough University of Technology, Leicestershire, England. The listings are contained in two files CTLALG.MAC and LUTHED.MAC.

In retrospect it may have been more efficient to have written the control software in a high level language such as the version of PL/M available for the Intel 8097 series of microcontrollers. The multi-tasking executive in this software support package relieves the programmer of some of the more time-consuming development work.

3.3.3 Mathematics Process Calculations
This section describes the calculations carried out by the mathematics process.

Analogue Inputs: Each of the analogue channels was sampled four times per update cycle and averaged. Scale and offset calibrations were applied to convert the numbers to suitable engineering units and the scaled values were digitally filtered (1st order filter) to reduce the effects of signal noise as follows:

\[ \text{New Filtered Value} = B(\text{Previous Filtered Value}) + (B-1)(\text{New Value}) \]  

(3.13)
Look-Up-Table Interpolation: Engine speed and driver-demand pointers to the 32x32 look-up-table are calculated as follows:

\[
x = \frac{31 \times N}{N_{\text{max}}}
\]

\[
y = \frac{31 \times D}{100\%}
\]

With reference to Figure 3.27, the operating point \(g\) is pointed to by \(x\) and \(y\). The four closest look-up-table points \(a\), \(b\), \(c\) & \(d\) are therefore given by the coordinates \((x',y')\), \((x'+1,y')\), \((x',y'+1)\) & \((x'+1,y'+1)\). The distances \(ae\), \(eb\), \(eg\) and \(gf\), which are normalized with respect to the look-up-table grid size, are calculated as follows:

\[
ae = \frac{N - N'}{N}
\]

\[
\text{eb} = 1 - ae
\]

\[
eg = \frac{D - D'}{D}
\]

\[
gf = 1 - eg
\]

The linearly interpolated value of \(z_g\) is calculated as follows:

\[
z_g = gf((z_a \times \text{eb}) + (z_b \times \text{ae})) + eg((z_c \times \text{eb}) + (z_d \times \text{ae}))
\]

Given look-up-table values for \(\phi\) and \(\beta\), and a measured value of \(\dot{m}_a\), the required mass-flow-rates for petrol and hydrogen (\(\dot{m}_p\) and \(\dot{m}_h\)) are calculated as follows:

\[
\dot{m}_p = \frac{\phi \dot{m}_a (1 - \beta)}{(1 - \beta)((a/f)_p)_{\text{stoic}}} + \frac{\beta((a/f)_h)_{\text{stoic}}}{e}
\]
The required petrol injection duration was determined by first computing $\Delta m_p$ and then applying Equation 3.10 as follows:

$$\Delta t_p = \frac{60 \dot{m}_p}{N} \times 1.0266 \times 10^{-1} + 9.4281 \times 10^{-1}$$

(3.23)

The required hydrogen injection duration was determined by first computing $\Delta m_h$, applying Equation 3.12 to correct for the pressure regulator set-point sensitivity to flow-rate and finally applying Equation 3.11 as follows:

$$\Delta t_h = \left[ 30 \dot{m}_h \right] \left[ \frac{451.3}{(P_h - 0.1544 \dot{m}_h)} \right] \left[ \sqrt{\frac{T_h}{293}} \right] \times 4.2374 + 0.54$$

(3.24)

The first bracketed group in Equation 3.24 represents the required $\Delta m_h$. The second group corrects for changes in ambient pressure ($P_h$ set as gauge pressure) and the effects of flow-rate on the regulation set-point. The third group corrects for changes in ambient temperature but it should be noted that the actual temperature of the hydrogen upstream of the injectors was also dependent on the temperature of the hydride pack and on the desorption and expansion cooling effects of the hydrogen. The correction for pressure and temperature assumed that the injectors behaved as critical-flow nozzles having the following characteristic:

$$\dot{m}_h = \frac{K P_h}{\sqrt{T_h}}$$

(3.25)

Petrol injection timing was specified in degrees of crankshaft rotation relative to TDC but was set in terms of a time delay (in tens of $\mu$s) after the MARK pulse which occurred at 90° BTDC. The calculation is as follows:

$$\Delta t_{ijk} = \frac{60}{N} \left[ \frac{90^\circ - \alpha_{ij}}{360^\circ} \right] \times 10^5$$

(3.26)
Although not a control variable, the hydrogen injection timing was fixed at 80° BTDC to allow the longest possible injection duration (\(\alpha_{ij}>80°\) was not possible because of hardware constraints). The calculations were identical to those for petrol injection.

### 3.3.4 Fault Tolerance

Seeger\(^{[12]}\) describes various software and hardware measures which may be implemented to minimize the effects of electrostatic and electromagnetic interference within the control system.

**Digital Filtering:** Low-pass digital filters may be implemented in software, as described in Section 3.3.3, to minimize the effects of undesirable voltage transients.

**Variable Limits:** RAM faults in variables can be detected if the values exceed known maximum or minimum limits. This is useful in situations where a variable's new value is dependent on its previous value or when it is used as a table pointer.

**2-State Flags:** 2-state flags should always be tested for exact value only: If a flag is not in either state then it should either be recomputed or set to the 'safe' state.

**Fault Indicators:** RAM fault indicators, such as 10101010\(B\) and 01010101\(B\), may be interspersed throughout memory. These can be checked periodically and any change indicates a fault.

**Tables:** Tables should not be too long as they may look like code if a Program Counter (PC) error occurs. It is a good idea to follow tables with a few consecutive 'RST' instructions which will ensure a jump to an error recovery procedure.

**Unused Memory:** Unused memory should be filled with 'NOP' instructions and terminated with an 'RST' instruction to ensure a jump to an error recovery procedure.

**Watch-Dog Timers:** If a PC error leads to execution of a long table it could be several tens of milliseconds before the error is detected. In fact the processor could find itself in an infinite loop. It is common practice to use a hardware timer which resets the processor periodically, say every 50 ms. In normal operation the software re-initializes the timer within the 50 ms period so that system RESET does not occur. Should the PC becomes corrupted and cause data to be executed, the timer will attain 'terminal count' and reset the system.

No spare hardware timers were available so software timers were used as a compromise. The interrupt routine servicing the crankshaft GATE pulses was used to decrement two software
counters. One counter was used to check the integrity of the interrupt driven UPDATE cycle and the second was used to check the integrity of the free-running STATUS cycle. If either of these two counters was allowed to reach ZERO, program execution passed to the RST70 interrupt vector and from there to an entry point in the program initialization code. This interrupt vector corresponded to the RST instruction 0FFH which was also interspersed between the data tables.

**Separate Stacks:** Each of the interrupt driven processes was given its own stack so that they were completely independent of each other. This overcame some of the development problems associated with the overflow of 'operating system' stacks.

### 3.3.5 Update Cycle Timing

The frequency of the update cycle determines the response characteristic of the control system. Too low a frequency will result in a sluggish system with significant control errors following transients. Frequencies above the maximum rate at which controlled events take place are of no additional benefit so this maximum rate is taken as the optimum frequency.

In most engine management systems the ignition frequency corresponds to the optimum and in the case of a four cylinder, four-stroke engine running at 6000 r/min this corresponds to 200 Hz (5 ms). This frequency would be an order of magnitude higher than necessary at idle speed but this would not be detrimental.

Theoretically it should be possible to determine the update frequency from the instruction timings but this can be dependent on conditional software branching and on the insertion of 'wait states' by the processor because of delays during access of 'off board' memory. The update period was therefore measured directly by toggling one of the 'bits' of one of the output ports each time the process repeated. An oscilloscope was used to measure the relevant time intervals and the results, with reference to Figure 3.28, are described in Table 3.1.

<table>
<thead>
<tr>
<th>Mode</th>
<th>A/D Update Time</th>
<th>Maths Update Time</th>
<th>Update Cycle Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Development</td>
<td>8 ms</td>
<td>168 ms</td>
<td>174 ms</td>
</tr>
<tr>
<td>Dedicated</td>
<td>6 ms</td>
<td>99 ms</td>
<td>105 ms</td>
</tr>
</tbody>
</table>

The excessive cycle times indicated are partly due to the use of an 'old technology' processor but are mainly due to the almost exclusive adoption of 32-bit floating-point arithmetic in the mathematics update process. These times could be reduced by an order of magnitude if 16-bit
fixed-point arithmetic were implemented in addition to the pre-calculation of all constants during initialization.

3.4 SUMMARY OF CONTROL STRATEGY

The preceding sections in this chapter dealt with the individual components of the control system but the opportunity is taken here to summarise the strategy used and the philosophy behind its selection.

3.4.1 Closed-Loop Versus Open-Loop Control

In general, a controller may be of two types; closed-loop or open-loop. Closed-loop controllers feedback one or more system 'outputs' so that they may be controlled accurately. Equivalence ratio control via an exhaust oxygen-concentration sensor is an example of closed-loop operation. This type of control is useful when a distinct target value has been determined for the parameter concerned because it reduces the requirements for accurate system calibrations and the problems associated with variable actuator characteristics. The difficulty with the implementation of a closed-loop controller lies in the choice of suitable feedback parameters, target values and sensors for the system, together with the determination of the form of the function relating 'inputs' to 'outputs'.

Open-loop controllers assume a fixed relationship between system 'inputs' and 'outputs' and therefore require accurate calibration. In the case of the control system described in this thesis, the required 'output' was defined as maximum thermal efficiency, with the constraint that certain defined maximum levels of NO$_x$ and C$_3$H$_8$ emissions must not be exceeded. Future development of combustion sensors may provide the means to 'close the loop' on these parameters but for the purposes of this work it was necessary to adopt an open-loop strategy.

An open-loop strategy may be based on a mathematically defined relationship between 'inputs' and 'outputs' but this assumes a prior complete understanding of the system's performance characteristics. Where the system's characteristics are complex or unknown, it is more practical to use a look-up-table technique to define the controller's calibration and this was the strategy adopted for the hydrogen/petrol control system.

The control system described in this thesis used engine speed and driver-demand coordinates into a look-up-table consisting of a 32 x 32 matrix of steady-state engine operating points. Each location in the matrix contained appropriate values of ignition timing, throttle angle, petrol injection timing, equivalence ratio and hydrogen energy fraction. The values of these control
variables which corresponded to the four matrix locations closest to the input coordinates were interpolated to yield the final control variable values. The relationship between torque and driver-demand at any engine speed was built into the look-up-table data by cross-reference between a driver-demand/speed map (defining engine torque as a function of speed and demand) and torque/speed maps for the control variables (defining control variables as functions of speed and torque).

Ignition timing, throttle angle and petrol injection timing were directly implemented in hardware from the control variables but, in conjunction with stored injector calibrations and the values of both equivalence ratio and hydrogen energy fraction, an additional system input, air mass-flow-rate, was required so that the hardware values of petrol and hydrogen injection durations could be determined and implemented.

3.4.2 Injector Sequencing

Hydrogen was injected at the throttle body four times every engine cycle in order to provide an even distribution between the four cylinders. This strategy also helped to minimize the pressure fluctuations upstream of the injectors thereby reducing the 'noise' on the flow-rate measurement system and eliminating some resonance effects at the injectors during low engine speed operation. The one major drawback of this method of sequencing was that it increased the linearity-limited minimum mass-flow-rate of the injectors.

The petrol injection sequencing was 'software selectable' to be one of three strategies. Simultaneous Twice per Cycle (SM2) operation is the most commonly used strategy because it is the least expensive and the simplest to implement. SM2 operation provides identical 'cycle resolved' injection timing for cylinders 1 and 4 but those for cylinders 2 and 3, although equal to each other, are 180° out of phase with respect to 1 and 4.

Sequential Twice per Cycle (SQ2) operation overcomes the 'phasing' difficulties of the SM2 strategy, thus helping to minimize cylinder-to-cylinder variations, but requires two injector driver stages instead of the one required by SM2 operation. With both of these sequencing patterns some fuel is always deposited in the inlet port thus creating a 'fuel reservoir-based' transient equivalence ratio problem. The reason for this problem is that the reservoir introduces a 'lag' into the fuel distribution system.

Sequential Once per Cycle (SQ1) sequencing, in conjunction with appropriate injection timing, can provide part-load operation for which the fuel is only injected while the inlet valve is open thus removing the 'fuel reservoir-based' transient problem. In addition to this benefit, SQ1 operation provides twice the injection duration for the same mass-flow-rate relative to the other
sequencing strategies and this halves the linearity-limited minimum mas-flow-rate of the injectors thus improving low-load calibration. Unfortunately, SQ1 operation requires an additional 'cycle position' sensor and is likely to produce in-cylinder equivalence ratio stratification which might just as easily be detrimental as it could be advantageous. Quader\textsuperscript{125} have shown that careful swirl control is a prerequisite for reliable stratification and therefore the further investigation of SQ1 operation for hydrogen/petrol operation was considered to be outside the scope of this work.

SM2 operation was used for all of the work presented in this thesis because of its simplicity and common usage.

3.4.3 Equivalence Ratio Transient Compensation
During throttle transients, the rate of change of air mass-flow-rate is usually much faster than the system can sense (because of the inertia of the moving vane in the air flow-meter) and faster than the fuel distribution system can react due to reservoir effects. With a multi-point 'port' fuel injection system, as used for this work, the fuel reservoir effects are relatively small and the main transient problem is associated with the lag in the measurement system. In order to compensate for this transient problem, the controller factored the measured air mass-flow-rate by the percentage change in throttle angle about to be commanded; the assumption being that a percentage change in throttle angle would yield an identical percentage change in air mass-flow-rate.

3.4.4 Transition Between Operation Modes
The operation switches between petrol-only and hydrogen/petrol operation according to the supply pressure from the vehicle's storage unit. When the hydrogen pressure is low, such as when the supply is exhausted or when the hydride pack is cold, the operation switches to the petrol-only look-up-table and when the pressure increases, the strategy reverts to hydrogen/petrol operation.
Chapter 4

LOOK-UP-TABLE CALIBRATION
4.1 CALIBRATION CONCEPT

Having described the design and function of the control system in Chapter 3, it is now necessary to describe the method by which the system was optimally calibrated for a particular engine.

Calibration consisted of measuring the engine's performance as it was affected by the control variables at a number of points on the engine's torque/speed map. For each operating point, contour maps describing the engine's performance as a function of equivalence ratio and hydrogen energy fraction were generated.

The contour maps typically indicated a conflict between thermal efficiency, NO\textsubscript{x} emissions and C\textsubscript{3}H\textsubscript{8} emissions. For the purposes of this research, no attempt was made to identify the appropriate maximum limits of exhaust emissions but such constraints, derived from driving cycle emission simulation studies, may be superimposed on the thermal efficiency map to facilitate the choice of optimal values of $\phi$ and $\beta$. Using these values as coordinates into the maps of ignition timing and throttle angle, yields the complete set of control variables for the optimized engine operating point.

Figure 4.1 describes a typical thermal efficiency map for the engine operating point of 2000 r/min and 53 Nm. The thick boundary defines the intersection of the areas within which the NO\textsubscript{x} and C\textsubscript{3}H\textsubscript{8} emissions are both less than 2 $\mu$gJ\textsuperscript{-1}. In this instance, the point of maximum thermal efficiency falls within the emissions constraints but it should be noted that these are high by current standards and would certainly lead to failure of the ECE 15.04 driving cycle emissions test. Figures 4.2 and 4.3 present the ignition timing and throttle angle maps corresponding to Figure 4.1.

Having collected calibration data in the above manner for points throughout the engine's torque/speed map, new contour maps were produced describing the control variables and performance parameters as steady-state functions of speed and torque. The control variable maps were then used in conjunction with a demand/speed map to generate the look-up-tables referenced by the control system.

4.2 EXPERIMENTAL PROCEDURE

In order to start the engine, it was necessary to have an initial calibration and this was effected by selecting a value of unity for $\phi$ and zero for $\beta$. Ignition timing was based on a linear function of engine speed and driver-demand. Throttle position was made proportional to demand and petrol injection timing was set to TDC.
Once the engine was warmed up, the speed was set to the desired value by means of the
dynamometer controller and the brake torque was set by adjusting the throttle angle via the
driver-demand potentiometer. Having attained the desired engine operating point, the look-up-table
was disabled to facilitate the manual adjustment of the remaining control variables. β values of 0%,
10%, 20% and 30% were selected, and at each of these a number of φ values between unity and the
lean extinction limit were set and data recorded. MBT ignition timing was implemented at each test
point. This data consisted of engine speed (N), brake torque (b.t.), equivalence ratio (φ), hydrogen
energy fraction (β), ignition timing (α_{ign}), throttle angle (θ), thermal efficiency (η_t), volumetric
efficiency (η_v), petrol mass-flow-rate (\dot{m}_p), hydrogen mass-flow-rate (\dot{m}_h), air mass-flow-rate
(\dot{m}_a), exhaust temperature (T_{ex}) and exhaust concentrations of CO, CO_2, NO_x and C_3H_8
emissions. Appendix F contains tables of data for various torque values between zero and full
load at speeds between 1000 r/min and 5000 r/min. The values of specific exhaust emissions given
in these tables were calculated with the aid of a fortran program (EMISSIONS.FORTRAN), the
listing of which is given in Appendix G. This program was developed from a similar one written
by Dr Richards[57].

4.3 CONTOUR GENERATION SOFTWARE

4.3.1 Surface Fitting
The task of generating the contour maps for this work consisted of two parts. The first part was
to generate 'z' values corresponding to a regular grid of points in the 'x' - 'y' plane and this was
effected by fitting a 'surface' to the experimental data and then re-evaluating at the required grid
points. Appendix H describes the fortran program (SURFIT.FORTRAN) which was
developed to perform this task.

To meet the requirements of the SURFIT.FORTRAN program, the input data had to be
collected in several constant 'x' groups spread over the 'x' - 'y' plane as shown in Figure
4.4. The limits of the final surface were defined by the minimum and maximum 'x' values and
the minimum and maximum 'y' values at each of the intermediate 'x' values in the input data.
In this way, the final surface was forced to have 'x' limits parallel to the 'y' axis but the 'y'
limits were permitted to be ragged as shown in Figure 4.5. In order to achieve a regular grid,
the 'y' data was transformed to 'y*' as follows:
In Equation 4.1 'y_{min}' and 'y_{max}' correspond to the current value of 'x' so that a regular grid was obtained with rectangular limits of 'x = x_{min}', 'x = x_{max}', 'y^* = 0' and 'y^* = 1'.

At each of the input values of 'x', the data points were interpolated with 'y^*' being considered as a function of 'y' to give points at each required grid position in the 'y^*' direction. The intermediate 'x' values in the grid were generated by interpolating with 'y' and 'z' each being considered as functions of 'x'.

Program SURFIT.FORTRAN provided a choice of linear or cubic-spline interpolation techniques but the former method was chosen for the maps described in this thesis so as to avoid giving a false impression concerning the reliability of the data between the original input points.

It was important that the 'z' versus 'y^*' relationships at each constant 'x' value be single-valued and, for this reason, it was sometimes necessary to discard some input data if this requirement was not met. An example of such a situation was in the case of 'x' being engine speed, 'y' being brake torque and 'z' being throttle angle. As throttle angle was increased at constant speed, the torque usually increased to a maximum before the throttle was fully open. At low engine speeds, the torque sometimes even fell away with increasing throttle angle, thus resulting in a multi-valued 'z' versus 'y^*' relationship. In this situation the points following the maximum torque point had to be discarded.

4.3.2 Contour Tracing

The second part of the task of generating contour maps was to select contour values and trace their occurrence through the rectangular grid generated by SURFIT.FORTRAN. Appendix I describes the fortran program (CONTPLOT.FORTRAN) which was written to carry out this procedure. This program was developed in collaboration with Tryphonos[122].

CONTPLOT.FORTRAN requested the user to provide the name of a file, generated by program SURFIT.FORTRAN, containing the definition of the surface under consideration. The input file was scanned to find the minimum and maximum 'z' limits and the user was then requested to specify the required contour levels.
The surface grid was scanned systematically until the required contour value was found to occur between two consecutive grid points. The contour was then traced through the grid by considering the four triangles made by the four corners of each grid square and the centre of the square as shown in Figure 4.6. The 'z' value corresponding to the centre was taken as the average of those for the four corners. Should the contour enter one side of a triangle, a simple calculation determined the exact crossing coordinates. The contour must then exit the triangle via one of the remaining two sides and again the exact coordinates were easily calculated. Alternatively the contour may pass exactly through one of the grid nodes thereby providing up to six options for the new exit direction. The four possible situations are indicated in Figure 4.7. Should the contour exit the grid instead of closing the loop, the original starting point was used to commence tracing in the opposite direction. Every grid square on the map was checked for a new starting point before a contour level was considered complete.

4.4 CONTROL VARIABLE OPTIMIZATION

Section 4.1 described in general terms the optimization process for selecting a set of control variables for a particular torque/speed engine operating point. Once values of $\beta$ and $\phi$ have been selected, determination of the corresponding values of the remaining control variables is straightforward. This section will therefore concentrate on the philosophy concerning the selection of values for $\beta$ and $\phi$ in order that optimum performance might be achieved. The main criterion for optimization was the maximization of thermal efficiency (or the minimization of the fuelling rate) although at some operating conditions the reduction of exhaust emission levels and conservation of the hydrogen storage capacity were also considered. The top and bottom of the load range were special cases where petrol-only operation was implemented and so these are dealt with first.

Zero-Load: At all engine speeds, petrol-only fuelling was used for zero-load operation and both ignition timing and throttle angle were optimized to maintain the desired speed with the minimum fuelling rate. The addition of hydrogen at this load condition may well have helped to reduce the energy usage rate and CO$_2$ emissions but it was considered that the benefits, in terms of driving cycle performance, would be outweighed by the requirement for hydrogen conservation which was imposed by the limitations of the vehicle's hydrogen storage system capacity. The experimental data corresponding to this optimization process is given in Appendix F under the headings "Zero Load, Petrol Only".

Minimum-Load: Minimum-load running is defined as operation on the engine's 'over-run' absorption torque line and it was therefore impossible to measure the engine's performance
without the use of a 'motoring' dynamometer facility. It was necessary, however, to define control variable settings for this operating condition. Thus, the zero-load settings (for the same engine speeds) were also chosen for this operating condition with the exception that the throttle angle was set to minimum and $\phi$ was set to 0.9. At this load condition it would have been possible to set the fuelling rate to zero however this would have allowed the cylinders to 'cool' during the over-run condition and, upon recommencement of fuelling, would have resulted in misfiring or partial-burning of the mixture.

**Idle:** The 'idle' condition may be defined as the operating point occurring at the intersection of the minimum-load and zero-load lines. Petrol-only operation was chosen on the basis of hydrogen conservation even though the idle condition makes a significant contribution to the total energy usage during typical driving cycles. An equivalence ratio of 0.9 was chosen to give a 'smooth' idle, while ignition timing was optimized to facilitate the smallest throttle angle for the chosen idle speed (this having been arbitrarily set at 800 r/min) thereby minimizing the fuelling rate. The requirement of a stable idle speed (assuming an open-loop controller) implies that the minimum-load line must continue to increase as engine speed decreases from the desired set-point. Should the engine speed drop, due to some small additional external loading, the subsequent increase in torque will help to compensate. Any increase in engine speed will be countered by a drop in torque as the engine enters the over-run condition.

**Cranking:** Due to engine 'roughness' and dynamometer limitations, no actual testing was carried out below idle hence the following arbitrary strategy was implemented. Petrol-only operation was adopted for simplicity. At zero engine speed, ignition timing and equivalence ratio were set to TDC and unity respectively. The values for these parameters were then varied linearly with engine speed to the appropriate values at 1000 r/min for the equivalent load conditions. Throttle angle was defined to be proportional to load at zero speed and, in an attempt to improve idle speed stability, the minimum-load angle was increased sharply from zero speed before falling steeply to the idle condition. This strategy succeeded in producing a significant back-up torque characteristic but due to 'coarse' throttle actuator resolution and controller 'lags' it resulted in idle speed oscillation. The final strategy used a constant throttle angle for the minimum-load line between zero speed and idle but, as a result of the increase in volumetric efficiency as engine speed decreased (causing the air flow to drop below the choking condition), some stability was still obtained.

**3/4-Load:** The fractional quantity 3/4 is used here to denote the load point at which the throttle has just been fully opened and above which equivalence ratio is the only means of load control. As thermal efficiency was not a major consideration and moreover because the use of hydrogen would have decreased volumetric efficiency and hence the maximum torque, petrol-only
operation was used. The lowest load point in this range corresponded to the lean equivalence ratio which gave maximum thermal efficiency at full throttle while the maximum load point corresponded to the equivalence ratio which gave maximum torque. MBT ignition timing was implemented at each point. The experimental data corresponding to this optimization process is given in Appendix F under the headings "Full Throttle, Petrol Only".

**Part-Load:** This load range, defined as the entire operating range between zero-load and 3/4-load operation, has the greatest impact on performance during typical driving cycles. It is here that the introduction of hydrogen supplementation is most beneficial and the use of the optimization maps, describing thermal efficiency, NO\textsubscript{x} and C\textsubscript{3}H\textsubscript{8} as functions of equivalence ratio and hydrogen energy fraction, is described below. MBT ignition timing was implemented for each point on these maps but future work could benefit from the inclusion of ignition timing as a third optimization variable. This would imply a more tedious programme of testing as well as the development of display techniques for three-dimensional contours but would greatly improve the understanding of the trade-off between thermal efficiency and NO\textsubscript{x} emissions.

