PERFORMANCE OF A DUAL-FUEL PRECHAMBER
DIESEL ENGINE WITH NATURAL GAS

BY

SEAHO SONG

B.Sc., The University of British Columbia, 1982

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF
THE REQUIREMENTS FOR THE DEGREE OF
MASTER OF APPLIED SCIENCE

in
THE FACULTY OF GRADUATE STUDIES
THE DEPARTMENT OF MECHANICAL ENGINEERING

We accept this thesis as conforming
to the required standard

THE UNIVERSITY OF BRITISH COLUMBIA
May 1984
©Seaho Song, 1984
In presenting this thesis in partial fulfilment of the requirements for an advanced degree at the University of British Columbia, I agree that the Library shall make it freely available for reference and study. I further agree that permission for extensive copying of this thesis for scholarly purposes may be granted by the Head of my Department or by his or her representatives. It is understood that copying or publication of this thesis for financial gain shall not be allowed without my written permission.

Department of Mechanical Engineering

The University of British Columbia
2324 Main Mall
Vancouver, Canada
V6T 1W5

Date: June 1984
ABSTRACT

The feasibility of dual-fuel operation with natural gas in a prechamber diesel engine was studied with special emphasis on fuel consumption and cylinder pressure development. The effects of air restriction, pilot diesel flow rate and injection timing were also studied. Dual-fuel operation showed poor part-load fuel consumption; near full load the fuel consumption was close to that of straight diesel operation. In the absence of injection timing adjustment the maximum power output of dual-fuel operation was severely limited by the maximum cylinder pressure. Retarding the injection timing was effective in reducing the maximum cylinder pressure to a safe level. The analysis of apparent energy release indicates the differences in combustion mechanism between auto-ignition of diesel fuel in straight diesel operation and propagation of flame fronts in dual-fuel operation.
Table of Contents

Abstract .............................................................. ii
List of Tables ......................................................... iv
List of Figures ......................................................... v
Acknowledgements ..................................................... viii
Nomenclature .......................................................... ix
I. INTRODUCTION ...................................................... 1
  1.1 Background .................................................... 1
  1.2 Present Study .................................................. 7
II. REVIEW OF LITERATURE ............................................ 10
  2.1 History of Dual-Fuel Diesel Engine ......................... 10
  2.2 Review of Research ........................................... 12
III. APPARATUS AND INSTRUMENTATION ............................... 26
  3.1 Engine and Test Bed ........................................... 26
  3.2 Instrumentation ............................................... 38
  3.3 Fuel ............................................................ 43
  3.4 Data Process .................................................. 45
IV. EXPERIMENTAL RESULTS ........................................... 47
  4.1 Fuel Consumption ............................................. 47
    4.1.1 Fuel Consumption with Unmodified Engine ............... 47
    4.1.2 Effect of Restricting Intake Air ......................... 55
    4.1.3 Effect of Varying Injection Timing ....................... 56
  4.2 Cylinder Pressure ............................................. 63
    4.2.1 Cylinder Pressure in Unmodified Engine ................. 63
    4.2.2 Effect of Restricting Intake Air ......................... 74
    4.2.3 Effect of Varying Injection Timing ....................... 79
V. ANALYSIS OF APPARENT ENERGY RELEASE .......................... 84
  5.1 General ....................................................... 84
  5.2 Method of Calculation ....................................... 85
    5.2.1 Definitions, Equations, and Assumptions ............... 85
    5.2.2 Computation Procedure .................................... 98
  5.3 Analysis ...................................................... 107
    5.3.1 Operation with Unmodified Engine ....................... 107
    5.3.2 Effect of Restricting Intake Air ......................... 124
    5.3.3 Effect of Varying Injection Timing ....................... 130
VI. CONCLUSIONS AND RECOMMENDATIONS ............................. 135
  6.1 Conclusions .................................................. 135
  6.2 Recommendations ............................................ 138

BIBLIOGRAPHY ......................................................... 139
APPENDIX A - CALIBRATION CURVES ................................ 142
APPENDIX B - COMPUTATION OF INDICATED MEAN EFFECTIVE PRESSURE ......................................................... 146
APPENDIX C - COMPUTER PROGRAM FOR DATA ACQUISITION ........ 148
APPENDIX D - COMPUTER PROGRAM FOR DATA PROCESS ............... 157
APPENDIX E - COMPUTER PROGRAM FOR APPARENT ENERGY RELEASE ......................................................... 162
List of Tables

2.1 Summary of Past Experimental Work ....................... 13
3.1 Engine Specification ....................................... 29
3.2 Typical composition of the Natural Gas Used ........ 44
3.3 Typical Output of Computer Program for Data Processing ...................................................... 46
5.1 Comparison of Actual and Computed Fuel Energy Consumed ......................................................... 105
5.2 Effect of Intake Air Restriction on Mixture Temperature at Top Dead Center .............................. 126
List of Figures

1.1 Combustion Chambers of Direct-Injection and Prechamber Engines ........................................... 5
2.1 Effect of Gas-Air Mixture Strength on Ignition delay ................................................................. 17
2.2 Typical Pressure-Time Trace of Non-Knocking and Knocking Operation ........................................ 21
2.3 Variation of Power Output with the Overall Mixture Strength for Different Intake Temperatures ........ 22
2.4 Typical Thermal Efficiencies of Dual-Fuel and Straight Diesel Operation .................................... 25
3.1 Apparatus and Instrumentation .............................................................. 27
3.2 Flow of Air, Fuel, and Exhaust Gas .............................................................. 28
3.3 Shape of Combustion Chambers .............................................................. 30
3.4 Sleeve Metering Fuel System .............................................................. 32
3.5 Fuel Injection Pump and Housing .............................................................. 32
3.6 Sequence of Injection Events .............................................................. 33
3.7 Governor Components of Sleeve Metering .............................................................. 33
3.8 Fuel Injection Nozzle .............................................................. 34
3.9 Turbocharger Cutaway View .............................................................. 36
3.10 Gas Mixer .............................................................. 37
3.11 Mounting of Cylinder Pressure Transducer .............................................................. 39
4.1 Effect of Pilot Diesel Flow Rate on Brake Thermal Efficiency .............................................................. 48
4.2 Fuel Consumption at Idling Operation .............................................................. 50
4.3 Effect of Pilot Diesel Flow Rate on Indicated Thermal Efficiency .............................................................. 52
4.4 Comparison of Brake Thermal efficiencies for Dual-Fuel and Straight Diesel Operation ............ 54
4.5 Effect of Intake Air Restriction on Brake Thermal Efficiency .............................................................. 57
4.6 Typical Cylinder Pressure Trace and Apparent Point of Ignition Start .............................................................. 58
4.7 Apparent Point of Ignition Start at Various Loads .............................................................. 59
4.8 Effect of Varying Injection Timing on Brake Thermal Efficiency .............................................................. 61
4.9 P-V Diagram of Straight Diesel Operation ............. 64
4.10 Ln P-V Diagram of Straight Diesel Operation ............. 66
4.11 Comparison of P-V Diagrams for Dual-Fuel and Straight Diesel Operation .............................................................. 67
4.12 Comparison of Ln P-V Diagrams for Dual-Fuel and Straight Diesel Operation .............................................................. 69
4.13 Comparison of Maximum Cylinder Pressures for Dual-Fuel and Straight Diesel Operation ............ 70
4.14 Maximum Cylinder Pressure at Various Loads .............................................................. 71
<table>
<thead>
<tr>
<th>Section</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.15</td>
<td>Comparison of Maximum Rate of Cylinder Pressure Rise for Dual-Fuel and Straight Diesel Operation</td>
<td>72</td>
</tr>
<tr>
<td>4.16</td>
<td>Maximum Rate of Cylinder Pressure Rise at Various Loads</td>
<td>73</td>
</tr>
<tr>
<td>4.17</td>
<td>Effect of Intake Air Restriction on Maximum Cylinder Pressure</td>
<td>75</td>
</tr>
<tr>
<td>4.18</td>
<td>Effect of Intake Air Restriction Pressure Prior to Combustion</td>
<td>76</td>
</tr>
<tr>
<td>4.19</td>
<td>Effect of Intake Air Restriction on Maximum Rate of Cylinder Pressure Rise</td>
<td>78</td>
</tr>
<tr>
<td>4.20</td>
<td>Effect of Varying Injection Timing on Maximum Cylinder Pressure and Rate of Pressure Rise</td>
<td>80</td>
</tr>
<tr>
<td>4.21</td>
<td>Effect of Varying Injection Timing on Pressure Prior to Combustion and Point of Ignition Start</td>
<td>81</td>
</tr>
<tr>
<td>4.22</td>
<td>Effect of Varying Injection Timing on P-V Diagram</td>
<td>82</td>
</tr>
<tr>
<td>5.1</td>
<td>Control Volume for Apparent Energy Release Analysis</td>
<td>86</td>
</tr>
<tr>
<td>5.2</td>
<td>Apparent Heat Transfer Rate and Heat Transfer Model</td>
<td>91</td>
</tr>
<tr>
<td>5.3</td>
<td>Effect of Heat Transfer Model on Apparent Rate of Energy Release</td>
<td>93</td>
</tr>
<tr>
<td>5.4</td>
<td>Effect of Equilibrium Calculation on Apparent Rate of Energy Release</td>
<td>97</td>
</tr>
<tr>
<td>5.5</td>
<td>Effect of Smoothing Pressure Data on Apparent Rate of Energy Release</td>
<td>101</td>
</tr>
<tr>
<td>5.6</td>
<td>Flowchart of Computer Program for Apparent Energy Release</td>
<td>103</td>
</tr>
<tr>
<td>5.7</td>
<td>Typical Output of Computer Program for Apparent Energy Release Analysis</td>
<td>104</td>
</tr>
<tr>
<td>5.8</td>
<td>Rate of Energy Release of Straight Diesel Operation at Various Loads</td>
<td>108</td>
</tr>
<tr>
<td>5.9</td>
<td>Cumulative Energy Release of Straight Diesel Operation at Various Loads</td>
<td>109</td>
</tr>
<tr>
<td>5.10</td>
<td>Effect of Air-Fuel Ratio on Maximum Rate of Energy Release in Straight Diesel Operation</td>
<td>111</td>
</tr>
<tr>
<td>5.11</td>
<td>Rate of Energy Release of Dual-Fuel Operation at Various Loads</td>
<td>112</td>
</tr>
<tr>
<td>5.12</td>
<td>Effect of Gas-Air Mixture Strength on Maximum Rate of Energy Release in Dual-Fuel Operation</td>
<td>113</td>
</tr>
<tr>
<td>5.13</td>
<td>Cumulative Energy Release of Dual-Fuel Operation at Various Loads</td>
<td>115</td>
</tr>
<tr>
<td>5.14</td>
<td>Comparison of Rate of Energy Release for Straight Diesel and Dual-Fuel Operation</td>
<td>116</td>
</tr>
<tr>
<td>5.15</td>
<td>Comparison of Cumulative Energy Release for Straight Diesel and Dual-Fuel Operation</td>
<td>117</td>
</tr>
<tr>
<td>5.16</td>
<td>Rate of Energy Release of Dual-Fuel Operation at Various Pilot Diesel Flow Rates</td>
<td>118</td>
</tr>
<tr>
<td>Section</td>
<td>Title</td>
<td>Page</td>
</tr>
<tr>
<td>---------</td>
<td>----------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>5.17</td>
<td>Cumulative Energy Release of Dual-Fuel Operation at Various Pilot Diesel Flow Rates</td>
<td>120</td>
</tr>
<tr>
<td>5.18</td>
<td>Fraction of Fuel Burnt in Low Load Dual-Fuel Operation</td>
<td>121</td>
</tr>
<tr>
<td>5.19</td>
<td>Rate of Energy Release of Dual-Fuel Operation at Various Pilot Diesel Flow Rates</td>
<td>122</td>
</tr>
<tr>
<td>5.20</td>
<td>Cumulative Energy Release of Dual-Fuel Operation at Various Pilot Diesel Flow Rates</td>
<td>123</td>
</tr>
<tr>
<td>5.21</td>
<td>Effect of Restricting Intake Air on Rate of Energy Release</td>
<td>125</td>
</tr>
<tr>
<td>5.22</td>
<td>Effect of Restricting Intake Air on Cumulative Energy Release</td>
<td>127</td>
</tr>
<tr>
<td>5.23</td>
<td>Effect of Restricting Intake Air on Rate of Energy Release</td>
<td>128</td>
</tr>
<tr>
<td>5.24</td>
<td>Effect of Restricting Intake Air on Cumulative Energy Release</td>
<td>129</td>
</tr>
<tr>
<td>5.25</td>
<td>Effect of Advancing Injection Timing on Rate of Energy Release</td>
<td>131</td>
</tr>
<tr>
<td>5.26</td>
<td>Effect of Advancing Injection Timing on Cumulative Energy Release</td>
<td>132</td>
</tr>
<tr>
<td>5.27</td>
<td>Effect of Retarding Injection Timing on Rate of Energy Release</td>
<td>133</td>
</tr>
<tr>
<td>5.28</td>
<td>Effect of Retarding Injection Timing on Cumulative Energy Release</td>
<td>134</td>
</tr>
</tbody>
</table>
Acknowledgement

The author wishes to acknowledge a sincere gratitude to Dr. P.G. Hill for his advice and encouragement. Numerous discussions with Dr. Roger Milane have and will remain valuable to the author. Thanks are also due to Messrs Stan Mah, Shu Osaka, and John Hoar for technical advice and assistance in setting up the equipments. Stan Mah was responsible for the installation of the engine, and for development of the instrumentation before the project commenced. Provision of technical information and assistance by Mr. Jim Bare of Finnings Tractors is greatly appreciated. Further thanks are due to the members of the thesis committee, Dr. B. Ahlborn, Dr. B. Evans and Dr. E.G. Hauptmann.

This work was financially supported by the Federal Department of Energy Mines and Resources.
# Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>area</td>
<td>m²</td>
</tr>
<tr>
<td>ATDC</td>
<td>after top dead center</td>
<td></td>
</tr>
<tr>
<td>BMEP</td>
<td>brake mean effective pressure</td>
<td>kPa</td>
</tr>
<tr>
<td>BTDC</td>
<td>before top dead center</td>
<td></td>
</tr>
<tr>
<td>CA</td>
<td>crank angle</td>
<td></td>
</tr>
<tr>
<td>CE</td>
<td>chemical energy</td>
<td>kJ</td>
</tr>
<tr>
<td>D</td>
<td>bore</td>
<td>mm</td>
</tr>
<tr>
<td>E</td>
<td>energy</td>
<td>kJ</td>
</tr>
<tr>
<td>e_c</td>
<td>internal energy of combustion</td>
<td>kJ/kg</td>
</tr>
<tr>
<td>K</td>
<td>equilibrium constant</td>
<td></td>
</tr>
<tr>
<td>k</td>
<td>thermal conductivity</td>
<td>kW/(m·°C)</td>
</tr>
<tr>
<td>m_f</td>
<td>mass of fuel</td>
<td>kg</td>
</tr>
<tr>
<td>N</td>
<td>number of moles</td>
<td>kmole</td>
</tr>
<tr>
<td>P</td>
<td>pressure</td>
<td>MPa</td>
</tr>
<tr>
<td>Q</td>
<td>heat transfer</td>
<td>kJ</td>
</tr>
<tr>
<td>q</td>
<td>heat transfer rate</td>
<td>kW</td>
</tr>
<tr>
<td>R</td>
<td>Reynolds Number</td>
<td></td>
</tr>
<tr>
<td>T</td>
<td>temperature</td>
<td>°C</td>
</tr>
<tr>
<td>U</td>
<td>internal energy</td>
<td>kJ</td>
</tr>
<tr>
<td>V</td>
<td>volume</td>
<td>m³</td>
</tr>
<tr>
<td>V̅</td>
<td>mean velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>W</td>
<td>work</td>
<td>kJ</td>
</tr>
<tr>
<td>V̅_e</td>
<td>equilibrium composition</td>
<td></td>
</tr>
<tr>
<td>µ</td>
<td>viscosity</td>
<td>kg/m·s</td>
</tr>
<tr>
<td>ρ</td>
<td>density</td>
<td>kg/m³</td>
</tr>
<tr>
<td>φ</td>
<td>equivalence ratio</td>
<td></td>
</tr>
<tr>
<td>λ</td>
<td>inverse of equivalence ratio</td>
<td></td>
</tr>
</tbody>
</table>

**Subscripts:**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>g</td>
<td>gas</td>
</tr>
<tr>
<td>i</td>
<td>state</td>
</tr>
</tbody>
</table>
CH.1 Introduction

1.1 Background

Dual-Fuel Diesel Engines with Natural Gas

Dual-fuel diesel engines are here defined as those which burn either gaseous fuels or diesel, or both at the same time. The mode of operation is defined as straight diesel if only diesel fuel is used, and dual-fuel if two fuels are used at the same time. In dual-fuel operation the gaseous fuel is mixed with air at lean gas-air ratio and the mixture is then compressed during the compression stroke. Near the end of the compression stroke, diesel fuel is injected and auto-ignites, initiating the combustion of the gas-air mixture. Because of its function to initiate the combustion the diesel in dual-fuel operation is often referred to as pilot diesel. The changeover of the mode of operation, either from dual-fuel to straight diesel or straight diesel to dual-fuel can take place while the engine operates.

Combustion characteristics of dual-fuel operation differ from those of straight diesel operation. In diesel operation the combustion takes places within small zones where the fuel-air ratio is suitable for combustion. As a stream of diesel fuel is injected into the cylinder, it is mixed with air to be disintegrated into fine droplets which in turn vapourize and
auto-ignite due to the high temperature of the compressed air. The time period during which liquid diesel is mixed with air and vapourized is referred to as 'physical delay' and the time taken from then to the point just prior to ignition is referred to as 'chemical delay'. These two delay periods are combined and commonly termed 'ignition delay'. Combustion in dual-fuel operation, in contrast, occurs in a nearly homogeneous fuel-air mixture. During the intake stroke, a nearly uniform mixture of gas and air is drawn into the cylinder, then compressed by piston movement to high temperature and pressure but not high enough to elicit auto-ignition. A small amount of diesel fuel is injected into the homogeneous gas-air mixture near the end of the compression stroke. The injected pilot diesel subsequently goes through the ignition delay before it disintegrates into diesel vapour to initiate flame fronts which propagate through the gas-air mixture. The propagation of flame fronts is largely responsible for subsequent combustion of the remaining gas-air mixture. It is in this regard that the combustion process of dual-fuel operation differs from that of straight diesel operation. The combustion in straight diesel operation is largely due to auto-ignition of diesel fuel, whereas that of dual-fuel operation depends heavily on both the auto-ignition characteristics of pilot diesel and the propagation of flame fronts.

The gaseous fuel we are concerned with here is natural gas. Natural gas is available in most localities, and is in many places more abundant than other kinds of fuel. The temperature at which the natural gas auto-ignites is higher than that of
other available gaseous fuels. Because of the high auto-ignition temperature the gas-air mixture can be compressed to high compression ratio without auto-ignition. This, with the low cost of natural gas, makes dual-fuel diesel operation with natural gas an attractive means for power production, especially in places where the gas supply may possibly be interrupted.

Dual-fuel diesel engines with natural gas have been employed extensively in power generating stations where engines operate constantly at near full load. The fuel consumption rate of dual-fuel diesel operation with natural gas at full load has been shown to be as good as or sometimes better than that of straight diesel operation. Typically, the amount of pilot diesel used in the past has consisted of less than 10% of total energy input. In applications such as pipe line industries, where the required power is nearly constant, dual-fuel diesel operation has been shown to be satisfactory.

The maximum power of dual-fuel diesel operation is limited by the occurrence of knock with its high rate of cylinder pressure rise. Knock in dual-fuel engines is believed to be the same type as in spark-ignition engine. If the dual-fuel engine has to operate over a range of load, the economic advantage of dual-fuel diesel operation at full load may be offset by poor combustion characteristics at low load, which result in poor fuel consumption rate. At low load significant amounts of natural gas survive combustion and escape through the exhaust because of lean gas-air mixture strength. Low load fuel consumption rate can be improved by restricting intake air,
which effectively increases gas-air mixture strength, or by increasing the pilot diesel flow rate. Other possible methods which improve low load fuel consumption rate are preheating the intake charge and advancing the injection timing of pilot diesel. Preheating the intake charge results in higher mixture temperature and thus assists the oxidation reaction of the gaseous fuel. Advancing the injection timing provides longer time for the gaseous fuel to react subsequent to the initial reaction of the pilot diesel.

Prechamber Diesel Engine

Diesel engines can be classified into two types: direct-injection or indirect-injection types depending on whether the combustion involves one or two chambers. A prechamber diesel engine is an indirect-injection engine consisting of two chambers, prechamber and main chamber. Fig. 1.1 shows the typical shape of the combustion chambers for direct-injection and prechamber diesel engines. The volume of the prechamber is typically 20 to 30 percent of the clearance volume. The main objective of the prechamber design is to burn a small fraction of the injected fuel in the prechamber so that the resulting pressure rise will drive the mixture of partially burnt and unburned fuel into the main chamber as a high speed jet whose turbulence will promote rapid and complete combustion.

