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Abstract

Due to the increased emphasis on vehicle fuel economy and performance, there is ongoing discussion about the potential for optimizing cylinder geometry as a means to improve engine efficiency and performance. In this thesis, a thermal efficiency model was developed that explicitly evaluated the effects of cylinder geometry on the combustion process. This thermal efficiency model was combined with the upgraded volumetric efficiency model and the mechanical efficiency model from a previous study to determine performance and fuel consumption as a function of engine speed, manifold absolute pressure, compression ratio, bore-stroke ratio, and displaced volume.

With a constant mean piston speed and at wide-open-throttle, best performance was obtained at a compression ratio near 8.5:1, a bore-stroke ratio near 0.8, and a displaced volume of about 460 cubic centimeters per cylinder. With the mean piston speed set to give maximum performance and at wide-open-throttle, best performance was obtained at a compression ratio near 8:1, a bore-stroke ratio near 0.75, and a displaced volume of about 425 cubic centimeters per cylinder. The results indicate that a higher compression ratio, lower bore-stroke ratio, and higher displaced volume will yield reduced fuel consumption. The best design configuration was found to be a compromise between desired performance and economy. These results demonstrate the type of studies and comparisons that could be made with the model.

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Title: Professor of Mechanical Engineering
        Director, Sloan Automotive Laboratory
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This is the space where the author gets to thank all the people who helped in the production of this study. A task such as this can appear more difficult than the research itself. To those who helped but are not mentioned by name here, my apologies.

First I'd like to thank my advisors, Professor John B. Heywood and Dr. Victor Wong for their patience and assistance throughout the evolution of this project. My knowledge of internal combustion engines has evolved from knowing little to a fairly competent understanding of the fundamentals of spark-ignition engines in large part due to their help. In addition, I must also thank Ron Nitschke, who developed the first part of the project, and Ken Patton, who is currently improving the model, for their practical insight and discussions.

I owe a tremendous thanks to the students and staff of Sloan Automotive Lab for their technical support and friendship. In particular, the computer jockeys of the lab, Mark Sztenderowicz and Rick Frank, deserve my gratitude for their help in dealing with the VAX750 computer. Finally, I must thank all the present and past members of room 31-061, or the "Mass Pike", for their friendship, help, support, and diversions they provided. The members of this proud group were: Ray Stanten, Djamel Hamiroune, Ron Nitschke, Francois Facon, Ken Patton, Pyongwan Park, Dennis Szydloeski, and Steve Billian.

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<tr>
<td>MAP</td>
<td>inlet manifold absolute pressure, kPa</td>
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<tr>
<td>$S_L$</td>
<td>laminar flame speed for combustion, m/s</td>
</tr>
<tr>
<td>$u'$</td>
<td>turbulent velocity for combustion, m/s</td>
</tr>
<tr>
<td>$\rho_u$</td>
<td>density of unburned mixture, m$^3$/kg</td>
</tr>
<tr>
<td>$A_f$</td>
<td>flame front area, m$^2$</td>
</tr>
<tr>
<td>$m$</td>
<td>total mass in cylinder, kg</td>
</tr>
<tr>
<td>$x_e$</td>
<td>total entrained mass fraction</td>
</tr>
<tr>
<td>$x_b$</td>
<td>total burned mass fraction</td>
</tr>
<tr>
<td>$\dot{x}_e$</td>
<td>mass fraction entrainment rate, 1/s</td>
</tr>
<tr>
<td>$\dot{x}_b$</td>
<td>mass fraction burning rate, 1/s</td>
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<tr>
<td>$\tau_b$</td>
<td>characteristic time scale for combustion, s</td>
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<tr>
<td>$V_d$</td>
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<td>cylinder bore, mm</td>
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<td>$L$</td>
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<td>$V_c$</td>
<td>clearance volume, cm$^3$</td>
</tr>
<tr>
<td>$r_c$</td>
<td>compression ratio</td>
</tr>
<tr>
<td>$h$</td>
<td>height of combustion chamber, mm</td>
</tr>
<tr>
<td>$a$</td>
<td>radius of piston, mm</td>
</tr>
<tr>
<td>$R_{sp}$</td>
<td>radius of sphere defining combustion chamber, mm</td>
</tr>
<tr>
<td>$h_s$</td>
<td>height of spark plug electrode, mm</td>
</tr>
<tr>
<td>$a_s$</td>
<td>radius of spark plug electrode, mm</td>
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<tr>
<td>IMEP</td>
<td>indicated mean effective pressure, kPa</td>
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<tr>
<td>MBT</td>
<td>maximum brake torque, N-m</td>
</tr>
<tr>
<td>$\eta_{id}$</td>
<td>ideal thermal efficiency, %</td>
</tr>
<tr>
<td>$\Theta$</td>
<td>dummy parameter</td>
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<tr>
<td>$r_c$</td>
<td>compression ratio</td>
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<tr>
<td>$\gamma$</td>
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</tr>
<tr>
<td>$S/V$</td>
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<tr>
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<td>$\eta_{cr}$</td>
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<td>$C$, $C_1$</td>
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<tr>
<td>$\phi$</td>
<td>equivalence ratio</td>
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$\eta_t$: engine thermal efficiency, %
$\eta_c$: combustion efficiency, %
$\eta_{th}$: thermodynamic cycle efficiency, %
$\eta_\phi$: power enrichment sub-model thermal efficiency, %
$h$: convective heat transfer coefficient, W/m$^2$K
$p$: cylinder pressure, kPa
$T$: cylinder temperature, K
$W$: local mean gas velocity, m/s
$\Delta T$: temperature difference in convection equation, K
$N$: engine speed, revolutions/minute
$A$: convective heat transfer area, m$^2$
$a, b, c, d$: exponent constants in Woschni heat transfer equation
$Q_{map}$: heat transfer due to manifold absolute pressure, J
$V$: cylinder volume, cm$^3$
$m$: cylinder mass, kg
$R$: gas constant for cylinder gases, kJ/kg-K
$\zeta$: exponential constant for sub-model relations
$\eta_{map}$: MAP sub-model thermal efficiency, %
$Q_N$: heat transfer due to variation of engine speed, J
$\eta_N$: engine speed sub-model thermal efficiency, %
$B/L$: bore-stroke ratio
$Q_{B/L}$: heat transfer due to variation of bore-stroke ratio, J
$\eta_{B/L}$: bore-stroke ratio sub-model thermal efficiency, %
$Q_{Vd}$: heat transfer due to variation of displaced volume, J
$\eta_{Vd}$: displaced volume sub-model thermal efficiency, %
$\eta_{ti}$: maximum indicated engine thermal efficiency, %
$\bar{S}_p$: maximum allowable mean piston speed, m/s
$\text{ON}$: fuel octane number, RON
$T_{sp}$: spark timing, degrees
$\text{SR}$: spark retard required to avoid knock, degrees
$\Delta \eta_{ti}$: thermal efficiency loss due to spark retard, %
$F/A$: fuel-air ratio
$\eta_v$: volumetric efficiency, %
$m_a$: mass flow rate of air, kg/s
$\rho_{ai}$: inlet air density, kg/m$^3$
$Q_{HV}$: heating value of fuel, kJ/kg
SFC: specific fuel consumption (brake or indicated), μg/J

$n_R$: number of revolutions per power stroke

$P$: power (brake or indicated), kW

$T$: torque (brake or indicated), N·m

$MEP$: mean effective pressure (brake or indicated basis), kPa

$\eta_m$: mechanical efficiency, %

$WOT$: wide-open-throttle manifold pressure, kPa
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Chapter 1

Introduction

Many challenges face the American automotive industry both now and in the future. Competition, both domestic and foreign, is requiring automobile manufacturers to improve product quality and performance. In addition, rapidly changing consumer tastes and preferences require a quick and adaptable automotive industry. Fluctuations in the price of fuel greatly determine consumer preference and subsequently the view of automotive manufacturers toward economy and power [1].

Despite these market trends, spark ignition engine geometry, for a given application, varies substantially between engine manufacturers and within a single manufacturer. Cost and manufacturing considerations often contribute to the use of common components in engines of different applications. However, a lack of understanding of the optimum engine geometry has also contributed to the wide diversity of spark-ignition engine geometries. Both these factors contribute to the automotive industry's difficulty in meeting performance and economy objectives quickly and efficiently.

Bore, stroke, and compression ratio are examples of basic engine geometry parameters that vary substantially in spark-ignition engines. Factors such as fuel consumption and power which effect vehicle performance and economy are dependent on these basic engine geometry parameters. After more than a century of engine development and testing, substantial data and theory describing performance and fuel consumption exist. Using the existing data and theory, a more logical and quantitative methodology for determining optimum engine geometry parameters for a given application can be developed. While preliminary performance studies have always been used in engine design, a methodology that explicitly relates performance to engine geometry would greatly decrease the number of engine designs that need to be investigated with hardware, which is both expensive and time consuming [2]. This would help make the automotive industry more adaptable and quicker to respond to changing market demands.

1.1 Overall Program Goal

The overall goal of this project was to develop relationships that accurately describe the effect of spark-ignition engine geometry on power and efficiency. This required the development of quantitative relationships that scale performance and fuel
consumption with engine geometry variables. Combining these relationships defines the optimum geometry for a given application.

This project was the second of a three part study at M.I.T. designed to meet the overall goals described previously. The study was initiated by Wayne Daniel [3] at General Motors Advanced Engineering Staff and continued at M.I.T. by Ronald Nitschke [1]. To help understand the objectives of part of the M.I.T. program, the results of the previous two studies [1,3] will be briefly reviewed.

1.2 Background

The first step in accomplishing the overall goal described previously was taken in a study done by Wayne Daniel [3]. Relationships which showed how thermal efficiency, volumetric efficiency, and mechanical efficiency scaled with engine geometry were combined to predict the effect of geometry changes on performance and fuel consumption. The main drawback to the usefulness of this study was use of scaling according to a "reference" engine geometry.

The initial M.I.T. study was conducted by Ronald Nitschke [1] and was designed to address the same issues as the previous study but in a qualitative predictive manner without reliance on a "reference" engine geometry. Like the previous study, volumetric, mechanical, and thermal efficiencies were the parameters to be predicted. A combination of these three parameters was then used to determine the performance and fuel consumption of the engine geometry. The input variables considered in this analysis were engine speed, manifold absolute pressure, bore, stroke, number of cylinders, compression ratio, equivalence ratio and valve lift and timing. The predicted mechanical and volumetric efficiencies showed acceptable agreement with data in the range of validity of the model.

However, the thermal efficiency model was limited to a small range of data generated by a cycle simulation computer program developed by Watts and Heywood [4]. The primary drawback of this model was that the Watts and Heywood cycle simulation was restricted to one engine geometry configuration. As a result, the thermal efficiency model used a geometric scaling method similar to the previous study. Although the results of this thermal efficiency model were necessary to obtain performance and economy characteristics, it did not explicitly contain important engine geometry parameters.

Also, because of the limited data sets used to develop the volumetric and mechanical efficiencies, some problems were found to occur outside the prescribed data range. In particular, at low inlet manifold absolute pressure (MAP), the volumetric
efficiency model did not predict realistic values. The data used to generate the volumetric efficiency model did not include low load data; the addition of low load data would increase the usefulness of the volumetric efficiency model and the model as a whole.

1.3 Objective

The objectives of this study were: (1) to develop an improved thermal efficiency model that includes the effects of engine geometry explicitly; (2) to develop a methodology for implementing the constraints that limit engine performance; and (3) improve the operating range of the volumetric efficiency model currently being used.

Once these objectives were accomplished, they were implemented into the model developed by Nitschke. With the combined model, the performance and fuel consumption were determined as a function of engine geometry. Comparison and validation was accomplished using production engine data.

1.4 Definition of Thermal Efficiency

For this study, thermal efficiency was defined as the amount of work produced per cycle over the compression and expansion strokes divided by the amount of fuel energy supplied per cycle that could be released in the combustion process [5]; this is often called gross indicated thermal efficiency. Note that this definition does not include pumping work; pumping work was included under the mechanical efficiency model designed by Nitschke.

1.5 Methodology

The methodology for developing the thermal efficiency model was complicated by the fact that this quantity is not normally measured directly during engine testing. To include the effects of geometry on thermal efficiency explicitly, a cycle simulation computer was used to develop a data base from which the model could be derived. The program was updated to include recent engine combustion modelling discoveries and modified to meet the requirements of our this study. The cycle simulation was then validated using experimental engine data previously generated at M.I.T. A test matrix was established and completed. With the data established, the individual effects of operating conditions and design variables were quantified and a thermal efficiency model developed and tested.

The constraints were developed using a combination of cycle simulation data and experimental engine data. The high speed constraint was developed using data from
the cycle simulation. The knock constraint was developed using production engine data and theories developed in other studies while the losses due to the spark retard required for knock avoidance were estimated using cycle simulation data.

Finally, the thermal efficiency model and the constraints were combined with the Nitschke volumetric and mechanical efficiency models. The range of valid inputs was increased to include low inlet pressure cases. The total model was then compared and validated against experimental engine results.

The remaining chapters in this document are arranged around this logic. Chapter 2 contains a brief description of the cycle simulation, the modifications made, the validation, and the resulting database. Chapter 3 shows the methodology followed in developing the thermal efficiency model and its comparison with the engine data. Chapter 4 covers the definition, development, and implementation of the engine constraints. Chapter 5 describes the improvements made in the model to increase the acceptable range of loads. Chapter 6 shows the results of the entire process, comparisons with real engines and the results of other similar studies, and an optimization study showing the capability of the model. Chapter 7 concludes with a summary and some final remarks.
Chapter 2

Cycle Simulation

An engine cycle simulation was used to account explicitly for the effects of cylinder geometry and operating contributions on gross indicated thermal efficiency. The cycle simulation was modified to include more recent engine research and to be more useful for the purposes of this study. Comparisons with actual engine test data were done to validate the cycle simulation. Finally, gross indicated thermal efficiency data was generated from the validated model.

2.1 Description of the SI Cycle Simulation

A engine cycle simulation is a mathematical model of the processes occurring in an engine over a complete cycle. The spark ignition engine cycle simulation used was a quasi-dimensional simulation developed by Poulos [6,7]. The quasi-dimensional simulation combined a zero-dimensional or thermodynamic model that follows the changes in the bulk properties of the cylinder charge with a phenomenological model for combustion which included the gross interaction between the propagating flame and the combustion chamber. This model allowed general changes in combustion chamber geometry and its influence on performance and fuel consumption to be investigated. The simulation was designed to predict the details of the combustion process over a broad range of operating and design conditions including the details of the combustion chamber geometry. Figure 2.1 shows the flow chart of the general thermodynamic analysis simulation.

2.1.1 Interaction between Propagating Flame and Chamber Geometry

The key feature of the cycle simulation used for this study was the combustion model that included the geometric interaction between propagating flame and combustion chamber geometry. To investigate the interaction between the flame and the combustion chamber, basic assumptions concerning the propagating flame, the model of the combustion chamber geometry, and the combustion calculation procedure were made.

The propagating flame in this cycle simulation was assumed to develop as a sphere with its center fixed at the spark plug electrodes and truncated by the chamber walls. In experimental engines, photographic evidence [8,9] indicates that the flame starts as a small, roughly spherical kernel centered at the electrode. Because of
turbulent cylinder flows, the flame shape becomes distorted and the flame center may be convected away from the electrodes in some cycles. In fact, this phenomenon is most likely a major cause of cycle-by-cycle variations in spark-ignition engines [10]. Nevertheless, since the cycle simulation has shown good agreement with experimental data and cycle-by-cycle variation was not a major concern of this study, possible convection of the flame center was not an issue.

As the flame propagated outward, it was truncated by the walls of the combustion chamber. The surface of the combustion chamber was approximated by flat triangular facets as shown in Figure 2.2. Each point was located in space by a set of x, y, z coordinates with the origin located at the intersection of the axis of symmetry of the engine cylinder with the mating plane between the cylinder and the cylinder head. Each facet was then described by the three points at its apices. The location of the spark plug was specified by its x, y and z coordinates. In addition, to account for piston motion, the facets on the piston surface were shifted vertically and the facets on the cylinder wall were stretched as crackangle varied. Thus, the combustion chamber shape was accurately described for all crankangles.

With the evolution of the propagating flame and the combustion chamber accurately defined, the interaction between the two was modelled using a geometric principle [6] and a pseudo-random number generator (see Figure 2.3). At a given crackangle, the flame front area was calculated for a given flame radius. The process was repeated for several flame radii at several different crankangles. In the process, inflamed volume and wetted wall area were also calculated. In the manner described above, the details of the combustion process and its interaction with the combustion chamber were determined.

2.2 Modifications

Although the cycle simulation allowed the investigation of the influence of engine geometry on thermal efficiency, some modifications were required to include recent research advances made in the modelling of the flame propagation process and to make the model more useful for the purpose of this study. The modifications of the cycle simulation included adding more recent combustion terms, automating the chamber geometry generating procedure, adding a more realistic fuel type, and modifying the heat transfer model. Each of these modifications will be discussed separately. The culmination of these changes resulted in a more realistic cycle simulation that was more useful for the purpose of this study. Figure 2.4 shows the overall program logic of the modified cycle simulation. The inputs include the geometric parameters (bore,
stroke, compression ratio, and spark location), the operating conditions (engine speed, inlet pressure, equivalence ratio, etc.), and system iteration constants. The geometric inputs were used to generate the geometric data, points and facets that describe the combustion chamber. This data was then used to generate flame propagation data depicting the interaction between the flame and the combustion chamber. The flame propagation data along with the operating conditions and engine design parameters were used by the cycle simulation to predict the performance and fuel consumption as well as detailed results for the four stroke cycle. This data was then used for the data base of our thermal efficiency model.

2.2.1 Modification of the Combustion Terms

The modification of the burning and entrainment rates in the cycle simulation was done to include the results of Beretta, Rashidi, and Keck [11]. In their research, they showed the effect of turbulence and laminar flame speed on burning and entrainment rates. As Figure 2.5 shows, the laminar flame speed, $S_1$, and turbulent intensity, $u'$, cause the leading edge of the flame to propagate forward with a roughly spherical leading edge. As this is occurring, the turbulence causes the flame front to become wrinkled and non-uniform with pockets of unburned charge mixed in with the burned mixture (see expansion in Figure 2.5). These pockets of unburned mixture then burn as the flame front continues to advance. At the start of combustion, the flame propagates as sphere centered at the spark electrode. In the early stages of combustion, turbulence has not had time to develop, and the flame propagates at the laminar flame speed with no effects of turbulence present. The revised burning and entrainment rate equations were developed to include these effects:

$$\dot{x}_b = \frac{\rho_u A_f S_1}{m} + \frac{(x_e - x_b)}{\tau_b}$$

$$\dot{x}_e = \frac{\rho_u A_f [u' (1 - e^{-t/\tau_b}) + S_1]}{m}$$

Equations 2-1 and 2-2 show the mathematical model of the physical effects described above. In the entrainment equation (Equation 2-2) the first term, $u' (1 - \exp(-t/\tau_b))$, represents the time lag required for turbulence to begin to have a significant impact on the entrainment rate while the second term, $S_1$, represents the flame entrainment rate at laminar flame speed. In the burning equation (Equation
2-1), the first term, \( \rho_u A_f S_l / m \), represents the burning of the entrained mass at laminar flame speed while the second term, \( (x_e - x_b) / \tau_b \), represents the burning of the pockets of unburned mixture in the burned region.

The addition of the new terms in the burning and entrainment formulas of the cycle simulation increased the accuracy of the flame development model and took into account the effects of turbulence more fully. As expected, when these changes were implemented into the simulation, the burning rate decreased accordingly (ie. occurred later in the cycle). Figure 2.6 shows the effect of the changes on the burning and entrainment rates. The additional entrainment term resulted in faster entrainment and burning while the additional exponential term in the burning equation resulted in slower initial development and subsequently slower entrainment and burning.

Combining these effects, the process was slower than previously modelled. These equations were thought to better model the burning and entrainment rates in a real engine.

2.2.2 Automation of Chamber Geometry

In the past, the points and facets that describe the chamber geometry were input by hand (ie. the user determined where to locate the points and facets). This method worked fine when only few geometries were investigated. For this study, such a procedure was inconvenient and time consuming. Because this study required the use of many different chamber sizes, the procedure of describing the chamber with points and facets was automated to make it easy to change the geometric inputs of the combustion chamber.

The chamber shape was assumed to be hemispherical, which means that the combustion chamber shape was essentially a portion of a sphere. In all cases, the piston was assumed to be a standard flat piston. With these assumptions and the standard geometric inputs (compression ratio, bore, and stroke), the points and facets describing the geometry could be found by applying simple geometric relationships [12].

The displaced volume was calculated from the given bore and stroke by:

\[
V_d = \frac{\pi}{4} B^2 L
\]  

(2-3)

The clearance volume was found from the displaced volume and compression ratio.
\[ V_c = \frac{V_d}{r_c - 1} \quad (2-4) \]

The height of the combustion chamber was found by solving the cubic equation involving the clearance volume and the radius of the piston:

\[ h^3 + 3 a^2 h - \frac{6 V_c}{\pi} = 0 \quad (2-5) \]

The hemispherical chamber was a portion of a sphere. The radius of this sphere was calculated from the following equation:

\[ R_{s p} = \frac{3 V_c}{\pi h^2} + \frac{h}{3} \quad (2-6) \]

The location of the spark plug in the x and y plane was input by the user. The height of the spark plug could then be found from the given spark plug location and the known sphere radius,

\[ h_s = h - R_{s p} \left( R_{s p}^2 - a_s^2 \right)^{1/2} \quad (2-7) \]

With the chamber height and radius known from equations 2-3 and 2-7, the points and facets were defined at incremental height levels to define fully the combustion chamber.

By approximating the chamber shape with flat triangular facets instead of curved facets, the effective volume of the combustion chamber was reduced. Equations 2-3 through 2-7 assumed the triangular facets could describe the chamber geometry exactly. To rectify the problem of using flat facets to approximate the chamber, an iterative procedure was performed that scaled the location of the points, and subsequently the facets, to give the correct piston area and clearance volume. The iteration was repeated until the calculated piston area and clearance volume using the points and
facets were within a tenth of a percent of the desired input values. Using this procedure, the chamber geometry was accurately described by the points and facets.

2.2.3 The Addition of Indolene Fuel

In order to simulate more accurately real engine operation, an additional fuel had to be added to the existing fuels in the cycle simulation. Indolene was a more realistic fuel because its properties closely resemble that of gasoline. Indolene fuel had a wide range of chemical compositions. From the sample present in the lab, the composition chosen for this study was $C_7H_{13}$. The properties and fuel enthalpy coefficients [5,13,14] were calculated and implemented into the cycle simulation. Table 2.1 shows the value of the chemical constants used for indolene fuel.

2.2.4 Valve Size

The size of the valves in the cycle simulation were initially set at a constant value. For smaller ranges of geometric variation, this requirement would not make a significant difference to the combustion process. However, our study requires a large range of geometric variables and the requirement of constant valve size would not simulate real engines. Therefore, a new modification had to be added to the program that scaled the valve size with the input geometry.

The valve scaling procedure was based on geometric relationships commonly used in engine design [5,15]. Table 2.2 shows the rules relating valve geometry to chamber geometry. The resulting changes made the valves more closely emulate real engine valves and made the cycle simulation output more realistic.

2.2.5 Characteristic Velocity

At low load and low speed, the cycle simulation appeared to give unreasonably high heat transfer values. Further investigation showed that the high heat transfer was primarily due to the presence of a very high characteristic velocity when backflow occurred through the exhaust valve. One of the components of the characteristic velocity was the velocity corresponding to the mean kinetic energy. This was shown to rise to an unrealistically high value when only a small amount of mass actually flowed back through the exhaust valve. Since the backflowing mass was so small, it seemed implausible that the kinetic energy could rise by such a large margin. To eliminate the problem, it was decided to make the derivative of kinetic energy equal to zero when backflow occurred. This resulted in a more logical curve at low loads and speeds and made "physical" sense since only a small mass was
backflowing. The fact that this change effectively under-predicts the effects of valve backflow was known, but the effects were small enough to be of little concern in this study. Because the problem occurred only at low loads and speeds, the modification did not affect the majority of the data base.

2.3 Validation

With the preceding modifications applied to the cycle simulation, it was capable of generating data in an easy and reliable fashion. However, to insure that the cycle simulation accurately simulated real engine data, it was necessary to calibrate the model with experimental engine test data. Several parameters were adjusted to obtain a good match with the engine data. The most important of these were the heat transfer constant, the turbulent intensity multiplier, and the wall and inlet temperatures. These parameters were varied so that the cycle simulation results closely matched the engine data.

2.3.1 Ricardo Engine

Data from a Ricardo Hydra NMK III engine data was used to calibrate the cycle simulation. This single-cylinder spark-ignition engine had a hemispherical combustion chamber with a convex hump in the center of the piston as seen in Figure 2.7. The geometric features of the combustion chamber include a roughly square bore and stroke with a normal compression ratio. The specific geometric details of the combustion chamber are listed in Table 2.3. It was assumed that this geometric configuration could be modelled as a hemispherical chamber with a flat piston.

2.3.2 Ricardo Engine Data

The data from the Ricardo engine were obtained at a variety of operating conditions [16]. Table 2.4 shows the range of operating conditions of the data used in this study. The typical experimental parameters measured during testing included: fuel type, engine speed, intake pressure, spark timing, fuel and air flow, exhaust emissions, and a continuous pressure measurement. From these measured quantities, engine performance and operating conditions were calculated: equivalence ratio based on fuel and air flow measurements, equivalence ratio based on exhaust emissions, indicated mean effective pressure (IMEP), and peak pressure and location (by crankangle). By simulating the operating and geometric conditions of the engine, the cycle simulation model was compared to the engine data and optimized to give the best possible match.
2.3.3 Validation of the Cycle Simulation

With the experimental data to compare with, the cycle simulation was validated to insure that it represented real engines as closely as possible. The validation procedure consisted of optimizing the cycle simulation at a median operating condition. The validated cycle simulation was then compared with experimental data at other operating conditions to see how well it performed.

The operating condition chosen as the median value was the first entry in Table 2.4. It represents a full load and median speed operating condition. To simulate this experimental data accurately, several parameters were varied. The varied parameters consisted of quantities not measured directly during the experiment, wall temperatures (cylinder head, cylinder wall, and piston top) and inlet air temperature, and computational parameters, heat transfer constant and turbulence multiplier. After considerable investigation, the optimum values for these parameters were found. Table 2.5 shows the values chosen to optimize the model while Figure 2.8 shows the comparison of cycle simulation results with the experimental data.

To clarify the values shown in Table 2.5, a brief explanation of each parameter is required. The wall temperatures vary considerably with operating conditions and are an important input to calculating the heat losses in the engine. As a result, the wall temperatures directly affect the thermal efficiency. Average values were found that resulted in good agreement between experimental data and cycle simulation results. The inlet air temperature, the air temperature at the intake port, determines the charge density during the intake stroke and ultimately determines the mass of air and fuel inducted into the cylinder. Since this quantity was not measured, a reasonable value that gave good agreement with the base conditions was used. The heat transfer constant represents the numerical scaling of the heat losses in the engine. This was scaled, along with the previous two parameters, so that the cycle simulation results matched those of the experimental data. The final parameter, the turbulence multiplier, is an indication of how much swirl (and/or swish) is present in the engine. In the experimental set-up, there was no specifically imposed swirl or squish, so this constant was set to unity. The optimized constants gave cycle simulation results that matched experimental performance data.

2.3.4 Comparison at Other Operating Conditions

The constants found for the "base" condition were used for all the other conditions too. Although in reality the constants vary with operating conditions, it was impossible to predict how they might change and their effect was not the purpose
of this study. For the other operating conditions in Table 2.4, the cycle simulation predictions were compared with the experimental results. A normalized error was found by dividing the difference between the experiment and the cycle simulation by the experimental value. For any parameter, Θ, the equation for the normalized error would be:

\[
\% \text{ Error in } \Theta = \frac{|\Theta_{\text{exp}} - \Theta_{\text{cs}}|}{\Theta_{\text{exp}}} \times 100
\]  

(2-8)

Figure 2.9 shows the normalized error in fuel consumption, thermal efficiency, indicated mean effective pressure, and fuel and air flow rates. The cycle simulation was able to predict indicated specific fuel consumption to within five percent under most operating conditions and ten percent in all cases. For thermal efficiency, the parameter of most interest in this study, an error of about four percent was obtained for all operating conditions. The calculated values of indicated mean effective pressure stayed within an error band of about three percent with a few cases substantially higher. The error in the masses of induced air and fuel was consistent with the previous error estimates where most operating condition incurred an error of less that five percent and a few cases with substantially higher discrepancies. With a few exceptions, the general magnitude of the error was less that five percent which was sufficiently accurate for this study.

The errors discussed previously had two probable sources, numerical modelling and experimental measurement. The numerical modelling error was attributed to two main causes. As discussed previously, the fact that the calibration factors did not change as the operating conditions varied induced small differences in comparison with the experimental results. In addition, the cycle simulation volumetric efficiency model, although adequate for a thermal efficiency study, was not sophisticated enough to model real engine flows. Consequently, the cycle simulation caused some air flow inaccuracies in the comparison with engine data. Also, experimental results inherently include small uncertainties due to the nature of the measurement process. The differences between the cycle simulation calculations and the engine experiments indicated that the cycle simulation adequately simulated real engine operation as closely as possible.
2.4 Thermal Efficiency Data Base

The base condition chosen was the median value of each parameter. To investigate the effect of a single parameter on thermal efficiency, that parameter was varied around the base condition with all the other parameters held constant.

2.4.1 Base Condition

Table 2.6 shows the condition selected as the base value [15]. The chamber shape was a hemispherical shape with the spark plug offset by one-sixth of the bore. The square cylinder two-valve design was run at moderate operating conditions with no induced swirl. These geometry and operating conditions represented a condition from which the parameters of interest in this study (the geometric parameters were displaced volume, bore-to-stroke ratio, and compression ratio; the operating conditions were inlet pressure and engine speed) could be varied. An additional operating condition considered was the equivalence ratio, but only to the extent that power enrichment was used at high loads. Although bore and stroke were not accounted for explicitly, they were implicitly contained in the displaced volume and the bore-stroke ratio parameters. Table 2.7 shows the test matrix used for this study and the range of parameter variation.

2.4.2 Data Generation

To perform an extensive parametric study of the various design and operating conditions required a simple and direct process of data generation. By defining the geometric parameters, the combustion chamber and flame interaction was found (see section 2.2.2). The flame data and the operating conditions were then used as inputs to the cycle simulation. Thus, for any geometry and operating conditions, a simple and easy process was derived.

For any operating condition, the inlet and exhaust pressures were required as part of the necessary cycle simulation inputs. Since the inlet pressure was one of the parameters of interest in this study, it was specified for each test point. However, the exhaust pressure was not explicitly known for each operating condition. In fact, in engine tests, the exhaust pressure is really a result of setting the other operating conditions [5,10,23]. The relationship noted by Heywood, Figure 2.10, was used to find the exhaust pressure as a function of engine speed and load (i.e. MAP) [5].