Having no experimental or analytical basis for selecting NO\textsubscript{x} and C\textsubscript{3}H\textsubscript{8} limits (other than the fact that the typical values observed were likely to be too high), an attempt was made to establish a pattern for optimum $\beta$ and $\phi$ values over the load/speed range. The thermal efficiency, NO\textsubscript{x} and C\textsubscript{3}H\textsubscript{8} maps (corresponding to specific speed/torque operating points) were laid out in a speed/torque grid pattern and studied with a view to establishing the occurrence of trends in $\beta$ and $\phi$. By avoiding any sharply rising emission levels, it was found that for half-load operation the optimum value of $\beta$ varied from 30\% at 1000 r/min to 15\% at 5000 r/min. Over the same range, the optimum value of $\phi$ varied from 0.6 to 0.65. It was considered that these trends, although apparent, represented an over-simplification of the control requirements. Further, in consideration of the known importance of low-speed operation to vehicle performance over driving cycles, it was decided that the actual optimum values of $\beta$ and $\phi$ should be used for all of the part-load points at 1000 r/min and 2000 r/min. For 3000, 4000 and 5000 r/min and part-load operation, $\beta$ was set at approximately 15\% and $\phi$ optimized for maximum thermal efficiency. The optimized data for the first two engine speeds is presented in Appendix F under the headings "Optimized Data" and for the remaining three engine speeds under the headings "Part Load". Presented under the same headings in Appendix F is the petrol-only operating data for which $\phi$ has been optimized for maximum thermal efficiency. This data, in conjunction with the other petrol-only data, forms the basis for the petrol-only look-up-table to which the controller reverts when the vehicle's hydrogen supply is exhausted.
4.5 THE DRIVER-DEMAND MAP

In a conventional spark-ignition engine, the 'driver-demand' (D) is implemented in terms of accelerator pedal position which is related by a fixed mechanical linkage to throttle angle. The shape of the constant demand lines on the engine's torque-speed map defines the engine's open-loop torque characteristic. The engine is forced to follow the selected driver-demand line when its load, represented by the load line, increases or decreases with external conditions. Figure 4.8 describes a typical spark-ignition engine characteristic where the accelerator pedal and throttle are linked mechanically.

For the hydrogen/petrol system, the absence of any direct mechanical control of the engine by the driver required the design of a 'driver-demand' map in order to define the engine's torque/speed relationship for all demands from 0-100%. The 100% demand line was taken to be the maximum torque curve and the 0% demand line was taken to be the engine's absorption torque curve as measured by motoring the running engine at closed throttle. For simplicity, the intermediate demand lines were equi-spaced, as shown in Figure 4.9, to provide a proportional torque/demand characteristic at every engine speed. Figure 4.10 describes the torque/demand characteristics at several engine speeds for the spark-ignition engine map described in Figure 4.8 along with a proportional characteristic. The conventional characteristic exhibits torque saturation at high and low demands depending on engine speed and, as a consequence, the average gain over the effective range of driver-demand is higher than that of the proportional characteristic. The saturation at low speeds and large throttle openings is due to the ineffectiveness of the throttle plate when the engine's demand for air is low. The saturation at high speeds and low throttle openings is due to early inlet choking when the engine's demand for air is high.

The thick lines on the maps presented in Figures 4.8 and 4.9 represent typical steady-state vehicle load-lines. Assuming constant external conditions, the engine will operate on the appropriate load-line while acceleration remains zero. The intersection of the appropriate driver-demand line and vehicle load-line therefore defines the steady-state operating point of the engine for those conditions. Should a change in the external loading occur, without any change in driver-demand, the change in the engine operating point will be governed by the shape of the appropriate driver-demand line. If the driver-demand line is steep, the change in engine speed would be small, as compared with that which would occur if the driver-demand line is near to the horizontal. Conversely, should the external loading remain constant while driver-demand alters, the change in speed would be defined by the torque versus driver-demand characteristic described in Figure 4.10. The topography of the driver-demand map therefore defines the engine's open-loop steady-state response to driver-demand and engine loading and it is the
The author's opinion that this has significant impact on the driver's perception of the vehicle's performance. Future work will be necessary to evaluate the relative merits of various characteristics, other than the proportional version described here, such as one with steeply decreasing control lines (from left to right) similar to those obtained with 'all speed' governors on diesel engines. Work is continuing at Loughborough\textsuperscript{[123,124]} in the development of a vehicle, featuring drive-by-wire throttle control, to be used for driveability studies.

Appendix J describes the fortran program DIGITIZE.FORTRAN which was used to generate the driver-demand map shown in Figure 4.9. This program requested the user to enter the 0\% and 100\% demand lines by either specifying an input file or by entering the data manually via a digitizing facility. The intermediate demand lines for the proportional map were then generated automatically by the program.

4.6 LOOK-UP-TABLE GENERATION

Appendix K describes a fortran program (GENERATE.FORTRAN) which was used to combine the information from the engine control maps (generated by SURFIT.FORTRAN) with the driver-demand map (generated by DIGITIZE.FORTRAN) to generate the look-up-tables for the control system.

GENERATE.FORTRAN accepted input data from a control file which defined the structure of the required look-up-table. For each selected engine speed and driver-demand coordinate in the look-up-table, the program found the required engine torque from the driver-demand map. The engine control maps (one for each control variable) were used to find the set of control variables corresponding to the relevant engine speed and torque. In this way the complete look-up-table was evaluated.

Figures 4.11 - 4.14 describe the engine maps used to define the values of ignition timing, throttle angle, equivalence ratio and hydrogen energy fraction for hydrogen/petrol operation. The map for petrol injection timing has not been presented because it was decided to set this variable to zero for the entire engine operating map. The reason for this decision was that, not only was the optimization process already both complicated and time-consuming but, in addition, Quader\textsuperscript{[125]} and engineers at Toyota Motor Corporation\textsuperscript{[88]} have indicated that the control of engine swirl is necessary if the sequential timing of petrol injection is to be of any benefit.
The throttle angle map in Figure 4.12 is of special interest in that it clearly indicates the non-linearity of the relationship between torque and throttle angle. Also of interest, is the hydrogen energy fraction map in Figure 4.14 which indicates that hydrogen supplementation is most beneficial at part-load and low engine speed thus supporting the view of the author and other researchers[51] that hydrogen addition to hydrocarbon-air mixtures affects mainly the early stages of combustion before turbulent flame propagation becomes important. This topic is dealt with in more detail in Chapter 5.
Chapter 5

PERFORMANCE RESULTS
5.1 STEADY-STATE ENGINE TEST-BED PERFORMANCE

The data in this section was extracted from the engine test data tables in Appendix F and corresponds to the control variable maps described by Figures 4.11 - 4.14. The data are presented in the form of contour maps on the speed/torque plane and map boundaries correspond to practical constraints. It should be noted that the contour generation process was based on a limited quantity of data and it is therefore appropriate to draw only general conclusions based on the overall trends displayed by the maps.

5.1.1 Thermal Efficiency

Figure 4.15 describes the map of thermal efficiency for hydrogen/petrol operation. The map indicates a typical trend of thermal efficiency which increases from 0% at zero load to a maximum just short of maximum torque. The underlying reason for the improvement in thermal efficiency between these points may be explained as follows. At maximum-load the throttle is fully open resulting in minimum pumping losses and low residuals (which affects flame speed) while the equivalence ratio is relatively rich giving maximum torque. From this point the throttle remains fully open and the equivalence ratio is made leaner (increased $\gamma$) until maximum thermal efficiency is achieved. Between the maximum thermal efficiency point and zero-load the throttle is progressively closed and this is the major factor affecting thermal efficiency. As throttling increases so too does the pumping losses and residuals, the latter being due to the effective lowering of the compression ratio. This increase in residuals retards the combustion process thereby lowering the efficiency. The reduction in charge mass, due to throttling, results in a decrease in the value of $\gamma$ thus contributing to a reduction in efficiency. At zero-load, the engine is producing zero brake torque and therefore, by definition, the thermal efficiency is also zero.

This characteristic is also reflected in Figure 4.16 which describes the relationship between thermal efficiency and torque at 2000 r/min. The peak thermal efficiency was obtained at full throttle, petrol-only operation and lean equivalence ratio. The top 20% of the engine's torque was controlled solely by the equivalence ratio and this constituted the main reason for adopting a drive-by-wire approach to the engine control problem. Figure 4.16 also indicates the improvement in thermal efficiency at part-load between hydrogen/petrol and petrol-only operation and Figure 4.17 describes this improvement, in terms of fuel energy saving, for the entire engine map. Below zero-load and above approximately 75% load, petrol-only operation was implemented and therefore no improvement is indicated. Maximum energy saving occurred at approximately 15% load and at low engine speed.
The increased benefit of hydrogen addition at low engine speed may be explained in terms of the relative importance of the laminar burning velocity ($u_l$). At slow engine speeds, where the r.m.s. turbulent velocity ($u'k$) is low, the contribution of $u_l$ to the turbulent burning velocity ($u_t$) is very important and any increase in $u_l$ due to the addition of hydrogen will significantly increase the burn-rate of lean mixtures. At higher speeds the contribution of $u_l$ to $u_t$ becomes less significant and consequently there is less benefit in supplementing with hydrogen. Bradley et al.[126] have shown that as $u'k$ increases with engine speed, the value of $u_t$ also increases but that the effect of flame straining decreases the ratio of $u_t/ u'k$, thus increasing the combustion duration in terms of crank angle and hence combustion efficiency. At high engine speeds and lean equivalence ratios, the effects of flame straining, which Bradley et al.[126] have shown to affect mainly the early stages of combustion, may partially or totally quench the flame. Abdel-Gayed et al.[127] have shown that lean hydrogen flames benefit from flame straining due to hydrogen's high diffusion coefficient and therefore supplementation with hydrogen may help to prevent bulk quenching at high engine speeds and lean equivalence ratios.

The increased benefit of hydrogen addition at low load may be explained in terms of throttling and equivalence ratio. Relative to petrol-only running, the throttle is more open for hydrogen/petrol operation and this results in lower pumping losses and a smaller residual mass fraction which in turn contributes to an increase in combustion rate. In addition, the higher value of $\gamma$ for the leaner mixture results in improved combustion efficiency while the faster flame speed of hydrogen helps to maintain constant-volume combustion.

The controller's optimized hydrogen energy fraction map, described in Figure 4.14, clearly indicates that, of all the engine operating regimes contained in the map, the low-load, low speed regime derives the most benefit from hydrogen supplementation.

5.1.2 Volumetric Efficiency

Figure 4.18 describes the volumetric efficiency map for hydrogen/petrol operation. The map indicates an increase in volumetric efficiency as the throttle is opened. At low loads, the relatively small throttle angles tend to dampen the effects of inlet tuning and so there is no significant change in volumetric efficiency with speed. However, as full throttle operation is approached (at approximately 3/4 load), these effects become more apparent and higher volumetric efficiencies occur at speeds to which the engine has been tuned. The engine speed at which the tuning effects are most beneficial is determined by valve timing and manifold design and, from the map, it appears that the inlet tuning may have been optimized for high engine speed at which point values of volumetric efficiency in excess of 100% occur.
5.1.3 Exhaust Temperature

The exhaust temperature map described in Figure 4.19 indicates trends of increasing temperature with both speed and load. The increase in temperature with speed is due to the reduction in the time available for the exhaust gases to lose heat to the cylinder walls. The less significant increase in exhaust temperature with load is due to a corresponding increase in cylinder charge mass which therefore results in a smaller reduction in the burnt-gas temperature. This temperature information is important for the design of heat exchangers such as might be required for the on-board generation of hydrogen from the main fuel or for heating hydride storage systems.

5.1.4 NO\textsubscript{X} Emissions

The generation of NO\textsubscript{X} in the combustion process is controlled by a complex set of rate-controlled chemical equilibrium reactions. If equilibrium conditions are assumed, the concentration of NO\textsubscript{X} is dependent only on temperature, pressure and oxygen concentration but Daneshyar & Watfa\textsuperscript{[128]} and Benson & Whitehouse\textsuperscript{[129]}, among others, have modelled the dynamic process and shown that the formation and dissociation of NO\textsubscript{X} is limited by the forward and reverse reaction rates respectively. The forward rate-controlled reaction limits the maximum NO\textsubscript{X} concentration to a value lower than that predicted by equilibrium conditions but the slower reverse reaction causes the NO\textsubscript{X} concentration to 'freeze' as the burnt-gas temperature falls below about 1800 K. It can therefore be said that both high temperature and high pressure lead to high NO\textsubscript{X} concentrations and Quader\textsuperscript{[130]} has shown experimentally that long combustion time durations result in lower NO\textsubscript{X} concentrations.

The effect of engine speed on NO\textsubscript{X} concentration is complex but there are two contributing factors which take prominence. Increasing engine speed gives rise to higher NO\textsubscript{X} concentrations due to a decrease in combustion time duration but this is somewhat offset by the reduction in average temperature and pressure which results from the more advanced MBT ignition timing. Figure 4.20 describes the NO\textsubscript{X} performance for hydrogen/petrol operation and supports the trend of increasing emission with engine speed.

The increase of NO\textsubscript{X} emission with engine load, observed in Figure 4.20, is due to the following: (a) increased peak temperature and pressure - a result of increased charge mass and an equivalence ratio which is closer to stoichiometric and, (b) reduced residual mass-fraction, due to increased throttle opening, which results in less charge dilution and retarded MBT ignition timing thus leading to an increase in the average temperature and pressure.

The high initial level of NO\textsubscript{X} emission observed in Figure 4.20 at low-load and slow engine speed may be explained with reference to Figure 4.14. It is at this low speed, low-load
operating condition that hydrogen supplementation is greatest and the higher adiabatic flame temperature of hydrogen (about 200 °C higher than petrol) causes an increase in NO\textsubscript{X} concentration. Figure 4.21 presents further evidence that this high NO\textsubscript{X} emission results from adopting hydrogen supplementation, demonstrated by the fact that the map indicates a sharp increase in NO\textsubscript{X} at the low speed, low-load condition. The improvement in part-load NO\textsubscript{X} emission, over the speed range, can be attributed to very lean operation as is evidenced by Figure 4.13 which indicates the use of an equivalence ratio of approximately 0.6 over most of this operating range.

5.1.5 C\textsubscript{3}H\textsubscript{8} Emissions
The 'flame ionization detector' hydrocarbon analyser used for this work was calibrated for propane hence the designation C\textsubscript{3}H\textsubscript{8} will be used here even though a wide spread of molecules of differing structure and molecular weight may have actually been present in the exhaust gases. Exhaust hydrocarbons can arise from a variety of sources in the combustion chamber, such as crevice volumes, oil film absorption, wall quenching and bulk quenching. During the compression stroke, hydrocarbons are dissolved into the cylinder wall oil film and some mixture is compressed into crevice volumes in the piston ring pack, head gasket interface and even the sparkplug threads. During combustion, these fuel storage locations are protected from the flame by the relatively cool metal and towards the end of the exhaust stroke these reservoirs expel the unburnt hydrocarbons into the exhaust stream. Quenching occurs when the flame burns close to the relatively cool combustion chamber walls and it is here that the shorter quenching distance of hydrogen (see Table 1.1) shows some benefit over petrol. Bradley et al\cite{126} have shown that excessive turbulence and/or very lean equivalence ratios can lead to partial or total bulk quenching of the flame - especially during the early stages of flame propagation when the addition of hydrogen could help significantly.

Figure 4.22 describes the map for C\textsubscript{3}H\textsubscript{8} emissions during hydrogen/petrol operation and indicates a general increase as engine speed decreases. This trend may be attributed to the rate at which hydrocarbons can be dissolved into the cylinder wall oil film and to the temperature of the burnt gases on the expansion stroke. At high speeds, there is less time for the oil film absorption process and so only small quantities of hydrocarbons are stored. The high burnt gas temperature occurring at high engine speeds also helps to oxidize the hydrocarbons as they emerge from the oil film. At low engine speeds, the reverse is true since there is more time in which oil film absorption may take place and the burnt gas temperatures are lower. The high C\textsubscript{3}H\textsubscript{8} emissions at the low-load and high-load conditions for 1000 r/min (Figure 4.22) can be explained with reference to Figures 4.13 & 4.14. At these operating points, the equivalence ratio is relatively lean (approximately 0.6) but the hydrogen energy fraction is low and it is therefore most likely that partial bulk quenching is occurring.
Figure 4.23 indicates the improvement in C$_3$H$_8$ emissions made with hydrogen/petrol operation. With reference to Figure 4.14 the largest improvements occur at low-load which corresponds to the point of maximum hydrogen supplementation. Above half-load, hydrogen/petrol operation results in higher C$_3$H$_8$ emissions relative to petrol-only running and this is attributed to a decrease in hydrogen supplementation in conjunction with relatively lean equivalence ratios.

Figure 4.24 presents the hydrocarbon emissions, as a percentage of the petrol injected, for the engine operating point of 2000 r/min and 53 Nm (see Figures 4.1 - 4.3) and provides a possible clue to the further understanding of hydrogen/petrol combustion. The lower line on the graph represents the theoretical lean limit (see Section 2.1 and Figure 2.1) and remains approximately equi-distant from the experimental limit over the hydrogen energy fraction range. However, the percentage of unburnt petrol at the experimental limit increases from approximately 3% at $\beta = 0$ to approximately 18% at $\beta = 30\%$, thus indicatig that petrol combustion starts to 'quench' as the petrol-only lean limit equivalence ratio is approached despite hydrogen addition. This lends support to the possibility that the petrol and hydrogen combustion processes are chemically independent but still related by temperature and pressure. If this is the case, it is probable that two flame fronts cross the combustion chamber and detection of this phenomenon should be possible by the use of schlieren photographic techniques. The benefits of hydrogen supplementation are therefore limited by the separate combustion efficiency of the petrol and results in an optimum hydrogen energy fraction of approximately 15% at this engine operating point. At much higher values of $\beta$, the overall combustion efficiency would probably continue to increase but even less of the injected petrol would burn and, in any case, severe back-fire was experienced on this engine at values of hydrogen energy fraction in excess of 35%.

5.2 SIMULATED VEHICLE PERFORMANCE

5.2.1 Simulated Steady-State Fuel Consumption

Figure 4.25 presents simulated steady-state vehicle fuel consumption data, as predicted from the steady-state test-bed performance data, for a Ford 'Transit' Crew Bus fitted with the test-bed engine and hydrogen/petrol control system. The steady-state engine load lines for all four gear ratios were calculated by the following equations using constants obtained from the vehicle specification given in Appendix L and assuming a straight, level road with no wind and $\rho_a = 1.2$ kgm$^{-3}$:
Where,

\[ s = \frac{2\pi N_{r}}{60n_{a} n_{g}} \]  

(4.3)

The load lines were then superimposed on the steady-state thermal efficiency maps for petrol-only and hydrogen/petrol operation to enable the calculation of the energy-equivalent petrol consumption rate in litres per 100 km. Assuming an energy density for liquid petrol of 32.3 MJ/litre, the calculation was performed as follows:

\[
\text{fuel consumption (l/100km)} = \frac{(\text{b.t.})N_{a} \pi}{(3 \times 32.3) \eta_{t} s}
\]

(4.4)

These calculations do not take into account any energy losses in the driveline between the engine and tyres but this does not detract from the significant fuel savings obtained across the vehicle speed range with hydrogen/petrol operation as observed in Figure 4.25.

5.2.2 Simulated Transient Fuel Consumption

Wilkinson\textsuperscript{[131]} has developed a software simulation package for predicting the performance of a vehicle during driving cycles. The package uses steady-state performance maps for the vehicle's engine, in conjunction with the vehicle characteristics, to perform a quasi-steady-state simulation of the required driving cycle. Some simplifications are made concerning clutch take-up and gear changing but no attempt is made to anticipate the engine's true transient response to driver-demand. Should the engine under consideration operate with a 'perfect' transient compensation strategy, the quasi-steady-state analysis would be completely valid (except for clutch take-up and gear changes) and consequently any significant differences between predicted and measured performance are likely to be attributable to the transient compensation strategy. For example, when the throttle is opened quickly, the compensation strategy must increase the fuelling to keep pace with the increasing air-flow. Should the strategy not supply enough fuel, the simulation will overestimate the fuel used and conversely, should the strategy supply too much fuel, the simulation will underestimate the fuel used.
Table 5.1 describes the simulated energy-equivalent petrol consumption for the ECE-15.04 driving cycle and predicts a 10% reduction in energy consumption for hydrogen/petrol operation as compared with petrol-only operation.

<table>
<thead>
<tr>
<th>Mode of Operation</th>
<th>Average Fuel Consumption (litres / 100 km)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pet-Only</td>
<td>17.19</td>
</tr>
<tr>
<td>H2/Pet</td>
<td>15.43</td>
</tr>
</tbody>
</table>

Table 5.1 Simulated ECE-15.04 Fuel Consumption

5.2.3 Simulated Transient Emissions Performance

The transient simulation package was also used, in conjunction with exhaust mass-flow-rate maps of NO$_X$ and C$_3$H$_8$, to predict the performance of the ECE-15.04 emissions test. Table 5.2 describes the results of this simulation which indicate an 11% increase in NO$_X$ and a 40% decrease in C$_3$H$_8$ for hydrogen/petrol operation as compared with petrol-only operation. These results correspond well with the low-load, low speed areas of the NO$_X$ reduction and C$_3$H$_8$ reduction maps described in Figures 4.21 & 4.23 respectively.

<table>
<thead>
<tr>
<th>Mode of Operation</th>
<th>NO$_X$ Emissions (g / test)</th>
<th>C3H8 Emissions (g / test)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pet-Only</td>
<td>11.36</td>
<td>12.59</td>
</tr>
<tr>
<td>H2/Pet</td>
<td>12.60</td>
<td>7.61</td>
</tr>
</tbody>
</table>

Table 5.2 Simulated ECE-15.04 Exhaust Emissions

5.3 TRANSIENT VEHICLE PERFORMANCE

After completion of the test-bed calibration phase of the work, the control system was fitted to a Ford 'Transit' Crew Bus having specifications as described in Appendix L and shown in Photograph 1. The engine, shown in Photograph 2, was a 2-litre OHIC fuel injected engine capable of being controlled by a Ford EEC-IV engine management system. Operation could be
quickly switched (5 minutes) from one control system to the other by changing a few electrical and mechanical connections. Hydrogen was supplied from the hydride pack which is shown in Photograph 3 along with the two control system racks.

The crew bus was tested on a 'full-control' regenerative chassis dynamometer and measurements were made of 'total' exhaust emissions and fuel consumption during ECE 15.04 drive cycle tests. This work was carried out by Lister[132] as a 'final year' project and Tables 5.3 & 5.4 were taken from this study. Table 5.3 compares the exhaust emission performances for three controller configurations. The EEC-IV configuration used a commercial petrol-only fuel injection system while the other configurations used the control system described in this thesis.

<table>
<thead>
<tr>
<th>Mode of Operation</th>
<th>r.m.s. Speed Error (km/h)</th>
<th>CO</th>
<th>CO2</th>
<th>NOx</th>
<th>C3H8</th>
<th>NOx + C3H8</th>
</tr>
</thead>
<tbody>
<tr>
<td>EEC-IV Pet-Only</td>
<td>1.89</td>
<td>25.8</td>
<td>846.1</td>
<td>8.15</td>
<td>3.50</td>
<td>11.7</td>
</tr>
<tr>
<td>New Pet-Only</td>
<td>2.57</td>
<td>103.8</td>
<td>682.6</td>
<td>12.9</td>
<td>24.4</td>
<td>37.3</td>
</tr>
<tr>
<td>New H2/Pet</td>
<td>2.54</td>
<td>70.5</td>
<td>820.1</td>
<td>15.7</td>
<td>17.4</td>
<td>33.1</td>
</tr>
<tr>
<td>Simulated Pet-Only</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>11.4</td>
<td>12.6</td>
<td>24.0</td>
</tr>
<tr>
<td>Simulated H2/Pet</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>12.6</td>
<td>7.61</td>
<td>20.2</td>
</tr>
<tr>
<td>ECE 15.04 Limits</td>
<td>-</td>
<td>93.0</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>25.0</td>
</tr>
</tbody>
</table>

Table 5.3 ECE-15.04 Emissions Results

Table 5.4 compares the fuel consumption performance for the same configurations. The r.m.s. error included in the data represents the error between the 'driver's aid' target value for the cycle and the actual dynamometer roller speed. The corresponding simulated transient data from Section 5.2 is included in these tables for completeness.
Table 5.4 ECE-15.04 Fuel Consumption

<table>
<thead>
<tr>
<th>Mode of Operation</th>
<th>r.m.s. Speed Error (km/h)</th>
<th>Average Fuel Consumption (litres / 100 km)</th>
</tr>
</thead>
<tbody>
<tr>
<td>EEC-IV Pet-Only</td>
<td>1.92</td>
<td>12.0</td>
</tr>
<tr>
<td>New Pet-Only</td>
<td>2.32</td>
<td>14.9</td>
</tr>
<tr>
<td>New H2/Pet</td>
<td>2.14</td>
<td>14.4</td>
</tr>
<tr>
<td>Simulated Pet-Only</td>
<td>-</td>
<td>17.2</td>
</tr>
<tr>
<td>Simulated H2/Pet</td>
<td>-</td>
<td>15.4</td>
</tr>
</tbody>
</table>

5.3.1 Driveability

Although no specific driveability assessment tests were carried out on the chassis dynamometer, the values of r.m.s. speed error give some indication as to the difficulty encountered when attempting to follow the driver's aid. The commercial fuel injection system consistently gave a lower error as compared with that for the hydrogen/petrol controller and this is attributable to the use, by the EEC-IV controller, of a more sophisticated and better calibrated transient compensation strategy.

Driveability was also affected by the method of operation of the stepper-motor control module and the relatively slow response of the motor itself. The control module was only capable of responding to the latest throttle position command provided that the previously commanded position had been achieved. The motor therefore had to ramp up from and, back down to, zero speed before the module could be made aware of the next required position and in the interim period, this change in position could become quite large or even be in the reverse direction. The resulting motion of the throttle was very 'jerky' and therefore contributed to the transient equivalence ratio problem as well as producing undesirable torque fluctuations.

The use of the 'proportional' driver-demand map, described in Figure 4.9, also contributed to the driveability problems in that the low torque/demand gain gave a sluggish 'feel' to the accelerator pedal. This map was later changed in favour of one with a torque/demand gain similar to that of the demand map described in Figures 4.8 & 4.10. This new map made a substantial difference to the 'feel' of the system and certainly improved the sensitivity of the
5.3.2 Exhaust Emissions

The CO performance of the hydrogen/petrol controller was considerably worse than that of the commercial system and exceeded the legislative limit in the case of petrol-only operation. The legislative limit for combined NO\(_x\) and C\(_2\)H\(_8\) was exceeded for both hydrogen/petrol and petrol-only operation with the new controller. It should be noted, however, that the ECE 15.04 emissions test is carried out with a 'cold soaked' vehicle and it is therefore hardly surprising that the hydrogen/petrol controller, without any form of cold-start transient strategy, yielded poor results. What can be said for the new controller is that both CO and C\(_3\)H\(_8\) emissions were lower for hydrogen/petrol operation while NO\(_x\) emissions were lower for petrol-only operation.