The advantages of the prechamber diesel engine compared to the direct-injection type are better emission characteristics
Figure 1.1 - Combustion Chambers of Direct-Injection and Prechamber Engines
and less tendency to knock. Because of the better emission characteristics the prechamber engines are preferred for higher speed operations. The disadvantages are mainly associated with higher surface-to-volume ratio which enhances heat loss and throttling between prechamber and main chamber. These result in higher fuel consumption rate.

The type of diesel engine used in dual-fuel operation with natural gas has been almost exclusively direct-injection, so that the behaviour of a prechamber diesel engine with dual-fuelling over a range of load appears to be virtually unknown.
1.2 Present Study

Objectives

The primary objective of this study was to determine the feasibility of dual-fuel operation with natural gas in a prechamber diesel engine. Observations were made of fuel consumption and cylinder pressure development which may critically affect engine durability. The effects on fuel consumption and cylinder pressure of the following variables were studied:

a. flow rate of pilot diesel
b. gas-air mixture strength
c. injection timing of pilot diesel

Computer analysis of apparent energy release was employed to study the combustion characteristics, first for straight diesel operation and then with regard to the above three operating variables. Past experience with direct-injection engines was considered in anticipating possible operating difficulties and in analyzing the observed combustion characteristics.

Experimental Work

A Caterpillar 3304, four-cylinder prechamber marine engine, with turbocharger was used in the course of the study. The experimental study was done at constant engine speed because load changes rather than speed changes were considered to be of
chief concern in examining the feasibility of dual-fuel operation. Throughout the experiments measurements were taken to produce fuel consumption and cylinder pressure data.

The first phase of tests was performed with various loads and pilot diesel fuel rates. It was found that the fuel consumption rate of dual-fuel operation is considerably higher than that of straight diesel operation at loads much lower than the 'full load' specified by the engine manufacturer. As the load was increased to near full load the fuel consumption rate approached that of straight diesel operation. It was observed that at some point beyond full load the fuel consumption rate of the dual-fuel operation would become lower than that of straight diesel operation. The flow rate of pilot diesel was shown to greatly affect the fuel consumption rate at low loads. As the load was increased the effect of pilot diesel flow rate became smaller. The maximum cylinder pressure and pressure rise were observed to increase rapidly with increase in load. The maximum power output was severely limited by the maximum cylinder pressure. It was found that when the flow rate of pilot diesel was below a certain limit the operation became erratic with misfirings. For a range of loads and pilot diesel flow rates, a region of unstable operation due to insufficient pilot diesel flow rate was established.

The second phase of the experiments was intended for study of the effect of intake air restriction. During the initial stage of the experiments it was noticed that excessive air restriction can cause surge of the compressor in the
turbocharger. Since surge can easily cause mechanical damage of the turbocharger the intake air restriction had to be limited to a small range of air flow reduction.

In the last stage of experiments the effect of injection timing was studied primarily because of concern over maximum pressures associated with dual-fuelling and normal injection timing. It was found that retarding the injection timing results in significant reduction in both the maximum cylinder pressure and pressure rise. The change in fuel consumption rate due to the retarded injection was found to be small.

Computer Analysis of Apparent Energy Release

A computer program which computes apparent energy release due to combustion was developed in order to study the combustion characteristics. The cylinder pressure data obtained during the course of above three phases of experiments were used in the analysis. The analysis showed that the excessive maximum cylinder pressure was mainly associated with high rate of combustion energy release. The results of the analysis are consistent with different mechanisms of combustion: auto-ignition of diesel in straight diesel operation and propagation of flame fronts in dual-fuel operation.
2.1 History of the Dual-Fuel Diesel Engine

The earliest practical use of gas as an engine fuel dates back to the end of the 19th century. Spark-ignited engines called 'gas engines' operated in much the same way as modern gas engines. In these early engines the gas-air mixtures were nearly stoichiometric and the compression ratios were about 6:1. Commercial production of engines of various sizes began in about the year 1900. Jones (1944) states that by 1920 in Britain engines with maximum power ranging from 5 to 2000 horse power (4 to 1500 kW) were manufactured for use mainly on waste gases, particularly blast furnace gas.

The first attempt to burn gas in a compression ignition engine appears to have been made by the C.&G. Cooper Company in 1927. According to Boyer and Crooks (1951), the first test consisted of injecting natural gas alone at high pressure at the end of the compression stroke. This resulted in irregular firing of the gas. In the next test a small portion of diesel was injected in addition to the gas. This was the birth of the so called 'gas-diesel' engine, in which the gaseous fuel was injected into the cylinder at high pressure (about 1000 psi or 7 MPa). The first commercial installation of such engine was achieved by the Nordberg Company in 1935. A 1,665 horse power (1,241 kW) engine was installed at Lubbock, Texas, and the operation was successful; at full load the specific fuel
consumption was as low as that of diesel operation.

The high pressure gas injection equipment needed for the 'gas-diesel' engines gave rise to problems: the equipment was costly and thus limited to use on large engines, and it was rather difficult to maintain. In 1938 the National Gas and Oil Company developed an 8-cylinder 440 horse power engine, which used town gas. In this engine the gas was admitted to the cylinder at low pressure through a separate passage from the air inlet. The operation was successful and led to the conversion of existing engines at the Coleshill Works of the Birmingham Tame and Rea District Drainage Board.

An alternative means of admitting the gas at low pressure became commonly used. Mitchell and Whitehouse (1954) described one such scheme developed by the English Electric Company to provide better gas-air mixing. Instead of being directly admitted into the cylinder, gas was mixed with air in the intake manifold after upstream injection through a 'flutter' valve. This flutter valve acted as non-return valve to prevent pressure pulsations or explosions passing back into the gas supply pipes.

The literature of the 1950's reveals some fundamental studies of dual-fuel operation on direct-injection diesel engines. These are discussed in detail in the following section.
2.2 Review of Research

An extensive summary of the results of dual-fuel diesel combustion research is provided in review papers by Karim (1980) and Karim (1982). Both research results and applications have been reviewed by O'Neal (1982). The literature contains a considerable amount of information concerning the operating experience and combustion processes of dual-fuel diesel operation though this is restricted to direct-injection engines only. The research experiences are discussed in this section in chronological order as indicated in Table 2.1 which mentions the main features of each project. The review here is restricted to studies which involve methane-based gases.

The importance of gas-air mixture strength in dual-fuel operation was studied by Elliott & Davis (1951). In their experiments with a CFR* diesel engine at a compression ratio of 21:1, selected pilot diesel rates were held constant to determine the effect of the concentration of natural gas (88.9% methane, 10.6% other hydrocarbons) in the intake. Their experiments showed that when the gas-air mixture strength was below a certain limit, the proportion of gas reacting increased with diesel fuel-air ratio and gas-air mixture strength. They found that if the concentration of gas is below the lower limit of flammability (which is commonly defined as the concentration

*CFR (Cooperative Fuel Research) engine: single-cylinder engine, with 3.25 in. (82.5 mm) bore and 4.5 in. (114 mm) stroke, manufactured by the Waukeska Engine Co. of Waukeska, Wis., the standard engine used for detonation measurement and generally for detonation research.
<table>
<thead>
<tr>
<th>AUTHOR</th>
<th>DATE</th>
<th>ENGINE</th>
<th>GASEOUS FUEL</th>
<th>MAIN FINDINGS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elliot &amp; Davis</td>
<td>1951</td>
<td>CFR diesel c.r. 16:1, 21:1</td>
<td>natural gas</td>
<td>lower limit of flammability, dependence of amount of gas reacted on gas-air mixture strength and diesel fuel-air ratio.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>methane 88.9% other hydrocarbon 10.6%</td>
<td></td>
</tr>
<tr>
<td>Lewis</td>
<td>1954</td>
<td>single-cylinder direct-injection c.r. 14.7:1 bore 105mm stroke 152mm</td>
<td>sludge gas methane 86.8% nitrogen 4.5% carbon dioxide 5.5%</td>
<td>effect of intake air restriction and intake air preheating on efficiency.</td>
</tr>
<tr>
<td>Simonson</td>
<td>1955</td>
<td>single-cylinder direct-injection c.r. 14.7:1 bore 105mm stroke 152mm</td>
<td>methane</td>
<td>effect of intake air preheating, intake air restriction, and varying injection timing on efficiency.</td>
</tr>
<tr>
<td>Moore &amp; Mitchell</td>
<td>1955</td>
<td>single-cylinder direct-injection bore 105mm stroke 152mm</td>
<td>methane</td>
<td>effect of intake air preheating, intake air restriction, and varying injection timing on efficiency.</td>
</tr>
<tr>
<td>Mitchell &amp; Whitehouse</td>
<td>1955</td>
<td>four-cylinder direct-injection c.r. 13.5:1 bore 254mm stroke 305mm</td>
<td>sludge gas methane 87.9% nitrogen 4.4% carbon dioxide 5.5%</td>
<td>effect of intake air restriction on efficiency</td>
</tr>
<tr>
<td>Author</td>
<td>Year</td>
<td>Engine Type</td>
<td>Compression Ratio</td>
<td>Fuel Composition</td>
</tr>
<tr>
<td>-----------------</td>
<td>------</td>
<td>----------------------</td>
<td>-------------------</td>
<td>------------------</td>
</tr>
<tr>
<td>Felt &amp; Steele</td>
<td>1962</td>
<td>Single-cylinder direct-injection</td>
<td>16.2:1</td>
<td>Methane 87.1%, Nitrogen 7.1%, Other hydrocarbons 5.1%</td>
</tr>
<tr>
<td>Karim &amp; Kahn</td>
<td>1968</td>
<td>Single-cylinder direct-injection</td>
<td>105mm bore, 152mm stroke</td>
<td>Methane</td>
</tr>
</tbody>
</table>

* Gas composition based on volume

Table 2.1 - Summary of Past Experimental Work
of gaseous fuel in the intake air at which the minimum liquid fuel-air ratio yields consistent and close-to-complete combustion) the gas does not react completely with oxygen unless it is in, or immediately adjacent to, an inflamed or high temperature region. In the absence of pilot diesel, a stoichiometric gas-air mixture does not appear to react to a significant extent; the exhaust gas analysis shows no sign of the presence of carbon dioxide, carbon monoxide, or aldehydes. Therefore, unless the natural gas is in a comparatively high temperature region, it is unlikely that the gas would react when its concentration is lower than the lower limit concentration.

The tests of Elliot and Davis in several diesel engines showed that the lower limit of flammability of natural gas in air under conditions existing at the end of compression is approximately 4 to 5 percent by volume. The corresponding ratio for the stoichiometric mixture strength was 9.1% by volume.

The experiments by Lewis (1954) were focused mainly on operations with weak gas-air mixture strength (below 8% by volume). Lewis used sludge gas (86.6% methane, 5.45% carbon dioxide, 4.5% nitrogen by volume) in a single cylinder direct-injection engine with compression ratio of 14.7:1 and engine speed of 1000 rpm. The lower limit of flammability in his work was shown to be about 6.2% by volume. This is somewhat higher than the result of earlier work by Elliot and Davis (1951), and may be due to the high concentration of carbon dioxide and nitrogen in the sludge gas used by Lewis. The experiments by Lewis showed that when the gas-air mixture strength is below the flammability limit, restricting intake air results in
significant increase in the amount of gas reacted and in increase in ignition delay. Preheating of intake air was also found to increase the amount of gas reacted, but resulted in decreased ignition delay. With preheating of the intake charge to 225 deg.C substantially complete combustion was achieved (95% of gas reacted at 4.5% gas-air mixture strength ,and 70% at 1.0% mixture strength). The ignition delay of pilot diesel in relation to gas-air mixture strength was also studied. The ignition point was identified from the pressure-time trace as the point of significant pressure rise due to combustion. Figure 2.1 shows the results obtained at constant pilot diesel rate of 0.415 lb/h (0.188 kg/h, 10% of the straight diesel full load fuel rate). Increase in gas-air mixture strength resulted initially in longer ignition delay of pilot diesel. Beyond the gas-air mixture strength of about 4% by volume, further increase in mixture strength showed sharp reduction in the ignition delay. The variation in ignition delay for the range of mixture strength of 0 to 8% by volume was about 3 deg. C.A.

The effect on combustion characteristics of gas-air mixture strength and intake charge temperature of a motored-engine was studied by Simonson(1954). His experiments were performed with methane in a direct-injection single-cylinder engine for intake charge temperatures ranging from 241 to 325 deg.C and gas-air mixture strength ranging from 0 to 5% by volume. The results from exhaust analysis of constant-speed motored tests revealed that increase in intake charge temperature results in more favourable conditions for flame propagation.
Figure 2.1 - Effect of Gas-Air Mixture Strength on Ignition Delay

speed
1000 rpm

pilot diesel
0.415 lb/hr
gas
methane

(Lewis, 1954)
Further work by Simonson (1955), with fired-engine operation included studies of the effects of air restriction, and changes in pilot diesel rate and injection timing. A direct-injection engine (the same engine as the one used by Lewis (1954)) with compression ratio of 14.7:1 was used with methane at the speed of 1000 rpm. Preheating of intake charge to 157 deg.C showed improvements of 20 to 30% in fuel consumption at part loads (10-60 psi or 70-410 kPa in brake mean effective pressure) with pilot diesel rate consisting 8.5% of diesel rate of straight diesel full load operation. With the same pilot diesel rate, advancing of injection timing by 6 deg. C.A. resulted in improvement of fuel consumption by 10 to 17% in the same part load range. Both intake air restriction and increase in pilot diesel rate gave significant improvements in fuel consumption. Maximum cylinder pressure was observed to increase with advanced injection timing. Tests at brake mean effective pressure of 115 psi (793 kPa) revealed that advancing the injection timing by 8 deg. C.A. increased the maximum cylinder pressure from 1000 to 1250 psi (7 to 8.7 MPa). Early ignition and rapid rates of pressure rise were reported to set a limit to the extent to which improved performance can be obtained by advancing the injection timing. With a pilot diesel injection rate of 0.8 lb/h (0.36 kg/h, 16% of straight diesel full load fuel rate) at 24 deg BTDC (10 deg advance) combustion was rough even at the brake mean effective pressure of 30 psi (210 kPa).

Experiments leading to improvement in part-load fuel consumption were done by Moore and Mitchell (1955). A single-cylinder direct-injection engine with 4.125 in (105.4 mm) bore
and 6.00 in (152 mm) stroke was used at the speed of 1000 rpm. Tests carried out with sludge gas on the effects of air restriction, intake charge preheating, and advancing injection timing showed results similar to those of the experiments by Simonson (1955). As much as 20% improvement in fuel consumption was reported in each separate test of the above methods. In reviewing past work they concluded that raising the intake temperature is the only practical way of extending the lower limit of flammability.

Work on a large engine was described by Mitchell and Whitehouse (1955). A four-cylinder direct-injection engine of 10 in (254 mm) bore and 12 in (305 mm) stroke with compression ratio of 13.5:1 was used at the speed of 600 rpm to determine the optimum conditions for reliability and efficiency. The gaseous fuel used was sludge gas (87.9% methane, 5.5% carbon dioxide, 4.4% nitrogen). The pilot diesel rate was a constant 6% of straight diesel full load fuel rate. Tests on the effect of air restriction showed a considerable improvement in fuel consumption: 46% improvement was reported at the brake mean effective pressure of 20 psi (140 kPa, 26% of full load). It was also found that with optimum air restriction the exhaust temperature remained approximately constant (within ± 50 deg.F or 30 deg.C) at all loads.

Experiments made by Felt and Steele (1962) showed that knock-limited maximum power is directly related to the anti-knock quality of the primary fuel. A three-cylinder direct-injection engine with compression ratio of 16.2:1 was used with
pilot diesel rate of 1.65 lb/h (0.74 kg/h, 12.8% of straight diesel full load fuel rate). Lead alkyl anti-knock compounds were found to be quite effective in enhancing the anti-knock quality of the primary fuel. A mixture consisting of 95% propane and 5% tetramethyllead was bled into the intake air stream. With addition of 5.5-6.0 gm of lead per therm (100,000 Btu or 106,000 kJ) of natural gas, it was possible to enhance the maximum power of dual-fuel operation with natural gas (87.1% methane, 5.1% other hydrocarbons, 7.1% nitrogen) by 26 percent without knock. The knock was described as 'audible high-frequency combustion loss', which was visible on the pressure-time trace. The knock was described from the observation of the shape of the pressure-time trace as 'end-gas knock': the knock arising from the autoignition of the end-gas ahead of the flame front. It is the type of knock which may occur in spark-ignition engines.

The influences of fuel-air mixture strength, pilot diesel rate and intake air temperature on knock-limited power were studied in detail by Karim et al. (1966/67) with a single-cylinder direct injection engine with compression ratio of 14.2:1 and 97.8% methane as the gaseous fuel. The knock was observed to be associated with a sharp change in the running regime of the engine and accompanied by loudly audible sound. The typical shape of the pressure diagram is shown in Figure 2.2. The knock-limited power, which is shown in Figure 2.3, was observed to decrease with increase in the intake air temperature and/or pilot quantity. It was found that the knocking occured only in a certain range of mixture strength; if
Figure 2.2 - Typical Pressure-Time Trace of Non-knocking and Knocking Operation with Methane as Gaseous Fuel
Figure 2.3 - Variation of Power Output with the Overall Mixture Strength for Different Intake Temperatures

(Karim et al., 1966/67)
the engine was operated on either side of that range of mixture strength, knock could be avoided. The region of knocking was on the lean side of stoichiometric mixture strength. The effect of the cetane number of the pilot diesel on the onset of knock was found to be small.

Karim and Kahn (1968) employed heat release analysis in an attempt to interpret the combustion processes. From the analysis with a single-cylinder direct-injection engine and methane as the gaseous fuel, they concluded that dual-fuel combustion generally undergoes two distinct phases. The first is mainly associated with the consumption of the pilot fuel together with part of the gaseous fuel. The second is associated mainly with the gaseous fuel and depends on its concentration and quality. The heat release analysis of very lean operation supported previous experimental evidences that poor combustion at low load operation is due mainly to the inability of the gaseous charge to supplement effectively the heat release of the first phase. The analysis of the operation with knock indicated that the knock was mainly associated with rapid simultaneous burning of the pilot diesel together with a substantial fraction of the gaseous charge.

Study of knock-limited maximum power in relation to compression ratio and engine speed is included in the review paper by O'Neal (1982). The knock-limited power increases very rapidly as the compression ratio is reduced. The trend for the knock-limited bmep is to increase with engine speed. As engine speed increases, less time is available for the end-gas to reach
the temperature for autoignition, and thus the onset of knock is suppressed.

In summary, the literature reveals that considerable research has been done on dual-fuel operation with the direct-injection type of diesel engine. At low loads, combustion suffers from weak gas-air mixture strength resulting in unburned gas escaping with the exhaust. Fig. 2.4 shows typical thermal efficiencies of dual-fuel operation with direct-injection diesel engines. Restricting or preheating intake air, increasing pilot diesel flow rate, and advancing the injection timing have been shown to be effective for improving the fuel consumption at low loads. Maximum power output of high load operation is limited by the occurrence of knock, which appears to be of the same kind as in spark-ignition engines. Adding lead alkyl anti-knock compounds or cooling the intake air were shown to be effective in improving knock-limited maximum power. The literature, however, seems silent on dual-fuel diesel operation with the prechamber type of diesel engine.
Figure 2.4 - Typical Thermal Efficiencies of Dual-Fuel and Straight Diesel Operation
CH.III Apparatus and Instrumentation

The arrangement of apparatus and instruments is shown in Fig. 3.1. The engine was coupled to an electromagnetic dynamometer. The signals from the cylinder pressure transducer were collected by a NEFF/620 data acquisition unit. The data obtained from various instruments were processed with a PDP 11/34 computer. The flow diagram of air, natural gas, and exhaust gas is provided in Fig. 3.2. Natural gas is mixed with air prior to entering the turbocharger. The gas-air mixture exiting from the turbocharger enters the cylinder during the intake stroke. The exhaust gas from the cylinder passes through the turbine side of the turbocharger, and then enters a muffler before discharging to open air.