For accurate comparisons between parameters, it was necessary to reference all data points to a particular time value. In common engine practice, the maximum brake torque (MBT) ignition timing point was used to compare engines of different
geometric and operating conditions. By using MBT ignition timing, the validity of this study was more useful and productive. Figure 2.11 shows an example of the timing curve for a particular operating condition. Timing curves show the peak value, or MBT point, as the obvious maximum value in both a local and global sense. Thus, for each data point, the MBT point was found before proceeding.

2.4.3 Parameter Variation

To study the effects of inlet pressure on thermal efficiency, the inlet pressure was varied from idle to full load conditions at each engine speed. At the higher loads, power enrichment was used to simulate the operation of a real engine. The power enrichment schedule was developed by Nitschke [1] and was based on production engine enrichment schedules. Figure 2.12 shows a typical curve shape for variation of inlet pressure at a particular engine speed. The results show that thermal efficiency slowly increases up to the point where power enrichment sets in and then decreases quickly as the higher equivalence ratio richens the mixture.

The effect of compression ratio was the most important factor in determining the thermal efficiency. Figure 2.13 shows the importance of compression ratio on thermal efficiency. As compression ratio increases, the thermal efficiency increases at a rate less than linear. The compression ratio introduced a non-linear relationship that was an important factor in determining thermal efficiency.

The effect of displaced volume on thermal efficiency is shown in Figure 2.14. Displaced volume had modest effect on thermal efficiency.

Figure 2.15 shows the effect of bore-stroke ratio on thermal efficiency at a fixed displaced volume. Bore-stroke ratio also had a moderate effect on thermal efficiency.
Chapter 3

Thermal Efficiency Correlation

The results generated by the cycle simulation (see Chapter 2) served as a data base for developing the thermal efficiency model. For each of the parameters in this study (inlet manifold pressure, engine speed, compression ratio, bore-stroke ratio, and displaced volume), a model was developed that described the individual effects of that parameter on thermal efficiency. In the remainder of this study, these individual models will be referred to as "sub-models". In this chapter, each of the sub-models will be discussed separately and then integrated together.

3.1 The Sub-Models

The development of each sub-model was a combination of theory and cycle simulation data. Theoretical expressions relating the effect of each parameter on thermal efficiency were derived from previous studies. The data from the cycle simulation was used to evaluate any constants in the theoretical relationships. A sub-model was developed for each of the following geometric parameters or operating conditions: compression ratio, power enrichment, manifold absolute pressure, engine speed, bore-stroke ratio, and displaced volume.

3.1.1 Compression Ratio

The compression ratio is the primary determinant of thermal efficiency in the ideal engine cycle. Assuming the cylinder gases remain as ideal gases, the relationship between compression ratio and thermal efficiency in the ideal cycle [5,18,19] is:

\[ \eta_{id} = \left[ 1 - \frac{1}{r_c^{\gamma - 1}} \right] \times 100 \]  \hspace{1cm} (3-1)

This relationship for the ideal cycle represents the best thermal efficiency an engine could obtain. In a real engine, and in the cycle simulation, the deviation from the ideal cycle for compression ratio variations was substantial. The biggest deviation from the ideal cycle was due to the presence of heat energy leaving the cylinder gases. To accurately model the effects of compression ratio on thermal efficiency, the ideal cycle prediction was corrected for losses due to heat transfer.
The effects of design parameters on thermal efficiency were studied extensively by Muranaka, Ishida, Takagi, and Nakagawa [21]. Their results showed a linear relationship between thermal efficiency and heat losses. In addition, heat losses were shown to be linearly related to surface-to-volume ratio. The combination of these two relationships produces the result that thermal efficiency is linearly related to compression ratio.

\[ \eta_{cr} \propto Q \propto S/V \propto r_c \]  \hspace{1cm} (3-2)

For compression ratio variations, heat transfer was the major correction to the ideal thermal efficiency. As a result, the thermal efficiency relation for compression ratio variation included an ideal thermal efficiency term minus a term due to heat losses. The functional form of the compression ratio thermal efficiency sub-model was:

\[ \eta_{cr} = c_1 \eta_{id} + c_2 r_c \]  \hspace{1cm} (3-3)

The data produced by the cycle simulation was used to determine the constants in this equation. Using the method of least squares, the final form of the compression ratio sub-model was:

\[ \eta_{cr} = 0.729 \eta_{id} - 0.226 r_c \]  \hspace{1cm} (3-4)

Figure 3.1 shows the comparison between the cycle simulation data and the sub-model for various compression ratios.

3.1.2 Power Enrichment

Power enrichment was accomplished by increasing the equivalence ratio at high load (manifold pressure). After surveying several production engines, power enrichment was generally found to begin at 85 kPa manifold absolute pressure (MAP). For manifold pressures less than this value, the equivalence ratio was set at stoichiometric for optimum performance of the three-way catalyst. For power enrichment, the schedule developed by Nitschke [1] was used. The power enrichment schedule was:
\[ \phi = 1 \quad \text{for MAP} \leq 85 \text{ kPa} \]
\[ \phi = 1 + [0.0419(MAP - 85)]^2 \quad \text{for MAP} > 85 \text{ kPa} \]  
(3-5)

The effect of equivalence ratio on thermal efficiency was investigated by thinking of thermal efficiency as the product of two efficiencies [2].

\[ \eta_t = \eta_c \eta_{th} \]  
(3-6)

Combustion efficiency, \( \eta_c \), is defined as the fraction of fuel energy supplied which is released in the combustion process. In practice, this quantity represents the amount of incomplete combustion products that are contained in the exhaust gas. Under lean operating conditions, the amount of incomplete combustion products is small and the combustion efficiency remains near one hundred percent. Under rich operating conditions, this quantity decreases substantially due to the lack of sufficient oxygen to complete combustion. The thermodynamic cycle efficiency, \( \eta_{th} \), is defined as the actual work per cycle divided by the amount of fuel chemical energy released during the combustion process. The thermodynamic cycle efficiency remains about constant as equivalence ratio richens. However, as Figure 3.7 shows, the combustion efficiency decreases rapidly as the equivalence ratio increases past stoichiometric. The slope of this decrease, according to Heywood [5], was proportional to the inverse of the equivalence ratio:

\[ \eta_c \propto \frac{1}{\phi} \quad \text{for} \ \phi > 1.0 \]  
(3-7)

This explains the sharp decrease in the thermal efficiency as the equivalence ratio increases past stoichiometric.

The functional form of the power enrichment thermal efficiency sub-model was predicted based on Equation 3-7:

\[ \eta_\phi = c_1 + c_2 \frac{1}{\phi} \]  
(3-8)
The constants in the above relationship were evaluated using the power enriched cycle simulation data. The final form of the power enrichment thermal efficiency sub-model was found using the method of least squares:

\[
\eta_\phi = 2.230 + 35.166 \frac{1}{\phi}
\]  \hspace{1cm} (3-9)

Figure 3.8 shows the relationship between the model and the cycle simulation results.

3.1.3 Manifold Absolute Pressure

The intake manifold absolute pressure (MAP) is a primary component in determining the cylinder pressures and temperatures throughout the engine cycle. The effect of manifold absolute pressure on thermal efficiency was developed assuming the results of Muranaka, Ishida, Takagi, and Nakagawa [19] held for variation of manifold pressure. To review, their results indicated that thermal efficiency was linearly related to heat losses. Therefore, the effect of manifold pressure on heat loss could be linearly related to thermal efficiency.

The relationship between heat transfer coefficient, engine geometry, and operating conditions was found in studies done by Woschni [17]. For his particular design and operating conditions, the heat transfer coefficient relationship was:

\[
h = B^{-0.3} p^{0.8} T^{-0.53} W^{0.8}
\]  \hspace{1cm} (3-10)

This equation found the heat transfer coefficient \( h \) as a function of bore \( b \), pressure \( p \), temperature \( T \), and mean gas velocity \( W \). The exponents in Equation 3-10 will change depending on the operating conditions and geometry. A more useful form of Equation 3-10 was developed by applying the heat transfer coefficient to the convective heat transfer equation. In functional form, the convective heat transfer equation was:

\[
Q = B^a p^b T^c W^d \Delta T A \frac{1}{N}
\]  \hspace{1cm} (3-11)
where $A$ is the heat transfer area, $\Delta T$ is the temperature difference between the cylinder gas and the cylinder walls, and $1/N$ represents the characteristic time for heat transfer. In our calculations, the temperature difference did not vary significantly with operating conditions or geometric configuration and was assumed constant for this study. The relationship between heat loss, engine geometry, and operating conditions was defined by Equation 3-11.

For variation of manifold pressure, the cylinder geometry (bore-stroke ratio, displaced volume, and compression ratio) and the engine speed were held constant. With these parameters constant, Equation 3-11 was simplified to include the effect of manifold pressure only:

$$Q_{MAP} \sim p^b T^c$$

The ideal gas law was used to relate temperature and pressure. Assuming the heat transfer takes place at a fixed volume, the ideal gas law showed that temperature was proportional to pressure and cylinder mass:

$$T \propto \frac{p}{m}$$

The cylinder mass in Equation 3-13 can be found from complicated relationships involving compressible flow phenomena through the intake valves. Assuming stagnation conditions, valve geometry, and discharge coefficient remain constant for variations in manifold absolute pressure, the total cylinder mass was approximated by:

$$m \sim p^\Theta$$

Substituting Equation 3-14 into the ideal gas relation, Equation 3-13, resulted in the relationship between pressure and temperature:

$$T \sim p^{1-\Theta}$$
Substituting Equation 3-15 into Equation 3-12 shows the effect of manifold pressure on heat transfer:

\[ Q_{\text{map}} \sim p^b p^{1-\theta} \sim p^\zeta \]  \hspace{1cm} (3-16)

where \( \zeta \) represents an exponential constant. The linear relationship between thermal efficiency and heat loss was used to generate the functional form of the manifold absolute pressure thermal efficiency sub-model:

\[ \eta_{\text{map}} = c_1 + c_2 p^\zeta \]  \hspace{1cm} (3-17)

Heat transfer data form the cycle simulation was used to find the exponential constant in Equation 3-16. Figure 3.4 shows a logarithmic plot of heat transfer as a function of manifold pressure at several different speeds. The exponent, \( \zeta \), was found from the slope of these curves. Using the method of least squares, the value of the constant exponent was found to be -0.258. With the exponent established, the remaining constants in Equation 3-17 were found from the thermal efficiency data generated by the cycle simulation. The final form of the manifold absolute pressure thermal efficiency sub-model was determined using the method of least squares:

\[ \eta_{\text{map}} = 42.1 - 4.36 (\text{MAP}/101.3)^{-0.258} \]  \hspace{1cm} (3-18)

Figure 3.5 shows the comparison between this model and the cycle simulation.

3.1.4 Engine Speed

The engine speed determines the relative time scale of the engine cycle. The effect of engine speed on thermal efficiency was found by assuming thermal efficiency was linearly related to heat losses (see previous section). For variations of engine speed, the cylinder geometry and manifold pressure were held constant. Applying these conditions to Equation 3-11, the heat transfer due to variation of engine speed was:
\[ Q_N \sim W^d \frac{1}{N} \] (3-19)

Assuming the local mean gas velocity term was directly proportional to piston speed [22] and using the condition that geometry remained constant, the heat transfer due to engine speed variation was further simplified:

\[ Q_N \sim N^\xi \] (3-20)

From the assumed linear relationship between heat transfer and thermal efficiency, the functional form of the engine speed thermal efficiency sub-model was:

\[ \eta_N = c_1 + c_2 N^\xi \] (3-21)

The cycle simulation data was used to evaluate the constants in Equations 3-20 and 3-21. Cycle simulation heat transfer data was used to find the constant exponent, \( \xi \). Figure 3.6 shows a logarithmic plot of the heat transfer data for variations of engine speed at several loads. The slope of these curves represents the value of the exponent. Using the method of least square, the value of the constant exponent was -0.088. The remaining constants were found from the cycle simulation thermal efficiency data at various engine speeds. Using the method of least squares, the final form of the engine speed thermal efficiency sub-model was:

\[ \eta_N = 44.6 - 15.1 N^{-0.088} \] (3-22)

Figure 3.7 shows the comparison between the model results and the cycle simulation data.

3.1.5 Bore-Stroke Ratio

Bore-stroke ratio and displaced volume are key parameters in determining the effect of geometry on thermal efficiency. The effects of bore-stroke ratio on thermal
efficiency was found assuming a linear relationship between heat loss and thermal efficiency (see Section 3.1.3). For variations of bore-stroke ratio, the operating conditions and other geometry parameters were held constant. The total heat transfer as a function of bore-stroke ratio was found by applying these conditions to Equation 3-11:

\[ Q_{B/L} \sim B^a \ A \ W^d \]  \hspace{1cm} (3 - 23)

The area for heat transfer was assumed to be proportional to piston area and the mean gas velocity was assumed to be proportional to piston speed. These assumptions were related to bore and bore-stroke ratio by:

\[ A \sim \frac{\pi}{4} \ B^2 \]  \hspace{1cm} (3 - 24)

\[ W \sim \frac{S}{P} \sim 2LN \sim L \sim \frac{B}{B/L} \]  \hspace{1cm} (3 - 25)

Substituting Equations 3-24 and 3-25 into the Equation 3-23, heat transfer as a function of bore-stroke ratio and bore was found to be:

\[ Q_{B/L} \sim B^a \left( \frac{B}{B/L} \right) \left( \frac{\pi}{4} \ B^2 \right) \sim B^{2a +d} \left( \frac{1}{B/L} \right)^d \]  \hspace{1cm} (3 - 26)

To relate Equation 3-26 to bore-stroke ratio only, the definition of displaced volume was used:

\[ V_d = \frac{\pi}{4} \ \frac{B^3}{B/L} \rightarrow B = \left[ V_d \frac{1}{(B/L) \ \frac{4}{\pi}} \right]^{1/3} \]  \hspace{1cm} (3 - 27)

Since displaced volume remained constant, a relationship between bore and bore-stroke ratio was found,
\[ B \sim (B/L)^{1/3} \]  \hspace{1cm} (3-28)

Substituting into the heat transfer equation, Equation 3-26, the heat transfer as a function of bore-stroke ratio was found:

\[ Q_{B/L} \sim (B/L)^{\xi} \]  \hspace{1cm} (3-29)

Since thermal efficiency and heat transfer are linearly related, the functional form of the bore-stroke ratio thermal efficiency sub-model was:

\[ \eta_{B/L} = c_1 + c_2 (B/L)^{\xi} \]  \hspace{1cm} (3-30)

Like the previous sub-models, the constant exponent, \( \xi \), was found from the slope of the logarithmic heat loss plot for bore-to-stroke ratio variation. Applying the method of least squares, the value of the constant exponent was found to be \(-0.020\). The remaining constants in Equation 3-30 were found from thermal efficiency cycle simulation data. The final form the the bore-to-stroke ratio thermal efficiency sub-model was determined using the method of least squares:

\[ \eta_{B/L} = -108.6 + 145.7(B/L)^{-0.020} \]  \hspace{1cm} (3-31)

Figure 3.18 shows the comparison between the bore-stroke ratio thermal efficiency sub-model and the cycle simulation data.

3.1.6 Displaced Volume

The displaced volume indicates the size of the engine cylinder. Along with the bore-stroke ratio (see previous section), the displaced volume determines the geometric relationship with thermal efficiency. To determine the effects of displaced volume on thermal efficiency, a linear relationship between heat losses and thermal efficiency was assumed (see Section 3.1.3). For variation of displaced volume, the operating conditions
and other geometric parameters were held constant. Similar to the previous section, Equation 3-11 was simplified to include geometry variables only:

\[ Q_{Vd} \sim B^a W^b A \]  \hspace{1cm} (3-32)

Making the same assumptions (Equations 3-24 and 3-25) about the heat transfer area and the mean gas velocity as the bore-stroke ratio thermal efficiency sub-model, the heat transfer due to changes in displaced volume was further simplified to include bore and bore-stroke only:

\[ Q \sim B^a \left( \frac{B}{B/L} \right)^b \left( \frac{\pi}{4} B^2 \right) \]  \hspace{1cm} (3-33)

For variation of displaced volume, the bore-stroke ratio remained constant and the relationship between displaced volume and bore (Equation 3-27) was still valid. Substituting Equation 3-27 into Equation 3-27, the heat transfer as a function of displaced volume was found to be:

\[ Q_{Vd} \sim B^{3ab} \sim V_d^{ab} \sim V_d^b \]  \hspace{1cm} (3-34)

Using the linear relationship between heat loss and thermal efficiency, the displaced volume thermal efficiency sub-model in functional form was:

\[ n_{Vd} = c_1 + c_2 V_d^b \]  \hspace{1cm} (3-35)

The constant exponent was found from the logarithmic relationship between cycle simulation heat loss and displaced volume. A statistical analysis using the method of least squares produced a value of -0.083 for the constant exponent when displaced volume was varied. Cycle simulation thermal efficiency results were then used to find the remaining constant in Equation 3-35. The final form of the displaced volume thermal efficiency sub-model was derived using the method of least squares:
\[ \eta_{Vd} = 45.7 - 13.8 V_d^{-0.083} \quad (3-36) \]

Figure 3.9 shows the comparison between the displaced volume thermal efficiency sub-model and the cycle simulation data.

3.3 Total Thermal Efficiency Model

The parametric sub-models developed in the previous section were integrated together to give a total thermal efficiency model. The resulting total thermal efficiency model allowed general changes in operating conditions as well as geometric parameters to be investigated quickly and easily.

The integration of the six sub-models into one total thermal efficiency model was based on a superposition principle. Since each of the sub-models was developed independent of the other parameters, the superposition principle should hold. Applying the superposition principle, the form of the total engine model was:

\[ \eta_{t_i} = C + C_1 \eta_{CR} + C_2 \eta_{MAP} + C_3 \eta_N + C_4 \eta_{B/L} + C_5 \eta_{Vd} + C_6 \eta_\phi \quad (3-37) \]

If the sub-models were truly independent of each other, the sub-model constants (C, C_1, C_2,...) would be unity. Because there exists some inter-dependence between the sub-models, it was concluded that the sub-model constants should be of order unity if the sub-models were developed correctly.

To evaluate the constants in the total thermal efficiency model, the complete data set generated by the cycle simulation was used in a statistical analysis to find the proper value of the constants. The resulting total thermal efficiency model constants were found using the method of least squares. The total thermal efficiency model was:

\[ \eta_{t_i} = -157.5 + 0.962 \eta_{CR} + 0.854 \eta_{MAP} + 0.647 \eta_N + 0.647 \eta_{B/L} + 1.037 \eta_{Vd} + 1.016 \eta_\phi \quad (3-41) \]

The resulting coefficients were all of order unity which indicates that the sub-model thermal efficiencies were close to correct.
3.4 Total Thermal Efficiency Results

The results of the thermal efficiency model were compared with the cycle simulation values at a variety of geometrics and operating conditions. Figure 3.10 shows the comparison of the total model with cycle simulation data for various loads ranging from idle conditions to full load at three different engine speeds. In all cases, the thermal efficiency model follows the data accurately throughout the speed and load range. Figure 3.11 shows the comparison between model and data at a wide range of bore-stroke ratios. Again, the engine model indicates good accuracy compared with the data. Figure 3.12 through 3.15 shows the model comparison at various compression ratios, strokes, bores, and displaced volumes. In all these cases, the total thermal efficiency model closely reproduces the cycle simulation results.
Chapter 4

Model Constraints

In real engines, there are several constraints that limit the operating range of the engine. The causes of these constraints range from structural effects to chemical reaction effects. Of the many constraints with actual engines, two were appropriate for study and implementation in this model. These were the maximum rated speed and the knock-limited thermal efficiency which are important constraints on engine performance.

4.1 Maximum Rated Speed

The maximum rated speed was defined as the speed where maximum brake power occurs. It would be inappropriate to run an engine at a speed much higher than the maximum rated speed since the power decreases from that point on and the engine would be producing less power at a higher cost. Also, exceeding this speed by an extreme amount for an extended period of time would result in mechanical failure of the engine parts.

To quantify more accurately the maximum rated piston speed constraint, several different engine geometries were tested using the Nitschke engine model [1]. Figure 4.1 shows the brake power as a function of engine speed for several bore-stroke ratios. For a constant displaced volume, the maximum rated speed increases as the bore-stroke ratio decreases. Figure 4.2 shows the composite results for bore-stroke ratio variation. As the bore-stroke ratio increases, the maximum rated speed increases linearly. This linear relationship results in a constant maximum rated mean piston speed regardless of the geometry of the engine. In Figure 4.3, the value of the constant maximum mean piston speed is approximately 13.5 meters per second. In actual engine design, this value is usually exceeded by as much as ten percent, so a more meaningful maximum mean piston speed constraint would be one of 15 meters per second.

\[
\bar{S}_p \leq 15 \text{ m/s} \quad (4-1)
\]

To implement the maximum rated engine speed constraint into the engine model was quite simple. In the spreadsheet version of the model, the mean piston speed was
calculated. If it exceeded the maximum rated piston speed, that value was flagged by a warning message calling attention to the user.

4.2 Knock Limitations

Engine performance at wide-open-throttle is usually limited by knock. Knock can vary in intensity from mild to severe. Where mild knock will do little structural damage to the engine, severe knock can cause structural damage to the engine. Knock limits the engine compression ratio which is a primary determinant of performance and economy. The causes and conditions of knock are not clearly understood and cannot be easily quantified. In most engine applications, the knock problem is handled by retarding the ignition timing enough to avoid knock while still striving for the maximum brake torque. This results in an spark ignition setting that is retarded from the non-knocking MBT timing and represents the knock-limited MBT timing. Accompanying the spark retard from MBT is a loss of thermal efficiency due to retarding the spark from the optimum setting.

4.2.1 Spark Retard Data

High manifold pressure spark retard data from several production engines is shown in Figure 4.4. The data in this graph was generated by finding the non-knocking MBT timing point at each speed using a high octane fuel such as indolene clear. The amount of spark retard was then found by using a gasoline type fuel (ie. Amoco 91) and noting the knock-limited MBT timing point. The amount of spark retard required was the difference between the non-knocking MBT value and the knocking MBT value. Following the procedure described above, the spark retard results were shown to be highly variable and engine dependant as shown in the figure.

4.2.2 Spark Retard Methodology

The spark retard model developed by Nitschke was compared to the data presented in the earlier section in Figure 4.5. Because of the transient nature of knock, the Nitschke results sometimes compared well with the data. However, in general, the slope on the Nitschke spark retard model does not show the same trends as the data. Thus, a new spark retard model was developed to account more explicitly for the knock limited spark retard.

The development of a spark retard model to correct knock problems is discussed in this section. The spark retard model was based on studies by other researchers
describing the effect of knock. According to recent research, the parameters of most significance on knock were compression ratio, bore, engine speed, and load [23].

The effect of compression ratio on the octane requirement of an engine was summarized by Blackmore and Thomas [27]. As Figure 4.6 shows, every unit increase in compression ratio requires an additional five fuel octane number to avoid knock in passenger cars on the road today. From this relationship, the octane required for any compression ratio can be approximated by:

\[ ON_{cr} = 89 + (r_c - 9.0) 5.0 \]  

(4-2)

The effects of bore and speed on spark retard were found by Peters [25]. A useful discovery by Peters was the relationship between speed, bore, and octane requirement:

\[ OR_{req \cdot d} \sim B^{0.29} N^{-0.076} \]  

(4-3)

The manifold absolute pressure and octane requirement were combined to quantify the required spark timings in a study by Renault [26]. In this relationship, the inlet pressure and octane requirement determined the required spark timing, as shown in Figure 4.8. In equation form, the graph was represented by,

\[ \theta_{sp} = 0.9251(OR) - 0.579\text{(MAP)} \]  

(4-4)

The total amount of spark retard was found using the definition of spark retard (knocking MBT spark timing minus non-knocking spark timing):

\[ SP = \theta_{\text{knocking}} - \theta_{\text{non-knocking}} \]  

(4-5)

In experimental spark retard tests, the non-knocking spark timing point was found using a high quality fuel. For most engine configurations, a fuel with an octane rating of 98 was adequate to prevent knock under the most severe conditions. The most severe condition for knock occurs at high load since cylinder pressure is
roughly proportional to inlet pressure. Therefore, the non-knocking octane number at the worst knock condition was used as a reference:

$$\text{ON}_{\text{non-knocking}} = 98 \text{ at MAP} = 95 \text{ kPA} \quad (4-6)$$

Substituting Equations 4-6 into Equation 4-5, results in a relationship between spark retard, octane number, and manifold absolute pressure:

$$\text{SR} = 0.9251[98 - \text{ON}] - 0.579[95 - \text{MAP}] \quad (4-7)$$

Simplifying, the required octane number at any manifold pressure and spark retard can be found by the equation:

$$\text{ON}_{\text{req'd}} = \frac{35.65 + 0.579\text{MAP} - \text{SR}}{0.9251} \quad (4-8)$$

Combining the effects of compression ratio, bore, engine speed, manifold absolute pressure, and spark retard, a total octane number relationship was found from scaled versions of Equations 4-2, 4-3, and 4-8:

$$\text{ON} = [1 + \frac{(r_c - 9)}{94}] [\text{constant B}^{0.29}N^{-0.076}] \\
\times \left[ \frac{35.65 + 0.579 \text{MAP} - \text{SR}}{0.9251 \text{ OR}} \right] \quad (4-9)$$

The experimental data in Figure 4.4 gives octane requirement, compression ratio, bore, engine speed, load, and spark retard for a variety of engine configurations and operating conditions. The constant in Equation 4-9 was found using this data in a method of least squares statistical analysis. Since the relationships in equation 4-9 were scaled, this constant should approximate the average octane rating required. From the statistics, the value of the constant was 89.77.

Substituting the constant into equation 4-9, a relationship between spark retard, octane number, compression ratio, bore, engine speed, and load was found:
$$SR = \frac{-0.925 \cdot ON^2}{\left[1 + \frac{(r_c - 9) \times 5}{94}\right][89.77 B^{0.29} N^{-0.076}] + 35.65 + 0.579 \text{ MAP}} \quad (4-10)$$

The spark retard calculated here includes the effect of compression ratio, bore, engine speed, and inlet pressure. If the calculated spark retard is less than zero, no spark retard is imposed on the system.

4.2.3 Spark Retard Results

The results of the spark retard model formulated in Equation 4-14 was found to predict spark retard as accurately as possible. Figure 4.8 shows the comparison between measured spark retard and predicted spark retard. The results of the spark retard model agree with intuitive predictions based on the fundamental physical relationships because increasing speed lessens the amount of spark retard (due to the decreasing time available for knocking) and decreasing load decreases the amount of spark retard (due to lower cylinder pressure).

The effect of knock-limited spark advance primarily affects engine performance through thermal efficiency. As more spark retard is required, the thermal efficiency decreases at an increasing rate. Using the cycle simulation thermal efficiency data, the effect of spark retard on thermal efficiency was quantified. Figure 4.10 shows the reduction of thermal efficiency as a function of spark retard for a variety of geometric configurations. These results were approximated by a single curve using the method of least squares. The equation representing this curve was:

$$\Delta \eta_t = -0.151(SR) - 0.0154(SR)^2 \quad (4-11)$$

To implement this model into the engine model, the required spark retard was found for the given operating condition, geometric configuration, and fuel type using Equation 4-10. The subsequent loss of thermal efficiency was found using Equation 4-11.
Chapter 5

Volumetric Efficiency

To integrate the thermal efficiency model developed in this study with the volumetric and mechanical efficiency models developed by Nitschke [1], the Nitschke models were compared and validated against experimental data. The mechanical efficiency model was thought to be adequate for this study with modifications and improvement being made in the future [32]. The methodology behind the volumetric efficiency accurately describes the flow losses in an engine in the range of the model. To expand the useful range of the volumetric efficiency model and the model as a whole, the volumetric efficiency was improved.

5.1 Volumetric Efficiency Methodology and Results

The Nitschke volumetric efficiency model predicts volumetric efficiency in terms of important engine design and operating variables such as engine speed, load, bore, stroke, valve timing, and valve lift. In the development of this model, the volumetric efficiency was broken down into several quasi-steady flow processes and valve timing effects. Induction flow friction, valve flow friction, flow choking, and heat transfer were modelled as quasi-steady flow processes while the valve timing effects were calculated using the correlations developed by Young [27].

When compared with the actual engine data, the Nitschke volumetric efficiency model compared favorably at high loads. The individuality of each induction system makes exact predictions quite difficult. Nevertheless, the Nitschke model does a relatively good job of estimating the volumetric efficiency at high loads in the absence of intake system tuning. However, the Nitschke model substantially underpredicts the volumetric efficiency at lower loads. In some cases, the model even predicts negative volumetric efficiency as shown in Figure 5.1. To use the model over the full load range, the Nitschke volumetric efficiency model was modified to give reasonable results at low loads.

5.2 Modifications

The modification of the volumetric efficiency model was based on experimental data from engines. Since the Nitschke model predicted wide-open-throttle volumetric efficiency well, the calculation procedure at wide-open-throttle was left unchanged. At lower loads, the volumetric efficiency was scaled relative to the wide-open-throttle
volumetric efficiency. As Figure 5.2 shows, all data available for this study falls in a wide band with virtually the same slope regardless of the engine speed and induction system design. Using a method of least squares statistical analysis, a scaling law dependence between manifold pressure and volumetric efficiency was found.

\[
\frac{\eta_v}{\eta_{v,wot}} = (\frac{\text{MAP}}{\text{MAP}_{wot}})^{1.25}
\]  

(5-1)

Thus, the volumetric efficiency at any load could be found relative to the wide-open-throttle volumetric efficiency using the relation described above. This method retains the accuracy of the Nitschke model at full load while avoiding the gross inaccuracies at low loads.

5.3 Model Results

The results of the modified volumetric efficiency model were compared against the data available for this study. The results of this comparison are shown in Figure 5.3. As shown, the modified volumetric efficiency model retained accuracy at high loads while obtaining reasonable results at lower loads.
Chapter 6

Performance and Fuel Consumption

The preceding chapters developed the new thermal efficiency model and the revised volumetric efficiency model. The integration of these models into the Nitschke model [1,28] provides the ability to predict performance and fuel consumption for any operating conditions and geometry. The remaining sections in this chapter will cover the formulation of performance and fuel consumption in the model, the comparison with measured engine data, the comparison with other studies, a parametric study, and a geometric optimization study. Comparison with measured engine data were done on an indicated basis, while the parametric and geometric optimization study were done on a brake basis.

6.1 Formulation of Performance and Fuel Consumption

This study, along with the work by Nitschke, predicts indicated thermal efficiency and volumetric efficiency. These efficiencies were then related to performance and fuel consumption. The common performance and fuel consumption relationships were established in texts by Taylor [18] and Obert [19], while the introduction of efficiency into these relationships was noted by Heywood [5] and Matthews [29]. The characteristics of interest in this study were power, torque, mean effective pressure (MEP), and specific fuel consumption (SFC).