The simulated transient emissions data underestimates the measured data by approximately 40% and this is attributed to the lack of a cold-start transient strategy for the hydrogen/petrol control system.

5.3.3 Fuel Consumption

The fuel consumption figures in Table 5.4 indicate that, with the new controller, a slight improvement over petrol-only running was obtained when hydrogen/petrol operation was implemented. However, the performance of the new controller was nevertheless not as good as that obtained with the use of the EEC-IV system.

The poor performance of the new system, as compared with the commercial system, may again be attributed to the differences in transient strategies but there was also some concern over the measurement technique for the EEC-IV system fuel consumption. The hydrogen/petrol system, in development mode, provided values of integrated fuel mass which were based on accurate injector calibrations. For hydrogen/petrol operation, the integrated hydrogen mass was converted to an energy equivalent mass of petrol. Fuel flow measurement for the commercial fuel injection system was performed by inserting a positive displacement flow meter in each of the supply and return fuel lines. Modern fuel injection systems use a high (approximately 150 litres per hour) bypass flow-rate as a means of preventing fuel vapourization and this posed a problem as far as measurement accuracy was concerned in that the nett fuel used represented the (small) difference of two large values. It was calculated that an error of 1% in the calibration of one transducer would result in a measurement error of 10% for the nett fuel used and, with the transducer calibrations quoted to be within ±1%, the accuracy of the EEC-IV fuel consumption figures quoted may be assumed to be within ±20%. Assuming the 'worst case' (favouring a
higher true flow-rate) results in an average fuel consumption figure of 14.4 litres per 100 km for the EEC-IV controller; this comparing more favourably with the results for the hydrogen/petrol system.

From Table 5.4, it is evident that the transient simulation package has slightly overestimated fuel consumption which implies that the transient strategy for the hydrogen/petrol control system was calibrated too 'lean'.

5.4 SYSTEM RELIABILITY

The hydrogen/petrol control system suffered from some reliability problems associated with the crankshaft encoder and the stepper motor throttle actuator.

Engine vibration at speeds in excess of 4000 r/min frequently resulted in broken wires in the crankshaft encoder but this may be overcome in the future by replacing the offending 'solid' conductors with the more flexible 'multi-strand' type. Alternatively, the timing signals could be taken from magnetic sensors mounted at the flywheel. The improved reliability of the latter technique would probably outweigh the resulting reduction in resolution.

The stepper motor was a relatively heavy unit and its position on the inlet system plenum chamber subjected it to severe vibration which resulted in a number of broken wires and support fixings. The latest support bracket, shown in Photograph 2, has been trouble-free but the development of throttle-bodies with integral servo-actuators will provide a solution which is superior in most respects relative to stepper motors.

The main computer rack also suffered from poor reliability in that the 'bus' connections between the card edge connectors and the system backplane were sometimes unreliable. Frequent cleaning or 'exercising' of the connectors helped to control the problem but this treatment would, in time, wear away the gold flashing on the connectors thus making matters worse.
6.1 PHILOSOPHY OF HYDROGEN/PETROL OPERATION

In Chapter 1 the concept of hydrogen supplementation was proposed as a means of conserving petroleum-based fuels in the short term. The addition of small quantities of hydrogen to the combustion process increases the burn-rate of the mixture while enabling more efficient lower equivalence ratio operation. The backfire problems associated with hydrogen-only operation are avoided by limiting the level of supplementation and this in turn implies a requirement for a relatively compact and lightweight hydrogen storage unit.

Hydrogen supplementation is applicable to a wide variety of fuels and it was noted that several researchers have contributed to the development of fuelling systems based on 'reformed' methanol. In these systems, the methanol, which is the only fuel stored in the vehicle, is heated by waste exhaust gases and catalytically split into hydrogen and carbon monoxide; both of which are combustable fuels. The catalytic converter acts as a chemical turbocharger in that the endothermic reaction absorbs exhaust energy in order to raise the effective calorific value of the fuel thus providing an immediate thermodynamic benefit. The hydrogen made available by the converter is fed to the engine where it is used to improve combustion. In addition to the two gaseous fuels, the presence of water in the methanol results in the formation of carbon dioxide and the effects of this, and the carbon monoxide on the engine's fuel consumption and emissions performance, are still being investigated. The major disadvantage of using methanol as the main fuel is that it has an energy density which is half that of petrol implying a doubling in storage tank volume and an increase in mass of approximately 30 kg compared with that for a 35 litre petrol tank.

The decision was taken to use petrol as the main fuel instead of methanol as this implied fewer modifications to the engine and avoided the additional complexities of controlling the converter. As a topic for future research, the hydrogen/petrol controller, as described in this thesis, could be adapted to control a reformed methanol engine which might yield superior performance compared to petrol-only or methanol-only operation. However, the focus of this work was the development of a suitable engine management system for the optimal control of hydrogen supplementation.

6.2 CONTROL REQUIREMENTS

6.2.1 Control For Maximum Thermal Efficiency

In Chapter 2 a review of the literature concluded that optimization of hydrogen energy fraction ($\beta$), equivalence ratio ($\phi$) and ignition timing was required in order to achieve maximum thermal efficiency over the engine's load/speed range. Typically, thermal efficiency increases as $\phi$ decreases from unity until some peak value just short of the lean-misfire limit. Increasing $\beta$
results in an extension of the lean-misfire limit thus allowing leaner operation and raising the value of peak thermal efficiency. In theory, peak thermal efficiency should continue to rise until $\beta$ reaches 100% but in practice it has been found that a relatively low optimum value of $\beta$ exists; the reason for this being attributed to the quenching of the petrol oxidation process as the equivalent petrol-only lean-misfire equivalence ratio is reached.

6.2.2 Control For Minimum Exhaust Emissions
It was noted from the literature that there are conflicts between the requirements for maximum thermal efficiency and minimum NO$_x$ and C$_3$H$_8$ exhaust emissions and this, therefore, required the development of a suitable optimization procedure for calibrating the control system. It was found that C$_3$H$_8$ emissions could be controlled by equivalence ratio but that typically the conditions for minimum C$_3$H$_8$ corresponded to near peak emission of NO$_x$. A number of researchers demonstrated that retardation of ignition timing, from MBT, resulted in significant reductions in NO$_x$ with only a small penalty in terms of thermal efficiency; this being due to the sensitivity of the dissociation reactions to peak temperatures and pressures.

6.2.3 Drive-By-Wire Requirements
At a given speed/torque operating point, the throttle angle must be altered to maintain the set point while equivalence ratio is changed. The requirement for equivalence ratio optimization at any chosen engine speed/torque operating point therefore necessitated the implementation of driver-independent throttle control.

In conventional spark-ignition engine control, where there is a mechanical linkage between the accelerator pedal and throttle, the engine's performance is a direct function of the driver-demand. In the case of driver independent throttle control however, the driver-demand input to the control system must be interpreted, via a pre-defined demand map, as a torque demand which must be met at the current engine speed by a suitable set of engine controls (ignition timing, throttle angle, $\phi$ and $\beta$).

The definition of the demand map has a direct bearing on the 'feel' of the control system in that the torque/position sensitivity of the accelerator pedal is a 'learned' characteristic of the system. Another characteristic of the demand map is that it defines the rate at which back-up torque is generated when the engine load is increased at constant driver-demand. The extent to which the inherent demand maps of today's vehicles have 'evolved' or been consciously optimized is not clear but it is the author's opinion that the commercial application of this new technology will necessitate a thorough understanding of the driver's perception of the optimum demand map if poor driveability is to be avoided.
In the past, the use of drive-by-wire throttle control has been limited to cruise-control and anti-wheel-spin applications and, more recently, some automotive manufacturers have been developing dynamic drive-by-wire systems in an effort to control low frequency power-train vibration. However, the author's contribution to the definition and interpretation of steady-state driver-demand maps is new and work at Loughborough University is continuing with the further development of practical drive-by-wire systems. An interesting possibility for the future is that drivers may be given the option of selecting any one of several demand maps giving a choice of 'performance feel'.

The reliability of any commercial drive-by-wire system will probably be the over-riding consideration affecting its acceptance. The further development of self-diagnostics, error correction and multiple-redundancy will be required to ensure fail-safe operation but the system cost must not increase significantly if the technology is to be commercially viable.

6.2.4 Sequentially Timed Fuel Injection

Some researchers have demonstrated the use of timed-sequential port fuel injection as a means of generating beneficial equivalence ratio stratification but, although this function was built into the hydrogen/petrol controller, the increased magnitude of the optimization problem, implied by the addition of another variable, resulted in the decision not to add further complication to the study. It should also be noted that consistent cylinder charge motion is a prerequisite for reliable equivalence ratio stratification and this may imply design changes to the engine's inlet system.

Bradley et al\textsuperscript{[126]} have shown that the further development of petrol-only, homogenous, lean-burn operation is limited by the flame straining effects of high turbulence which occur during the early stages of combustion. It has been shown that both the addition of hydrogen to the combustion process and the use of high energy, long duration ignition systems contribute to the extension of the usable lean-limit equivalence ratio but the development of the axially-stratified-charge engine by Quader\textsuperscript{[125]} and Toyota\textsuperscript{[88]} has pointed the way forward.

May et al\textsuperscript{[59]} have demonstrated the use of sequentially timed hydrogen injection as a means of controlling backfire at high hydrogen energy fractions. Hydrogen is only injected during the inlet-valve-open period thus eliminating the presence of a large volume of pre-mixed charge in the manifold. Another technique might be to inject a small quantity of hydrogen to the sparkplug at the time of ignition as this could provide the level of stratification required to maintain stable combustion at very lean equivalence ratios. Any further work on the hydrogen/petrol control system would therefore benefit from the implementation of sequentially timed injection of both petrol and hydrogen.
6.3 CONTROLLER IMPLEMENTATION

Having established the necessity for control of the engine's ignition timing, throttle position, equivalence ratio and hydrogen energy fraction, the literature relating to engine management systems was studied to provide suggestions for a suitable controller design and ideas for control strategies. A commercial microprocessor-based controller was chosen as the basis for the engine management system because of the flexibility which it offered in terms of strategy development, while the interface between the controller and the engine was implemented in purpose-built electronic hardware. The detail design of the control system was presented in Chapter 3 but the opportunity is taken here to discuss some of the important issues.

6.3.1 The Look-Up-Table Strategy
The closed-loop control of a particular measurable variable is the only means of achieving reliably accurate performance but the adoption of closed-loop control strategies for the optimization of the hydrogen/petrol engine's control variables was discounted on the basis that no suitable sensor(s) existed. Consequently, an open-loop control strategy was implemented which was based on the use of a look-up-table containing a matrix of control variable values. These values corresponded to the superimposition of the driver demand map upon the optimized steady-state torque/speed maps of the control variables. Engine speed and driver-demand were used as inputs to the look-up-table which then returned a set of control variable values corresponding to the desired torque output.

The disadvantage of using an open-loop look-up-table control strategy is that it cannot adapt to changes in system characteristics such as fuel quality, engine wear and component malfunction. Much research effort is being directed at adaptive control strategies which will, once given an initial calibration, perform optimally during the life of the engine and provide indication of system degradation.

6.3.2 Transient Compensation
The main emphasis of this thesis was concerned with the steady-state performance of the control system but it was considered desirable that the system should be installed in a vehicle and evaluated over typical driving cycles. It was therefore necessary to develop a suitable transient strategy to compensate for fuelling 'lags' during rapid throttle angle excursions. During these excursions, the air mass-flow-rate responds instantaneously but the corresponding measurement system may introduce a 'lag' and, when the fuelling system does respond, the effects of fuel reservoirs in the inlet port will contribute to an equivalence ratio excursion. A suitable transient
compensation strategy will detect the start of a throttle perturbation and suitably compensate for system lags by momentarily adjusting the fuelling rate. In the case of the hydrogen/petrol control system, the transient compensation strategy was implemented by factoring the measured air mass-flow-rate by the percentage change in throttle angle.

No attempt was made to develop a 'cold-start' transient strategy because of the considerable calibration task implied. Instead, a simple fixed-duration, fixed enrichment strategy was provided by operation of a switch on the interface electronics rack. This feature proved adequate for starting and driving but resulted in high exhaust emissions during 'cold-soaked' transient testing.

6.3.3 Dedicated And Development Systems
Two versions of the control software were written. The 'dedicated' version included only those features necessary for controlling the engine whereas the 'development' system additionally included certain features such as the acquisition and display of performance parameters and the facility for altering the control variables independently of the look-up-table. The development system proved to be invaluable during the calibration procedure in that the effect of altering one control variable could be observed without affecting the settings of the other variables.

It is a highly desirable feature of modern engine management systems that on-line development tools be made available to alter in-service calibrations and strategies. Such tools can also provide valuable diagnostic information during system development.

6.3.4 Software Development
The software development represented a major part of the work and the advantages of approaching such a complex task by thoroughly testing each new section of code, before progressing further, cannot be stressed too strongly. The debugging of interrupt-driven software can be very difficult and it is highly recommended that such work be carried out with the aid of an In-Circuit-Emulator (ICE) facility.

The future development of the hydrogen/petrol control system would benefit from the use of a more modern microcontroller, such as the Intel 8096 device, which provides a higher overall performance in addition to special hardware features suitable for engine control. Efficient high level languages and pre-written multi-tasking executives are available for these new single-chip controllers thus greatly simplifying the software development task.

The recent development of transputers promises to bring the benefits of true parallel processing to the engine control problem. The high computational performance offered by these devices
promises to make possible the real-time calculation of i.m.e.p. and burn-rate which, with the development of suitable, low-cost transducers, will facilitate the closed-loop control of these combustion parameters. The use of these combustion performance targets overcomes the problems of calibration drift due to engine wear, variable fuel quality and component failure.

6.3.5 Hardware Development

The development of reliable interface electronics is vital to overall system robustness and the practical experience gained during the course of this work suggests that radiated interference, 'ground-loops' and vibration are the main areas of concern.

Radiated interference from the engine's ignition system is a significant problem but commercial engine management systems have, in addition, to cope with radar, radio and power-line interference. The only reliable means of protecting the circuitry from radiated interference is to enclose it in a 'Faraday cage' and to 'decouple' all analogue and digital inputs as they enter the enclosure.

Ground-loops are the result of slight potential differences between multiple system grounds and may be avoided by grounding the system at one point only. The presence of ground-loops results in analogue signals drifting without apparent reason and consequently, as in the case of this work, can result in many hours of wasted calibration studies.

Vibration was a major cause of system failure during the course of this work and future designs would benefit from careful packaging and strain-relief of connections. In particular, the crankshaft encoder and throttle actuator suffered from fatigued electrical connections and these system components require careful consideration in the future.

The use of a commercial crankshaft encoder was an advantage to the development work in that it provided the required resolution and performed the necessary signal conditioning. However, a re-design of the control system would benefit from the simplicity and cost saving afforded by the use of proximity sensors to detect flywheel teeth and a TDC position marker. The same crank-angle resolution could be achieved by applying a phase-locked-loop to the input signal.

A stepper motor throttle actuator was used for the control system on the basis of cost but in retrospect the finite throttle angle resolution, 'jerky' action and relatively slow response would indicate the future adoption of a d.c. servo-motor actuator. Rugged throttle-body servo-motor actuators are now being developed which can provide full-motion response times of 200 ms.
6.4 SYSTEM CALIBRATION

The calibration of the hydrogen/petrol controller's look-up-table, described in Chapter 4, consisted of two stages. The first stage involved the specification of the optimization target while the second involved the optimization of the engine controls in order to achieve the specified target. Having determined optimum control variable maps for the entire operating range of the engine, a specially written software package was used to superimpose the demand map upon the control variable maps and thereby create the matrix of data for the look-up-table.

6.4.1 3/4-Load To Full-Load
At operating conditions above 3/4 load, the emphasis was on a smooth progression from maximum thermal efficiency to maximum torque and, consequently, petrol-only operation was indicated with load control being effected by means of equivalence ratio adjustment. The addition of hydrogen would have limited the maximum torque by reducing the volumetric efficiency. MBT ignition timing was chosen to give maximum torque. Direct cylinder injection of hydrogen would have overcome the volumetric efficiency problems but would have been expensive. In addition, the importance of fuel economy is of secondary importance at high loads and conservation of the hydrogen supply is a major concern.

6.4.2 Zero-Load
At the zero-load condition the only reliably measurable performance parameter was the fuelling rate and, because of the relative unimportance of this load condition to the driving cycle transient performance, petrol-only operation was adopted and the fuelling rate minimized by optimizing throttle angle and ignition timing.

6.4.3 Minimum-Load
The minimum-load or over-run condition is important to any driving cycle since it is experienced whenever the accelerator pedal is released. Unfortunately, the lack of a 'motoring' engine dynamometer facility precluded the measurement of data in this operating regime. Petrol-only operation at φ = 0.9 was arbitrarily chosen and ignition timing was set to the corresponding zero-load values for the same speeds. Throttle angle was set to minimum. In order to keep the cylinders hot some fuelling was maintained otherwise there would have been a number of poor engine cycles once fuelling recommenced.

6.4.4 Idle
Inspite of the importance of the idle condition to vehicle operation during typical driving cycles, the implementation of petrol-only operation was chosen on the basis of hydrogen supply conservation. The equivalence ratio was set to 0.9 to give smooth operation while throttle angle
and ignition timing were optimized to yield the minimum fuelling rate while maintaining idle speed.

6.4.5 Cranking
No testing was performed below idle speed because of excessive engine vibration and dynamometer limitations hence throttle angle was made proportional to driver-demand and petrol-only operation was implemented with ignition timing and φ varying linearly from TDC and unity respectively to the appropriate values at idle.

6.4.6 Part-Load
Considering the importance of part-load operation to system performance during driving cycles, this operating regime was optimized for maximum thermal efficiency with hydrogen/petrol operation. Contour maps were produced for selected part-load engine operating points to describe the thermal efficiency, NOx and C3H8 performance with respect to φ and β. The φ and β coordinates corresponding to optimum operation were then read off the maps along with the corresponding values of MBT ignition timing and throttle angle which were also displayed in the form of contour maps on the φ/β plane. Due to the absence of any firm criteria for selecting maximum allowable emissions limits, the philosophy for optimization consisted of avoiding any steeply increasing emission trends but otherwise selecting the point of maximum thermal efficiency.

It is proposed that a future programme of work should be instituted to study the influence of individual engine operating points on the fuel economy and emissions performance of driving cycles. Such a programme of study could use the drive-cycle simulation program developed by Wilkinson[131].

A complete study of the relative sensitivities of thermal efficiency and NOx emissions to MBT-retarded ignition timing requires the development of three-dimensional contour display techniques. Such techniques would facilitate the simultaneous optimization of φ, β and ignition timing.

Any further calibration work for the hydrogen/petrol control system would greatly benefit from the development of an automated testing procedure which would allow the optimization of a larger number of engine operating conditions. This would increase the overall performance of the system and facilitate more accurate prediction of driving cycle performance.
6.5 SYSTEM PERFORMANCE

Chapter 5 describes the steady-state test-bed performance as well as the transient performance of the system when installed in a vehicle. Additionally, simulations were employed to predict the steady-state vehicle fuel consumption as well as the transient fuel consumption and emissions performance.

6.5.1 Fuel Consumption
The steady-state test-bed data indicated improvements in fuel economy of up to 20% for hydrogen/petrol operation compared with petrol-only running and the transient simulation program predicted that this would translate into a 10% reduction in fuel consumption over the ECE-15.04 driving cycle. In fact, the chassis dynamometer experiments only indicated a 3% gain in fuel economy. Comparative tests were made with a commercial engine management system for the same engine but there was some doubt as to the accuracy of fuel measurement.

In commercial terms, a 3% gain in fuel economy would not warrant the additional cost of the hydrogen storage and metering system. However, the lack of greater improvement may be due to the coarseness of the torque/speed grid used for the optimization process. With a finer grid and more attention paid to the areas of the engine map corresponding to the driving cycle, it should be possible to improve on this performance.

6.5.2 Exhaust Emissions
The steady state test-bed data indicated improvements in NO\textsubscript{X} and C\textsubscript{3}H\textsubscript{8} emissions of up to 40% and 60% respectively for the change to hydrogen supplementation although in some areas of the engine map corresponding increases in specific emissions were in excess of 40% and 20% for NO\textsubscript{X} and C\textsubscript{3}H\textsubscript{8} respectively.

The transient simulation program predicted that both petrol-only operation and hydrogen/petrol operation would meet the ECE-15.04 emissions regulations for combined NO\textsubscript{X} and C\textsubscript{3}H\textsubscript{8} emissions and that hydrogen/petrol operation would result in a 16% improvement over petrol-only operation. In practice, the reduction in combined NO\textsubscript{X} and C\textsubscript{3}H\textsubscript{8} emission, for hydrogen/petrol operation relative to petrol-only operation, was only 11%. In addition, both methods of operation failed to meet the ECE-15.04 standards with hydrogen/petrol operation producing 32% more emissions than the maximum allowed.

The ECE-15.04 emissions driving cycle is carried on 'cold-soaked' vehicles and so the poor performance of both modes of operation with the hydrogen/petrol control system was attributed to the lack of a suitable cold-start transient control strategy. It is therefore clear that any future
work on the hydrogen/petrol control system will have to include the development and calibration of an adequate cold-start strategy.

### 6.5.3 Driveability

The initial subjective driveability assessments concluded that the response of the control system was poor and this prompted a re-design of the demand map. The initial 'proportional' map was replaced by one imitating a typical spark-ignition engine; the main effect of which was to increase the torque/position sensitivity of the accelerator pedal. The adoption of this demand map made a significant improvement to the 'feel' of the control system.

The 'jerky' action of the stepper motor throttle actuator made a major contribution to the driveability problem and the only solution will be to replace this actuator with a d.c. servo-motor.

### 6.6 RECOMMENDATIONS FOR FUTURE WORK

1) The work of Coward and Jones\textsuperscript{[46]}, Yu et al\textsuperscript{[61]}, Metghalchi and Keck\textsuperscript{[62]} and Sher and Hacohen\textsuperscript{[64-66]} should be extended, in the fashion indicated in Section 2.1, to provide a mathematical expression for the laminar flame speeds of hydrogen/petrol/air mixtures at elevated temperatures and pressures over the entire working ranges of $\phi$ and $\beta$. It will be necessary to support this proposed work with laminar flame speed measurements taken over the same ranges of $\phi$ and $\beta$ and schlieren photography should be used to test the possibility that two flames occur in hydrogen/petrol/air combustion.

2) Future studies should include ignition timing as an optimization variable in addition to equivalence ratio and hydrogen energy fraction. These studies would require the development of software for displaying three-dimensional contour maps. Such maps would facilitate the understanding of the ignition timing trade-off between thermal efficiency and NO\textsubscript{x} emissions.

3) A parametric simulation study should be carried out to determine appropriate maximum allowable levels of exhaust emissions in terms of the engine maps in order to ensure that the vehicle meets the emission legislation requirements.

4) Future calibration work will require the use of an engine dynamometer capable of motoring the test engine so that the over-run condition can be suitably optimized. This type of dynamometer will also facilitate calibration at the 'idle' condition.
5) The current transient strategy of the hydrogen/petrol control system requires a programme of calibration and further development to include cold-start compensation. This is perhaps the most important task facing the further development of the controller.

6) It is strongly recommended that an indepth study of the driver-demand map be carried out to thoroughly establish its effect on driveability and on the driver's perception of performance. The author suggests that future cars would benefit from drive-by-wire systems providing engine management, cruise control, anti-wheel-spin and the choice of 'performance feel'.

7) The engine controller described in Chapter 3 is very much a prototype system and future work on the hydrogen/petrol engine would benefit greatly if the system were to be repackaged in a more robust form. Such a system could utilize 'hall effect' 'pickups' at the flywheel instead of the crankshaft encoder and should use a d.c. servo-motor, instead of a stepper motor, for throttle actuation.
Chapter 7

CONCLUSIONS
Chapter 1 of this thesis concluded that the use of hydrogen to supplement petrol/air combustion in the spark-ignition engine was a useful means of obtaining good fuel economy while helping to conserve petroleum-based fuels. In Chapter 2 a study of the literature identified the requirement for an engine management system to optimally control the ignition timing, throttle angle, equivalence ratio and hydrogen energy fraction. Chapter 3 presented the design of and discussed the strategies for the open-loop, look-up-table based microprocessor control system while Chapter 4 described the optimization process by which the controller was calibrated for steady-state operation. The optimization process varied for different areas of the engine map but the strategy for part-load operation (the only operating regime to use hydrogen supplementation) consisted of maximizing thermal efficiency within the limiting constraints of NOx and C3H8 exhaust emissions. Having calibrated the control system, Chapter 5 presented and discussed the results of steady-state test-bed operation and transient operation of the engine when installed in a vehicle. Finally, Chapter 6 summarised and discussed the important aspects of the work and suggestions were made concerning the need for future research.

7.2 AUTHOR'S ORIGINAL CONTRIBUTIONS

In Section 2.1, the models of Yu et al[61] and Metghalchi & Keck[62] were combined to predict the laminar flame speed of hydrogen/petrol/air mixtures at elevated temperatures and pressures as a function of equivalence ratio ($\phi$) and hydrogen energy fraction ($\beta$). Even considering the limitations imposed by Yu et al, the combined model was shown, by the author, to be deficient in terms of lean equivalence ratio prediction. The author has also presented some evidence concerning the possibility that the hydrogen combustion process is independent of the petrol combustion process and has suggested that future work should therefore attempt to establish the occurrence of two flames. A new concept, that of steady-state drive-by-wire torque control, has been proposed in this thesis as it relates to the implementation of driver-independent throttle control which was utilized by the hydrogen/petrol engine management system. Furthermore, recognition of the importance of the demand map and the discussion of its implications and influence upon the driver's perception of the 'feel' and driveability of the vehicle also broke new ground. Although some of the inherent characteristics of demand maps for conventional spark-ignition engines were known, it was the author's detailed scrutiny which revealed other characteristics which had been previously overlooked.
Chapter 4 presented a novel graphical technique for the optimization of engine performance in terms of equivalence ratio and hydrogen energy fraction. The technique itself, together with all of the supporting analysis software, was developed by the author.

