Combustion modeling of a two cylinder cycle reciprocating engine.

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COMBUSTION MODELING OF A TWO CYLINDER CYCLE RECIPROCATING ENGINE

BY

Victor Chrjapin

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June 1984
COMBUSTION MODELING OF A TWO CYLINDER CYCLE RECIPROCATING ENGINE

by

VICTOR CHRJAPIN

B.S. Mech. Eng., Purdue University
(1975)

SUBMITTED TO THE DEPARTMENT OF OCEAN ENGINEERING
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COMBUSTION MODELING OF A TWO CYLINDER CYCLE RECIPIROCATING ENGINE

by

VICTOR CHRJAPIN

Submitted to the Department of Ocean Engineering on May 11, 1984, in partial fulfillment of the requirements for the Degrees of Master of Science in Naval Architecture and Marine Engineering and Master of Science in Mechanical Engineering.

ABSTRACT

A simple mathematical model was developed to simulate the closed portion of the cycle for a quiescent chamber compression ignition engine utilizing the assumption of perfect gases and the first law of thermodynamics. Various input parameters were used in trend analysis to check the model. The output from the computer program was compared to test data from a four inch bore, open chamber semi-quiescent diesel engine run at the Sloan Test Laboratory. This computer model was then modified to simulate the expansion stroke of a newly developed, two cylinder cycle reciprocating engine. The model was then run to determine the optimum point of fuel injection for the new engine.

Thesis Supervisor: Professor A. Douglas Carmichael
Title: Professor of Power Engineering
I wish to express my sincere gratitude to Professor A. Douglas Carmichael for his understanding, assistance and guidance which made this project a most valuable experience. The discussions with Professors Warren M. Rohsenow and John B. Heywood were also most helpful in keeping me and this project on track.

I wish to thank Mr. Peter Birkholz for the use of his data processing facilities and technical advise during the preparation of the final draft of this thesis.

For their everlasting love and encouragement to do better than they, this project is dedicated to my parents, Wasil and Stella.

Last, but not least by any stretch of the imagination, I am most deeply indebted to my wife, Karen, and my daughters, Kelly and Valery, for their love, patience, understanding and encouragement during the course of this project and for the period of study at M.I.T.

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Chapter 1
INTRODUCTION

The compression ignition (CI) engine is thriving in newfound popularity amongst automobiles, medium-duty and heavy-duty freight transport trucks, marine propulsion and auxiliary systems, and various other industrial applications. The United States Department of Energy (DOE) has recently predicted that diesel fuel consumption will exceed gasoline consumption in this country by the year 2000. This is primarily due to the shift to diesel power in the automotive and truck freight transport industries to take advantage of the high efficiency, high power-to-weight ratio engines. The present daily diesel fuel consumption of the order of $10^8$ liters\(^1\) is expected to increase by as much as 50 percent by the turn of the century. The increasing fuel consumption rate, coupled with the constant concern of diminishing oil reserves, has prompted renewed interest in improving the operating efficiency of the conventional compression ignition engine. Any small improvement in engine efficiency will obviously result in an enormous savings in petroleum.
The approaches currently pursued to improve compression ignition engine efficiency include increasing the compression ratio and the development of the "adiabatic" engine. The former involves turbocharging and the improvement to piston ring technology. The latter approach concentrates the most emphasis on insulating the engine. This requires the use of temperature-resistant ceramic cylinder liners for combustion cylinders whose gas wall temperatures can be of the order of 1200 degrees Kelvin. In addition to these two approaches, there are many other avenues of research in progress that involve improvements that will increase the compression ratio, decrease the heat loss from the engine, or increase the combustion efficiency through improved combustion chamber design.

Instead of improving upon the conventional compression ignition engine, a new cycle engine design is under development. This new design, proposed by Carmichael\(^2\), consists of a two cylinder cycle which divides the functions of a conventional four-stroke diesel cycle into two parts. The new engine has one cylinder which compresses the incoming air charge and another cylinder which acts as the combustion chamber and expansion cylinder. These two cylinders are interconnected by a regenerative heat exchanger. The regenerator acts as the heart of the new design. Through the use of new ceramic materials, the regenerator will act as a heat transfer medium by
transfering a portion of the heat from the exhaust gases to the incoming air charge. The temperature of the incoming air will be elevated twice. The first temperature increase is due to the compression process in the first cylinder. This cylinder, in turn, will transfer its air charge through the ceramic matrix of the regenerative heat exchanger, thus boosting the temperature for the second time. After passing through the regenerator, the incoming air charge will be of sufficient temperature to accommodate spontaneous combustion. With this high temperature, the high compression ratio of the conventional compression ignition engine is not required to obtain work from the cycle. Figures 1 and 2 depict the pressure vs. volume and temperature vs. volume diagrams for the new cycle as compared to a conventional diesel cycle. The preliminary design of this new two cylinder cycle engine indicates that an improvement to thermal efficiency can be achieved over the conventional compression ignition engine.

An important element of the engine design process is the capability to predict, with an acceptable degree of accuracy, the energy release during combustion as a function of time. This process is extremely complex in that it involves the injection and atomization of fuel, the evaporation and mixing of the fuel with the air charge, followed by the various phases of combustion. The ability to accurately predict the heat release rate is vital to the engine designer when analyzing a new engine design.
This thesis is an attempt to assimilate various diesel engine combustion models to produce a simple, yet accurate, model to be used in the continuing evaluation of the new two cylinder cycle reciprocating engine. The proposed model can be utilized on a personal computer system to determine the optimum point of fuel injection for the new engine. The model has the capability to evaluate two different fuel types (i.e. iso-octane and propane).
Chapter 2
DESCRIPTION OF THE TWO CYLINDER CYCLE RECIPROCATING ENGINE

The Two Cylinder Cycle Reciprocating Engine consists of one compression cylinder and a pair of combustion/expansion cylinders, see figures 3 and 4. The compression cylinder consists of an intake valve and two exhaust valves, one to each expansion cylinder. Each expansion cylinder has its own fuel injector. The regenerator cavity contains an exhaust valve in addition to the ceramic matrix regenerator. The pistons of both the compression cylinder and the expansion cylinders are considered to be of simple geometry with flat heads. The expansion cylinder piston incorporates no unique features to increase turbulence or swirl, thus it is similar to a direct injection, quiescent chamber diesel engine cylinder. The five valves and three pistons are actuated by a camshaft that allows the compression piston to operate at twice the speed of an expansion piston. The compression cylinder will alternately provide a compressed air charge to each expansion cylinder via the regenerator. A typical cycle can be illustrated by referring to figures 3 and 4.
-13-

The compression cylinder is at top dead center.

Step 1:

Valves 2A and 3A are already closed.

Valves 2B is closed, valve 2B is the exhaust stroke, cylinder is not just completed an impulse air charge transfer to

and has just completed an impulse air charge transfer to

Step 2:

The compression cylinder has just reached the

Step 3:

The compression cylinder is approx. 90°.

Step 4:

The compression cylinder is approx. 90°.

Step 5:

The compression cylinder has just completed an impulse air charge.

2A, 3A and 2B are closed, valves 2B is still open.

2A, 3A and 2B are closed, valves 2B is still open.

The regenerator. Valves 1 is open, valves 2A, 3A, 2B are closed.

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The regenerator. Valves 1 is open, valves 2A, 3A, 2B are closed.

The regenerator. Valves 1 is open, valves 2A, 3A, 2B are closed.

The regenerator. Valves 1 is open, valves 2A, 3A, 2B are closed.

The regenerator. Valves 1 is open, valves 2A, 3A, 2B are closed.
has just closed (it only opened for a very short time just before the compression cylinder reached TDC), valve 3A just opened, valves 2A and 3B are closed. (This is the same as step 1 except that cylinders A and B are reversed.)

**Step 6:** The compression cylinder is approximately 90° ATDC and is in the middle of an air charge induction. Cylinder A is still exhausting and cylinder B is in the expansion process. Valve 1 is open, valves 2B, 3B, 2A are closed and valve 3A is open. (This is the same as step 2 except that cylinders A and B are reversed.)

**Step 7:** The compression cylinder has just reached BDC and has completed induction of an air charge. Cylinder A is exhausting through the regenerator and cylinder B is still expanding. Valve 1 just closed, valves 2B, 3B, 2A are closed and valve 3A is open. (This is the same as step 3 except that cylinders A and B are reversed.)

**Step 8:** The compression stroke is approximately 90° BTDC and in the middle of compressing an air charge. Cylinder A is completing its exhaust stroke and cylinder B is ending its expansion stroke. Valve 1 is closed, valve 2B, 3B and 2A are closed, valve 3A is still open. (This is the same as step 4 except that cylinders A and B are reversed.)
Step 9: This is the same as step 1.

Figures 5 and 6 show the temperature and pressure as a function of cylinder volume for a cycle.

The table below summarizes the sequencing of the valves for a complete cycle of an expansion cylinder.