3.1 Engine and Test Bed

Engine

A caterpillar 3304 four-cylinder engine coupled with an electromagnetic dynamometer was used in this project. The engine speed was adjusted to 1600 rpm throughout the experiments. Full load at this speed was specified by the manufacturer as 124 psi (856 kPa) in brake mean effective pressure. The engine is of prechamber type and is equipped with a turbocharger. The specification of the engine is provided in
Figure 3.1 - Layout of Apparatus and Instrumentation
Figure 3.2 - Flow Diagram of Air, Fuel, and Exhaust Gas
<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>BORE</td>
<td>12.1 cm (4.75 in.)</td>
</tr>
<tr>
<td>STROKE</td>
<td>15.2 cm (6.0 in.)</td>
</tr>
<tr>
<td>DISPLACEMENT</td>
<td>6970 cm³ (425 cu.in.)</td>
</tr>
<tr>
<td>COMPRESSION RATIO</td>
<td>17.5 : 1</td>
</tr>
<tr>
<td>NUMBER OF CYLINDERS</td>
<td>4</td>
</tr>
<tr>
<td>MAXIMUM POWER</td>
<td>93.2 kW (125 hp)</td>
</tr>
<tr>
<td></td>
<td>at 2000 rpm</td>
</tr>
<tr>
<td>TYPE</td>
<td>prechamber</td>
</tr>
<tr>
<td>ASPIRATION</td>
<td>turbo-charged</td>
</tr>
</tbody>
</table>

Table 3.1 - Engine Specification
Figure 3.3 - Shape of Combustion Chambers
Table 3.1. The volume of the prechamber is 27 percent of the total volume when the piston is at the top dead center. The cross-section of prechamber and main chamber is shown in Fig. 3.3.

**Diesel Injection System**

The engine was equipped with a sleeve metering type of fuel system. Fig. 3.4 shows the layout of the system. The main components of the sleeve metering diesel-injection pump are shown in Fig. 3.5. The plunger is moved up and down inside the barrel by the action of the pump camshaft. Fig. 3.6 shows the effective stroke of the plunger and the sequence of the injection events. The sleeve metering system uses centrifugal governor flyweights (shown in Fig. 3.7) in order to prevent any change in engine speed due to the variation in load. Thus once the governor control is set at certain position in the rack setting, the engine speed is maintained constant regardless of possible change in load. Fig. 3.8 shows the single hole diesel-injection nozzle used in the engine. The injection timing was adjusted to desired angle according to the service manual(NO.SER7053-01, pp80).
Figure 3.4 - Sleeve Metering Fuel System

Figure 3.5 - Fuel Injection Pump and Housing
Figure 3.6 - Sequence of Injection Events

Figure 3.7 - Governor Components of Sleeve Metering
Figure 3.8 - Fuel Injection Nozzle
Turbocharger

The engine was equipped with a T1210 model turbocharger manufactured by AiResearch. A cutaway view of the turbocharger is shown in Fig. 3.9.

Dynamometer

The engine was coupled to an electromagnetic dynamometer (General Electric, model 1G136). The absorption capacity of the dynamometer was 200 horse power (150 kW).

Gas-mixer

In order to introduce the natural gas, a simple gas-mixer was installed on upstream of the inlet to the turbocharger. Fig. 3.10 shows the gas-mixer.

Intake Air Restriction

A simple 'butterfly' type of valve was installed near the gas-mixer to control the amount of air intake.
Figure 3.9 - Turbocharger Cutaway View
3.2 Instrumentation

Torque

The torque applied by the engine shaft to dynamometer was obtained by placing a strain gage load cell (Interface Inc., model 1420-4F) on the dynamometer housing. The maximum allowable load specified by the manufacturer was 500 lb. A bridge amplifier meter (Ellis Associates, model BAM-1) was used to amplify the response from the load cell. The load cell was calibrated by placing various weights on the arm of the dynamometer housing and reading the voltage from the bridge amplifier meter. The calibration curve for the load cell is provided in Appendix A. The relation between the response of the load cell and the applied weight was very nearly linear.

Cylinder Pressure

The no. 1 cylinder was instrumented with an AVL piezoelectric pressure transducer (model 8QP500c). The transducer was cooled with water and mounted in a steel sleeve through the cylinder head. Fig. 3.11 shows the location of the mounted transducer. The signal from the transducer was transmitted by a low noise cable to a charge amplifier (Kistler, model 5004) and then to a data acquisition system. The system consisted of a NEFF, System 620, analogue to digital converter which was connected to a PDP 11/34 minicomputer. A computer program was
Figure 3.11 - Mounting of Cylinder Pressure Transducer
written (see Appendix C) to sample the pressure signal at intervals of one degree crank angle, along with a bottom dead center signal, drawn from an optical sensor mounted on the toothed wheel at the front of the engine. The program computed an ensemble-averaged value of pressure collected over 30 to 50 cycles for each degree of crank angle. The averaged values were then used to compute indicated mean effective pressure and to analyze apparent energy release. The pressure transducer was calibrated using a dead-weight tester for a pressure range of 0 to 2000 psi (14 MPa). The calibration curve is provided in Appendix A.

Air Flow Rate

A laminar flow element (Meriam Instrument, model 50MC2-4F, range 0-400SCFM) was used to measure the air flow rate. It was mounted between the air filter and turbocharger. The pressure drop across the element was read in inches of water on a water-filled U-tube manometer and translated to volumetric flow rate. The calibration curve provided by the element manufacturer is shown in Appendix A.

Gas Flow Rate

The natural gas was drawn in from the mains supply at a pressure of 5 psi. The gas was then passed through a pressure regulator which reduced the pressure to a pressure a few inches
of water higher than atmospheric pressure. A laminar flow element (Meriam Instrument, model 50MH10-1.25NT, range 0-15SCFM) was mounted to measure the flow rate. The pressure drop across the element was read in inches of water on a water-filled U-tube manometer and translated to volumetric flow rate. The flow rate was then corrected for natural gas by multiplying the ratio of viscosities for air and natural gas. The amount of gas admitted to intake was controlled manually with a tapered type of gas valve. The calibration curve provided by the element manufacturer is shown in Appendix A.

**Diesel Flow Rate**

A positive displacement type of flow meter (American Meter, model 1A, range 0.05-5GPH) which measures cumulated flow rate was used with a stop watch for a time period of 10 to 20 minutes to obtain the volumetric flow rate of diesel. The accuracy of the flow meter was confirmed by using a graduated cylinder.

**Turbocharger Inlet Pressure**

As a precaution to prevent the possibility of turbocharger surge due to excessive air restriction, the air pressure at the inlet of compressor side of the turbocharger was measured for each change of air flow rate. A water-filled U-tube manometer was used. The limiting measure of the compressor inlet pressure specified by the engine manufacturer was 24 inches of water.
below the atmospheric pressure.

**Intake Air Pressure**

In order to measure the intake air pressure after the turbocharger, a bourdon-tube type of pressure gage (Marquette, model 41-123, range 30 inches of water vacuum to 15 psi above atmospheric pressure) was mounted on the intake manifold near no. 4 cylinder.

**Engine Speed**

The engine speed was measured by a hand digital tachometer (Shimpo, model DT-205) which sends out a continuous light beam and counts the pulses reflected off a piece of reflective tape attached on the engine shaft.
3.3 Fuel

Diesel

The diesel fuel used throughout the experiments had the following typical properties:

API gravity - 31
specific gravity - 0.871
lower heating value - 45,263 kJ/kg

In the analysis of energy release and computation of stoichiometric fuel-air ratio dodecane (C₁₂H₂₅) was assumed to be the representing hydrocarbon for the diesel fuel.

Natural Gas

The natural gas used in the experiments had the following properties:

density - 0.766 kg/m³
  at 101.3 kPa,
  25 deg. C

viscosity - 108.96 micropoise
  at 21.1 deg. C

lower heating value - 48,558 kJ/kg

A typical composition of the natural gas used here is given in Table 3.2.
<table>
<thead>
<tr>
<th>COMPOSITION</th>
<th>RELATIVE VOLUME (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>methane</td>
<td>94.00</td>
</tr>
<tr>
<td>ethane</td>
<td>3.30</td>
</tr>
<tr>
<td>propane</td>
<td>1.00</td>
</tr>
<tr>
<td>iso-butane</td>
<td>0.15</td>
</tr>
<tr>
<td>n-butane</td>
<td>0.20</td>
</tr>
<tr>
<td>iso-pentane</td>
<td>0.02</td>
</tr>
<tr>
<td>n-pentane</td>
<td>0.02</td>
</tr>
<tr>
<td>nitrogen</td>
<td>1.00</td>
</tr>
<tr>
<td>carbon dioxide</td>
<td>0.30</td>
</tr>
<tr>
<td>hexane</td>
<td>0.01</td>
</tr>
</tbody>
</table>

Table 3.2 - Typical Composition of the Natural Gas Used
3.4 Data Process

The measurements obtained from various instruments for each experiment were fed into the PDP 11/34 computer for computation of the following:

- power output
- thermal efficiency
- volumetric efficiency
- proportion of diesel to total fuel input based on heating values
- gas-air, diesel-air, total fuel-air ratio

A typical output from the data process is shown in Table 3.3. A listing of the computer program used for the computations is included in Appendix II.
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>engine speed</td>
<td>1601 rpm</td>
</tr>
<tr>
<td>air flow rate</td>
<td>170 cu.ft/min</td>
</tr>
<tr>
<td>diesel flow rate</td>
<td>1.82 l/hr</td>
</tr>
<tr>
<td>gas flow rate</td>
<td>6.32 cu.ft/min</td>
</tr>
<tr>
<td>fuel consumption rate</td>
<td>11,683 BTU/hr-hp</td>
</tr>
<tr>
<td>power output</td>
<td>35.6 hp</td>
</tr>
<tr>
<td>brake mean effective pressure</td>
<td>41.4 psi</td>
</tr>
<tr>
<td>thermal efficiency</td>
<td>21.8 %</td>
</tr>
<tr>
<td>volumetric efficiency</td>
<td>86.3 %</td>
</tr>
<tr>
<td>diesel input proportion</td>
<td>15.5 %</td>
</tr>
<tr>
<td>gas-air equivalence ratio</td>
<td>0.365</td>
</tr>
<tr>
<td>diesel-air equivalence ratio</td>
<td>0.068</td>
</tr>
<tr>
<td>fuel-air equivalence ratio (total fuel)</td>
<td>0.433</td>
</tr>
</tbody>
</table>

Table 3.3 - Typical Output of Computer Program for Data Processing
CH. IV. Experimental Results

4.1 Fuel Consumption

4.1.1 Fuel Consumption with Unmodified Engine

Tests were carried out to study the change in fuel consumption rate with variation in load and pilot diesel flow rate. The engine speed was set at 1600 rpm for all the experiments. The injection timing of the diesel fuel remained constant at 12.3 degrees before top dead centre.

Tests always started from straight diesel operation. While the engine was running on 100 percent diesel fuel, the speed was adjusted with the diesel fuel governor rack setting to 1600 rpm, and the load to the predetermined setting. The gas was then gradually added by opening the gas control valve. As the amount of gas was increased, the flow of diesel fuel was reduced to maintain the speed at 1600 rpm.

In Fig. 4.1, brake thermal efficiencies based on lower heating value are plotted against the fractional diesel energy input, which is here defined as the ratio of diesel to total energy input. The shaded area on the left corresponds to the region where stable operation is not possible because of insufficient pilot diesel flow rate. In this region, the operation was erratic with misfired cycles observed on the cylinder pressure-time trace on the oscilloscope. The interpolated paths of constant pilot diesel operation are
Figure 4.1 - Effect of Pilot Diesel Rate on Brake Thermal Efficiency
indicated by the dotted lines. As previously described in Chapter III, the diesel injection rate of the engine was controlled by two mechanisms – the governor rack setting, which is adjusted manually, and the centrifugal governor flyweights, which respond to any small change in applied load in order to maintain a constant speed. Thus even though the rack setting is held fixed, addition of gas in the intake would result in change in diesel injection rate. For this reason it was not possible to carry out tests for constant pilot diesel flow rate.

It is seen from Fig. 4.1 that the decrease of thermal efficiency due to reduction of pilot diesel flow rate becomes smaller as the load increases. At the brake mean effective pressure of 714 kPa (84% of full load), the change in thermal efficiency is less than 1 percent over the range of fractional diesel energy input of 7 to 100 percent. The extent to which the fuel consumption rate depends on the pilot diesel flow rate for idling operation is seen in Fig. 4.2. The idling operation with the fractional diesel energy input of 15 percent requires nearly twice as much energy input than that of straight diesel operation.

To obtain another picture of the role of pilot diesel flow rate in dual-fuel operation, calculations were made of indicated thermal efficiencies. The indicated thermal efficiency, being based on total power produced (including the power used to overcome the frictional and pumping losses), should be more closely correlated with the effectiveness of the combustion process than the engine efficiency based on shaft power output.
Figure 4.2 - Fuel Consumption at Idling Operation
This assumption is consistent with the work of Simonson (1955) in which the measured values of indicated thermal efficiency and the proportion of gaseous fuel reacted showed qualitatively similar trends. Appendix B includes a description of the method used in present work to obtain indicated mean effective pressure.

The indicated thermal efficiencies were interpolated and plotted in Fig. 4.3. The efficiencies are shown as a function of the flow rate of pilot diesel at constant gas-air mixture strength. The equivalence ratio $\Phi_g$ of the gas was computed as the ratio of the mass of the stoichiometric amount of air required for combustion of the gas alone to the mass of the actual amount of air drawn in: i.e.

$$\Phi_g = \frac{\text{stoichiometric air mass flow rate}}{\text{actual air mass flow rate}}$$

It can be seen from Fig. 4.3 that the pilot diesel flow rate has a very significant effect on the thermal efficiency at low gas-air mixture strength. As the gas-air mixture strength becomes richer, the thermal efficiency becomes less sensitive to the change in pilot diesel flow rate. With gas equivalence ratio of 0.6, an increase of pilot diesel flow rate from 1.4 to 3.5 kg/h results in less than 1 percent improvement in indicated thermal efficiency. If it is assumed that the measured values of the indicated thermal efficiencies are largely a function of the amount of gas burned, then the gas-air equivalence ratio of 0.6 would be a good approximation for the lower limit of flammability for this particular type of engine. The gas
Figure 4.3 - Effect of Pilot Diesel Flow Rate on Indicated Thermal Efficiency
equivalence ratio of 0.6 corresponds to the gas-air volumetric ratio of about 4 percent. This approximate value is close to the lower inflammability limit of 4 to 5 percent suggested by Elliott and Davis (1951) for a direct-injection diesel engine.

The dotted lines in Fig. 4.3 indicate the paths of constant load operation with variation in pilot diesel rate. The line at the bottom traces the operation at idling, and the one at the top the operation at about 84 percent of full load. Operating with a constant pilot diesel rate of 3.4 kg/h would correspond to 100 percent diesel fuelling at idling and 20 percent diesel fuelling at 85 percent of full load. The shaded area on the left is the region of unstable operation due to misfiring.

Fig. 4.4 shows brake thermal efficiency for straight diesel and dual-fuel operations. The straight diesel operation shows the fuel consumption characteristics typical of compression ignition engines. As the load is increased from idling, the efficiency improves because of the increase in power output while the frictional loss of the engine remains relatively constant for a fixed speed. At the brake mean effective pressure of about 700 kPa, the thermal efficiency reaches a peak. With further increase in load, the curve starts to decline, presumably due to the decreased access of fuel to oxygen. One of the dual-fuel efficiency curves corresponds to operation with the minimum pilot diesel flow rate needed to assure stable operation without misfiring. This curve is obtained from Fig. 4.1 by interpolation. The thermal efficiency of the dual-fuel operation at part load is substantially lower
Figure 4.4 - Comparison of Brake Thermal Efficiencies for Dual-Fuel and Straight Diesel Operation

Values for dual-fuel operations are interpolated from Fig. 4.1
than that of straight diesel operation. At the brake mean effective pressure of 850 kPa, which is about the rated full load at 1600 rpm, the thermal efficiencies of both operations are same. Extrapolation of the dual-fuel operation curve to higher loads suggests the trend of the thermal efficiency surpassing that of straight diesel operation. This trend may be due to the homogeneous nature of gas-air mixture in dual-fuel operation, which allows better access of fuel to oxygen.

4.1.2 Effect of Restricting Intake Air

Tests to determined the effect on thermal efficiency of restricting intake air were performed for several load settings ranging from idling to about 50 percent of full load. For each load setting, the rate of gas flow was controlled so that the flow rate of pilot diesel would remain approximately constant. The air restriction was limited by the minimum allowable pressure of the air at the inlet of the turbocharger. Excessive restriction of air beyond the limit resulted in surging of the turbocharger, which in turn caused violent unsteadiness of the engine. Because of this, the air-gas flow ratio reduction had to be confined to about 10 percent. Sufficient reduction of air, either by using gating or eliminating the turbocharger, would have increased the mixture strength to near the lower limit of flammability. Such an increase in the mixture strength may have led to much improved fuel consumption.
Fig. 4.5 shows the effect of air restriction on brake thermal efficiency at various load settings. The gas-air mixture strength is represented by the equivalence ratio based on the gas alone. The tests for all the load settings exhibit improvements when air restriction is imposed. The increase in brake thermal efficiency however was for all cases less than 1 percent for about 10 percent air reduction.

4.1.3 Effect of Varying Injection Timing

The ignition delay of pilot diesel was studied by observing the averaged cylinder pressure vs crank angle trace. In determining the point of the start of ignition (the point at which combustion has proceeded far enough to affect the pressure noticeably), the earliest significant deviation of pressure from the expected compression curve near top dead centre was sought by close examination. In most cases this method allowed identification of the point without difficulty. Fig. 4.6 shows a typical pressure trace and identification of the apparent point of ignition.

With the injection timing fixed at 12.3 degree BTDC, the change in ignition delay of pilot diesel with addition of gas was investigated at a range of load settings. Fig. 4.7 shows the crank angle at the start of ignition with the pilot diesel rate varying from 10-20 to 100 percent of total energy input. For straight diesel operation at low load, the ignition does not
Figure 4.5 - Effect of Intake Air Restriction on Brake Thermal Efficiency
Figure 4.6 - Typical Cylinder Pressure Trace and Apparent Point of Ignition Start
Figure 4.7 - Apparent Point of Ignition Start at Various Loads
start until 2 degrees after top dead centre. As the load is raised to half of full load, the ignition point is advanced by a degree probably due to the increased end-gas temperature. The pressure vs crank angle trace reveals the increase in end-gas pressure of about 50 kPa for the above load increase. The addition of gas extends the ignition delay by 1 to 2 degrees. The corresponding result obtained by Moore and Mitchell (1955) in a direct injection engine was about 2 degrees and by Karim (1980) also in a direct injection engine about 1.5 degrees. These agreements in magnitude of the extended ignition delay seems to suggest that the higher level of turbulence prior to the ignition of pilot diesel in prechamber engine does not have a significant effect on ignition delay of the pilot diesel. In the absence of air restriction, the point of ignition is no later than 3 degrees ATDC for all the loads and pilot diesel flow rates tested.

To study the effect on thermal efficiency of varying injection timing, the timing was advanced by 5 and 10 degrees for two load settings, 143 and 278 kPa in brake mean effective pressure. The flow rate of pilot diesel was controlled to be about 20 percent of total energy input. The lower graph of Fig. 4.8 shows that advancing the injection timing by 5 degrees advances the point of ignition to top dead center for both of the load conditions. The improvement in brake thermal efficiency for the corresponding change in injection timing is very little, as shown by the upper graph. Further advance of the timing by 5 degrees results in advancement of the apparent point of ignition start to 4 degrees BTDC. The thermal
Figure 4.8 - Effect of Varying Injection Timing on Brake Thermal Efficiency and Apparent Point of Ignition Start
efficiency of the low load operation of 143 kPa in brake mean effective pressure shows no improvement, while that of the operation of 278 kPa in brake mean effective pressure exhibits a trend to deteriorate. The tests for higher load conditions were avoided in fear of excessive increase in maximum cylinder pressure.
4.2 Cylinder Pressure

4.2.1 Cylinder pressure in Unmodified Engine

For each randomly selected cycle cylinder pressure was measured at every crank angle. The pressure values at each angle were then averaged over 30-50 cycles. The engine speed and the injection timing were fixed at 1600 rpm and 12.3 degree BTDC respectively. Fig. 4.9 shows the P-V diagrams for straight diesel operation with various load settings ranging from idling to full load. The low load operations reveal the delay until after the top dead centre of significant rise in pressure due to combustion. The maximum cylinder pressure reached at full load operation is about 7870 kPa (1140 psi).

The cylinder pressure and volume of compression and expansion processes in internal combustion engines can be approximately related by following relationships:

\[
\frac{n}{P \, V} = \text{const}
\]

or

\[
\ln(P) = n \, \ln(V) + \text{const}
\]

The exponent 'n' would be exactly the ratio of specific heats if the working fluid was an ideal gas with constant properties, and the compression or expansion process was adiabatic and frictionless. For a mixture of real gases at combustion pressures, the ideal gas is a good approximation and for small
Figure 4.9 - P-V Diagram of Straight Diesel Operation
frictional effect, small heat transfer to the walls, slowly changing properties the above relationship is valid with nearly constant value of 'n'. The ln(P)-ln(V) plot thus provides an approximate but convenient means to identify the points of the beginning and the end of the combustion. Fig. 4.10 shows the ln(P)-ln(V) plot of the straight diesel operation. The figure indicates the separate prechamber and main-chamber stages of combustion through their effects on the pressure development. At full load, the initial combustion starts near top dead centre. A short period of expansion which is recognized by a short straight line near the ln(volume ratio) of 0.65. The subsequent stage of combustion is identified by the deviation of pressure from a straight line. This characteristic is less distinctive at lower loads. The idling operation shows no sign of the 2-stage combustion characteristic. It seems that the combustion in the first stage corresponds to the initial incomplete combustion in the prechamber. The short period of expansion process following the first stage is no doubt associated with the escape from the prechamber of the hot partially burned diesel-air mixture and mixing with air which has resided in the main chamber. The second stage combustion then takes place in the main chamber. As the load is reduced to idling, the flow rate of diesel fuel is reduced to a small amount and the combustion in the prechamber is probably nearly complete, requiring little further combustion in the main chamber.