The design of the hydrogen/petrol engine management system and the development of the control software is representative of one facet of the author's original contribution although a large proportion of the detailed electronics design was performed by others.

7.3 RELEVANCE OF WORK

The significant steady-state fuel consumption reductions afforded by the adoption of hydrogen supplementation bear the promise of future commercial exploitation but further refinement of the hardware and transient strategies is required in order to realize the commercial viability of transient operation. The control of NOx emissions, by means of ignition timing retardation, is likely to be necessary in order that future legislative requirements regarding emissions might be met.

The adaptation of the engine management system described in this thesis to the control of a 'reformed methanol' powered vehicle is likely to show increased benefits over hydrogen/petrol operation. Such a system would combine the benefits of hydrogen supplementation with the 'chemical turbocharger' action of the methanol converter.

The development of the engine mapping, optimization and transient performance prediction techniques, as described in this thesis, has encouraged further investigations which are now underway at Loughborough University with a view to the establishment of a comprehensive vehicle performance simulation package.

The author's contributions to the development of the steady-state drive-by-wire concept are currently being exploited at Loughborough University and a second vehicle has been equipped with a drive-by-wire throttling system in order to study the influence of the demand-map upon the driver's perception of performance.

7.4 SUMMARY OF MAIN CONCLUSIONS

1) Hydrogen supplementation of petrol combustion has been shown to yield steady-state reductions in fuel consumption of up to 20%.
2) The use of MBT ignition timing at lean equivalence ratios resulted in high NO\textsubscript{x} emissions during hydrogen supplementation and this implies the need for a future study regarding the simultaneous optimization of equivalence ratio, hydrogen energy fraction and ignition timing.

3) Hydrogen supplementation afforded reductions in steady-state C\textsubscript{3}H\textsubscript{8} emissions of up to 60% but in some areas of the operating range there were increases of up to 20%.

4) Predictions concerning the transient fuel consumption and emissions performance over the ECE-15.04 driving cycle were over-optimistic with respect to the measured performance but accurately estimated the trends.

5) The measured ECE-15.04 exhaust emissions exceeded the required limits but this was attributed to the lack of a suitable cold-start transient compensation strategy.

6) A subjective driveability assessment concluded that the performance was poor and that this was due partly to a crudely calibrated transient compensation strategy and partly to the 'jerky' action of the stepper motor throttle actuator. It was consequently recommended that future work would benefit by the replacement of this unit with a d.c. servo-motor actuator.

7) A study of the concept of drive-by-wire engine torque control concluded that the detailed design of the demand map had a significant influence on driver-perceived 'feel' and driveability.

8) Consideration of the hydrogen/petrol/air combustion process, coupled with the observation of steady-state test-bed performance, suggested the possibility that the hydrogen and petrol oxidation processes were independent and could result in two flames.
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Appendix A

PAL Decoder Boolean Equations
Device Type = PAL16L6
Memory Type = 8K*8 (2764 EPROM)
QUAD BEGINNING ADDRESS IS 0000H

Pin No = 1, 2, 3, 4, 5, 6, 7, 8,
Signal = A13, A12, A11, A10, AF, AE, AD, AC,

Pin No = 9, 10, 11, 12, 13, 14, 15, 16,
Signal = AB, MBS, /BHEN, GND, B16, QEI, S2, /CS3,

Pin No = 17, 18, 19, 20, 21, 22, 23, 24
Signal = /CS2, /SWP8, /QE, /CS1, /CS0, TE, A0, VCC

CS0 = /AF * /AE * BHEN * B16 * MBS * QEI * TE
+ /AF * /AE * BHEN * B16 * MBS * QEI * TE /A0
+ /AF * /AE * AD * /BHEN * /B16 * MBS * QEI * TE
(= 0000H TO 1FFFH)

CS1 = /AF * /AE * BHEN * B16 * MBS * QEI * TE
+ /AF * /AE * /BHEN * B16 * MBS * QEI * TE * /A0
+ /AF * /AE * AD * /BHEN * /B16 * MBS * QEI * TE
(= 2000H TO 3FFFH)

CS2 = /AF * AE * BHEN * B16 * MBS * QEI * TE
+ /AF * AE * /BHEN * B16 * MBS * QEI * TE * /A0
+ /AF * AE * AD * /BHEN * /B16 * MBS * QEI * TE
(= 4000H TO 5FFFH)

CS3 = /AF * AE * BHEN * B16 * MBS * QEI * TE
+ /AF * AE * /BHEN * B16 * MBS * QEI * TE * /A0
+ /AF * AE * AD * /BHEN * /B16 * MBS * QEI * TE
(= 6000H TO 7FFFH)

SWP8 = /AF * /AD * /BHEN * /B16 * MBS * QEI * TE

QE = /AF * MBS * QEI * TE
QUAD-I

Device Type = PAL16L6
Memory Type = 4K*8 (2732 EPROM)

QUAD BEGINNING ADDRESS IS 8000H

Pin No =  1,  2,  3,  4,  5,  6,  7,  8,
Signal = A13, A12, A11, A10, AF, AE, AD, AC,

Pin No =  9,  10, 11, 12, 13, 14, 15, 16,
Signal = AB, MBS, /BHEN, GND, B16, QEI, S2, /CS3,

Pin No = 17, 18, 19, 20, 21, 22, 23, 24
Signal = /CS2, /SWP8, /QE, /CS1, /CS0, TE, A0, VCC

CS0 = AF * /AE * /AD * BHEN * B16 * MBS * QEI * TE
    + AF * /AE * /AD * /BHEN * B16 * MBS * QEI * TE * /A0
    + AF * /AE * /AD * /AC * /BHEN * /B16 * MBS * QEI * TE
(= 8000H TO 8FFFH)

CS1 = AF * /AE * /AD * BHEN * B16 * MBS * QEI * TE
    + AF * /AE * /AD * /BHEN * B16 * MBS * QEI * TE * /A0
    + AF * /AE * AD * /AC * /BHEN * /B16 * MBS * QEI * TE
(= 9000H TO 9FFFH)

CS2 = AF * /AE * AD * BHEN * B16 * MBS * QEI * TE
    + AF * /AE * AD * /BHEN * B16 * MBS * QEI * TE * /A0
    + AF * /AE * AD * /AC * /BHEN * /B16 * MBS * QEI * TE
(= A000H TO AFFFH)

SWP8 = AF * /AE * /AC * /BHEN * /B16 * /MBS * QEI * TE

QE = AF * /AE * MBS * QEI * TE
**QUAD-2**

Device Type = PAL16L6

Memory Type = 8K*8 (6264 RAM)

**QUAD BEGINNING ADDRESS IS 0000H**

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**CS0**

\[
/AF \cdot /AE \cdot BHEN \cdot B16 \cdot MBS \cdot QEI \cdot TE \\
+/AF \cdot /AE \cdot /BHEN \cdot B16 \cdot MBS \cdot QEI \cdot TE * /A0 \\
+/AF \cdot /AE \cdot /AD \cdot /BHEN \cdot /B16 \cdot MBS \cdot QEI \cdot TE \\
(= 0000H TO 1FFFH)
\]

**CS1**

\[
/AF \cdot /AE \cdot BHEN \cdot B16 \cdot MBS \cdot QEI \cdot TE \\
+/AF \cdot /AE \cdot /BHEN \cdot B16 \cdot MBS \cdot QEI \cdot TE * /A0 \\
+/AF \cdot /AE \cdot /AD \cdot /BHEN \cdot /B16 \cdot MBS \cdot QEI \cdot TE \\
(= 2000H TO 3FFFH)
\]

**CS2**

\[
/AF \cdot AE \cdot BHEN \cdot B16 \cdot MBS \cdot QEI \cdot TE \\
+/AF \cdot AE \cdot /BHEN \cdot B16 \cdot MBS \cdot QEI \cdot TE * /A0 \\
+/AF \cdot AE \cdot /AD \cdot /BHEN \cdot /B16 \cdot MBS \cdot QEI \cdot TE \\
(= 4000H TO 5FFFH)
\]

**CS3**

\[
/AF \cdot AE \cdot BHEN \cdot B16 \cdot MBS \cdot QEI \cdot TE \\
+/AF \cdot AE \cdot /BHEN \cdot B16 \cdot MBS \cdot QEI \cdot TE * /A0 \\
+/AF \cdot AE \cdot /AD \cdot /BHEN \cdot /B16 \cdot MBS \cdot QEI \cdot TE \\
(= 6000H TO 7FFFH)
\]

**SWP8**

\[
/AF \cdot /AD \cdot /BHEN \cdot /B16 \cdot /MBS \cdot QEI \cdot TE \\
\]

**QE**

\[
/AF \cdot MBS \cdot QEI \cdot TE \\
\]
QUAD-3

Device Type = PAL16L6
Memory Type = 8K*8 (2764 EPROM)

QUAD BEGINNING ADDRESS IS 8000H

Pin No =  1,  2,  3,  4,  5,  6,  7,  8,
Signal = A13,  A12,  A11,  A10,  AF,  AE,  AD,  AC,

Pin No =  9, 10, 11, 12, 13, 14, 15, 16,
Signal = AB,  MBS,  /BHEN,  GND,  B16,  QEI,  S2,  /CS3,

Pin No =  17, 18, 19, 20, 21, 22, 23, 24
Signal = /CS2,  /SWP8,  /QE,  /CS1,  /CS0,  TE,  A0,  VCC

CS0 = AF */AE * BHEN * B16 * MBS * QEI * TE
+ AF */AE */BHEN * B16 * MBS * QEI * TE */A0
+ AF */AE */AD */BHEN */B16 * MBS * QEI * TE
(= 8000H TO 9FFFH)

CS1 = AF */AE * BHEN * B16 * MBS * QEI * TE
+ AF */AE */BHEN * B16 * MBS * QEI * TE */A0
+ AF */AE */AD */BHEN */B16 * MBS * QEI * TE
(= A000H TO BFFFH)

CS3 = AF */AE * BHEN * B16 * MBS * QEI * TE
+ AF */AE */BHEN * B16 * MBS * QEI * TE */A0
+ AF */AE */AD */BHEN */B16 * MBS * QEI * TE
(= C000H TO DFFFH)

SWP8 = AF */AD */BHEN */B16 */MBS * QEI * TE

QE = AF * MBS * QEI * TE
Device Type = PAL10L8

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</table>

**IOADRS**

\[
\text{IOADRS} = /A7 \times A6 \times /A5 \times A4 \times \text{TEINP} \times \text{IOBS} \\
+ /A7 \times A6 \times A5 \times /A4 \times \text{TEINP} \times \text{IOBS}
\]

(50H)

**ERCS**

\[
\text{ERCS} = /A7 \times A6 \times /A5 \times A4 \times A3 \times /A2 \times /A1 \times /A0 \times \text{TEINP} \times \text{IOBS}
\]

(58H TO 5BH)

**PICCS**

\[
\text{PICCS} = /A7 \times A6 \times /A5 \times A4 \times A3 \times A2 \times A1 \times \text{TEINP} \times \text{IOBS}
\]

(5EH TO 5FH)

**PARALLELCS**

\[
\text{PARALLELCS} = /A7 \times A6 \times /A5 \times A4 \times /A3 \times A2 \times \text{TEINP} \times \text{IOBS}
\]

(54H TO 57H)

**SERIALCS**

\[
\text{SERIALCS} = /A7 \times A6 \times A5 \times /A4 \times A3 \times A2 \times /A1 \times \text{TEINP} \times \text{IOBS}
\]

(60H TO 67H)

**ISBXCS0**

\[
\text{ISBXCS0} = /A7 \times A6 \times A5 \times /A4 \times A3 \times \text{TEINP} \times \text{IOBS}
\]

(68H TO 6FH)

**ISBXCS1**

A - 6
Appendix B

I/O Address Mapping
<table>
<thead>
<tr>
<th>I/O Address</th>
<th>J/O Device</th>
<th>Output Function</th>
<th>Input Function</th>
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</thead>
<tbody>
<tr>
<td>0F9H (11111xx1B)</td>
<td>Am94/1541</td>
<td>-</td>
<td>Feedback High</td>
</tr>
<tr>
<td>0F8H (11111xx0B)</td>
<td>Stepper Motor</td>
<td>-</td>
<td>Feedback Low</td>
</tr>
<tr>
<td>0F1H (11110xx1B)</td>
<td>Controller</td>
<td>Command</td>
<td>Read ERCO Register</td>
</tr>
<tr>
<td>0F0H (11110xx0B)</td>
<td>J6 iSBC 80/24</td>
<td>Write Data</td>
<td>Read Data</td>
</tr>
<tr>
<td>0EFH (0EDH)</td>
<td>Serial Port 8251A</td>
<td>Control</td>
<td>Status</td>
</tr>
<tr>
<td>0EEH (0ECH)</td>
<td>U20 iSBC 80/24</td>
<td>Write Data</td>
<td>Read Data</td>
</tr>
<tr>
<td>0EBH</td>
<td>Parallel Port</td>
<td>Control</td>
<td>-</td>
</tr>
<tr>
<td>0EAH</td>
<td>8255A</td>
<td>Write Port C</td>
<td>Read Port C</td>
</tr>
<tr>
<td>0E9H</td>
<td>U18</td>
<td>Write Port B</td>
<td>Read Port B</td>
</tr>
<tr>
<td>0E8H</td>
<td>J1 iSBC 80/24</td>
<td>Write Port A</td>
<td>Read Port A</td>
</tr>
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<td>0E7H</td>
<td>Parallel Port</td>
<td>Control</td>
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</tr>
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<td>0E6H</td>
<td>8255A</td>
<td>Write Port C</td>
<td>Read Port C</td>
</tr>
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<td>0E5H</td>
<td>U15</td>
<td>Write Port B</td>
<td>Read Port B</td>
</tr>
<tr>
<td>0E4H</td>
<td>J1 iSBC 80/24</td>
<td>Write Port A</td>
<td>Read Port A</td>
</tr>
<tr>
<td>0DFH</td>
<td>Interval Timer</td>
<td>Control</td>
<td>-</td>
</tr>
<tr>
<td>0DEH</td>
<td>8253</td>
<td>Load Counter #2</td>
<td>Read Counter #2</td>
</tr>
<tr>
<td>0DDH</td>
<td>U19</td>
<td>Load Counter #1</td>
<td>Read Counter #1</td>
</tr>
<tr>
<td>0DCH</td>
<td>iSBC 80/24</td>
<td>Load Counter #0</td>
<td>Read Counter #0</td>
</tr>
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<td>0DBH (0D9H)</td>
<td>Interrupt Controller</td>
<td>OCW1, ICW2, ICW4</td>
<td>IMR</td>
</tr>
<tr>
<td>0DAH (0D8H)</td>
<td>8259A U17 iSBC 80/24</td>
<td>OCW2, OCW3, ICW1</td>
<td>IRR, ISR, etc</td>
</tr>
<tr>
<td>0D7H</td>
<td>iSBC 80/24</td>
<td>Initiate TRACE</td>
<td>-</td>
</tr>
<tr>
<td>0D6H</td>
<td>-</td>
<td>Flash LED 1ms</td>
<td>-</td>
</tr>
<tr>
<td>0D5H</td>
<td>-</td>
<td>Bus Overide</td>
<td>-</td>
</tr>
<tr>
<td>0D4H</td>
<td>-</td>
<td>Reset PWRFL Latch</td>
<td>PWRFL Status</td>
</tr>
<tr>
<td>0C1H (11000xx1B)</td>
<td>iSBX 311 A/D</td>
<td>-</td>
<td>L-Byte, Status</td>
</tr>
<tr>
<td>0C0H (11000xx0B)</td>
<td>Module</td>
<td>-</td>
<td>H-Byte, Reset Int.</td>
</tr>
<tr>
<td>0C0H (11000xxxB)</td>
<td>J5 iSBC 80/24</td>
<td>Channel Select</td>
<td>-</td>
</tr>
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B - 2
<table>
<thead>
<tr>
<th>I/O Address</th>
<th>I/O Device</th>
<th>Output Function</th>
<th>Input Function</th>
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<td>7FH</td>
<td>Am95/6120</td>
<td>Unit Code</td>
<td></td>
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<tr>
<td>7EH</td>
<td>Dual-Density</td>
<td>Track, H-Byte</td>
<td></td>
</tr>
<tr>
<td>7DH</td>
<td>Floppy Disk</td>
<td>Sector, L-Byte</td>
<td></td>
</tr>
<tr>
<td>7CH</td>
<td>Controller</td>
<td>Control</td>
<td>Status</td>
</tr>
<tr>
<td>61H (01100xx0B)</td>
<td>SBX Maths Module</td>
<td>Control</td>
<td>Status</td>
</tr>
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<td>60H (01100xx1B)</td>
<td>P4 Am96/5232</td>
<td>Write Data</td>
<td>Read Data</td>
</tr>
<tr>
<td>5FH</td>
<td>U34 Am96/5232</td>
<td>OCW1, ICW2, ICW4</td>
<td>IMR</td>
</tr>
<tr>
<td>5EH</td>
<td>8259A</td>
<td>OCW2, OCW3, ICW1</td>
<td>IRR, ISR, etc</td>
</tr>
<tr>
<td>5DH</td>
<td>Serial Port 8251A</td>
<td>Control</td>
<td>Status</td>
</tr>
<tr>
<td>5CH</td>
<td>U32 Am96/5232</td>
<td>Write Data</td>
<td>Read Data</td>
</tr>
<tr>
<td>5BH (5AH)</td>
<td>System Timer 9513</td>
<td>Control</td>
<td>Status</td>
</tr>
<tr>
<td>59H (58H)</td>
<td>U38 Am96/5232</td>
<td>Write Data</td>
<td>Read Data</td>
</tr>
<tr>
<td>57H</td>
<td>Parallel Port</td>
<td>Control</td>
<td></td>
</tr>
<tr>
<td>56H</td>
<td>8255A</td>
<td>Write Port C</td>
<td>Read Port C</td>
</tr>
<tr>
<td>55H</td>
<td>U33</td>
<td>Write Port B</td>
<td>Read Port B</td>
</tr>
<tr>
<td>54H</td>
<td>P3 Am96/5232</td>
<td>Write Port A</td>
<td>Read Port A</td>
</tr>
<tr>
<td>50H</td>
<td>Am96/5232</td>
<td>QUAD-EN, LED</td>
<td>OPT0, OPT1, MPST</td>
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Appendix C

Modifications To Computer Hardware
INTEL iSBC 80/24 SINGLE-BOARD COMPUTER

The iSBC 80/24 single-board computer is described fully in INTEL manual Order No. 142648-001. This manual describes the configuration in which the board is shipped from the factory, and in particular, pages 2.5 to 2.9 describe the default jumper connections. Following is a list of the jumper modifications carried out for this work.

Connect J1/p32 (CLK - crankshaft degree pulses) to CLKO (counter #0)
Remove - E129-E133, E35-E50
Connect - E132-E133, E35-E38

Connect J1/p30 (GATE - crankshaft timing pulse) to INTR 7.5 on processor and to GATEO (counter #0)
Remove - E34-E46, E34-E36, E120-E122, E93-E121
Connect - E45-E46, E34-E39, E121-E122, E120-E121, E36-E51

Connect - Output of counter #0 (ignition timing controller) to J1/p18 (IGN)
Remove - E100-E118, E31-E45
Connect - E31-E118

Set counter #1 as 'real time' counter (14.88 μs < period < 487.8 μs) for artificial encoder
Remove - E130-E134
Connect - E134-E128

Connect output of counter #1 (computer CLK source) to J1/p20 (computer CLK out)
Remove - E30-E44
Connect - E30-E103

Connect J1/p28 (MARK - cycle marker pulse) to IR0 on 8259A PIC
Remove - E33-E48
Connect - E33-E102

Connect RXRDY of 8251APCI (VDU serial port - OECH, OEDH) to INTR 5.5 on processor
Remove - NIL
Connect - E117-E119
Connect P1/p42 (INT 1 on Multibus) (RXRDY of 'off-board' 8251A PCI) (AUX serial port - 5CH, 5DH) to INTR 6.5 on processor
Remove - NIL
Connect - E107-E120

Set for 8K 'on-board' RAM
Remove - E148-E149
Connect - NIL

Disable 'on-board' RAM by removing 'max RAM address' jumper
Remove - E154-E155
Connect - NIL

Select EPROM type 2764
Remove - J8:1-4, J7:7
Connect - J7:2-6

Disable 'on-board' EPROM by grounding PROMENABLE
Remove - NIL
Connect - E54-E26

Connect J5/MINTR0 (EOC from iSBX 311 A/D module) to IR3 on 8259A PIC
Remove - NIL
Connect - E88-E99

Connect J6/MINTR0 ((IBE .OR. OBF) from Am94/1541 stepper motor controller module) to IR2 on 8259A PIC
Remove - NIL
Connect - E90-E100

Connect J6/MINTR1 (ERR from Am94/1541 stepper motor controller module) to IR1 on 8259A PIC
Remove - E101-E106
Connect - E91-E101

C - 3
In addition to the above jumpering, the following modifications were carried out to the iSBBC 80/24 board.

The iSBX bus connectors J5 and J6 were connected to an iSBX 311 A/D module and an Am94/1441 stepper motor controller module respectively.

The 'on-board' RAM capacity was increased from 4K to 8K by the installation of the iSBBC 301 optional RAM expansion module.

The dil header in socket J4 was replaced with one having p1-p14, p2-p13, p3-p4 and p5-p6 connections made.

Sockets U4, U5 and U6 were dedicated as outputs and fitted with 7437 TTL line drivers (quad 2 input NAND buffers - 48 mA sink current)

Socket U3 was dedicated as input and fitted with a dil header. It was found necessary to 'schmitt' three of the inputs to the dil header (CLK, GATE & MARK) and so a 40106 hex schmitt inverter device was inserted into the spare socket position U1. Two gates were used for each signal.

A small 'piggy-back' printed circuit board was fitted to the iSBBC 80/24 board to provide a hardware TRACE facility in conjunction with the system monitor program. This board generates a TRAP interrupt two machine cycles after a WRITE to port 0D7H is performed.

iSBX 311 ANALOGUE INPUT MULTIMODULE BOARD

The default configuration for this board is described in INTEL manual Order No. 142913-001. The following jumper modifications were carried out.

Isolate analogue signal 'RETURN' from 'board GND'

Remove - E10-E13, E11-E12
Connect  - E12-E13

Configure for single ended input
Remove - NIL
Connect  - E8-E9, E19-E20
Configure for unipolar (0-5V) operation
Remove - E15-E16
Connect - E14-E15, E16-E17

Am95/6120 DUAL-DENSITY DISK CONTROLLER BOARD

The default configuration for this board is described in Advanced Micro Devices manual Order No. 059901341-001. The following modifications were carried out.

Set 7CH as base address
SW2/1 = 0
SW2/2 = 1
SW2/3 = 1
SW2/4 = 1
SW2/5 = 1
SW2/6 = 1

Select 3 ms head step rate
Remove - jumper 15
Connect - jumper 17

The Am95/6120 does not monitor the 'DISK CHANGE' signal (P4/p12) from the drives, and so this signal was .OR.'ed into the 'WRITE PROTECT' signal (P4/p44) at the edge connector. This was done by inserting two diodes and cutting a land.

Am96/5232 PROGRAMMABLE RAM/EPROM AND I/O BOARD

The default configuration for this board is described in Advanced Micro Devices manual Order No. 059920039-001. The following jumper modifications were carried out.

Select 9-bit wide memory operation
Remove - 84-85
Connect - NIL
Select 20-bit memory addressing
Remove - 94-95
Connect - NIL

Select QUAD-0 on power-on or RESET
Remove - NIL
Connect - 12-13

Select 8-bit I/O addressing
Remove - 79-80
Connect - NIL

For XACK timing - 300ms < longest time < 400ms
Remove - NIL
connect - 89-87

Configure QUAD-0 for 2764 EPROM's - pin 26 to VCC & pin 23 to A11
Remove - NIL
Connect - 1-2, 9-7

Configure QUAD-1 for 2732 EPROM's - pin 26 to VCC & pin 23 to A11
Remove - NIL
Connect - 33-34, 43-41

Configure QUAD-2 for 6264 RAM's - pin 26 to VCC & pin 23 to A11
Remove - NIL
Connect - 76-77, 93-91

Configure QUAD-3 for 6264 RAM's - pin 26 to VCC & pin 23 to A11
Remove - NIL
Connect - 96-97, 102-100

Disconnect protective GND on edge connector P3 from board ground
Remove - 4-5
Connect - NIL
Isolate PORT C (56H) bits 0-4 from P3/(p23, p21, p19, p17 & p25)
Remove - 26-25, 28-27, 30-29, 32-31, 24-23
Connect - NIL

Connect timer #1 output (OUT1) to P3/p23 (Petrol Injector #1)
Remove - NIL
Connect - 72-25

Connect timer #2 output (OUT2) to P3/p21 (Petrol Injector #2)
Remove - NIL
Connect - 70-27

Connect timer #3 output (OUT3) to P3/p19 (Petrol Injector #3)
Remove - NIL
Connect - 68-29

Connect timer #4 output (OUT4) to P3/p17 (Petrol Injector #4)
Remove - NIL
Connect - 66-31

Connect timer #5 output (OUT5) to P3/p25 (Hydrogen Injector)
Remove - NIL
Connect - 64-23

Connect external input of timer #1 (IN1) to P3/p9 (CLK) via two schmitt inverters (hex schmitt inverter device installed in spare socket U51)
Remove - NIL
Connect - U26/p8-U51/p1, U51/p2-U51/p3, U51/p4-53

Ground external inputs of timers #2, #3 & #4 (IN2, IN3 & IN4)
Remove - NIL
Connect - 54-55-56-130

Connect external input of timer #5 (IN5) to FOUT
Remove - NIL
Connect - 57-U38/p7
Enable 9513 oscillator
Remove - NIL
Connect - 82-83

Connect PORT C (56H) bit-0 to timer #1 gate input (G1)
Remove - NIL
Connect - 49-26

Connect PORT C (56H) bit-1 to timer #2 gate input (G2)
Remove - NIL
Connect - 48-28

Connect PORT C (56H) bit-2 to timer #3 gate input (G3)
Remove - NIL
Connect - 50-30

Connect PORT C (56H) bit-3 to timer #4 gate input (G4)
Remove - NIL
Connect - 51-32

Connect PORT C (56H) bit-4 to timer #5 gate input (G5)
Remove - NIL
Connect - 52-24

Connect RXRDY of 8251A PCI (AUX serial port - 5CH, 5DH) to P1/p42 (INT 1 on Multibus)
Remove - NIL
Connect - 62-103, 125-126

Select hardware baud rate (speed selectable via S3)
Remove - NIL
Connect - 44-46

In addition to the above jumper selections, the following modifications were carried out to the Am96/5232 board.

