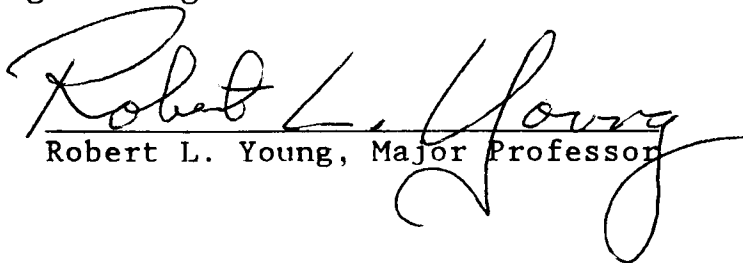
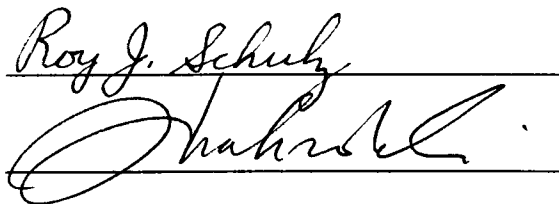
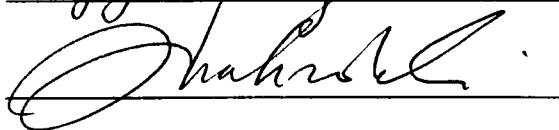


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Robert L. Young, Major Professor

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Date 7/27/88

PERFORMANCE STUDY OF A CONTINUOUSLY-
VARIABLE STROKE INTERNAL
COMBUSTION ENGINE

A Thesis
Presented for the
Master of Science
Degree
The University of Tennessee, Knoxville

Burt Randall Jennings

August 1988

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ABSTRACT

The performance of a continuously-variable stroke internal combustion engine was simulated with a computer model. The engine has a mechanism which provides a different stroke length for each of the four strokes in the engine cycle. Specifically, the variable-stroke engine has a smaller clearance volume at the start of the intake stroke than at the end of the compression stroke, and the expansion stroke is lengthened to get more work from the combustion products.

A conventional engine was also modeled to provide a reference for comparison of the computer results. The models include subroutines for calculation of piston movement, intake and exhaust mass flow, combustion, heat transfer, friction, and shaft power.

The program results indicated that the variable-stroke engine yields a 12% gain in output horsepower. The gain is attributed to the engine's higher volumetric efficiency.

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CHAPTER I

INTRODUCTION

Historical Background

There have been many advances in the technology of spark ignition engines. Many recent advances in production engine design were conceived years ago, and became practical because of new materials and manufacturing technologies, or because of economic changes. Many design concepts are incorporated in laboratory engines, and some of these concepts will be used to improve the production engines of the future.

Some of the more successful advances in the design of spark ignition engines are turbocharging, rotary engines, computer-controlled electronic ignition, multi-valved cylinders, and fuel injection. The future may see production of plastic engines, ceramic cylinders, variable valve timing, variable stroke, and variable compression ratio.

Variable stroke and variable compression ratio engines rely on some type of mechanism to modify the travel of the piston. These mechanisms add cost, weight, and friction to the engines. To date, none of these mechanisms have been incorporated in production engines.

One promising design is the K-cycle engine, which uses opposed pistons and a cam to control the piston positions, resulting in a longer expansion stroke than compression stroke and slow piston motion during combustion (ref. 4, pg. 48). Another very efficient design is Teledyne's AVCR-1360-2 (diesel) engine, which uses hydraulic actuators in the piston to vary compression ratio to respond to changing engine load (ref. 4, pg. 45).

Objective

The objective of this study is to examine a variable stroke engine concept to determine whether it might improve engine efficiency and performance enough to make it a feasible alternative to conventional fixed-stroke design. A small clearance volume at the end of the exhaust stroke is expected to more thoroughly purge the exhaust gases, resulting in the ingestion of a greater mass of reactants during the intake stroke. A longer expansion stroke is expected to deliver more shaft output power from the combustion products.

A computer program will model the operation of a conventional engine and a variable stroke engine of the same size and under the same conditions to reveal the predicted changes in efficiency and performance.

Description

The continuously-variable stroke, reciprocating piston, spark ignition internal combustion engine has a mechanism which varies the piston travel during the four-stroke cycle so that each stroke has a different starting place and a different length. The intake stroke begins with a small clearance volume in the cylinder and ends before the piston reaches the bottom of the cylinder. The compression stroke is shorter than the intake stroke, and when the piston stops at the end of the compression stroke, the cylinder has a larger clearance volume than at the start of the intake stroke. The expansion stroke is longer than the compression stroke and the piston travels to the bottom of the cylinder. The exhaust stroke is longest, and the small cylinder clearance volume at the end of the exhaust stroke marks the start of the next intake stroke.