<table>
<thead>
<tr>
<th>Valve</th>
<th>1</th>
<th>2A</th>
<th>3A</th>
<th>2B</th>
<th>3B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Step 1:</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>0</td>
</tr>
<tr>
<td>Step 2:</td>
<td>0</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>0</td>
</tr>
<tr>
<td>Step 3:</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>0</td>
</tr>
<tr>
<td>Step 4:</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>0</td>
</tr>
<tr>
<td>Step 5:</td>
<td>X</td>
<td>X</td>
<td>0</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Step 6:</td>
<td>0</td>
<td>X</td>
<td>0</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Step 7:</td>
<td>X</td>
<td>X</td>
<td>0</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Step 8:</td>
<td>X</td>
<td>X</td>
<td>0</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Step 9:</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>0</td>
</tr>
</tbody>
</table>

where X = Valve closed

and O = Valve open

As can be readily seen, the valve timing sequence is rather complex. The timing sequence must be such as to allow the impulse transfer of the air charge to occur
without possible blow-down to the atmosphere or charging the wrong cylinder. A shift of the crank angle must be considered to optimize the air charge transfer sequence to the on-line expansion cylinder. Thus, the valve timing sequence is a critical factor in the correct and efficient operation of this new engine design and must be dealt with appropriately.
Chapter 3
COMBUSTION AND COMBUSTION MODELING
(An overview)

3.1 Description of Diesel Engine Combustion

The diesel engine combustion process is exceedingly complex and not very well understood. Combustion in the diesel engine is characterized by compression ignition, a non-uniform fuel and air distribution in the combustion chamber, and a continuous mixing throughout the period in which combustion occurs. Due to the initial conditions in the chamber when fuel is first injected, the air charge in the cylinder is of sufficient temperature and pressure to support a chain-reaction. However, combustion in the compression ignition engine is governed by the local conditions in each part of the charge and not dependent on the spread of the flame from one point to another. Therefore, the rate of combustion is dependent on the state and distribution of the fuel and upon the pressure and temperature within the cylinder. \(^3\)

3.1.1 The Phases of Combustion
Ricardo described the diesel engine combustion process as taking place in three stages; namely the delay period, a period of rapid combustion, followed by burning at a controlled rate.\textsuperscript{3} Lyn\textsuperscript{4} described the burning process in three slightly different phases. The first phase is a period of rapid combustion which lasts for only three degrees crank angle. The second stage is characterized by a decreased rate of heat release lasting approximately 40 degrees crank angle. The third period consists of the fuel burning at a very slow rate which may persist through the remainder of the expansion stroke.

A combination of the descriptions of Ricardo and Lyn may be more appropriate. The stages of combustion could be divided into ignition delay, premixed burning, diffusion controlled combustion and the tail of combustion.\textsuperscript{5,6} Figure 7 depicts the four stages of combustion in a heat release diagram.

3.1.1.1 Ignition Delay

The term ignition delay, or ignition lag, describes the time required by the preliminary reactions that occur prior to the appearance of flame. The ignition delay is broken down into a physical delay and a chemical delay. The physical delay period occurs between the beginning of fuel injection and the onset of chemical reactions. During this period, the fuel is atomized, vaporized, mixed with air and
raised in temperature. This process is sometimes collectively referred to as preparation. The chemical delay period immediately follows the physical delay period and terminates at inflammation or ignition. This period is characterized by chemical reactions starting slowly with pre-flame oxidation of the fuel followed by local ignition.

The ignition delay will vary according to cylinder temperature, cylinder pressure, the type of fuel, the initial temperature of the fuel, the characteristics of the fuel injectors and the turbulence in the cylinder. The physical delay is small for light fuels but can become the controlling factor for heavy, viscous fuels. The physical delay can be significantly reduced by using high injection pressures and high turbulence to expedite the breakup of the fuel jet.

Semi-empirical relationships have been developed to describe the ignition delay. An estimate for ignition delay was developed by Wolfer in 1938:[7]

\[
t = 0.44P^{-1.19}\exp(4650/T)
\]

where: \( t \) = ignition delay in milliseconds
\( P \) = cylinder pressure in atmospheres
and \( T \) = temperature in degrees K at ignition.
An estimate by Clarke\cite{8} in 1970 is quite similar to that by Wolfer:

\[ t = 0.22\exp(5500/T)P^{-0.727} \]

where: \( t \) = ignition delay in seconds

\( T \) = cylinder temperature in degrees K

and \( P \) = cylinder pressure in N/m\(^2\).

Still another empirical expression for ignition delay was developed by Spadaccini and Tevelde\cite{9} from experiments for NASA in 1979 with diesel fuel in a steady flow facility:

\[ t = 2.43\times10^{-9}P^{-2}\exp(41560/RT) \]

where \( t \) = ignition delay in seconds

\( P \) = pressure in atmospheres

\( T \) = mixture temperature in degrees K

and \( R \) = gas constant in atm cm\(^3\)/gmole°K.

Figure 8 represents the effects of temperature and pressure on ignition delay as determined from the estimates by Wolfer. The Spadaccini and Tevelde and Clarke relationships yield somewhat similar results.

When using ignition delay expressions, it must be emphasized that differences in engines, fuel properties (especially cetane number), fuel injectors and actual engine temperatures and pressures make the calculation rather approximate. These formulas are also very limited by their use of bulk temperatures, with no consideration of local compositions or temperatures.\cite{10}
3.1.1.2 Premixed Burning

In the premixed burning stage, flame occurs at one or more locations and spreads turbulently. The rate and amount of combustion during this stage is directly related to the fuel preparation rate and the length of the ignition delay period. Since this stage of combustion is one of premixed combustion, little carbon (soot) is produced resulting in little radiation heat transfer. However, since the combustion rate is so intense, combustion generated noise is controlled by this stage of combustion.\(^{11}\) Figure 9 depicts premixed burning in a cylinder.

3.1.1.3 Diffusion Controlled Burning

Once the prepared, or premixed, fuel has burned, the combustion process slows down. The combustion rate in this stage will be dominated by the rate of local air entrainment. Since the temperature in the cylinder is favorable for ignition in this stage, the air/fuel mixing process will control the rate of combustion. This preparation of the fuel will be governed by the turbulence and swirl in the cylinder. Lyn\(^{4}\) estimated that approximately 40 percent of the heat release from the combustion of fuel comes from this stage. Figure 10 shows the diffusion burning process in a cylinder.

3.1.1.4 Combustion Tail Stage

This last stage of combustion is characterized by the cylinder pressure and temperature falling as the expansion process continues. The rate of combustion tails off due to the chemical kinetic effects as
the chemical reaction rate slows. In this stage, the reaction rate will become the controlling factor instead of the air/fuel mixing process. This stage is also characterized by diffusion combustion with a high production and combustion of soot particles with a resultant high rate of radiation heat transfer. This last stage of combustion can proceed through the completion of the expansion stroke and can contribute up to 20 percent of the total heat release.\(^4\) Figure 11 represents a typical heat release rate diagram showing the four stages of combustion.

3.2 Combustion Modeling

The combustion process is often considered the most important aspect of an internal combustion engine, but, at the same time, the least understood and most complex. A mathematical model depicting combustion would require good models of the fuel system to include the injection/fuel pump, the injector nozzles, and fuel lines. Additionally, models of fuel atomization, vaporization, fuel/air mixing, cylinder air motion, chemical kinetics and pre-mixed and diffusion mixing would be required. A model as comprehensive as this has yet to be developed. Spaulding\(^{12}\) states that this type of "combustion modeling is impossible." He justifies this by pointing out that the number of governing restraints and rules outnumber the degrees of freedom and, in addition, the requirements of low cost, speed and accuracy must also be met. Since the complexity of the real combustion process is so overwhelming, substantial simplifying assumptions must be made to obtain solutions.
3.2.1 Types of Models and Uses

Bracco\(^{[13]}\) categorized combustion models into three categories based on their uses in examining different engine problems. The categories are the zero-dimensional (or thermodynamic) model, the quasi-dimensional (or entrainment) model, and the multi-dimensional (or detailed) model.

3.2.1.1 Zero-dimensional Model\(^{[11]}\)

The zero-dimensional model is structured around a thermodynamic analysis of the engine cylinder contents during the cycle. The assumptions include one-dimensional flow, isentropic adiabatic flow through nozzles simulating flow past valves, and unburned mixtures as mixtures of air, fuel vapor and residual gases. Specific heats of the gas mixture are modeled using polynomial functions of temperature. Compression is assumed to be adiabatic. Combustion assumes thermochemical equilibrium and progressive burning via mass elements. The expansion process assumes thermochemical equilibrium.

Heat transfer is modeled using correlations between the Nusselt, Prandtl- and Reynolds numbers from heat transfer in steady turbulent flow over flat plates and pipes. These relationships are in the form of:

\[ \text{Nu} = a \text{Re}^b \text{Pr}^c \]

where \(a\), \(b\), and \(c\) are obtained from experimental data for a specific engine.
The combustion process is generally modeled from an apparent heat release or an experimentally obtained fuel burning rate. One of the most widely used correlations is based on the Wiebe Function. In this function, the fuel burned is expressed as a fraction of the total fuel injected.\(^5\)

\[
FB = 1 - \exp\left(-K_2(t) ^{K_1+1}\right)
\]

where \(FB\) = fraction of fuel burned/total injected

\(t\) = time from ignition

\(K_1\) = shape factor for combustion curve

\(K_2\) = combustion efficiency coefficient.

Another typical function form is the cosine function.\(^{11}\)

\[
X(\Theta) = \frac{(1/2)}{1 - \cos \pi [(\Theta - \Theta_0)/\Delta \Theta_b]}
\]

where \(X(\Theta)\) = mass fraction burned at crank angle \(\Theta\)

\(\Theta_0\) = crank angle at the start of combustion

and \(\Delta \Theta_b\) = burn duration.