Fig. 4.11 compares typical indicator diagrams of dual-fuel and straight diesel operations. The dual-fuel operation
Figure 4.10 - Ln P-V Diagram of Straight Diesel Operation
Figure 4.11 - Comparison of P-V Diagrams for Dual-Fuel and Straight Diesel Operation
exhibits a much slimmer P-V diagram, with much of the combustion taking place within a narrow range of cylinder volume. The $\ln(P)-\ln(V)$ diagrams in Fig. 4.12 shows the very different combustion characteristics of dual-fuel and straight diesel operations. The $\ln(P)-\ln(V)$ diagram of dual-fuel operation shows no obvious intermediate expansion period which is quite distinctively recognized in straight diesel operation. The effect is particularly noticeable at high load where the combustion duration of dual-fuel operation is much shorter than that of straight diesel operation. This would be consistent with rapid propagation of flame fronts in the gas-air mixture, in contrast to the slower combustion in straight diesel operation.

Fig. 4.13 shows the change in maximum cylinder pressure with load variation for straight diesel and dual-fuel operation with pilot diesel consisting 20 percent of total energy input. At high load the maximum cylinder pressure of dual-fuel operation is very much higher than that of straight diesel operation. At brake mean effective pressure of 714 kPa, which corresponds to 84 percent of full load, the maximum pressure of dual-fuel operation is about 35 percent higher than that of the straight diesel operation.

The maximum cylinder pressures of dual-fuel operation at various loads and pilot diesel rates are shown in Fig. 4.14.

Fig 4.15 shows the comparison of maximum rate of cylinder pressure rise at various load settings. Fig. 4.16 shows the maximum rate of pressure rise at various pilot diesel rates.
Figure 4.12 - Comparison of Ln P-V Diagrams for Dual-Fuel and Straight Diesel Operation
Comparison of Maximum Cylinder Pressures for Dual-Fuel and Straight Diesel Operation

Figure 4.13 - Comparison of Maximum Cylinder Pressures for Dual-Fuel and Straight Diesel Operation
Figure 4.14 - Maximum Cylinder Pressure at Various Loads
Figure 4.15 - Comparison of Maximum Rate of Cylinder Pressure Rise for Dual-Fuel and Straight Diesel Operation
Figure 4.16 - Maximum Rate of Cylinder Pressure Rise at Various Loads

- Fractional Diesel Energy Input (%)
- Injection Timing
- Max. Rate of Cyl. Press. (MPa/deg CA)

Legend:
- 143 kPa
- 278 kPa
- 428 kPa
- 571 kPa
- Bmp 713 kPa
One of the most serious features of dual-fuel operation with natural is the very large increase in mechanical loading of the engine, unless the pilot diesel injection is retarded. In section 4.2.3, it will be shown how peak pressures and maximum rates of pressure rise can be greatly reduced by retarding the injection timing.

4.2.2 Effect of Restricting Intake Air

The effect on maximum cylinder pressure of restricting intake air was considered for two load conditions, 428 kPa (50 percent of full load) and 571 kPa (67 percent of full load) in brake mean effective pressure. The flow rate of pilot diesel was 15 percent of total energy input. Fig. 4.17 shows the change in maximum cylinder pressure with restriction of intake air. As mentioned before the maximum amount of allowable air restriction was limited by the turbocharger surging characteristics. Substantial reduction in maximum cylinder pressure was achieved without exceeding the throttling limitation. With maximum air restriction when the load is below 67 percent of full load the maximum cylinder pressure does not exceed that of straight diesel operation at full load, which is about 8000 kPa.

Fig. 4.18 shows the change in pressure just before combustion with reduction of manifold pressure due to throttling. The pressure just prior to combustion was obtained
Figure 4.17 - Effect of Intake Air Restriction on Maximum Cylinder Pressure
Figure 4.18 - Effect of Intake Air Restriction Pressure Prior to Combustion

Flow rate of intake air: 4.6 cum/min
Pressure prior to combustion (MPa): 4.0, 4.4, 4.8, 5.2
BMEP 571 kPa, 4.28 kPa
15% diesel
from the cylinder pressure vs crank angle trace as the point just prior to the significant pressure rise due to combustion. For both of the load conditions the change in maximum cylinder pressure is of the same order of the change in the pressure prior to combustion. At the brake mean effective pressure of 428 kPa the reduction of air flow rate from 4.9 to 4.3 m³/min resulted in the drop in the pressure prior to combustion of 400 kPa and in maximum cylinder pressure of 500 kPa. At the brake mean effective pressure of 571 kPa the reduction is 450 kPa for the pressure prior to combustion and 500 kPa for the maximum cylinder pressure when the air flow is restricted to 4.38 from 5.01 m³/min. This seems to suggest that the reduction of maximum cylinder pressure when the intake air is restricted is due mainly to the decreased pressure prior to combustion (due to reduction in manifold pressure).

The effect on the maximum rate of cylinder pressure rise of intake air restriction is shown in Fig. 4.19. The restriction of intake air seems to result in higher maximum rate of cylinder pressure rise. This trend is believed to be the consequence of the increased gas-air mixture strength which favours flame propagation.
Figure 4.19 - Effect of Intake Air Restriction on Maximum Rate of Cylinder Pressure Rise
4.2.3 Effect of Varying Injection Timing

The effect on maximum cylinder pressure of retarding injection timing was considered for a brake mean effective pressure of 571 kPa. The flow rate of pilot diesel used was 10 percent of total energy input. The reduced maximum cylinder pressure as a result of injection timing retardation by 2 and 4 degrees CA are shown in Fig. 4.20. Retardation of merely 4 degrees reduced the maximum pressure by 1850 kPa. The change in the pressure prior to combustion for the corresponding retardation was 460 kPa as shown in Fig. 4.21.

Fig. 4.22 shows the P-V diagram for different injection timings. As the injection timing is retarded the point of the significant pressure rise is retarded further from the top dead centre exhibiting less steep and wider trace. The maximum rate of pressure rise was also reduced as shown in Fig. 4.20. The change in thermal efficiency due to injection timing retardation was less than 0.5 percent.

Test at higher load, at a brake mean effective pressure of 714 kPa (83% of full load), with 10 percent fractional diesel energy input and 4 degrees retardation showed reduction of peak pressure from 10.2 to 7.7 MPa and maximum rate of pressure rise from 2.1 to 1.4 MPa/deg. The loss in brake thermal efficiency due to the injection retardation was about 1 percent.

It is estimated that safe full-load dual-fuel operation would require 4 to 6 degrees of injection timing retardation and this would result in loss of thermal efficiency of 1 to
Figure 4.20 - Effect of Varying Injection Timing on Maximum Cylinder Pressure and Rate of Pressure Rise
Figure 4.21 - Effect of Varying Injection Timing on Pressure Prior to Combustion and Point of Ignition Start
Figure 4.22 - Effect of Varying Injection Timing on P-V Diagram
2 percent. At loads beyond full load, since it is expected that the thermal efficiency of dual-fuel operation without injection retardation would surpass that of straight diesel operation (see section 4.1.1), the dual-fuel operation with sufficient injection retardation to assure safe level of peak pressure would still result in thermal efficiency close to that of straight diesel operation. Retarding injection timing seems to be a practical, and necessary means of ensuring the safe dual-fuel operation at high load.
5.1 General

One of the most effective means of interpreting the combustion processes in internal combustion engines is the estimation of the rate of chemical energy release, or burning rate, from the measured pressure distribution. The combustion energy released affects the internal energy of the gas mixtures inside the cylinder, the heat transfer through the cylinder walls, and the work done on the piston head. By appropriate estimation of heat transfer and thermodynamic properties of gas mixtures, the apparent energy release due to combustion can be estimated from measured values of cylinder pressure and change in cylinder volume. The analysis provides a qualitative picture of the combustion processes of dual-fuel and straight diesel operation.
5.2 Method of calculation

5.2.1 Definitions, Equations, and Assumptions

When both the intake and exhaust valves are closed the mixtures of air and burned and unburned fuels can be considered as a system undergoing a change of state, which is bounded by cylinder walls and piston head (see Fig. 5.1). The first law of thermodynamics then can be applied to the system for a small time change $\delta t$:

\[
\text{first law} \quad \delta Q = dE + \delta W
\]

where

- $Q$ - heat transfer to the system
- $E$ - energy of the system
- $W$ - work done by the system

Since the only significant energies of the system involved here are the internal and chemical energy, the energy of the system can be assumed to consist of the following:

\[
E = U + CE
\]

where

- $U$ - internal energy of the system
- $CE$ - chemical energy of the system
Figure 5.1 - Control Volume for Apparent Energy Release Analysis
Then the corresponding first law becomes:

\[ \delta Q = dU + dCE + \delta W \]

For a finite change of time, \( \Delta t \), with the system undergoing a change from state \( i \) to state \((i+1)\), the first law can be integrated to yield the following:

\[ iQ_{i+1} = \Delta U + \Delta CE + i W_{i+1} \]

where

\[ \Delta U = U_{i+1} - U_i \]
\[ \Delta CE = CE_{i+1} - CE_i \]

If \( \overline{P} \) is defined to be the average pressure of the system:

\[ i\overline{P}_{i+1} = \frac{P_i + P_{i+1}}{2} \]

then the work done on the system can be approximated as:

\[ iW_{i+1} \approx i\overline{P}_{i+1} (V_{i+1} - V_i) \]

where \( V_i \) - cylinder volume at \( i^{th} \) state

The change in chemical energy of the system can be estimated as:

\[ \Delta CE \approx \sum_{j}^{} m_{fj} u_{cj} \]
where
\[ m_{fj} \] - mass of \( j \)th fuel burned during i\textsuperscript{th} state, (\( j = 1 \) for diesel, j = 2 for natural gas)
\[ u_{cj} \] - internal energy of combustion of \( j \)th fuel

Now a finite difference form of the first law may be written as:

\[ \sum m_{fj} u_{cj} = Q_{i+1} - \Delta U - \bar{P}_{i+1} \Delta V \] (Eqn 5.1)

Evaluation of these terms will be discussed in subsequent sections.

Assumptions

In developing the method of computation several assumptions were made, namely:

1. The constituents of the mixture in the cylinder behave as ideal gases with temperature-dependent thermodynamic properties.

2. The gaseous constituents of the mixture are considered to be homogeneous and uniform in thermodynamic state: spatial non-uniformity in the rate of chemical energy release is ignored.
3. The composition of the combustion products corresponds to equilibrium dissociation.

4. The continuous variation of thermodynamic properties with time can be adequately represented by stepwise variation over a small time interval corresponding to 1 degree crank angle.

5. Presence of residual gas during the intake is neglected.

6. Overlap of intake and exhaust valve is ignored.

7. At any given instant the burned fractions of the natural gas and diesel fuels are the same.

Heat Transfer

In order to account for the heat transfer between the gas mixture and the cylinder walls, both convective and radiative modes were considered, following the procedure of Annand (1963) whose equation is:

\[ \frac{q}{A} = a \frac{k}{D} (R) \frac{b}{b} (T - T_{\text{wall}}) + c (T^4 - T_{\text{wall}}^4) \]

where

- \( q \) - heat transfer rate
- \( A \) - surface area of cylinder walls
- \( k \) - thermal conductivity of the mixture
- \( D \) - bore
- \( R \) - Reynolds number defined as \( \rho \text{VD}/\mu \)
  
  where

- \( \rho \) - density of the mixture
\( \nabla \) - mean piston velocity
\( \mu \) - viscosity of the mixture

\( T \) - mixture temperature

\( T_{\text{wall}} \) - cylinder wall temperature

\( a, b, c \) - constants

The first term with the first order of temperature accounts for the convective heat transfer and the second term with the fourth order temperature for the radiative. The constants 'a' and 'b' for the convective term were selected to yield a fit with least-square-errors to the apparent heat transfer during the compression stroke. A nonlinear least-squares-fit technique was adopted in optimizing the two constants for a large set of apparent heat transfer data obtained (from equation 5.1 and measured pressures with no combustion and known constituents of the mixture) for dual-fuel and straight diesel operation over a range of loads. Fig. 5.2 shows the fitted curve and data for straight diesel operation at brake mean effective pressure of 571 kPa. The optimized values for the dimensionless constants 'a' and 'b' were 0.47 and 0.7 respectively. The values suggested by Annand(1963) for 'a' was 0.35-0.8 and for 'b' was 0.7.

The constant 'c' for the radiation term was taken to be \( 3.3 \times 10^{-11} \text{ kJ/K}^4 \) as suggested by Annand for diesel engines. This value would correspond to the product of the Stefan-Boltzmann constant \( \sigma \) and an emissivity of 0.58, appropriate to grey body radiation.
Figure 5.2 - Apparent heat transfer rate and heat transfer model

--- selected model

x apparent heat transfer rate

--- selected model

x apparent heat transfer rate

Speed - 1600rpm
Bump - 571 kPa
Straight diesel operation
It was assumed that the temperature of the wall, at a given load, stays constant throughout the cycle, and varies linearly with the applied load. The measurements of wall temperature made by Kamel and Watson (1979) on an indirect-injection Ricardo swirl engine showed that the change in wall temperature throughout the cycle at full load was less than 10 percent of the mean temperature. Their data also suggested that the wall temperature of both prechamber and main chamber varied nearly linearly with applied load. In the present work the wall temperature was calculated from

\[ T_{\text{wall}} = 0.071 \times \text{bmeP} + 540 \]

bmeP in kPa

\[ T_{\text{wall}} \text{ in K} \]

The numbers obtained from this formula for \( T_{\text{wall}} \) are well within 10 percent of those measured by Kamel and Watson at the same engine speed.

Fig. 5.3 shows a typical calculation of the apparent rate of energy release with and without the adopted heat transfer model; the computation procedure is presented later.

**Dissociation**

In computing the constituents of the combustion products, equilibrium dissociation was assumed. The dissociation reactions considered are as follows:
Figure 5.3 - Effect of Heat Transfer Model on Apparent Rate of Energy Release

- with heat transfer model
- without heat transfer model

straight diesel operation
bmep - 571 kPa
speed - 1600 rpm
It may be noted that in his engine mixture dissociation calculations, Campbell (1977) considered the above dissociations, and in addition the dissociation of $\text{O}_2$, $\text{H}_2$, and $\text{OH}$. As will be shown the degree of dissociation is small at the relatively high pressures and low temperatures of compression ignition engines. Hence only 4 dissociation processes were considered in the equilibrium calculation.

For each step of incremental time $\Delta t$ and given pressure and temperature of the mixture, the following set of nonlinear equations were solved for the number of moles of combustion products:

1. \[
\frac{Y_{\text{CO}}V_{\text{O}_2}^{1/2}}{Y_{\text{CO}_2}} = K_A\left(\frac{P^0}{P}\right)^{1/2} \\
\text{or} \\
\frac{(N_{\text{CO}}+A)(N_{\text{O}_2}+1/2(A+C-D))^{1/2}}{(N_{\text{CO}_2}-A)} (N_{\text{tot}})^{-1/2} = K_A\left(\frac{P^0}{P}\right)^{1/2}
\]

2. \[
\frac{Y_{\text{H}_2}V_{\text{OH}}^{1/2}}{Y_{\text{H}_2\text{O}}} = K_B\left(\frac{P^0}{P}\right)^{1/2} \\
\text{or} \\
\frac{(N_{\text{H}_2}+1/2B+C)^{1/2}(N_{\text{OH}}+B)}{(N_{\text{H}_2\text{O}}-B-C)} (N_{\text{tot}})^{-1/2} = K_B\left(\frac{P^0}{P}\right)^{1/2}
\]
3. \[
\frac{Y_{H_2O}}{Y_{H_2}} = K_C \left(\frac{P^o}{P}\right)^{1/2}
\]

or

\[
\frac{(N_{H_2} + 1/2B + C) \left(\frac{N_{O_2}}{2} + 1/2(A + C - D)\right)^{1/2}}{(N_{H_2O} - B - C)} \left(\frac{N_{tot}}{2}\right)^{1/2} = K_C \left(\frac{P^o}{P}\right)^{1/2}
\]

4. \[
\frac{Y_{NO}}{Y_{N_2O_2}} = K_D
\]

or

\[
\frac{(N_{NO} + D)}{(N_{N_2} - 1/2D) \left[\frac{N_{O_2}}{2} + 1/2(A + C - D)\right]} = K_D
\]

5. \[
N_{tot} = \sum N_i + 1/2(A + B + C)
\]

where \(K_A, K_B, K_C, K_D\) are equilibrium constants for reactions a,b,c,d.

\(Y_{CO}, Y_{NO}\), etc are equilibrium compositions.

\(N_{CO}, N_{NO}\), etc are the number of moles of constituents present prior to the current step of combustion.

\(A, B, C, D\) are numbers of moles of \(CO_2, H_2O, H_2O, N_2\) or \(O_2\) dissociated in reactions a,b,c,d, respectively.

\(P^o\) is atmospheric pressure, 101.3 kPa.

\(P\) is the gas mixture pressure.

\(N_{tot}\) is the total number of moles of the mixture.
Given the pressure and temperature of the mixture, the five equations were simultaneously solved for A, B, C, D, and $N_{\text{tot}}$ using a modified Newton's method. The values of the equilibrium constants, which are functions of temperature, were calculated from fitted curves based on thermodynamic data given in the JANAF Thermodynamics Tables.

Fig. 5.4 shows a typical calculation of the apparent rate of energy release computed with and without dissociation.
Figure 5.4 - Effect of Equilibrium Dissociation Calculation on Apparent Rate of Energy Release
5.2.2 Computation Procedure

The apparent energy release is obtained by solving the equations of mass and energy conservation for the mixture temperature and the fraction of fuel burned. The work done on the piston is computed from the smoothed pressure data and change in cylinder volume. The rate of heat transfer is estimated as described in section 5.2.1. The composition of the mixture is computed with the equilibrium dissociation assumption, and with the equations provided in the previous section.

Consider a small step in the calculation during which the state of the mixture changes from state \( i \) to state \( (i+1) \). The calculations for state \( (i+1) \) start with complete knowledge of state \( i \); the following conditions are given:

\[
T_i, P_i, (n_{CH_4})_i, (n_{C_{12}H_{26}})_i, \ldots, (n_j)_i
\]

where the \( n_j \)'s are the numbers of moles of \( CH_4, C_{12}H_{26}, N_2, O_2, H_2O, CO_2, CO, H_2, OH, NO \).

The calculation procedure is as follows:

1. Assume \( T_{i+1} \), \( (fr)_i \) to \( i+1 \) (fraction of fuel burned in one step).
2. Obtain the composition molar fractions \( (n_j)_{i+1} \) which would exist in state \( (i+1) \) were there no dissociation.
3. With the assumed $T_{i+1}$ and the measured pressure $P_{i+1}$ perform equilibrium dissociation calculations to obtain the values of $(n_j)_{i+1}$ which satisfy the dissociation relationships in section 5.2.1.

4. For each species calculate the change in the number of moles

$$\Delta n_j = (n_j)_{i+1} - (n_j)_i$$

5. Compute the changes in chemical energy $\Delta CE$ and internal energy $\Delta U$ as

$$\Delta CE = \sum_j \Delta n_j (u_f^o j + \Delta u_j)$$

where

$$u_f^o j = \text{internal energy of formation at 298 K}$$

$$\Delta u_j = u_j(T_{i+1}) - u_j(298K)$$

$$\Delta U = \sum_j (n_j)_i (u_j(T_{i+1}) - u_j(T_i))$$

where $u_j$ - internal energy of $j^{th}$ constituent of gas mixture

6. Compute $i Q_{i+1}$ and $i W_{i+1}$

7. Check the following two conservation equations:

$$\Delta CE = i Q_{i+1} + \Delta U + i W_{i+1}$$

$$P_{i+1} V_{i+1} = (\sum_j (n_j)_{i+1}) RT_{i+1}$$

8. If the above two equations are satisfied then computation is completed. If not, repeat from 1.
Cylinder Pressure Data

The cylinder pressure was recorded at every degree of crank angle. The measured values were then averaged over 30-50 randomly selected cycles. It required 20-30 minutes to obtain an averaged pressure trace of 30 cycles with the NEFF data acquisition unit and the PDP/11 computer. Because of large cycle-to-cycle variations, the resulting pressure-crank angle curves were not smooth enough to provide a smooth calculated curve of rate of energy release. The pressure-crank angle curves were smoothed by fitting a curve between the obtained data. The technique involved fitting a piece-wise cubic polynomial (continuous to the second derivative) between the pressure measurements over a crank angle range of 180 degrees with minimization in square errors. The inverse of the variation in the slope of the pressure trace for four neighbouring points were used in providing the weight for the least-squares fit.