Sockets U24 and U25 were fitted with 7437 TTL line drivers (quad 2 input NAND buffers - 48 mA sink current)
The default I/O addresses conflicted with those for the iSBC 80/24 board, and so the factory installed PAL decoder was replaced. As there were no factory installed PAL memory address decoders, four devices had to be programmed in addition to the new I/O decoder. The boolean equations for these five PAL's are given in Appendix A.
Appendix D

Sensor And Actuator Calibration Data
Table D.1 A/D Calibration For Engine Speed

<table>
<thead>
<tr>
<th>ATDSPD</th>
<th>N (r/min)</th>
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<td>50</td>
</tr>
<tr>
<td>30</td>
<td>500</td>
</tr>
<tr>
<td>60</td>
<td>1001</td>
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<td>90</td>
<td>1499</td>
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<tr>
<td>120</td>
<td>2002</td>
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<td>149</td>
<td>2506</td>
</tr>
<tr>
<td>179</td>
<td>3011</td>
</tr>
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<td>209</td>
<td>3513</td>
</tr>
<tr>
<td>237</td>
<td>3981</td>
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<td>266</td>
<td>4479</td>
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<td>327</td>
<td>5512</td>
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<td>360</td>
<td>6073</td>
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Table D.2 A/D Calibration For Accelerator Pedal Potentiometer

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<th>ATDCJL</th>
<th>% DEMAND</th>
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<td>512</td>
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Table D.3  A/D Calibration For HITACHI
Hot-Wire Aneomometer Output Voltage

<table>
<thead>
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<th>ATDHIv (volts)</th>
<th>183</th>
<th>1.75</th>
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<tr>
<td></td>
<td>224</td>
<td>2.03</td>
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<td></td>
<td>251</td>
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<td>270</td>
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<td></td>
<td>318</td>
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<td>476</td>
<td>4.45</td>
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Table D.4 Calibration For BOSCH VAF85G Vane Air Flow Meter

<table>
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<th>v (volts)</th>
<th>( V_a ) (m³ h⁻¹)</th>
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<td>0.64505</td>
<td>13</td>
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<tr>
<td>0.86820</td>
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<tr>
<td>1.05285</td>
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<td>1.21035</td>
<td>22</td>
</tr>
<tr>
<td>1.38985</td>
<td>26</td>
</tr>
<tr>
<td>1.54360</td>
<td>30</td>
</tr>
<tr>
<td>1.70920</td>
<td>35</td>
</tr>
<tr>
<td>1.85270</td>
<td>40</td>
</tr>
<tr>
<td>2.09245</td>
<td>50</td>
</tr>
<tr>
<td>2.28830</td>
<td>60</td>
</tr>
<tr>
<td>2.45395</td>
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<tr>
<td>2.59740</td>
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</tr>
<tr>
<td>2.72395</td>
<td>90</td>
</tr>
<tr>
<td>2.83720</td>
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<td>3.03310</td>
<td>120</td>
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### Table D.5  A/D Calibration For BOSCH Vane Air Flow Meter Output Voltage

<table>
<thead>
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<th>ATDVAF (volts)</th>
<th>v (volts)</th>
</tr>
</thead>
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<tr>
<td>174</td>
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<tr>
<td>223</td>
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<tr>
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<td>2.2349</td>
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<tr>
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<tr>
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<td>ATDHYD (°C)</td>
<td>$V_H$ (L/min)</td>
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<tr>
<td>------------</td>
<td>---------------</td>
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Table D.8 Petrol Injector Calibration
(4 Injectors, Sequencing=Sim. 2/cy, \( \rho_p=750 \text{ kg/m}^3 \))

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<th>( V ) (cm(^3))</th>
<th>( t ) (s)</th>
<th>( \Delta m ) (mg)</th>
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D-7
Table D.9 Hydrogen Injector Calibration  
(4 Injectors, Sequencing=Sim. 4/cy)

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<th>mh (mg/s)</th>
<th>Δm (mg)</th>
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### Table D.10  Hydrogen Pressure Regulator Calibration

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Appendix E

Ford T88 2 Litre OHC Fuel Injected Engine
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<td>Model:</td>
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<td>Type:</td>
<td>4-cylinder, 4-Stroke, Spark Ignition</td>
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<tr>
<td>Stroke:</td>
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<tr>
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<td>Petrol Fuel Injection (EEC-IV)</td>
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<td>Maximum Torque:</td>
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</tr>
<tr>
<td>@ Speed:</td>
<td>4097 (r/min)</td>
</tr>
<tr>
<td>Maximum Power:</td>
<td>85.0 (kW)</td>
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<tr>
<td>@ Speed:</td>
<td>5400 (r/min)</td>
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<td>Maximum Speed:</td>
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<td>Firing Order:</td>
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<tr>
<td>Valve Timing:</td>
<td>i vo  i vc  e vo  e vc</td>
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<tr>
<td></td>
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Appendix F

Calibration Data
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<th>N (r/min)</th>
<th>b.L (cor) (Nm)</th>
<th>Φ</th>
<th>β (°)</th>
<th>θ (°)</th>
<th>V/C</th>
<th>( \eta_s )</th>
<th>( \eta_b )</th>
<th>( \dot{m}_p ) (mg/s)</th>
<th>( \dot{m}_b ) (mg/s)</th>
<th>( \theta_s ) (°C)</th>
<th>CO (%)</th>
<th>CO₂ (%)</th>
<th>NOX (ppm)</th>
<th>CO₂ (µg/J)</th>
<th>CO₃ (µg/J)</th>
<th>NO₃ (µg/J)</th>
<th>C₂H₄ (µg/J)</th>
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1000 r/min, Full Throttle, Petrol Only (Press=101.9 kPa, Temp=20°C)

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<th>β (°)</th>
<th>θ (°)</th>
<th>V/C</th>
<th>( \eta_s )</th>
<th>( \eta_b )</th>
<th>( \dot{m}_p ) (mg/s)</th>
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<th>( \theta_s ) (°C)</th>
<th>CO (%)</th>
<th>CO₂ (%)</th>
<th>NOX (ppm)</th>
<th>CO₂ (µg/J)</th>
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1000 r/min, Zero Load, Petrol Only (Press=99.7 kPa, Temp=20°C)
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<th>Φ</th>
<th>β</th>
<th>γ_d (deg)</th>
<th>η_l</th>
<th>η_r</th>
<th>n_p (m/s)</th>
<th>n_b (m/s)</th>
<th>n_a (g/s)</th>
<th>θ_E (°C)</th>
<th>CO (%)</th>
<th>CO_2 (%)</th>
<th>NOx (ppm)</th>
<th>CO (µg/l)</th>
<th>CO_2 (µg/l)</th>
<th>N_2 (µg/l)</th>
<th>C_3H_8 (µg/l)</th>
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### OPTIMIZED DATA

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<th>γ_d (deg)</th>
<th>η_l</th>
<th>η_r</th>
<th>n_p (m/s)</th>
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<th>CO_2 (%)</th>
<th>NOx (ppm)</th>
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1000 r/min, 73Nm, Petrol Only (Press=97.7 kPa, Temp=20°C)

1000 r/min, 73Nm, β =10% (Press=97.7 kPa, Temp=20°C)

1000 r/min, 73Nm, β =20% (Press=97.7 kPa, Temp=20°C)

1000 r/min, 73Nm, β =30% (Press=97.7 kPa, Temp=20°C)

Petrol

OPTIMIZED

r/mln
| N (r/min) | b.t. (cor) (Nm) | Φ | ρ (g/L) | q | θ (deg) | ηp (%) | ηp (m²/s) | m (m²/s) | $\eta_s$ (g/s) | $\eta_s$ (g/s) | CO (%) | CO₂ (%) | NO₅ (ppm) | C₅H₅ (ppm) | CO (µg/L) | CO₂ (µg/L) | NO₅ (µg/L) | C₅H₅ (µg/L) |
|----------|----------------|---|-------|---|---------|--------|----------|---------|-------------|-------------|--------|---------|-----------|------------|-----------|---------|-----------|-----------|-----------|
| 1000     | 52.4           | 1.006 | 15    | 15 | 13.32  | 27.7   | 33.3     | 405     | 15.3        | 6.44        | 1450   | 760     | 0.0       | 0.0        | 165       | 1.61     | 1.67      | 1.67      |
| 1000     | 53.3           | 0.993 | 20    | 15 | 13.72  | 28.5   | 36.0     | 391     | 15.2        | 6.95        | 1850   | 690     | 0.0       | 0.0        | 147       | 2.25     | 1.42      | 1.42      |
| 1000     | 54.1           | 0.886 | 20    | 15 | 14.40  | 29.3   | 40.4     | 390     | 15.2        | 7.80        | 1100   | 700     | 0.0       | 0.0        | 145       | 1.47     | 1.56      | 1.56      |
| 1000     | 52.7           | 0.683 | 50    | 15 | 14.64  | 36.1   | 44.1     | 372     | 14.0        | 8.59        | 450    | 825     | 0.0       | 0.0        | 142       | 0.679    | 2.04      | 2.04      |
| 1000     | 51.8           | 0.484 | 9.7   | 30  | 16.75  | 29.0   | 52.9     | 383     | 14.8        | 10.21       | 120    | 1100    | 0.0       | 0.0        | 159       | 0.223    | 3.29      | 3.29      |
| 1000     | 52.2           | 1.007 | 25.4  | 15  | 13.31  | 26.5   | 34.8     | 368     | 32.4        | 6.57        | 1750   | 635     | 0.0       | 0.0        | 136       | 1.98     | 1.27      | 1.27      |
| 1000     | 53.6           | 0.999 | 26.1  | 15  | 13.95  | 27.2   | 37.5     | 369     | 33.2        | 7.25        | 1510   | 600     | 0.0       | 0.0        | 137       | 1.85     | 1.27      | 1.27      |
| 1000     | 52.8           | 0.814 | 20.8  | 15  | 14.51  | 27.6   | 41.0     | 358     | 33.6        | 7.92        | 670    | 610     | 0.0       | 0.0        | 135       | 0.920    | 1.43      | 1.43      |
| 1000     | 51.1           | 0.713 | 21.7  | 25  | 15.07  | 28.5   | 44.4     | 336     | 33.3        | 8.57        | 490    | 705     | 0.0       | 0.0        | 129       | 0.744    | 1.79      | 1.79      |
| 1000     | 53.4           | 0.608 | 21.3  | 30  | 16.34  | 29.0   | 52.5     | 340     | 32.3        | 10.12       | 140    | 880     | 0.0       | 0.0        | 132       | 0.248    | 2.55      | 2.55      |
| 999      | 53.2           | 0.500 | 20.0  | 35  | 24.30  | 26.0   | 70.7     | 382     | 34.1        | 13.64       | 25     | 3000    | 0.0       | 0.0        | 139       | 0.666    | 11.7      | 11.7      |

**OPTIMIZED DATA**

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<th>Φ</th>
<th>ρ (g/L)</th>
<th>q</th>
<th>θ (deg)</th>
<th>ηp (%)</th>
<th>ηp (m²/s)</th>
<th>m (m²/s)</th>
<th>$\eta_s$ (g/s)</th>
<th>$\eta_s$ (g/s)</th>
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<th>CO₂ (%)</th>
<th>NO₅ (ppm)</th>
<th>C₅H₅ (ppm)</th>
<th>CO (µg/L)</th>
<th>CO₂ (µg/L)</th>
<th>NO₅ (µg/L)</th>
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N
(r/mln)

b.t. (cor)
(Nm)

<I>

(~l_

a,'

(d••~

•

(de~)

(~)

~~:,

(m~fs)

..,
(mg/s)

m,

(Ris)

Tox
("C)

eo

COz

(%)

(%)

eo

NO,
(ppm)

(ppm)

(~oiJ)

9.0
8.5
7.6
6.6
5.4

840
820
880
370

710
1610
680
950

3.80

ss

2600

8.7
7.9
7.0
6.2
5.0

1000
820
1120
480
72

630
575
610
900

1.25

2080

0.0

171
170
166
167
167

8.0
7.2
6.4
5.7
4.8

1120

520
480
550
630
900

0.0
0.0
0.0
0.0
0.0

153
154
149
151
149

400
412
480
528
680
1400

0.0

C31f3

COz

(~RIJ)

NO,

C3IIs

(~RIJ)

~~·I.J)

1.14

1.61

1.19

3.85

1000 r/mln, 37 Nm, Petrol Only (Press=101.9 kPa, Temp=20"C)

11
37.7
38.2

1.003

1001
1001

36.0

0.803

1003
1007

37.0

0.702

36.6

0.602

1001

0.903

o.o
o.o
0.0
0.0
0.0

20
20
30
30
30

11.92

24.6

27.5

12.37

25.8

29.6

12.60

26.0
26.1
23.4

31.0

13.62
15.30

36.8
47.2

374
362
338
352
387

o.o
o.o
0.0

o.o
o.o

5.54

5.95
6.25
7.44
9.53

478
476
455
457
446

0.3

o.o
o.o
0.0

o.o

0.0

o.o
o.o
o.o

179
181
181
183 .
195

1.43

1.80

0.701
0.135

2.89

1.34

1.45

1.20

1.42
1.64

1.01

1000 r/mln, 37 Nm, P =10% (Press=101.9 kPa, Temp=20"C)

11
1.003

10.4

10.0
10.7
10.2
10.6

1001

37.4
37.0

1000

36.5

6.900
0.806

1003

37.4
37.1

6.603

1001

999

0.701

20
20
35
35
35

11.92

12.37
12.ss
13.57
14.88

24.2

27.2

24.8

29.3
31.8
36.5
44.5

25.9
26.1
24.1

340
329
316
320
333

14.1

5.49

13.0

5.90

13.5
13.0

6.37
7.38

14.1

8.!16

473
466
442
438
430

0.1
0.0
0.0
0.0
0.0

0.0
0.0

o.o

1.81

I

0.884
0.164

2.71

1.46

0.633
0.361

1.19
1.20
1.48
1.90
3.12

140
134
137
133
133
135

1.51
0.821
0.627
0.649
0.208
0.0297

0.989
1.07
1.31
1.64
2.42
6.10

---....

1.05
0.196

2.37
2.66

1.65

1000 r/mln, 37 Nm, P =20% (Press=101.9 kPa, Temp=20"C)

I
1001

37.3

0.999

19.9

1002

37.3
37.5
36.8
37.6

0.895
0.795
0.697
0.599

u.s
19.5

1002

1002
1000

19.6

19.8

20
20
25
30
40

11.81
12.34
12.78
13.46
14.59

23.7

24.5
25.3
25.9
25.4

27.0
29.6
32.2
35.7
42.5

304
301
291
282
286

26.9
26.1
25.2
24.5
25.3

5.42

5.98

6.50
7.19
8.52

459
453
437
425
411

0.0
0.0
0.0
0.0
0.0

940
700
350
170

1.37
1.11

I

1000 r/mln, 37 Nm, P =30% (Press=101.9 kPa, Temp=20"C)

11
1004
1003
1001
1001
1002
1004

37.2

0.918

37.4
37.4

0.832
0.745
0.651

37.0
37.0

0.556
0.465

28.4
28.9
29.5
29.4
29.1
29.4

37.0
37.0

0.750

0.0

0.550

30.0

37.1

15
15
20
30
35
40

13.76
14.63
16.31

30
35

13.14
14.74

12.30

12.60
13.05

24.4
25.1
25.7
26.3
26.8
25.9

42.6
52.5

272
260
251
244
241
249

34.0
43.3

345
242

28.8
30.7
33.4

37.0

38.6

37.8
37.6
36.2
35.5
36.9

5.82
6.18
6.71
7.45

8.60
10.64

449
443
430
410
395
376

0.0
0.0
0.0
0.0
0.0
0.0

6.8
6.1
5.5
4.9
4.2
3.4

1080

456
394

....
....

....

....

550
370
350
96
11

o.o
0.0
0.0

o.o
o.o

OPTIMIZED DATA

11
1000
1000

26.1

26.7

0.0

6,87

35.6

8.73

I

----

----

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....

....

I


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<th>$\beta$ (m/s)</th>
<th>$\gamma$ (deg)</th>
<th>$\dot{n}_P$ (m/s)</th>
<th>$\dot{n}_S$ (m/s)</th>
<th>$\dot{n}_A$ (m/s)</th>
<th>$\gamma$ (C)</th>
<th>CO (%)</th>
<th>CO₂ (%)</th>
<th>NO (ppm)</th>
<th>C₅H₈ (ppm)</th>
<th>CO (ppm)</th>
<th>NO (ppm)</th>
<th>C₅H₈ (ppm)</th>
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**1000 r/min, 10 Nm, Petrol Only (Press=101.9 kPa, Temp=20°C)**

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<th>$\dot{n}_S$ (m/s)</th>
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**1000 r/min, 10 Nm, β =10% (Press=101.9 kPa, Temp=20°C)**

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<th>$\dot{n}_A$ (m/s)</th>
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**1000 r/min, 10 Nm, β =30% (Press=101.9 kPa, Temp=20°C)**

**OPTIMIZED DATA**

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<th>$\dot{n}_A$ (m/s)</th>
<th>$\gamma$ (C)</th>
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<th>CO₂ (%)</th>
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<th>C₅H₈ (ppm)</th>
<th>CO (ppm)</th>
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<td>CO (ug/J)</td>
<td>NO (ug/J)</td>
<td>C_2H_4 (ug/J)</td>
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2000 r/min, Full Throttle, Petrol Only (Press=99.7 kPa, Temp=20°C)

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<th>N_p (mpa)</th>
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<th>n_p (g/s)</th>
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<th>CO_2 (%)</th>
<th>NO_x (ppm)</th>
<th>C_3H_8 (ppm)</th>
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2000 r/min, Zero Load, Petrol Only (Press=99.7 kPa, Temp=20°C)
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2000 r/min, 73 Nm, \( \beta \) =10% (Press=97.7 kPa, Temp=19°C)

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<th>β (deg)</th>
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<th>( \eta_b ) (g/s)</th>
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<th>( \eta_b ) (g/s)</th>
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<th>( \eta_a ) (g/s)</th>
<th>( \eta_b ) (g/s)</th>
<th>CO (%)</th>
<th>CO (%)</th>
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2000 r/min, 73 Nm, \( \beta \) =20% (Press=97.7 kPa, Temp=19°C)

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**2000 r/min, 17 Nm, Petrol Only (Press=99.7 kPa, Temp=20°C)**

| Year | h, b, (Nm) | β (°) | α (°) | η (°/s) | D2 (°) | η2 (°/s) | η1 (°/s) | Θ (°) | Θ2 (°) | Θ1 (°) | CO (%) | CO2 (%) | NO (%) | NO2 (%) | CO (µg/l) | NO (µg/l) | C2H4 (µg/l) | C2H6 (µg/l) | C3H8 (µg/l) |
|------|-----------|------|------|--------|-------|--------|--------|------|-------|--------|--------|--------|--------|--------|--------|----------|----------|-----------|-----------|-----------|
| 1996 | 17.0      | 1.014| 21.3 | 25     | 13.50 | 15.9   | 19.2   | 415  | 40.3  | 7.41   | 601    | 0.6    | 8.1    | 5.8    | 4.9    | 16.4     | 221       | 1.08      | 1.61      |
| 1997 | 17.2      | 0.917| 21.6 | 30     | 13.72 | 16.6   | 19.8   | 385  | 37.6  | 7.62   | 577    | 0.1    | 7.8    | 6.2    | 4.9    | 1.88     | 230       | 1.71      |
| 2001 | 15.9      | 0.714| 21.5 | 35     | 13.95 | 16.8   | 21.3   | 368  | 36.1  | 8.21   | 560    | 0.0    | 7.2    | 6.7    | 4.2    | 1.51     | 238       | 1.63      |
| 1994 | 16.1      | 0.712| 18.8 | 35     | 14.62 | 18.5   | 25.5   | 397  | 32.9  | 9.81   | 544    | 0.0    | 6.5    | 4.9    | 3.0    | 1.42     | 225       | 2.01      |
| 1998 | 18.6      | 0.901| 30.1 | 25     | 13.50 | 16.3   | 19.7   | 379  | 58.3  | 7.59   | 583    | 0.3    | 7.4    | 8.2    | 4.0    | 6.0      | 202       | 1.49      | 1.31      |
| 1998 | 18.2      | 0.908| 30.2 | 30     | 13.55 | 16.9   | 21.1   | 366  | 56.5  | 8.14   | 568    | 0.0    | 7.2    | 8.2    | 3.5    | 1.61     | 205       | 1.20      |
| 2004 | 18.0      | 0.799| 29.0 | 30     | 14.17 | 17.0   | 22.5   | 347  | 52.9  | 8.70   | 548    | 0.0    | 6.5    | 8.1    | 3.8    | 1.77     | 209       | 1.44      |
| 1996 | 18.3      | 0.703| 30.4 | 30     | 14.62 | 17.5   | 25.6   | 344  | 53.6  | 9.84   | 534    | 0.0    | 5.8    | 3.9    | 4.5    | 0.95     | 208       | 1.87      |
| 2008 | 18.3      | 0.604| 30.6 | 35     | 15.07 | 17.9   | 28.5   | 331  | 52.0  | 11.04  | 514    | 0.0    | 5.1    | 1.8    | 4.6    | 0.50     | 212       | 2.24      |
| 1998 | 16.2      | 0.594| 30.7 | 40     | 15.97 | 16.2   | 34.5   | 321  | 52.4  | 13.28  | 496    | 0.0    | 4.2    | 3.0    | 2.0    | 1.13     | 233       | 7.51      |

**Optimized Data**

<p>| Year | h, b, (Nm) | β (°) | α (°) | η (°/s) | D2 (°) | η2 (°/s) | η1 (°/s) | Θ (°) | Θ2 (°) | Θ1 (°) | CO (%) | CO2 (%) | NO (%) | NO2 (%) | CO (µg/l) | NO (µg/l) | C2H4 (µg/l) | C2H6 (µg/l) | C3H8 (µg/l) |
|------|-----------|------|------|--------|-------|--------|--------|------|-------|--------|--------|--------|--------|--------|--------|----------|----------|-----------|-----------|-----------|
| 2000 | 17        | 0.8  | 0.0  | 35     | 14.88 | 14.4   | 26.4   | 550  | 0.0   | 10.20  | 432    | 0.0    | 0.5    | 1.5    | 1.0    | 1.27     | 8.57      |
| 2000 | 17        | 0.7  | 20.0 | 36     | 14.71 | 18.5   | 26.0   | 397  | 33.0  | 10.62  | 541    | 0.0    | 0.0    | 1.3    | 0.0    | 1.38     | 2.37      |</p>
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<th>C&lt;sub&gt;L&lt;/sub&gt; (dps)</th>
<th>θ (dps)</th>
<th>η&lt;sub&gt;p&lt;/sub&gt; (%)</th>
<th>η&lt;sub&gt;n&lt;/sub&gt; (%)</th>
<th>V&lt;sub&gt;p&lt;/sub&gt; (m/s)</th>
<th>V&lt;sub&gt;n&lt;/sub&gt; (m/s)</th>
<th>m&lt;sub&gt;a&lt;/sub&gt; (g/s)</th>
<th>m&lt;sub&gt;n&lt;/sub&gt; (g/s)</th>
<th>CO (%)</th>
<th>CO&lt;sub&gt;2&lt;/sub&gt; (%)</th>
<th>NO&lt;sub&gt;x&lt;/sub&gt; (ppm)</th>
<th>C&lt;sub&gt;3&lt;/sub&gt;H&lt;sub&gt;8&lt;/sub&gt; (ppm)</th>
<th>CO (μg/J)</th>
<th>CO&lt;sub&gt;2&lt;/sub&gt; (μg/J)</th>
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* These values are obviously too low and the carbon dioxide analyser reading is suspected.
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<th>φ</th>
<th>b</th>
<th>α</th>
<th>θ</th>
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<th>η_R</th>
<th>m_H</th>
<th>m_B</th>
<th>t_C</th>
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<th>CO₂ (%)</th>
<th>NOₓ (ppm)</th>
<th>C₃H₈ (ppm)</th>
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<th>CO₂ (ppm)</th>
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4000 r/min, Full Throttle, Petrol Only (Press=98.0 kPa, Temp=20°C)

4000 r/min, Part Load (Press=101.4 kPa, Temp=22°C)

4000 r/min, Zero Load, Petrol Only (Press=99.7 kPa, Temp=20°C)

* These values are obviously too low and the carbon dioxide analyser reading is suspected
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<th>θ (deg)</th>
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<th>η_s (%)</th>
<th>n_p (mg/s)</th>
<th>n_h (mg/s)</th>
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<td>0.859</td>
<td>0.0</td>
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*These values are obviously too low and the carbon dioxide analyser reading is suspected.*
Appendix G

EMISSIONS.FORTRAN
PROGRAM EMISIONS - Copyright A L Entage, 1987.

The basis of this program was taken from a program called EXHEM
written by Dr W L Richards during his work at Loughborough.