There are numerous other combustion models that utilize various heat release patterns. Some replace the heat release curve with two straight lines. In this type of combustion model, one line simulates the rapid combustion of the bulk of the injected fuel and the other line represents the slower combustion phase further down the expansion stroke.
An empirical model developed by Whitehouse and Way\textsuperscript{14} is based on elementary combustion principles. Fuel is assumed to be prepared for combustion as a result of fuel-air mixing. The reaction rate calculates the burn rate in the premixed stage of combustion. The preparation rate becomes governing during the diffusion burning phase as the fuel is assumed to burn as rapidly as it is prepared. (The Whitehouse and Way model will be dealt with in detail in a later chapter.)

In general, thermodynamic combustion models are useful when performing a design trade off or comparison analysis to evaluate the effects of change in engine design and operation. Since, however, the details of the combustion process are an input to the model, the results can only indicate what will transpire if the engine burns in the specified manner. These models cannot address the feasibility of the engine operating in the prescribed manner because the details of the burning process are not linked to the engine design and operation.\textsuperscript{15}

3.2.1.2 Quasi-dimensional Model\textsuperscript{11}

Quasi-dimensional models are also structured around a thermodynamic analysis of the engine cylinder during the cycle. Many of the same assumptions are utilized to describe the various portions of the process as are used in the thermodynamic model. The combustion process, on the other hand, is based on more fundamental physical quantities such as turbulent intensity, turbulent mixing, jet characteristics in jet mixing and the kinetics of the fuel-oxidation process.

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The quasi-dimensional models can be utilized for the same purposes as the zero-dimensional models except that they can now be used where changes in the combustion process can be a dominant factor. The major drawback of the quasi-dimensional model is its inability to examine, in detail, the interaction between fluid flow and engine geometry.\(^{(14)}\)

3.2.1.3 **Multi-dimensional Model**\(^{(11)}\)

In a multi-dimensional model, the governing partial-differential equations describing conservation of mass, momentum, energy and species, and the sub-models describing turbulence, chemical kinetics, and etc. are numerically solved subject to boundary conditions and other restraints. These models have the potential for examining the interaction between fluid flow and engine geometry that is lacking in the quasi-dimensional model. The detailed model will predict engine performance and emission characteristics from the first principles with virtually no empirical relationships. Unfortunately, solving the relevant conservation equations in three-dimensional, time dependent formulation, coupled with the state equations and sub-models leads to a computer program that will tax even the most capable computer system.
Chapter 4

THE TWO CYLINDER CYCLE COMBUSTION MODEL

Since the two cylinder cycle reciprocating engine is a totally new concept, combustion modeling can be even more difficult than for a compression ignition engine. However, the approach taken models the expansion cylinder of the new cycle after a diesel engine cylinder. The beginning of the expansion stroke will simulate a diesel engine with its piston at TDC with a charge of air. For this initial combustion model, the air will be assumed to be contained within the cylinder, at pressure, with no additional air added after expansion, as in the actual new engine cycle.

4.1 Assumptions

The assumptions for this single zone combustion model are essentially those previously mentioned for the thermodynamic type of models.

a. The First Law of Thermodynamics is used to establish an energy balance to determine the temperature at the end of each step.

b. The working fluid is treated as an ideal gas.

c. The system contents are homogeneous and of uniform temperature and pressure.
d. The changes in gas properties due to the rate of change of the gas composition are considered to be negligible.

e. Combustion is treated as a reversible heat release process.

f. Combustion products are formed in the proportions according to the law of perfect combustion.

g. No dissociation of the products of combustion occurs.

h. Only four gases are considered to be present and are varied as required for perfect combustion.

i. The incoming air charge is assumed to be pure air plus a fraction of the residual gases remaining in the cylinder.

4.2 Thermodynamics of Internal Combustion Engines

4.2.1 Ideal Gas

The assumed thermally ideal gas obeys the state equation

\[ pV = MRT \]

where \( p \) = pressure
\( V \) = volume
\( M \) = number of moles
\( \bar{R} \) = universal gas constant

and \( T \) = temperature.

The specific gas constant, \( R \), can be written in terms of \( \bar{R} \) and \( m_w \), the molecular weight of the gas.

\[ R = \frac{\bar{R}}{m_w}. \]
If the mass of the gas, \( m = M_m \), then the state equation can be written as:

\[ pV = mRT. \]

The specific internal energy for an ideal gas can be represented as a function of temperature:

\[ u = f(T) \]

where \( u \) = specific internal energy
and \( f(T) \) = function of temperature dependent on the gas.

If the function \( f(T) \) is expressed in the form of a limited power series, then \( \{17\} \)

\[ u = u_0 + \sum_{n=1}^{n=S} a_n T^n \]

where \( a_1 \) to \( a_5 \) are constants which vary depending on the gas
and \( u_0 \) = internal energy at absolute zero.

The specific heat at constant volume can be defined as:

\[ C_v = (dq/dT)_v = (du/dT)_v. \]

Thus, following the same procedures as for the internal energy, above:\( \{17,18\} \)

\[ C_v = \sum_{n=1}^{n=S} n a_n T^{n-1} \]

The specific enthalpy, \( h \), for an ideal gas is given by:

\[ h = u + RT. \]

It follows that:\( \{17,18\} \)

\[ h = h(T) = u_0 + \sum_{n=1}^{n=S} a_n T^n + RT. \]
At absolute zero, $T=0$:

$$h = h_0 = u_0.$$  

Therefore, for a perfect gas, the internal energy varies linearly with temperature as:

$$h = h_0 + C_V T + R T.$$  

The specific heat at constant pressure, $C_P$, is defined by:

$$C_P = (dq/dT)_P = (dh/dT)_P.$$  

For a perfect gas:

$$C_P = C_V + \bar{R}.$$  

Now, enthalpy can be expressed by:

$$h = h_0 + C_P T.$$  

For thermodynamic processes with gases of constant composition and specific heats undergoing state changes;

$$h_0 = u_0 = 0.$$  

Then,

$$u = C_V T;$$

$$h = C_P T;$$

$$h - u = (C_P - C_V) T = R T;$$

and $C_P - C_V = \bar{R}.$

Gas data are often given in terms of enthalpy vice internal energy.

The conventional form is:

$$h(T)/RT = (h - h_0)/RT$$

$$= a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4.$$  

and the internal energy is expressed as:

$$u(T)/RT = (a_1-1) + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4.$$  

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The values for the polynomial coefficients, \( a_0 \) to \( a_5 \) are provided in Table 2. Other formulations for the calculation of enthalpy and specific heat are available in the literature.\(^{(5,26,27)}\)

**Table 2: Polynomial Coefficients**

**Range: 500 - 3000 Degrees Kelvin**

<table>
<thead>
<tr>
<th></th>
<th>( a_1 )</th>
<th>( a_2 )</th>
<th>( a_3 )</th>
<th>( a_4 )</th>
<th>( a_5 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \text{C}_2 \text{H}_18 )</td>
<td>3.0959</td>
<td>2.73114E-03</td>
<td>-7.88542E-07</td>
<td>8.66002E-11</td>
<td>0.0</td>
</tr>
<tr>
<td>( \text{H}_2\text{O} )</td>
<td>3.74292</td>
<td>5.65590E-04</td>
<td>4.95240E-08</td>
<td>-1.81802E-11</td>
<td>0.0</td>
</tr>
<tr>
<td>( \text{O}_2 )</td>
<td>3.25304</td>
<td>6.52350E-04</td>
<td>-1.49524E-07</td>
<td>1.53897E-11</td>
<td>0.0</td>
</tr>
<tr>
<td>( \text{N}_2 )</td>
<td>3.34435</td>
<td>2.94260E-04</td>
<td>1.95300E-09</td>
<td>-6.57470E-12</td>
<td>0.0</td>
</tr>
<tr>
<td>( \text{C}_3\text{H}_8 )</td>
<td>-0.71993</td>
<td>4.6426E-02</td>
<td>-1.68385E-05</td>
<td>-2.67009E-09</td>
<td>0.0</td>
</tr>
</tbody>
</table>

4.2.2 **Properties of Gas Mixtures**\(^{(18)}\)

Mixtures of gases obey the following.

a. The gas mixture as a whole obeys the equation of state,

\[ pV = MRT, \] where \( M \) is the total number of moles of all species.

b. The total pressure of the mixture is equal to the sum of the pressures which the individual components/species exert.

c. The internal energy, enthalpy and entropy of the mixture equals the sum of the internal energies, enthalpies and entropies which each individual component/species would have if it separately occupied the
entire volume of the mixture at the same temperature.

Thus, for mixtures of ideal gases the mole fraction is given by:

\[ x_i = \frac{M_i}{M} \]

where \( M_i \) = moles of a specie

and \( M \) = total number of moles.

Then,

\[ \sum x_i = 1.0. \]

Enthalpy is given by:

\[ H = \sum M_i h_i = M \sum x_i h_i. \]

Internal energy is given by:

\[ U = \sum M_i u_i = M \sum x_i u_i. \]

Specific Heats are given by:

\[ C_p = \Sigma x_i C_{pi} \]

\[ C_v = \Sigma x_i C_{vi}. \]

4.2.3 The First Law of Thermodynamics \(^{[17]}\)

The emphasis of this model is the closed portion of the cycle. Therefore, the First Law of Thermodynamic for closed systems is simply:

\[ dQ - dW = dU' \]

where \( dQ \) = heat energy transfer

\( dW \) = work energy transfer

\( dU' \) = change in internal energy.