Fig. 5.5 shows the rate of energy release calculated from unsmoothed and smoothed pressure-crank angle curves. The smoothed pressure curve showed very small visually detectable change, except in the region near the peak of the combustion pressure.

Computer Program for Apparent Energy Release

The main function of the computer program was to execute the computation procedure described in the previous section. The program initially read in the cylinder pressure data and flow
Figure 5.5 - Effect of Smoothing Pressure Data on Apparent Rate of Energy Release
rates of air and fuels along with data for operating conditions. The pressure data were then smoothed and volumes of the cylinder for all crank angles were computed. Subsequently, for each degree of crank angle, equations for energy and mass conservation were simultaneously and iteratively solved for the temperature and the fraction of fuel burned. A modified Newton's method was used in solving the system of nonlinear equations. The program assumed that combustion may take place at anytime after the injection of diesel fuel. Before the diesel injection point, the program took an alternate route and merely computed the mixture temperature directly from ideal gas law. Fig. 5.6 shows the flowchart of the procedures adopted in the computer program. A listing of the computer program is provided in Appendix E. Fig. 5.7 shows a typical output of the program.

Check of Computation

In order to confirm quantitatively the correctness of the method used, the computed amount of consumed fuel energy per averaged cycle from the computer output was compared with actual amount of fuel energy input. Because the computed rate of chemical energy release is essentially zero except for the crank angle interval of -10 to +90 degrees after top dead center, it was necessary to integrate the energy release only in this range.

Table 5.1 shows the ratios of the computed to actual energy consumed for various operations. From all the ratios shown in the table, it is seen that the agreements between the computed and
Figure 5.6 - Flowchart of Computer Program for Apparent Energy Release Analysis
Figure 5.7- Typical Output of Computer Program for Apparent Energy Release Analysis
<table>
<thead>
<tr>
<th>LOAD (kPa)</th>
<th>MODE OF OPERATION</th>
<th>FRACTIONAL DIESEL ENERGY INPUT (%)</th>
<th>ACTUAL ENERGY CONSUMED (kJ)</th>
<th>COMPUTED ENERGY CONSUMED (kJ)</th>
<th>RATIO OF COMPUTED TO ACTUAL ENERGY CONSUMED</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>straight diesel</td>
<td>-</td>
<td>0.79</td>
<td>0.83</td>
<td>1.06</td>
</tr>
<tr>
<td>279</td>
<td>straight diesel</td>
<td>-</td>
<td>1.74</td>
<td>1.67</td>
<td>0.96</td>
</tr>
<tr>
<td>571</td>
<td>straight diesel</td>
<td>-</td>
<td>3.05</td>
<td>2.84</td>
<td>0.93</td>
</tr>
<tr>
<td>713</td>
<td>straight diesel</td>
<td>-</td>
<td>3.75</td>
<td>3.48</td>
<td>0.93</td>
</tr>
<tr>
<td>856</td>
<td>straight diesel</td>
<td>-</td>
<td>4.58</td>
<td>4.26</td>
<td>0.93</td>
</tr>
<tr>
<td>571</td>
<td>dual-fuel</td>
<td>20.9</td>
<td>3.17</td>
<td>3.01</td>
<td>0.95</td>
</tr>
<tr>
<td>571</td>
<td>dual-fuel</td>
<td>10.2</td>
<td>3.14</td>
<td>2.95</td>
<td>0.94</td>
</tr>
</tbody>
</table>

Table 5.1 - Comparison of Actual and Computed Fuel Energy Consumed
actual energy consumed are quite good. The small disagreements are probably due to the variation in heating values of fuels, inadequacy of heat transfer model, unburned fuel, and various assumptions made during the course of method development. It should be noted that the amount of unburned gas escaping the cylinder was not subtracted from the actual amount of fuel energy input.
5.3 Analysis

5.3.1 Operations with Unmodified Engine

The calculated rate of energy release for straight diesel operation with loads ranging from idling to full load is shown in Fig. 5.8. At idling near the top dead center the curve declines to negative values prior to the peak. This may be the result of the inadequacy of the heat transfer model. At low loads the model seems to underestimate the rate of heat transfer, resulting in negative values for the apparent rate of energy release. The roughness of the curves during and towards the end of combustion is due to roughness in the pressure data. The roughness could have been reduced by further smoothing the pressure data.

At full load (bmept = 856 kPa) the rate of energy release reveals two stages of combustion. The first stage lasts until 12 to 14 degrees after top dead centre. As the load decreases the second stage combustion becomes less distinctive. It seems that the first peak corresponds to combustion in the prechamber and the second to combustion in the main chamber. The calculated cumulative energy release in Fig. 5.9 further supports this view. Near 13 degree C.A. after the top dead centre, which is approximately the point separating the two stages of combustion, the cumulative energy release for the loads other than idle is nearly the same at about 1.3 kJ. This is about 30% of the total cumulative energy release at full load. The volumetric ratio of prechamber to the total volume at
Figure 5.8 - Rate of Energy Release of Straight Diesel Operation at Various Loads
Figure 5.9 - Cumulative Energy Release of Straight Diesel Operation at Various Loads
the top dead centre is about 25 percent. Thus the analysis suggests that the combustion in straight diesel operation of prechamber engine consists of two distinct and subsequent stages—prechamber and main chamber. From Fig. 5.9 it can be seen that the time period during which the combustion takes place increases with the increase in load. The maximum rate of energy release is the smallest at full load and increases as the load is decreased. At very low loads the rate of energy release rises very rapidly although the maximum rate of energy release is limited by the total diesel energy input. Fig. 5.10 shows the dependence on air-fuel ratio of the maximum rate of energy release. It is seen that the rapidity of the combustion in straight diesel operation increases nearly linearly with air-fuel ratio.

Fig. 5.11 shows the rate of energy release with dual-fuel operation in which pilot diesel fuel accounts for about 20 percent of the total energy input, and when no change has been made in diesel injection timing. The most noteworthy feature is the nearly twofold increase in the maximum rate of energy release. It is observed from the figure that the change in combustion duration with varying load is relatively small when compared to that of straight diesel operation. The shapes of the rate of energy release curves exhibit very little of the two-staged combustion characteristic which is observed in straight diesel operation. The rapidity of rise of the rate of energy release increases with the increase in load. Fig. 5.12 shows the maximum rate of energy release plotted against the gas-air mixture strength. The figure indicates strong
In Straight Diesel Operation

Figure 5.10 - Effect of Air-Fuel Ratio on Maximum Rate of Energy Release

\[
\frac{\text{stoichiometric air-fuel mass ratio}}{\text{actual air-fuel mass ratio}} = \chi
\]

Max. Rate of Energy Release (k/J(deg.CA))

Bump - 856 kPa (full load)

426 kPa

714 kPa

571 kPa

0.10

0.15

0.20

0.25
Figure 5.11 - Rate of Energy Release of Dual-Fuel Operation at Various Loads
Figure 5.12 - Effect of Gas-Air Mixture Strength on Maximum Rate of Energy Release in Dual-Fuel Operation

\[ \Phi_g = \frac{\text{actual gas-air mass ratio}}{\text{stoichiometric gas-air mass ratio}} \]
dependence of the rapidity of combustion on the gas-air mixture strength. The cumulative energy release of dual-fuel operation corresponding to the curves in Fig. 5.11 is shown in Fig. 5.13.

Fig. 5.14 and Fig. 5.15 compare straight diesel and dual-fuel operation at 85 and 33 percent of full load. From Fig. 5.14 it is seen that for high load operation the combustion duration of the dual-fuel operation is much shorter than that of the straight diesel operation, and the maximum rate of energy release is about 3.7 times higher. The remarkable difference in the shape of the rate of energy release for the two modes of operations supports the view mentioned in section 4.2.1 that the mechanisms of combustion are different. Evidently the combustion in dual-fuel operation is mainly carried out by flame propagation through the premixed gas-air mixture, ignited by the burning diesel spray which penetrates the main chamber as a hot jet. In straight diesel operation, combustion occurs in the fuel-air mixture adjacent to evaporating fuel drops rather than propagating as a turbulent flame through the entire mixture. Since most of the fuel does not require evaporation in the former case it is reasonable that this mode of combustion should be faster.

Fig. 5.16 shows the rate of energy release for a low load operation (brake mean effective pressure of 279 kPa) with various flow rates of pilot diesel. The apparent ignition points, which may be identified as the points where the curves start rising, agree quite well with those observed from the pressure-crank angle trace. The ignition delay is increased as
Figure 5.13 - Cumulative Energy Release of Dual-Fuel Operation at Various Loads

Crank Angle (deg ATDC)

-100
tdc
30.0
10.0
5.0
0.0
30
60
90
120
150
180
210
240
270
300
330
360

Cumulative Energy Release (kJ)

35
30
25
20
15
10
5
0
-0.5
-1.0
-1.5
-2.0
-2.5
-3.0
-3.5
-4.0

9.2 KPa
7.7 KPa
6.7 KPa
5.1 KPa
Bump - 7.1 KPa

Approx. 20% diesel operation
Figure 5.14 - Comparison of Rate of Energy Release for Straight Diesel and Dual-Fuel Operation
Figure 5.15 - Comparison of Cumulative Energy Release for Straight Diesel and Dual-Fuel Operation
Figure 5.16 - Rate of Energy Release of Dual-Fuel Operation at Various Pilot Diesel Flow Rates
much as 2 degree C.A. as the flow rate of pilot diesel is decreased from 100 to 13.5 percent of total energy input. The first stage of combustion in straight diesel operation consumes about 70 percent of the diesel energy input (see Fig. 5.17). A reduction of pilot diesel to 32 percent of total energy input results in a higher peak for rate of energy release with no indication of the two-staged combustion. Further reduction of pilot diesel results in lower and wider peaks. Fig. 5.18 shows the fraction of fuel burned at different pilot quantities. The operation with low pilot diesel flow rate shows a large amount of unburned fuel. The amount of unburned fuel decreases as the flow rate of pilot diesel is increased.

Fig. 5.19 shows the rate of energy release for a high load operation (brake mean effective pressure of 571 kPa). Reduction of pilot diesel flow rate appears to increase ignition delay and decrease the maximum rate of energy release. The cumulative energy release shown in Fig. 5.20 indicates that at this load combustion occurs in two stages at all pilot diesel energy ratios. The small second-stage combustion in high load dual-fuel operation seems to be the consumption, in the main chamber, of the unburned and/or partially burned fuel remaining from the flame propagation in the first stage.
Figure 5.17 - Cumulative Energy Release of Dual-Fuel Operation at Various Pilot Diesel Flow Rates
Figure 5.18 - Fraction of Fuel Burnt in Low Load Dual-Fuel Operation
Figure 5.19 - Rate of Energy Release of Dual-Fuel Operation at Various Pilot Diesel Flow Rates
Figure 5.20 - Cumulative Energy Release of Dual-Fuel Operation at Various Pilot Diesel Flow Rates
5.3.2 Effect of Restricting Intake Air

Fig. 5.21 shows the rate of energy release for low load operation with and without air restriction. It indicates that restriction of intake air lengthens ignition delay by 1-2 degrees. Table 5.2 illustrates the temperature and pressure of mixture at top dead centre with and without restriction of air. When the intake air is restricted a drop in the pressure at top dead centre is noticed. A rather surprising phenomenon is that there appears from the calculation to be an increase in temperature. This is contrary to the assumption that the temperature would drop as the pressure drops, which was the basis used by Lewis (1953) in explaining the increase in ignition delay when the restriction of air is imposed. It seems that the increase in ignition delay is not the result of the change in chemical delay since the chemical delay would be shortened if the temperature is increased. This would suggest that the change in ignition delay is probably more sensitive to the change in physical delay. Fig. 5.22 shows the cumulative energy release for the corresponding operations.

The effect of air restriction for high load operation is shown in Fig 5.23 and Fig. 5.24. The operation at the brake mean effective pressure of 571 kPa (67% full load) exhibits a radical increase in maximum rate of energy release when the intake air is restricted. The gas-air mixture strength for the corresponding operation is 0.606 in equivalence ratio. This value is slightly above the previously mentioned value for the lower limit of flammability. Thus the sudden increase in the
Figure 5.21 - Effect of Restricting Intake Air on Rate of Energy Release

- ----- bmep = 279 kPa, air restricted ($\bar{\phi}_g = 0.400$), 14.5% diesel
- ----- bmep = 279 kPa, air unrestricted ($\bar{\phi}_g = 0.364$), 13.5% diesel
- ----- bmep = 143 kPa, air restricted ($\bar{\phi}_g = 0.308$), 20.9% diesel
- ----- bmep = 143 kPa, air unrestricted ($\bar{\phi}_g = 0.279$), 19.5% diesel

Figure 5.21 - Effect of Restricting Intake Air on Rate of Energy Release
<table>
<thead>
<tr>
<th>LOAD (kPa)</th>
<th>AIR RESTRICTION</th>
<th>FRACTIONAL DIESEL ENERGY INPUT (%)</th>
<th>$\phi_{gas}$</th>
<th>$T_{tdc}$ (K)</th>
<th>$P_{tdc}$ (kPa)</th>
<th># OF MOLES OF INTAKE MIXTURE</th>
</tr>
</thead>
<tbody>
<tr>
<td>143</td>
<td>unrestricted</td>
<td>19.5</td>
<td>0.279</td>
<td>968</td>
<td>4810</td>
<td>0.0631</td>
</tr>
<tr>
<td>143</td>
<td>restricted</td>
<td>20.9</td>
<td>0.308</td>
<td>999</td>
<td>4370</td>
<td>0.0555</td>
</tr>
<tr>
<td>279</td>
<td>unrestricted</td>
<td>13.5</td>
<td>0.364</td>
<td>951</td>
<td>4860</td>
<td>0.0648</td>
</tr>
<tr>
<td>279</td>
<td>restricted</td>
<td>14.5</td>
<td>0.400</td>
<td>1010</td>
<td>4470</td>
<td>0.0564</td>
</tr>
<tr>
<td>429</td>
<td>unrestricted</td>
<td>15.0</td>
<td>0.437</td>
<td>944</td>
<td>5000</td>
<td>0.0673</td>
</tr>
<tr>
<td>429</td>
<td>restricted</td>
<td>15.3</td>
<td>0.483</td>
<td>1010</td>
<td>4680</td>
<td>0.0586</td>
</tr>
<tr>
<td>571</td>
<td>unrestricted</td>
<td>10.2</td>
<td>0.526</td>
<td>955</td>
<td>5210</td>
<td>0.0692</td>
</tr>
<tr>
<td>571</td>
<td>restricted</td>
<td>10.7</td>
<td>0.606</td>
<td>1010</td>
<td>4780</td>
<td>0.0601</td>
</tr>
</tbody>
</table>

Table 5.2 - Effect of Intake Air Restriction on Mixture Temperature at Top Dead Center
Figure 5.22 - Effect of Restricting Intake Air on Cumulative Energy Release
Figure 5.23 - Effect of Restricting Intake Air on Rate of Energy Release

Rate of Energy Release (kJ/deg ATDC) vs. Crank Angle
Figure 5.24 - Effect of Intake Air Restriction on Cumulative Energy Release

- bmep - 429 kPa, air unrestricted ($\bar{\phi} = 0.437$), 15.0\% diesel
- bmep - 429 kPa, air restricted ($\bar{\phi} = 0.483$), 15.3\% diesel
- bmep - 571 kPa, air unrestricted ($\bar{\phi} = 0.526$), 10.2\% diesel
- bmep - 571 kPa, air restricted ($\bar{\phi} = 0.606$), 10.7\% diesel
maximum rate of energy release seems to be the result of fuel-air ratio on propagation of a flame through a premixed gas-air mixture.

5.3.3 Effect of Varying Injection Timing

Fig. 5.25 and Fig. 5.26 show the effect of advancing the injection timing at low load. Advancing the injection timing by 10 degree C.A. results in the maximum rate of energy release occurring at top dead center. For this timing the second stage combustion is more distinct. For the operations with injection timing of 12.3 and 17.3 degree C.A. BTDC the combustion is taking place only after the piston has started moving downwards, thus assisting the flow of mixture from prechamber to the main chamber. For the operation with injection timing of 22.3 degree C.A. BTDC, the combustion takes place when the piston is nearly motionless, and the second stage combustion would be more distinctive.

The effect of retarding injection timing at high load is shown in Fig 5.27 and Fig. 5.28. Significant reduction in the maximum rate of energy release is observed when the timing is retarded. Again the two-staged combustion characteristics become less distinctive as the timing is retarded.
Figure 5.25 - Effect of Advancing Injection Timing on Rate of Energy Release

bmep - 279 kPa, 20% diesel

- injection at 12.3 BTDC
- injection at 17.3 BTDC
- injection at 22.3 BTDC
Figure 5.26 - Effect of Advancing Injection Timing on Cumulative Energy Release.

Cumulative Energy Release (kJ)

-0.5 0.0 0.5 1.0 1.5 2.0 2.5 3.0 3.5 4.0

0.0 300 Crank Angle (deg ATDC)

-100 tdc

--- bme = 279 kPa, 208 diesel injection at 12.3 BTDC

--- 17.3 BTDC

--- 22.3 BTDC

132
Figure 5.27 - Effect of Retarding Injection Timing on Rate of Energy Release
Figure 5.28 - Effect of Retarding Injection Timing on Cumulative Energy Release
6.1 Conclusions

Safe operation of a prechamber diesel engine with dual-fuelling with natural gas is severely limited by maximum cylinder pressure. Even at half load the maximum cylinder pressure of dual-fuel operation is as high as that of straight diesel operation at full load. The excessive maximum cylinder pressure is associated with the high rate of energy release by combustion which takes place within a nearly homogeneous gas-air mixture. The maximum cylinder pressure, as well as the rate of cylinder pressure rise, can be reduced to a safe level by retarding the injection timing by about 4 to 6 degrees of crank angle. The change in thermal efficiency due to the retardation is small (less than 0.5 percent with 4 degree retardation at 67 percent of full load). Restricting the intake air can reduce also the maximum cylinder pressure, but this results in higher maximum rate of cylinder pressure rise.

Stable dual-fuel operation requires sufficient flow rate of pilot diesel fuel; insufficient amount of pilot diesel fuel results in erratic operation with misfired cycles. The minimum pilot diesel fuel required in order to ensure a stable operation is typically 8 to 15 percent of the total energy input, depending on engine load.
Dual-fuel operation at part load showed generally higher fuel consumption than that of straight diesel operation. The apparent energy release analysis revealed that the higher fuel consumption rate is mainly due to gas surviving unburned through the combustion chamber. The main cause of this poor combustion is weak gas-air mixture strength. Increase in pilot diesel flow rates reduces the amount of unburned gas and thus improves the fuel consumption rate. The dependence of total fuel consumption rate on pilot diesel flow rate is less with higher gas-air mixture concentration. With restriction on turbocharger inlet pressure (to prevent surge) less than about 10 percent air restriction was possible in the tests conducted; this showed some improvement (perhaps one percent) on fuel consumption rate. Advancing injection timing showed no significant effect on fuel consumption rate.

The fuel consumption rate during dual-fuel operation near full load approached that of straight diesel operation. Extrapolated beyond full load diesel operation, the fuel consumption rate of dual-fuel operation would become lower than that of straight diesel operation.

The combustion characteristics of straight diesel and dual-fuel operation differ in that in the former the combustion consists mainly of auto-ignition of diesel fuel, whereas in the latter the combustion is carried out by the propagation of flame fronts. This distinction was clearly exhibited in the analysis of apparent energy release, where straight diesel operation showed a two-staged combustion and dual-fuel operation a
shorter, single-staged combustion.
6.2 Recommendations

• Without difficult modifications the current engine can be converted into a direct-injection engine. Further tests on this converted engine would lead to comparative study between prechamber and direct-injection engines.

• Further studies at different engine speeds are required.

• Further study of the effect of restricting air intake without the turbocharger may show some improvement in fuel consumption at low loads.