The mean effective pressure is a useful parameter which indicates how well an engine design uses its displaced volume. Physically, this quantity represents the constant pressure which, if exerted on the piston throughout the power stroke, would yield work equal to the work of the cycle [18,19]. Mean effective pressure is related to thermal and volumetric efficiency by [5,29]:

\[ MEP = \eta_t \eta_v Q_{HV} \rho_{\text{ai}} \frac{(F/A)}{V} \]  \hspace{1cm} (6-1)

The specific fuel consumption (SFC) is a measure of how efficiently an engine is converting fuel into work [5,18,19]. In practice, this parameter is used instead of thermal efficiency since all quantities are measured in standard and physical units. Specific fuel consumption is related to thermal efficiency by:
\[ SFC = \frac{1}{\eta t Q_{HV}} \quad (6.2) \]

Power and torque are related to thermal and volumetric efficiency by [29]:

\[ p = \frac{\eta_t \eta_v Q_{HV} d \rho a_i (F/A) V_d n R}{n_R} \quad (6.3) \]
\[ T = \frac{\eta_t \eta_v Q_{HV} d \rho a_i (F/A) V_d}{2\pi n_R} \quad (6.4) \]

The mean effective pressure, specific fuel consumption, power, and torque equations are valid on both a brake and indicated basis. Indicated power and brake power are related through the mechanical efficiency:

\[ P_b = \eta_m P_i \quad (6.5) \]

Thus, to evaluate the parameters in equations 6.1 through 6.5 on a brake basis, the relationships were simply multiplied by the mechanical efficiency.

6.2 Comparison with Engine Data

With the ability to predict performance and fuel consumption in the engine model, the engine model was compared to experimental engine data. Table 6.1 shows the general specifications for fuel consumption calculations were made from the measured test quantities. Engine performance and fuel consumption were then compared with the engine model characteristics on an indicated basis. The comparison was made on an indicated basis for two reasons: (1) the models developed in this study (thermal and volumetric efficiency) could be evaluated separately from the mechanical efficiency model and (2) the mechanical efficiency was being revised to alleviate some shortcomings in its structure [32].
6.2.1 Calculations from Engine Data

Indicated performance and fuel consumption were calculated from the quantities measured during testing. Each engine test consisted of varying speed and load throughout the normal driving ranges. At each speed and load point, several quantities were measured: brake torque, fuel rate, manifold pressures, spark advance, motored torque, exhaust gas recirculation (EGR), and exhaust emissions. Exhaust emissions were used to calculate the fuel-air ratio using an oxygen chemical balance. Wide-open-throttle (WOT) torque data was adjusted to atmospheric conditions [5] while throttled torque was not. Frictional torque was measured as wide-open-throttle motoring friction at each operating condition.

Brake power was calculated from the measured torque. From the brake power, the brake specific fuel consumption (BSFC) was determined. Indicated torque was found from the sum of the frictional torque and the brake torque data:

\[ T_i = T_b + T_f \]  \hspace{1cm} (6-6)

Indicated power, indicated mean effective pressure, and mechanical efficiency were calculated from the indicated torque. Indicated specific fuel consumption was calculated from the product of brake specific fuel consumption and mechanical efficiency.

6.2.2 Correction for EGR and Pumping

To compare the engine model predictions with test data, some of the engine data had to be corrected for the effects of exhaust gas recirculation (EGR) which was not included in the model predictions. In addition, the engine model calculations were expanded to include net results that matched the corresponding test engine quantities.

As shown in Table 6-1, two of the engines used for comparison in this study included exhaust gas recirculation (EGR) in their test results. For wide-open-throttle comparisons, exhaust gas recirculation was not a problem. At lower loads, exhaust gas recirculation became quite substantial (up to 20%). To correct for the influence of exhaust gas recirculation, the partial pressure of the incoming charge was used. The incoming charge consists primarily of recirculated exhaust gas, air, and fuel vapor. In mathematical form, the partial pressure relationship was:

\[ P_{MAP, measured} = P_{air} + P_{EGR} + P_{fuel} \]  \hspace{1cm} (6-7)
In the engine tests, exhaust gas recirculation was measured as a percentage of the total incoming charge. Therefore, the adjusted manifold pressure without exhaust gas recirculation was found by subtracting this percentage from the total incoming charge:

\[
P_{\text{MAP, adjusted}} = P_{\text{MAP, measured}} \times (1 - \text{EGR})
\] (6-8)

This adjustment for exhaust gas recirculation is consistent with the engine test procedure where inlet pressure is increased as exhaust gas recirculation increases with all other conditions held constant.

As discussed previously, the experimental wide-open-throttle motoring torque was used to describe the frictional losses at each engine speed. Thus, the indicated calculations based on this assumption were net quantities - indicated quantities that included all four strokes of the engine cycle. However, for the engine model, all indicated quantities were calculated on a gross basis - indicated quantities that included only the compression and the expansion strokes of the engine cycle. To overcome the difficulty in comparing net test values with gross model predictions, the engine model predictions were converted to net predictions by subtracting the pumping losses calculated in the Nitschke mechanical efficiency model:

\[
\text{IMEP}_{\text{net}} = \text{IMEP}_{\text{gross}} - \text{PMEP}
\] (6-9)

The appropriate changes were also made in the other performance and fuel consumption calculations to convert them to a net indicated basis. These changes allowed the engine model predictions to be accurately compared with the engine test results.

To increase the versatility of the engine model, several user defined inputs were created. Fuel heating value and octane number were included so that valid comparisons could be made with engine data of differing fuel types. In addition, atmospheric air temperature and pressure were added as inputs so that variations in air density could be included.

The culmination of the corrections discussed above produced engine model predictions and engine test results that represented quantities calculated in a similar manner. Since the quantities have the same meaning, valid performance and fuel consumption comparisons could be made.
6.2.3 Performance and Fuel Consumption Comparison

Comparisons between the engine model predictions and the test data results was performed to determine how well the model could predict real engine performance and fuel consumption. Performance was primarily a concern at wide-open-throttle, but for the sake of completeness, values were calculated for two part load conditions. Fuel consumption was primarily a concern at part load, but wide-open-throttle results were also found. For each of these loads (MAP), the engine speed was varied from low to high speed.

Performance and fuel consumption comparisons were also made at constant speeds. For these comparisons, engine speed was held constant at three values (1000, 3000, and 5000 RPM) while load was varied from low load to wide-open-throttle. Because of the variety of data sources, all cases are not compared at identical conditions, but were matched as closely as the data would allow. It should be noted that the engines used in this study were run at different fuels and exhaust gas recirculation. In particular, engine A was run with no exhaust gas recirculation and high quality fuel while engines B and C were run with exhaust gas recirculation and lower quality fuel. Table 6.1 shows the general operating conditions for each of these engines.

6.2.4.1 Constant Pressure

Figure 6.1 shows the comparison between the test engine results at wide-open-throttle. Performance, fuel consumption, efficiencies, and spark retard are shown as functions of speed for each of the three engines. Both net and gross indicated predictions are shown for comparison. The model predicts performance and fuel consumption adequately. The major cause of any discrepancy between the model prediction and the test results is clearly shown to be linked to the volumetric efficiency prediction which did not model induction system tuning. The thermal efficiency models the test results adequately with some difference due to engine configuration.

Figure 6.2 and 6.3 show performance and fuel consumption at part load conditions. The main source of difference remains in the volumetric efficiency model even at low loads. Thermal efficiency predictions perform as well as in the full load case.
6.2.4.2 Constant Engine Speed

Figure 6.4 shows the comparison between engine model and test results for performance and fuel consumption at medium engine speed. The corrected exhaust gas recirculation data is shown where appropriate. In general, the engine model adequately predicts performance and fuel consumption at this speed. Most of the differences in performance and fuel consumption were caused by disagreement between the predicted and measured volumetric efficiency results. For the most part, thermal efficiency predictions closely model the engine data.

Figure 6.5 and 6.6 show the comparison between the engine model predictions and the engine test results at low and high speed, respectively. Model predictions at low speeds match the data better than the mid-range speed predictions. High speed comparisons show some substantial differences between the model predictions and the test results. The biggest difference is most likely caused by high speed tuning effects that the volumetric efficiency model could not predict.

6.2.4 Sources of Difference

The differences between the model predictions and the test data were quite small in most cases. Of the three engines, the model was able to predict the performance and fuel consumption of engine A better than engine B and C throughout the speed and load range tested. In general, the engine model predicts performance and fuel consumption for a "generic" engine. Specific engines present individual design characteristics that the engine model cannot predict.

As the preceding sections demonstrated, the common cause of many of the differences was the absence of induction system tuning in the volumetric efficiency model. In addition, other sources of differences influenced the comparison. The engine test results introduced some uncertainty through motoring friction and data evaluation. The engine model produced some inaccuracy through the fuel-air ratio and wide-open-throttle manifold pressure assumptions.

The assumption that the motoring torque of the engine represents the frictional torque loss in a firing engine is not clearly defined. Despite every attempt to keep all the operating conditions the same for the motored and firing cases, the major source of inaccuracy with this method is that the gas pressure forces on the piston and rings are considerably lower in the motored test than the firing engine. In addition, the lower cylinder wall oil temperature under motored conditions causes some inaccuracy [5].

The engine data itself was the probable cause for some difference between the model and the data. At high speeds, the data for engine C seemed to produce
inconsistent trends for thermal efficiency and fuel consumption. Due to the limited number of data sources, this inconsistency did not eliminate this engine from the comparison.

An additional difference in the engine comparison was the fuel-air ratio. In the engine test data, the fuel-air ratio varied with manifold pressure. At high inlet manifold pressure, the mixture was enriched to obtain more power. At intermediate and low manifold pressures, the mixture became leaner. The power enrichment schedule was quite variable and engine dependent with some engines having substantially more enrichment than others. In contrast, the engine model fuel-air ratio remained at stoichiometric for intermediate and low loads with power enrichment at high loads as in Equation 3-5. The difference in fuel-air ratio between the model and the data can contribute significant difference to the comparison.

The difference between wide-open-throttle manifold pressure in the model and the data was a potential source of some inaccuracy. In the engine model, wide-open-throttle was assumed to be at 95 kPa. However, for the engine data, wide-open-throttle varied from 100 kPa at low speeds to 95 kPa at high speeds. This causes some uncertainty when comparing engine data with model predictions at wide-open-throttle and low speed.

The inaccuracies discussed in this section help to explain some of the differences in magnitude between the engine model and the data. Despite these discrepancies, the engine model predicted the trends of the data well. For this reason, the engine model represents a model that can predict engine performance and fuel consumption for a parametric study.

6.3 Comparison with Other Studies

To further insure that the engine model was performing as expected, the model predictions were compared with results from several other studies. These studies were primarily concerned with the effect of geometric variables on performance and fuel consumption. The geometric conditions of the studies were simulated as closely as possible in the engine model.

Muranaka, et. al. [30] investigated the effect of compression ratio on indicated thermal efficiency. For their comparison, a disk chamber with a square bore-stroke ratio and fixed displaced volume was modelled using a cycle simulation. The geometric variables in the model were simulated to closely match the geometry used in the study except that no information about the valves was available. Figure 6.7 shows the comparison between the engine model predictions and the Muranaka results.
The trends of the engine predictions compare well with the Muranaka study, but the magnitudes are slightly different due most likely to the different combustion chamber shapes in the model and the study.

Mattavi [31] also studied the effects of compression ratio on indicated thermal efficiency. The study results were obtained using an experimental single cylinder engine with a hemispherical chamber and a domed piston. Compression ratio changes were accomplished by varying the height of the domed piston. The geometric parameters and operating conditions of the study were unknown. Model predictions were generated at a fixed geometry and constant operating conditions. Figure 6.8 shows the results of the comparison between the model and the Mattavi study. The slope of the predictions and the data appear to agree well with the magnitude of the results being quite a bit different due presumably to the unknown geometry (bore and stroke) and operating conditions (speed, load, and equivalence ratio).

Engine simulation computations were used to study the effects of bore-stroke ratio on thermal efficiency, fuel consumption, and mean effective pressure. In this study, bore, compression ratio, valve size and timing, piston speed, and inlet pressure were held constant as bore-stroke ratio varied. Using these parameters, the engine model predictions were generated. However, this study used a "double-hemi" combustion chamber and extensive exhaust gas recirculation while the model predictions were generated with no exhaust gas recirculation and a hemispherical combustion chamber. Comparison between the model predictions and study are shown in Figure 6.9. Considering the difference in combustion chamber shape and exhaust gas recirculation, the results compare well in both magnitude and shape.

In comparison with the other studies, the engine model predicted the study results accurately considering the difference in geometry and operating conditions. This supports the validity of the model.

6.4 Parametric Study

One of the primary goals of this study was to examine the effect of geometry on performance and fuel consumption. Performance and fuel consumption were studied on a brake basis. This required integration of the Nitschke mechanical efficiency model with the models developed here. The performance characteristics were evaluated at wide-open-throttle since this represents the best performance for a particular geometric configuration. Fuel consumption was studied at a highway cruise condition (MAP = 70 kPa) where efficiency is most important. The geometric parameters of interests were compression ratio, bore-stroke ratio, and displaced volume.
6.4.1 Compression Ratio

The effect of compression ratio on performance is shown in Figure 6.10. Both the knock-limited predictions and the predictions not limited by knock are shown. For the cases not limited by knock, performance increases as compression ratio increases because of the higher pressures generated by the high compression ratio. For the knock-limited cases, performance is worse at low speeds where spark retard is high and improves as engine speed increases. At high compression ratios, the amount of required spark retard increases causing the magnitude of performance to drop significantly from the predictions not limited by knock. Figure 6.11 shows the required spark retard corresponding to each of the compression ratios in Figure 6.10 and shows the increasing spark retard with higher compression ratio.

The effect of compression ratio on fuel consumption is shown in Figure 6.12. As expected, higher compression ratio engines result in lower fuel consumption since their thermal efficiency increases with compression ratio. This trend is expected since the ideal thermal efficiency (Equation 3-1) is inversely proportional to compression ratio.

6.4.2 Bore-Stroke Ratio

Figure 6.13a shows the effect of bore-stroke ratio on performance. As the bore-stroke ratio increases at a fixed displaced volume, the peak performance remains about the same magnitude, but occurs at a higher speed. The peak performance seems to occur at the same mean piston speed with small differences in magnitude regardless of the bore-stroke ratio as shown in Figure 6.13b.

The effect of bore-stroke ratio on fuel consumption is shown in Figure 6.14a. Higher bore-stroke ratios cause the fuel consumption to increase rapidly at higher compression ratios. Figure 6.14b shows fuel consumption as a function of bore-stroke ratio and mean piston speed. At low mean piston speeds, larger bore-stroke ratios produce lower fuel consumption. At high piston speeds, the opposite is true.

6.4.3 Displaced Volume

Figure 6.15a shows the performance characteristics at a fixed bore-stroke ratio as a function of displaced volume and speed. As displaced volume increases, the peak performance occurs at higher speeds. At some point, the displaced volume becomes too large and the performance is severely limited by spark retard throughout the speed range. Figure 6.15b shows the performance characteristics at a fixed bore-stroke ratio.
as a function of displaced volume and mean piston speed. At high speeds, the large
displaced volume performs better than the smaller displaced volume.

The effect of the displaced volume on fuel consumption is shown in Figure 6.16a. As displaced volume increases, the fuel consumption decreases throughout the entire engine speed range. Figure 6.16b shows the effect of mean piston speed on fuel consumption. The large displaced volume achieves lower fuel consumption than the other two smaller displaced volumes throughout the mean piston speed range.

6.5 Geometric Optimization

The optimum configuration for a particular application can be determined using the engine model. The optimum geometry was found in two ways: (1) at specified operating conditions and (2) at maximum output. To specify the operating conditions, the engine speed or mean piston speed and the inlet pressure or brake mean effective pressure (BMEP) must be specified. The maximum value for a particular geometry was found by varying the operating conditions until the maximum output (maximum BMEP and minimum BSFC) was found.

6.5.1 Specified Operating Conditions

At a specified operating condition, the best performance and lowest fuel consumption for a particular geometry were found. One geometric parameter was varied while the other two remained constant. For these comparisons, the operating conditions were set to correspond to mid-range engine speeds and wide-open-throttle for performance and highway cruise for fuel consumption. The performance was found at wide-open-throttle since it represented the best possible performance at that geometric condition. The fuel consumption was found at a constant load (BMEP) of 550 kPa to simulate highway cruise conditions. The mean piston speed was specified at a constant value of 6.8 m/s which corresponded to a mid-range engine speed for most geometries. To simulate real engine operation, knock-limited values were used.

Figure 6.17 shows the performance and fuel consumption curves for variation of compression ratio. As compression ratio rises, thermal efficiency continues to increase and fuel consumption, being inversely related to thermal efficiency, continues to decrease. As a result, lower fuel consumption was obtained at higher compression ratios. The performance shows peak performance near a compression ratio of 8. Compression ratios higher than this value become more and more knock-limited as compression ratio increases. Figure 6.18 shows the thermal efficiency, volumetric
efficiency, mechanical efficiency, and spark retard as a function of compression ratio for both performance and fuel consumption.

Performance and fuel consumption as a function of bore-stroke ratio are shown in Figure 6.19. As bore-stroke ratio increases, fuel consumption increases, for a fixed displaced volume, due to a constantly decreasing thermal efficiency. Again, the lowest fuel consumption point is not clearly defined but lower fuel consumption is obtained at lower bore-stroke ratio. As bore-stroke increases, performance peaks around a value of 0.8 and then decreases. The peak performance point is a compromise between increasing volumetric efficiency and decreasing thermal and mechanical efficiency. Figure 6.20 shows the thermal, volumetric, and mechanical efficiencies and spark retard as a function of bore-stroke ratio.

Figure 6.21 shows the performance and fuel consumption results as a function of displaced volume. The performance peaks at a value of 460 cubic centimeters per cylinder and then decreases. The peak performance is a compromise between increasing mechanical and volumetric efficiencies and decreasing thermal efficiency. The slightly increasing thermal and mechanical efficiencies at part load cause the fuel consumption to continuously decrease as displaced volume increases. Therefore, better fuel consumption is obtained at higher displaced volume. Figure 6.22 shows the mechanical, thermal, volumetric efficiency, and spark timing for peak performance and lowest fuel consumption as a function of displaced volume.

6.5.2 Maximum Performance and Minimum Fuel Consumption

The maximum performance and minimum fuel consumption were found for variations in the geometric parameters. The maximum brake mean effective pressure and minimum brake specific fuel consumption for a particular geometry were found at any speed and load condition. Maximum brake mean effective pressure occurs at wide-open-throttle while minimum brake specific fuel consumption occurs at the manifold pressure just before power enrichment begins (85 kPa MAP).

Figure 6.23 shows the maximum performance and minimum fuel consumption for variation of compression ratio. Peak performance occurs near a compression ratio of 8.0 and is a trade-off between thermal, volumetric, and mechanical efficiency. Both thermal and volumetric efficiency have a peak near the peak performance point while mechanical efficiency declines with increasing compression ratio. For fuel consumption purposes, compression ratio for minimum fuel consumption is not clearly defined. The two components of fuel consumption, mechanical and thermal efficiency, have opposite trends as compression ratio increases - thermal efficiency increases and mechanical
efficiency decreases. Near a compression ratio of 13:1, the mechanical efficiency causes the fuel consumption to increase and the minimum fuel consumption point is found. However, this value is outside the range of practical spark-ignition engine compression ratios. Therefore, higher compression ratio gives lower fuel consumption. Figure 6.24 shows the efficiencies and spark retard as a function of compression ratio.

Figure 6.25 shows the maximum performance and minimum fuel consumption configuration as bore-stroke ratio varies. The efficiencies and spark retard that accompany bore-stroke ratio variation are shown in Figure 6.26. Peak performance occurs at a small bore-stroke ratio value of 0.75. Despite the improving volumetric efficiency as bore-stroke ratio increases, the decreasing mechanical and thermal efficiencies result in decreasing power at high bore-stroke ratios. The lowest fuel consumption point is not explicitly defined because of the decreasing thermal efficiency as bore-stroke ratio increases. Again, this is a trade-off between the decreasing thermal efficiency and the slightly increasing mechanical efficiency. Therefore, lower fuel consumption is obtained at lower bore-stroke ratio.

The effect of displaced volume on maximum performance and minimum fuel consumption is shown in Figure 6.27. The trends and results of this figure can best be explained by examining the efficiencies and spark retard that constitute performance and fuel consumption. Figure 6.28 shows these parameters as a function of displaced volume. Peak performance occurs at a displaced volume of about 425 cubic centimeters per cylinder. This point is the result of a trade-off between the increasing volumetric efficiency and the decreasing thermal efficiency. The displaced volume for lowest fuel consumption continues to increase in the range tested due to a trade-off between the increasing mechanical efficiency and the slightly decreasing thermal efficiency. For the range of practical displaced volumes, better fuel consumption is obtained at larger displaced volumes.
Chapter 7

Summary and Conclusions

The objective of this study was to define more explicitly the impact of geometry on thermal efficiency. A cycle simulation was used to generate data where individual operating conditions and geometric parameters could be varied. To insure that this cycle simulation was producing correct and valid data, it was modified and validated using experimental engine data from a Ricardo engine. With the data base available for each parameter variation, individual sub-models were developed that defined each parameters individual effect on thermal efficiency. These sub-models were combined so that the thermal efficiency could be found for any combination of parameters.

Constraints were implemented into the model to simulate more accurately real engine operation. A knock constraint was developed that found the required spark retard for any particular engine configuration and operating conditions. This constraint was based on several knock studies and spark retard data from production engines. A high speed constraint was also developed that limited the piston speed to values of about ten percent over the maximum brake torque.

The thermal efficiency model and constraint models were integrated with the remaining parts of the Nitschke model. The volumetric efficiency model was revised to include low load operation so that the full range of the thermal efficiency model could be used. The total model was able to predict performance and fuel consumption for any engine configuration and operating condition.

To insure that the completed engine model was producing accurate and reliable results, it was tested against engine data and the results from other studies. Comparison with engine data required correcting the model and data for differences in exhaust gas recirculation and pumping. Results were then found for the full load and speed range on a net indicated basis. Comparison with both engine data and other geometry studies showed good results and supported the accuracy of the engine model.

Parametric and optimization studies were performed to predict the effect of variations in compression ratio, bore-stroke ratio, and displaced volume. Optimization was performed for constant operating conditions (speed and load) and maximum performance and minimum fuel consumption. With constant mean piston speed and wide-open-throttle manifold pressure, best knock-limited performance was found at:
\[ r_c = 8.5 \]

\[ B/L = 0.8 \]

\[ V_d = 460 \text{ cc/cylinder} \]

With the mean piston set to give maximum performance and wide-open-throttle manifold pressure, best knock-limited performance was found at:

\[ r_c = 8.0 \]

\[ B/L = 0.75 \]

\[ V_d = 425 \text{ cc/cylinder} \]

The lowest fuel consumption point was not found explicitly. For both constant mean piston speed and the mean piston speed giving minimum fuel consumption at a fixed load (brake mean effective pressure), lower fuel consumption was obtained with higher compression ratio, lower bore-stroke ratio, and higher displaced volume. The best design configuration was found to be a compromise between desired performance and economy.

While the conclusions presented here are valid only for this particular configuration and may vary depending on the operating conditions and cylinder geometry, the results demonstrate the usefulness of this type of model in quantifying important engine cylinder design trends.
References


Table 2.1 Values of chemical constants for Indolene [5].

\[(F/A)_s = 0.068\]

\[(A/F)_s = 14.6\]

\[M = 97.34 \text{ kg/kmol}\]

Enthalpy Polynomial Constants:

\[A_1 = -24.09 \quad A_4 = 61.26\]

\[A_2 = 249.2 \quad A_5 = 0.5276\]

\[A_3 = -193.8 \quad A_6 = -19.21\]

Table 2.2 Scaling rules for relating valve size to engine geometry in the cycle simulation [5].

\[D_{iv} = 0.5 \, B\]

\[A_{ex} = 0.75 \, A_{iv} \quad [D_{ev} = 0.43 \, B]\]

Lift = 0.25 D

Rocker Arm Ratio = 1.5

Valve Thickness = 1.5 mm

Seat Angle = 45 degrees

Note: D = diameter of valve

A = cross-sectional area of valve

B = bore

iv = intake valve

ex = exhaust valve
Table 2.3 Geometric details of Ricardo NMK III experimental single-cylinder engine.

Valve Lift - Intake = 9.96 mm

Valve Lift - Exhaust = 10.23 mm

Spark Plug Location = (17 mm, 0 mm, 15.24 mm)

Bore = 85.7 mm

Stroke = 86.0 mm

Connecting Rod Length = 157.9 mm

Compression Ratio = 8.29

Displaced Volume = 496 cm$^3$

Clearance Volume = 68 cm$^3$

Intake Valve Opening = 7 dbtc

Intake Valve Closing = 51 dabc

Exhaust Valve Opening = 101 dbbc

Exhaust Valve Closing = 39 datc
Table 2.4 Operating conditions of Ricardo NMK III experimental data.

<table>
<thead>
<tr>
<th>Run</th>
<th>Speed [rpm]</th>
<th>P(in) [atm.]</th>
<th>T(spark) [dbtc]</th>
<th>Phi (intake)</th>
<th>Phi (exhst.)</th>
<th>Phi (ACART)</th>
<th>Fuel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Exp - 51</td>
<td>1500.00</td>
<td>1.00</td>
<td>22.0</td>
<td>0.998</td>
<td>0.986</td>
<td>1.017</td>
<td>Indolene</td>
</tr>
<tr>
<td>Exp - 59</td>
<td>1500.00</td>
<td>0.70</td>
<td>25.0</td>
<td>1.000</td>
<td>1.025</td>
<td>1.020</td>
<td>Indolene</td>
</tr>
<tr>
<td>Exp - 47</td>
<td>1000.00</td>
<td>1.00</td>
<td>12.5</td>
<td>1.000</td>
<td>1.029</td>
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</tr>
<tr>
<td>Exp - 49</td>
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<td>25.0</td>
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<td>1.012</td>
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<tr>
<td>Exp - 55</td>
<td>2500.00</td>
<td>0.70</td>
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<td>0.998</td>
<td>0.977</td>
<td>1.016</td>
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<tr>
<td>Exp - 53</td>
<td>2500.00</td>
<td>1.00</td>
<td>25.0</td>
<td>0.998</td>
<td>0.979</td>
<td>1.011</td>
<td>Indolene</td>
</tr>
<tr>
<td>Exp - 110</td>
<td>1500.00</td>
<td>1.00</td>
<td>23.0</td>
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<td>1.196</td>
<td>1.138</td>
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</tr>
<tr>
<td>Exp - 111</td>
<td>1500.00</td>
<td>1.00</td>
<td>32.0</td>
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<td>0.800</td>
<td>0.810</td>
<td>Indolene</td>
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<tr>
<td>Exp - 112</td>
<td>1500.00</td>
<td>1.00</td>
<td>23.0</td>
<td>---</td>
<td>0.800</td>
<td>0.810</td>
<td>Indolene</td>
</tr>
<tr>
<td>Exp - 113</td>
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<tr>
<td>Exp - 114</td>
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<td>0.58</td>
<td>30.0</td>
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<td>1.000</td>
<td>1.001</td>
<td>Indolene</td>
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<tr>
<td>Exp - 108</td>
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<td>1.00</td>
<td>23.0</td>
<td>---</td>
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<td>1.011</td>
<td>Indolene</td>
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<td>Group1.30</td>
<td>1800.00</td>
<td>1.00</td>
<td>26.0</td>
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<td>1.197</td>
<td>1.170</td>
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<td>Group1.20</td>
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<td>0.996</td>
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<td>Group1.25</td>
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<td>0.822</td>
<td>0.814</td>
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<td>Group1.28</td>
<td>1800.00</td>
<td>1.00</td>
<td>32.0</td>
<td>0.914</td>
<td>0.917</td>
<td>0.914</td>
<td>Indolene</td>
</tr>
</tbody>
</table>
Table 2.5 Optimized constants for cycle simulation.

\[
\begin{align*}
&u' = 1.0 \\
&c_\beta = 1.5 \\
&T_{\text{head}} = 450 \text{ K} \\
&T_{\text{wall}} = 370 \text{ K} \\
&T_{\text{piston}} = 500 \text{ K} \\
&T_{\text{intake}} = 320 \text{ K}
\end{align*}
\]

Table 2.6 Base conditions for generation of thermal efficiency data base.

Bore = 85 mm

Stroke = 85 mm

Connecting Rod Length = 148.75 mm

Displaced Volume = 482.33 cm\(^3\)/cylinder

Clearance Volume = 53.59 cm\(^3\)/cylinder

Spark Location = 14.17 mm, 0.0 mm

Hemispherical Combustion Chamber Shape

Valve Timing:
- IVO = 12.5 btdc
- IVC = 50.0 atdc
- EVO = 55.0 bbdc
- EVC = 15.0 atdc

Inlet Valve Data:
- Diameter = 41.40 mm
- Lift = 9.30 mm
- Seat Angle = 45 degrees

Exhaust Valve Data:
- Diameter = 33.99 mm
- Lift = 9.27 mm
- Seat Angle = 45 degrees
Table 2.7 Test matrix for data generation using cycle simulation.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Speed [RPM]</th>
<th>Load - MAP [atm.]</th>
<th>Comp. Ratio</th>
<th>B/L</th>
<th>Dspl Vol [cc]</th>
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<tr>
<td>Speed</td>
<td>1000 - 5000</td>
<td>0.30 - 0.94</td>
<td>10.0</td>
<td>1.0</td>
<td>482.33</td>
</tr>
<tr>
<td>Load</td>
<td>1000 - 5000</td>
<td>0.30 - 0.94</td>
<td>10.0</td>
<td>1.0</td>
<td>482.33</td>
</tr>
<tr>
<td>Comp Rat</td>
<td>3000</td>
<td>0.75</td>
<td>8.5 - 11.5</td>
<td>1.0</td>
<td>482.33</td>
</tr>
<tr>
<td>B/L</td>
<td>3000</td>
<td>0.70</td>
<td>10.0</td>
<td>0.5 - 1.5</td>
<td>482.33</td>
</tr>
<tr>
<td>Dspl Vol</td>
<td>3000</td>
<td>0.75</td>
<td>10.0</td>
<td>1.0</td>
<td>215.7 - 909.2</td>
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Table 2.8 Engine specifications for engines used in model comparison.

<table>
<thead>
<tr>
<th>Engine</th>
<th>EGR</th>
<th>Fuel</th>
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<tr>
<td>A</td>
<td>NO</td>
<td>Indolene Clear</td>
</tr>
<tr>
<td>B</td>
<td>YES</td>
<td>Amoco 91</td>
</tr>
<tr>
<td>C</td>
<td>YES</td>
<td>Amoco 91</td>
</tr>
</tbody>
</table>
Figure 2.1 Flow chart showing the thermodynamic analysis procedure used in the cycle simulation [6].
Figure 2.2 Approximate representation of the combustion chamber geometry in the cycle simulation [6].
principle:

application:

Figure 2.3 Calculation of the flame front area in the cycle simulation [6].
Figure 2.4 Logic of modified cycle simulation program.
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Figure 2.6 The effect of the modified combustion terms on (a) burned mass fraction and (b) entrained mass fraction.
Figure 2.7 Combustion chamber of Ricardo NMK III experimental single-cylinder engine.
Operating Conditions

Exp. Title: Ricardo.51
Model Title: Rnl32_new_press
Intake Pressure: 1.0 atm.
Spark Timing: 22 dbTC
Fuel-Air Equivalence Ratio: 0.998
Engine Speed: 1500 rpm

Results

<table>
<thead>
<tr>
<th></th>
<th>gIMEP (atm)</th>
<th>M(fuel) (g/cyc)</th>
<th>M(air) (g/cyc)</th>
<th>m(fuel) (g/s)</th>
<th>m(air) (g/s)</th>
<th>Resid. Fract.</th>
<th>Fuel Type</th>
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<td>Experiment</td>
<td>9.630</td>
<td>0.0313</td>
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<td>Model</td>
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<td>0.4979</td>
<td>0.4113</td>
<td>6.222</td>
<td>5.00</td>
<td>Iso-Octan</td>
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</table>

Figure 2.8 Comparison between cycle simulation and Ricardo NMK III results.
Figure 2.9 Comparison between the cycle simulation and Ricardo NMK III results at different operating conditions: (a) fuel consumption, (b) thermal efficiency, (c) indicated mean effective pressure, and (d) air and fuel flow.
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Figure 2.13 Gross thermal efficiency as a function of compression ratio.