The internal energy is defined by:

\[ U' = U + KE + PE \]

where \( U \) = the intrinsic internal energy
KE = kinetic energy
PE = potential energy.

For a closed system, we can assume that PE = KE = 0. Therefore,
\[ \mathrm{d}Q - \mathrm{d}W = \mathrm{d}U \]

where \( U = M \sum x_i u_i \)
- \( M \) = total number of moles
- \( x_i \) = mole fraction of gas \( i \)
- \( u_i \) = specific internal energy of gas \( i \).

For non-reacting closed systems, we can write:
\[ \mathrm{d}Q - \mathrm{d}W = \mathrm{d}U \]

where \( \mathrm{d}W = p \mathrm{d}V = (\sum x_i p) \mathrm{d}V \)

and \( \mathrm{d}U = M \mathrm{d}(\sum x_i u_i) \).

For a reacting closed system, we can expand this to:
\[ \mathrm{d}Q - p \mathrm{d}V = (U_{op} - U_{or}) + U_p(T) - U_r(T) \]

where \( (U_{op} - U_{or}) = \Delta U_o \)
- \( \Delta U_o \) = heat of reaction
- \( U_p(T) \) = energy of products as a function of time
- \( U_r(T) \) = energy of reactants as a function of temperature.

4.3 Heat Transfer from the gas to the Cylinder

To be able to balance the energy in a real system, the heat transfer from the combustion gas to the walls of the cylinder must be considered. Two basic equations are generally accepted for use in cycle
calculations. These are the correlations developed by Annand and Woschni. The relationship by Woschni{19} is based upon a forced convection model.

\[ \frac{q}{A} = C_3 d^{-0.2} p^{0.8} T_g^{-0.053} (C_1 V_p + C_2 (p-p_0) V_T / p')^{0.8} (T_g - T_w) \]

where \( C_1, C_2, \) and \( C_3 = \) constants

\( A = \) area
\( D = \) cylinder bore
\( p = \) pressure
\( T_g = \) mean gas temperature
\( T_w = \) wall temperature
\( V_p = \) piston velocity
\( p_0 = \) motoring pressure
\( p' = \) trapped pressure
\( V' = \) trapped volume
\( T' = \) trapped temperature.

Although Woschni's expression is readily accepted, it does not separately distinguish between convection and radiation.

The Annand equation is also largely based on turbulent convection. Unlike the Woschni correlation, Annand claims that the Reynolds number is the major parameter affecting convection. Convection is the first term in his equation. The second term in Annand's equation is a radiation term assuming grey body radiation. Thus:{20}

\[ \frac{q}{A} = a (k/D) (Re)^b (T_g - T_w) + c (T_g^4 - T_w^4) \]

where \( q = \) heat transfer rate
\( A = \) area
\( a, b, c = \text{constants} \)

\( k = \) thermal conductivity

\( D = \) bore

\( \text{Re} = \) Reynolds Number = \( \rho V_p D / \mu \)

\( \rho = \) density

\( V_p = \) piston velocity

\( \mu = \) viscosity

\( T_g = \) temperature of gas (mean)

\( T_w = \) temperature of wall.

The range of values for Annand's constants are:

for a four stroke engine:

\( a = 0.26 \)

\( b = 0.75 \pm 0.15 \)

\( c = 3.88 \pm 1.39 \times 10^{-8} \) J/sm\(^2\)K\(^4\)

for a two stroke engine:

\( a = 0.26 \)

\( b = 0.64 \pm 0.10 \)

\( c = 3.03 \pm 1.06 \times 10^{-8} \)

Since Annand's equation separates the convective term from the radiation term, it is believed that the Annand correlation is better suited to the new cycle calculations.

4.4 The Combustion Model
In the process of heat release from combustion, both physical and chemical effects are involved. Liquid fuel injected into an engine must be heated, vaporized, and mixed with oxygen in the preparation process prior to combustion. Once the fuel is prepared, it may then burn at a rate controlled by chemical kinetics. It has been demonstrated that the time required for combustion of the prepared fuel is negligible as compared to the preparation time.

At the beginning of the burning period, chemical kinetics are important due to the low temperatures. When fuel is first injected into a cylinder of a diesel engine, the temperature is generally such that rapid burning will not occur. Additionally, the heat transferred to the incoming fuel causes the temperature to drop in the cylinder. As the temperature rises in the cylinder, the combustion rate rises, thus increasing the temperature. The heat release rate continues to rise until the lack of prepared fuel becomes the controlling factor. When the excess prepared fuel is depleted, combustion will proceed at the rate of fuel preparation. Figure 12 represents the effects of preparation rate and reaction rate in premixed burning as a function of crank angle.

4.4.1 Preparation of Fuel

After injection, the fuel is physically prepared for combustion. As mentioned before, this process involves the atomization, vaporization and mixing of the fuel with air. The rate of preparation can be assumed to be proportional to the total surface area of the fuel spray droplets. If all the droplets are assumed to be of identical size, then it
follows: \{7, 14, 21\}

\[ M_i = np \pi D_o^2/6 \]
\[ M_u = np \pi D^2 \]

where \( M_i \) = Mass of fuel injected
\( M_u \) = Mass of fuel unburned
\( n \) = number of fuel droplets
\( p \) = fuel droplet density
\( D_o \) = Initial droplet diameter
\( D \) = Droplet diameter.

The total area

\[ \text{Area} = n \pi D^2 = n \pi (6M_u/np\pi)^{2/3} \]
\[ \text{Area} = (6M_i/npD_o)^{3/3} (6M_u/p\pi)^{2/3} \]
\[ \text{Area} = 6M_i^{1/3}M_u^{2/3}/pD_o. \]

Assuming that the density, \( p \), and initial diameter, \( D_o \), are constant, then the

\[ \text{Area} \propto M_i^{1/3}M_u^{2/3}. \]

Allowing for the effect of oxygen availability on the mixing of the fuel, the preparation rate, \( PR \), can be written as:

\[ PR = KM_i^{1-x}M_u^xp_{O2}^m \]

where \( x \) = empirical constant
\( m \) = empirical constant
\( p_{O2} \) = partial pressure of oxygen
\( K \) = constant.

The constant \( K \) is a function of the characteristics of fuel injection, air movement and combustion chamber shape. Typical values for four
stroke engine are: \( K = 0.008 - 0.020 \)
\( x = 2/3 \)
\( m = 0.4 \).

4.4.2 Reaction of Fuel

Since diesel fuel is not a pure substance, it is impossible to ascertain the exact chemical equations involved since the actual compounds in the fuel are unknown. The temperatures that are available from experiments are only average cylinder temperatures. With these approximations/estimations, the equations for reaction rate are highly empirical. The degree of approximation involved may be justified due to the short time period during which chemical kinetics is of importance. Also, the total fuel that is burned is equal to the amount of fuel that is prepared. The reaction rate equation that was proposed by Whitehouse and Way \( \{7, 14, 21\} \) is based on the Arrhenius equation.

\[
R = \frac{(K'P_2^2)}{(N/\bar{T})} \int (PR-R)dx \exp(-act/T)
\]

where \( R = \) reaction rate per degree crank angle
\( K' = \) empirical constant
\( act = \) empirical constant
\( P_2 = \) partial pressure of oxygen
\( PR = \) preparation rate
\( N = \) engine speed in rpm
\( T = \) cylinder temperature.

The effect of the ignition delay period is incorporated in the Arrhenius type expression \( \exp(-act/T) \). Typical values of \( K' \) and \( act \) are:
act = 1.4 x 10^4

K' = 1.2 x 10^{10} for two stroke engines

K' = 65 x 10^{10} for four stroke engines

4.5 Verification of the Model

The model was converted to computer code using TRS-80 Model III Disk Basic. The program listing is presented in Appendix B.

In an effort to set the empirical coefficients, the average value was used for all coefficients that had a range of values for four stroke engines. The program was run and compared to the data obtained by Remley\(^\text{22}\) in actual engine testing in the Sloan Automotive Laboratory. Figure 13 represents the pressure versus volume curve for the model and for the engine run by Remley. Appendix A provides specifications of the test engine.
Chapter 5

SELECTION OF FUEL INJECTION POINT

In order to obtain the maximum work and highest efficiency from the new two cylinder cycle, the time of fuel injection should be optimized. To obtain this optimum, a number of cycles were run on the computer.

5.1 Selection of the Model Coefficients

The model was run assuming the expansion cylinder at TDC with an air charge at a temperature and pressure of 1090°C and 10 atmospheres while the engine speed of 850 rpm and air/fuel ratio were held constant. The selected fuel was C8H18 (iso-octane) with a lower heating value of 4.2 x 10^7 joules/kilogram and a residual air fraction of 0.05.

The model was run several times to obtain a value of K in the equation:

\[ PR = K M_i (1-x) M_u x P_{O2} \]

The values of x and m were held constant at 2/3 and 0.04, respectively, as the values used for four-stroke diesel engines. When searching for a value of K, a diffusion combustion period of 70 - 120 degrees of crank angle was sought. This was found through several iterations to occur at a value of K = 0.012.
The values of constants for the reaction rate equation:

\[ R = \left[ K' P_{O2}/N \right] \exp\left(\frac{-a}{T}\right) / (P - R) \int dx, \]

were selected as the values for four-stroke diesel engines.