• Exhaust gas analysis would provide some important information such as the lower limit of flammability.
BIBLIOGRAPHY


33. OBERT, E.F., Internal Combustion Engines and Air Pollution, 3rd ED., Harper & Row, 1973


APPENDIX A - CALIBRATION CURVES

Load Sensor

* slope = 22.2 N/mV
Air Flow Element

\[
\text{Volumetric Air Flow Rate (l/sec)} \quad \text{Differential Pressure (kPa)}
\]

* slope = 104.3 l/sec-kPa
Cylinder Pressure Transducer

Pressure (MPa)

Charge (pC)

* slope = 0.00831 MPa/pC
APPENDIX B - COMPUTATION OF INDICATED MEAN EFFECTIVE PRESSURE

Definition

An indicated mean effective pressure, imep, is defined as that theoretical constant pressure which can be imagined exerted during each power stroke of the engine to produce work equal to the indicated work:

\[ \int_{V_1}^{V_2} P \, dV = P_{\text{ind}} \Delta V \]

where
- \( P \) - cylinder pressure
- \( P_{\text{ind}} \) - indicated mean effective pressure, imep
- \( V \) - cylinder volume
- \( V_1 \) - \( V \) at the beginning of the cycle
- \( V_2 \) - \( V \) at the end of the cycle
- \( \Delta V = V_{\text{bdc}} - V_{\text{tdc}} \)

Indicated mean effective pressure, in this project, was computed according to the following:

\[ P_{\text{ind}} = \frac{\sum_{i=1}^{n} \left( \frac{P_i + P_{i+1}}{2} \right) (V_{i+1} - V_i)}{(V_{\text{bdc}} - V_{\text{tdc}})} \]

where \( P_i, V_i \) are cylinder pressure and volume at \( i \)th stage, and each stage corresponds to a degree of crank angle throughout a cycle.
Indicated Mean Effective Pressure of Straight Diesel Operation

![Graph showing the relationship between indicated mean effective pressure (kPa) and brake mean effective pressure (kPa)].
Listing of APP.PROG2 at 22:10:05 on APR 11, 1984 for CCid=AFPH

C Acquires data from Diesel Engine
C
EXTERNAL QTQIO
EXTERNAL GETADR
EXTERNAL ASNLUN

INTEGER LIST(3002), IDAT(3002), IPARM(6), ISTAT(2)
INTEGER YES, NO, ANS, IPARR(5), IBDC(3)
REAL LOAD, PMEAN(721), VOLUME(5)

YES = 1
NO = 0
SCALE = -32768.0
NPOINT = 3001
NPCYC = 720
NPCYCE = 750
NPCYCS = 700
NPCY2 = 360
STROKE = 6.0
ARM = 3.0
ROD = 9.595
VCLEAR = 6.444
PTAREA = 3.1416 * (4.75 / 2.0) ** 2
NPC1 = 721
FNPC1 = FLOAT(NPC1)
NPCY3 = NPCY2 + 50
RDPDEG = 0.0174533
PTH = 400.0
PMINRF = 14.7
STHR = 0.6

DO 5 I=1,3002
   IDAT(I) = 0
   CONTINUE

DO 6 I=1,NPC1
   PMEAN(I) = 0.0
   CONTINUE

CALL ASNLUN(3, 'NI', 0)
CALL ASSIGN*1, 'PAR2.DAT')

READ(1,100) CLOCK
DWELL = 1. / CLOCK
HERTZ = 1. / XRATE(DWELL, IRATE, IPRSET, 1)
CALL CLOCKB(IRATE, IPRSET, 1, IND, 1)
DELT = 1. / HERTZ
WRITE(5,200) IND, HERTZ

READ(1,101) NCHAN
READ(1,102) NDPCH

NDTOT = NPOINT
ISTRT = 1
ILAST = NDTOT + 1

C read in port address
READ(1,103) LIST(1)
read in scan instructions
DO 10 I=2,NCHAN+1
CALL GETADR(IPARM(1), IDAT)
CALL WTQI0("1002,3,10,1,ISTAT,IPARM,IDS)
WRITE(5,201)
WRITE(5,202) ISTAT(1), ISTAT(2), IDS
C
IRSA = 1
C write scan list to RAM, read back and check
IPARM(2) = (ILAST-ISTRT+1) * 2
IPARM(3) = IRSA
CALL GETADR(IPARM(1), LIST(ISTART))
CALL WTQI0("400,3,10,1,ISTAT,IPARM,IDS)
C read back
CALL GETADR(IPARM(1), IDAT(ISTRT))
CALL WTQI0("1000,3,10,1,ISTAT,IPARM,IDS)
C print any discrepancies
IERR = 0
DO 20 1=ISTRT,ILAST
IF (IDAT(I) .EQ. LIST(I)) GO TO 20
IERR = IERR + 1
WRITE(5,203) LIST(I), IDAT(I)
CONTINUE
WRITE(5,204) IERR
WRITE(5,202) ISTAT(1), ISTAT(2), IDS
C acquire data
IWCN = NDTOT + 1
CALL IDATE(ID1, ID2, ID3)
WRITE(5,205) ID1, ID2, ID3
C calibration of Pressure Measurement
C
PCPPSI = 0.830
CHMUPV = 1000.0
PCPMU = 1.415
GAIN = 1.0
CIP = CHMUPV * PCPMU / PCPPSI / SCALE * GAIN
DO 600 I=1,100
WRITE(5,270)
READ(5,170) SPEED
FREQRQ = SPEED / 60.0 / 2.0 * FLOAD(NPCYC) * 2
DMELL = 1. / FREQRQ
HERTZ = 1. / IRATE(DMELL, IRATE, IPRSET, 1)
CALL CLOCKB(IRATE, IPRSET, 1, IND, 1)
DELN = 1. / Hertz * 2
DIPDEG = DELT
WRITE(5,200) IND, Hertz
NCYCLE = 10
Listing of APP.PROG2 at 22:10:05 on APR 11, 1984 for CCid=AFPH
Listing of APP.PROG2 at 22:10:05 on APR 11, 1984 for CCid=AFPH

117 WRITE(5,271) NCYCLE
118 READ(5,171) NCYCLE
119 IPMAX = 0
120 FNCYC = FLOAT(NCYCLE)
121 NSET = 1
122 DO 610 1610=1,NPC1
123 PMEAN(1610) = 0.0
124 610 CONTINUE
125 PMAX0 = 0.0
126 DPDTM0 = 0.0
127 RPMIN0 = 0.0
128 RIMEPO = 0.0
129 SUMRP = 0.0
130 SUN2RP = 0.0
131 SUM1M = 0.0
132 SUMPX = 0.0
133 SUM2PX = 0.0
134 SUMPN = 0.0
135 SUM2PN = 0.0
136 SUMDN = 0.0
137 SUN2DN = 0.0
138 WRITE(5,269)
139 READ(5,170) PINTAK
140 PINTAK = PINTAK + 14.7
141 WRITE(5,210)
142 READ(5,110) CR
143 DO 650 M650=1,500
144 IF (NSET .GT. NCYCLE) GO TO 651
145 CALL GETADR(I  PARK(1), 
146 I DAT)
147 IPARM(2) =  IWCT *2
148 IPARM(3) = IRSA
149 CALL WTQIOC3001 ,3,10, 
150 1, I STAT,IPARM,IDS)
151 MB = 3
152 MF = 751
153 DO 680 M680=1,2
154 DO 681 M=MB,MF,2
155 S = ABS(S/SCALE)
156 IF (S .LT. STHR) GO TO 681
157 P = IDAT(M+359)
158 P = P * C1P
159 IF (P .LT. PTHR) GO TO 682
160 IBDC(1) = M
161 GO TO 683
162 682 CONTINUE
163 MB = M + 680
164 MF = MB + 80
165 GO TO 680
166 681 CONTINUE
167 680 CONTINUE
168 GO TO 650
169 683 CONTINUE
170 INDXRD = 2
171 MB = IBDC(1) + 680
172 MF = MB + 80
Listing of APP.PROG2 at 22:10:05 on APR 11, 1984 for CCid=AFPH

175     DO 685 M=685*1,2
176     DO 686 M=MB,MF,2
177     S = IDAT(M)
178     S = ABS(S/SCALE)
179     IF (S .LT. STHR) GOTO 686
180     IBDC(INDXBD) = M
181     MB = M + 680
182     MF = MB + 80
183     INDEXBD = INDEXBD + 1
184     GO TO 685
185 686     CONTINUE
186     GO TO 650
187 685     CONTINUE
188     IDIFFA = IBDC(2) - IBDC(1)
189     IDIFFB = IBDC(3) - IBDC(2)
190     MB = IBDC(2) + 681
191     MF = MF + 80
192     PMIN = 0.0
193     CONTINUE
194     IDIFF = IDIFFA + IDIFFB
195     RPMIND = 60.0 / (FLOAT(IDIFF) * DELT) * 4.0
196     MM = IBDC(3) - 81
197     DO 625 M=1,40
198     PM = IDAT(MM)
199     PMIN = PMIN + PM
200     MM = MM + 2
201 625     CONTINUE
202     PMIN = PMIN / 40.0 * C1P
203     CONTINUE
204     IBDC1P = IBDC(1) - 1
205     IBDC3P = IBDC(3) - 1
206     PMAX = 0.0
207     DPDTMX=0.0
208     RIMEP = 0.0
209     THETA = -180.0
210     DTHETA = 720. / FLOAT(IDIFF) * 2.0
211     DO 632 J1=IBDC1P,IBDC3P,2
212     PJ1P1 = IDAT(J1+2)
213     PJ1 = IDAT(J1)
214     PJ1P1 = PJ1P1 * C1P - PMIN + PINTAK
215     PJ1 = PJ1 * C1P - PMIN + PINTAK
216     DPDT = ABS((PJ1P1-PJ1) / DELT) * DTPDEG
217     IF (DPDT.GT. DPDTMX) DPDTMX = DPDT
218     RAD1 = THETA * RDPDEG
219     RAD2 = (THETA + DTHETA) * RDPDEG
220     X1 = -ARM * COS(RAD1)
221     1 X2 = -ARM * COS(RAD2)
222     - SQRT(ROD*ROD - (ARM * SIN(RAD1)) ** 2)
223     - SQRT(ROD*ROD - (ARM * SIN(RAD2)) ** 2)
224     PAVER = (PJ1 + PJ1P1) / 2.0
225     RIMEP = RIMEP + PAVER / STROKE * (X2 - X1)
226     IF (PJ1 .GT. PMAX) PMAX = PJ1
227     Theta = Theta + DTHETA
228     CONTINUE
229     WRITE(5,914) IBDC1P, IBDC3P
230 914     FORMAT(15,' Cycle lies between ','15,' <--- ',15,' degrees ')
Listing of APP.PROG2 at 22:10:05 on APR 11, 1984 for CCid=AFPH

233    WRITE(5,274) PMAX, PINTAK, RIMEP, DPDTMX
234    C
235    WRITE(5,279)
236    READ(5,120) ANS
237    IF (ANS .EQ. NO) GO TO 650
238    FIBDC1 = FLOAT(IBDC1)
239    DO 637 J1=1,721
240      FFJ1 = FLOAT(J1-1) / DTHETA * 2.0 + FIBDC1
241      INDX = FFJ1 / 2.0
242      INDX = INDX * 2
243      RINDX = (FFJ1 - FLOAT(INDX)) / 2.0
244      PINDX = IDAT(INDX)
245      PINDX = PINDX * C1P - PMIN + PINTAK
246      P = PINDX * RINDX * (PINDX1 - PINDX)
247      PMEAN(J1) = PMEAN(J1) + P / FNCYC
248    CONTINUE
249    PMAX0 = PMAX0 + PMAX
250    RIMEPO = RIMEPO + RIMEP
251    DPDTMO = DPDTMO + DPDTMX
252    RPM1NO = RPM1NO + RPMIND
253    SUMRP = SUMRP + RPMIND
254    SUM2RP = SUM2RP + RPMIND*RPMIND
255    SUM1M = SUM1M + RIMEP
256    SUM21M = SUM21M + RIMEP*RIMEP
257    SUMPX = SUMPX + PMAX
258    SUM2PX = SUM2PX + PMAX*PMAX
259    SUMPN = SUMPN + PMIN
260    SUM2PN = SUM2PN + PMIN*PMIN
261    SUMDP = SUMDP + DPDTMX
262    SUM2DP = SUM2DP + DPDTMX*DPDTMX
263    NSET = NSET + 1
264    IF (FNCYC1 .EQ. 0) FNCYC1=1
265    SDIMEP = SQRT((SUM2IM-SUM1M*SUM1M/FNCYC)/FNCYC1)
266    SDPMAX = SQRT((SUM2PX-SUMPX*SUMPX/FNCYC)/FNCYC1)
267    SDPMIN = SQRT((SUM2PN-SUMPN*SUMPN/FNCYC)/FNCYC1)
268    SDPDT = SQRT((SUM2DP-SUMDP*SUMDP/FNCYC)/FNCYC1)
269    SDRPM = SQRT((SUM2RP-SUMRP*SUMRP/FNCYC)/FNCYC1)
270    WRITE(5,275)
271    WRITE(5,276) NCYCLE, SPEED, HERTZ
272    WRITE(5,277)
273    WRITE(5,278) RPM1NO, PMAX0, PINTAK, RIMEPO, DPDTMO
274    WRITE(5,286) SDRPM, SDPMAX, SDPMIN, SDIMEP, SDDPDT
275    C
276    CONTINUE
277    WRITE(5,290)
278    READ(5,120) ANS
279    IF (ANS .EQ. NO) GO TO 998
Listing of APP.PROG2 at 22:10:05 on APR 11, 1984 for CCid=AFPH

291       600 CONTINUE
292              C
293       998 CONTINUE
295       WRITE(5,291)
296       READ(5,120) ANS
297       IF (ANS .EQ. NO) GO TO 999
298       CALL ASSIGN(2, 'P.DAT')
299       WRITE(5,292)
300       READ(5,192) NDEG
301       IPVDMG = NO
302       INTERD = NDEG / 180
303       IF (IPVDGM .EQ. YES) INTERD = INTERD * 2
304       NDAT = NDEG / INTERD + 1
305       NDEG2 = NDEG / 2
306       ITDC = 181
307       IPBEG = ITDC - NDEG2
308       IPEND = ITDC + NDEG2
309       IF (NEG .EQ. 720) IPBEG = 1
310       IF (NEG .EQ. 720) IPEND = 721
311       WRITE(2,700)
312              PMAX = 0.0
313              NSKIP = 5 * INTERD
314       DO 660 1660=IPBEG,IPEND,NSKIP
315                   JB = 1660
316                   JF = 1660 + NSKIP - 1
317                   JL = 1
318       DO 661 J=JB,JF,INTERD
319                      P = PMEAN(J)
320                      IPARR(JL) = P
321                      JL = JL + 1
322       661 CONTINUE
323              JLM1 = JL - 1
324       WRITE(2,701) (IPARR(LL), LL=1,JLM1)
325       660 CONTINUE
326              MINX = IPBEG - 181
327              MAXX = IPEND - 181
328       WRITE(2,702)
329       IF (IPVDGM .EQ. YES) GO TO 670
330       WRITE(2,703) NDAT, MINX, INTERD
331       WRITE(2,704) MINX, MAXX
332       WRITE(2,705)
333       WRITE(2,706) RPMIN0
334       WRITE(2,707)
335       GO TO 674
336       670 CONTINUE
337       WRITE(2,710)
338          THETA = MINX
339          FINTER = FLOAT(INTERD)
340          THETA = THETA - FINTER
341       DO 662 1662=IPBEG,IPEND,NSKIP
342                   JB = 1662
343                   JF = 1662 + NSKIP - 1
344                   JL = 1
345       DO 663 J=JB,JF,INTERD
346          THETA = THETA + FINTER
347          RAD = THETA * RDPDEG
348             X = ROD + ARM - ARM * COS(RAD)
Listing of APP.PROG2 at 22:10:05 on APR 11, 1984 for CCid=AFPH

1 - SQRT(ROD * ROD - (ARM * SIN(RAD)) ** 2)

VOLUME(JL) = X * PTAREA + VCLEAR

JL = JL + 1

WRITE(2,711) (VOLUME(LL), LL=1,JLM1)

WRITE(2,706) RPMINO

WRITE(2,717)

WRITE(2,708)

WRITE(2,709)

999 CONTINUE

STOP

C

100 FORMAT(1X,F16.5)

101 FORMAT(1X,12)

102 FORMAT(1X,14)

103 FORMAT(05)

110 FORMAT(A5)

120 FORMAT(11)

150 FORMAT(F12.4)

170 FORMAT(F7.2)

171 FORMAT(I12)

180 FORMAT(F5.1,I5,F5.3,I5,F5.1)

192 FORMAT(I13,I12)

195 FORMAT(I11)

C

200 FORMAT(I1I3,F8.1,' Hz')

201 FORMAT(I1I3,SERIES 500 BUS RESET!!!')

202 FORMAT(I1DRIVER COMPLETION CODE = ',06,' (OCTAL)' )

203 FORMAT(I1DIRECTIVE STATUS = ',06,' (OCTAL)' )

204 FORMAT(I1WRITE TO RAM AND READBACK COMPLETE',I13,' ERRORS',)

205 FORMAT(I1date: ',I13,I12,I12,' )

206 FORMAT(I1?'To start scanning, enter RETURN!',$)

207 FORMAT(I1,2X,E14.6)

208 FORMAT(I1,3(2X,FB.3),4X,F11.2)

209 FORMAT(I1' # of Cycles Engine Speed Data aqusit Freq',/)

210 FORMAT(I1,1I3,' (cycles) (rpm) (Hz) ')

211 FORMAT(I1P max (psi) P intake IMEP dpdt max (psi/deg')

212 FORMAT(I1,3(2X,F8.3),4X,F11.2)

213 FORMAT(I1,' Ideal # of Cycles is ',12,' Enter desired # 12 ',

214 FORMAT(I1,' # of data points in ',13,'th cycle is ',14,'/

215 FORMAT(I1Indicated Engine Speed is ',F6.1,' rpm')

216 FORMAT(I1,' Mean: Indicated Speed P max P intake IMEP

217 FORMAT(I1,1I3,' cycles) (rpm) (Hz) '

218 FORMAT(I1P max P intake IMEP dpdt max '

219 FORMAT(I1,1I3,F8.3,5X(11.2))

220 FORMAT(I1,' Do you want to select this cycle? (1/0):',$)

221 FORMAT(I1,'?? To start scanning, enter RETURN!',$)

222 FORMAT(I1,15X,' # of Cycles Engine Speed Data aqusit Freq',/)

223 FORMAT(I1,15X,' (cycles) (rpm) (Hz) ')

224 FORMAT(I1P max P intake IMEP dpdt max (psi/deg')

225 FORMAT(I1,3(2X,F8.3),4X,F11.2)

226 FORMAT(I1,' # of Cycles Engine Speed Data aqusit Freq',/)

227 FORMAT(I1,15X,' (cycles) (rpm) (Hz) ')

228 FORMAT(I1P max P intake IMEP dpdt max ',')

229 FORMAT(I1,' Do you want to select this cycle? (1/0):',$)
280 FORMAT(1X, '>>------- Enter M.U. per V, pC per M.U., Gain:', $)
286 FORMAT(1X, 'Stand. Dev:', F8.3, 5X, 5(F11,3))
290 FORMAT(1X, 'Enter Crank Angle Range 13:', $)
295 "Do you want a P-V diagram? Enter 1 or 0:', $)
299 FORMAT(' Cycle NOT found .', $)
300 FORMAT(1X, 'EN PRES &')
301 FORMAT(1X, 'EN VOL &')
302 FORMAT(1X, 'EN ANGL SHOR ')
304 FORMAT(1X, 'GR ANGL PRES;YR -200 1600;XR ', I3, 1X, 13, ' ;&')
305 FORMAT(1X, 'GR VOL PRES;YR -200 1600;XR 5 115;&')
306 FORMAT(1X, 'TI ''Cylinder pressure vs Crank Angle'';&')
307 FORMAT(1X, 'TI ''Cylinder pressure vs Volume'';&')
308 FORMAT(1X, 'DA ''F6.1, rpm'';&')
309 FORMAT(1X, 'XTIT ''Crank Angle (deg)'', &')
310 FORMAT(1X, 'XTIT ''Volume (cu. in.)''; &')
311 FORMAT(1X, 'YTIT ''Pressure psi''; &')
312 FORMAT(1X, 'YG;XG')
313 END
APPENDIX D - COMPUTER PROGRAM FOR DATA PROCESSING
This program processes data from diesel engine

EXTERNAL WTQIO
EXTERNAL GETADR
EXTERNAL ASNLUN

INTEGER LIST(200), IDAT(200), IPARM(6), ISTAT(2)
INTEGER YES, NO, ANS, IACTIV(2)
REAL LOAD

YES = 1
NO = 0
SCALE = 32768.0

calibration constants for gas flow
C1GAS = 2.22
C2GAS = -0.0194

CALL PERFRM(VOLDSL, VOLGAS, VOLAIR, SPEED, LOAD, QINPPW, BMEP, 1
POWER, THRMEF, VOLEFF, PERDSL, RAF, RAD, RAG, 0)