This program calculates exhaust emissions in g/kWh and micro-g/J
from exhaust gas analyser data

------------------------------------------------------------------

C PROGRAM DECLARATIONS
IMPLICIT REAL (A-Z)
PARAMETER (PI=22.0/7.0)
CHARACTER ANSWER*50

C Set relevant molecular weights
MWC3H8 = 44.095
MWPET = 111
MWCO  = 28.01
MN02  = 28.0134
MNO2  = 31.9988
MN2   = 30.01
MWH2  = 2.0158
MWAIR = 28.964

C Enter speed and torque data
PRINT *
10 PRINT *
CALL COUA('Enter engine speed (r/min) ')
READ(*,*) SPEED
CALL COUA('Enter brake torque (Nm) ')
READ(*,*) TORQUE

C Calculate brake power in kW
BP = TORQUE * SPEED * 2.0 * PI / (60.0 * 1000.0)

C Enter mass flow rate data
PRINT *
20 CALL COUA('Enter petrol mass flow rate (mg/s) ')
READ(*,*) MPET
MPET = MPET / 1000.0
CALL COUA('Enter hydrogen mass flow rate (mg/s) ')
READ(*,*) MH2
MH2 = MH2 / 1000.0
CALL COUA('Enter air mass flow rate (g/s) ')
READ(*,*) MAIR

C Calculate molar flow rates of air, oxygen, nitrogen, petrol and hydrogen
MOLPET = MPET / MWPET
MOLH2  = MH2  / MWH2
MOLAIR = MAIR / MWAIR
MOLO2  = 0.21 * MOLAIR
MOLN2  = 0.79 * MOLAIR

C Calculate molar flow rate of exhaust CO2 (assume complete combustion)
EXHC02 = 8.0 * MOLPET

G - 2
C Calculate molar flow rate of exhaust N2 (assume no NOx)
EXHN2 = MOLN2

C Calculate molar flow rate of exhaust H2O
EXHH2O = MOLH2 + (MOLPET * 14.6 / 2.0)

C Calculate molar flow rate of exhaust O2 (assume none for rich combustion)
REQO2 = EXHCO2 + (EXHH2O / 2.0)
IF(MOLO2.LE.REQO2) THEN
  EXHO2 = 0.0
ELSE
  EXHO2 = MLO2O - REQO2
END IF

C Calculate total molar flow rate in exhaust (DRY)
MLTOT1 = EXHO2 + EXHCO2 + EXHN2

C Calculate total molar flow rate in exhaust (WET)
MLTOT2 = MLLTOT1 + EXHH20

C Enter analyser data
CALL COUA(‘Enter CO concentration (%) ’)
READ(*,*) PERCO
CALL COUA(‘Enter CO2 concentration (%) ’)
READ(*,*) PERC02
CALL COUA(‘Enter NOx concentration (ppm) ’)
READ(*,*) PPMNO
CALL COUA(‘Enter C3H8 concentration (ppm) ’)
READ(*,*) PPMC3HB

C Calculate molar flow rates of exhaust constituents
CO = PERCO * MLLTOT1 / 100.0
C3H8 = PPMC3HB * MLLTOT2 / 1000000.0
CO2 = PERC02 * MLLTOT1 / 100.0
NO = PPMNO * MLLTOT1 / 1000000.0
N2 = EXHN2 - 0.5 * NO
O2 = EXHO2 - 0.5 * NO

C Calculate g/kWh for exhaust gasses
D1CO = CO * MWCO * 3600.0 / BP
D1C02 = CO2 * MMCO2 * 3600.0 / BP
D1NO = NO * MN0 * 3600.0 / BP
D1C3H8 = C3H8 * MC3H8 * 3600.0 / BP

C Calculate micro-g/J for exhaust gasses
D2CO = D1CO / 3.6
D2C02 = D1C02 / 3.6
D2NO = D1NO / 3.6
D2C3H8 = D1C3H8 / 3.6

C Calculate check value of the percentage carbon dioxide in the exhaust
CHKC02 = EXHC02 - CO - 3.0 * C3H8
ERRC02 = -1*(CO2 - CHKCO2) * 100.0 / CO2
C Display results
WRITE(*,1000) D1CO, D2CO, D1CO2, D2CO2, D1NO, D2NO, D1C3H8, D2C3H8, ERRCO2

C Program flow control
PRINT *
30 CALL COUA('S(top) R(restart) N(ext) ')
READ(*,*) ANSWER
CALL UPCASE(ANSWER)
IF(ANSWER.EQ.'R') THEN
   GOTO 10
ELSE IF(ANSWER.EQ.'N') THEN
   GOTO 20
ELSE IF(ANSWER.NE.'S') THEN
   GOTO 30
END IF

C Terminate program
STOP

C Format statements
1000 FORMAT('-----------------------------------------------•
            CO 1, E9.3, g/kWh = ',E9.3,' micro-g/J',
            CO2 1, E9.3, g/kWh = ',E9.3,' micro-g/J',
            NOx 1, E9.3, g/kWh = ',E9.3,' micro-g/J',
            C3H8 1, E9.3, g/kWh = ',E9.3,' micro-g/J',
            CO2 molar error check = ',E9.3,' %' )
END
Appendix H

SURFIT.FORTRAN
SURFIT IS ONE OF A SUITE OF FOUR PROGRAMS DEVELOPED BY A L EMTAGE FOR DEFINING ENGINE MAPS AND GENERATING LOOK-UP TABLES FOR ENGINE MANAGEMENT MICROPROCESSOR CONTROL SYSTEMS.

SURFIT ACCEPTS ENGINE TEST DATA AS FILE INPUT REPRESENTING A PARTICULAR VARIABLE IN TERMS OF SPEED AND TORQUE COORDINATES. THE DATA MUST BE RECORDED AND STORED IN THE FOLLOWING FORMAT:

i) Select, set and record an engine speed.

ii) Adjust control/driver demand to give lowest possible torque.

iii) Record current torque and output variable.

iv) Maintain set engine speed and increment control/driver demand to increase torque.

v) Repeat iii) and iv) while maintaining speed set point until no higher torque is obtainable. (NB. After the maximum torque point some engines may indicate a drop in torque with further increases in control/driver demand. Data from this region is to be avoided if the variable of interest is to be single valued with respect to speed and torque.)

vi) Repeat i) to v) for various speed settings between minimum and maximum.

vii) Record minimum and maximum speed values to be used for the surface fitting procedure.

-- STORAGE DECLARATIONS --

IMPLICIT DOUBLE PRECISION (A-H, O-Z)
CHARACTER FILE01*50, FILE02*50, ANSWER*50
INTEGER PARAM1, PARAM2, PARAM3, NUNIT1, NUNIT2
PARAMETER (PARAM1=51, PARAM2=51, PARAM3=51)
PARAMETER (NUNIT1=6, NUNIT2=7)
PARAM1 = GRID SIZE IN 'Y' DIRECTION
PARAM2 = GRID SIZE IN 'X' DIRECTION
PARAM3 = LARGEST OF PARAM1 AND PARAM2
COMMON /SURFT1/ XYZDAT(PARAM1,PARAM2,3), SPDMIN &
SDPMAX, NXPTS, NYPTS(PARAM2)
COMMON /SURFT2/ X(PARAM3), Y(PARAM3), Z(PARAM3)
GET INPUT FILENAME AND OPEN IT

PRINT *
CALL COUA('Enter input filename. ',  
READ (UNIT=*, FMT=*) FILE01
OPEN(UNIT=NUNIT1, FILE=FILE01, FORM='FORMATTED'  
& , CARRIAGE=.FALSE.)

INPUT DATA

READ(UNIT=NUNIT1, FMT=*) NXPTS
DO 20, J=1, NXPTS
   All speed values for this J are the same so only store first.
   The rest will be filled in when data is expanded in the 'Y'  
   direction.
   READ(UNIT=NUNIT1, FMT=*) XYZDAT(I,J,1)
   READ(UNIT=NUNIT1, FMT=*) NYPTS(J)
   NY = NYPTS(J)
   DO 10, I=1, NY
      READ(UNIT=NUNIT1, FMT=*) XYZDAT(I,J,2), XYZDAT(I,J,3)
   10 CONTINUE
20 CONTINUE
READ(UNIT=NUNIT1, FMT=*) SPDMIN
READ(UNIT=NUNIT1, FMT=*) SPDMAX

CLOSE INPUT FILE

CLOSE(UNIT=NUNIT1)

CHOOSE CURVE FITTING TECHNIQUE

PRINT *
CALL COUA('Use linear interpolation? ',  
READ (UNIT=*, FMT=*) ANSWER
CALL UPCODE(ANSWER)
IF(ANSWER.EQ.'Y'.OR.ANSWER.EQ.'YE'.OR.ANSWER.EQ.'YES') THEN
   IFLAG = -1
ELSE IF(ANSWER.EQ.'N'.OR.ANSWER.EQ.'NO') THEN
   IFLAG = 1
ELSE
   GOTO 25
END IF

H-3
NSIZ = PARAM1
MPTS = PARAM1
DO 30, J=1, NXPTS
   NY = MPTS(J)
   TRKMIN = XYZDAT(1, J, 2)
   TRKMAX = XYZDAT(NY, J, 2)
   IF(IFLAG.EQ.-1) THEN
      CALL LININT2(XYZDAT(1, J, 2), XYZDAT(1, J, 3), XYZDAT(1, J, 1)
                   , NSIZ, NY, MPTS, TRKMIN, TRKMAX, 1)
   ELSE
      CALL EXSPL2(XYZDAT(1, J, 2), XYZDAT(1, J, 3), XYZDAT(1, J, 1)
                   , NSIZ, NY, MPTS, TRKMIN, TRKMAX, 1)
   END IF
30 CONTINUE

NSIZ = PARAM2
MPTS = PARAM2
NY = PARAM1
DO 60, I=1, NY
   DO 40, J=1, NXPTS
      X(J) = XYZDAT(I, J, 1)
      Y(J) = XYZDAT(I, J, 2)
      Z(J) = XYZDAT(I, J, 3)
40 CONTINUE
   IF(IFLAG.EQ.-1) THEN
      CALL LININT2(X, Y, Z, NSIZ, NXPTS, MPTS
                   & SPDMIN, SPDMAX, 2)
   ELSE
      CALL EXSPL2(X, Y, Z, NSIZ, NXPTS, MPTS
                   & SPDMIN, SPDMAX, 2)
   END IF
   DO 50, J=1, MPTS
      XYZDAT(I,J,1) = X(J)
      XYZDAT(I,J,2) = Y(J)
      XYZDAT(I,J,3) = Z(J)
50 CONTINUE
60 CONTINUE

PRINT *
CALL COUA( 'Enter output filename. ' )
READ(UNIT=*, FMT=*) FILE02
OPEN(UNIT=NUNIT2, FILE=FILE02, FORM='UNFORMATTED')
C ----------
C OUTPUT DATA
C ----------
WRITE(UNIT=NUNIT2) SPDMIN, SPDMAX
NXPTS = PARAM2
WRITE(UNIT=NUNIT2) (XYZDAT(I,J,2), J=1, NXPTS)
& , (XYZDAT(PARAM1,K,2), K=1, NXPTS)
NY = PARAM1
WRITE(UNIT=NUNIT2) (XYZDAT(I,J,3), I=1, NY), J=1, NXPTS)
C -----------------
C CLOSE OUTPUT FILE
C -----------------
CLOSE(UNIT=NUNIT2)
C
C -----------------
C PROGRAM TERMINATION
C -----------------
STOP
END
Appendix I

CONTPLOT.FORTRAN
CONTPLOT IS ONE OF A SUITE OF FOUR PROGRAMS DEVELOPED BY A L EMTAGE FOR DEFINING ENGINE MAPS AND GENERATING LOOK-UP TABLES FOR ENGINE MANAGEMENT MICROPROCESSOR CONTROL SYSTEMS.

CONTPLOT ACCEPTS DATA FROM FILES ('.cal') DEFINING SOME VARIABLE WITH RESPECT TO ENGINE SPEED AND ENGINE BRAKE TORQUE. THE PROGRAM FIRST FINDS THE MAX/MIN LIMITS FOR THIS VARIABLE AND THEN REQUESTS THE USER TO DEFINE THE RANGE AND NUMBER OF CONTOUR LEVELS TO BE FOUND AND PLOTTED. THE CONTOUR COORDINATES ARE DETERMINED BY A CONTOUR TRACING ALGORITHM DEVELOPED BY A TRYPHONOS (FINAL YEAR PROJECT, DEPARTMENT OF TRANSPORT TECHNOLOGY, LOUGHBOROUGH UNIVERSITY, 1986). THE PROGRAM PRODUCES AN OUTPUT FILE ('.plt') IN THE 'include' FORMAT SUITABLE FOR USE WITH THE 'TELLAGRAF' GRAPHICS FACILITY AT LOUGHBOROUGH UNIVERSITY.

---

### STORAGE DECLARATIONS
---

IMPLICIT DOUBLE PRECISION (A-H, O-Z)
CHARACTER FILCAL*50, FILPLT*50, ANSWER*50, TITLE*lOO
INTEGER PARAM1, PARAM2, PARAM3, PARAM4, NUNIT1, NUNIT2

INTEGER IPATH(5,2,3,4)
DATA IPATH/ 2, 3, 6, 0, 0, 4, 2, 5, 0, 0
       , 4, 2, 5, 0, 0, 3, 1, 1, 0,-1
       , 3, 1, 1, 0,-1, 2, 3, 6, 0, 0

       , 3, 3, 7, 0, 0, 1, 2, 6, 0, 0
       , 1, 2, 6, 0, 0, 4, 1, 2,-1, 0

       , 4, 1, 2,-1, 0, 3, 3, 7, 0, 0

       , 4, 3, 8, 0, 0, 2, 2, 7, 0, 0
       , 2, 2, 7, 0, 0, 1, 1, 3, 0,+1

       , 1, 1, 3, 0,+1, 4, 3, 8, 0, 0

       , 1, 3, 5, 0, 0, 3, 2, 8, 0, 0
       , 3, 2, 8, 0, 0, 2, 1, 4,+1, 0

       , 2, 1, 4,+1, 0, 1, 3, 5, 0, 0/

REAL CNTXCO, CNTYCO
CHARACTER VALLAB(ll)*lO
PARAMETER (PARAM1=51, PARAM2=51, PARAM3=5000, PARAM4=50)
PARAMETER (NUNIT1=6, NUNIT2=7)

---
PARAM1 = GRID SIZE IN 'Y' DIRECTION
PARAM2 = GRID SIZE IN 'X' DIRECTION
PARAM3 = MAXIMUM NUMBER OF CONTOUR POINTS
PARAM4 = MAX NUMBER OF CONTOUR SEGMENTS

COMMON /CNTPL1/ GRDINF(PARAM1,PARAM2)
COMMON /CNTPL2/ GRIDWRK(PARAM2,PARAM1,3)
COMMON /CNTPL3/ SPDMIN, SPDMAX, TRKMIN(PARAM2), TRKMAX(PARAM2)
COMMON /CNTPL4/ CNTXCO(PARAM3), CNTYCO(PARAM3)
COMMON /CNTPL5/ CNTVAL, NWORK(PARAM2,PARAM1), NPOINT(PARAM4)
& , NSEG(101)
COMMON /CNTPL6/ XIAB(PARAM4*11), YLAB(PARAM4*11)
& , IFLAG(PARAM4*11)

---------------------------------------------------
GET INPUT FILENAME, OPEN IT, READ DATA AND CLOSE IT
---------------------------------------------------

PRINT *
CALL COUA('Enter '.cal' input filename. ')
READ(UNIT=*, FMT=*) FILCAL
L = LENG(FILCAL)
IF(FILCAL(L-3:1).NE.''.cal') FILCAL = FILCAL // '.cal'
OPEN(UNIT=NUNIT1, FILE=FILCAL, FORM='UNFORMATTED')
READ(UNIT=NUNIT1) SPDMIN, SPDMAX
NXPTS = PARAM2
READ(UNIT=NUNIT1) (TRKMIN(J), J=1, NXPTS)
& , (TRKMAX(K), K=1, NXPTS)
NYPTS = PARAM1
READ(UNIT=NUNIT1) ((GRDINF(I,J), I=1, NYPTS), J=1, NXPTS)
CLOSE(UNIT=NUNIT1)

--------------------------------
REARRANGE 'X' AND 'Y' COORDINATES
--------------------------------
NYPTS = PARAM1
NXPTS = PARAM2
DO 20, I=1, NYPTS
SCALEY = (I - 1.0) / (NYPTS - 1.0)
DO 10, J=1, NXPTS
GRIDWRK(J,I,1) = SPDMIN + (SPDMAX - SPDMIN) * (J - 1.0) / (NXPTS - 1.0)
& GRIDWRK(J,I,2) = TRKMIN(J) + (TRKMAX(J) - TRKMIN(J)) * SCALEY
& GRIDWRK(J,I,3) = GRDINF(I,J)
10 CONTINUE
20 CONTINUE

-----------------------------------
SELECT CONTOUR LIMITS AND INTERVALS
-----------------------------------
NPTS = PARAM1 * PARAM2
CALL DMNMAX(GRDWRK(1,1,3), NPTS, CNTMIN, CNTMAX)
PRINT *
CALL COUA('Data limits are: MIN = ')
WRITE(UNIT=*, FMT=('E10.4,5') CNTMIN

I-3
CALL COUA(' and MAX = ')  
WRITE(UNIT=*, FMT=*(E10.4)) CNTMAX  
PRINT *  
CALL COUA('Enter desired minimum value. ')  
READ(UNIT=*, FMT=*) CNTMIN  
PRINT *  
CALL COUA('Enter desired maximum value. ')  
READ(UNIT=*, FMT=*) CNTMAX  
PRINT *  
CALL COUA('Enter number of major intervals (<11). ')  
READ(UNIT=*, FMT=*) MAJINT  
MAJINT = MAJINT + 1  
DO 21, I=1, MAJINT  
WRITE(UNIT=*, FMT=9900) I  
READ(UNIT=*, FMT='(A)') VALLAB(I)  
CONTINUE  
PRINT *  
CALL COUA('Enter number of minor intervals (<11). ')  
READ(UNIT=*, FMT=*) MININT  
C GET OUTPUT FILENAME AND OPEN IT  
C -------------------------------  
C WRITE HEADER INFORMATION TO PLOT FILE  
C -------------------------------  
WRITE(UNIT=NUNIT2, FMT=1000)  
C WRITE ENGINE MAP LIMITS TO PLOT FILE  
C -------------------------------------  
NYPTS = PARAM1  
NXPTS = PARAM2  
WRITE(UNIT=NUNIT2, FMT=2000) 0.0, 0  
J = 1  
DO 22, I=1, (NYPTS-1)  
WRITE(UNIT=NUNIT2, FMT=3000) GRDWRK(J,I,1), GRDWRK(J,I,2)  
CONTINUE  
I = NYPTS  
DO 24, J=1, (NXPTS-1)  
WRITE(UNIT=NUNIT2, FMT=3000) GRDWRK(J,I,1), GRDWRK(J,I,2)  
CONTINUE  
J = NXPTS  
DO 26, I=NYPTS, 2, -1  
WRITE(UNIT=NUNIT2, FMT=3000) GRDWRK(J,I,1), GRDWRK(J,I,2)  
CONTINUE  
I = 1
DO 28, J=NXPTS, 1, -1
WRITE(UNIT=NUNIT2, FMT=30000) GRDWRK(J, I, 1), GRDWRK(J, I, 2)
28   CONTINUE

C -------------------------------------------------
C TRACE CONTOURS AND WRITE COORDINATES TO PLOT FILE
C -------------------------------------------------

NCONT = 0
NLAB = 0
MCONT = ((MAJINT - 1) * MININT) + 1
DO 60, MAJ=1, MAJINT
   DO 50, MIN=1, MININT
      CNTVAL = CNTMIN + (CNTMAX - CNTMIN) * (NCONT - 1.0) / (MCONT - 1.0)
   5   NXPTS = PARAM2
      NYPTS = PARAM1
      NXYCO = PARAM3
      NPSIZ = PARAM4
      CALL TRACE(GRDWRK, NWORK, NXPTS, NYPTS, 3, CNTVAL
      , CNTXCO, CNTYCO, NXYCO, NPOINT, NPSIZ
      , IPATH, 5, 2, 3, 4)
      NSEG(NCONT) = 0
   30   NPTS1 = 1
      NPTS2 = NPOINTINSEG(NCONT, 1)
   40   NSEG(NCONT) = NSEG(NCONT) + 1
      WRITE(UNIT=NUNIT2, FMT=2000) CNTVAL, NSEG(NCONT)
   45   CONTINUE
      IF(MIN.EQ.1) THEN
         NLAB = NLAB + 1
         IF((NPTS2-NPTS1) .GT. 5) THEN
            NL = (NPTS1 + NPTS2) / 2
            XLAB(NLAB) = CNTXCO(NL)
            YLAB(NLAB) = CNTYCO(NL)
            IFLAG(NLAB) = 1
         ELSE
            IFLAG(NLAB) = 0
         END IF
   END IF
      END IF
   60   CONTINUE

C -------------------------------------------------
C INPUT TITLE FOR ENGINE MAP AND WRITE FORMATING INFORMATION TO PLOT FILE
C -------------------------------------------------

PRINT *
CALL COUA( 'Enter title for engine map. ')
READ(UNIT=*, FMT='(A)') TITLE

I-5
PRINT *
CALL COUA('Speed-Torque map? ', 1)
READ(UNIT=*, FMT='(A)') ANSWER
CALL UPCASE(ANSWER)

WRITE(UNIT=NUNIT2, FMT=4000)
WRITE(UNIT=NUNIT2, FMT=4001)
IF(ANSWER.EQ.'Y'.OR.ANSWER.EQ.'YE'.OR.ANSWER.EQ.'YES') THEN
  WRITE(UNIT=NUNIT2, FMT=4010)
ELSE
  WRITE(UNIT=NUNIT2, FMT=4011)
END IF

LTITLE = LENG(TITLE)
WRITE(UNIT=NUNIT2, FMT=4002) TITLE(1:LTITLE)
WRITE(UNIT=NUNIT2, FMT=4003)
WRITE(UNIT=NUNIT2, FMT=4004)
WRITE(UNIT=NUNIT2, FMT=4005)

C --------------------------------------------------------
C WRITE PLOT FILE CONTOUR DEFINITIONS
C --------------------------------------------------------
NCONT = 0
NLAB = 0
NCURVE = 1
DO 90, MAJ-1, MAJINT
  DO 80, MIN-1, MININT
    IF((MAJ.EQ.MAJINT).AND.(MIN.GT.1)) GOTO 90
    NCONT = NCONT + 1
    MSEG = NSEG(NCONT)
    IF(MSEG.EQ.0) GOTO 80
    DO 70, I=1, MSEG
      NCURVE = NCURVE + 1
      IF(MIN.EQ.1) THEN
        NLAB = NLAB + 1
        WRITE(UNIT=NUNIT2, FMT=5000) NCURVE
        IF(IFLAG(NLAB).EQ.1) THEN
          LENLAB = LENG(VALLAB(MAJ))
          WRITE(UNIT=NUNIT2, FMT=5100)
          NLAB, XLAB(NLAB), YLAB(NLAB)
          NLAB, VALLAB(MAJ)(1:LENLAB)
        END IF
      ELSE
        WRITE(UNIT=NUNIT2, FMT=6000) NCURVE
      END IF
    70 CONTINUE
  80 CONTINUE
90 CONTINUE
WRITE(UNIT=NUNIT2, FMT=7000)
WRITE(UNIT=NUNIT2, FMT=8000)

C --------------------------------------------------------
C CLOSE PLOT FILE
C --------------------------------------------------------
CLOSE(UNIT=NUNIT2)
C PLOT FILE FORMAT STATEMENTS
C

1000 FORMAT('case is ascii.'
&/ 'centimeter on.'
&/ 'generate a plot.'
&/ 'input data.')

2000 FORMAT('"",E10.4,2X,E10.4)
3000 FORMAT(E10.4,2X,E10.4)
4000 FORMAT('end of data.'
C Page setup
&/ 'page x 29.7 y 21.0.'
&/ 'page border off.'

4001 FORMAT(
C Axes setup
&/ 'existence x on y on.'
&/ 'x mode normal.'
&/ 'y mode normal.'
&/ 'origin x 4.35 y 4.0.'
&/ 'length x 21.0 y 13.0.'
&/ 'minimum x 0.0.'
&/ 'no axis frame.'
&/ 'alphabet x standard y standard.'
&/ 'style x swiss-light y swiss-light.'
&/ 'height x 0.4 y 0.4.'

4010 FORMAT(
&/ 'x label text "Engine Speed (r/min)".'
&/ 'y label text "Brake Torque (Nm)".')

4011 FORMAT(
&/ 'x label text "Hydrogen Energy Fraction".'
&/ 'y label text "Equivalence Ratio".')

4002 FORMAT(
C Title setup
&/ 'title existence on.'
&/ 'title alphabet standard.'
&/ 'title style swiss-medium.'
&/ 'title units centimeters.'
&/ 'title height 0.7.'
&/ 'title box 4.0 25.7 0.8 1.8.'
&/ 'title connect ct.'
&/ 'title text ":A,"')

4003 FORMAT(
C Legend setup
&/ 'legend existence no.'

4004 FORMAT(
C Curve setup
&/ 'every curve symbol count 0.'
&/ 'every curve scattered no.'
&/ 'every curve color black.')

4005 FORMAT(
C Message setup
&/ 'every message units coordinate.'
&/ 'every message height 0.3.'
&/ 'every message style swiss-light.'
&/ 'every message connect ct.'
&/ 'every message blanking on.'