With this input data and selection of constants, the model yields a heat release rate curve which closely resembles that described by Ricardo, Lyn and Whitehouse et al, see figure 14. The premixed burning phase yields approximately 45 percent of the heat release, the diffusion controlled burning phase yields approximately 45 percent of the heat release with the tail of combustion providing the remaining 10 percent.

5.2 Optimizing Fuel Injection

Intuitively, the maximum work and highest efficiency would be expected with fuel injection and combustion occurring at TDC, or immediately thereafter. This, however, does not appear to be the case when the data is evaluated. See figures 15 through 21. While the fuel injection is varied from 180 (TDC) to 205 degrees crank angle, with an injection period of 20 degrees, the thermal efficiency rises. For fuel injection occurring from 180 to 195 degrees, the temperature at 360 degrees (BDC) is not sufficient to heat the regenerator matrix to a temperature which will pre-heat the incoming air charge to 1090 degrees Kelvin as specified by the input data. For fuel injection occurring at 205 degrees, and later, incomplete combustion will result.

From this approach, the optimum point of fuel injection occurs at 200 degrees crank angle for a fuel injection period of 20 degrees.
When the fuel injection period is reduced to 10 degrees, a similar pattern is observed. A fuel injection point on, or before, 200 degrees results in the cylinder gas temperature dropping too low to support sufficient air charge pre-heat. Fuel injection on, or after, 210 degrees results in incomplete combustion. See figures 22 through 24. In this case, the optimum point of fuel injection occurs at 205 degrees. The thermal efficiency for this case is higher that the case of a 20 degree injection period. Also, the specific fuel consumption is lower in the case of 10 degree injection as compared to 20 degree injection.

Through a similar analysis, the case of an air/fuel ratio of 25 yields an optimum fuel injection point of 195 degrees crank angle for a period of 20 degrees. For this air/fuel ratio, the value of K in the preparation rate equation was selected as 0.018 to achieve a similar heat release rate pattern.
Chapter 6
COMMENTS AND RECOMMENDATIONS

As can be readily seen, the output from this type of thermodynamic model is dependent on the value of the empirical coefficients. The characteristics of the heat release rate curve will shift as a function of air/fuel ratio, temperature, pressure and engine speed. Therefore, only a comparison of results from a defined heat release should be used for qualitative comparison analysis.

Since this computer model was written for a personal computer, the time required for one run is excessively long for a detailed comparison analysis. The run time for one run with five degree increments is approximately 1 hour and 20 minutes. A motoring analysis (no combustion) requires approximately 15 minutes. The amount of time required in the combustion iteration process is the difference between the two. These times were obtained when running the program with no remark statements and elimination of all unnecessary spaces in the program. Undoubtedly, the efficiency of the program can be somewhat increased by utilizing some clever programming techniques. However, the use of a small computer strictly dedicated to a comparison analysis with crank angle increments of one or two degrees can occupy the machine for an inordinate period of time.
6.1 Recommendations

This single zone model allows for cycle studies. However, a problem that must be explored is the formation of soot and gaseous pollutants. This can be accomplished by expanding the model to a two or four zone model.\footnote{23,24} During this model expansion, the effect of chemical kinetics should be further examined to display a more realistic combustion process. The values of the coefficients for the polynomial expression of enthalpy for the other products of combustion are readily available.\footnote{17,18}

The effects of heat transfer from the system may be more appropriately modeled by the use of the widely accepted Woschni correlations.\footnote{19} The use of Annand's correlation, however, does allow for the separation of convection and radiation.

The effects of mixing of the air charge with the fuel must be further explored to determine the effects on combustion intensity and efficiency.\footnote{25}

The use of a larger computer system would be most beneficial in a comparison analysis. Single runs can be easily done on a personal computer system, however, many runs using small crank angle increments are best, although more costly, performed on a main frame system capable of performing numerous simultaneous calculations.

Lastly, to obtain realistic coefficients for the empirical constants in the preparation and reaction rate equations, experiments using a rapid compression machine are considered appropriate. This
would provide for realistic data with minimum cost.
REFERENCES


2. Carmichael, A.D., Professor of Power Engineering, Massachusetts Institute of Technology, September 1983.


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Figures
Figure 1: Pressure vs. Volume for New Cycle vs. Diesel Engine
Figure 2: Temperature vs. Volume for New Cycle vs. Diesel
Figure 5: Potential Arrangement of Components for New Engine\(^{(2)}\)
Figure 4: Cutaway View of New Cycle Engine\(^{(2)}\)
Figure 5: Temperature vs. Volume for New Cycle
Figure 6: Pressure vs. Volume for New Cycle
Figure 7: The Four Phases of Combustion
Figure 8: Effects of Temperature and Pressure on Ignition Delay
Figure 9: Direct-Injection Compression Ignition Engine Combustion during the PREMIXED Phase
BURNED GAS: NO
WHITE/YELLOW FLAME: SOOT OXIDATION
RICH ZONES IN FUEL JET: SOOT FORMATION
FLAME QUENCH ON WALLS: HC

Figure 10: Direct-Injection Compression Ignition Engine Combustion during the MIXING CONTROLLED Phase
Figure 11: Typical Heat Release Diagram Showing Four Stages of Combustion
Figure 12: Heat Release Rates Calculated by Whitehouse Ray Model
Figure 15: Comparison of Model with results from Remley Test (11)
Figure 14: Heat Release Rate Obtained from Computer
DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE - MOTORING

INPUT DATA:

CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 850 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M+2
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG
STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .502636

RESIDUAL

CYLINDER PRESSURE CYLINDER TEMPERATURE CYLINDER PRESSURE CYLINDER TEMPERATURE CYLINDER PRESSURE CYLINDER TEMPERATURE

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Figure 15

-65-
### DIESSEL ENGINE COMBUSTION CYCLE

#### RUN: CARMICHAEL ENGINE

**INPUT DATA:**
- **CYLINDER BORE =** 0.3725 METERS
- **STROKE =** 0.3725 METERS
- **CONNECTING ROD LENGTH =** 0.745 METERS
- **ENGINE SPEED =** 350 RPM
- **ENGINE COMPRESSION RATIO =** 5
- **AIR / FUEL RATIO =** 30
- **TRAPPED PRESSURE =** 1.01325E+06 N/M²
- **TRAPPED TEMPERATURE =** 1090 DEG KELVIN
- **RESIDUAL AIR FRACTION =** 0.05
- **FUEL SELECTED FOR THIS ANALYSIS =** C8H18 (ISO OCTANE)
  - WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG
- **STOICHIOMETRIC AIR / FUEL RATIO =** 15.1151
- **FUEL / AIR EQUIVALENCE (PHI) =** 0.503836

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<th>CYLINDER TEMPERATURE (DEG K)</th>
<th>CYLINDER AIR (VOLLES)</th>
<th>FUEL IN (VOLLES)</th>
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**Figure 16**

-66-
**DIESEL ENGINE COMBUSTION CYCLE**

**RUN: CARMICHAEL ENGINE**

**INPUT DATA:**

- CYLINDER BORE = 0.3725 METERS
- STROKE = 0.3725 METERS
- CONNECTING ROD LENGTH = 0.3725 METERS
- ENGINE SPEED = 850 RPM
- ENGINE COMPRESSION RATIO = 5
- AIR / FUEL RATIO = 30
- TRAPPED PRESSURE = 1.01325E+06 N/MT²
- TRAPPED TEMPERATURE = 1090 DEG KELVIN
- RESIDUAL AIR FRACTION = 0.05
- FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
- WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG
- STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
- FUEL / AIR EQUIVALENCE (PHI) = 0.503836

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**FUEL INJECTION:**

- FUEL INJECTION START = 0 DEGREES
- FUEL INJECTION END = 0 DEGREES

**EXHAUST VALVE OPEN - CYCLE COMPLETE**

Figure 17

-67-
**DIESEL ENGINE COMBUSTION CYCLE**

**RUN: CARMICHAEL ENGINE**

**INPUT DATA:**

- CYLINDER BORE = .3725 METERS
- STROKE = .3725 METERS
- CONNECTING ROD LENGTH = .745 METERS
- ENGINE SPEED = 850 RPM
- ENGINE COMPRESSION RATIO = 5
- AIR / FUEL RATIO = 30
- TRAPPED PRESSURE = 1.01325E+06 N/M²
- TRAPPED TEMPERATURE = 1090 DEG KELVIN
- RESIDUAL AIR FRACTION = .05
- FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE) WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG

STOICHIOMETRIC AIR / FUEL RATIO = 15.1151

FUEL / AIR EQUIVALENCE (PHI) = .503836

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**Figure 18**

-68-
DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:
- CYLINDER BORE = .3725 METERS
- STROKE = .3725 METERS
- CONNECTING ROD LENGTH = .745 METERS
- ENGINE SPEED = 850 RPM
- ENGINE COMPRESSION RATIO = 5
- AIR / FUEL RATIO = 30
- TRAPPED PRESSURE = 1.01325E+06 N/M²
- TRAPPED TEMPERATURE = 1090 DEG KELVIN
- RESIDUAL AIR FRACTION = .05
- FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
  WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG
- FUEL / AIR EQUIVALENCE (PHI) = .503836