Figure 2.14 Thermal efficiency as a function of displaced volume.
Figure 2.15 Thermal efficiency as a function of bore-stroke ratio.

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Figure 3.2 Variation of engine combustion efficiency with fuel-air equivalence ratio [5].
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Figure 3.4 Logarithmic plot of heat transfer as a function of manifold absolute pressure. [Note: Squares - data at 1000 RPM; Pluses - data at 3000 RPM; Diamonds - data at 5000 RPM; Triangles - sub-model prediction]
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Figure 3.9 Displaced volume thermal efficiency sub-model prediction of cycle simulation results. [Note: Squares – data; Pluses – sub-model predictions]

Figure 3.10 Total model thermal efficiency prediction as a function of manifold absolute pressure: (a) 1000 RPM, (b) 3000 RPM, and (c) 5000 RPM.
Figure 3.10 (cont'd)
Figure 3.11 Total model thermal efficiency prediction as a function of bore-stroke ratio.

Figure 3.12 Total model thermal efficiency prediction as a function of compression ratio.
Figure 3.13  Total model thermal efficiency prediction as a function of stroke.

Figure 3.14  Total model thermal efficiency prediction as a function of bore.
Figure 3.15 Total model thermal efficiency prediction as a function of displaced volume.

Figure 4.1 Brake power for several different engine geometries.
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Figure 4.3 Maximum rated mean piston speed for various engine geometry configurations.
Figure 4.4 Required spark retard for several production engines.

Figure 4.5 Nitschke spark retard prediction of engine data. [Note: Squares - engine data; Pluses - Nitschke model predictions]
Figure 4.6 Octane requirement versus compression ratio for passenger cars on the road today [29].

Figure 4.7 The effect on inlet pressure and octane requirement on spark retard [31].
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Figure 4.9 Reduction of thermal efficiency due to spark retard.
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Figure 5.1 (cont'd)
Figure 5.2 Normalized volumetric efficiency as a function of manifold absolute pressure for several production engines.
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Figure 5.3 (cont'd)
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Figure 6.3 Model comparison with engine data at low load: performance, fuel consumption (economy), and efficiencies.
Figure 6.4 Model comparison with engine data at mid-range speed: performance, fuel consumption (economy), and efficiencies. Note: data corrected for EGR is also shown.
Figure 6.5 Model comparison with engine data at low speed: performance, fuel consumption (economy), and efficiencies. Note: data corrected for EGR is also shown.
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Figure 6.20 Effect of bore-stroke ratio on efficiencies at specified operating conditions.
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Figure 6.22 Effect of displaced volume on efficiencies at specified operating conditions.
Figure 6.23 Effect of compression ratio on maximum performance and minimum fuel consumption.

Figure 6.24 Effect of compression ratio on efficiencies for maximum performance and minimum fuel consumption.
Figure 6.25 Effect of bore-stroke ratio on maximum performance and minimum fuel consumption.

Figure 6.26 Effect of bore-stroke ratio on efficiencies for maximum performance and minimum fuel consumption.
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Appendix A

New Combustion Terms in Cycle Simulation Subroutine
SUBROUTINE CMBSLT

PURPOSE
CALCULATES THE TIME RATE OF CHANGE OF PRESSURE, TEMPERATURE, MASS, HEAT TRANSFER, WORK TRANSFER, MEAN KINETIC ENERGY, AND TURBULENT KINETIC ENERGY IN THE CYLINDER DURING COMBUSTION.

USAGE
CALL CMBSLT (DT, DY, DYP)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>DT</td>
<td>YES</td>
<td>NO</td>
<td>TIME (DEG)</td>
</tr>
<tr>
<td>DY(1)</td>
<td>YES</td>
<td>NO</td>
<td>MASS INDUCED INTO CHAMBER THROUGH INTAKE VALVE (G)</td>
</tr>
<tr>
<td>DY(2)</td>
<td>YES</td>
<td>NO</td>
<td>MASS EXHAUSTED FROM CHAMBER THROUGH EXHAUST VALVE (G)</td>
</tr>
<tr>
<td>DY(3)</td>
<td>YES</td>
<td>NO</td>
<td>MASS FRACTION ENTRAINING (-)</td>
</tr>
<tr>
<td>DY(4)</td>
<td>YES</td>
<td>NO</td>
<td>MASS FRACTION BURNED (-)</td>
</tr>
<tr>
<td>DY(5)</td>
<td>YES</td>
<td>NO</td>
<td>MASS FRACTION OF FRESH CHARGE (-)</td>
</tr>
<tr>
<td>DY(6)</td>
<td>YES</td>
<td>NO</td>
<td>MEAN KINETIC ENERGY IN CHAMBER (F)</td>
</tr>
<tr>
<td>DY(7)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - PISTON TOP (KJ)</td>
</tr>
<tr>
<td>DY(8)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - CYLINDER HEAD (KJ)</td>
</tr>
<tr>
<td>DY(9)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - CYLINDER WALL (KJ)</td>
</tr>
<tr>
<td>DY(10)</td>
<td>YES</td>
<td>NO</td>
<td>CYLINDER PRESSURE (ATM)</td>
</tr>
<tr>
<td>DY(11)</td>
<td>YES</td>
<td>NO</td>
<td>TEMPERATURE OF UNBURNED MIXTURE DURING COMBUSTION (K)</td>
</tr>
<tr>
<td>DY(12)</td>
<td>YES</td>
<td>NO</td>
<td>VOLUME OF UNBURNED MIXTURE DURING COMBUSTION (CM**3)</td>
</tr>
<tr>
<td>DY(13)</td>
<td>YES</td>
<td>NO</td>
<td>TEMPERATURE OF BURNED PRODUCTS DURING COMBUSTION (K)</td>
</tr>
<tr>
<td>DY(14)</td>
<td>YES</td>
<td>NO</td>
<td>TOTAL WORK TRANSFER (KJ)</td>
</tr>
<tr>
<td>DY(15)</td>
<td>YES</td>
<td>NO</td>
<td>MASS FRACTION OF (NO) IN ADIABATIC CORE ZONE (MNOC/MAC)</td>
</tr>
<tr>
<td>DY(16)</td>
<td>YES</td>
<td>NO</td>
<td>MASS FRACTION OF (NO) IN BOUNDARY LAYER ZONE (MNOSL/MBL)</td>
</tr>
<tr>
<td>DY(17)</td>
<td>YES</td>
<td>NO</td>
<td>MASS FRACTION IN BOUNDARY LAYER ZONE (-)</td>
</tr>
<tr>
<td>DYP(20)</td>
<td>YES</td>
<td>NO</td>
<td>RATE AT WHICH MASS IS INDUCED THROUGH THE INTAKE VALVE (G/DEG)</td>
</tr>
<tr>
<td>DYP(21)</td>
<td>YES</td>
<td>NO</td>
<td>RATE AT WHICH MASS IS EXHAUSTED THROUGH THE EXHAUST VALVE (G/DEG)</td>
</tr>
<tr>
<td>DYP(22)</td>
<td>YES</td>
<td>NO</td>
<td>RATE OF CHANGE OF MASS FRACTION ENTRAINING (1/DEG)</td>
</tr>
<tr>
<td>DYP(23)</td>
<td>YES</td>
<td>NO</td>
<td>RATE OF CHANGE OF MASS FRACTION BURNED (1/DEG)</td>
</tr>
<tr>
<td>DYP(24)</td>
<td>YES</td>
<td>NO</td>
<td>RATE OF CHANGE OF MASS FRACTION OF FRESH CHARGE (1/DEG)</td>
</tr>
<tr>
<td>DYP(25)</td>
<td>YES</td>
<td>NO</td>
<td>RATE OF CHANGE OF MEAN KINETIC ENERGY (KJ/DEG)</td>
</tr>
<tr>
<td>DYP(26)</td>
<td>YES</td>
<td>NO</td>
<td>RATE OF HEAT TRANSFER - CYLINDER HEAD (KJ/DEG)</td>
</tr>
<tr>
<td>DYP(27)</td>
<td>YES</td>
<td>NO</td>
<td>RATE OF HEAT TRANSFER - PISTON TOP (KJ/DEG)</td>
</tr>
<tr>
<td>DYP(28)</td>
<td>YES</td>
<td>NO</td>
<td>RATE OF HEAT TRANSFER - CYLINDER WALL (KJ/DEG)</td>
</tr>
<tr>
<td>DYP(29)</td>
<td>YES</td>
<td>NO</td>
<td>RATE OF CHANGE OF CYLINDER PRESSURE (ATM/DEG)</td>
</tr>
<tr>
<td>DYP(30)</td>
<td>YES</td>
<td>NO</td>
<td>RATE OF CHANGE OF UNBURNED MIXTURE TEMPERATURE DURING COMBUSTION (K/DEG)</td>
</tr>
</tbody>
</table>
DYP(14) NO YES RATE OF CHANGE OF UNBURNED MIXTURE
DYP(15) NO YES RATE OF CHANGE OF RUBBLIZED PRODUCTS
DYP(16) NO YES RATE OF TOTAL WORK TRANSFER (KJ/DEG)
DYP(18) NO YES RATE OF CHANGE OF (NO) MASS FRACTION
DYP(19) NO YES RATE OF CHANGE OF (NO) MASS FRACTION
DYP(20) NO YES RATE OF CHANGE OF MASS FRACTION

REMARKS
NONE

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
UTHRMO UTRANS CSADV

METHOD
SEE S. G. POULOS SM THESIS

WRITTEN BY S. H. MANSOURI AND S. G. POULOS
EDITED BY S. G. POULOS

EDITED BY ROBERT T. CHANG 10 DECEMBER 1986
ADDITIONS: NEW COMBUSTION TERMS WERE ADDED FOR THE MASS FRACTION
BURNED AND ENTRAINHE.

SUBROUTINE CMBSTN_NEW (DT, DY, DYP)

LOGICAL SPBURN, ENSTOP
REAL8 DT, LY(20), DYP(20), D4(4), XMOfR(14), TMP, PRG
REAL MW, M2IM, M2IMM, EINVIS, MASS, MDOT, MDOTFR, MSTART, MACRSC, MACRSP,
& MIRCSC, MASSU, MASSAG, MWBL, MMAC, MMNO, MMN, MM02, MM0H, K, K1, K2, K3
DIMENSION Y(20), YP(20)
COMMON/EPARAM/ BORE, STROKE, CONRL, CULCA, CSATDC, CMRTIO, CLVTDC
COMMON/HTRC/ CNH, EXPHT
COMMON/TEMPS/ 7PSTON, THEAD, TCW
COMMON/BURN/ SPBURN, FIRE, RPM
COMMON/DDTDH/ ESPDI
COMMON/MANFP/ PIM, TIM, EGR, PEM, MSTART
COMMON/TIMES/ TIV, TEVC, TIVC, TSARK, TEVO
COMMON/IMT/ HIM, M2IM, GIM, RHOIM
COMMON/FLAG/ INFAG
COMMON/HEATS/ CVHTRN, HTRCOE, HTPAPI, HTPAHM, HTPACW, HTRAPI,
& HTRHM, HTRCM, HTTRAN, QFRPI, QFRH, QFRCW
COMMON/UBHEAT/* UHTRCO, UHTRCO
COMMON/TURBU/ CBI, MACRSC, UPRIME, VMKE
COMMON/VALUE/ VIV, VEV
COMMON/KINET/ TURBKE, MIRCSC, SSUBL, BTMSC
COMMON/SPARK/ UPRISP, RHOSP, MACRSP, CMULT
COMMON/CHDM/ D
COMMON/SPECB/ DTBRC, CONSB, ESXSPB
COMMON/NOX/ YAC, YNO, XNOAC, XNOBL, XNO, PPMA, PPMBL, PPW0
COMMON/ACBL/ VACONV, VBLONV, DBLONB, TWALLB, TAC, TBLAYR
COMMON/FLOQ/MVONV, VBRONV, AVLONB, APONB, AHONB,
& ABONB, ACUONB, ACBONB, DFLONB
COMMON/XSTOP/ XSTOP, XBSTOP, ENSTOP, IFCONT
COMMON/RADS/ RENTRA, RBURND

DO 10 I = 1, 20
Y(I) = DY(I)
10 CONTINUE
T = DT
DO 20 I = 1, 20
20 CONTINUE

C FIND THERMODYNAMIC AND TRANSPORT PROPERTIES IN CYLINDER

RESFRK = 1. - Y(5)
CALL THERMO (T, Y(13), Y(12), RESFRK, HU, CSUBPU, CSUBTU,
& RHOU, DRODTU, DRODPU, XXA, XXB, ADUMU, BDUMU, XXC, XXD)
CALL THERMO (T, Y(15), Y(12), 1., HB, CSUBPB, CSUBTB,
& RHOB, DRODTB, DRODPB, GAMAB, XXB, ADUMB, BDUMB, XXC, XXD)
CALL UTRANS (Y(13), UDYVIS, UTHRCN)
CALL STRANS (Y(15), GAMAB, CSUBPB, BDDYVIS, BTHRCN)
UKNVIS = UDYVIS/RHOU
BKNVIS = BDDYVIS/RHOB

C CALCULATE TOTAL MASS AND UNBURNED MASS IN CYLINDER

MASS = MSTART + Y(1) - Y(2)
MASSU = MASS*(1. - Y(4))

CALL CSAADV (T, AHEAD, APSTON, ACW, VOLUME, DVDT)
VENTRA = VOLUME - (MASS/RHOU)*(1. - Y(3))
VBURNC = VOLUME - Y(14)

C IF ENTRAINMENT IS ALMOST COMPLETE, USE SIMPLE FLAME GEOMETRY

IF (.NOT. ENSTOP) GO TO 25
23 AFLAM = AFSTOP * ((TEVO - T)*(TEVO - T)/((TEVO - TSTOP)*)
& (TEVO - TSTOP))
APISU = 0.0
AHEADU = 0.0
ACWU = 0.0
FLMDEL = 0.0
APISB = APSTON
AHEADB = AHEAD
ACWB = ACW
GO TO 27

C CALCULATE GEOMETRIC INTERACTION BETWEEN FLAME AND CHAMBER

25 CALL FLAMEGE (T, VENTRA, VBURNC, RENTRA, RBURNC, AFLAM, APISU,
& APISB, AHEADU, AHEADB, ACWU, ACWB, FLMDEL)
27 VENDOR = VENTRA/VOLUME
VBRON = VBURNC/VOLUME
AFLONB = AFLAM/(BORE*BORE)
APUONB = APISU/(BORE*BORE)
APBONB = APISB/(BORE*BORE)
AHUONB = AHEADU/(BORE*BORE)
AHBONB = AHEADB/(BORE*BORE)
ACUONB = ACWU/(BORE*BORE)
ACBONB = ACWB/(BORE*BORE)
DFLONB = FLMDEL/BORE
IF (((Y(3) .GT. 0.09) .AND. (IFCNT .LE. 0)) GO TO 28
IF (((AFLONB .GT. 1.0E-8) .OR. (T .LE. TSPARK) .OR.
& (IFCNT .GT. 0) .OR. (SPBURN)) GO TO 29
C C STORE FINAL FLAME AREA PRIOR TO USING SIMPLE FLAME GEOMETRY

28 AFSTOP = AOLDF
TSTOP = TOLDF
ENSTOP = .TRUE.
IFCNT = IFCNT + 1
GO TO 23
29 AOLDF = AFLAM
TOLDF = T

C CALCULATE TURBULENT FLOW AND COMBUSTION MODEL PARAMETERS
MACRSC = MACRSP * (RHOUSP/RHOU)**(1./3.)
IF (MACRSC .GE. (BORE/2.)) MACRSC = BORE/2.
CALL LAMFSP (Y(13), Y(12), RESFRK, SSUBL)
UPRIME = CMULT * UPRISP * (RHOU/RHOUSP)**(1./3.)

CMULT: ARTIFICIAL FACTOR WHICH MULTIPLIES U' DURING COMBUSTION

MICRSC = MACRSC * SORT(15. * UKNVIS/(UPRIME * MACRSC))
BTIMSC = MICRSC/SSUBL
IF (SPBURN) GO TO 30

PREDICTED BURN RATE COMBUSTION MODEL
CALCULATE ENTRAINMENT AND BURN RATES

ADDITIONS ON 11 DECEMBER 1986:
1. THE UPRIME TERM IN THE MASS FRACTION ENTRAINTED EQUATION
   (YP(3)) WAS MULTIPLIED BY A DECAYING EXPONENTIAL FUNCTION
   TO ACCOUNT FOR THE SMALL TURBULENCE PRESENT IN THE
   INITIAL STAGES OF COMBUSTION.
2. THE MASS FRACTION BURNED EQUATION (YP(4)) WAS MODIFIED TO
   INCLUDE A LAMINAR FLAME SPEED TERM TO ACCOUNT FOR THE
   ADVANCEMENT OF FLAME AT LAMINAR FLAME SPEED THROUGH THE
   ENTRAINMENT REGION.
3. THE MASS FRACTION BURNED EQUATION (YP(4)) REQUIRED A RAMP
   TOWARD THE END OF COMBUSTION TO ENHANCE A SMOOTH TRANSITION
   FROM PARTIAL BURNED MASS TO COMPLETELY BURNED.

TFSOC=(T-TSPARK)/(6.8*RPM)
YP(3) = RHOU*AFLAM * (UPRIME*(1-EXP(-TFSOC/BTIMSC)) +
& SSUBL/MASS)
IF (Y(3) .GE. ESTOP) YP(3) = YP(3)/1.5
IF (Y(3) .GE. 0.998) YP(3) = YP(3)/1.5
IF (Y(3) .GE. 0.999) YP(3) = YP(3)/1.5
IF (Y(3) .GE. 0.9999) YP(3) = 0.0
YP(4) = (Y(3) - Y(4))/BTIMSC +(RHOU*AFLAM*SSUBL)/MASS
IF (Y(4) .GE. XBSTOP) YP(4) = YP(4)/1.5
IF (Y(4) .GE. 0.998) YP(4) = YP(4)/1.5
IF (Y(4) .GE. 0.999) YP(4) = YP(4)/1.5
IF (Y(4) .GE. 0.9999) YP(4) = 0.0
GO TO 40

SPECIFIED BURN RATE COMBUSTION MODEL

30 TONDB = (T-TSPARK)/DTBNR
IF (TONDB .GT. 1.0) TONDB = 1.0
YP(4) = CONSPB*(EXSPB + 1.)*(TONDB)**(EXSPB)*EXP(-CONSPB*
& TONDB***(EXSPB + 1.) )/DTBNR*ESPD)
IF (Y(4) .GE. XBSTOP) YP(4) = YP(4)/1.5
IF (Y(4) .GE. 0.998) YP(4) = YP(4)/1.5
IF (Y(4) .GE. 0.999) YP(4) = YP(4)/1.5
IF (Y(4) .GE. 0.9999) YP(4) = 0.0
YP(3) = YP(4)

40 TURBKE = 1.5 * MASS * UPRIME**2.
YP(6) = -.3387 * CBETA/MACRSC * Y(6) * SORT( TURBKE/MASS )

CVHTRN: CHARACTERISTIC VELOCITY 'N CYLINDER; (CM/SEC).

PI = 3.141592654
CONSTR = CONRL/STROKE
SINTH = SIN( T*PI/180. )
COSTH = COS( T*PI/180. )
VONVPM = ABS( PI * SINTH * ( 1. + COSTH/SQRT( 4. * CONSTR*CONSTR
& - SINTH*SINTH ) )/2. )
VPMEAN = STROKE/(180. * ESPDI)
VPSTO = VPMEAN + VONVPM
VMKE = SQRT( 2. * Y(6)/MASS )
CVHTRN = SQRT( 0.25*VPSTO*VPSTO + VMKE*VMKE + UPRIME*UPRIME )

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CALCULATE HEAT TRANSFER RATE.

UHTRCO = CONHT*( (CVHTRN+MACRSC/UKNVIS)**EXPHT ) * UTHRCH/MACRSC
BHTRCO = CONHT*( (CVHTRN+MACRSC/UKNVIS)**EXPHT ) * BTHRCH/MACRSC
UHPAPI = UHTRCO * ( Y(13) - TPSTON )
BHPAPI = BHTRCO * ( Y(15) - TPSTON )
UHPAHD = UHTRCO * ( Y(13) - THEAD )
BHPAHD = BHTRCO * ( Y(15) - THEAD )
UHPACW = UHTRCO * ( Y(13) - TCW )
BHPACW = BHTRCO * ( Y(15) - TCW )

UHTRPI = UHPAPI * APISU
BHTRPI = BHPAPI * APISB
UHTRHD = UHPAHD * AHEADU
BHTRHD = BHPAHD * AHEADB
UHTRCW = UHPACW * ACWU
BHTRCW = BHPACW * ACWB

HTPAPI = (UHTRPI + BHTRPI)/(APISU + APISB)
HTPAHD = (UHTRHD + BHTRHD)/(AHEADU + AHEADB)
HTPCW = 0.0
IF ( T .EQ. 360.) GO TO 50
HTPCW = (UHTRCW + BHTRCW)/(ACWU + ACWB)
50 CONTINUE

UHTRAN = UHTRPI + UHTRHD + UHTRCW
BHTRAN = BHTRPI + BHTRHD + BHTRCW

YP(8) = 1.E-10 * (UHTRPI + BHTRPI)
YP(9) = 1.E-10 * (UHTRHD + BHTRHD)
YP(10) = 1.E-10 * (UHTRCW + BHTRCW)

CALCULATE RATES OF CHANGE OF TEMPERATURE AND PRESSURE IN THE CYLINDER. THEN CALCULATE RATE OF DOING WORK.

CDUMU = 1. - ( DROTU * BDUMU )/( RHOU * ADUMU )
CDUMB = 1. - ( DROTB * BDUMB )/( RHOB * ADUMB )
DDUMY = YP(4)/(Y(4) - 1.) - (1. - 1./CDUMU) * UHTRAN/
& (BDUMU+MASSU)
EDUMY = ( VBURN/Y(14) )* ( DVDT/DBURN - YP(4)/Y(4) -
& (1. - 1./CDUMB)* ( YP(4)*(HU - HB)/(Y(4)*BDUMB) -
& BHTRAN/(BDUMB*Y(4)*MASS) ) )
FDUMY = VBURN*DRDPS/* ( Y(14)+CDUMB*RHOB ) + DRDPU/(CDUMB*RHOU)
YP(12) = ( DDUMU - EDUMY ) / FDUMY
IF ( MASSU .EQ. 0.0 ) MASSU = 0.00005 * MASS
YP(14) = Y(14)* ( DDUMY - DRDPU*YP(12)/(CDUMU+RHOU) )
YP(13) = (BDUMU/ADUMU)* ( YP(4)/(Y(4) - 1.) - YP(14)/Y(14) -
& UHTRAN/( BDUMU+MASSU ) )
DVDTB = DVDT - YP(14)
YP(15) = (BDUMB/ADUMB)* ( YP(4)*(1. + (HU - HB)/BDUMB )/Y(4) -
 & DVDTB/DBURN - BHTRAN/( BDUMB*Y(4)*MASS ) )
YP(16) = Y(12) * DVDT * .101325E-3

END OF PERFORMANCE CALCULATION

START OF NOX CALCULATION;
BURNT ZONE IS DIVIDED INTO AN ADIABATIC
CORE AND A THERMAL BOUNDARY LAYER

IF ( (APISB + AHEADB + ACWB) .GT. 0.0 ) GO TO 53
TWALLB = THEAD
GO TO 55
53 TWALLB = ( (APISB+TPSTON) + (AHEADB+THEAD) + (ACWB+TCW) )/
 & (APISB + AHEADB + ACWB)
FOR ADIABATIC ENGINE USE SINGLE ZONE MODEL (NO B. L.)

IF (CONHT .LE. 0.0) GO TO 55

TBLAYR = 0.5*(TWALLB + Y(15))
CALL THERMO (T, TBLAYR, Y(12), 1., HBL, XXA, XXB, RHobl, DRDTBL,
& DRDPBL, MMBl, XXD, ADUMBBL, BDUMBBL, XXG, XXH)

TDOTTBL = 0.5*YP(15)

CALCULATE RATE OF CHANGE OF BOUNDARY LAYER MASS

YP(20) = ((1. - Y(20)/Y(4))*((ADUMBBL*Y(20)*TDOTTBL - BDUMBBL*
& (DRDTBL*TDOTTBL + DRDPBL*YP(12))/(RHobl*ADUMBBL)) +
& BHTRAN/MASS ) )/(HB - HBL)

GO TO 57

55 YP(20) = 0.0

YP(19) = 0.0

TBLAYR = Y(15)

RHobl = RHob

57 VBLAYR = MASS*Y(20)/RHobl

VBLONV = VBLAYR/VOLUME

DELDTBL = 0.0

IF ((APIsB + AHEADB + ACWB) .GT. 0.0)
& DELDTBL = VBLAYR/(APIsB + AHEADB + ACWB)

DBLONB = DELDTBL/BORE

VADIAC = VBUWND - VBLAYR

VACONV = VADIAC/VOLUME

MASSAC = MASS*Y(4)*(1. - Y(20)/Y(4))

HAC = (Y(4)*HB - Y(20)*HBL)/(Y(4) - Y(20))

TAC = 1.03*Y(15)

IF (CONHT .LE. 0.0) TAC = Y(15)

CALCULATE ADIABATIC CORE TEMPERATURE

CALL ITRATE (T, TAC, Y(12), 1.0, HAC, XXA, XXB, ROAOAC,
& XXC, XXD, XXE, MWAc, XXF, XXG, XXH, XXI)

MMNO = 30.

MMN = 14.

MMO2 = 32.

MMOH = 17.

TMP = TAC

PRS = Y(12)

OBTAIN EQUILIBRIUM MOLE FRACTIONS OF NO, N, O2, AND OH

CALL PTCHEM (TMP, PRS, D, XMOfR, ISENT)

IF (ISENT .EQ. 2) GO TO 60

EQNO = XMOfR(9)*(MMNO/MMAC)

EQN = XMOfR(10)*(MMN/MMAC)

EQO2 = XMOfR(14)*(MMO2/MMAC)

EQOH = XMOfR(5)*(MMOH/MMAC)

K1 = 7.6E+13 * DEXP(-38.00+3/TMP)

K2 = 1.5E+09 * TMP * DEXP(-19.50+3/TMP)

K3 = 4.1E+13

R1 = K1 * EQN * EQUO

R2 = K2 * EQN * EQO2

R3 = K3 * EQN * EQOH

K = R1/(R2 + R3)

CALCULATE RATES OF CHANGE OF NO IN BOUNDARY LAYER
AND IN ADIABATIC CORE

IF (CONHT .GT. 0.0) YP(19) = YP(20) * Y(18)/Y(20)
\[
YP(18) = (2.0*MWNO/RHOAC)*(1.0 - Y(18)*Y(18)/(EQNO*EQNO))^{1.0}/
\]
\[+ (1.0 + K*Y(18)/EQNO) = MASS*Y(20)*Y(18)/MASSAC
\]
\[
YAC = Y(4) - Y(20)
\]
\[YNO = Y(18)*YAC + Y(19)*Y(20)
\]
\[
XNOAC = Y(18) - MWAC/MWNO
\]
\[XNOBL = Y(19) - MWBL/MWNO
\]
\[XNO = (XNOAC+VADIAC/TAC + XNOBL+VBLAYR/TBLAYR)/
\]
\[+ (VADIAC/TAC + VBLAYR/TBLAYR + Y(14)/Y(13))
\]
\[
PPMAC = 1.0E+06 * XNOAC
\]
\[PPMBL = 1.0E+06 * XNOBL
\]
\[PPMNO = 1.0E+06 * XNO
\]
\[END OF NOX CALCULATION
\]
\[60 \ YP(12) = YP(12)/1.01325*E+6
\]
\[UHTRCO = UHTRCO * 1.0E-6
\]
\[BHTRCO = BHTRCO * 1.0E-6
\]
\[HTAPI = HTAPI * 1.0E-6
\]
\[HTHDI = HTHDI * 1.0E-6
\]
\[HTACW = HTACW * 1.0E-6
\]
\[HTRAPI = YP(8)
\]
\[HTRAHAD = YP(8)
\]
\[HTRACW = YP(18)
\]
\[THTRAN = HTRAPI + HTRAHAD + HTRACW
\]
\[QFRPI = 0.0
\]
\[QFRHD = 0.0
\]
\[QFRCW = 0.0
\]
\[IF ((CONHT .LE. 0.0) .OR. (THTRAN .EQ. 0.0)) GO TO 70
\]
\[QFRPI = 100. * HTRAPI/THTRAN
\]
\[QFRHD = 100. * HTRAHAD/THTRAN
\]
\[QFRCW = 100. * HTRACW/THTRAN
\]
\[THTRAN = 1000. * THTRAN * ESPDI
\]
\[CONVERT ALL TIME DERIVATIVES TO RATE PER CRANK ANGLE DEGREE.
\]
\[70 \ DO 80 I = 1, 20
\]
\[DYP(I) = YP(I) * ESPDI
\]
\[80 \ CONTINUE
\]
\[RETURN
\]
\[END
\]
Appendix B

Chamber Geometry Input Program
PROGRAM GEOMINP

THIS PROGRAM GENERATES THE DATA FOR USE AS THE INPUT TO THE
"GEOMGEN" PROGRAM. THE USER INPUTS THE BASIC GEOMETRIC
PARAMETERS: BORE, STROKE, CONNECTING ROD LENGTH, SPARK
LOCATION, AND COMPRESSION RATIO. IN ADDITION, THE USER
CAN INPUT ITERATION CONSTANTS IF DESIRED. THE RESULTS WILL
BE USED BY "CHAMBER" TO GENERATE FLAME DATA FOR USE BY
"QUASI", A COMBUSTION MODELLING PROGRAM BY STEVE POULOS.
FOR MORE INFORMATION ON "CHAMBER" AND "QUASI". SEE STEVE
POULOS' THESIS, MASSACHUSETTS INSTITUTE OF TECHNOLOGY,
JULY 1983.