The variable-stroke engine has the same carburetor and intake design, valving, ignition system, cooling system, and exhaust system as a conventional engine. The four-stroke cycle is also very nearly the same. The difference is in the design of the connecting rods which use the reciprocating motion of the piston to produce torque in the output shaft. The variable-stroke engine has an off-set link connected to a rotating shaft which

continuously varies the relationship of the piston and connecting rod to the crankshaft (see figure 1). The shaft rotates at exactly one-half the speed of the crankshaft, so the variation repeats once for every two rotations of the crankshaft, which is also once for every four-stroke cycle. An alternative way of looking at the concept is to consider both shafts to be crankshafts--one rotating once with every cycle, and the other rotating twice with every cycle.

The variations in the stroke lengths yield advantages to the variable stroke engine. The small clearance volume at the end of the exhaust stroke and the start of the intake stroke reduces the amount of residual exhaust gas which mixes with the fresh air/fuel charge. The charge therefore is cooler and contains a higher percentage of volatile mixture.

The intake stroke is longer than the compression stroke, so for a given compressed volume, the air/fuel charge may have more time to enter through the intake port (provided the speed of the piston during the intake stroke does not increase proportionately.) The increased intake interval reduces the speed of the charge through the carburetor and intake passages, resulting in lower intake friction and improved volumetric efficiency.

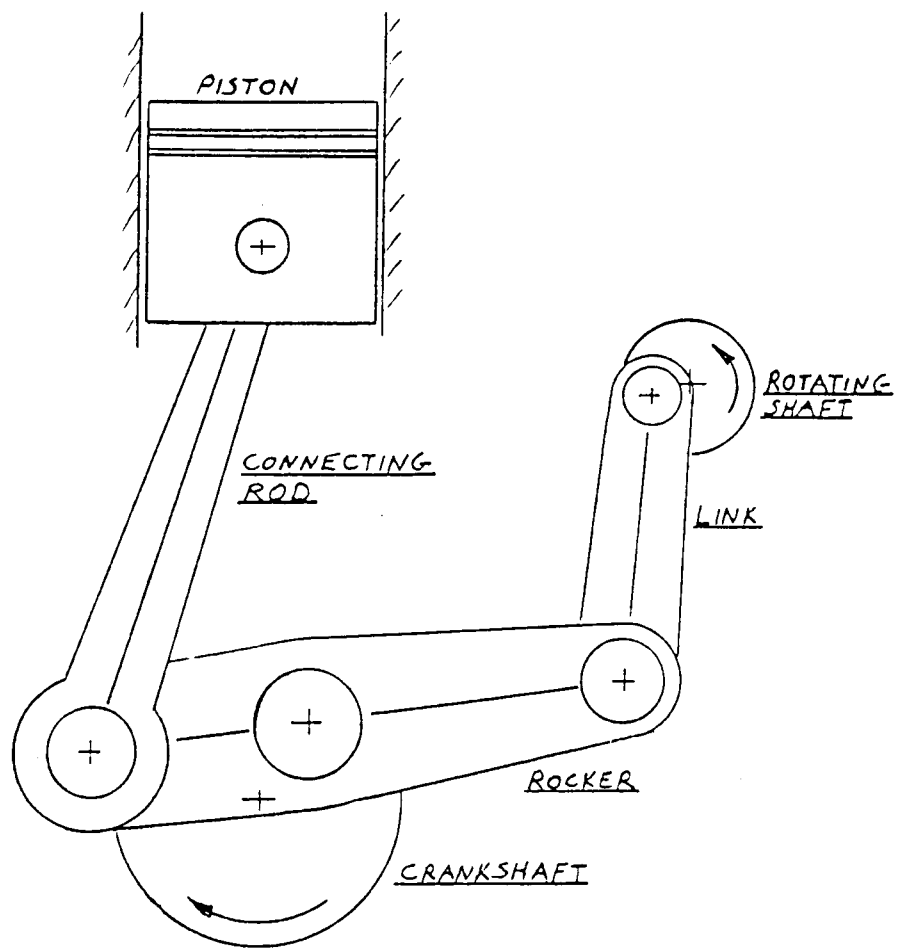


Figure 1. Variable-Stroke Engine Mechanism

The expansion stroke is longer than the compression stroke, which allows more complete expansion of the exhaust products. More available energy of the exhaust products is used to do useful work, and less of it is lost in the exhaust system as heat.

The long exhaust stroke reduces the speed of the exhaust gases through the exhaust port (unless the piston speed during the exhaust stroke increases proportionately), reducing the exhaust friction losses.

The disadvantages of the variable-stroke engine include the added weight, complexity, and cost of the additional shaft and linkages. The long expansion and exhaust strokes increase the heat load on the cylinder walls. The mechanism increases the engine's mechanical friction and adversely affects the dynamic balance of the engine.