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Figure 19
# DIESEL ENGINE COMBUSTION CYCLE

**RUN: CARMICHAEL ENGINE**

**INPUT DATA:**

- **CYLINDER BORE** = 0.3725 METERS
- **STROKE** = 0.3725 METERS
- **CONNECTING ROD LENGTH** = 0.745 METERS
- **ENGINE SPEED** = 850 RPM
- **ENGINE COMPRESSION RATIO** = 5
- **AIR / FUEL RATIO** = 30
- **TRAPPED PRESSURE** = 1.01325E+06 N/M²
- **TRAPPED TEMPERATURE** = 1090 DEG KELVIN
- **RESIDUAL AIR FRACTION** = 0.05
- **FUEL SELECTED FOR THIS ANALYSIS** = C8H18 (ISO OCTANE)

**STOICHIOMETRIC AIR / FUEL RATIO** = \(-4.2E+07\) JOULES/KG

**FUEL / AIR EQUIVALENCE (\(\phi\))** = 0.503836

<table>
<thead>
<tr>
<th>CYLINDER LOCATION</th>
<th>CYLINDER VALVE OPEN</th>
<th>CYLINDER VALVE CLOSE</th>
<th>CYLINDER VALVE IN (DEG K)</th>
<th>CYLINDER VALVE OUT (DEG K)</th>
<th>FUEL IN (LITRE)</th>
<th>FUEL OUT (LITRE)</th>
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</thead>
<tbody>
<tr>
<td>CYLINDER LOCATION</td>
<td>CYLINDER VALVE OPEN</td>
<td>CYLINDER VALVE CLOSE</td>
<td>CYLINDER VALVE IN (DEG K)</td>
<td>CYLINDER VALVE OUT (DEG K)</td>
<td>FUEL IN (LITRE)</td>
<td>FUEL OUT (LITRE)</td>
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**EX-PET VALUE OPEN** = -3 CYCLE COMPLETE

**EX-PET VALUE CLOSE** = -3 CYCLE COMPLETE

*Figure 20*

-70-
DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:
CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 850 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M²
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG
STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .503836

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<th>CYLINDER VOLUME (L)</th>
<th>CYLINDER PRESSURE (BAR)</th>
<th>CYLINDER TEMPERATURE (K)</th>
<th>CYLINDER DATA (JOULES)</th>
<th>FUEL IN STROKE (FRACTION)</th>
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Figure 21

-71-
DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:
- CYLINDER BORE = 0.3725 METERS
- STROKE = 0.3725 METERS
- CONNECTING ROD LENGTH = 0.745 METERS
- ENGINE SPEED = 850 RPM
- ENGINE COMPRESSION RATIO = 5
- AIR / FUEL RATIO = 30
- TRAPPED PRESSURE = 1.01325E+06 N/M²
- TRAPPED TEMPERATURE = 1090 DEG KELVIN
- RESIDUAL AIR FRACTION = 0.05
- FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
- WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG
- STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
- FUEL / AIR EQUIVALENCE (PHI) = 0.503836

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<th>CYLINDER</th>
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Figure 22
DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:
CYLINDER BORE = .3725 METERS
STROKE = .3725 METERS
CONNECTING ROD LENGTH = .745 METERS
ENGINE SPEED = 850 RPM
ENGINE COMPRESSION RATIO = 5
AIR / FUEL RATIO = 30
TRAPPED PRESSURE = 1.01325E+06 N/M²
TRAPPED TEMPERATURE = 1090 DEG KELVIN
RESIDUAL AIR FRACTION = .05
FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE)
WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG
STOICHIOMETRIC AIR / FUEL RATIO = 15.1151
FUEL / AIR EQUIVALENCE (PHI) = .503836

Figure 23
DIESEL ENGINE COMBUSTION CYCLE

RUN: CARMICHAEL ENGINE

INPUT DATA:

| CYLINDER BORE = 0.3725 METERS |
| STROKE = 0.3725 METERS |
| CONNECTING ROD LENGTH = 0.745 METERS |
| ENGINE SPEED = 850 RPM |
| ENGINE COMPRESSION RATIO = 5 |
| AIR / FUEL RATIO = 30 |
| TRAPPED PRESSURE = 1.01325E+06 N/M² |
| TRAPPED TEMPERATURE = 1090 DEG KELVIN |
| RESIDUAL AIR FRACTION = 0.05 |
| FUEL SELECTED FOR THIS ANALYSIS = C8H18 (ISO OCTANE) WITH A LOWER HEATING VALUE = -4.2E+07 JOULES/KG |
| STOICHIOMETRIC AIR / FUEL RATIO = 15.1151 |
| FUEL / AIR EQUIVALENT (PHI) = 0.502836 |

<table>
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<tr>
<th>COMPRESSION CYLINDER AIR (DEGREE)</th>
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<th>CYLINDER (VOL%)</th>
<th>CYLINDER FUEL (VOL%)</th>
<th>CYLINDER (VOL%)</th>
<th>CYLINDER FUEL (VOL%)</th>
<th>CYLINDER (VOL%)</th>
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Figure 24
## Appendix A

**Specifications of Test Remley Engine**

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<tr>
<th>Specification</th>
<th>Value</th>
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<tbody>
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<tr>
<td>Bore</td>
<td>4.0 inches</td>
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<tr>
<td>Stroke</td>
<td>2.5 inches</td>
</tr>
<tr>
<td>Cylinder Displacement</td>
<td>31.41 cubic inches</td>
</tr>
<tr>
<td>Connecting Rod Length</td>
<td>6.25 inches</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>14.3 : 1</td>
</tr>
<tr>
<td>Number of Compression Rings</td>
<td>2</td>
</tr>
<tr>
<td>Number of Oil Rings</td>
<td>1</td>
</tr>
<tr>
<td>Number of Inlet Valves</td>
<td>2</td>
</tr>
<tr>
<td>Number of Exhaust Valves</td>
<td>2</td>
</tr>
<tr>
<td>Valve Diameter</td>
<td>1.286 inches</td>
</tr>
<tr>
<td>Valve Lift</td>
<td>0.280 inches</td>
</tr>
<tr>
<td>Inlet Valve Timing open/close</td>
<td>15°BTDC/50°ABDC</td>
</tr>
<tr>
<td>Exhaust Valve Timing open/close</td>
<td>50°BBDC/15°ATDC</td>
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<tr>
<td>Diameter of Intake Manifold Pipe</td>
<td>2.00 inches</td>
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<tr>
<td>Diameter of Exhaust Manifold Pipe</td>
<td>1.60 inches</td>
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Data Collected from Test Run

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<tbody>
<tr>
<td>Engine Speed in RPM</td>
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<td>Inlet Pressure</td>
<td>13.5 inches Hg gage</td>
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<td>Exhaust Pressure</td>
<td>13.4 inches Hg gage</td>
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<td>Inlet Air Temperature</td>
<td>186° F</td>
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<td>Air to Fuel Ratio</td>
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<td>IMEP</td>
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<td>Start of Injection</td>
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<td>Ignition Delay</td>
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<td>Period of Fuel Injection</td>
<td>17.5°</td>
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Appendix B

Computer Model

The computer program is written in TRS-80 Model III Disk Basic and consists of a main program and nine subroutines. The program listing has numerous remarks statements inserted to make the algorithm and computer code easier to understand. Since the program takes a considerable length of time to run, it is recommended that the remark statements be deleted before running. Samples of output are presented in figures 15 to 24.
'******************************************************
' MAIN PROGRAM
'******************************************************
'This program is written in TRS-80 Model III Disk Basic.
'Remove all remarks spaces before running to speed up run time.
'Dimension arrays.
'Array U contains the thermodynamic polynomial coefficients.
'Array F(5) and FD(5) are used in subroutines for calculating
'Thermodynamic data (i.e. enthalpy, internal energy, and moles).
'Arrays A(5) and B(5) contain number of moles of the
'Five species at the beginning of step, A, and at
'The end of the step, B.
'DIM U(5,5), F(5), FD(5), A(5), B(5)
'Define all variables starting with I & J as integers.
'DEFINT I,J
'Input data is requested from the operator -- WATCH UNITS
'INPUT"ENTER TODAY'S DATE";DATE$
'INPUT"ENTER RUN NUMBER";NUMBS
'INPUT"ENTER CYLINDER BORE IN METERS";D
'INPUT"ENTER STROKE IN METERS";S
'INPUT"ENTER CONNECTING ROD LENGTH IN METERS";L
'INPUT"ENTER ENGINE SPEED IN RPM";RPM
'INPUT"ENTER ENGINE COMPRESSION RATIO";CR
'INPUT"ENTER AIR / FUEL RATIO";AFR
'INPUT"ENTER PRESSURE AT START OF COMPRESSION IN N/M^2";P1
'INPUT"ENTER TEMPERATURE AT START OF COMPRESSION IN DEG K";T1
'INPUT"ENTER RESIDUAL GAS FRACTION";F
'INPUT"ENTER CRANK ANGLE FOR INTAKE VALVE SHUT";ALPHA
'INPUT"ENTER CRANK ANGLE FOR EXHAUST VALVE OPEN";AEVO
'INPUT"ENTER CRANK ANGLE FOR FUEL INJECTION ";AIJECT
180 INPUT "ENTER PERIOD OF FUEL INJECTION (DEGREES)";IJECT
185 INPUT "ENTER CRANK ANGLE INCREMENTS FOR THIS RUN";ADELT
190 INPUT "SELECT FUEL: (1) FOR C8H18 (OCTANE) (2) FOR C3H8 (PROPANE)";NN
195 'Load in fuel data for selected fuel.
200 IF NN=1 THEN GOSUB 4000 ELSE GOSUB 4100
205 'Load in constants with subroutine 4200
210 GOSUB 4200
215 'Subroutine 5000 calculates cylinder volume and surface area.
220 GOSUB 5000
225 V1=V
227 'Calculate phi and number of moles of fuel based on perfect combustion
230 GOSUB 5100
235 'Print out input data.
240 GOSUB 5200
250 PRES=P1*PBAR
255 IF AIJECT=ALPHA THEN LPRINT, "FUEL INJECTION START AT ";AIJECT; "COMBUSTION COMMENCED"
260 LPRINT,ALPHA,V1,PRES,T1,WRKT,DMF,KO
265 'Convert heat of reaction from J/Kg to J/Kgmole.
270 QVS=QVS*WF
275 'Calculate moles of all species at beginning of run.
280 A(1)=MOLE*CA*F  'Moles of Carbon dioxide
290 A(2)=MOLE*F*HA/2  'Moles of Water Vapor
300 A(3)=MOLE*SOX*(1-F)/PHI  'Moles of Oxygen
310 A(4)=3.76*A(3)  'Moles of Nitrogen
320 A(5)=0.0  'Moles of Fuel
330 FORII=1T05
350 NEXTII
355 'Y, I1, & I2 are values that are fed into subroutines 5500 & 6000
356 'These subroutines are used to calculate internal energy, enthalpy and moles.
The temperatures are raised to powers in the polynomial expressions. The values of I1 & I2 tell the subroutines for what species to solve.