WRITTEN BY ROBERT T. CHANG
20 FEBRUARY 1987
MIT SLOAN AUTOMOTIVE LABORATORY
DR. VICTOR W. WONG, ADVISOR

*********************

VARIABLE IDENTIFICATION

ANS = RESPONSE CHARACTER (Y=YES;N=NO)
NAME = NAME OF GEOMETRY OUTPUT FILE
PLOC = FILE NAME DESCRIPTION OF SPARK LOCATION
VNUM = VERSION NUMBER FOR FILE NAME DESCRIPTION
BORE = BORE OF CHAMBER GEOMETRY
STROKE = STROKE OF CYLINDER GEOMETRY
CNR = CONNECTING ROD LENGTH OF PISTON MECHANISM
CR = COMPRESSION RATIO OF CHAMBER
SX,SY = COORDINATES OF SPARK LOCATION
SZ = VERTICAL DISTANCE FROM CHAMBER WALL TO SPARK
CAINT = CRANK ANGLE INTERVAL FOR ITERATION
CAMAX = MAXIMUM CRANK ANGLE FOR COMBUSTION MODEL
RADSTP = RADIUS STEP SIZE FOR ITERATION
RADPNT = RANDOM POINT DENSITY FOR ITERATION
MINPNT = MINIMUM NUMBER OF RANDOM POINTS FOR ITERATION
MAXPNT = MAXIMUM NUMBER OF RANDOM POINTS FOR ITERATION
ITS = NUMBER OF FLAME AREA ITERATIONS

*********************

TYPE DECLARATIONS AND STORAGE ALLOCATION

CHARACTER NAME*21,ANS*1,PLOC*5,VNUM*2
DATA CAINT,CAMAX,RADSTP,RADPNT/14.0,70.0,24.0,0.0,300.0/
DATA MINPNT,MAXPNT,ITS/200,2200,9/

*********************

PRINT INTRODUCTION

PRINT 100
100 FORMAT(1X,'PROGRAM GEOMINP'.//)
PRINT 110
110 FORMAT(1X,'THIS PROGRAM GENERATES INPUT DATA FOR USE BY THE'.//
  1 1X,'"GEOMGEN" PROGRAM. IT ASSUMES A HEMISPHERICAL CHAMBER'.//
  2 1X,'SHAPE WITH BASIC GEOMETRICAL PARAMETERS INPUT BY THE'.//
  3 1X,'USER.'//)
PRINT 120
120 FORMAT(1X,'DO YOU WISH TO USE THIS PROGRAM? Y/N')
READ (5,125)ANS
125 FORMAT(A1)
IF (ANS.EQ.'N') GOTO 890

-133-
**********  INPUT BASIC GEOMETRICAL PARAMETERS  **********

200 PRINT 100
   NUM=0
PRINT 210
210 FORMAT(1X,'THE FILE NAME FOR THE GEOMETRY FILE IS OF THE/',
    1X,'GENERAL FORM "HEMI_XXXXX_.GEOVAT". THE "XXXXX" IS/',
    1X,'PORTION DESCRIBES THE SPARK PLUG LOCATION. THE/',
    1X,'#" DESCRIBES A TWO-DIGIT VERSION NUMBER. FOR/',
    1X,'EXAMPLE, "HEMI_THIRD_01" WOULD BE VALID./')
PRINT 220
220 FORMAT(1X,'PLEASE INPUT A FIVE CHARACTER NAME DESCRIBING THE/',
    1X,'SPARK PLUG LOCATION..')
READ(5,225)LOC
225 FORMAT(A5)
PRINT 230
230 FORMAT(1X,'PLEASE INPUT A TWO-DIGIT VERSION NUMBER FOR THE FILE.')
READ(5,232) VNUM
232 FORMAT(A2)

* * * CREATE FILE NAME * * *
*
* NAME='HEMI_'//PLOC//'_'//VNUM//'_.INP'
*
* INPUT GEOMETRICAL DATA * * *
*
PRINT 235
235 FORMAT(1X,'PLEASE INPUT THE BORE OF THE CHAMBER.')
READ(5,*) BORE
PRINT 240
240 FORMAT(1X,'PLEASE INPUT THE STROKE OF THE CYLINDER.')
READ(5,*) STROKE
PRINT 250
250 FORMAT(1X,'PLEASE INPUT CONNECTING-ROD LENGTH.')
READ(5,*) CNRL
PRINT 255
255 FORMAT(1X,'PLEASE INPUT DESIRED COMPRESSION RATIO.')
READ(5,*) CR
PRINT 260
260 FORMAT(1X,'PLEASE INPUT CARTESIAN COORDINATES OF SPARK LOCATION /
    1X, '(X,Y). IN A PLANE PERPENDICULAR TO THE CYLINDER AXIS.',
    1X,'THE ORIGIN IS AT THE CENTER OF THE PISTON.').' 
READ(5,*) SX, SY
PRINT 265
265 FORMAT(1X,'IS THE SPARK PLUG FLUSH ON THE CHAMBER WALL? Y/N /
    1X,'(Y = FLUSH AT WALL, N = OFFSET FROM WALL).')
READ(5,125) ANS
IF (ANS.EQ. 'Y') THEN
   SZ=0.0
ELSE
   PRINT 267
267 FORMAT(1X,'PLEASE INPUT THE VERTICAL OFFSET OF THE SPARK PLUG.')
   READ(5,*) SZ
ENDIF

* * ITERATION DATA * *
*
PRINT 270
270 FORMAT(1X,'DO YOU WANT TO USE DEFAULT ITERATION VALUES? Y/N')
READ(5,125) ANS
IF (ANS.EQ. 'Y') GOTO 285
PRINT 275
275 FORMAT(1X,'INPUT CRANK ANGLE INTERVAL, MAXIMUM CRANK ANGLE, '
    1 'RADIUS'/1X,'STEP SIZE, AND RANDOM POINT DENSITY,'
    2 'SEPERATED BY COMMAS.')
READ(5,*) CAINT, CAMAX, RADSTP, RNDPNT
PRINT 280
280 FORMAT(1X,'PLEASE INPUT MINIMUM NUMBER OF RANDOM POINTS, MAXIMUM.'/

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1X, 'NUMBER OF RANDOM POINTS, AND NUMBER OF FLAME AREA' /
2X, 'ITERATIONS SEPERATED BY COMMAS AND IN INTERGER FROM' /
3X, '(NO DECIMAL POINTS).' )
READ(5,*) MINRPT,MAXRPT,ITS

* ECHO DATA *

285 PRINT 286
286 FORMAT (/////) PRINT 288
288 FORMAT(1X, 'INPUT DATA:') /
WRITE(6,289)NAME,BORE,STROKE,CNRL,CR
290 FORMAT(1X, 'FILE NAME = ',A21/) 
1X, 'BORE = ',F10.5/
2X, 'STROKE = ',F10.5/
3X, 'CONNECTING ROD LENGTH = ',F10.5/
4X, 'COMPRESSION RATIO = ',F10.5)
WRITE(6,292)SX,SY
292 FORMAT(1X, 'PLANAR SPARK LOCATION = ',F10.5, ',',F10.5)
WRITE(6,293)SZ
293 FORMAT(1X, 'VERTICAL SPARK LOCATION OFFSET = ',F10.5)
IF (ANS.EQ.'N') THEN WRITE(6,294)CAINT,CAMAX,RADSTP,RNDPNT,MINRPT,MAXRPT,ITS
294 FORMAT(1X, 'CRANK ANGLE INTERVAL = ',F10.4/) 
1X, 'MAXIMUM CRANK ANGLE = ',F10.4/
2X, 'RADIUS STEP SIZE = ',F10.4/
3X, 'RANDOM POINT DENSITY = ',F10.4/
4X, 'MINIMUM # OF RANDOM POINTS = ',I6/ 
5X, 'MAXIMUM # OF RANDOM POINTS = ',I6/ 
6X, 'OF ITERATIONS = ',I6) 
ENDIF PRINT 296
296 FORMAT(1X, 'IS THIS DATA ACCEPTABLE? Y/N') READ(5,125)ANS IF (ANS.EQ.'N') THEN PRINT 298
298 FORMAT(1X, 'DO YOU WANT TO TRY AGAIN? Y/N') READ(5,125)ANS IF (ANS.EQ.'N') GOTO 890 
GOTO 200 ENDIF

* INITIILIZE FILES *
OPEN(9,FILE = NAME,STATUS = 'NEW',FORM = 'FORMATTED')

* WRITE OUTPUT DATA *
WRITE (9,410)BORE,STROKE,CNRL,CR,SX,SY,SZ
410 FORMAT(F10.5)
WRITE (9,410)CAINT,CAMAX,RADSTP,RNDPNT
WRITE (9,420)MINRPT,MAXRPT,ITS
420 FORMAT(I6)

* TEST FOR REPEAT *
CLOSE(9)
PRINT 510
510 FORMAT(1X, 'DO YOU WISH TO CREATE ANOTHER FILE? Y/N') READ (5,125)ANS IF (ANS.EQ.'Y') GOTO 200
END OF PROGRAM

END
Appendix C

Automated Chamber Geometry Program
PROGRAM GEOMGEN

***********

THIS PROGRAM GENERATES THE GEOMETRY DATA FOR USE AS THE
INPUT TO THE "CHAMBER" PROGRAM. THE PROGRAM ASSUMES A
HEMISPHERICAL COMBUSTION CHAMBER SHAPE. THE USER INPUTS
THE BASIC GEOMETRIC PARAMETERS: BORE, STROKE, CONNECTING
ROD LENGTH, SPARK LOCATION, AND COMPRESSION RATIO VIA AN
INPUT FILE CREATE BY "GEOMINP". IN ADDITION, THE USER CAN
INPUT ITERATION CONSTANTS IF DESIRED. THE RESULTS OF THIS
PROGRAM WILL BE USED BY "CHAMBER" TO GENERATE FLAME DATA
FOR USE BY "QUASI", A COMBUSTION MODELLING PROGRAM BY
STEVE Poulos. FOR MORE INFORMATION ON "CHAMBER" AND
"QUASI" SEE STEVE Poulos' THESIS, MASSACHUSETTS INSTITUTE
OF TECHNOLOGY, JULY 1983.

PROGRAMMER'S NOTE: THIS PROGRAM IS DESIGNED TO BE RUN IN
BATCH MODE. THIS CAN MOST EASILY BE ACCOMPLISHED BY USING
"GEOMGEN_LIST" TO CREATE AND SUBMIT THE BATCH PROCESS.
THIS METHOD REQUIRES AN INPUT FILE CREATED BY "GEOMINP".

WRITTEN BY ROBERT T. CHANG
21 NOVEMBER 1986
MIT SLOAN AUTOMOTIVE LABORATORY
DR. VICTOR W. WONG, ADVISOR

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VARIABLE IDENTIFICATION
***********

ANS    = RESPONSE CHARACTER (Y=YES; N=NO)
NAME   = NAME OF GEOMETRY OUTPUT FILE
PLOC   = FILE NAME DESCRIPTION OF SPARK LOCATION
VNUM   = VERSION NUMBER FOR FILE NAME DESCRIPTION
BORE   = BORE OF CHAMBER GEOMETRY
STROKE = STROKE OF CYLINDER GEOMETRY
CNRL   = CONNECTING ROD LENGTH OF PISTON MECHANISM
CR     = COMPRESSION RATIO OF CHAMBER
SX, SY, SZ = COORDINATES OF SPARK LOCATION
DZ     = VERTICAL SPARK LOCATION OFFSET
RSP    = RADIUS OF SPARK IN PLANE PERPENDICULAR TO CYL. AXIS
ZH     = VERTICAL DISTANCE FROM TOP OF HEAD
CAINT  = CRANK ANGLE INTERVAL FOR ITERATION
CAMA   = MAXIMUM CRANK ANGLE FOR COMBUSTION MODEL
RADSTP = RADIUS STEP SIZE FOR ITERATION
RADPNT = RANDOM POINT DENSITY FOR ITERATION
MINPNT = MINIMUM NUMBER OF RANDOM POINTS FOR ITERATION
MAXPNT = MAXIMUM NUMBER OF RANDOM POINTS FOR ITERATION
ITS    = NUMBER OF FLAME AREA ITERATIONS
PI     = NUMERICAL VALUE (3.141592654)
DVOL   = DISPLACED VOLUME (CYLINDER VOLUME)
VCL    = CLEARANCE VOLUME (COMBUSTION CHAMBER VOLUME)
H      = CLEARANCE VOLUME HEIGHT
H0     = THEORETICAL SPARK LOCATION
R      = RADIUS OF IMPOSED COMBUSTION CHAMBER SPHERE
DELT   = CHANGE IN ANGLE AROUND CIRCLE
TH     = ANGLE MEASURED FROM POSITIVE X-AXIS
Z      = HEIGHT OF PLANE WHERE POINTS ARE LOCATED
RC     = RADIUS OF PARTICULAR CIRCLE WHERE POINTS ARE LOCATED
IDP    = IDENTIFICATION OF POINTS (0=MOVEAVBLE; 1=FIXED)
APEX   = APICES OF FACETS
NUM    = FACET NUMBER
CONST   = CONSTANT FOR FACET CALCULATIONS
CONST1  = CONSTANT FOR FACET CALCULATIONS
XCOF   = COLUMN MATRIX OF POLYNOMIAL COEFFICIENTS
COF    = WORKING VECTOR OF COEFFICIENTS
MORD   = ORDER OF POLYNOMIAL FOR SUBROUTINE CALL
ROOTR  = RESULTANT VECTOR OF REAL ROOTS OF POLYNOMIAL

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SUBROUTINE CHAMBER VARIABLES USED IN THIS PROGRAM

POINT = MATRIX OF THE THREE POINTS THAT MAKE UP EACH FACETS
FACET = FACET NUMBER
POINTS = TOTAL NUMBER OF POINTS USED TO DESCRIBE CHAMBER
FACETS = TOTAL NUMBER OF FACETS USED TO DESCRIBE CHAMBER
HOLD = VECTOR CONTAINING STATUS OF EACH POINT
(1 = CYLINDER HEAD; 0 = PISTON)
ZONE = VECTOR CONTAINING STATUS OF EACH FACET
(3 = CYLINDER WALL; 2 = PISTON; 1 = CYLINDER HEAD)
COORD = MATRIX OF X, Y, AND Z COORDINATES OF EACH POINT
VTOTAL = VECTOR OF TOTAL VOLUME AT EACH CRANK ANGLE
VIN = MATRIX OF VOLUME ENGULFED BY FLAME AT EACH CRANK ANGLE AND RADIUS STEP
VOUT = MATRIX OF VOLUME NOT ENGULFED BY THE FLAME AT EACH CRANK ANGLE AND RADIUS STEP
AOUTP = AREA OF PISTON
LIMIT = MATRIX OF LIMITING VALUES OF FACET COORDINATES
NHEAD = NUMBER OF FACETS ON HEAD OF CHAMBER
NPIST = NUMBER OF FACETS ON PISTON
NOWALL = NUMBER OF FACETS ON CYLINDER WALL
XSARK = X-AXIS LOCATION OF SPARK PLUG
YSARK = Y-AXIS LOCATION OF SPARK PLUG
ZSARK = Z-AXIS LOCATION OF SPARK PLUG
PSPBOT = LIMITING VALUE FOR PISTON TRAVEL
CONRL = CONNECTING ROD LENGTH FOR CHAMBER

SUBROUTINE AND FUNCTION IDENTIFICATION

XCONV = CONVERTS X-COORDINATE FROM CYLINDRICAL TO CARTESIAN
YCONV = CONVERTS Y-COORDINATE FROM CYLINDRICAL TO CARTESIAN
POLRT = SOLVE J HEIGHT OF HEMISPHERE POLYNOMIAL
CHAMBER = USES POINTS AND FACETS TO GENERATE CORRECTION FACTORS FOR FACET GENERATION (MODIFIED FROM S. Poulos CHAMBER PROGRAM)

TYPE DECLARATIONS AND STORAGE ALLOCATION

CHARACTER NAME*21,ANS*1,PLOC*5,VNUM*2
COMPLEX ROOTI(3)
INTEGER APEX(3),CONST,CONST1,CONST4
INTEGER POINT(400,3),FACET,POINTS,FACETS,HOLD(200),ZONE(400)
REAL COORD(200,3),VTOTAL(30),VIN(30,50),VOUT(30,50),AOUTP(30,50)
REAL LIMIT(2,3)
DIMENSION XCOF(4),COF(4),ROOTR(3),APISTR(2),VCLEAR(2),C(3)
DATA PI,NUM,CONST,CONST1,CONST4/3.141592654,0,16,15,15/
DATA CINT,CMAX,RADSTP,RNDPNT/14.0,70.0,24900,300.0/
DATA MINT,MAXRPT,ITS/200,2200,9/
DATA ICOUNT/0/
COMMON/DIMENS/BORE,STROKE,CONRL
COMMON/FACETS,NUM,FACET,POINTS,NHEAD,NPIST,NWALL,POINTS
COMMON/SPARK/XSARK,YSARK,ZSARK
COMMON/FPNTS/POINT
COMMON/VOLUMES/VTOTAL,VIN,VOUT
COMMON/AREAS/AOUTP
COMMON/CTRL/Caint, CAMAX, RADSTP, RNDPTN, MINRPT, MAXRPT, ITS
COMMON/COORD/COORD
COMMON/ZONE/ZONE
COMMON/FUXPNT/HOLD
COMMON/LIMITS/LIMIT, PISBOT
C(1)=1.0
C(2)=1.0
C(3)=1.0

READ INPUT DATA

READ(B,*)BORE,STROKE,CNRL,CR,SX,SY,DZ
READ(B,*)CAIN1,CAMAX,RADSTP,RNDPTN,MINRPT,MAXRPT,ITS

CALCULATE CHAMBER GEOMETRY

DVol=PI*(BORE**2)*STROKE/4.0
VCL=DVol/(CR-1)
VCLEAR(1)=VCL
VCLEAR(2)=VCL
APIS(1)=PI*BORE*BORE/4.0
RC1=BORE/2.0

FIND HEMISPHERE HEIGHT FROM VCL=(PI/6)[3*(BORE/2)*H+(H*3)]

XCOF(4)=1.0
XCOF(3)=0.0
XCOF(2)=3.0*(BORE/2.0)**2
XCOF(1)=-(6.0*VCL)/PI
MORD=3
CALL POLRT(XCOF,COF,MORD,ROOTR,ROOTI,IER)
H=ROOTR(1)
H0=H

FIND SPHERE RADIUS FROM VCL=(PI*(H**2)*(3*R-H))/3

R=((3.0*VCLEAR(2))/(PI*(H**2)+(H))/3.0

EXPAND HEIGHT TO ALLOW FOR FACETS

H0=C(3)
write(10,*) 'h=',h
write(10,*) 'r=',r
write(10,*) 'vCLEAR(2)=',vCLEAR(2)

RESET CONSTANTS

CONST=16
CONST1=15
CONST4=15

DEFINE POINTS

POINTS AT BASE ON HEAD

IDP=1
DELTH=2.0*PI/16.0
TH=0.0
Z=0.9

ADJUST FOR FACET SPACE
RC1=RC1*(SORT(C(2)))
write(10,*)'rc1 = ',rc1
DO 630 K=1,16
   IF (K.EQ.1) THEN
      TH=PI/16.00
   ELSE
      TH=TH+DETH
   ENDIF
   X=XCONV(RC1,TH)
   Y=YCONV(RC1,TH)
   COORD(K,1)=X
   COORD(K,2)=Y
   COORD(K,3)=Z
   HOLD(K)=IDP
630 CONTINUE

POINTS AT BASE ON CYLINDER WALL

IDP=0
TH=0.0
DO 640 K=17,32
   IF (K.EQ.17) THEN
      TH=PI/16.00
   ELSE
      TH=TH+DETH
   ENDIF
   X=XCONV(RC1,TH)
   Y=YCONV(RC1,TH)
   COORD(K,1)=X
   COORD(K,2)=Y
   COORD(K,3)=Z
   HOLD(K)=IDP
640 CONTINUE

POINTS AT FIRST LEVEL ON HEAD

Z=0.5445305+H
IDP=1
TH=0.0
HH=H+Z
VCL=(1.0/3.0)*PI*(HH+2)*((3*R)-HH)
RC=SQRT(((6+VCL)/(PI+HH))-((HH+2))/3.0)

ADJUST FOR FACET SPACE

DO 650 K=33,48
   IF (K.EQ.33) THEN
      TH=0.0
   ELSE
      TH=TH+DETH
   ENDIF
   X=XCONV(RC,TH)
   Y=YCONV(RC,TH)
   COORD(K,1)=X
   COORD(K,2)=Y
   COORD(K,3)=Z
   HOLD(K)=IDP
650 CONTINUE

POINTS AT SECOND LEVEL ON HEAD

DETH=2.0*PI/8.0
Z=0.88693+H
IDP=1
TH=0.0
HH=H+Z
VCL=(1.0/3.0)*PI*(HH+2)*((3*R)-HH)
RC=SQR((((6*VCL)/(PI*HH))-(HH**2))/3.0)

ADJUST FOR FACET SPACE

DO 659 K=49,56
   IF (K.EQ.49) THEN
      TH=0.0
   ELSE
      TH=TH+DELTH
   ENDIF
   X=XCNV(RC,TH)
   Y=YCNV(RC,TH)
   COORD(K,1)=X
   COORD(K,2)=Y
   COORD(K,3)=Z
   HOLD(K)=IDP
   659 CONTINUE

POINT AT CENTER OF HEAD AND PISTON

K=57
X=0.0
Y=0.0
Z=H
IDP=1
COORD(K,1)=X
COORD(K,2)=Y
COORD(K,3)=Z
HOLD(K)=IDP
K=58
Z=0.0
IDP=0
COORD(K,1)=X
COORD(K,2)=Y
COORD(K,3)=Z
HOLD(K)=IDP

FACET INFORMATION

CYLINDER WALL FACETS (STRETCHABLE)

IDF=3
DO 730 N=1,16
   NUM=NUM+1
   APEX(1)=N+16
   APEX(2)=N
   APEX(3)=N+1
   IF (APEX(3).EQ.17) APEX(3)=1
   POINT(NUM,1)=APEX(1)
   POINT(NUM,2)=APEX(2)
   POINT(NUM,3)=APEX(3)
   ZONE(NUM)=IDF
   730 CONTINUE

DO 740 N=17,32
   NUM=NUM+1
   APEX(1)=N
   APEX(2)=N+1
   IF (APEX(2).EQ.33) APEX(2)=17
   APEX(3)=APEX(2)+16
   POINT(NUM,1)=APEX(1)
   POINT(NUM,2)=APEX(2)
   POINT(NUM,3)=APEX(3)
   ZONE(NUM)=IDF
   740 CONTINUE
FACETS ON HEAD (BASE TO FIRST LEVEL)

IDF=1
DO 745 N=33,48
   NUM=NUM+1
   APEX(1)=N
   APEX(2)=N-32
   APEX(3)=N-32+1
   IF (APEX(3).EQ.17) APEX(3)=1
   POINT(NUM,1)=APEX(1)
   POINT(NUM,2)=APEX(2)
   POINT(NUM,3)=APEX(3)
   ZONE(NUM)=IDF
745 CONTINUE
DO 750 N=33,48
   NUM=NUM+1
   APEX(1)=N
   APEX(2)=N+1
   IF (APEX(2).EQ.49) APEX(2)=33
   APEX(3)=APEX(2)-32
   POINT(NUM,1)=APEX(1)
   POINT(NUM,2)=APEX(2)
   POINT(NUM,3)=APEX(3)
   ZONE(NUM)=IDF
750 CONTINUE

FACETS ON HEAD (LEVEL 1 TO LEVEL 2)

DO 759 N=49,56
   NUM=NUM+1
   APEX(1)=N
   APEX(2)=N+1
   IF (APEX(2).EQ.57) APEX(2)=49
   APEX(3)=APEX(2)+CONST
   IF (APEX(2).EQ.49) APEX(3)=48
   CONST=CONST-1
   POINT(NUM,1)=APEX(1)
   POINT(NUM,2)=APEX(2)
   POINT(NUM,3)=APEX(3)
   ZONE(NUM)=IDF
759 CONTINUE
DO 760 N=34,48,2
   NUM=NUM+1
   APEX(1)=N
   APEX(2)=N+1
   IF (APEX(2).EQ.49) APEX(2)=33
   APEX(3)=APEX(2)+CONST1
   IF (APEX(3).EQ.41) APEX(3)=49
   CONST1=CONST1-1
   POINT(NUM,1)=APEX(1)
   POINT(NUM,2)=APEX(2)
   POINT(NUM,3)=APEX(3)
   ZONE(NUM)=IDF
760 CONTINUE
DO 763 N=33,47,2
   NUM=NUM+1
   APEX(1)=N
   APEX(2)=N+1
   APEX(3)=APEX(2)+CONST4
   CONST4=CONST4-1
   POINT(NUM,1)=APEX(1)
   POINT(NUM,2)=APEX(2)
   POINT(NUM,3)=APEX(3)
   ZONE(NUM)=IDF
763 CONTINUE

FACETS ON HEAD (LEVEL 2 TO LEVEL 3)
**APEX(1)=57
DO 785 N=9,56
NUM=NUM+1
APEX(2)=N
APEX(3)=N+1
IF (APEX(3).EQ.57) APEX(3)=49
POINT(NUM,1)=APEX(1)
POINT(NUM,2)=APEX(2)
POINT(NUM,3)=APEX(3)
ZONE(NUM)=IDF
785 CONTINUE

**FACETS ON PISTON SURFACE

**APEX(1)=58
IDF=2
DO 790 N=17,32
NUM=NUM+1
APEX(2)=N
APEX(3)=N+1
IF (APEX(3).EQ.33) APEX(3)=17
POINT(NUM,1)=APEX(1)
POINT(NUM,2)=APEX(2)
POINT(NUM,3)=APEX(3)
ZONE(NUM)=IDF
790 CONTINUE

*******************************************************************************

*************** ACTUAL CLEARANCE VOLUME & PISTON AREA ***************

********** PREPARE FOR CALL TO CHAMBER SUBROUTINE **********

POINTS=58
FACETS=112
NHEAD=64
NPIST=16
NCWALL=32
CONRL=CNRL
XSPARK=0.0
YSFARK=0.0
ZSPARK=0.0

SET LIMITING VALUES

LIMIT(1,1)=999999.
LIMIT(2,1)=999999.
LIMIT(1,2)=999999.
LIMIT(2,2)=999999.
LIMIT(1,3)=999999.
PISBOLT=999999.

DO 870 M=1,112
IF (ZONE(M).GT.4) GOTO 880
IF (ZONE(M).LT.3) GOTO 830
GOTO 870
830 IF (ZONE(M).EQ.2) GOTO 850
DO 840 N=1,3
K=POINT(M,N)
IF (COORD(K,3).LT.LIMIT(2,3)) LIMIT(2,3)=COORD(K,3)
IF (COORD(K,1).GT.LIMIT(2,1)) LIMIT(2,1)=COORD(K,1)
IF (COORD(K,1).LT.LIMIT(1,1)) LIMIT(1,1)=COORD(K,1)
IF (COORD(K,2).GT.LIMIT(2,2)) LIMIT(2,2)=COORD(K,2)
IF (COORD(K,2).LT.LIMIT(1,2)) LIMIT(1,2)=COORD(K,2)
840 CONTINUE
GOTO 870
850 DO 880 N=1,3
L=POINT(M,N)
880 Continue

IF (COORD(L,3).GT.LIMIT(L,3)) LIMIT(L,3)=COORD(L,3)
IF (COORD(L,3).LT.PISBOT) PISBOT=COORD(L,3)
860 CONTINUE
870 CONTINUE

* CALL CHAMBER SUBROUTINE *

880 CALL CHAMBER

* PROCESS SUBROUTINE OUTPUT *

VCLEAR(2)=VOUT(1,1)
APIST(2)=AOUTP(1,1)
WRITE(10,'(A10)')'VOUT(1,1)='',VOUT(1,1)
WRITE(10,'(A10)')'VTOTAL(1)='',VTOTAL(1)
WRITE(10,'(A10)')'APIST(1)='',APIST(1)
WRITE(10,'(A10)')'VCLEAR(1)='',VCLEAR(1)
WRITE(10,'(A10)')'APIST(2)='',APIST(2)
WRITE(10,'(A10)')'VCLEAR(2)='',VCLEAR(2)

* CALCULATE NEW CONSTANTS AND TEST *

IF (((C(1).GT.1.001).OR.(C(1).LT.0.999)).OR.(ICOUNT.EQ.0)) THEN
  C(1)=VCLEAR(1)/VCLEAR(2)
ENDIF
IF (((C(2).GT.1.001).OR.(C(2).LT.0.999)).OR.(ICOUNT.EQ.0)) THEN
  C(2)=APIST(1)/APIST(2)
ENDIF
C(3)=C(1)/C(2)
WRITE(10,'(A10)')'C(1)=',C(1)
WRITE(10,'(A10)')'C(2)=',C(2)
WRITE(10,'(A10)')'C(3)=',C(3)
ICOUNT=ICOUNT+1
IF (ICOUNT.EQ.10) THEN
  WRITE(10,'(A10)')'NO CONVERSION'
goto 900
ENDIF
IF (((C(2).GT.1.001).OR.(C(2).LT.0.999)).OR.(C(1).GT.1.001).
  1 OR.(C(1).LT.0.999)) GOTO 800

***********************************************************************

* FIND REAL SPARK LOCATION BLOCK 0100 *

900 RSP=SORT((SX**2)+(SY**2))
  ZHR=SORT((R**2)-(RSP**2))
  SZ=H0-ZH
WRITE(10,'(A10)')'RSP=',RSP
WRITE(10,'(A10)')'R=',R
WRITE(10,'(A10)')'ZH=',ZH
WRITE(10,'(A10)')'H0=',H0
WRITE(10,'(A10)')'SZ=',SZ
WRITE(10,'(A10)')'SX=',SX
WRITE(10,'(A10)')'SY=',SY
WRITE(10,'(A10)')'DZ=',DZ

* ADJUST FOR VERTICAL OFFSET *

SZ=SZ-DZ

***********************************************************************

* STORE INITIAL DATA BLOCK 0500 *

WRITE(9,505)
505 FORMAT(' C')
WRITE(9,507)
507 FORMAT(' C',3X,'BASIC ENGINE DIMENSIONS AND SPARK LOCATION:')

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WRITE(9,505)
WRITE(9,510)
510 FORMAT('C',3X,'BORE, STROKE, CONNECTING-ROD LENGTH,' 1 'XSPARK, YSPARK, & ZSPARK:')
WRITE(9,505)
WRITE(9,525)
525 FORMAT('C',1X,'*',5(8X,'**'))
WRITE(9,530)BORE,STROKE,CNRL,SX,SY,SZ
530 FORMAT(2X,6(F10.5))
WRITE(9,505)
WRITE(9,535)
535 FORMAT('C',3X,'CRANK ANGLE INTERVAL; MAXIMUM CRANK ANGLE; RADIUS ' 1 'STEP SIZE;'/' C',3X,'RANDOM POINT DENSITY; MINIMUM # OF RANDOM ' 2 'POINTS; MAXIMUM #;'/' C',3X,'OF RANDOM POINTS; # OF ITERATIONS IN ' 3 'AFLAME CALCULATIONS:')
WRITE(9,540)
540 FORMAT('C',47X,3('RJ->',4X))
WRITE(9,542)
542 FORMAT('C',1X,'*',4(8X,'**').6X,2('**',6X),'**')
WRITE(9,545)CAINT,CAVG,RADSTP,RNDPNT,MINPNT,MAXPNT,ITS
545 FORMAT(2X,4(F10.3),3(I8))

PRINT OUTPUT FILE

WRITE(9,505)
WRITE(9,910)
910 FORMAT('C',3X,'POINTS:')
WRITE(9,915)
915 FORMAT('C',3X,'RJ->',35X,'RJ->')
WRITE(9,920)
920 FORMAT('C',1X,'*',3X,'**'.3(10X,'**').3X,'*')
DO 940 K=1,POINTS
WRITE(9,930)K,COORD(K,1),COORD(K,2),COORD(K,3),HOLD(K)
930 FORMAT(2X,15,3(F12.3),I5)
940 CONTINUE

WRITE STOP POINT

K=59
IDP=2
X=0
Y=0
Z=0
WRITE(9,930)K,X,Y,Z,IDP

FACET INFORMATION

WRITE(9,505)
WRITE(9,950)
950 FORMAT('C',3X,'FACETS:')
WRITE(9,955)
955 FORMAT('C',1X,'*',5(3X,'**'))
DO 970 K=1,FACETS
WRITE(9,960)K,POINT(K,1),POINT(K,2),POINT(K,3),ZONE(K)
960 FORMAT(2X,5(I8))
970 CONTINUE

WRITE SIGNAL FACET

NUM=113
APEX(1)=1
APEX(2)=1
APEX(3)=1
IDF=4
WRITE(9,985)NUM,APEX(1),APEX(2),APEX(3),IDF

-146-
FUNCTION XCONV(R,TH)

FUNCTION XCONV

This function converts from polar coordinates to the x-
dimension of Cartesian coordinates.