I-7
SUBROUTINE TRACE(GRID, NWRK, NX, NY, NSUB
 & , Z, X, Y, NXY, NPT, NS
 & , IPATH, IP1, IP2, IP3, IP4)
IMPLICIT DOUBLE PRECISION (A-H, O-Z)
INTEGER NX, NY, NWRK(NX,NY), NXY, NPT(NS), NS
 & , ORUTE
INTEGER IPATH(IP1,IP2,IP3,IP4)
REAL X(NXY), Y(NXY)
DOUBLE PRECISION GRID(NX,NY,NSUB)
DO 20, IY=1, NY
   DO 10, IX=1, NX
      NWRK(IX,IY) = 0
   10 CONTINUE
20 CONTINUE
DO 30, IS=1, NS
   NPT(IS) = 0
30 CONTINUE
IS = 1
DO 90, IY=1, (NY-1)
   DO 80, IX=1, (NX-1)
      NWRK = 3 means AB & AC crossings already traced
      IF(NWRK(IX,IY).NE.3) THEN
         C
         C
NWRK = 1 means AB crossing already traced
IF(NWRK(IX,IY),NE.1) THEN
   ITYPE = 1
   CALL DEFCRS(GRID,NX,NY,NSUB,IX,IY,ITYPE
   ,X1,X2,Y1,Y2,Z1,Z2)
   CALL CROSS(X1,X2,Y1,Y2,Z1,Z2,Z,X(NPT(IS)+1)
   ,Y(NPT(IS)+1),NPT(IS),NWRK,NX,NY,IX,IY
   ,NCROSS,ITYPE)
END IF
C
Initial crossing point found
IF(NCROSS.EQ.1) THEN
   IDIR = 1
   ITRANG = 1
   INRUTE = 1
   OURUTE = 1
   IYY = IY
   IXX = IX
   ITYPE = IPATH(IS,OURUTE,INRUTE,ITRANG)
   CALL DEFCRS(grid,NX,NY,NSUB,IXX,IYY,ITYPE
   ,X1,X2,Y1,Y2,Z1,Z2)
   CALL CROSS(X1,X2,Y1,Y2,Z1,Z2,Z,X(NPT(IS)+1)
   ,Y(NPT(IS)+1),NPT(IS),NWRK,NX,NY,IXX,IYY
   ,NCROSS,ITYPE)
C
Not a crossing point so it must be 2nd route...
...out of triangle (NB. no error trapping on 2nd route)
IF(NCROSS.EQ.0.AND.OURUTE.EQ.1) THEN
   OURUTE = 2
   GOTO 50
END IF
C
Not a crossing point so it must be 2nd route...
...out of triangle (NB. no error trapping on 2nd route)
IF(NCROSS.EQ.0.AND.OURUTE.EQ.1) THEN
   OURUTE = 2
   GOTO 50
END IF
C
Contour closed if it crosses initial crossing point
IF((IXX.EQ.IX).AND.(IYY+1).EQ.IY).AND.
   (ITRANG.EQ.3).AND.(IDIR.EQ.1)) THEN
   ICOMPL = 1
ELSE
   ICOMPL = 0
END IF
IXXADD = IPATH(4,OURUTE,INRUTE,ITRANG)
IYYADD = IPATH(5,OURUTE,INRUTE,ITRANG)
IXX = IXX + IXXADD
IYY = IYY + IYYADD
C
If boundary reached or contour closed
IF((IXX.LT.1.OR.IXX.GT.(NX-1)).OR.IYY.LT.
   1.OR.IYY.GT.(NY-1).OR.ICOMPL.EQ.1) THEN
   IDIR = -1
   IYY = IY - 1
   ITRANG = 3
   INRUTE = 1
C
OURUTE = 1
IS = IS + 1
NPT(IS) = NPT(IS-1)
GOTO 40
END IF

Contour segment is incomplete
ELSE

NXTITR = IPATH(1,OURUTE,INRUTE,ITRANG)
NXTINR = IPATH(2,OURUTE,INRUTE,ITRANG)
ITRANG = NXTITR
INRUTE = NXTINR
OURUTE = 1
GOTO 50
END IF

All possibilities are exhausted for this initial...

...point so go on to find a new initial point
IS = IS + 1
NPT(IS) = NPT(IS-1)
END IF
END IF

C NWKR = 2 or 3 means AC crossing already traced
IF((NWRK(INX,INY).EQ.0) .OR. (NWRK(INX,INY).EQ.1)) THEN

ITYPE = 2
CALL DEFCRS(GRIO,NX,NY,NSUB,IX,IY,ITYPE,
X1,X2,Y1,Y2,Z1,Z2)
CALL CROSS(X1,X2,Y1,Y2,Z1,Z2,Z,X(NPT(IS)+1),
Y(NPT(IS)+1),NPT(IS),NWRK,NX,NY,IX,IY,INRUTE,
NXTITR,ITRANG,ITYPE)
END IF

C Initial crossing point found
IF(NCROSS.EQ.1) THEN

IDIR = 1
ITRANG = 2
INRUTE = 1
OURUTE = 1
IXX = IX
IYY = IY
60

ITYPE = IPATH(3,OURUTE,INRUTE,ITRANG)
CALL DEFCRS(GRIO,NX,NY,NSUB,IXX,IYY,ITYPE,
X1,X2,Y1,Y2,Z1,Z2)
CALL CROSS(X1,X2,Y1,Y2,Z1,Z2,Z,X(NPT(IS)+1),
Y(NPT(IS)+1),NPT(IS),NWRK,NX,NY,IXX,IYY,INRUTE,
NXTITR,ITRANG,ITYPE)
END IF

C Not a crossing point so it must be 2nd route...

...out of triangle (NB. no error trapping on 2nd route)
IF(NCROSS.EQ.0.AND.OURUTE.EQ.1) THEN

OURUTE = 2
GOTO 70
END IF

C Contour closed if it crosses initial crossing point
IF((IXX+1).EQ.IX).AND.(IYY.EQ.IY).AND.(
ITRANG.EQ.4).AND.(IDIR.EQ.1)) THEN

ICOMPL = 1
ELSE

ICOMPL = 0
END IF

I- 10
IHXADD = IPATH(4, CURUTE, INRUTE, ITRANG)
IHYADD = IPATH(5, CURUTE, INRUTE, ITRANG)
IXX = IXX + IHXADD
IYY = IYY + IHYADD

C If boundary reached or contour closed
IF (IXX .LT. 1 OR IXX .GT. (NX-1) OR IYY .LT. 1 OR IYY .GT. (NY-1) OR ICOMPL.EQ.1) THEN
   6
C If last direction was +ve and start point was not...
   IF (IDIR.EQ.1) AND (IX.NE.1) AND (ICOMPL.EQ.0)) THEN
      END IF
C Contour segment is incomplete
ELSE
   NXTITR = IPATH(1, CURUTE, INRUTE, ITRANG)
   NXTINR = IPATH(2, CURUTE, INRUTE, ITRANG)
   ITRANG = NXTITR
   INRUTE = NXTINR
   CURUTE = 1
   GOTO 70
END IF
C All possibilities are exhausted for this initial...
C ...point so go on to find a new initial point
   IS = IS + 1
   NPT(IS) = NPT(IS-1)
   END IF
END IF

80 CONTINUE
90 CONTINUE

NPT(IS) = 0
RETURN
END

C ---------------------------------------------------------------
C SUBROUTINE TO TEST FOR CROSS OVER AND STORE COORDINATES IF TRUE
C ---------------------------------------------------------------
SUBROUTINE CROSS(X1, X2, Y1, Y2, Z1, Z2, Z, X, Y, NPT, NWRK, NX, NY, IXX, IYY
   & , NCRESS, NTYP)
IMPLICIT DOUBLE PRECISION (A-H, O-Z)
INTEGER NXY, NSEG, NWRK(NX,NY), NCRESS, NTYP

I-11
REAL X, Y

Crossing point found if Z is within range
IF( ((Z1.LE.Z) .AND. (Z.LT.Z2)) .OR,
& ((Z1.LE.Z) .AND. (Z.LT.Z2)) ) THEN
  IF(Z1.EQ.Z .AND. (ITYPE.EQ.1 .OR. ITYPE.EQ.2)) THEN
    NWRK(IXX, IYY) = 3
    IF(IYY.GT.1) THEN
      IF(NWRK(IXX-1, IYY-1).EQ.1) THEN
        NWRK(IXX-1, IYY-1) = 3
      ELSE IF(NWRK(IXX-1, IYY-1).EQ.0) THEN
        NWRK(IXX-1, IYY-1) = 2
      END IF
    END IF
    IF(IXX.GT.1) THEN
      IF(NWRK(IXX-1, IYY).EQ.2) THEN
        NWRK(IXX-1, IYY) = 3
      ELSE IF(NWRK(IXX-1, IYY) .EQ.O) THEN
        NWRK(IXX-1, IYY) = 1
      END IF
    END IF
    GOTO 10
  ELSE IF( (Z1.EQ.Z .AND. ITYPE.EQ.3) .OR, (Z2.EQ.Z .AND. ITYPE.EQ.2)) THEN
    NWRK(IXX, IYY+1) = 3
    IF(IYY.GT.1) THEN
      IF(NWRK(IXX, IYY-1).EQ.1) THEN
        NWRK(IXX, IYY-1) = 3
      ELSE IF(NWRK(IXX, IYY-1).EQ.0) THEN
        NWRK(IXX, IYY-1) = 2
      END IF
    END IF
    IF(IXX.GT.1) THEN
      IF(NWRK(IXX-1, IYY+1).EQ.2) THEN
        NWRK(IXX-1, IYY+1) = 3
      ELSE IF(NWRK(IXX-1, IYY+1) .EQ.O) THEN
        NWRK(IXX-1, IYY+1) = 1
      END IF
    END IF
    GOTO 10
  ELSE IF( (Z1.EQ.Z .AND. ITYPE.EQ.4) .OR, (Z2.EQ.Z .AND. ITYPE.EQ.1)) THEN
    NWRK(IXX+1, IYY) = 3
    IF(IYY.GT.1) THEN
      IF(NWRK(IXX+1, IYY-1).EQ.1) THEN
        NWRK(IXX+1, IYY-1) = 3
      ELSE IF(NWRK(IXX+1, IYY-1).EQ.0) THEN
        NWRK(IXX+1, IYY-1) = 2
      END IF
    END IF
    IF(IXX.GT.1) THEN
      IF(NWRK(IXX+1, IYY).EQ.2) THEN
        NWRK(IXX+1, IYY) = 3
      ELSE IF(NWRK(IXX+1, IYY) .EQ.O) THEN
        NWRK(IXX+1, IYY) = 1
      END IF
    END IF
    GOTO 10
  ELSE IF( (Z2.EQ.Z .AND. (ITYPE.EQ.3 .OR. ITYPE.EQ.4)) THEN
    NWRK(IXX+1, IYY+1) = 3
    IF(NWRK(IXX+1, IYY).EQ.1) THEN
      I+12
NWRK(IXX+1, IYY) = 3
ELSE IF(NWRK(IXX+1, IYY) .EQ. 0) THEN
  NWRK(IXX+1, IYY) = 2
END IF
IF(NWRK(IXX, IYY+1) .EQ. 2) THEN
  NWRK(IXX, IYY+1) = 3
ELSE IF(NWRK(IXX, IYY+1) .EQ. 0) THEN
  NWRK(IXX, IYY+1) = 1
END IF
GOTO 10
END IF

ELSE IF(NWRK(IXX, IYY) .EQ. 0 .AND. ITYPE .EQ. 1) THEN
  NWRK(IXX, IYY) = 1
ELSE IF(NWRK(IXX, IYY) .EQ. 0 .AND. ITYPE .EQ. 2) THEN
  NWRK(IXX, IYY) = 2
ELSE IF(NWRK(IXX, IYY) .EQ. 1 .AND. ITYPE .EQ. 2) THEN
  NWRK(IXX, IYY) = 3
END IF

10  X = REAL(X1 + (X2 - X1) * (Z2 - Z1) / (Z2 - Z1))
    Y = REAL(Y1 + (Y2 - Y1) * (Z2 - Z1) / (Z2 - Z1))
    NPT = NPT + 1
    NCROSS = 1
ELSE
  NCROSS = 0
END IF

RETURN

END
Y1 = GRID(IX, IY, 2)
Y2 = GRID(IX+1, IY, 2)
Z1 = GRID(IX, IY, 3)
Z2 = GRID(IX+1, IY, 3)

C If AC crossing
ELSE IF (ITYPE.EQ.2) THEN
X1 = GRID(IX, IY, 1)
X2 = GRID(IX+1, IY+1, 1)
Y1 = GRID(IX, IY, 2)
Y2 = GRID(IX+1, IY+1, 2)
Z1 = GRID(IX, IY, 3)
Z2 = GRID(IX+1, IY+1, 3)
C If CD crossing
ELSE IF (ITYPE.EQ.3) THEN
X1 = GRID(IX+1, IY, 1)
X2 = GRID(IX+1, IY+1, 1)
Y1 = GRID(IX+1, IY, 2)
Y2 = GRID(IX+1, IY+1, 2)
Z1 = GRID(IX+1, IY, 3)
Z2 = GRID(IX+1, IY+1, 3)
C If BD crossing
ELSE IF (ITYPE.EQ.4) THEN
X1 = GRID(IX+1, IY+1, 1)
X2 = GRID(IX+1, IY+1, 1)
Y1 = GRID(IX+1, IY+1, 2)
Y2 = GRID(IX+1, IY+1, 2)
Z1 = GRID(IX+1, IY+1, 3)
Z2 = GRID(IX+1, IY+1, 3)
C If BE crossing
ELSE IF (ITYPE.EQ.5) THEN
X1 = GRID(IX+1, IY+1, 1)
X2 = (GRID(IX, IY+1, 1) + GRID(IX+1, IY+1, 1)) / 4.0
Y1 = GRID(IX+1, IY+1, 2)
Y2 = (GRID(IX, IY+1, 2) + GRID(IX+1, IY+1, 2)) / 4.0
Z1 = GRID(IX+1, IY+1, 3)
Z2 = (GRID(IX+1, IY+1, 3) + GRID(IX+1, IY+1, 3)) / 4.0
C If AE crossing
ELSE IF (ITYPE.EQ.6) THEN
X1 = GRID(IX, IY+1, 1)
X2 = (GRID(IX, IY+1, 1) + GRID(IX+1, IY+1, 1)) / 4.0
Y1 = GRID(IX, IY+1, 2)
Y2 = (GRID(IX, IY+1, 2) + GRID(IX+1, IY+1, 2)) / 4.0
Z1 = GRID(IX, IY+1, 3)
Z2 = (GRID(IX, IY+1, 3) + GRID(IX+1, IY+1, 3)) / 4.0
C If EC crossing
ELSE IF (ITYPE.EQ.7) THEN
X1 = GRID(IX, IY+1, 1)
X2 = (GRID(IX, IY+1, 1) + GRID(IX+1, IY+1, 1)) / 4.0
Y1 = GRID(IX, IY+1, 2)
Y2 = (GRID(IX, IY+1, 2) + GRID(IX+1, IY+1, 2)) / 4.0
Z1 = GRID(IX, IY+1, 3)
Z2 = (GRID(IX, IY+1, 3) + GRID(IX+1, IY+1, 3)) / 4.0

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GRID(IY+1,IX+1,2) + GRID(IY+1,IX+1,3)) / 4.0
Z2 = (GRID(IY,IX,3) + GRID(IY+1,IX,3)
+ GRID(IY,IX+1,3) + GRID(IY+1,IX+1,3)) / 4.0

If ED crossing
ELSE IF (ITYPE.EQ.8) THEN
X1 = GRID(IX+1,IX+1,1)
X2 = (GRID(IX,IX,1) + GRID(IX+1,IX,1))
+ GRID(IX,IX+1,1) + GRID(IX+1,IX+1,1)) / 4.0
Y1 = GRID(IX+1,IX+1,2)
Y2 = (GRID(IX,IX,2) + GRID(IX+1,IX,2))
+ GRID(IX+1,IX+1,2) + GRID(IX+1,IX+1,2)) / 4.0
Z1 = GRID(IX+1,IX+1,3)
Z2 = (GRID(IX,IX,3) + GRID(IX+1,IX,3)
+ GRID(IX,IX+1,3) + GRID(IX+1,IX+1,3)) / 4.0
END IF

RETURN
END
Appendix J

DIGITIZE.FORTRAN
DIGITIZE IS ONE OF A SUITE OF FOUR PROGRAMS DEVELOPED BY A L EMTAGE FOR DEFINING ENGINE MAPS AND GENERATING LOOK-UP TABLES FOR ENGINE MANAGEMENT MICROPROCESSOR CONTROL SYSTEMS.

DIGITIZE ALLOWS THE USER TO USE THE NESTLER DIGITIZER IN THE DEPT. OF TRANS. TECM. AT LIT IN ORDER TO DEFINE THE SHAPES OF DRIVER DEMAND/CONTROL LINES ON THE ENGINE'S TORQUE VERSUS SPEED MAP. THE PROCEDURE IS AS FOLLOWS:

i) The user prepares a scaled graph of constant driver demand lines from 0% demand to 100% demand. These lines are drawn with respect to axes of Brake Torque (BT) in units of Nm and of Engine Speed (SPD) in units of rev/min. The positions and coordinate values of the axes' limits must be indicated.

ii) The prepared graph is fixed to the digitizer surface and upon request the axes' limits are indicated by the digitizer cursor. This procedure defines the necessary scaling and angular transformations for the graphical data.

iii) Upon request and starting with 0% demand the user will enter the percentage demand and then indicate up to 500 points on that demand line. Up to 100 demand lines in increasing order may be defined in this way including that for 100% demand.

iv) Upon request the user supplies a file name from which the program will generate two new file names the following extensions:

   a = '.plt'
   b = '.gen'

File 'a' is a 'TELLAGRAF' 'include.' file defining driver/control demand as a function of torque and engine speed while file 'b' will contain torque as a function of demand and speed suitable as input data to program GENERATE.

STORAGE DECLARATIONS

IMPLICIT DOUBLE PRECISION (A-H, O-Z)
CHARACTER FILEOl*50, FILEO2*50, ANSWER*50, TITLE*100
INTEGER PARAMl, PARAM2, PARAM3, PARAM4, NUNITl, NUNIT2
PARAMETER (PARAM1=500, PARAM2=50, PARAM3=51, PARAM4=51)
PARAMETER (NUNITl=6, NUNIT2=7)
COMMON /DIGIT!/ ALFA, XSO, YSO, XS2PP, YSlPP, XGO, YGO, XG2, YGl
COMMON /DIGIT2/ XYINP(PARAM1,PARAM2,2), DEMAND(PARM2), SPDMIN & SPDMAX, NDEMAND, NXYINP(PARAM2)
COMMON /DIGIT3/ XYZDAT(PARAM3,PARAM4,3)
CHOOSE CURVE FITTING TECHNIQUE

1 PRINT *
CALL COUA('Use linear interpolation? ')
READ (UNIT=*, FMT=*) ANSWER
CALL UPCASE(ANSWER)
IF(ANSWER.EQ.'Y'.OR.ANSWER.EQ.'YE'.OR.ANSWER.EQ.'YES') THEN
  INTFLG = -1
ELSE IF(ANSWER.EQ.'N'.OR.ANSWER.EQ.'NO') THEN
  INTFLG = 1
ELSE
  GOTO 1
END IF

CHOOSE TO DIGITIZE DATA OR READ FROM FILE

5 PRINT *
CALL COUA('Digitize data? ')
READ(*,*) ANSWER
CALL UPCASE(ANSWER)
IF(ANSWER.EQ.'N'.OR.ANSWER.EQ.'NO') THEN
  PRINT *
  CALL COUA('Enter input file name. ')
  READ(*,*) ANSWER
  OPEN(UNIT=9,FILE=ANSWER,FORM='FORMATTED'
      ,CARRIAGE=.FALSE.)
  READ(9,*) SPMIN
  READ(9,*) SPDMAX
  READ(9,*) N0
  DO 6, I=1, N0
     READ(9,*) XYINP(I,1,1), XYINP(I,1,2)
  CONTINUE
  DEMAND(1) = 0.0
  MPTS - PARAM4
  NSIZ = PARAM1
  NXYINP(1) = N0
  IF(INTFLG.EQ.-1) THEN
     CALL LININT1(XYINP(1,1,1),XYINP(1,1,2),NSIZ,NXYINP(1)
       ,MPTS,SPDMIN,SPDMAX)
  ELSE
     CALL EXSPL1(XYINP(1,1,1),XYINP(1,1,2),NSIZ,NXYINP(1)
       ,MPTS,SPDMIN,SPDMAX)
  END IF
  READ(9,*), N100
  DO 7, I=1, N100
     READ(9,*) XYINP(I,PARAM2,1), XYINP(I,PARAM2,2)
  CONTINUE
  DEMAND(PARAM2) = 100.0
  NDMAND = 2
  NXYINP(PARAM2) = N100
IF(INTFLG.EQ.-1) THEN
   CALL LININT1(XYINP(1,PARAM2,1),XYINP(1,PARAM2,2),NSIZ
6   ,NXYINP(PARAM2),MPTS,SPDMIN,SPDMAX)
ELSE
   CALL EXSPL1(XYINP(1,PARAM2,1),XYINP(1,PARAM2,2),NSIZ
6   ,NXYINP(PARAM2),MPTS,SPDMIN,SPDMAX)
END IF
READ(9,*), ANSWER
CALL UPCASE(ANSWER)
IF(ANSWER.EQ.'Y'.OR.ANSWER.EQ.'YES') THEN
   CLOSE(UNIT=9)
   GOTO 40
END IF
READ(9,*), NDMAND
NDMAND = NDMAND + 2
DO 9, J=2, (NDMAND-1)
   READ(9,*), DEMAND(J)
   READ(9,*), NDEM
DO 8, I=1, NDEM
   READ(9,*), XYINP(I,J,1), XYINP(I,J,2)
     CONTINUE
   NXYINP(J) = NDEM
ENDIF
9   CONTINUE
   CLOSE(UNIT=9)
   GOTO 90
ELSE IF(ANSWER.NE.'Y'.AND.ANSWER.NE.'YE'.AND.
   .ANSWER.NE.'YES') THEN
   GOTO 5
END IF

C -------------------
C LOCATE AXES' LIMITS
C -------------------
PRINT *
CALL COUA('Please enter maximum Y value.  ')
READ (UNIT=*, FMT=*) YG1
PRINT *
CALL COUA('Please enter X co-ordinate of axes crossing point.  ')
READ (UNIT=*, FMT=*) XG0
PRINT *
CALL COUA('Please enter Y co-ordinate of axes crossing point.  ')
READ (UNIT=*, FMT=*) YG0
PRINT *
CALL COUA('Please enter maximum X value.  ')
READ (UNIT=*, FMT=*) XG2

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CALL COU('Now point to maximum Y value on axis.')
10 CALL DIGINP(XS1, YS1, IREPLY)
   IF (IREPLY.EQ.0) THEN
      PRINT *, CHAR(7)
   ELSE
      GOTO 10
   END IF
PRINT *
CALL COU('Now point to axes crossing point.')
20 CALL DIGINP(XSO, YSO, IREPLY)
   IF (IREPLY.EQ.0) THEN
      PRINT *, CHAR(7)
   ELSE
      GOTO 20
   END IF
PRINT *
CALL COU('Now point to maximum X value on axis.')
30 CALL DIGINP(XS2, YS2, IREPLY)
   IF (IREPLY.EQ.0) THEN
      PRINT *, CHAR(7)
   ELSE
      GOTO 30
   END IF
PRINT *
C SETUP DATA TRANSFORMATION
C
ALFA1 = ATAN((XS1 - XSO) / (YS1 - YSO))
ALFA2 = ATAN((YSO - YS2) / (XS2 - XSO))
ALFA = (ALFA1 + ALFA2) / 2.0
CALL SUB1(XS1, YS1, XS1PP, YS1PP)
CALL SUB1(XS2, YS2, XS2PP, YS2PP)
C
C INPUT MINIMUM AND MAXIMUM SPEED VALUES
C
PRINT *
CALL COUA('Enter minimum speed value. ')
READ (UNIT=*, FMT=*) SPDMIN
PRINT *
CALL COUA('Enter maximum speed value. ')
READ (UNIT=*, FMT=*) SPDMAX
C
C INPUT MINIMUM AND MAXIMUM DEMAND LINES
C
PRINT *
CALL COU('Now digitize 0% DEMAND line with up to 500 points.')
NSIZ = PARAM1
CALL LININP(XYINP(1,1,1), XYINP(1,1,2), NSIZ, NXYINP(1))
DEMAND(1) = 0.0
MPTS = PARAM4
IF(INTFLG.EQ.-1) THEN
    CALL LININT1(XYINP(1,1,1), XYINP(1,1,2), NSIZ, NXYINP(1), MPTS &
                  , SPDMIN, SPDMAX)
ELSE
    CALL EXSPL1(XYINP(1,1,1), XYINP(1,1,2), NSIZ, NXYINP(1), MPTS &
                  , SPDMIN, SPDMAX)
END IF

PRINT *
CALL COU('Now digitize 100% DEMAND line with up to 500 points.')
CALL LININT1(XYINP(1,PARAM2,1), XYINP(1,PARAM2,2), NSIZ &
                  , NXYINP(PARAM2), MPTS, SPDMIN, SPDMAX)
ELSE
    CALL EXSPL1(XYINP(1,PARAM2,1), XYINP(1,PARAM2,2), NSIZ &
                  , NXYINP(PARAM2), MPTS, SPDMIN, SPDMAX)
END IF

DEMAND(PARAM2) = 100.0
NDMAND = 2

--------------------------------
CHOOSE FULL OR MINMAX DEFINITION
--------------------------------

PRINT *
CALL COU('You now have the choice of fully defining the map')
CALL COU('with the digitizer or only defining the MINIMUM and')
CALL COU('MAXIMUM DEMAND lines. If the latter is chosen the')
CALL COU('program will generate linearly spaced intermediate')
CALL COUA('DEMAND values. Do you want to use MINMAX? ')
READ(UNIT=*, FMT=*) ANSWER
CALL UPCASE(ANSWER)
IF(ANSWER.EQ.'Y'.OR,ANSWER.EQ.'YE'.OR,ANSWER.EQ.'YES') THEN
    NXPTS = PARAM4
    NYPTS = PARAM3
    DMDMIN = 0.0
    DMDMAX = 100.0
    DO 60, J=1, NXPTS
        SPD = XYINP(J,1,1)
        TRKMIN = XYINP(J,1,2)
        TRKMAX = XYINP(J,PARAM2,2)
        DO 50, I=1, NYPTS
            XYZDAT(I,J,1) = SPD
            SCALE = (I - 1.0) / (NYPTS - 1.0)
            XYZDAT(I,J,2) = TRKMIN + (TRKMAX - TRKMIN) * SCALE
            XYZDAT(I,J,3) = DMDMIN + (DMDMAX - DMDMIN) * SCALE
            50 CONTINUE
        60 CONTINUE
    GOTO 120
ELSE IF(ANSWER.NE.'N'.AND,ANSWER.NE.'NO') THEN
    GOTO 40
END IF
INPUT INTERMEDIATE DEMAND LINES

70 PRINT *
    CALL COUA('Enter % value of next intermediate DEMAND line. '
    READ(UNIT=*, FMT=*) DEMAND(NDMAND)
    CALL COU('Now digitize this DEMAND line with up to 500 points.
    NSIZ = PARAM1
    CALL LININP(XYINP(1,NDMAND,1), XYINP(1,NDMAND,2), NSIZ
        &      , NXYINP(NDMAND))
    MPTS = PARAM4
    IF(INTFLG.EQ.-1) THEN
        CALL LININT1(XYINP(1,NDMAND,1), XYINP(1,NDMAND,2), NSIZ
            &      , NXYINP(NDMAND), MPTS, SPDMIN, SPDMAX)
    ELSE
        CALL EXSPL1(XYINP(1,NDMAND,1), XYINP(1,NDMAND,2), NSIZ
            &      , NXYINP(NDMAND), MPTS, SPDMIN, SPDMAX)
    END IF
    NDMAND = NDMAND + 1
    IF(NDMAND.EQ.PARAM2) GOTO 90