CHAPTER II

DISCUSSION OF THE MODEL

The computer model used in this analysis has two parts. One part of the model simulates the performance of a conventional four-stroke internal combustion engine. The second part uses most of the same engine parameters as the first part, but the mechanical linkage is changed to simulate the piston movement of the continuously-variable stroke engine.

The performance results from the conventional engine model were compared to existing experimental data to validate the model's ability to simulate characteristics of actual engine operation. Comparison of the performance results of the variable-stroke model to the conventional engine model was used to demonstrate the predicted differences in the two engines.

Simplifying assumptions were made in both parts of the model which affected the absolute performance results, but they did not significantly affect the differences between the two parts:

1. Carburation: stoichiometric air/fuel ratio with perfect mixing in the intake manifold at atmospheric temperature and pressure.

2. Valving : sinusoidal opening and closing of intake and exhaust valves, dependent only on crankshaft position, with timing optimized for maximum engine output horsepower.
3. Combustion: complete combustion of gasoline into water (gaseous) and carbon dioxide. Spark timing was optimized for maximum engine output horsepower.
4. Mass flow: flow through valves driven only by bulk pressure differentials, no gas mixture inertial effects, and instantaneous complete mixing of all gas constituents in the cylinder. No blow-by leakage occurred past piston rings.
5. Preignition: no preignition of the gas mixture.
6. Output horsepower: no engine power is required to drive auxiliary devices, such as a water pump, alternator, etc.

Other assumptions are identified in the discussions of the program subroutines.

The computer models were developed using the BASIC programming language (see Appendix C and Appendix D for program listings). The program structure is outlined on the flowchart shown in figure 2.

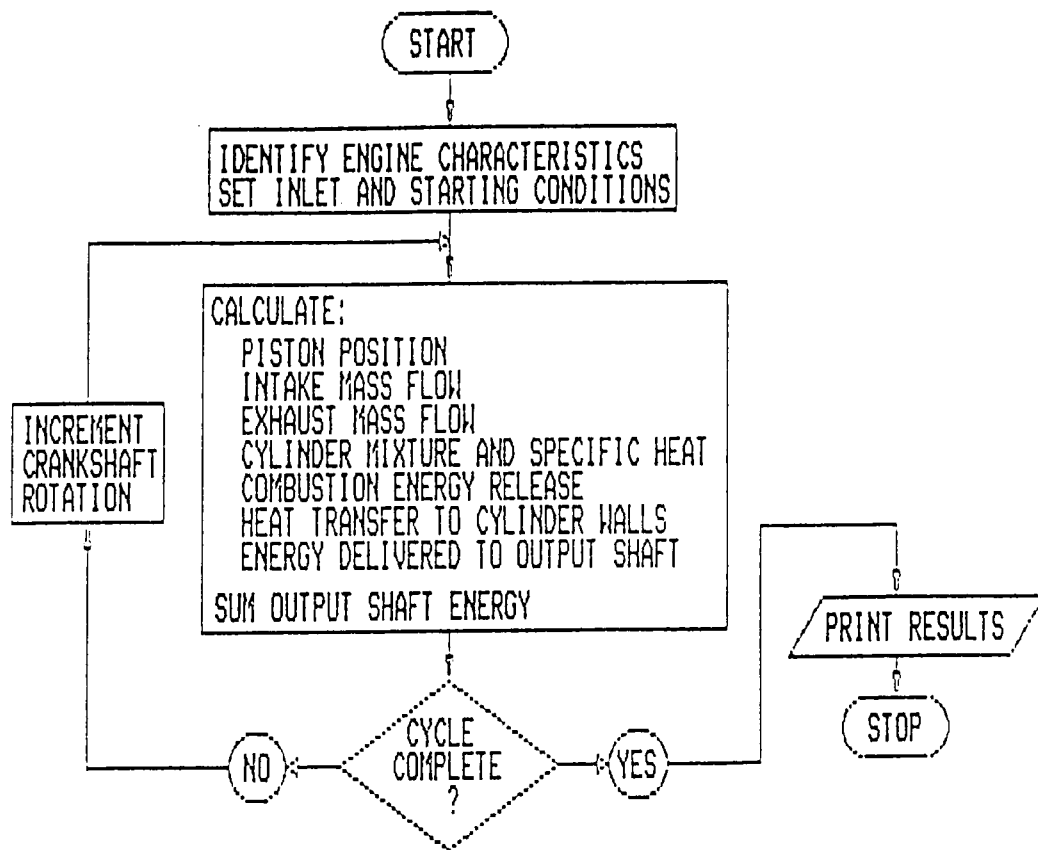


Figure 2. Program Flowchart

Execution of a program began by identifying the geometric parameters of the engine and a set of starting conditions. Rotation of the crankshaft was incremented through a small angle which affected movement of the piston, and the mass and energy transfer were calculated.