WRITE(5,248)
READ(5,150) VLOAD
LOAD = 5.0 * VLOAD
WRITE(5,249)
READ(5,150) SPEED
WRITE(5,247)
READ(5,150) DPQAIR
VOLAIR = 2.173 + 0.221 * DPQAIR
WRITE(5,245)
READ(5,150) DPQGAS
VOLGAS = DPQGAS / 248.8
VOLGAS = C1GAS * VOLGAS + C2GAS * VOLGAS * VOLGAS
WRITE(5,246)
READ(5,150) VOLDSL
VOLDSL = VOLDSL * 60.0
CALL PERFRM(VOLDSL, VOLGAS, VOLAIR, SPEED, LOAD, QINPPW, BMEP, 1
POWER, THRMEF, VOLEFF, PERDSL, RAF, RAD, RAG, 1)
WRITE(5,250)
WRITE(5,251) SPEED, LOAD, VOLAIR, VOLDSL
WRITE(5,254)
WRITE(5,255) QINPPW, POWER, BMEP, THRMEF, VOLEFF
WRITE(5,256)
WRITE(5,257) VOLGAS, PERDSL
WRITE(5,258)
WRITE(5,259) RAF, RAD, RAG

save data in a file?
WRITE(5,224)
READ(5,120) ANS
IF (ANS .EQ. NO) GO TO 70
IF (L .EQ. 1) CALL ASSIGN(2, 'OUT.DAT')
WRITE(2,250)
WRITE(2,251) SPEED, LOAD, VOLAIR, VOLDSL
Listing of APP.PROG1 at 22:58:59 on APR 6, 1984 for CCid=AFPH

59 WRITE(2,252)
60 WRITE(2,253) DPTURB, DPCOMP, T1TURB, T2TURB, T1COMP, T2COMP
61 WRITE(2,254)
62 WRITE(2,255) QINPPW, POWER, BMEP, THRMEF, VOLEFF
63 WRITE(2,256)
64 WRITE(2,257) VOLGAS, PERDSL
65 WRITE(2,258)
66 WRITE(2,259) RAF, RAD, RAG
67 C
68 70 CONTINUE
69 WRITE(5,226)
70 READ(5,120) ANS
71 IF (ANS .EQ. NO) GO TO 999
72 500 CONTINUE
73 C
74 C
75 999 CONTINUE
76 STOP
77 C
78 100 FORMAT(1X,F16.5)
79 101 FORMAT(1X,12)
80 102 FORMAT(1X,12)
81 103 FORMAT(05)
82 110 FORMAT(A5)
83 120 FORMAT(II)
84 150 FORMAT(F12.4)
85 C
86 C
87 224 FORMAT(1X,'???? Enter 1 or 0 to save data!',$)
88 225 FORMAT(1X,I5,2X,E14.6)
89 226 FORMAT(//,'???? To rerun enter 1 or 0!',$)
90 245 FORMAT(1X,' >> > Enter Gas Flow in Pascal :',$)
91 246 FORMAT(1X,' >> > Enter Diesel Flow in litre/min:',$)
92 247 FORMAT(1X,' >> > Enter Air Flow in Pascal:',$)
93 248 FORMAT(1X,' >> > Enter Engine Speed in rpm:',$)
94 249 FORMAT(1X,' >> > Enter Load in Voltage:',$)
95 250 FORMAT(1X,' Speed (rpm) Load (lb) Air Flow (ft3/min) ',
96 1 ' Diesel Flow (ltr/hr)')
97 251 FORMAT(3X,F7.2,7X,F7.3,7X,F10.3,5X,F10.4)
98 253 FORMAT(1X,6(F8.2,2X))
99 254 FORMAT(5X,'Heat cons Power out BMEP Therm eff Vol eff ',
100 1 ' (BTU/hr-hp) (hp) (%) (%)
102 256 FORMAT(10X,'Gas Flow diesel input proportion (heat)',/,
103 1 ' (ft3/min) (percent total heat) '
104 257 FORMAT(10X,F12.3,5X,F8.2,' %')
105 258 FORMAT(10X,' LAMDA (tot) LAMDA (dsl) LAMDA (gas) ')
106 259 FORMAT(10X,3F16.2)
107 END
108 C
109 C
110 C
111 SUBROUTINE PERFRM(VOLDSL,VOLGAS,VAOLAIR,SPEED,LOAD,QINPPW, 
112 1 BMEP,POWER,THRMEF,VOLEFF,PERDSL,RAF,RAD,RAG,INDEX)
113 C
114 C computes performance characteristics
115 C
116 REAL LOAD,LHVDSL,LHVGAS
Listing of APP.PROG1 at 22:58:59 on APR  6, 1984 for CCid=AFPH

    C
  118  IF (INDEX .NE. 0) GO TO 10
  119  C
  120  C initialize const values
  121  C
  122  VISCAG  =  1.669
  123  DENDSL  =  0.8697 * 62.27
  124  DENGAS  =  0.044386
  125  DENAIR  =  0.07541
  126  STCDSL  =  15.0
  127  HHVDSL  =  1058288.4
  128  STCGAS  =  16.7
  129  LHVDSL  =  1002560.0
  130  HHVGAS  =  1024.7
  131  LHVGAS  =  926.0
  132  DSPLMT  =  425.04
  133  ARMLEN  =  17.5 / 12.0
  134  C
  135  C conv. factors
  136  C
  137  F3PLTR  =  0.03531
  138  HPPFPM  =  1.0 / 33000.
  139  BTUPHP  =  2544.433 / 1.01387 / 0.986315
  140  PI   =  3.1415
  141  C
  142  RETURN
  143  C
  144  10 CONTINUE
  145  C
  146  C process data
  147  C
  148  TQDYNO  =  LOAD * ARMLEN
  149  POWER  =  TQDYNO * SPEED * 2.0 * PI * HPPFPM
  150  VSWEPT  =  DSPLMT * SPEED / 2.0
  151  BMEP   =  (POWER / HPPFPM * 12.0) / VSWEPT
  152  DSLFLW  =  VOLDSL * F3PLTR
  153  GASFLW  =  VOLGAS * VISCAG
  154  FMAIR   =  DENAIR * VOLAIR
  155  FMDSL   =  DENDSL * DSLFLW / 60.0
  156  FMGAS   =  DENGAS * GASFLW
  157  RAD     =  FMAIR / (STCDSL * FMDSL)
  158  RAG     =  0.0
  159  IF (FMGAS .GT. 0.0) RAG = FMAIR / (STCGAS * FMGAS)
  160  RAF    =  FMAIR / (STCDSL * FMDSL + STCGAS * FMGAS)
  161  VOLEFF  =  VOLAIR / (VSWEPT / 12.0 / 12.0 / 12.0) * 100.0
  162  PERDSL  =  LHVDSL*DSLFLW / (LHVDSL*DSLFLW*LHVGAS*GASFLW*60.0)
  163  PERDSL  =  PERDSL * 100.0
  164  C
  165  C branch out no-load ie. idling
  166  C
  167  IF (ABS(LOAD) .LT. 0.00001) GO TO 11
  168  QINPPW  = (LHVDSL*DSLFLW + LHVGAS*GASFLW*60.0) / POWER
  169  THRMEF  =  BTUPHP / QINPPW * 100.0
  170  RETURN
  171  C
  172  11 CONTINUE
  173  C
  174  QINPPW  =  0.0
Listing of APP.PROG1 at 22:58:59 on APR 6, 1984 for CCid=AFPH

175  THEMEF = 0.0
176  C
177  RETURN
178  END
APPENDIX E - COMPUTER PROGRAM FOR APPARENT ENERGY RELEASE
Listing of DIG.HEAT.N at 20:48:34 on MAY 28, 1984 for CCid=AFPH Page 1

Rate of Heat Release Analysis Program
written by: Seaho Song
Dec / 1983

This program reads in the cylinder pressure data

to compute the rate of heat release for every C.A.

IMPLICIT REAL*8(A-H,O-Z)
REAL*8 CYLVOL(180)
REAL*8 GAS(10), GASNEW(10), P(180)
REAL*8 HEATRT(180), ANGLE(180), GASNEW(10), DH0(10), DH0(10)
REAL*8 F(2), DFDX(2,2), X(2), DX(2), FEP(2), XEP(2),
WORKAR(2,2)
REAL*8 NTOT, SAVGAS(180,10)
INTEGER I, PERM(4)
COMMON / GEOM / ARM, ROD, BORE, STROKE, VCLEAR
COMMON / EXPMT/ SPEED, BMEP
COMMON / PROP/ DENAIR, DENDSL, DENNG, WTDSL, WTNG,
1 WTAIR
COMMON /THDYPR/ HOF(10), R0, WT(10), NGAS

specify cylinder geometry in inches, then convert
into metric.

CONVF1 = 0.0254
ARM = 3.0 * CONVF1
ROD = 9.595 * CONVF1
BORE = 4.75 * CONVF1
STROKE = 6.0 * CONVF1
VCLEAR = 6.444 * CONVF1**3
To smoothen pressure data SET ISMOOT = 1
To use heat transfer model SET IHTRSF = 1
To consider dissociation SET IDSSOC = 1
To obtain output appropriate
for plotting SET IPLOT = 1
Setting ISMOOT = 0; IHTRSF = 0; IDSSOC = 0 will
assume unsmoothed, adiabatic with no dissociation.

ISMOOT = 1
IHTRSF = 1
IDSSOC = 1
I PLOT = 1

EPSIL = 0.1E-1

compute cylinder volume at every C.A.

CALL GEOMTR(CYLVL, ANGLE)

assign coefficients for thermodynamic properties of various gases.

CALL READPR

read in cylinder pressure data, injected diesel amount, CH4 amount, and injection & ignition characteristics.

CALL DATAIN(GAS,P,DSLAMT,INJBEG,INJEND,IGNBEG)

write out the mode of operation and bmep in kPa

BMEPKP = BMEP * 6.895D0

IF (GAS(2) .LT. 0.1D-12)
1 WRITE(6,210) SPEED, BMEPKP
IF (GAS(2) .GE. 0.1D-12)
2 WRITE(6,211) SPEED, BMEPKP

smooth the P data

IF (ISMOOT .NE. 1) GO TO 10

CALL SMOOTP(P, ANGLE, IGNBEG, IPOK)

IF (IPOK  .EQ. 0) WRITE(6,914)
IF (IPOK  .EQ. 0) STOP

CONTINUE

set up for initial stage

FRREM keeps track of fraction of fuel remaining.
FRBURN burnt.

The subscript 1 refers to the previous step

2 present ..

FRREM = 1.0
FRBURN = 0.0
QCACCM = 0.0
P1 = P(1)
V1 = CYLVL(1)
T1 = P1 * V1 / R0

.. total mass of gases present
Listing of DIG.HEAT.N at 20:48:34 on MAY 28, 1984 for CCid=APFH Page 3

117 C NTOT - total number of Kmoles of gases present
118 C GAS(1-10) - # of Kmoles of each gas present
119 C GASNEW(1-10) - used to update GAS(1-10)
120 C
121 C TOTMAS = 0.D0
122 NTOT = 0.D0
123 DO 29 J=1,NGAS
124 GASNEW(J) = GAS(J)
125 TOTMAS = TOTMAS + GAS(J) * WT(J)
126 NTOT = NTOT + GAS(J)
127 29 CONTINUE
128 C DHOTOT - Enthalpy at T1 minus the Enthalpy at 25 C for the total gas
129 C
130 C CALL DH0FN(T1,DH01)
131 DHOTOT = 0.0
132 DO 31 I=1,NGAS
133 DHOTOT = DHOTOT + GASNEW(I) * DH01(I)
134 31 CONTINUE
135 U2 = DHOTOT - NTOT * R0 * T1
136 C write headings for the output
137 C WRITE(6,200)
138 C calculation of rate of heat release is carried out for each C.A. degree.
139 C
140 DO 50 ITH=2,180
141 P2 = P(ITH)
142 V2 = CYLVOL(ITH)
143 C update the number of Kmoles of diesel injected.
144 C IF (ITH .EQ. INJBEG)
145 1 GAS(1) = GAS(1) + DSLAMT
146 IF (ITH .NE. INJBEG) GOTO 750
147 C TOTMAS = 0.D0
148 NTOT = 0.D0
149 DO 732 J=1,NGAS
150 TOTMAS = TOTMAS + GAS(J)*WT(J)
151 NTOT = NTOT + GAS(J)
152 732 CONTINUE
153 C IF (ITH .EQ. INJBEG) GOTO 750
154 C no combustion.
155 C processes compression stroke.
156 C 25 CONTINUE
157 FRAC = 0.0
158 RN = NTOT * R0
159 T2 = P2 * V2 / RN
compute rate of heat transfer, internal energy.

CALL UPROD(P1,P2,T1,T2,V1,V2,GAS,GASNEW,DH01,FRAC,
1    DH0,NTOT,TOTMAS,U2RES,QC,QHT,0,1HTRSF) GO TO 58

combustion is taking place, processes combustion and expansion stroke.

CONTINUE

assign initial guess values for fraction of fuel burnt and the gas mixture temperature.

IF (FRAC<0.1D-20) FRAC = 0.1D-6

use modified Newton's method, the following two are calculated iteratively:

- FRAC - fraction of fuel burnt
- T2 - gas mixture temperature

the following system of two nonlinear equations are solved

\[
\begin{bmatrix}
P - \frac{nRT}{V}
2

U - U + work - Qhtr
2
\end{bmatrix}
\]

for

\[
\bar{X} = \begin{bmatrix}
FRAC \\
T2
\end{bmatrix}
\]

DO 650 L650=1,50

given \bar{X}, compute \bar{F}

CALL GETF(X,F,GAS,T1,P1,P2,V1,V2,NTOT,
1    GASNEW,QHT,QC,TOTMAS,DH01,DH0,1HTRSF,IDSSOC)

if solution is found, terminate the iteration.

IF ((DABS(F(1)).LT.1.0).AND.(DABS(F(2)).LT.0.1E-4))

formulate the Jacobian matrix of \bar{F} as:

\[
\frac{dF}{dX} = \begin{bmatrix}
\frac{dF}{dx} & \frac{dF}{dx} \\
1 & 1 & 2
\end{bmatrix}
\]

\[
\frac{dF}{dx} = \begin{bmatrix}
2 & 1 & 2
\end{bmatrix}
\]
Listing of DIG.HEAT.N at 20:48:34 on MAY 28, 1984 for CCid=AFPH Page 5

233 C
234 DO 651 LJ=1,2
235 DO 652 LI=1,2
236 XEP(LI) = X(LI)
237 652 CONTINUE
238 XEP(LJ) = XEP(LJ) + EPSIL
239 CALL GETF(XEP,FEP,GAS,T1,P1,P2,V1,V2,NTOT,
240 1 GASNEW,QMT,QC,TOTMAS,DH01,DH01,HTRSF,IDSSOC)
241 DO 653 LI=1,2
242 DFDX(LI,LJ) = (FEP(LI)-F(LI)) / EPSIL
243 653 CONTINUE
244 651 CONTINUE
245 C
246 C the iteration scheme is as follows:
247 C
248 C X = X + DX
249 C
250 C DX is obtained by solving
251 C
252 C (dFdX) DX = -F
253 C
254 C
255 C F(1) = -F(1)
256 F(2) = -F(2)
257 C
258 C the routine SLE is a UBC Library subroutine
259 C which solves a system of linear equations.
260 C
261 C CALL SLE(2,2,DFDX,1,2,F,DX,IPERM,2,WORKAR,
262 1 DET,JEXP)
263 C
264 DO 654 LI=1,2
265 X(LI) = X(LI) + DX(LI)
266 654 CONTINUE
267 650 CONTINUE
268 66 CONTINUE
269 C
270 C iteration failed, terminate the execution.
271 C
272 WRITE(6,913) L650, X(1), X(2)
273 GO TO 850
274 C
275 C solution found
276 C
277 59 CONTINUE
278 FRAC = X(1)
279 T2 = X(2)
280 58 CONTINUE
281 C
282 C compute the accumulated heat release,
283 C accumulated fraction of fuel burnt.
284 C
285 QCACCM = QCACCM + QC
286 IANGLE = ITH - 90
287 FRBURN = FRBURN + FRAC * FRREM
288 FRREM = 1.0 - FRBURN
Listing of DIG.HEAT.N at 20:48:34 on MAY 28, 1984 for CCid=APFH Page 6

291 C write out the solution for the current C.A.
292 C WRITE(6,201) IANGLE, P2, T2, QHT, QC, QCACCM,
293 1 FRAC, FRBURN, V2
294 C shift index for next C.A. computation.
295 C T1 = T2
296 P1 = P2
297 V1 = V2
298 DO 71 I=1,NGAS
299 DOH0(I) = DHO(I)
300 71 CONTINUE
301 IF (ITH .LT. 80) GO TO 851
302 IF (ITH .GT. 170) GO TO 851
303 C write out the solution for plotting purpose.
304 C IF (IPLOT .NE. 1) GO TO 851
305 WRITE(1,205) ANGLE(ITH), P2
306 WRITE(2,205) ANGLE(ITH), T2
307 WRITE(3,205) ANGLE(ITH), QCACCM
308 WRITE(4,205) ANGLE(ITH), QC
309 851 CONTINUE
310 C save the instantaneous gas mixture composition to write out at the end of the rate of heat release output.
311 C IF (ITH .LT. (INJBEG - 1)) GO TO 50
312 DO 61 J=1,NGAS
313 GAS(J) = GASNEW(J)
314 SAVGAS(ITH,J) = GAS(J)
315 61 CONTINUE
316 C this is the end of the process for one C.A.. C.A. is incremented end the process proceeds to the next C.A..
317 C 50 CONTINUE
318 850 CONTINUE
319 C end of all the processes.
320 C write out the gas mixture composition for each C.A. from the C.A. just prior to the diesel injection.
321 C WRITE(6,202)
322 INJBM1 = INJBEG - 1
323 DO 72 ITH=INJBM1,180
324 IANGLE = ITH - 90
325 WRITE(6,203) IANGLE, (SAVGAS(ITH,J), J=1,NGAS)
326 72 CONTINUE
327 STOP
328 200 FORMAT(' CA P kPa T deg K, Q htr. (kJ), Q r' 
329 'elease, Q accum Frac Frac cume Vol')
330 201 FORMAT(1X,15,10E14.5)
331 202 FORMAT('Gas Comp(Kmol) Dsl CH4 N2 O2 ')
SUBROUTINE READPR

This routine assigns values, for various gases, density, molecular weight, enthalpy of formation, number of different kind of gases considered, and the ideal gas constant.

IMPLICIT REAL*8(A-H,O-Z)
COMMON /PROP1/ DENAIR,DENDSL,DENNG,WTDSL,WTNG,WTAIR
COMMON /THDYPR/ H0F(10),RO,WT(10),NGAS

density
DENAIR = 0.337600E-01
DENDSL = 0.848900E+00
DENNG = 0.186760E+01

molecular weight of intake gases
WTDSL = 0.170000E+03
WTNG = 0.160000E+02
WTAIR = 0.137280E+03

number of different kind of gases considered, and the ideal gas constant.
NGAS = 10
RO = 0.831425E+01

enthalpy of formation at standard condition.
H0F(1) = -.290871E+06
H0F(2) = -.748730E+05
H0F(3) = 0.000000E+00
H0F(4) = 0.000000E+00
H0F(5) = -.393522E+06
H0F(6) = -.241827E+06
H0F(7) = 0.000000E+00
H0F(8) = 0.394630E+05
C
HOF(9) = -.110525E+06
HOF(10) = 0.905920E+05
C
molecular weight
C
WT(1) = 0.170000E+03
WT(2) = 0.160400E+02
WT(3) = 0.280130E+02
WT(4) = 0.319990E+02
WT(5) = 0.440100E+02
WT(6) = 0.180150E+02
WT(7) = 0.201600E+01
WT(8) = 0.170070E+02
WT(9) = 0.280100E+02
WT(10) = 0.460000E+02
C
RETURN
END
C
SUBROUTINE GEOMTR(CYLVOL, ANGLE)
C
This routine assigns values for the engine geometry and computes cylinder volume at each C.A.
C
COMMON / GEOM / ARM, ROD, BORE, STROKE, VCLEAR
REAL*8 CYLVOLO80), ANGLE(180)
Computes V, ANGLE for Theta=-90,90 (Crank Angle)
V in cu. meter
XAREA = 3.14 * BORE * BORE / 4.0
ARMSQ = ARM * ARM
RODSQ = ROD * ROD
RADPDG = 3.14 / 180.0
/~ compute cylinder volume
ITH1 = -89
ITH2 = 90
DO 100 ITH=ITH1,ITH2
TH = DFLOAT(ITH)
RD = TH * RADPDG
X = ROD*ARM*(1.0-DCOS(RD))-DSQRT(RODSQ-ARMSQ*DSIN(RD)**2)
CYLVOLOITH+90) = XAREA * X + VCLEAR
ANGLEITH+90) = TH
100 CONTINUE
SUBROUTINE DATAIN(GAS, P, DSLAMT, INJBEG, INJEND, IGNBEG)

This routine reads in the data for injection and ignition characteristics, cylinder pressure, engine speed, BMEP, flow rates of air, gas, diesel. The flow rates are converted to number of Kmoles per cycle.