Y=TS:I1=1:I2=4
FOR I=I1 TO I2
GOSUB 5500
NEXTI
GOSUB 6000

'Calculate internal energy, E, at reference temperature, TS.
ES1=RMOL*TS*F1
Y=T1
FOR I=I1 TO I2
GOSUB 5500
NEXT I
GOSUB 6000
M1=F3

'Calculate internal energy and Specific heat at temperature T1
E1=RMOL*T1*F1
C1V=RMOL*F2/F3

'Add crank angle interval to go to end of step
ALPHA=ALPHA+ADELT

'Calculate cylinder volume and area.
GOSUB 5000
V2=V
IF K0 >= 1.0 GOTO 540
IF ALPHA >= AIJECT THEN GOTO 560
DMF=0.0
GOTO 590
KK=1

'If injection has occurred, then go to Combustion Subroutine.
570 GOSUB 6500
580 IF KK = 2 GOTO 600
585 'First approximation at temperature.
590 T2=T1*(<V1/V2>+(RMOL/C1V))-(DMF*QVS*MOLES)/(C1V*M1)
595 'Calculate the internal energy after combustion at T5.
600 Y=TS+I2=4
610 FOR II=1 TO 5
620 X(II)=B(II)
630 NEXT II
640 FOR I=I1 TO I2
650 GOSUB 5500
660 NEXT I
670 GOSUB 6000
680 EES2=RMOL*TS*F1
685 'Calculate the internal energy & specific heat at T2.
690 Y=T2
700 FOR I=I1 TO I2
710 GOSUB 5500
720 NEXT I
730 GOSUB 6000
740 E2=RMOL*T2*F1
750 M2=F3
760 C2V=RMOL*F2/F3
765 'Calculate pressure at end of step - Ideal gas.
770 P2=(M2/M1)*(T2/T1)*(V1/V2)*P1
775 'Calculate heat transfer in subroutine 7000
780 GOSUB 7000
785 'Calculate work
790 DW=0.5*(P1+P2)*(V2-V1)
795 'Calculate error for Newton-Raphson iteration.
800  FE=(E2-ES2)-(E1-ES1)+DW-DQ+(DMF*MOLE*QVS)
810  EARER=FE/(M2*C2V)
815  \text{NRACC is the allowable error for Newton-Raphson Iteration.}
820  IFABS(EARER)<NRACCGOTO0860
830  T2=T2-EARER/2
835  \text{Recalculate energies and specific heats at "new" temperature.}
840  IF ALPHA < AIJECT GOTO 690
850  KK=2:GOSUB 6600
855  GOTO 600
856  \text{Convert pressure to bars.}
860  PRES=P2*PBAR
865  \text{'Cumulative work}
870  WRKT=WRKT+DW
875  \text{'Cumulative heat transfer.}
880  Q=Q+DQ
882  \text{'Cumulative heat release.}
885  KO=KO+DMF
890  LPRINT,ALPHA,V2,PRES,T2,WRKT,DMF,KO
892  IF YZ=10. GOTO 900
895  IF ZZ=1 THEN LPRINT,,,,"COMBUSTION COMPLETED" ELSE GOTO 900
896  YZ=10.
900  IF ALPHA = AEVO GOTO 2000
905  \text{'Shift end of step data to beginning of next step.}
910  P1=P2
920  V1=V2
930  T1=T2
940  E1=E2
950  ES1=ES2
960 C1V=C2V
970 M1=M2
975 PEP=PPEP
976 RACT=RRCT
980 FOR II=1 TO 5
990 A(II)=B(II)
1000 NEXT II
1010 GOTO 500
2000 LPRINT, "EXHAUST VALVE OPEN -- CYCLE COMPLETE"
2010 'Calculate power, IMEP, efficiency, and sfc.
2020 PWR=WRKT*RPM*1.2E-05
2030 MEP=WRKT*PBAR/VS
2040 EFFTH=100.0*WRKT/(-QVS*MOLE)
2050 SFC=(3.6E06*WF)/(-QVS*KO*EFFTH)
2060 LPRINT: LPRINT
2070 LPRINT TAB(20)"IMEP =";MEP;" BARS"
2080 LPRINT TAB(20)"POWER (4 STROKE) =";PWR;" KILOWATTS"
2090 LPRINT TAB(20)"SPECIFIC FUEL CONSUMPTION =";SFC;" KG/KW-HR"
2100 LPRINT TAB(20)"THERMAL EFFICIENCY =";EFFTH;" PERCENT"
2110 END
FUEL DATA

FUELS="C8H18 (ISO OCTANE)"

Polynomial coefficients for iso-octane

U(5,1)=-0.71993
U(5,2)=4.6426E-02
U(5,3)=-1.68385E-05
U(5,4)=-2.67009E-09
U(5,5)=0.0

CA=8: HA=18  'Carbon atoms = 8, Hydrogen atoms = 18
QVS=-4.2E07  'Lower heating value in J/Kg.

RETURN

Polynomial coefficients for propane.

FUELS="C3H8 (PROPANE)"

U(5,1)=1.13711
U(5,2)=1.45532E-02
U(5,3)=-2.95876E-06
U(5,4)=0.0
U(5,5)=0.0

QVS=-4.63E07  'Lower heating value in J/Kg.