VARIABLE IDENTIFICATION

XCONV = value in Cartesian coordinates
R = radial value in polar coordinates
TH = angular value in polar coordinates
(measured from positive x-axis)

FUNCTION CALCULATION

XCONV = R \cdot \cos(TH)

RETURN
END

FUNCTION YCONV(R,TH)

FUNCTION XCONV

This function converts from polar coordinates to the x-
dimension of Cartesian coordinates.

VARIABLE IDENTIFICATION

XCONV = value in Cartesian coordinates
R = radial value in polar coordinates
TH = angular value in polar coordinates
(measured from positive x-axis)

FUNCTION CALCULATION

YCONV = R \cdot \sin(TH)

RETURN
END

SUBROUTINE POLRT

PURPOSE
Computes the real and complex roots of a real polynomial

USAGE
CALL POLRT(XCOF,COF,M,ROOTR,ROOTI,IER)
DESCRIPTION OF PARAMETERS

XCOF - VECTOR OF N+1 COEFFICIENTS OF THE POLYNOMIAL
ORDERED FROM SMALLEST TO LARGEST POWER

COF - WORKING VECTOR OF LENGTH N+1
M - ORDER OF POLYNOMIAL
ROOTR - RESULTANT VECTOR OF LENGTH M CONTAINING REAL ROOTS
OF THE POLYNOMIAL
ROOTI - RESULTANT VECTOR OF LENGTH M CONTAINING THE
CORRESPONDING IMAGINARY ROOTS OF THE POLYNOMIAL
IER - ERROR CODE WHERE
IER=0 NO ERROR
IER=1 M LESS THAN ONE
IER=2 M GREATER THAN 36
IER=3 UNABLE TO DETERMINE ROOT WITH 500 INTERATIONS
ON 5 STARTING VALUES
IER=4 HIGH ORDER COEFFICIENT IS ZERO

REMARKS
LIMITED TO 36TH ORDER POLYNOMIAL OR LESS.
FLOATING POINT OVERFLOW MAY OCCUR FOR HIGH ORDER
POLYNOMIALS BUT WILL NOT AFFECT THE ACCURACY OF THE RESULTS.

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED
NONE

METHOD
NEWTON-RAPHSON ITERATIVE TECHNIQUE. THE FINAL ITERATIONS
ON EACH ROOT ARE PERFORMED USING THE ORIGINAL POLYNOMIAL
RATHER THAN THE REDUCED POLYNOMIAL TO AVOID ACCUMULATED
ERRORS IN THE REDUCED POLYNOMIAL.

SUBROUTINE POLRT(XCOF,COF,M,ROOTR,ROOTI,IER)
DIMENSION XCOF(1),COF(1),ROOTR(1),ROOTI(1)
DOUBLE PRECISION X0,Y0,X,Y,XPR,YPR,UX,UY,V,XT,UT,XT2,UT2,SUMSQ,
1 DX,DY,TEMP,ALPHA,DABS

IF A DOUBLE PRECISION VERSION OF THIS ROUTINE IS DESIRED, THE
C IN COLUMN 1 SHOULD BE REMOVED FROM THE DOUBLE PRECISION
STATEMENT WHICH FOLLOWS.

DOUBLE PRECISION XCOF,COF,ROOTR,ROOTI

THE C MUST ALSO BE REMOVED FROM DOUBLE PRECISION STATEMENTS
APPEARING IN OTHER ROUTINES USED IN CONJUNCTION WITH THIS
ROUTINE.
THE DOUBLE PRECISION VERSION MAY BE MODIFIED BY CHANGING THE
CONSTANT IN STATEMENT 78 TO 1.0D-12 AND IN STATEMENT 122 TO
1.0D-10. THIS WILL PROVIDE HIGHER PRECISION RESULTS AT THE
COST OF EXECUTION TIME

IFI=0
IFI=M
IER=0
IF(XCOF(N+1))10,25,10
10 IF(N) 15,15,32
C
SET ERROR CODE TO 1
C
15 IER=1
20 RETURN
C SET ERROR CODE TO 4

25 IER=4
   GO TO 20
C
C SET ERROR CODE TO 2

30 IER=2
   GO TO 20
32 IF(N-36) 35,35,30
35 NX=N
   NXX=N+1
   N2=1
   KJ1 = N+1
   DO 40 L=1,KJ1
   MT=KJ1-L+1
40 COF(MT)=XCOF(L)
C
C SET INITIAL VALUES

45 XO=.00500101
   YO=0.01000101
C
C ZERO INITIAL VALUE COUNTER

IN=0
50 X=XO
C
C INCREMENT INITIAL VALUES AND COUNTER

XO=10.0+YO
   YO=10.0+X
C
C SET X AND Y TO CURRENT VALUE

X=XO
   Y=YO
   IN=IN+1
   GO TO 59
55 IFIT=1
   XPR=X
   YPR=Y
C
C EVALUATE POLYNOMIAL AND DERIVATIVES

59 ICT=0
60 UX=0.0
   UY=0.0
   V = 0.0
   YT=0.0
   XT=1.0
   U=COF(N+1)
   IF(U) 65,130,65
65 DO 70 I=1,N
   L =N-I+1
   TEMP=COF(L)
   XT2=X*XT--YT*YT
   YT2=X*YT+Y*XT
   U=U+TEMP*XT2
   V=V+TEMP*YT2
   FI=I
   UX=UX+FI*XT*TEMP
   UY=UY-FI*YT*TEMP
   XT=XT2
70 YT=YT2
   SUMSQ=UX+UX+UY+UY
   IF(SUMSQ) 75,110,75
75 DX=(V*UY-U*UX)/SUMSQ
X=X+DX
DY=(U*UY+V*UX)/SUMSQ
Y=Y+DY

78 IF(DABS(DY)+DABS(DX)-1.0D-05) 100,80,80

C
C     STEP ITERATION COUNTER
C
80 ICT=ICT+1
     IF(ICT-500) 80,85,85
85 IF(IFIT)=100,90,100
90 IF(IN=5) 50,95,95

C
C     SET ERROR CODE TO 3
C
95 IER=3
     GO TO 20
100 DO 105 L=1,NXX
     MT=KJ1-L+1
     TEMP=XCOF(MT)
     XCOF(MT)=COF(L)
105 COF(L)=TEMP
     ITEMP=N
     N=N+1
     NX=ITEMP
     IF(IFIT) 120,55,120
110 IF(IFIT) 115,50,115
115 X=XPR
     Y=YPR
120 IFIT=0
122 IF(DABS(Y)-1.0D-4*DABS(X)) 135,125,125
125 ALPHA=X+X
     SUMSQ=X*X+Y*Y
     N=N-2
     GO TO 140
130 X=0.0
     NX=NX-1
     NXX=NXX-1
135 Y=0.0
     SUMSQ=0.0
     ALPHA=X
     N=N-1
140 COF(2)=COF(2)+ALPHA*COF(1)
145 DO 150 L=2,N
150 COF(L+1)=COF(L+1)+ALPHA*COF(L)-SUMSQ*COF(L-1)
155 ROOT(N2)=Y
     ROOTR(N2)=X
     N2=N2+1
     IF(SUMSQ) 160,165,160
160 Y= Y
     SUMSQ=0.0
     GO TO 155
165 IF(N) 20,20,45
     END

C
C
C
C
C
C
C
C
C
C

MODEL FOR THE GEOMETRIC INTERACTION

BETWEEN A SPHERICAL PROPAGATING FLAME AND

A COMBUSTION CHAMBER OF ARBITRARY SHAPE

-150-
BY

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VERSION 1.0 4/26/82

PROGRAM CHAMBER

PURPOSE

THIS PROGRAM CALCULATES THE GEOMETRY OF A SPHERICAL FLAME
WITH A FIXED CENTER AS IT PROPAGATES THROUGH THE COMBUSTION
CHAMBER OF A RECIPROCATING ENGINE. THE CHAMBER MAY HAVE
ANY ARBITRARY SHAPE, AND THE CALCULATION TAKES INTO
ACCOUNT THE EFFECT OF PISTON MOTION ON THE INSTANTANEOUS
SHAPE OF THE CHAMBER. GIVEN A DETAILED DESCRIPTION OF
THE CHAMBER SHAPE, THE CRANK ANGLE (WHICH TOGETHER WITH
THE ENGINE STROKE AND CONNECTING ROD LENGTH DETERMINES THE
PISTON POSITION), AND A FLAME RADIUS, THE PROGRAM CALCULATES
THE AREAS OF THE CYLINDER HEAD, PISTON, AND CYLINDER WALL ON
EACH SIDE OF THE FLAME, AND THE FLAME SURFACE.

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>OFLAME</td>
<td>NO</td>
<td>YES</td>
<td>BASE FLAME AREA ABOVE HIGHEST POINT</td>
</tr>
<tr>
<td>AFLAME</td>
<td>NO</td>
<td>YES</td>
<td>ON PISTON @ TDC</td>
</tr>
<tr>
<td>AINH</td>
<td>NO</td>
<td>YES</td>
<td>AREA OF HEAD INSIDE FLAME</td>
</tr>
<tr>
<td>AOUTH</td>
<td>NO</td>
<td>YES</td>
<td>AREA OF HEAD OUTSIDE FLAME</td>
</tr>
<tr>
<td>AINP</td>
<td>NO</td>
<td>YES</td>
<td>AREA OF PISTON INSIDE FLAME</td>
</tr>
<tr>
<td>AOUTP</td>
<td>NO</td>
<td>YES</td>
<td>AREA OF PISTON OUTSIDE FLAME</td>
</tr>
<tr>
<td>AINOW</td>
<td>NO</td>
<td>YES</td>
<td>AREA OF CYL. WALL INSIDE FLAME</td>
</tr>
</tbody>
</table>

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AOUTCW NO YES AREA OF CYL. WALL OUTSIDE FLAME
VIN NO YES VOLUME INSIDE THE FLAME
VOUT NO YES VOLUME OUTSIDE THE FLAME
VTOTAL NO YES TOTAL CHAMBER VOLUME

REMARKS
NONE

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED
INPUT ROUIT FLAME CHANGE
WALL REPORT

METHOD
THE SURFACE OF THE COMBUSTION CHAMBER IS APPROXIMATED BY A
_LARGE NUMBER OF FLAT TRIANGULAR FACETS. THE FACETS WHICH LIE
ON THE CYLINDER WALL ARE "STRETCHED" AS THE PISTON MOVES AND
THOSE ON THE PISTON CROWN TRANSLATE WITH THE PISTON. AT EACH
CRANK ANGLE, SPHERES WITH INCREMENTAL RADII ARE MADE TO INTER-
SECT THE INSTANTANEOUS SURFACE OF THE CHAMBER. A LARGE NUMBER
OF POINTS ARE DISTRIBUTED RANDOMLY ON THE SURFACE OF EACH
SPHERE AND EACH POINT IS TESTED TO SEE IF IT LIES WITHIN THE
CHAMBER. THE FRACTION OF POINTS ON THE TEST SPHERE WHICH ARE
INSIDE THE CHAMBER TIMES THE SURFACE AREA OF THE TEST SPHERE
IS THE TOTAL FLAME AREA. THE SURFACES OF THE 3 ZONES OF THE
CHAMBER (HEAD, PISTON, CYLINDER WALL) ON EACH SIDE OF THE
FLAME ARE FOUND BY IDENTIFYING WHICH FACETS LIE ON EACH SIDE
OF THE FLAME AND ADDING UP THEIR AREAS TO GET THE TOTALS.

WRITTEN BY S. G. Poulos
EDITED BY S. G. Poulos
EDITED FOR USE AS SUBROUTINE IN PROGRAM GEOMGEN
BY ROBERT T. CHANG 19 FEBRUARY 1986

SUBROUTINE CHAMBER

INTEGER SPNTS, SPMIN, SPMAX, OLDTHT, OLDRAD, ACOUNT, BCOUNT,
& COUNT
REAL LIMIT(2,3), OFLAME(50), AINH(30,50), AINP(30,50),
& AINCW(30,50), AOUTH(30,50), AOUTP(30,50), AOUTCW(30,50),
& AFRAME(30,50), VIN(30,50), VTOTAL(30), VOUT(30,50), MTHETA
REAL ZLMT(2,3), ZINH(30,50), ZINP(30,50),
& ZINCW(30,50), ZOUTH(30,50), ZOUTP(30,50), ZOUTCW(30,50),
& ZFRAME(30,50), ZIN(30,50), ZOUT(30,50)
COMMON/DIMENS/ BORE, STROKE, CONRL
COMMON/LIMIT/ LIMIT, PISBOT
COMMON/ACNT/ ACOUNT
COMMON/BCNT/ BCOUNT
COMMON/CCNT/ CCOUNT
COMMON/VOLUMES/ VTOTAL, VIN, VOUT
COMMON/CNTROL/ DTHETA, MTHETA, RSTEPS, PDEST, SPMIN,
& SPMAX, MAXITS
COMMON/RSEED/ IX
COMMON/AREAS/AOUTP

READ IN CHAMBER GEOMETRY AND PARAMETERS FOR CONTROLLING
CALCULATION. INPUTS ARE TRANSFERRED THROUGH COMMON BLOCKS.

INITIALIZE TO 0.0 ALL VOLUMES AND AREAS TO BE CALCULATED

ITHETA = 1
VTOTAL(ITHETA) = 0.0
DO 4 IRAD = 1, 50
  OFRAME(IRAD) = 0.0
  AFRAME(ITHETA, IRAD) = 0.0
  AINH(ITHETA, IRAD) = 0.0
  AOUTH(ITHETA, IRAD) = 0.0
4 CONTINUE
AINP(ITHETA,IRAD) = 0.0
AOUTP(ITHETA,IRAD) = 0.0
AINCW(ITHETA,IRAD) = 0.0
AOUTCW(ITHETA,IRAD) = 0.0
VIN(ITHETA,IRAD) = 0.0
VOUT(ITHETA,IRAD) = 0.0

4 CONTINUE

C CALCULATE MAXIMUM FLAME RADIUS FOR PISTON AT BDC

C ACCOUNT = 0

C SET INITIAL RANDOM NUMBER SEED FOR SUBROUTINE 'FLAME'

IX = 11261949
CALL RQUT (RMAX)

C CALCULATE RADIUS INCREMENT AND NORMALIZE ENGINE DIMENSIONS

DELTAR = RMAX/RSTEPS
STBRAT = STROKE/BORE
CTBRAT = CONRL/BORE
DTBRAT = DELTAR/BORE

C CALCULATE FLAME AREAS FOR BASE CASE: PISTON AT TDC.
C REGION OF INTEREST FOR BASE CASE IS BETWEEN THE HIGHEST
C POINT IN THE CHAMBER AND THE HORIZONTAL PLANE PASSING
C THROUGH THE HIGHEST POINT ON THE PISTON AT TDC.

RADIUS = 0.0

DO 20 IRAD = 1, 50
   IF (IRAD .EQ. 1) GO TO 20
   RADIUS = RADIUS + DELTAR

20 CONTINUE

C CALCULATE NUMBER OF POINTS TO GENERATE ON FLAME SPHERE

SPNTS = INT( 16. * PDENST * (RADIUS/BORE)**2. )
IF (SPNTS .LT. SPMIN) SPNTS = SPMIN
IF (SPNTS .GT. SPMAX) SPNTS = SPMAX

C CALL FLAME (RADIUS, SPNTS, FSURF, STADEV)

OFLAME(IRAD) = FSURF
IF (((IRAD .GT. 1) .AND. (OFLAME(IRAD) .LE. 0.00001)) GO TO 30

20 CONTINUE

C BEGIN MAIN CALCULATION LOOP FOR INCREASING VALUES OF THETA.
C FOR EACH INCREASING VALUE OF THETA, INCREASE RADIUS IN STEPS
C OF DELTAR.

30 LIMIT(2,3) = LIMIT(1,3)
LIMIT(1,3) = PISBTO
OLDTHT = 0.0
THETA = 0.0
BCOUNT = 0

C ITHETA = 1

40 CALL RQUT (RMAX)
OLDTHT = ITHETA - 1
RADIUS = 0.0
CCOUNT = 0

C DO 70 IRAD = 1, 50
MAXRAD = IRAD

C CALCULATE CHAMBER SURFACE AREAS ON EACH SIDE OF THE
CALL FLAME (ITHETA, IRAD, RADIUS, AINH, AOUTH, AINP, AOUTP, AINCW, AOUTCW)

IF (OLDHT .LT. 1) GO TO 50

IF FLAME HAS NOT IMPINGED ON PISTON AT THIS RADIUS,
FLAME AREA = FLAME AREA FROM PREVIOUS CRANK ANGLE.

IF (((AINP(OLDHT, IRAD)/BORE**2) .GT. 0.00001) GO TO 50
AFLAME(ITHETA, IRAD) = AFLAME(OLDHT, IRAD)
GO TO 60

IF (IRAD .EQ. 1) GO TO 60
SPNTS = INT((16. - PDENST * (RADIUS/BORE)**2.)
IF (SPNTS .LT. SPMIN) SPNTS = SPMIN
IF (SPNTS .GT. SPMAX) SPNTS = SPMAX

CALL FLAME (RADIUS, SPNTS, FSURF, STADEV)

TOTAL AREA = BASE AREA + FSURF

AFLAME(ITHETA, IRAD) = OFLAME(IRAD) + FSURF

OLDRAD = IRAD - 1
OLDFLM = 0.0
OLDVIN = 0.0
IF (OLDRAD .GE. 1) OLDFLM = AFLAME(ITHETA, OLDRAD)
IF (OLDRAD .GE. 1) OLDVIN = VIN(ITHETA, OLDRAD)

CALCULATE INCREMENTAL VOLUME DUE TO DELTAR

IF ((IRAD .GT. 1) .AND.
& (AFLAME(ITHETA, IRAD) .LE. 0.00001)) GO TO 80
DELV = .5 * (AFLAME(ITHETA, IRAD) + OLDFLM) * DELTAR
VIN(ITHETA, IRAD) = OLDVIN + DELV
RADIUS = RADIUS + DELTAR
CONTINUE

USE REDUCED DELTAR FOR LAST VOLUME INCREMENT

DELV = .5 * (AFLAME(ITHETA, MAXRAD) + OLDFLM) * 
& (MAX - FLOAT(MAXRAD - 2) * DELTAR)
VIN(ITHETA, MAXRAD) = OLDVIN + DELV
VTOTAL(ITHETA) = VIN(ITHETA, MAXRAD)
CMRTIO = (VTOTAL(1) + 3.1415926 * BORE * BORE * STROKE/4.) / VTOTAL(1)

DO 90 IRAD = 1, MAXRAD
VOUT(ITHETA, IRAD) = VTOTAL(ITHETA) - VIN(ITHETA, IRAD)
CONTINUE

COMPLETE SOLUTION MATRIX FOR REMAINING RADII

MXRD1 = MAXRAD + 1
DO 91 IRAD = MXRD1, 50
AINH(ITHETA, IRAD) = AINH(ITHETA, MAXRAD)
AOUTH(ITHETA, IRAD) = AOUTH(ITHETA, MAXRAD)
AINP(ITHETA, IRAD) = AINP(ITHETA, MAXRAD)
AOUTP(ITHETA, IRAD) = AOUTP(ITHETA, MAXRAD)
AINCW(ITHETA, IRAD) = AINCW(ITHETA, MAXRAD)
AOUTCW(ITHETA, IRAD) = AOUTCW(ITHETA, MAXRAD)
VIN(ITHETA, IRAD) = VIN(ITHETA, MAXRAD)
VOUT(ITHETA, IRAD) = VOUT(ITHETA, MAXRAD)
CONTINUE

RETURN
END

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SUBROUTINE FLAME

PURPOSE
THIS SUBROUTINE CALCULATES THE INTERSECTION BETWEEN A
GENERAL CHAMBER SHAPE AND A SPHERE OF GIVEN RADIUS. THE
SURFACE OF THE SPHERE WITHIN THE CHAMBER IS FOUND BY
GENERATING RANDOM POINTS ON THE SURFACE OF THE SPHERE
AND THEN TESTING THE POINTS TO SEE IF THEY LIE INSIDE THE
CHAMBER. THE INTERIOR SPHERE SURFACE IS THEN EQUAL TO THE
FRACTION OF POINTS FOUND INSIDE THE CHAMBER TIMES THE
TOTAL SURFACE AREA OF A SPHERE OF THE GIVEN RADIUS.

USAGE
CALL FLAME (RADIUS, SPNTS, FSURF, STADEV)

DESCRIPTION OF PARAMETERS
PARAMETER INPUT OUTPUT DESCRIPTION
RADIUS YES NO INSTANTANEOUS SPHERE RADIUS
SPNTS YES NO NUMBER OF RANDOM POINTS TO BE GENER-
ATED ON THE SURFACE OF THE SPHERE
FSURF NO YES FLAME SURFACE
STADEV NO YES STANDARD DEVIATION OF ITERATIONS
ABOUT MEAN (MEAN IS FSURF)

REMARKS
NONE

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED
INTERS

METHOD
ALGORITHM FOR GENERATING RANDOM POINTS ON THE SURFACE OF
A SPHERE IS TAKEN FROM: MARSAGLIA, GEORGE, "CHOOSING A
POINT FROM THE SURFACE OF A SPHERE", THE ANNALS OF MA-
THEMATICAL STATISTICS, 1972, VOL. 43, NO. 2, 645-646.

WRITTEN BY S. G. POULOS
EDITED BY S. G. POULOS

SUBROUTINE FLAME (RADIUS, SPNTS, FSURF, STADEV)

INTEGER SPNTS, FACET, FACETS, A, B, C, POINT(400,3),
& PNTA, PNTB, PNTC, AA, BB, CC, POINTS
REAL COORD(200,3), LIMIT(2,3)
COMMON/DIMENS/ BORE, STROKE, CONRL
COMMON/SPARK/ XSPARK, YSPARK, ZSPARK
COMMON/QUANTS/ FACETS, NHEAD, NPIST, NNOWALL, POINTS
COMMON/CONTROL/ DTHETA, MTHETA, RSTEPS, PDENST, SPMIN,
& SMAX, MAXITS
COMMON/LIMITS/ LIMIT, PISBOT
COMMON/FPNTS/ POINT
COMMON/COORDI/ COORD
COMMON/RSEED/ IX

ISTAT=LIB$INT_OVER(0) ! Disable integer overflow

SUM = 0.0
SUMSQ = 0.0

DO 50 ITER = 1, MAXITS

ITERS = ITER
NCOUNT = 0
INPTS = 0

DO 40 IPOINT = 1, SPNTS

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GENERATE A SINGLE RANDOM POINT

10
IX = IX*65539
IF (IX .LT. 0) IX = IX + 2147483647 + 1
XR = IX
V1 = 1. - XR*0.9313224E-9

IX = IX*65539
IF (IX .LT. 0) IX = IX + 2147483647 + 1
XR = IX
V2 = 1. - XR*0.9313224E-9

S = (V1*V1) + (V2*V2)
IF (S .GE. 1.0) GO TO 10
R = 2.*SORT(1. - S)

ZTEST = R*V1*RADIUS + ZSPARK

REJECT THE POINT IF IT IS OUTSIDE OF LIMITING PLANES SURROUNDING THE INSTANTANEOUS CHAMBER.

IF (ZTEST .GT. LIMIT(2,3)) GO TO 40
IF (ZTEST .LT. LIMIT(1,3)) GO TO 40
YTEST = R*V2*RADIUS + YSPARK
IF (YTEST .GT. LIMIT(2,2)) GO TO 40
IF (YTEST .LT. LIMIT(1,2)) GO TO 40
XTEST = (S + S - 1.)*RADIUS + YSPARK
IF (XTEST .GT. LIMIT(2,1)) GO TO 40
IF (XTEST .LT. LIMIT(1,1)) GO TO 40

NCOUNT = 0

DO 30 FACET = 1, FACETS

FIND INTERSECTION OF VECTOR THROUGH TEST POINT ON SPHERE AND SPARK LOCATION WITH PLANE OF A FACET

CALL INTERS (FACET, XTEST, YTEST, ZTEST, XINT, YINT, ZINT)

IF (XINT .EQ. 99999.) GO TO 30
AA = 3
BB = 1
CC = 2

ENTER LOOP TO DECIDE WHETHER INTERSECTION POINT FOUND ABOVE LIES WITHIN THE FACET. CONSIDER 2 APICES OF A FACET. IF REMAINING APEX IS ON OPPOSITE SIDE OF LINE THROUGH FIRST 2 APICES FROM TEST POINT, THEN TEST POINT IS OUTSIDE OF FACET. REPEAT FOR ALL 3 PAIRS OF APICES. TEST POINT MUST PASS ALL 3 TESTS TO BE INSIDE FACET.

DO 20 L = 1, 3
A = BB
B = CC
C = AA
AA = A
BB = B
CC = C
PNTA = POINT(FACET,A)
PNTB = POINT(FACET,B)
PNTC = POINT(FACET,C)

DNA = COORD(PNTA,1) - COORD(PNTB,1)
DNB = COORD(PNTA,2) - COORD(PNTB,2)
DNB = COORD(PNTA,3) - COORD(PNTB,3)

XCA = COORD(PNTC,1) - COORD(PNTA,1)
YCA = COORD(PNTC,2) - COORD(PNTA,2)
ZCA = COORD(PNTC,3) - COORD(PNTA,3)

DISTCL = (DNB+ZCA - DNC+YCA - DNB+ZCA)*INP
+ (DNA+ZCA - DNC+XCA)*INP
+ (DNB+XCA - DNA+YCA)*INP

YIA = YINT - COORD(PNTA,2)
ZIA = ZINT - COORD(PNTA,3)

DISTIL = (DNB+YIA - DNB+ZIA)*(DNC+YIA - DNB+ZIA)

IF (DISTIL .GT. DISTCL) GO TO 30

XIA = XINT - COORD(PNTA,1)

DISTIL = DISTIL + (DNA+ZIA - DNC+XIA)*INP
+ (DNA+ZIA - DNC+XIA)

IF (DISTIL .GT. DISTCL) GO TO 30

DISTIL = DISTIL + (DNB+XIA - DNA+YIA)*INP
+ (DNB+XIA - DNA+YIA)

IF (DISTIL .GT. DISTCL) GO TO 30

XCI = COORD(PNTC,1) - XINT
YCI = COORD(PNTC,2) - YINT
ZCI = COORD(PNTC,3) - ZINT

DISTCI = (DNB+YCI - DNB+ZCI)*(DNC+YCI - DNB+ZCI)
+ (DNA+ZCI - DNC+XCI)*INP
+ (DNB+XCI - DNA+YCI)*INP

IF (DISTCI .LT. DISTIL) GO TO 20

IF (DISTCI .LT. DISTCL) GO TO 20
GO TO 30

20 CONTINUE

REJECT FACET UNLESS IT IS BETWEEN THE TEST POINT AND THE SPARK LOCATION

ADIST = SQRT((XTEST - XPARK)*(XTEST - XPARK) +
+ (YTEST - YSPARK)*(YTEST - YSPARK) +
+ (ZTEST - ZSPARK)*(ZTEST - ZSPARK))

BDIST = SQRT((XTEST - XINT)*(XTEST - XINT) +
+ (YTEST - YINT)*(YTEST - YINT) +
+ (ZTEST - ZINT)*(ZTEST - ZINT))

CDIST = SQRT((XSPARK - XINT)*(XSPARK - XINT) +
+ (YSPARK - YINT)*(YSPARK - YINT) +
+ (ZSPARK - ZINT)*(ZSPARK - ZINT))

IF (BDIST .GT. ADIST) GO TO 30

IF (CDIST .GT. ADIST) GO TO 30

FACET IS INTERSECTED BY TEST VECTOR; INCREMENT NCOUNT

NCOUNT = NCOUNT + 1

30 CONTINUE

IF VECTOR INTERSECTS AN ODD NUMBER OF FACETS, THEN THE TEST POINT IS OUTSIDE THE CHAMBER. OTHERWISE, IT IS INSIDE. TRY A NEW POINT.