IMPLICIT REAL*8(A-H,O-Z)

COMMON / EXPMT / SPEED, BMEP
COMMON / PROP 1 / DENAIR, DENDSL, DENNG, WTDSL, WTNG, WTAIR
REAL*8 GAS(10), P(180)

READ(10,100) SPEED, BMEP
READ(10,101) QAIR, QDSL, QNG
READ(10,102) INJBEG, INJEND, IGNBEG

JA = 1
DO 10 L=1,36
   JB = JA + 4
   READ(10,103) (P(J), J=JA,JB)
   JA = L * 5 + 1
10 CONTINUE

convert pressure data from psi to kPa

DO 20 J=1,180
   P(J) = P(J) * 6.895
20 CONTINUE

compute # of moles per cycle:
C GAS(2) - nat gas
  (3) - N2
  (4) - O2

DO 30 L=1,20
  GAS(L) = 0.0
30 CONTINUE

C making sure of # of cylinder = 4

GAS(2) = QNG * DENNG / SPEED * 2.0 / WTNG / 4.0
AIRMOL = QAIR * DENAIR / SPEED * 2.0 / WTAIR
GAS(3) = 3.76 * AIRMOL / 4.0
GAS(4) = AIRMOL / 4.0

C compute amount of diesel injected in kmoles

DSLAMT = QDSL / 60.0 * DENDSL / SPEED * 2.0 / WTDSL / 4.0

RETURN

100 FORMAT(1X, F6.1, 1X, F5.1)
101 FORMAT(1X, F5.1, 1X, F5.2, 1X, F5.2)
102 FORMAT(1X, 5(F6.1, 1X))
END

SUBROUTINE UPROD(P1,P2,T1,T2,V1,V2,GAS,GASNEW,DH01,FRAC,
  1 DH0,NTOT,TOTMAS,U2RES,QC,QHT,ICOMB,1HTRSF)

IMPLICIT REAL*8(A-H,O-Z)

REAL*8 GAS(10),GASNEW(10),R1(10),DH01(10),DH0(10),NTOT
COMMON /THDYPR/ H0F(10), R0, WT(10), NGAS

WORK = 0.5D0 * (P1 + P2) * (V2 - V1)
CALL DH0FN(T2,DH0)

IF (ICOMB .EQ. 0) GOTO 5

compute the heat release due to combustion during
the current C.A. interval.

QC = 0.0D0
DO 20 I=1,NGAS
  QC = QC + (GASNEW(I) - GAS(I))
  QC = QC + (H0F(I) + DH0(I) - R0 * T2)
CONTINUE
QC = -QC
IF (ICOMB .EQ. 0) QC = 0.0

DO 30 I=1,NGAS
DU = DU + GAS(I) * (DHO(I) - DHO1(I) - R0DT)
CONTINUE

IF (I COMB .EQ. 0) QC = 0.0
C
C
DU = 0.DO
R0DT = R0 * (T2 - T1)
DO 30 I=1,NGAS
DU = DU + GAS(I) * (DHO(I) - DHO1(I) - R0DT)
CONTINUE

comput heat trasfer.
if IHTRSF is set to 0, adiabatic processe is assumed.
QHT = 0.DO
IF (IHTRSF .NE. 1) GO TO 10
QHT = QHTRSF(T2,V2,GASNEW,TOTMAS)
GO TO 10
CONTINUE
U2RES = DU + WORK - QC - QHT
RETURN
END

SUBROUTINE STCHPD(GAS,FRAC,GASNEW)
This routine computes the number of Kmoles of stoichiometric combustion product, and yield the updated composition of the gas mixture in the cylinder.

IMPLICIT REAL*8(A-H,O-Z)
COMMON /THDYPR/ HOF(10), R0, WT(10), NGAS
REAL*8 GAS(10), GASNEW(10), N, M
M = GAS(1) * FRAC
N = GAS(2) * FRAC
GASNEW(1) = GAS(1) - M
GASNEW(2) = GAS(2) - N
GASNEW(3) = GAS(3)
GASNEW(4) = GAS(4) - 18.5*M - 2.0*N
GASNEW(5) = GAS(5) + 12.0*M + N
GASNEW(6) = GAS(6) + 13.0*M + 2.0*N
DO 10 I=7,NGAS
   GASNEW(I) = GAS(I)
10  CONTINUE
C
RETURN
END

SUBROUTINE SMOOTP(P, ANGLE, IGNBEG, IPOK)

This routine uses Cubic-Spline-Least-Squares-Fit to smooth the cylinder pressure data. The rate of pressure rise is numerically computed from the UNSMOOTHED data, and this is used in weight to control the degree of local smoothness. The weighting is based on scattering the of slope of the pressure data.

IMPLICIT REAL*8(A-H,O-Z)

REAL*8 P(180), DPDTH(180), TOL(180), ANGLE(180)
REAL*8 PD1(180), PD2(180)
REAL*8 W(2000)

compute dp/dTheta by fitting a quadratic through 3 points

DO 5 J=2,179
   DPDTH(J) = (P(J+1) - P(J-1)) / 2.DO
5  CONTINUE

DPDTH(1) = (4.DO*P(2) -3.DO*P(1) -P(3)) /2.DO
DPDTH(180) = (3.DO*P(180)-4.DO*P(179)+P(178))/2.DO

TOLO controls the local smoothness.

SVAL the global ..

TOLO = 1.0
SVAL = 1000.0
IPOK = 1
DO 10 I=4,177
   JS = I - 3
   JF = I + 2
   AV = 0.0
10  DO 11 J=JS,JF
   AV = AV + DPDTH(J)
11  CONTINUE

AV = AV / 10.0

SD is a measure of scattering in the slope of the pressure data. This is computed by considering 4 adjacent points.

SD = 0.0
DO 12 J=JS,JF
   SD = SD + DABS(DPDTH(J) - AV)**2
12  CONTINUE
C The degree of smoothness is less forced for
    the points before the start of ignition.
C IF (DABS(DFLOAT(I-IGNBEG)).LT.5.0) TOL(I)=TOL(I)/10.0
CONTINUE
DO 20 1=1,3
   TOL(I) = TOL(4)
20  CONTINUE
DO 30 1=178,180
   TOL(I) = TOL(177)
30  CONTINUE
C The routines DSPLFT and DSPLN are UBC Library
    subroutines, which performs Least-Squares-Fit
    with Cubic-Spline as basis functions.
CALL DSPLFT(ANGLE,P,TOL,SVAL,180,5,613)
CALL DSPLN(ANGLE,P,PD1,PD2,180,5,613)
RETURN
613 CONTINUE
C The fit has failed
IPOK = 0
RETURN
END
C Computes mean viscosity of gas mixtures.
IMPLICIT REAL*8(A-H,O-Z)
REAL*8 GAS(10)
COMMON /THDYPR/ HOF(10),R0,WT(10),NGAS
TM = T**0.645
VISC = GAS(1) * WT(1) * 1.33
VISC = VISC + GAS(2) * WT(2) * 3.35
VISC = VISC + GAS(3) * WT(3) * 4.57
VISC = VISC + GAS(4) * WT(4) * 5.09
VISC = VISC + GAS(5) * WT(5) * 3.71
VISC = VISC + GAS(6) * WT(6) * 3.26
TOTW = GAS(1)*WT(1)+GAS(2)*WT(2)+GAS(3)*WT(3)
1 + GAS(4)*WT(4)+GAS(5)*WT(5)+GAS(6)*WT(6)
DOUBLE PRECISION FUNCTION QHTRSF(T2,V2,GASNEW,TOTMAS)

This routine computes the heat transfer at current Annand's model is used.

IMPLICIT REAL*8(A-H,O-Z)

COMMON / GEOM / ARM,ROD,BORE,STROKE,VCLEAR
COMMON /EXPMT / SPEED, BMEP
REAL*8 GASNEW(10)

This routine employs Annand's model to compute the rate of heat transfer.

A = 0.47D0
C = 1.6D-12
30 CONTINUE
PISEVEL = SPEED * STROKE / 30.0
CALL VISCS(T2,GASNEW,VISC)
DENTOT = TOTMAS / V2
RENUM = DENTOT * PISVEL * BORE / VISC
REKD = CP * VISC / 0.7 / BORE * RENUM**(0.7)
TW = 0.484 * BMEP + 540.0
The surface area of the cylinder
SURFA = (V2 - VCLEAR) * 4.DO / BORE + 0.0304D0
QCONVC = A * SURFA * REKD* (T2 - TW)
QRAD = (1.6E-12)*(10.76)*C*SURFA*(T2**4-TW**4)
QHTRSF = -(QCONVC + QRAD) * (60./SPEED/360.)
RETURN
END
SUBROUTINE DSSOCN(P,T,GAS1,GAS2,NTOT)

This routine computes the equilibrium dissociation products. The theoretical details including the numerical methods are described in the external documentation.

The reactions considered are:

1. \( CO \leftrightarrow CO + \frac{1}{2} O \)
2. \( HO \leftrightarrow \frac{1}{2} H + OH \)
3. \( HO \leftrightarrow H + \frac{1}{2} O \)
4. \( \frac{1}{2} N + \frac{1}{2} O \leftrightarrow NO \)

IMPLICIT REAL*8(A-H,O-Z)

COMMON /THDYPR/ HOF(10),R0,WT(10),NGAS
REAL*8 GAS1(10), GAS2(10), NTOT
REAL*8 K(4), F(5), FEP(5), KPOP(4), DX(5), XEP(5)
REAL*8 WORKDB(5,5), DXDBL(5), FDBL(5),DFDXDB(5,5),DETDBL
INTEGER I,PERM(10)

SUMN = 0.0
DO 5 I=1,NGAS
GAS2(I) = GAS1(I)
SUMN = SUMN + GAS1(I)
5 CONTINUE

IF (GAS1(5) .GE. 0.1E-20) GO TO 500
NTOT = SUMN
RETURN

compute the equilibrium constants for the given temperature.

K(1) = DEXP(DLOG(T)**(-7.4721)*(-0.65549E+8)+10.53)
K(2) = DEXP(DLOG(T)**(-7.0457)*(-0.30372E+8)+10.159)
K(3) = DEXP(DLOG(T)**(-6.8674)*(-0.18879E+8)+8.7095)
K(4) = DEXP(DLOG(T)**(-7.3355)*(-0.16593E+8)+1.80127)

POP = 101.325D0/P
KPOP(1) = K(1) * K(1)*POP
KPOP(2) = K(2) * K(2)*POP
KPOP(3) = K(3) * K(3)*POP
Listing of DIG.HEAT.N at 20:48:34 on MAY 28, 1984 for CCid=AFPH Page 16

871 KPOP(4) = K(4) * K(4)
872 C if the reactions are insignificant, then skip.
873 C
874 C RMK = 0.0
875 DO 10 I=1,4
876 IF (DABS(KPOP(I)) .GT. RMK) RMK = DABS(KPOP(I))
877 CONTINUE
878 IF (RMK .GT. 0.1E-5) GO TO 20
879 NTOT = SUMN
880 RETURN
881 C
882 C
883 C 20 CONTINUE
884 C
885 C initial guess
886 C
887 C X(1) = K(1)*DSQRT(P0P/SUMN/GAS1(4))*GAS1(5)
888 X(1) = X(1) - GAS1(9) / SUMN
889 X(3) = K(3)*DSQRT(P0P/SUMN/GAS1(4))*GAS1(6)
890 X(3) = X(3) - GAS1(7) / SUMN
891 X(2) = K(2)*DSQRT(P0P/SUMN)*GAS1(6)
892 X(2) = X(2)/DSQRT(DABS(GAS1(7)+X(3)*SUMN))
893 X(2) = X(2) - GAS1(8)/SUMN
894 X(4) = K(4)*DSQRT(GAS1(3)*GAS1(4))/SUMN
895 X(4) = X(4) - GAS1(10)/SUMN
896 X(5) = SUMN
897 40 CONTINUE
898 C
899 C Solve for \( \bar{X} \) using modified Newton's method.
900 C The method is precisely the same as that in
901 C the main routine, and the notations are also
902 C nearly the same.
903 C
904 C TOL = DABS(KPOP(4)) * 0.1D-4
905 DO 50 L=1,100 
906 CALL EVALF(X,GAS2,KP0P,SUMN,F)
907 C
908 C It is sufficient to check only the F(4),
909 C since it is the most slowly converging term.
910 C
911 C IF (DABS(F(4)) .LT. TOL) GOTO 300
912 C
913 C DO 51 LJ=1,5
914 EPSIL = DSQRT(0.1D-12+0.1D0*DABS(X(LJ))))
915 DO 52 LI=1,5
916 XEP(LI) = X(LI)
917 CONTINUE
918 XEP(LJ) = XEP(LJ) + EPSIL
919 CALL EVALF(XEP,GAS2,KP0P,SUMN,FEP)
920 DO 53 LI=1,5
921 DFDXDB(LI,LJ) = (FEP(LI)-F(LI)) / EPSIL
922 CONTINUE
923 53 CONTINUE
924 51 CONTINUE
925 C
926 C DO 55 LI=1,5
927 FDBL(LI) = -F(LI)
928 55 CONTINUE
Again, SLE is a UBC Library subroutine, which
solves a system of linear equations.
In solving the systems of equations, the routine
retains the decomposed matrix.

CALL SLE(5,5,DFDXDB,1,5,FDBL,DXDBL,IPERM,5,WORKDB,
DETDBL,JEXB)

DO 54 LI=1,5

X(LI) = X(LI) + DXDBL(LI)

CONTINUE

NTOT = X(5)
A = X(1) * NTOT
B = X(2) * NTOT
C = X(3) * NTOT
D = X(4) * NTOT

GAS2(3) = DABS(GAS1(3) - 0.5*D)
GAS2(4) = DABS(GAS1(4) + 0.5*(A+C-D))
GAS2(5) = DABS(GAS1(5) - A)
GAS2(6) = DABS(GAS1(6) - B - C)
GAS2(7) = DABS(GAS1(7) - A)
GAS2(8) = DABS(GAS1(8) + B)
GAS2(9) = DABS(GAS1(9) + A)
GAS2(10) = DABS(GAS1(10) + D)

Here, once a Jacobian matrix is formed for F,
it is used for 3-4 iterations, thus reducing
the cost.

DO 70 J70=1,3

CALL EVALF(X,GAS2,KP0P,SUMN,F)

DO 71 LI=1,5
FDBL(LI) = -F(LI)

CONTINUE

The routine DBS is also a UBC Library routine.
The routine uses the decomposed matrix by the
routine SLE to very economically compute new
solution with newly given FDBL

CALL DBS(5,1,5,FDBL,DXDBL,IPERM,5,WORKDB)

DO 73 LI=1,5
X(LI) = X(LI) + DXDBL(LI)

CONTINUE

update the composition of the gas mixture.

NTOT = X(5)
A = X(1) * NTOT
B = X(2) * NTOT
C = X(3) * NTOT
D = X(4) * NTOT

GAS2(3) = DABS(GAS1(3) - 0.5*D)
GAS2(4) = DABS(GAS1(4) + 0.5*(A+C-D))
GAS2(5) = DABS(GAS1(5) - A)
GAS2(6) = DABS(GAS1(6) - B - C)
GAS2(7) = DABS(GAS1(7) + 0.5*B + C)
GAS2(8) = DABS(GAS1(8) + B)
GAS2(9) = DABS(GAS1(9) + A)
GAS2(10) = DABS(GAS1(10) + D)
70 CONTINUE
50 CONTINUE
C
Iteration has failed. The execution will terminate
with a proper error message.
WRITE(6,213)
STOP
mission completed. Exit.
CONTINUE
RETURN
213 FORMAT('he succes Fm %XX%XX Fail to Converge in Dssociation xx% XXX')
END
C
C
C
SUBROUTINE EVALF(X,N,KP0P,SUMN,F)
This routine checks the appropriateness of the
given possible solution for the equilibrium dissociation. The deviation is
designated by the vector F.
IMPLICIT REAL*8(A-H,O-Z)
REAL*8 X(5), N(10), KP0P(4), F(5), NTOT
NTOT = X(5)
TERM1 = N(4)/NTOT+0.5*(X(1)+X(3)-X(4))
TERM2 = (N(7)/NTOT + 0.5*X(2) + X(3))
TERM3 = (N(3)/NTOT-0.5*X(4))*(N(4)/NTOT +0.5*(X(1)+X(3)-X(4)))
F(1) = (N(9)/NTOT+X(1))**2*TERM1/(N(5)/NTOT-X(1))**2
1 - KP0P(1)
F(2) = TERM2*(N(8)/NTOT+X(2))**2/(N(6)/NTOT-X(2)-X(3))**2
1 - KP0P(2)
F(3) = TERM2**2 * TERM1 / (N(6)/NTOT-X(2)-X(3))**2 - KP0P(3)
F(4) = (N(10)/NTOT+X(4))**2/TERM3
1 - KP0P(4)
F(5)= (SUMN-NTOT)-0.5D0*(X(1)+X(2)+X(3))*NTOT
1
RETURN
END
SUBROUTINE GETF(X,F,GAS,T1,P1,P2,V1,V2,NTOT,
     1     GASNEW,QHT,QC,TOTMAS,DH01,DH0,ITHTRSF,IDSSOC)

C This routine checks the appropriateness of the given s
C possible solutions for the requirements for the first s
C law and the ideal gas law. This routine is used in s
C the main routine for computing the fraction of fuel s
C burnt and T2.

C

IMPLICIT REAL*8(A-H,0-Z)
REAL*8 X(2),F(2),GAS(10),GASNEW(10),DH0(10),DH01(10)
REAL*8 NTOT,GASNEW(20)
COMMON /THDYPR/ H0F(10),R0,WT(10),NGAS

FRAC = X(1)
T2 = X(2)

IF (IDSSOC .EQ. 0) GO TO 50
CALL STCHPD(GAS,FRAC,GASNEW)
CALL DSSOCN(P2,T2,GASNEW,GASNEW,NTOT)
GO TO 51
CONTINUE

CALL STCHPD(GAS,FRAC,GASNEW)
CONTINUE

CALL UPROD(P1,P2,T1,T2,V1,V2,GAS,GASNEW,DH01,FRAC,
     1     DH0,NTOT,TOTMAS,U2RES,QC,QHT,1,ITHTRSF)

F(1) = P2 - NTOT * R0 * T2 / V2
F(2) = U2RES

RETURN

END

DOUBLE PRECISION FUNCTION CP0VAL(T,GAS)

C This routine computes the mean value of the specific f
C heat Cp.

IMPLICIT REAL*8(A-H,0-Z)
COMMON /THDYPR/ H0F(10),R0,WT(10),NGAS
REAL*8 GAS(10), CP0(10), NTOT
Listing of DIG.HEAT.N at 20:48:34 on MAY 28, 1984 for CCid=AFPH  Page 20

1103 C
1104 TH = T / 100.0
1105 TH2 = TH * TH
1106 TQ = TH**(0.25)
1107 TQ2 = TQ * TQ
1108 TQ3 = TQ * TQ * TQ
1109 TQ6 = TQ3*TQ3
1110 C
1111 CP0(1) = 104.18 + 465.5 * (T / 1000.0)
1112 CP0(2) = -672.87 + 439.74*TQ - 24.875*TQ3 + 323.88/TQ2
1113 CP0(3) = 37.432 + 0.020102*TQ6-178.57/TQ6+236.88/TH2
1114 CP0(5) = -3.7357+30.529*TQ2-4.1034*TH+0.024198*TH2
1115 CP0(6) = 143.05-183.54*TQ+82.751*TQ2-3.6989*TH
1116 CP0(7) = 56.505-702.74/TQ3+1165.0/TH-560.7/TQ6
1117 CP0(8) = 81.546-59.35*TQ+17.766*TQ3-4.266*TH
1118 CP0(9) = 69.145-0.70463*TQ3-200.77/TQ2+176.76/TQ3
1119 CP0(10) = 46.045+216.1/TQ2-363.66/TQ3+232.55/TH2
1120 C
1121 CPOVAL = 0.0
1122 DO 20 I=1,NGAS
1123 DO 20 I=2,NGAS
1124 CPOVAL = CPOVAL + CP0(I) * GAS(I)
1125 20 CONTINUE
1126 C
1127 C
1128 RETURN
1129 END

SUBROUTINE DH0FN(T,DH0)

This routine computes the change in enthalpy between the given temperature and 298 deg K. The unit for DH0(1-10) is kJ/Kmol-K

C

IMPLICIT REAL*8(A-H,O-Z)

REAL*8 DH0(10)

T2 = T*T
T3 = T*T*T
TQ2 = DSQRT(T)
TQ = DSQRT(TQ2)
TQ3 = TQ * TQ2
TQ5 = T * TQ
TQ6 = T * TQ2
TQ7 = T * TQ3

DH0(1) = 104.18*T+0.23276*T2-51714.3
Listing of DIG.HEAT.N at 20:48:34 on MAY 28, 1984 for CCid=AFPH Page 21

1161 C
1162 C
1163 1
1164 C
1165 1
1166 C
1167 C
1168 C
1169 1
1170 C
1171 1
1172 C
1173 C
1174 1
1175 C
1176 C
1177 1
1178 C
1179 C
1180 1
1181 C
1182 C
1183 1
1184 C
1185 C
1186 1
1187 C
1188 C
1189 C
1190 C
1191 C

END