RETURN
4200 '*********************************************************************
4201 '********** Constants and other input data **********
4202 '*********************************************************************
4208 'This subroutine loads various constants and
4209 'polynomial coefficients for CO2, H2O, N2 and O2.
4210 P0=101325  'Reference pressure in N/M^2
4220 T5=298  'Reference temperature in degrees Kelvin.
4230 PI=3.1415927
4240 RD=180/PI  'Converts from degrees to radians
4250 RMOL=8314.3  'Newton-meter/(kg-mole)K
4260 WRKT=0.0;Q=0.0
4270 PBAR=1E-05  'Conversion from N/m^2 to bars.
4271 K1=0.014  'Preparation rate coefficient 0.008-0.022.
4272 K2=2/3  'Preparation rate coefficient x.
4273 K3=6.5E11  'Reaction rate coefficient K'.
4274 K4=1.5E4  'Reaction rate coefficient act.
4275 NRACC=1.0  'Newton-Raphson Accuracy.
4276 G1=0.26;G2=0.75;G3=3.88E-08  'Anand equation coefficients a,b,&c.
4277 TW=750;PR=0.7  'Wall temperature (assumed) and Prandtl Number.
4278 M(1)=24E-06;M(2)=20E-06;M(3)=32E-06;M(4)=29E-06  'viscosity of species.
4280 FOR II=1 TO 5
4290 A(II)=0.0:B(II)=0.0
4300 NEXT II
4310 'Thermodynamic data preparation  U(I,J)
4320 'I=Species  J=Coefficient
4330 'Species: 1=CO2;  2=H2O;  3=O2;  4=N2;  5=FUEL
4350 'Carbon Dioxide
4360 U(1,1)=3.0959
4370 U(1,2)=2.73114E-03
4380 U(1,3)=-7.88542E-07
4390 U(1,4)=8.66002E-11
4400 U(1,5)=0.0
4430 'Water Vapor
4440 U(2,1)=3.74292
4450 U(2,2)=5.65590E-04
4460 U(2,3)=4.95240E-08
4470 U(2,4)=-1.81802E-11
4480 U(2,5)=0.0
4510 'Oxygen
4520 U(3,1)=3.25304
4530 U(3,2)=6.52350E-04
4540 U(3,3)=-1.49524E-07
4550 U(3,4)=1.53897E-11
4560 U(3,5)=0.0
4590 'Nitrogen
4600 U(4,1)=3.34435
4610 U(4,2)=2.94260E-04
4620 U(4,3)=1.95300E-09
4630 U(4,4)=-6.57470E-12
4640 U(4,5)=0.0
4700 'Volume of cylinder at BDC
4710 VS=PI*S*(D/2.0)^2
4715 'Volume at TDC.
4720 VC=VS/(CR-1)
4730 N=L/(S/2)
4735 'Total cylinder volume.
4740 VT=VS+VC
4745 'Areas in cylinder.
4750 AC=4*VC/D
4760 AS=S*PI*D
4770 AT=AS+AC
4780 RETURN
5000  *********** VOLUME & AREA OF CYLINDER
5001  ***********
5010  SWEEP=(1+N)^(N+2.0-(SIN(ALPHA/RD))/1.0/0.5-COS(ALPHA/RD))
5020  V=VT-(VS/2)*SWEEP
5030  AREA=AT-(AS/2)*SWEEP
5040  RETURN
5100 '*****************************************************************************
5101 '********** FUEL CALCULATIONS **********
5102 '*****************************************************************************
5105 'Calculate Number moles of fuel.
5110 SOX=CA*HA/4
5120 WF=12.0*CA*HA
5125 'Calculate air fuel ratio - stoichiometric.
5130 ASTF=4.76*SOX*28.96/WF
5140 PHI=ASTF/AFR
5145 'Calculate mole of fuel for perfect combustion.
5150 MOLE=P1*V1*PHI/(4.76*SOX*RMOL*T1)
5160 RETURN
5200 '**************************************************************************
5201 '*****PRINT SUBROUTINE*************
5202 '**************************************************************************
5205 LPRINT CHR$(27);"6";LPRINT CHR$(27);CHR$(49);CHR$(30)
5210 LPRINT:LPRINTTAB(25)"DIESEL ENGINE COMBUSTION CYCLE";LPRINT
5220 LPRINT TAB(25);"RUN: ";NUMBS;" ";DATES
5230 LPRINT TAB(10)"INPUT DATA:"
5240 LPRINT TAB(20)"CYLINDER BORE = ";D;"METERS"
5250 LPRINT TAB(20)"STROKE = ";S;"METERS"
5260 LPRINT TAB(20)"CONNECTING ROD LENGTH = ";L;"METERS"
5270 LPRINT TAB(20)"ENGINE SPEED = ";RPM;"RPM"
5280 LPRINT TAB(20)"ENGINE COMPRESSION RATIO = ";CR
5290 LPRINT TAB(20)"AIR / FUEL RATIO = ";AFR
5300 LPRINT TAB(20)"TRAPPED PRESSURE = ";P1;"N/M²"  
5310 LPRINT TAB(20)"TRAPPED TEMPERATURE = ";T1;"DEG KELVIN"
5320 LPRINT TAB(20)"RESIDUAL AIR FRACTION = ";F
5330 LPRINT TAB(20)"FUEL SELECTED FOR THIS ANALYSIS = ";FUELS
5340 LPRINT TAB(20)" WITH A LOWER HEATING VALUE = ";QVS;"JOULES/KG"
5350 LPRINT TAB(15)"STOICHIOMETRIC AIR / FUEL RATIO = ";ASTF
5360 LPRINT TAB(15)"FUEL / AIR EQUIVALENCE (PHI) = ";PHI
5400 LPRINT CHR$(27);CHR$(48);CHR$(29)
5405 LPRINT CHR$(27);"B"
5410 LPRINT,"COMPRESSION"," CYLINDER"," CYLINDER"," CYLINDER"," CYLINDER"," FUEL","CUMULAT:  
5420 LPRINT," ANGLE"," VOLUME"," PRESSURE"," TEMPERATURE"," WORK"," IN STEP"," FUEL"
5430 LPRINT," (DEGREES)"," (M³)"," (BAR)"," (DEG K)"," (JOULES)"","(FRACTION)"","(FRACTION"
5440 LPRINT,------------",------------",------------",------------",------------",------------"
5450 RETURN
5500 '*****************************************************************************
5505 '****** THERMODYNAMIC PROPERTIES OF MIXTURES ******
5510 '*****************************************************************************
5520 F=0.0
5530 FD=0.0
5550 FOR J=1 TO 5
5560 LET Z=J
5570 LET L=J-1
5580 F=F+U(I,J)*Y*(Z-1.0)
5600 FD=FD+2*U(I,J)*Y*(Z-1.0)
5620 NEXT J
5625 F(I)=F
5626 FD(I)=FD
5630 RETURN

6000 '*****************************************************************************
6005 '****** THERMODYNAMIC PROPERTIES OF MIXTURES ******
6006 '*****************************************************************************
6010 F1=0.0
6020 F2=0.0
6030 F3=0.0
6040 FOR I=I1 TO I2
6050 F1=F1+X(I)*(F(I)-1.0)
6070 F2=F2+X(I)*(FD(I)-1.0)
6090 F3=F3+X(I)
6100 NEXT I
6120 RETURN
6500 '*****************************************************************************
6501 '********** COMBUSTION SUBROUTINE **********
6502 '*****************************************************************************
6506 IF ALPHA = AIJECT LPRINT, , "FUEL INJECTION START AT ";AIJECT,"COMBUSTION COMMENCED"
6507 IF ALPHA = (AIJECT+IJECT+ADELT) LPRINT, ,"FUEL INJECTION STOP AT ";ALPHA-ADELT
6508 IF KO >= 1.0 GOTO 540
6510 FOR II=1 TO 5
6515 X(II)=A(II)
6520 NEXT II
6540 'Variable MIJECT is the total fuel injected to this point.
6545 IF MIJECT = (MOLE*WF) THEN GOTO 6600
6550 MIJECT = MIJECT+MOLE*WF*ADELT/IJECT
6590 'Calculate Partial Pressure of Oxygen
6600 P02=P2*PBAR*A(3)/M1
6605 'Calculate the fuel that has been prepared in this step.
6610 MASUNB=MIJECT-PEP
6615 Z1=MIJECT*(1-K2)
6616 Z2=MASUNB*K2
6617 Z3=P02+0.4
6620 PN1=K1*Z1*Z2*Z3
6625 'Calculate the cumulative fuel Prepared kg.
6630 PPEP=PEP+(PN1*ADEL)
6635 'Check to see if reaction rate less than preparation
6640 IF PPEP <= RACT GOTO 6690
6650 COEF=K3*P02/(RPM*SQR(TM))
6660 R1N=COEF*(PPEP-RACT)*EXP(-K4/TM)
6665 'Calculate cumulative fuel reacted kg.
6670 RRCT=RACT+(R1N*ADEL)
6680 IF RRCT < PPEP THEN DF=R1N*ADEL ELSE DF=PN1*ADEL:GOTO6691
6690 DF=PN1*ADELT
6691 DMF=DF/(WF*MOLE)
6692 IF (K0+DMF)>1.0 THEN DMF=(1.0-K0)
6693 IF (K0+DMF)=1.0 THEN ZZ=1
6700 'Calculate the amount of fuel burned in this step.
6701 A(5)=DMF*MOLE:Y=TS:I2=5
6702 FOR I=1 TO I2:GOSUB 5500:NEXT I:GOSUB 6000
6703 ES1=RMOL*TS*F1:Y=T1
6704 FOR I=1 TO I2:GOSUB 5500:NEXT I:GOSUB 6000
6705 M1=F3-A(5)
6706 E1=RMOL*T1*F1
6707 C1V=RMOL*F2/F3
6709 'Calculate moles of products after combustion
6710 B(1)=A(1)+(DMF*MOLE*CA)
6720 B(2)=A(2)+(DMF*MOLE*HA/2)
6730 B(3)=A(3)-(DMF*MOLE*SOX)
6740 B(4)=A(4)
6750 B(5)=0.0
6760 RETURN
7000 '***********************************************
7010 '********** HEAT TRANSFER-ANNAND EQUATION **********
7020 '***********************************************
7055 'Calculate piston speed
7060 VP=2*S*RPM/60
7065 'Calculate mean temperature
7070 TM=(T1+T2)/2
7100 'Calculate the mixture viscosity from the individual specie's viscosity.
7120 MU=X(1)*M(1)+X(2)*M(2)+X(3)*M(3)+X(4)*M(4)
7125 'Calculate Specific Heat Cp
7130 CP=C2V+RMOL/M2
7135 'Calculate Conductivity
7140 K=CP*MU/PR
7145 'Calculate density
7150 R0=P2*M2/(RMOL*T2)
7155 'Calculate Reynold's Number
7160 RE=R0*D*VP/MU
7165 'Convective Term from Annand's Equation
7170 CVECT=G1*K*(RE+G2)*(TM-TW)/D
7175 'Radiation Term from Annand's Equation
7180 CRAD=G3*(TM^4.0-TW^4.0)
7190 IF ALPHA < AIJECT THEN QDT=CVECT ELSE QDT=CVECT+CRAD
7195 'Calculate heat loss this iteration
7200 DG=AREA*QDT*ADELT/(6*RPM)
7210 RETURN
Combustion modeling of a two cylinder cycle reciprocating engine.