IF ((NCOUNT/2)*2 .LT. NCOUNT ) GO TO 40

INPTS = INPTS + 1

40 CONTINUE

CALCULATE FLAME SURFACE AREA FOR THIS ITERATION

FSURFI = ( 4. * 3.1415926 * RADIUS*RADIUS * FLOAT(INPTS) )
+ /FLOAT(SPNTS)

SUM = SUM + FSURFI

SUMSQ = SUMSQ + ( FSURFI*FSURFI )

50 CONTINUE

CALCULATE FLAME AREA AVERAGED OVER ALL ITERATIONS.
FSURF = SUM/FLOAT(ITTERS)
STADEV = SQRT(SUMSQ(FLOAT(ITTERS) - (FSURF*FSURF)))
C
RETURN
END
C

SUBROUTINE WALL
C
PURPOSE
C CALCULATES THE SURFACE AREAS ON EACH SIDE OF THE FLAME
C FOR EACH OF THE THREE ZONES OF THE CHAMBER (I.E. HEAD,
C PISTON, CYLINDER WALL).
C
USAGE
C CALL WALL (ITHETA, IRAD, RADIUS, AINH, AOUTH, AINP,
C & AOUTP, AINOW, AOUTCW)
C
DESCRIPTION OF PARAMETERS
C PARAMETER INPUT OUTPUT DESCRIPTION
C ITHETA YES NO CRANK ANGLE INDEX
C IRAD YES NO RADIUS INDEX
C RADIUS YES NO FLAME RADIUS
C AINH NO YES HEAD AREA INSIDE THE FLAME
C AOUTH NO YES HEAD AREA OUTSIDE THE FLAME
C AINP NO YES PISTON AREA INSIDE THE FLAME
C AOUTP NO YES PISTON AREA OUTSIDE THE FLAME
C AINOW NO YES CYL. WALL AREA INSIDE THE FLAME
C AOUTCW NO YES CYL. WALL AREA OUTSIDE THE FLAME
C BCOUNT YES NO 1 = START OF THETA LOOP, ELSE > 1
C CCOUNT YES NO 1 = START OF RADIUS LOOP, ELSE > 1
C
REMARKS
C THE PROCEDURE USED IN THIS CALCULATION IS INDEPENDENT OF
C THE NUMBER OF RANDOM POINTS ON THE SURFACE OF THE SPHERE.
C ACCURACY DEPENDS MAINLY ON HOW CLOSELY THE CHAMBER IS RE-
C PRESENTED BY FLAT TRIANGULAR FACETS.
C
SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED
C TRIARE  INTSPH
C
METHOD
C 'WALL' CONSIDERS EACH FACET AT A TIME. BY COMPARING THE DI-
C STANCE FROM THE FACET APICES TO THE SPARK LOCATION WITH THE
C FLAME RADIUS, IT DECIDES WHETHER THE FACET LIES INSIDE OR
C OUTSIDE THE FLAME. IF 2 OF THE APICES LIE ON ONE SIDE AND
C THE OTHER APICES LIE ON THE OTHER, 'WALL' CALLS SUBROUTINE
C 'INTSPH' TO LOCATE THE POINTS AT WHICH THE FLAME RADIUS
C INTERSECTS 2 SIDES OF THE FACET. IT THEN CALCULATES THE
C APPROXIMATE AREA OF THE FACET ON EACH SIDE OF THE FLAME.
C THE TOTAL WALL AREA IN ANY ZONE IS THE SUM OF THE CONTRIBU-
C TIONS FROM THE INDIVIDUAL FACETS IN THAT ZONE.
C
WRITTEN BY S. G. Poulos
C EDITED BY S. G. Poulos
C
SUBROUTINE WALL (ITHETA, IRAD, RADIUS, AINH, AOUTH, AINP,
C & AOUTP, AINOW, AOUTCW)
C
INTEGER FACET, FACETS, ZONE(400), FLAG, BCOUNT, CCOUNT,
C & A, B, C, AA, BB, CC, FIRST, SECOND, OTHER, APEX,
C & POINT(400,3), PNT, POINTS
REAL COORD(200,3), AREA(400), RADDIF(3), AINH(30,50), AINP(30,50)
C & AINOW(30,50), AOUTH(30,50), AOUTP(30,50), AOUTCW(30,50)
COMMON/DIMENS/ BORE, STROKE, CONRL
COMMON/SPARK/ XSPARK, YSPARK, ZSPARK
COMMON/QUANTS/ FACETS, NHEAD, NPIST, NCWALL, POINTS

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COMMON/FPNTS/ POINT
COMMON/COORDI/ COORD
COMMON/ZONES/ ZONE
COMMON/BCNT/ BCOUNT
COMMON/CCNT/ CCOUNT

C
AINH(ITHETA,IRAD) = 0.0
AOUTH(ITHETA,IRAD) = 0.0
AINP(ITHETA,IRAD) = 0.0
AOUTP(ITHETA,IRAD) = 0.0
AINCW(ITHETA,IRAD) = 0.0
AOUTCW(ITHETA,IRAD) = 0.0

C
BCOUNT = BCOUNT + 1
CCOUNT = CCOUNT + 1

C
CONSIDER ONE FACET ON THE CHAMBER WALL

C
DO 110 FACET = 1, FACETS
   IF (ZONE(FACET) .NE. 3) GO TO 10
   IF (CCOUNT .GT. 1) GO TO 20
C
   RECALCULATE FACET AREA IF FACET HAS JUST BEEN STRETCHED
C
   CALL TRIARE (FACET, TRIA)
C
   AREA(FACET) = TRIA
   GO TO 20
C
10   IF (BCOUNT .GT. 1) GO TO 20

C
   CALCULATE FACET AREA IF THIS HAS NOT YET BEEN DONE.
C
   CALL TRIARE (FACET, TRIA)
C
   AREA(FACET) = TRIA
   20   IF (ZONE(FACET) .NE. 1) GO TO 30
   IF (ITHETA .GT. 1) GO TO 110
C
   ADDAREA = AREA(FACET)
   AINSP = 0.0
   AOUTSP = 0.0
   FLAG = 0

C
   FIND DIFFERENCE BETWEEN DISTANCE FROM EACH FACET APEX
   TO SPARK LOCATION AND FLAME RADIUS
C
   DO 40 APEX = 1, 3
   PNT = POINT(FACET, APEX)
   RADDIF(APLEX) =
     & SQRT((COORD(PNT,1) - XSPARK)*(COORD(PNT,1) - XSPARK) +
     & (COORD(PNT,2) - YSPARK)*(COORD(PNT,2) - YSPARK) +
     & (COORD(PNT,3) - ZSPARK)*(COORD(PNT,3) - ZSPARK))
     & - RADIUS
   CONTINUE
C
40   IF ( (RADDIF(1) .GE. 0.0) .AND. (RADDIF(2) .GE. 0.0) .AND.
     & (RADDIF(3) .GE. 0.0) ) GO TO 100
     & IF ( (RADDIF(1) .LE. 0.0) .AND. (RADDIF(2) .LE. 0.0) .AND.
     & (RADDIF(3) .LE. 0.0) ) GO TO 90

C
   SET FLAG = 1 IF FACET LIES LIES NEITHER FULLY OUTSIDE
   NOR FULLY INSIDE OF FLAME
C
   FLAG = 1
   AA = 3
   BB = 1
   CC = 2

C
   ENTER LOOP TO IDENTIFY WHICH SIDES OF THE FACET ARE

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INTERSECTED BY THE FLAME RADIUS

DO 50 II = 1, 3
   A = BB
   B = CC
   C = AA
   AA = A
   BB = B
   CC = C
   IF ( (RADDIF(A) .GE. 0.0) .AND. (RADDIF(B) .GE. 0.0) )
      GO TO 60
   IF ( (RADDIF(A) .LE. 0.0) .AND. (RADDIF(B) .LE. 0.0) )
      GO TO 60
50 CONTINUE

POINTS 'FIRST' AND 'SECOND' LIE ON ONE SIDE OF THE FLAME,
WHILE POINT 'OTHER' LIES ON THE OTHER SIDE.

60 FIRST = A
   SECOND = B
   OTHER = C

FIND INTERSECTION BETWEEN FLAME RADIUS AND SIDES OF FACET

CALL INTSPH (RADIUS, FACET, OTHER, FIRST, XX1, YY1, ZZ1)
CALL INTSPH (RADIUS, FACET, OTHER, SECOND, XX2, YY2, ZZ2)

CALCULATE AREA OF MINOR TRIANGLE OF INTERSECTED FACET

PNT = POINT(FACET, OTHER)
SIDEA = SQRT((COORD(PNT,1) - XX1)*(COORD(PNT,1) - XX1) +
   (COORD(PNT,2) - YY1)*(COORD(PNT,2) - YY1) +
   (COORD(PNT,3) - ZZ1)*(COORD(PNT,3) - ZZ1))
SIDEB = SQRT((COORD(PNT,1) - XX2)*(COORD(PNT,1) - XX2) +
   (COORD(PNT,2) - YY2)*(COORD(PNT,2) - YY2) +
   (COORD(PNT,3) - ZZ2)*(COORD(PNT,3) - ZZ2))
SIDEC = SQRT((XX1 - XX2)*(XX1 - XX2) +
   (YY1 - YY2)*(YY1 - YY2) +
   (ZZ1 - ZZ2)*(ZZ1 - ZZ2))
SDUMY = 0.5*(SIDEA + SIDEB + SIDEC)
IF (RADDIF(OTHER) .LE. 0.0) GO TO 70
AOUTSP = SQRT(ABS(SDUMY*(SDUMY - SIDEA)*(SDUMY - SIDEa)
                + (SDUMY - SIDEc)))
AINESS = AREAF(FACET) - AOUTSP
GO TO 80

70 AINESS = SQRT(ABS(SDUMY*(SDUMY - SIDEA)*(SDUMY - SIDEB)
                    + (SDUMY - SIDEc)))
AOUTSP = AREAF(FACET) - AINESS

80 ADDARE = 0.0

ALLOCATE AREA OR SUB-AREAS OF CURRENT FACET TO
APPROPRIATE ZONE OF THE CHAMBER

90 IF (ZONE(FACET) .EQ. 1) AINH(IHTHETA, IRAD) = AINH(IHTHETA, IRAD)
   + ADDARE + AINESS
   IF (ZONE(FACET) .EQ. 2) AINP(IHTHETA, IRAD) = AINP(IHTHETA, IRAD)
   + ADDARE + AINESS
   IF (ZONE(FACET) .EQ. 3) AINCW(IHTHETA, IRAD) =
   AINCW(IHTHETA, IRAD) + ADDARE + AINESS
   IF (FLAG .LT. 1) GO TO 110

100 IF (ZONE(FACET) .EQ. 1) AOUTH(IHTHETA, IRAD) =
      AOUTH(IHTHETA, IRAD) + ADDARE + AOUTSP
   IF (ZONE(FACET) .EQ. 2) AOUTP(IHTHETA, IRAD) =
      AOUTP(IHTHETA, IRAD) + ADDARE + AOUTSP
   IF (ZONE(FACET) .EQ. 3) AOUTCW(IHTHETA, IRAD) =
      AOUTCW(IHTHETA, IRAD) + ADDARE + AOUTSP

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110 CONTINUE

NO NEED TO RECALCULATE CYLINDER HEAD AREAS IF ALREADY DONE

IF (I.THETA .EQ. 1) GO TO 120
A.INH(I.THETA,IRAD) = A.INH(1,IRAD)
A.OUTH(I.THETA,IRAD) = A.OUTH(1,IRAD)

120 RETURN

END

SUBROUTINE RQUIT

PURPOSE

'ROUIT' IS CALLED EACH TIME THE PISTON POSITION IS CHANGED
IN ORDER TO EVALUATE THE SMALLEST RADIUS TO WHICH THE FLAME
WILL HAVE TO DEVELOP IN ORDER FOR IT TO SWEEP THROUGH THE
ENTIRE CHAMBER.

USAGE

CALL RQUIT (RMAX)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMAX</td>
<td>NO</td>
<td>YES</td>
<td>MAXIMUM FLAME RADIUS FOR FLAME TO</td>
</tr>
<tr>
<td>DIST</td>
<td>NO</td>
<td>NO</td>
<td>SWEEP THROUGH INSTANTANEOUS CHAMBER</td>
</tr>
<tr>
<td>ACCOUNT</td>
<td>YES</td>
<td>YES</td>
<td>DISTANCE FROM SPARK TO A CORNER OF</td>
</tr>
<tr>
<td>POINTS</td>
<td>YES</td>
<td>NO</td>
<td>LIMITING VOLUME SURROUNDING CHAMBER</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>1 FOR FIRST CALL, &gt;1 THEREAFTER</td>
</tr>
</tbody>
</table>

REMARKS

NONE

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED

METHOD

FOR EACH VALUE OF THETA (CRANK ANGLE), THE DISTANCE FROM THE
SPARK LOCATION TO THE 8 CORNERS OF A RECTANGULAR PARALLELEPIPED
SURROUNDING THE CHAMBER IS CALCULATED. THE MAXIMUM VALUE FROM
AMONG THESE DISTANCES IS THE OUTER LIMIT OF FLAME TRAVEL.

WRITTEN BY S. G. Poulos
EDITED BY S. G. Poulos

INTEGER ACCOUNT, PNT, POINTS, HOLD(200)
REAL LIMIT(2,3), COORD(200,3)
COMMON/DIMENS/ BORE, STROKE, CONRL
COMMON/QUANTS/ FACETS, NHEAD, NPIST, NCWALL, POINTS
COMMON/SPARK/ XSPARK, YSPARK, ZSPARK
COMMON/LIMITS/ LIMIT, PI.SBOT
COMMON/COORDI/ COORD
COMMON/FIXPNT/ HOLD
COMMON/AGNT/ ACCOUNT

RMAX = 0.0
ACCOUNT = ACCOUNT + 1

DO 20 PNT = 1, POINTS
    DUMYX = COORD(PNT,1)
    DUMYY = COORD(PNT,2)
    DUMYZ = COORD(PNT,3)

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WHEN CALLED FOR THE FIRST TIME (ACOUNT = 1), BASE
CALCULATION OF MAXIMUM RADIUS ON BDC PISTON POSITION

IF (ACOUNT .GT. 1) GO TO 10
IF (HOLD(PNT) .LT. 1) DUMYZ = DUMYZ - STROKE

FIND DISTANCE FROM SPARK TO CORNER OF LIMITING VOLUME
10  DIST = SQRT((DUMYX - XSPARK)*(DUMYX - XSPARK) +
              (DUMYY - YSPARK)*(DUMYY - YSPARK) +
              (DUMYZ - ZSPARK)*(DUMYZ - ZSPARK))

IF (DIST .GT. RMAX) RMAX = DIST
20 CONTINUE

RETURN
END

SUBROUTINE CHANGE

PURPOSE
'CHANGE' IS CALLED EACH TIME THE PISTON POSITION IS TO BE
CHANGED. THIS SUBROUTINE CALCULATES THE DISTANCE BY WHICH
THE PISTON DROPS AS A FUNCTION OF CRANK ANGLE. IT THEN
LOCATES THOSE POINTS ON THE CHAMBER SURFACE WHICH ARE LA-
BELED AS MOVABLE AND SHIFTS THEM APPROPRIATELY. IN ADDITION,
'CHANGE'-shifts the lower Z LIMIT USED IN SUBROUTINE FLAME.

USAGE
CALL CHANGE (THETA,OLDROP)

DESCRIPTION OF PARAMETERS
PARAMETER    INPUT   OUTPUT   DESCRIPTION
             YES     NO       CURRENT CRANK ANGLE
THETA        YES     YES     STORES DISTANCE TRAVELLED BY
              -------         PISTON FROM TDC POSITION
OLDROP

REMARKS
NONE

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED

METHOD
SEE PURPOSE, ABOVE

WRITTEN BY S. G. POULOS
EDITED BY S. G. POULOS

SUBROUTINE CHANGE (THETA,OLDROP)

INTEGER APEX, HOLD(200), PNT, POINTS
REAL LIMIT(2,3), COORD(200,3)
COMMON/DIMENS/ BORE, STROKE, CONRL
COMMON/QUANTS/ FACETS, NHEAD, NPIST, NCWALL, POINTS
COMMON/LIMITS/ LIMIT, PISBOT
COMMON/COORD1/ COORD
COMMON/FIXPNT/ HOLD

CONVERT THETA TO RADIANS

THETAR = 3.1415926*THETA/180.

CALCULATE DISTANCE TRAVELLED BY PISTON FROM TDC POSITION TO
CURRENT CRANK ANGLE POSITION, AND ADJUSTMENT FROM LAST THETA
ZDROP = -((CONRL + (STROKE/2.)*(1. - COS(THE TAR)) - SQRT(CONRL*
    &
    CONRL - (STROKE/2.)*(STROKE/2.)*SIN(THE TAR)*SIN(THE TAR) ))
DELTAZ = ZDROP - OLDROP

SHIFT LOWER Z LIMIT TO MATCH NEW VALUE OF THETA

LIMIT(1,3) = LIMIT(1,3) + DELTAZ

SHIFT Z COORDINATE OF ALL NON-FIXED POINTS BY DELTAZ

OLDROP = ZDROP

DO 10 PNT = 1, POINTS
   IF (HOLD(PNT) .GE. 1) GO TO 10
   COORD(PNT,3) = COORD(PNT,3) + DELTAZ
10 CONTINUE

RETURN
END

SUBROUTINE INTERS

PURPOSE
THIS SUBROUTINE CALCULATES THE INTERSECTION BETWEEN A VECTOR
PASSING THROUGH THE SPARK LOCATION AND A TEST POINT ON A
FLAME SPHERE AND THE PLANE OF A FACET ON THE CHAMBER WALL.

USAGE
CALL INTERS (FACET, XTEST, YTEST, ZTEST, XINT, YINT, ZINT)

DESCRIPTION OF PARAMETERS
PARAMETER INPUT OUTPUT DESCRIPTION

FACET YES NO FACET NUMBER
XTEST YES NO X COORDINATE OF POINT ON SPHERE
YTEST YES NO Y COORDINATE OF POINT ON SPHERE
ZTEST YES NO Z COORDINATE OF POINT ON SPHERE
XINT NO YES X COORDINATE OF FACET INTERSECTION
YINT NO YES Y COORDINATE OF FACET INTERSECTION
ZINT NO YES Z COORDINATE OF FACET INTERSECTION

REMARKS
NONE

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED

METHOD
'INTERS' PERFORMS AN ANALYTICAL SOLUTION OF 2 SIMULTANEOUS
EQUATIONS: 1) THE EQUATION OF A VECTOR DEFINED BY THE 3-D
COORDINATES OF 2 POINTS (SPARK & TEST); 2) THE EQUATION OF A
PLANE DEFINED BY THE 3-D COORDINATES OF THE 3 POINTS OF A
TEST FACET (PNT1, PNT2, PNT3)

WRITTEN BY S. G. Poulos
EDITED BY S. G. Poulos

SUBROUTINE INTERS (FACET, XTEST, YTEST, ZTEST, XINT, YINT, ZINT)

INTEGER FACET, POINT(400,3), PNT1, PNT2, PNT3
REAL COORD(200,3)
COMMON/SPARK/ XSPARK, YSPARK, ZSPARK
COMMON/FPNTS/ POINT
COMMON/COORD1/ COORD

XST = XSPARK - XTEST
IF (ABS(XST) .GE. 0.0000001) GO TO 10
XINT = 99999.
GO TO 30
10 YST = YSPARK - YTEST
ZST = ZSPARK - ZTEST

C
PNT1 = POINT(FACET,1)
PNT2 = POINT(FACET,2)
PNT3 = POINT(FACET,3)
C
YZBAR = (COORD(PNT2,2) - COORD(PNT1,2) )* (COORD(PNT3,3) -
1 COORD(PNT1,3) ) - (COORD(PNT2,3) - COORD(PNT1,3) )*
2 (COORD(PNT3,2) - COORD(PNT1,2) )
ZXBAR = (COORD(PNT2,3) - COORD(PNT1,3) )* (COORD(PNT3,1) -
1 COORD(PNT1,1) ) - (COORD(PNT2,1) - COORD(PNT1,1) )*
2 (COORD(PNT3,3) - COORD(PNT1,3) )
XYBAR = (COORD(PNT2,1) - COORD(PNT1,1) )* (COORD(PNT3,2) -
1 COORD(PNT1,2) ) - (COORD(PNT2,2) - COORD(PNT1,2) )*
2 (COORD(PNT3,1) - COORD(PNT1,1) )
C
IN THE EVENT THAT BOT = 0.0 (AN EXTREMELY UNLIKELY EVENT),
NEGLECT THIS FACET TO AVOID DIVISION BY ZERO
C
BOT = (ZXBAR * YST) + (XYBAR * ZST) + (YZBAR * XST)
IF (ABS(BOT) .GE. 0.0000001) GO TO 20
XINT = 99999.
GO TO 30
C
20 A = (YST * XTEST) + XST*(COORD(PNT1,2) - YTEST )
B = (ZST * XTEST) + XST*(COORD(PNT1,3) - ZTEST )
C = XST * COORD(PNT1,1)
C
TOP = (ZXBAR * A) + (XYBAR * B) + (YZBAR * C)
C
CALCULATE COORDINATES OF INTERSECTION BETWEEN TEST VECTOR
PLANE OF TEST FACET
C
XINT = TOP/BOT
YINT = YTEST + (YST/XST )*(XINT - XTEST)
ZINT = ZTEST + (ZST/XST )*(XINT - XTEST)
C
30 RETURN
END
C
SUBROUTINE INTSPH
C
PURPOSE
'INTSPH' IS CALLED BY SUBROUTINE 'WALL' WHEN A FACET IS
INTERSECTED BY THE FLAME AT THE CURRENT RADIUS. IN ORDER
TO PROPERLY ALLOCATE THE AREA OF THE FACET TO EACH SIDE OF
THE FLAME (INSIDE/OUTSIDE), THE POINTS OF INTERSECTION
BETWEEN THE RADIUS AND THE 2 INTERSECTED SIDES OF THE FACET
MUST BE FOUND. THIS SUBROUTINE CALCULATES THE COORDINATES
OF ONE OF THE INTERSECTING POINTS WHEN GIVEN THE END POINTS
OF ONE OF THE INTERSECTED SIDES OF THE FACET.
C
USAGE
CALL INTSPH (RADIUS, FACET, APEX1, APEX2, XX, YY, ZZ)
C
DESCRIPTION OF PARAMETERS
PARAMETER INPUT OUTPUT DESCRIPTION
C
RADIUS YES NO CURRENT FLAME RADIUS
FACET YES NO FACET NUMBER
APEX1 YES NO POINT ON INTERSECTED SIDE OF FACET
APEX2 YES NO POINT ON INTERSECTED SIDE OF FACET
XX NO YES X COORDINATE OF INTERSECTION POINT
YY NO YES Y COORDINATE OF INTERSECTING POINT
ZZ NO YES Z COORDINATE OF INTERSECTING POINT

REMARKS
NONE

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED

METHOD
TO FIND THE POINT WHERE THE RADIUS INTERSECTS A GIVEN SIDE OF
THE FACET, FIRST FIND WHICH OF THE TWO POINTS ON THE ENDS
OF THE SIDE OF THE FACET IS OUTSIDE THE FLAME. THIS IS
NEEDED TO PLACE THE INPUTS TO THE CALCULATION IN STANDARD
FORM. THEN TAKE A TEST POINT WHICH BISECTS THE SIDE OF THE
FACET AND COMPARE ITS DISTANCE TO THE SPARK LOCATION WITH THE
CURRENT FLAME RADIUS. WITH THE RESULT, DECIDE WHICH HALF OF
THE FACET SIDE MUST CONTAIN THE SOLUTION POINT. NEXT, BISECT
THIS HALF AND MAKE THE BISECTION THE NEW TEST POINT. REPEAT
THIS PROCESS UNTIL THE DISTANCE BETWEEN THE TEST POINT AND
THE SPARK PLUG IS SUFICIENTLY CLOSE TO THE FLAME RADIUS.

WRITTEN BY S. G. POULOS
EDITED BY S. G. POULOS

SUBROUTINE INTSPH (RADIUS, FACET, APEX1, APEX2, XX, YY, ZZ)

INTEGER PNT, POINT(400,3), APEX1, APEX2, FACET
REAL COORD(200,3)
COMMON/SPARK/ XSPARK, YSPARK, ZSPARK
COMMON/FPNTS/ POINT
COMMON/COORD1/ COORD

PNT = POINT(FACET, APEX1)
XA = COORD(PNT,1)
YA = COORD(PNT,2)
ZA = COORD(PNT,3)

PNT = POINT(FACET, APEX2)
XB = COORD(PNT,1)
YB = COORD(PNT,2)
ZB = COORD(PNT,3)

ARAD = SQRT( (XA - XSPARK) * (XA - XSPARK) + (YA - YSPARK) *
& (YA - YSPARK) + (ZA - ZSPARK) * (ZA - ZSPARK) )
& BRAD = SQRT( (XB - XSPARK) * (XB - XSPARK) + (YB - YSPARK) *
& (YB - YSPARK) + (ZB - ZSPARK) * (ZB - ZSPARK) )

CHECK TO SEE WHICH FACET POINT IS OUTSIDE OF FLAME

IF (ARAD .GT. BRAD) GO TO 10

DUMXA = XA
DUMYA = YA
DUMZA = ZA
XA = XB
YA = YB
ZA = ZB
XB = DUMXA
YB = DUMYA
ZB = DUMZA

10 DO 30 I = 1, 30

NEW TEST POINT BISECTS RANGE WHERE SOLUTION MUST EXIST

TESTX = (XA + XB)/2.
TESTY = (YA + YB)/2.
TESTZ = (ZA + ZB)/2.
TESTR = SQRT((TESTX - XSPARK)*(TESTX - XSPARK) +
           (TESTY - YSPARK)*(TESTY - YSPARK) +
           (TESTZ - ZSPARK)*(TESTZ - ZSPARK))

CHECK FOR CONVERGENCE

IF ( ABS((TESTR - RADIUS)/RADIUS) .LE. 0.0001 ) GO TO 40
IF (TESTR .GT. RADIUS) GO TO 20

XB = TESTX
YB = TESTY
ZB = TESTZ
GO TO 30

20
XA = TESTX
YA = TESTY
ZA = TESTZ
GO TO 30

CONTINUE

40

RETURN

END

SUBROUTINE TRIARE

PURPOSE

THIS SUBROUTINE CALCULATES THE SURFACE AREA OF A TRIANGULAR FACET. GIVEN THE FACET NUMBER, 'TRIARE' OBTAINS THE COORDINATES OF THE 3 APICES OF THE FACET AND THEN USES A STANDARD ANALYTICAL FORMULA TO CALCULATE IT'S SURFACE AREA.

USAGE

CALL TRIARE (FACET, TRIA)

DESCRIPTION OF PARAMETERS

PARAMETER INPUT OUTPUT DESCRIPTION
FACET YES NO FACET NUMBER
TRIA NO YES AREA OF FACET

REMARKS

NONE

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED

METHOD


WRITTEN BY S. G. POULOS
EDITED BY S. G. POULOS

SUBROUTINE TRIARE (FACET, TRIA)

INTEGER PNT1, PNT2, PNT3, POINT(400,3), FACET
REAL COORD(200,3)
COMMON/FPNT5/ POINT
COMMON/COORD/ COORD

PNT1 = POINT(FACET,1)
PNT2 = POINT(FACET,2)
PNT3 = POINT(FACET,3)
CALCULATE LENGTHS OF SIDES OF TRIANGLE

SIDEA = SQRT( 
  & (COORD(PNT1,1) - COORD(PNT2,1))*(COORD(PNT1,1) - COORD(PNT2,1)) + 
  & (COORD(PNT1,2) - COORD(PNT2,2))*(COORD(PNT1,2) - COORD(PNT2,2)) + 
  & (COORD(PNT1,3) - COORD(PNT2,3))*(COORD(PNT1,3) - COORD(PNT2,3)))

SIDEB = SQRT( 
  & (COORD(PNT1,1) - COORD(PNT3,1))*(COORD(PNT1,1) - COORD(PNT3,1)) + 
  & (COORD(PNT1,2) - COORD(PNT3,2))*(COORD(PNT1,2) - COORD(PNT3,2)) + 
  & (COORD(PNT1,3) - COORD(PNT3,3))*(COORD(PNT1,3) - COORD(PNT3,3)))

SIDEC = SQRT( 
  & (COORD(PNT2,1) - COORD(PNT3,1))*(COORD(PNT2,1) - COORD(PNT3,1)) + 
  & (COORD(PNT2,2) - COORD(PNT3,2))*(COORD(PNT2,2) - COORD(PNT3,2)) + 
  & (COORD(PNT2,3) - COORD(PNT3,3))*(COORD(PNT2,3) - COORD(PNT3,3)))

SDUMY = 0.5 * (SIDEA + SIDEB + SIDEC)

TRIA = SQRT( ABS( SDUMY*(SDUMY - SIDEA)*(SDUMY - SIDEB) 
  & *(SDUMY - SIDEC) ) )

RETURN
END
Appendix D

Addition of Indolene Fuel in Cycle Simulation Subroutine
SUBROUTINE FUELDT

PURPOSE

This subroutine is called to set the values of the fuel related parameters at the start of program execution. The only input required is the fuel type. The parameters which are set are: I) The atom ratios which specify the properties of the fuel-air mixture, the set of enthalpy coefficients associated with the fuel (used in the property routines), and the fuel heating value and stoichiometric fuel/air ratio; II) The atom ratios required by subroutine 'PTCHEM' for calculation of equilibrium burned gas composition.

USAGE

CALL FUELDT

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>FUELTP</td>
<td>YES</td>
<td>NO</td>
<td>FUEL TYPE</td>
</tr>
<tr>
<td>PSI</td>
<td>NO</td>
<td>YES</td>
<td>MOLAR N2 TO O2 RATIO FOR AIR</td>
</tr>
<tr>
<td>XI</td>
<td>NO</td>
<td>YES</td>
<td>MOLAR N2 TO O2 RATIO FOR AIR</td>
</tr>
<tr>
<td>CI</td>
<td>NO</td>
<td>YES</td>
<td># OF CARBON ATOMS/FUEL MOLECULE</td>
</tr>
<tr>
<td>DEL</td>
<td>NO</td>
<td>YES</td>
<td>CARBON/HYDROGEN RATIO OF FUEL</td>
</tr>
<tr>
<td>HY</td>
<td>NO</td>
<td>YES</td>
<td>HYDROGEN ATOMS PER FUEL MOLECULE</td>
</tr>
<tr>
<td>ENW</td>
<td>NO</td>
<td>YES</td>
<td>NITROGEN ATOMS PRE FUEL MOLECULE</td>
</tr>
<tr>
<td>OZ</td>
<td>NO</td>
<td>YES</td>
<td>OXYGEN ATOMS PER FUEL MOLECULE</td>
</tr>
<tr>
<td>QLOWER</td>
<td>NO</td>
<td>YES</td>
<td>LOWER HEATING VALUE OF THE FUEL</td>
</tr>
<tr>
<td>FASTO</td>
<td>NO</td>
<td>YES</td>
<td>STOICHIOMETRIC FUEL/AIR RATIO</td>
</tr>
<tr>
<td>AF(I)</td>
<td>NO</td>
<td>YES</td>
<td>FUEL COEFFICIENT ARRAY</td>
</tr>
<tr>
<td>D(I)</td>
<td>NO</td>
<td>YES</td>
<td>ATOM RATIO ARRAY (SEE PTCHEM)</td>
</tr>
</tbody>
</table>

REMARKS

Only isooctane and propane are available for use as fuels. Indolene is available for use as fuel. Added on 17 April 1987 by Robert T. Chang

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED

METHOD

See purpose, above

WRITTEN BY S. G. POULOS
EDITED BY S. G. POULOS

SUBROUTINE FUELDT

INTEGER FUELTP
REAL AF(6), D(4)

COMMON/FUEL/, FUELTP, ENW, CX, HY, OZ, DEL, PSI, PHI, QLOWER, FASTO
COMMON/FUPR, AF
COMMON/OXDANT/, XI
COMMON/CHM/, D

PSI = 3.76
XI = 3.76

IF (FUELTP .EQ. 2) GO TO 10
IF (FUELTP .EQ. 3) GO TO 15

FOLLOWING DATA FOR ISOOCTANE (FUELTP = 1)

CX = 8.0
DEL = 8.0/18.0
HY = 18.0
ENW = 0.0
OZ = 0.0
QLOWER = 44.392
FASTO = 1./15.11

SET COEFFICIENTS FOR USE BY PROPERTY Routines

AF(1) = -0.55313
AF(2) = 181.62
AF(3) = -97.787
AF(4) = 20.402
AF(5) = -0.03895
AF(6) = -60.518

GO TO 20

FOLLOWING DATA FOR PROPANE (FUELT = 2)

10 CX = 3.0
DEL = 3.0/8.0
HY = 8.0
ENW = 0.0
OZ = 0.0
QLOWER = 46.3
FASTO = 0.0638

AF(1) = -1.4867
AF(2) = 74.339
AF(3) = -39.0849
AF(4) = 8.05426
AF(5) = 0.0121948
AF(6) = -18.4811
GO TO 20

FOLLOWING DATA FOR INDOLENE (FUELT = 3)

15 CX = 7.0
DEL = 1/1.88
HY = 13.16
ENW = 0.0
OZ = 0.0
QLOWER = 43.079
FASTO = 0.08843

AF(1) = -16.9900
AF(2) = 206.8050
AF(3) = -149.4780
AF(4) = 44.5140
AF(5) = 0.3268
AF(6) = -55.0470

CALCULATE ATOM RATIOS FOR USE BY 'PTCHEM'

20 D(1) = 1./DEL
D(2) = 1.0
D(4) = 2.*(1. + 0.25/DEL)/PHI
D(3) = PSI*D(4)

RETURN
END
Appendix E

Modified Valve Size Subroutine
SUBROUTINE MFLRT

PURPOSE
CALCULATES MASS FLOW RATE THROUGH AN ORIFICE.