80 PRINT *
    CALL COUA('Another DEMAND line? '
    READ(UNIT=*, FMT=*)) ANSWER
    IF(ANSWER.EQ.'Y'.OR,ANSWER.EQ.'YE'.OR,ANSWER.EQ.'YES') THEN
        GOTO 70
    ELSE IF(ANSWER.NE.'N'.AND,ANSWER.NE.'NO') THEN
        GOTO 80
    END IF
    90 NXPTS = PARAM4
    NSIZ = PARAM3
    MPTS = PARAM3
    DMDMIN = 0.0
    DMDMAX = 100.0
    DO 110, J=1, NXPTS
        DO 100, I=1, (NDMAND-1)
            XYZDAT(I,J,1) = XYINP(J,PARAM2,1)
            XYZDAT(I,J,2) = XYINP(J,PARAM2,2)
            XYZDAT(I,J,3) = DEMAND(I)
        CONTINUE
        XYZDAT(NDMAND,J,1) = XYINP(J,PARAM2,1)
        XYZDAT(NDMAND,J,2) = XYINP(J,PARAM2,2)
        XYZDAT(NDMAND,J,3) = DEMAND(PARAM2)
        IF(INTFLG.EQ.-1) THEN
            CALL LININT2(XYZDAT(1,J,3), XYZDAT(1,J,2), XYZDAT(1,J,1)
                &      , NSIZ, NDMAND, MPTS, DMDMIN, DMDMAX, 1)
        ELSE
            CALL EXSPL2(XYZDAT(1,J,3), XYZDAT(1,J,2), XYZDAT(1,J,1)
                &      , NSIZ, NDMAND, MPTS, DMDMIN, DMDMAX, 1)
        END IF
    CONTINUE
INPUT FILENAME PREFIX FOR TWO OUTPUT FILES AND OPEN THESE

PRINT *
CALL COUA('Enter prefix for output filenames. ')  
READ(UNIT=*, FMT=*) ANSWER  
L = LEN(ANSWER)  
FILE01 = ANSWER(1:1) /' .plt'
FILE02 = ANSWER(1:1) /' .gen'

OPEN(UNIT=NUNIT1, FILE=FILE01, FORM='FORMATTED'  
& CARRIAGE=.FALSE.)  
OPEN(UNIT=NUNIT2, FILE=FILE02, FORM='UNFORMATTED')

WRITE '.plt' DATA TO '.plt' FILE

WRITE polt limits to plot file
WRITE(UNIT=NUNIT1, FMT=1000)
NYPTS = PARAM3
NXPTS = PARAM4
WRITE(UNIT=NUNIT1, FMT=2000)
I = 1
DO 130, J=1, (NYPTS-1)
    WRITE(UNIT=NUNIT1, FMT=3000) XYZDAT(J,I,1), XYZDAT(J,I,2)
130 CONTINUE
J = NYPTS
DO 140, I=1, (NXPTS-1)
    WRITE(UNIT=NUNIT1, FMT=3000) XYZDAT(J,I,1), XYZDAT(J,I,2)
140 CONTINUE
I = NXPTS
DO 150, J=NYPTS, 2, -1
    WRITE(UNIT=NUNIT1, FMT=3000) XYZDAT(J,I,1), XYZDAT(J,I,2)
150 CONTINUE
J = 1
DO 160, I=NXPTS, 1, -1
    WRITE(UNIT=NUNIT1, FMT=3000) XYZDAT(J,I,1), XYZDAT(J,I,2)
160 CONTINUE

Write DEMAND lines to plot file
NDMAND = 0
MAJINT = 10
MAJSIZ = (PARAM3 - 1.0) / MAJINT
MININT = 2
MINSIZ = MAJSIZ / MININT
NINT = MAJINT * MININT
NXPTS = PARAM4
DO 180, MAJ=1, (MAJINT+1)
    DO 170, MIN=1, MININT
        IF((MAJ.EQ.MAJINT+1).AND.(MIN.GT.1)) GOTO 180
        NDMAND = NDMAND + 1
        IY = 1 + ((MAJ-1.0) * MAJSIZ) + ((MIN - 1.0) * MINSIZ)
        WRITE(UNIT=NUNIT1, FMT=2001) INT(ANINT(XYZDAT(IY,1,3)))
        DO 165, I=1, NXPTS
            WRITE(UNIT=NUNIT1, FMT=3000) XYZDAT(IY,1,1)
            & XYZDAT(IY,1,2)
        165 CONTINUE
        CONTINUE
    170 CONTINUE
180 CONTINUE

C Input plot title and write formatting information to plot file
PRINT *
    CALL COUA('Enter title for engine map. ', )
    READ(UNIT=*, FMT='(A)') TITLE
    WRITE(UNIT=NUNIT1, FMT=4000) WRITE(UNIT=NUNIT1, FMT=4001)
    L = LENG(TITLE)
    WRITE(UNIT=NUNIT1, FMT=4002) TITLE(1:L)
    WRITE(UNIT=NUNIT1, FMT=4003)
    WRITE(UNIT=NUNIT1, FMT=4004)
END IF

C Write plot file DEMAND line definitions
NCURVE = 1
DO 200, MAJ=1, MAJINT
    DO 190, MIN=1, MININT
        IF((MAJ.EQ.MAJINT).AND.(MIN.GT.1)) GOTO 200
        NCURVE = NCURVE + 1
        IF(MIN.EQ.1) THEN
            WRITE(UNIT=NUNIT1, FMT=5000) NCURVE
        ELSE
            WRITE(UNIT=NUNIT1, FMT=6000) NCURVE
        END IF
    190 CONTINUE
200 CONTINUE

C Write plot file DEMAND line definitions
WRITE(UNIT=NUNIT1, FMT=7000)
WRITE(UNIT=NUNIT1, FMT=8000)

C Write '.gen' DATA TO '.gen' FILE
C ---------------------------
WRITE(UNIT=NUNIT2) SPDMIN, SPDMAX
WRITE(UNIT=NUNIT2) (XYZDAT(I,J,3), J=1, NXPTS)
& (XYZDAT(PARAM3,K,3), K=1, NYPTS)
WRITE(UNIT=NUNIT2) (XYZDAT(I,J,2), I=1, NYPTS), J=1, NXPTS)

C -----------
C CLOSE FILES
C -----------
CLOSE(UNIT=NUNIT1)
CLOSE(UNIT=NUNIT2)

J-9
PLOT FILE FORMAT STATEMENTS

1000 FORMAT('case is ascii.'
&/ 'centimeter on.'
&/ 'generate a plot.'
&/ 'input data.')</p>

2000 FORMAT('"dd"')
2001 FORMAT('"d",I3""')
3000 FORMAT(E10.4,2X,E10.4)
4000 FORMAT('end of data.')

C Page setup
&/ 'page x 29.7 y 21.0.'
&/ 'page border off.')</p>

4001 FORMAT(
C Axes setup
&/ 'existence x on y on.'
&/ 'x mode normal.'
&/ 'y mode normal.'
&/ 'origin x 4.35 y 4.0.'
&/ 'length x 21.0 y 13.0.'
&/ 'minimum x 0.0.'
&/ 'no axis frame.'
&/ 'alphabet x standard y standard.'
&/ 'style x swiss-light y swiss-light.'
&/ 'height x 0.4 y 0.4.'
&/ 'x label text "Engine Speed (r/min)".'
&/ 'y label text "Brake Torque (Nm)".')
4002 FORMAT(
C Title setup
&/ 'title existence on.'
&/ 'title alphabet standard.'
&/ 'title style swiss-medium.'
&/ 'title units centimeters.'
&/ 'title height 0.7.'
&/ 'title box 4.0 25.7 0.8 1.8.'
&/ 'title text ",A,".')
4003 FORMAT(
C Legend setup
&/ 'legend existence no.')</p>

4004 FORMAT(
C Curve setup
&/ 'every curve symbol count 0.'
&/ 'every curve scattered no.'
&/ 'every curve color black.'
5000 FORMAT(
C Major demand line
&/ 'curve ',I3,¨ existence yes, texture solid.'
6000 FORMAT(
C Minor demand line
&/ 'curve ',I3,¨ existence yes, texture 9.'
7000 FORMAT(
C Engine map limits
&/ 'curve 1 existence yes, texture 11.'
8000 FORMAT('go.')
SUBROUTINE TO CALCULATE XSPP AND YSPP

SUBROUTINE SUB1(XS, YS, XSPP, YSPP)
IMPLICIT DOUBLE PRECISION (A-H, O-Z)
COMMON /DIGIT1/ ALFA, XSO, YSO, XS2PP, YS1PP, XGO, YGO, XG2, YG1

XSPP = (XS * COS(ALFA)) - (YS * SIN(ALFA))
YSPP = (XS * SIN(ALFA)) + (YS * COS(ALFA))
RETURN
END

SUBROUTINE TO CALCULATE XG AND YG

SUBROUTINE SUB2(XS, YS, XG, YG)
IMPLICIT DOUBLE PRECISION (A-H, O-Z)
COMMON /DIGIT1/ ALFA, XSO, YSO, XS2PP, YS1PP, XGO, YGO, XG2, YG1

CALL SUB1(XS, YS, XSPP, YSPP)
XG = XGO + ((XG2 - XGO) / XS2PP) * XSPP
YG = YGO + ((YG1 - YGO) / YS1PP) * YSPP
RETURN
END
SUBROUTINE LININP(X, Y, NSIZ, NPTS)
IMPLICIT DOUBLE PRECISION (A-H, O-Z)
DOUBLE PRECISION X(NSIZ), Y(NSIZ)
INTEGER NSIZ, NPTS

PRINT *
CALL COU('You may now indicate as many points as you wish. To')
CALL COU('stop digitizing, enter ***' on the keyboard.')

NPTS = 0
10 IF(NPTS.EQ.NSIZ) THEN
    IREPLY = 3
ELSE
    CALL DIGINP(XD, YD, IREPLY)
END IF

IF (IREPLY.EQ.0) THEN
    CALL SUB2(XD, YD, X(NPTS+1), Y(NPTS+1))
    PRINT *
    PRINT *, X(NPTS+1), Y(NPTS+1)
    NPTS = NPTS + 1
    PRINT *, CHAR(7)
    GOTO 10
ELSE IF (IREPLY.EQ.3) THEN
    PRINT *
    PRINT *, 'No. of points read in =', NPTS
    PRINT *, CHAR(?)
ELSE
    GOTO 10
END IF
RETURN
END
SUBROUTINE FOR DIGITIZER INPUT

SUBROUTINE DIGINP(X, Y, IREPLY)
IMPLICIT DOUBLE PRECISION (A-H, O-Z)
CHARACTER STRING*14
DOUBLE PRECISION X, Y
INTEGER IREPLY, IX, IY

READ(UNIT=*, FMT=*) STRING
IF(STRING.EQ.'**') THEN
   IREPLY = 3
   RETURN
END IF

STRING = STRING(2:7) // ' ' // STRING(8:13)
DECODE(STRING, 1000) X, Y
1000 FORMAT (V)
IREPLY = 0
RETURN
END
Appendix K

GENERATE.FORTRAN
GENERATE IS ONE OF A SUITE OF FOUR PROGRAMS DEVELOPED BY A L EM TAGE
FOR DEFINING ENGINE MAPS AND GENERATING LOOK-UP TABLES FOR ENGINE
MANAGEMENT MICROPROCESSOR CONTROL SYSTEMS.

GENERATE COMBINES DATA FROM A FILE ('.gen') DEFINING THE SHAPE
OF THE ENGINE'S DRIVER/CONTROL DEMAND LINES WITH FILES ('.cal') DEFINING
THE ENGINE'S CONTROL VARIABLES TO CREATE AN ASSEMBLER FILE ('.lut')
DEFINING A LOOK-UP TABLE FOR USE IN A MICROPROCESSOR BASED CONTROL
SYSTEM. THE TASK IS DEFINED BY AN INPUT FILE ('.def') WHOSE FORMAT IS
AS FOLLOWS:

LINE 1  - filename of driver/control demand file ('.gen')
LINE 2  - filename for output assembler file ('.mac')
LINE 3  - size of look-up table in speed direction
LINE 4  - size of look-up table in driver/control demand
direction
LINE 5  - number (N1) of header lines for assembler file (N1>-1)
N1 LINES - containing the header text
1 LINE  - number (N2) of output format groups (N2>0)
N2 GROUPS- formatted as follows:
1 LINE  - containing text of assembler storage
directive
1 LINE  - number (N3) of control variables
associated with this directive (N3>0)
2*N3 LINES- containing alternately filenames ('.cal')
and scale factors defining the control
variables (in correct order)
1 LINE  - number (N4) of footer lines for assembler file
(N4>-1)
N4 LINES - containing the footer text

FOLLOWING IS AN EXAMPLE OF A '.def' FILE:

dummy.gen
dummy.lut
32
32
2
aseg
100H
2
db
ig.t.cal
1.0
pli.cal
1.0
dw
3
ta.cal
4444444
eqi.cal
1000.0
hcf.cal
STORAGE DECLARATIONS

IMPLICIT DOUBLE PRECISION (A-H, O-Z)
INTEGER PARAM1, PARAM2, PARAM3, NUNIT1, NUNIT2, NUNIT3
PARAMETER (PARAM1=51, PARAM2=51, PARAM3=10)
PARAMETER (NUNIT1=6, NUNIT2=7, NUNIT3=8)
PARAM1 = GRID SIZE IN 'Y' DIRECTION
PARAM2 = GRID SIZE IN 'X' DIRECTION
CHARACTER FILDEF*50, FILGEN*50, FILCAL(PARAM3)*50, FILLUT*50
& STRING*50, LINE(PARAM3)*50
COMMON /GENER1/ SPDMIN, SPDMAX, GENDAT(PARAM1,PARAM2)
& DDMIN(PARAM2), DDMAX(PARAM2)
COMMON /GENER2/ TRKMIN(PARAM2,PARAM3), TRKMAX(PARAM2,PARAM3)
COMMON /GENER3/ CALDAT(PARAM1,PARAM2,PARAM3)
COMMON /GENER4/ SCALE(PARAM3), NXSIZE, NYSIZE, NLINE(PARAM3)

GET '.def' FILENAME AND OPEN IT

PRINT *
CALL COUA('Enter DEFINE filename. '
READ(UNIT=*, FMT=*) FILDEF
L = LENG(FILDEF)
IF(FILDEF(L-3:L).NE.' .def') FILDEF = FILDEF // '.def'
OPEN(UNIT=NUNIT1, FILE=FILDEF, FORM='FORMATTED'
& , CARRIAGE=.FALSE.)

GET '.gen' FILENAME, OPEN IT, READ IT AND CLOSE IT

READ(UNIT=NUNIT1, FMT=*) FILGEN
L = LENG(FILGEN)
IF(FILGEN(L-3:L).NE.' .gen') FILGEN = FILGEN // '.gen'
OPEN(UNIT=NUNIT2, FILE=FILGEN, FORM='UNFORMATTED')
READ(UNIT=NUNIT2) SPDMIN, SPDMAX
NXPTS = PARAM2
READ(UNIT=NUNIT2) (DMIN(J), J=1, NXPTS)
& , (DMMAX(K), K=1, NXPTS)
NYPTS = PARAM1
READ(UNIT=NUNIT2) ((GENDAT(I,J), I=1, NYPTS), J=1, NXPTS)
CLOSE(UNIT=NUNIT2)

GET '.mac' FILENAME AND OPEN IT

READ(UNIT=NUNIT1, FMT=*) FILLUT
L = LENG(FILLUT)
IF(FILLUT(L-3:L).NE.' .lut') FILLUT = FILLUT // '.lut'
OPEN(UNIT=NUNIT3, FILE=FILLUT, FORM='FORMATTED'
& , CARRIAGE=.FALSE.)
INPUT SIZE OF LOOK-UP TABLE

READ(UNIT=UNIT1, FMT=*) NXSIZE
READ(UNIT=UNIT1, FMT=*) NYSIZE

INPUT AND WRITE OUTPUT FILE HEADER

READ(UNIT=UNIT1, FMT=*) N
IF (N.GT.0) THEN
   DO 10, I=1, N
      READ(UNIT=UNIT1, FMT='(A)') STRING
      WRITE(UNIT=UNIT3, FMT='(A)') STRING
   CONTINUE
END IF

INPUT AND WRITE OUTPUT FILE HEADER

READ(UNIT=UNIT1, FMT=*) MGROUP
NFILE = 0
DO 30, J=1, MGROUP
   READ(UNIT=UNIT1, FMT='(A)') LINE(J)
   READ(UNIT=UNIT1, FMT=*) NLINE(J)
   N = NLINE(J)
   DO 20, I=1, N
      NFILE = NFILE + 1
      READ(UNIT=UNIT1, FMT='(A)') SCALEINFILE
      READ(UNIT=UNIT1, FMT=*) STRING
   CONTINUE
30 CONTINUE

OPEN '.cal' FILES, CHECK SPEED LIMITS, READ DATA AND CLOSE

DO 40, K=1, NFILE
   L = LEN(FILCAL(K))
   IF (FILCAL(K)(L-3:L).NE.' .cal') THEN
      FILCAL(K) = FILCAL(K) // '.cal'
   END IF
   OPEN(UNIT=UNIT2, FILE=FILCAL(K), FORM='UNFORMATTED')
   READ(UNIT=UNIT2) DUMMY1, DUMMY2
   IF (DUMMY1.NE.SPDMIN .OR. DUMMY2.NE.SPDMAX) THEN
      PRINT *
      CALL COU('***FATAL***')
      CALL COUA('***ERROR*** Speed limits do not match on ')
      WRITE(UNIT=*, FMT='(A)') FILCAL(K)
      CALL COU('PROGRAM HALTED!')
      STOP
   END IF
   NXPTS = PARAM2
   READ(UNIT=UNIT2) (TRKMIN(J,K), J=1, NXPTS)
   (TRKMAX(J1,K), J1=1, NXPTS)
   NYPTS = PARAM1
   READ(UNIT=UNIT2) ((CALDAT(I,J,K), I=1, NYPTS), J=1, NXPTS)
   CLOSE(UNIT=UNIT2)
40 CONTINUE
GENERATE LOOK-UP TABLE

DO 80, I=1, NYSIZE
  Calculate Driver Demand coordinate
  DDCORD = 1.0 + (PARAM1 - 1.0) * (I - 1.0) / (NYSIZE - 1.0)
  INTDDC = INT(DDCORD)
  FRCDDC = DDCORD - INTDDC
DO 70, J=1, NXSIZE
  Calculate Engine Speed coordinate
  ESCORD = 1.0 + (PARAM2 - 1.0) * (J - 1.0) / (NXSIZE - 1.0)
  INTESEC = INT(ESCORD)
  FRCESEC = ESCORD - INTESEC
  Use these coordinates to find Brake Torque from '.gen' data
  BT = GENDAT(INTDDC, INTESEC)
  IXMIN = INTESEC
  IF(INTDDC.LT.PARAM1) THEN 
    BBT = GENDAT(INTDDC+1, INTESEC)
    IXMAX = INTESEC + 1 
  ELSE
    IXMAX = IXMIN
  END IF
  IF(INTDDC.LT.PARAM1.AND.INTDDC.LT.PARAM2) THEN 
    BBT = GENDAT(INTDDC+1, INTESEC+1)
  ELSE
    BBT = ABT
  END IF
  IF(INTDDC.LT.PARAM1.AND.INTDDC.LT.PARAM2.AND.INTDDC.LT.PARAM1) 
  BT = GENDAT(INTDDC+1, INTESEC+1)
  IF(INTDDC.GE.PARAM1) BBT = ABT
  IF(INTDDC.GE.PARAM1) DBT = BBT
  BT = ((ABT * (1.0 - FRCDDC)) + (BBT * FRCESEC))
     + ((CBT * (1.0 - FRCESEC)) + (DBT * FRCESEC))
     * (1.0 - FRCDDC)
  Calculate and write values of control variables
  NFILE = 0
DO 60, K=1, MGROUP
  LLINE = LENG(LINE(K))
  STRING = LINE(K)(1:LLINE)
  CALL UPCASE(STRING)
  IF(MGROUP.GT.1.OR.IB.EQ.1) THEN 
    WRITE(UNIT=NUNIT3, FMT='(A,$)')
    LINE(K)(1:LLINE) // ' '
  END IF
  N = NLINE(K)
DO 50, L=1, N
  NFILE = NFILE + 1
  BTMIN = TRKMIN(IXMIN, NFILE) * (1.0 - FRCDDC)
     + TRKMIN(IXMAX, NFILE) * FRCDDC
  BTMAX = TRKMAX(IXMIN, NFILE) * (1.0 - FRCESEC)
     + TRKMAX(IXMAX, NFILE) * FRCESEC
  BTCORD = 1 + (PARAM1-1.0) * (BT-BTMIN)
     / (BTMAX-BTMIN)
K-5
IF(ABS(BTCORD-1.0) .LT. 0.0001) BTCORD = 1.0
IF(ABS(PARAM2-BTCORD) .LT. 0.0001) BTCORD = PARAM2
INTBTC = INT(BTCORD)
FRCBTC = BTCORD - INTBTC
IF((INTBTC.GT.PARAM2).OR.(INTBTC.LT.1)) THEN
PRINT *
  CALL COU('***FATAL***')
  CALL COUA('***ERROR*** Torque limits for')
  CALL COU('...gen')
  CALL COUA('file are greater than those for')
  WRITE(UNIT=*, FMT=('(A)')) FILCAL(NFILE)
  CALL COU('PROGRAM HALTED!')
  STOP
END IF
AVR = CALDAT(INTBTC,INTESC,NFILE)
IF(INTESC.LT.PARAM2) THEN
  BVR = CALDAT(INTBTC,INTESC+1,NFILE)
ELSE
  BVR = AVR
END IF
IF(INTBTC.LT.PARAM1) THEN
  CVR = CALDAT(INTBTC+1,INTESC,NFILE)
ELSE
  CVR = AVR
END IF
IF(INTESC.LT.PARAM2.AND.INTBTC.LT.PARAM1) THEN
  DVR = CALDAT(INTBTC,INTESC+1,NFILE)
ELSE
  DVR = BVR
END IF
VAR = (((AVR * (1.0 - FRCESC)) + (BVR * FRCESC))
  * (1.0 - FRCBTC))
  + (((CVR * (1.0 - FRCESC)) + (DVR * FRCESC))
  * FRCBTC)
VAR = VAR * SCALE(NFILE)
IF(L.LT.N.OR.(MGROUP.EQ.1.AND.I8.LT.8)) THEN
  IF(STRING.EQ.'DB') THEN
    WRITE(UNIT=NUNIT3, FMT=(/I3,'',I5,'',$)) INT(VAR)
  ELSE
    WRITE(UNIT=NUNIT3, FMT=(/I5,'',I5,$)) INT(VAR)
  END IF
  I8 = I8 + 1
ELSE
  IF(STRING.EQ.'DB') THEN
    WRITE(UNIT=NUNIT3, FMT=(/I3)) INT(VAR)
  ELSE
    WRITE(UNIT=NUNIT3, FMT=(/I5)) INT(VAR)
  END IF
  I6 = 1
END IF
50 CONTINUE
60 CONTINUE
70 CONTINUE
80 CONTINUE

K - 6
READ(UNIT=NUNIT1, FMT=*) N
IF (N.GT.0) THEN
  DO 90, I=1, N
    READ(UNIT=NUNIT1, FMT=*) STRING
    WRITE(UNIT=NUNIT3, FMT='(A)') STRING
  CONTINUE
END IF

CLOSE '.lut' AND '.def' FILES

STOP
END
Appendix L

Ford Transit Crew Bus Specifications
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kerb Weight (TOTAL)</td>
<td>1770 kg</td>
</tr>
<tr>
<td>Kerb Weight (FRONT)</td>
<td>870 kg</td>
</tr>
<tr>
<td>Kerb Weight (REAR)</td>
<td>900 kg</td>
</tr>
<tr>
<td>( A_D )</td>
<td>0.0117</td>
</tr>
<tr>
<td>( B_D )</td>
<td>0.0002</td>
</tr>
<tr>
<td>( C_D )</td>
<td>0.480</td>
</tr>
<tr>
<td>( A )</td>
<td>3.47 m²</td>
</tr>
<tr>
<td>( r_r )</td>
<td>0.314 m</td>
</tr>
<tr>
<td>( I_w )</td>
<td>6.0 kgm² (i.e. 6 x 1.0 kgm²)</td>
</tr>
<tr>
<td>( I_p )</td>
<td>0.0 kgm²</td>
</tr>
<tr>
<td>( I_e )</td>
<td>0.225 kgm²</td>
</tr>
<tr>
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Petrol

Hydrogen

1200 r/min
MBT Spark Timing
CR = 8:1
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START

INITIALIZE

INITIATE INTERRUPT DRIVEN UPDATE CYCLE

CHECK STATUS

MODE SELECTION

UPDATE DISPLAY AND EXECUTE KEYBOARD COMMANDS
Figure 3.26 Main Update Cycle

START

SAMPLE SPEED DEMAND AND AIR FLOW

INITIATE MATHEMATICS INTERRUPT PROCESS

SAMPLE REMAINING A/D CHANNELS

STOP

START

CALCULATE POINTERS TO LOOK-UP-TABLE

OBTAIN LOOK-UP-TABLE VALUES

INTERPOLATE AND CALCULATE OUTPUT VALUES

SET OUTPUTS

CALCULATE DISPLAY VALUES

INITIATE A/D INTERRUPT PROCESS

STOP
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SYMBOL KEY

- $N = 1000 \text{ r/min}$
- $N = 2000 \text{ r/min}$
- $N = 3000 \text{ r/min}$
- $N = 4000 \text{ r/min}$
- $N = 5000 \text{ r/min}$
- $N = 6000 \text{ r/min}$
- Proportional
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HYDROGEN/PETROL CREW BUS