USAGE
CALL MFLRT (CD, AREA, PO, MW, TO, PS, GAMMA, FLRT)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>CD</td>
<td>YES</td>
<td>NO</td>
<td>DISCHARGE COEFFICIENT</td>
</tr>
<tr>
<td>AREA</td>
<td>YES</td>
<td>NO</td>
<td>AREA OF RESTRICTION (CM**2)</td>
</tr>
<tr>
<td>PO</td>
<td>YES</td>
<td>NO</td>
<td>UPSTREAM PRESSURE (ATM)</td>
</tr>
<tr>
<td>PS</td>
<td>YES</td>
<td>NO</td>
<td>DOWNSTREAM PRESSURE (ATM)</td>
</tr>
<tr>
<td>MW</td>
<td>YES</td>
<td>NO</td>
<td>MOLECULAR WEIGHT (G/MOLE)</td>
</tr>
<tr>
<td>TO</td>
<td>YES</td>
<td>NO</td>
<td>UPSTREAM TEMPERATURE (K)</td>
</tr>
<tr>
<td>GAMMA</td>
<td>YES</td>
<td>NO</td>
<td>RATIO OF SPECIFIC HEATS, CP/CV</td>
</tr>
<tr>
<td>FLRT</td>
<td>NO</td>
<td>YES</td>
<td>MASS FLOW RATE (G/S)</td>
</tr>
</tbody>
</table>

REMARKS
NONE

SUBROUTINE AND FUNCTION SUBPROGRAM REQUIRED
NONE

METHOD
FLOW THROUGH THE ORIFICE IS TREATED AS ONE-DIMENSIONAL, QUASI-STEADY, AND ISENTROPIC (MODIFIED BY A DISCHARGE COEFFICIENT)

WRITTEN BY S. H. MANSOURI AND K. RADHAKRISHNAN
EDITED BY S. H. MANSOURI AND S. G. POULOS

SUBROUTINE MFLRT (CD, AREA, PO, MW, TO, PS, GAMMA, FLRT)

REAL MW
C
FLRT = 0.0
IF (PO.EQ. PS) GO TO 20
GI = 1.0/GAMMA
SUM = GAMMA * MW/TO
CONST = 111.12272 * CD * AREA * PO * SQRT(SUM)

RATIO = PS/PO
CRIT = ( 2./(GAMMA + 1.) )**( GAMMA/(GAMMA - 1. ) )

CHECK IF FLOW IS CHOKED
IF (RATIO .LT. CRIT) GO TO 10

SUBSONIC FLOW
SUN = 2./((GAMMA - 1.) * ( RATIO**(GI + GI) - RATIO**(GI + 1.) )
FLRT = CONST * SQRT(SUN)
GO TO 20

CHOKED FLOW
10 FLRT = CONST * CRIT**( 0.5 * (1.0 + GI )

20 RETURN
END
SUBROUTINE IVACD

PURPOSE
CALCULATES AREA AND DISCHARGE COEFFICIENT
OF INTAKE VALVE

USAGE
CALL IVACD (T, PR, AREA, CD)

DESCRIPTION OF PARAMETERS

PARAMETER INPUT OUTPUT DESCRIPTION

T YES NO TIME (DEG)
PR YES NO PRESSURE RATIO ACROSS INTAKE
AREA NO YES EFFECTIVE AREA OF INTAKE
CD NO YES DISCHARGE COEFFICIENT

REMARKS

PHI
I CCT INTAKE CAM CONTOUR OPENING
I CDT INTAKE CAM CONTOUR CLOSING
THIVO INTAKE VALVE OPENING TIME (DEG)
THIMAX INTAKE VALVE MAXIMUM OPENING TIME (DEG)
THIVC INTAKE VALVE CLOSING TIME (DEG)
DIV DIAMETER OF INTAKE VALVE (IN)
SAIV SEAT ANGLE OF INTAKE VALVE (DEG)
RARM ROCKER ARM RATIO
DPORT DIAMETER OF INTAKE PORT (IN)
APORT AREA OF INTAKE PORT (CM**2)
DIVDP DIVS/DPORT
DIVS = DIV - 2 * EVATH * COS(SAIV)
VTDP EVATH/DPORT
EVATH EFFECTIVE VALVE HEAD THICKNESS (IN)
SIN(SAIV)
COS(SAIV)
SIN(2*SIV)
(1-DIVDP/SS2A)
LIMITING FLOW AREA VALUE
VDTP/(SSA+SSA)
LIMITING FLOW AREA VALUE

SCALE BORE SCALING PARAMETER
SCALE VALVE LIFT SCALER TO SIMULATE GM SPECS.

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
NONE

METHOD
SEE REPORT

WRITTEN BY S. H. MANSOURI AND K. RADHAKRISHNAN
EDITED BY S. H. MANSOURI AND S. G. POULOS

MODIFIED TO SCALE THE VALUE PARAMETERS WITH BORE. MODIFICATIONS
ARE BASED ON GENERAL MOTORS PRACTICES AND PROFESSOR JOHN B. HEYWOOD'S
RECOMMENDATIONS. THE "*ALING OF THE VALVE PARAMETERS WITH BORE IS
THOUGHT TO BE MORE REALISTIC WHEN INVESTIGATING THE EFFECT OF
GEOMETRY.
ON PERFORMANCE AND ECONOMY. ADDITIONS MADE FOR THIS PURPOSE WILL BE
PRECEDED BY "*" COMMENTS.
SUBROUTINE IVACD (T, PR, AREA, CD)

REAL ICCD T, ICCD T
DIMENSION PHIT(79), ICCD T(79), ICCD T(79)
DATA PHIT/1., 2., 3.0, 4.0, 5.0, 6.0, 7.0, 8.0, 9.0, 10.,
35., 36., 37., 38., 39., 40., 41., 42., 43., 44., 45., 46.,
47., 48., 49., 50., 51., 52., 53., 54., 55., 56., 57., 58.,
59., 60., 61., 62., 63., 64., 65., 66., 67., 68., 69., 70.,
71., 72., 73., 74., 75., 76., 77., 78., 79./
DATA ICCD T/2345., 23438., 23404., 23347., 23266., 23163.,
& 23637., 22888., 22716., 22522., 22305., 22065., 21883., 21519.,
& 21212., 20883., 20533., 20160., 19766., 19351., 18914., 18457.,
& 17979., 17482., 16966., 16431., 15878., 15308.,
& 14721., 14108., 13616., 12875., 12234., 11582., 10921., 10253.,
& 09579., 08972., 08225., 07552., 06889., 06240., 05610., 05006.,
& 04432., 03898., 03394., 02937., 02525., 02156., 01833., 01554.,
& 01318., 01125., 00970., 00849., 00756., 00662., 00572., 00476.,
& 00388., 00301., 00234., 00190., 00162., 00126., 00091., 00069.,
& 00046., 00026., 00005., 00001., 00000/0.
DATA ICCD T/2345., 23438., 23404., 23347., 23266., 23163.,
& 23637., 22888., 22716., 22522., 22305., 22065., 21883., 21519.,
& 21212., 20883., 20533., 20160., 19766., 19351., 18914., 18457.,
& 17979., 17482., 16966., 16431., 15878., 15308.,
& 14721., 14108., 13616., 12875., 12234., 11582., 10921., 10253.,
& 09579., 08972., 08225., 07552., 06889., 06240., 05610., 05006.,
& 04432., 03898., 03394., 02937., 02525., 02156., 01833., 01554.,
& 01318., 01125., 00970., 00849., 00756., 00662., 00572., 00476.,
& 00388., 00301., 00234., 00190., 00162., 00126., 00091., 00069.,
& 00046., 00026., 00005., 00001., 00000/0.
DATA TIMAX /100.0/
DATA RARAB/1.50/
DATA SSA,CSA,SS2A/.70711.,.70711,1.0/
DATA REFIO, REFIC/.00490.,.00572/

SCALING DATA REQUIRED FOR CHANGES

COMMON/EPARAM/BORE,STROKE,CONRL,CYLCA,CSATDC,CMTIO,CLVTDC

CALCULATE SCALING PARAMETER ASSUMING BORE = 10.3 AS DEFAULT BORE
FOR THIS LIFT CURVE.  SEE S. H. MANSOURI PH.D. THESIS FOR DEFAULT
VALUES.

SCALER = 0.25 / 0.165815
SCALE = BORE * SCALER / 10.3

SCALE NECESSARY VALVE PARAMETERS WITH BORE

EVATH = 0.08352
DIV = 0.5 * BORE / 2.54
DIVS = DIV - 2 * EVATH * CSA
DIVDP = 0.84269
DPORT = DIV/DIVDP
APORT = 3.14159 * (DPORT**2) * (2.54**2) / 4.0
VTDP = EVATH/DPORT
R2 = (1.0 - DIVDP)/SS2A
R3 = VTDP/(SSA * SSA)

CALCULATE VALVE LIFT

IF (T.GE. TIMAX) GO TO 10
PHI = 0.5 * (TIMAX - T) + 1.0

-174-
IT = IFIX( 0.5 * (TIMAX + 2.9999 - T) )
IF (IT .LE. 1) IT = 2
VLIFT = (ICCDT(IT) + (PHIT(IT) - PHI)*(ICCDT(IT-1) - ICCDT(IT)))
& - REFIC) * RARMR * SCALE
GO TO 20
10 PHI = 0.5 * (T - TIMAX) + 1.0
IT = IFIX( 0.5 * (T + 1.9999 - TIMAX) )
IF (IT .LE. 0) IT = 1
VLIFT = (ICCDT(IT) + (PHIT(IT) - PHI)*( ICCDT(IT) - ICCDT(IT+1) )
& - REFIC) * RARMR * SCALE
20 RISE = VLIFT/DPORT
IF (RISE .LE. 0.0) RISE = 0.0

CALCULATE CD

CDI = 0.96 - 5.47 * RISE * RISE

EFFECT OF PRESSURE RATIO ON DISCHARGE COEFFICIENT

CD = CDI + 0.8 * (1. - CDI) * (CDI - 0.1) * (PR - 1.)
IF ((RISE .LE. R3) .AND. (RISE .GE. R2)) GO TO 30
IF (RISE .GT. R3) GO TO 40
ATAP = 4.0 * (1.0 - 0.5 * RISE + SS2A) * RISE + CSA
GO TO 50
30 ATAP = 4.0 * (DIVDP + .5 * RISE + SS2A) * RISE + CSA
GO TO 50
40 ATAP = 4.0 * (DIVDP + VTDP*CSA/SSA) * SORT( RISE*RISE - 2.0*RISE*
& VTDP + VTDP*VTDP/(SSA + SSA) )

CALCULATE EFFECTIVE VALVE OPEN AREA (CM**2).

50 AREA = ATAP * APORT

RETURN
END

SUBROUTINE EVACD

PURPOSE
CALCULATES AREA AND DISCHARGE COEFFICIENT
OF EXHAUST VALVE

USAGE
CALL EVACD (T, PR, AREA, CD)

DESCRIPTION OF PARAMETERS

PARAMETER | INPUT | OUTPUT | DESCRIPTION
-----------|-------|--------|----------------
T          | YES   | NO     | TIME (DEG)
PR         | YES   | NO     | PRESSURE RATIO ACROSS EXHAUST
AREA       | NO    | YES    | EFFECTIVE AREA OF EXHAUST
CD         | NO    | YES    | DISCHARGE COEFFICIENT

REMARKS

PHIT
ECCDOT
---
ECCDT
---
THEVO
---
THEMAX
---

PHI TABLE
EXHAUST CAM CONTOUR OPENING
DATA TABLE (IN)
EXHAUST CAM CONTOUR CLOSING
DATA TABLE (IN)
EXHAUST VALVE OPENING TIME (DEG)
EXHAUST CAM CONTOUR MAXIMUM
OPENING TIME (DEG)
THEVC

EXHAUST VALVE CLOSING TIME (DEG)

DEV

DIAMETER OF EXHAUST VALVE (IN)

SAEV

SEAT ANGLE OF EXHAUST VALVE (DEG)

RARM

ROCKER ARM RATIO

DPORT

DIAMETER OF EXHAUST PORT (IN)

APORT

AREA OF EXHAUST PORT (CM^2)

DEVDP

DEVS/DPORT

DEVS

DEVS = DEV - 2 * EVATH * COS(SAEV)

VTDP

EVATH/DPORT

EVATH

EFFECTIVE VALVE HEAD THICKNESS (IN)

SSA

SIN(SAEV)

CSA

COS(SAEV)

SS2A

SIN(2*SAEV)

R2

(1.-DEVDP)/SS2A

R3

LIMITING VALUE FOR FLOW AREA CALCULATIONS

VTDP/(SSA*SSA)

SCALE

BORE SCALING PARAMETER

SCALER

LIFT SCALING PARAMETER TO MEET GM SPECS

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED

NONE

METHOD

SEE REPORT

WRITTEN BY S. H. MANSOURI AND K. RADHAKRISHNAN

EDITED BY S. H. MANSOURI AND S. G. POULOS

MODIFIED TO SCALE THE VALUE PARAMETERS WITH BORE. MODIFICATIONS

ARE BASED ON GENERAL MOTORS PRACTICES AND PROFESSOR JOHN B. HEYWOOD'S

RECOMMENDATIONS. THE SCALING OF THE VALVE PARAMETERS WITH BORE IS

THOUGHT TO BE MORE REALISTIC WHEN INVESTIGATING THE EFFECT OF

GEOMETRY ON PERFORMANCE AND ECONOMY. ADDITIONS MADE FOR THIS PURPOSE WILL BE

PRECEDED BY "*" COMMENTS.

MODIFIED BY R. T. CHANG 28 JULY 1987

SUBROUTINE EVACD (T, PR, AREA, CD)

DIMENSION PHIT(88), ECCDOT(88), ECCDOT(88)

DATA PHIT/1., 2., 3.0, 4.0, 5.0, 6.0, 7.0, 8.0, 9.0, 10.,
& 47., 48., 49., 50., 51., 52., 53., 54., 55., 56., 57., 58.,
& 59., 60., 61., 62., 63., 64., 65., 66., 67., 68., 69., 70.,
& 71., 72., 73., 74., 75., 76., 77., 78., 79., 80., 81., 82.,
& 83., 84., 85., 86., 87., 88./

DATA ECCDOT/.23500., .23491., .23466., .23423., .23364., .23287.,
& .19386., .19003., .18604., .18187., .17754., .17303.,
& .00000., .00000., .00000., .00000./

DATA ECCDOT/.23500., .23491., .23466., .23423., .23364., .23287.,
& .19386., .19003., .18604., .18187., .17754., .17303.,
& .00000., .00000., .00000., .00000./
&.06839, .06222, .05622, .05044, .04492, .03971, .03484, .03036, 
&.02626, .02258, .01931, .01645, .01399, .01133, .01023, .00888, 
&.00782, .00700, .00637, .00588, .00547, .00512, .00481, .00450, 
&.00420, .00390, .00360, .00330, .00300, .00270, .00240, .00210, 
&.00180, .00150, .00120, .00091, .00068, .00040, .00022, .00010, 
&.00003, .00001, .00000/
DATA TEMAX/666.0 /
DATA RARMR/1.50/
DATA SSA,CSA,SS2A/.70711,.70711,1.0/
DATA REFEC,REFEC/0.00547,.00670/

* SCALING DATA REQUIRED FOR CHANGES
* COMMON/EPARAM/BORE,STROKE,CONRL,CYLCA,CSATDC,CMRTIO,CLVTDC

* CALCULATE SCALING PARAMETER ASSUMING BORE = 10.3 AS DEFAULT BORE
* FOR THIS LIFT CURVE. SEE S. H. MANSOURI PH D. THESIS FOR DEFAULT
* VALUES.
* SCALER = 0.25 / .196839
SCALE = BORE * SCALER / 10.3

* SCALE NECESSARY VALVE PARAMETERS WITH BORE
* EVATH = 0.08352
DEV = 0.433013 * BORE / 2.54
DIVS = DIVS - 2 * EVATH * CSA
DIVDP = 0.03848
DPORT = DIVS/DIVDP
APORT = 3.14159 * (DPORT**2) * (2.54**2) / 4.0
VTDP = EVATH/DPORT
R2 = (1.0 - DEVDP)/SS2A
R3 = VTDP/(SSA * SSA)
C
TAT = T
IF (TAT .LT. 360.0) TAT = TAT + 720.0
C
C CALCULATE VALVE LIFT
C
IF (TAT .GE. TEMAX) GO TO 10
PHI = 0.5 * (TEMAX - TAT) + 1.0
IT = IFIX( 0.5 * (TEMAX + 2.9999 - TAT) )
IF (IT .LE. 1) IT = 2
VLIFT = (ECCD(TI) + (PHIT(I) - PHI)*(ECCD(IT-1) - ECCD(IT)))
& - REFEC) * RARMR * SCALE
GO TO 20
C
10 PHI = 0.5 * (TAT - TEMAX) + 1.0
IT = IFIX( 0.5 * (TAT + 1.9999 - TEMAX) )
IF (IT .LE. 0) IT = 1
VLIFT = (ECCD(IT) + (PHIT(IT) - PHI)*(ECCD(IT) - ECCD(IT+1)))
& - REFEC) * RARMR * SCALE
20 RISE = VLIFT/DPORT
IF (RISE .LE. 0.0) RISE = 0.0
C
C CALCULATE CD
C
30 CDI = 1.05 - 9.30 * RISE * RISE
40 IF (PR .GT. 1.86) GO TO 50
C
C EFFECT OF PRESSURE RATIO ON DISCHARGE COEFFICIENT
C
CD = CDI + 0.8 * (1. - CDI) * (CDI - 0.1) * (PR - 1.)
GO TO 60
50 CD = CDI + 0.7 * (1. - CDI) * (CDI - 0.1) + (1. - CDI) * 
& (0.27 - 0.1 * CDI) * (1. -3.46/(PR * PR))
60 IF (((RISE .LE. R3) .AND. (RISE .GE. R2)) GO TO 70
IF (RISE .GT. R3) GO TO 80

-177-
ATAP = 4.0 * (1.0 - .5 * RISE * SS2A) * RISE * CSA
GO TO 90
70 ATAP = 4.0 * (DEVDP + .5 * RISE * SS2A) * RISE * CSA
GO TO 90
80 ATAP = 4.0 * (DEVDP + VTDP*CSA/SSA) * SQRT( RISE*RISE - 2.0*RISE*
 & VTDP + VTDP * VTDP/(SSA * SSA) )

C CALCULATE EFFECTIVE VALVE OPEN AREA ( CM**2 ).
C
90 AREA = ATAP * APORT
C
RETURN
END
C
C

-178-
Appendix F

Modified Characteristic Velocity Subroutine
SUBROUTINE EXAUST

PURPOSE
CALCULATES THE TIME RATE OF CHANGE OF PRESSURE, TEMPERATURE, MASS, HEAT TRANSFER, WORK TRANSFER, MEAN KINETIC ENERGY, AND TURBULENT KINETIC ENERGY IN THE CYLINDER DURING EXAUST.

USAGE
CALL EXAUST (DT, DY, DYP)

DESCRIPTION OF PARAMETERS

<table>
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<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
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<tbody>
<tr>
<td>DT</td>
<td>YES</td>
<td>NO</td>
<td>TIME (DEG)</td>
</tr>
<tr>
<td>DY(1)</td>
<td>YES</td>
<td>NO</td>
<td>MASS INDUCTED INTO CHAMBER THROUGH INTAKE VALVE (G)</td>
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<tr>
<td>DY(2)</td>
<td>YES</td>
<td>NO</td>
<td>MASS EXHAUSTED FROM CHAMBER THROUGH EXHAUST VALVE (G)</td>
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<tr>
<td>DY(6)</td>
<td>YES</td>
<td>NO</td>
<td>MEAN KINETIC ENERGY IN CHAMBER (ERG)</td>
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<tr>
<td>DY(7)</td>
<td>YES</td>
<td>NO</td>
<td>TURBULENT KINETIC ENERGY IN CHAMBER (ERG)</td>
</tr>
<tr>
<td>DY(8)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - PISTON TOP (KJ)</td>
</tr>
<tr>
<td>DY(9)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - CYLINDER HEAD (KJ)</td>
</tr>
<tr>
<td>DY(10)</td>
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<td>NO</td>
<td>HEAT TRANSFER - CYLINDER WALL (KJ)</td>
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<tr>
<td>DY(11)</td>
<td>YES</td>
<td>NO</td>
<td>CYLINDER TEMPERATURE (K)</td>
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<tr>
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<td>NO</td>
<td>CYLINDER PRESSURE (ATM)</td>
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<tr>
<td>DY(16)</td>
<td>YES</td>
<td>NO</td>
<td>TOTAL WORK TRANSFER (KJ)</td>
</tr>
<tr>
<td>DY(17)</td>
<td>YES</td>
<td>NO</td>
<td>TOTAL ENTHALPY EXHAUSTED (KJ)</td>
</tr>
</tbody>
</table>

| DYP(1)    | NO    | YES    | RATE AT WHICH MASS IS INDUCTED THROUGH THE INTAKE VALVE (G/DEG) |
| DYP(2)    | NO    | YES    | RATE AT WHICH MASS IS EXHAUSTED THROUGH THE EXHAUST VALVE (G/DEG) |
| DYP(6)    | NO    | YES    | RATE OF CHANGE OF MEAN KINETIC ENERGY (KJ/DEG) |
| DYP(7)    | NO    | YES    | RATE OF CHANGE OF TURBULENT KINETIC ENERGY (KJ/DEG) |
| DYP(8)    | NO    | YES    | RATE OF HEAT TRANSFER - CYLINDER HEAD (KJ/DEG) |
| DYP(9)    | NO    | YES    | RATE OF HEAT TRANSFER - PISTON TOP (KJ/DEG) |
| DYP(10)   | NO    | YES    | RATE OF HEAT TRANSFER - CYLINDER WALL (KJ/DEG) |
| DYP(11)   | NO    | YES    | RATE OF CHANGE OF CYLINDER TEMPERATURE (K/DEG) |
| DYP(12)   | NO    | YES    | RATE OF CHANGE OF CYLINDER PRESSURE (ATM/DEG) |
| DYP(16)   | NO    | YES    | RATE OF TOTAL WORK TRANSFER (KJ/DEG) |
| DYP(17)   | NO    | YES    | RATE AT WHICH TOTAL ENTHALPY IS EXHAUSTED (KJ/DEG) |

REMARKS
NONE

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
UTHMRO  BTRANS  MFLRT
CSAADV  EVACD

METHOD
SEE REPORT
SUBROUTINE EXAUST (DT, DY, DYP)

LOGICAL FIRE
REAL*8 DT, DY(20), DYP(20)
REAL MW, MWIM, MWIMM, KINVIS, MASS, MDOT, MDOTFR, MSTART, MACRSC
DIMENSION Y(20), YP(20)
COMMON/EPARAM/ BORE, STROKE, CONRL, CYLCA, CSATDC, CMRTO, CLVTDC
COMMON/BURN/ SPBURN, FIRE, RPM
COMMON/HTRC/ CONHT, EXPHT
COMMON/TEMPS/ TPSTON, THEAD, TCW
COMMON/DTDTH/ ESPDI
COMMON/MANFP/ PIM, TIM, EGR, PEM, MSTART
COMMON/TIMES/ TIVO, TEVC, TIVC, TSPARK, TEVO
COMMON/IMTHP/ HIM, MWIM, CIM, RHOIM
COMMON/FLAG/ INFAG
COMMON/HEATS/ CVHTRN, HTRCOE, HTPAPI, HTPAD, HTPACW, HTRAPI,
& HTRADH, HTRACW, THTRAN, QFRPI, QFRHO, QFRCW
COMMON/TURBO/ CBETA, MACRSC, UPRIME, VMKE
COMMON/VALVE/ VIV, VEV
COMMON/RHMAS/ RHO, MASS, VOLUME, H, GAMMA

VEV = 0.0

DO 10 I = 1, 20
   Y(I) = DY(I)
10 CONTINUE
   T = DT
DO 20 I = 1, 20
   YP(I) = 0.0
20 CONTINUE

FIND THERMODYNAMIC AND TRANSPORT PROPERTIES IN CYLINDER

RESFRK = 1.
IF (.NOT. FIRE) RESFRK = 0.0
CALL THERMO (T, Y(11), Y(12), RESFRK, H, CSUBP, CSUBT,
& RHO, DHRDHT, DHRDHP, GAMMA, MW, ADUMY, BDUMY, GDUMY, HDUMY)
IF (FIRE) CALL BTRANS (Y(11), GAMMA, CSUBP, DYNVIS, THRCON)
IF (.NOT. FIRE) CALL UTRANS (Y(11), DYNVIS, THRCON)
KINVIS = DYNVIS/RHO
MASS = MSTART + Y(1) - Y(2)

IS EXHAUST VALVE STILL OPEN ?

IF (T .GE. TEVC) GO TO 50
   YES IT IS.
   ANY FLOW ACROSS IT ?
   IF (Y(12) - PEM) 30, 50, 40
   YES, FLOW INTO CYLINDER.
   FIND CD AND AREA FOR EXHAUST VALVE.

30 PR = PEM/Y(12)
CALL EVACD (T, PR, AREA, CD)

FIND MASS FLOW RATE.

CALL MFLRT (CD, AREA, PEM, MW, Y(11), Y(12), GAMMA, FRAEV)

CALCULATE RATES DUE TO THIS FLOW.

YP(2) = -FRAEV
IF (AREA .LE. 0.0) GO TO 35
VEV = -FRAEV/(RHO*AREA)

0.0 required due to overestimation of kinetic energy when backflow through the exhaust valve exists. This error caused a tremendously high characteristic velocity and therefore high heat transfer at low loads and low speeds.

Modified by R. T. Chang 9 July 1987

35 YP(6) = 0.0 * .5 * FRAEV * VEV*VEV
GO TO 50

FLOW FROM CYLINDER INTO EXHAUST MANIFOLD.
FIND AREA AND CD FOR EXHAUST VALVE.

40 PR = Y(12)/PEM
CALL EVACD(T, PR, AREA, CD)

FIND MASS FLOW RATE.

CALL MFLRT(CD, AREA, Y(12), MW, Y(11), PEM, GAMMA, FRAEV)

CALCULATE RATES DUE TO THIS FLOW

YP(2) = FRSV
IF (AREA .LE. 0.0) GO TO 45
VEV = FRAEV/(RHO*AREA)
45 YP(6) = -FRAEV * ( (2/MASS)
YP(7) = --FRAEV * (Y(7). ASS

FIND SURFACE AREAS AND VOLUME OF CHAMBER

50 CALL CSAVDV(T, AHEAD, APSTON, ACW, VOLTIME, DVT)
MACRS = VOLUME/(3.1415 * BORE * BORE/4.)
IF (MACRS .GE. (BORE/2.)) MACRS = BORE/2.
YP(6) = YP(6) - .3387 * CBETA/MACRS * Y(6) * SQRT(Y(7)/MASS)
YP(7) = YP(7) + .3387 * CBETA/MACRS * Y(6) * SQRT(Y(7)/MASS)
      - .5443 * Y(7)/MACRS * SQRT(Y(7)/MASS)
MDT = -YP(2)

CHARACTERISTIC VELOCITY IN CYLINDER; (CM/SEC).

PI = 3.141592654
CONSTR = CONRL/STROKE
SINTH = SIN( T*PI/180. )
COSTH = COS( T*PI/180. )
VONVP = ABS( PI * SINTH - ( 1. / COSTH/SQRT( 4. * CONSR*CONSTR & - SINTH*SINTH ) )/2. )
VPMEAN = STROKE/(180. * ESPDI)
VPSTO = VPMEAN * VONVP
VMAKE = SQRT( 2. * Y(6)/MASS )
UPRIME = SQRT( 0.666667 * Y(7)/MASS )

AND VELOCITY TERM DUE TO BLOWDOWN

VBDOWN = 4 * YP(2)/( 3.1415925 * RHO * BORE * BORE )
IF ( YP(2) .LT. 0.0 ) VBDOWN = 0.0
CVHTRN = SQRT( 0.25*VPSTO*VPSTO + VMAKE*VMAKE + UPRIME*UPRIME + & VMAKE*WSH+VBDOWN )

CALCULATE HEAT TRANSFEI. KATES

.RTCD = CONH:*(( CVHTRN+MACRS/KINVIS)**EXPHT )*THRCD/MACRS
HTPA = HTROE * ( Y(11) - 1*STIN )
HTPAHD = HTROE * ( Y(11) - THEAF )
HTPACW = HTROE * ( Y(11) - TCK )

H'RAPI = APSTON * HTPAPI
HTRAHD = AHEAD * HTPAH
HTRACW = ACW * HTPACW

THTRAN = HTRAPI + HTRAHD + HTRACW
QFRPI = 0.0
QFRHD = 0.0
QFRCW = 0.0
IF ( (CONHT .LE. 0.0) .OR. (THTRAN .EQ. 0.0) ) GO TO 60
QFRPI = 100. * HTRAPI/THTRAN
QFRHD = 100. * HTRAHD/THTRAN
QFRCW = 100. * HTRACW/THTRAN
C
C     CALCULATE RATES OF CHANGE OF TEMPERATURE AND PRESSURE IN
C     THE CYLINDER. THEN CALCULATE RATE OF DOING WORK.
C
60 YP(11) = - (BDUMY/ADUMY) * ( YP(2)/MASS + DVDT/VOLUME +
   &       THTRAN/(BDUMY*MASS) )
YP(12) = (1./1.01325E+6) * RHO/DRHCP * ( -DVDT/VOLUME
   &       - YP(11)*DRHDT/RHO + MDOT/MASS )
YP(16) = Y(12) * DVDT * .101325E-3
C
HTRCOE = HTRCOE * 1.E-6
HTPAPI = HTPAPI * 1.E-6
HTPAHD = HTPAHD * 1.E-6
HTPACW = HTPACW * 1.E-6
YP(8) = HTRAPI * 1.E-19
YP(9) = HTRAHD * 1.E-10
YP(10) = HTRACW * 1.E-10
THTRAN = HTRACW * 1.E-7 * ESPDI
C
YP(17) = YP(2) * H
VEV = VEV/100.
C
C     CONVERT ALL TIME DERIVATIVES TO RATE PER CRANK
C     ANGLE DEGREE.
C
DO 70 I = 1, 20
   DYP(I) = YP(I) * ESPDI
70 CONTINUE
C
RETURN
END
C
C
Appendix G

Engine Model Spreadsheet
<table>
<thead>
<tr>
<th>constraint</th>
<th>operating indep var</th>
<th>operating conditions</th>
<th>design independent variables</th>
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<td>THROTTLE POSITION</td>
<td>VALVETRAIN TYPE</td>
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<td>kPa</td>
<td>%</td>
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Fuel RON = 98 Fuel LHV = 43.20 Atm. Pr. = 99.15 Atm. T = 298.00 At Dens = 1.1394
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<th>Indicated TORQUE</th>
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