Incremental rotation of the crankshaft continued through several four-stroke cycles until the conditions at the start of a cycle were equivalent to the conditions at the end of the cycle, simulating steady-state operation of the engine. Examination of the energy delivered to the output shaft during the last engine cycle indicated the predicted performance of the engine. The starting parameters were varied, and the program was re-executed to optimize the horsepower output of the engine model.

Both parts of the model shared some of the same calculation procedures for evaluating the mass and energy transfer during incremental rotation of the crankshaft. These procedures were grouped into subroutines in the program structure. The following sections describe the function of each of the subroutines. Development of formulas in the text follows the format used in BASIC programming.

Piston Position

The piston position varied in relationship to the position of the crankshaft and the geometry of the connecting linkage. This was the only difference in the calculation procedure between the variable-stroke engine and the conventional engine.

In the conventional engine model (see figure 3), the value of the throw (crankshaft offset) was established by the displacement and the cylinder bore:

$$\text{AREA} = (\text{BORE})^2 \cdot \text{PI} / 4$$

$$\text{DISPL} = \text{STROKE} \cdot \text{AREA}$$

$$\text{THROW} = \text{STROKE} / 2$$

where:

AREA=cylinder cross-sectional area [sq. in.]

BORE=cylinder diameter [inches]

DISPL=total volume swept out by the piston [cubic in.]

PI=3.1416 (constant)

STROKE=total length of piston travel [inches]

THROW=crankshaft offset [inches]

The clearance volume was established by the value of the compression ratio, which is the ratio of the cylinder volume at the start of the compression stroke to the volume at the end of the compression stroke:

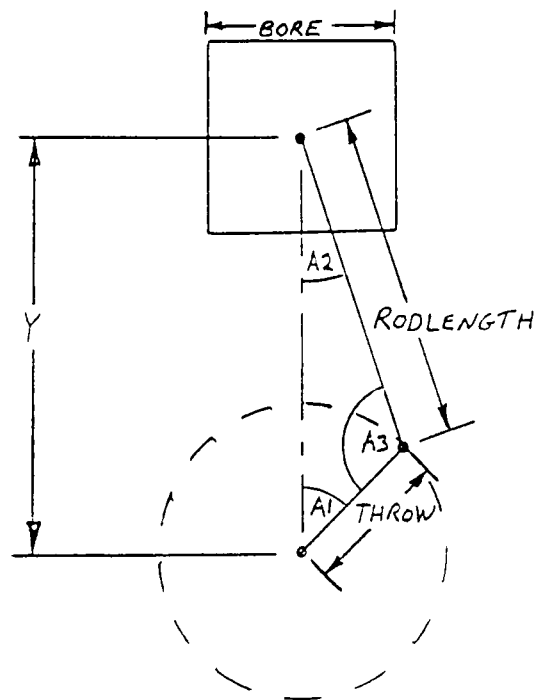


Figure 3. Conventional Engine Dimensional Parameters

$$\text{COMPRESS} = (\text{CLR VOL} + \text{DISPL}) / \text{CLR VOL}$$

rearranging:

$$\text{CLR VOL} = \text{DISPL} / (\text{COMPRESS} - 1)$$

where:

COMPRESS=compression ratio

CLR VOL=clearance volume [cubic in.]

The volume inside the cylinder was always the clearance volume plus the volume swept out as the piston moved from the top-dead-center (TDC) position. At TDC the connecting rod and the crankshaft throw were in-line. As the crankshaft rotated, the piston height decreased in accordance with the projected heights of the connecting rod and crankshaft throw:

$$\text{VOLUME} = \text{CLR VOL} + \text{AREA} * (\text{ROD LENGTH} + \text{THROW} - \text{Y3})$$

$$\text{Y3} = \text{ROD LENGTH} * \cos(\text{A2}) + \text{THROW} * \cos(\text{A1})$$

where:

VOLUME=the instantaneous cylinder volume [cubic in.]

ROD LENGTH=the connecting rod length [inches]

Y3=the instantaneous piston pin height [inches]

A2=the angle of the connecting rod from vertical
[radians]

A1=the angle of the crankshaft throw from vertical
[radians]

Angle A1 was incremented by the main program. Angles A2 and A3 were found geometrically:

$$\text{SIN}(A2) = \text{THROW} * \text{SIN}(A1) / \text{RODLENGTH}$$

$$A3 = \text{PI} - A1 - A2$$

Angle A3 was used in another subroutine to calculate a value for the friction in the crankpin bearing. Special programming techniques were required to make the values for A3 continuous throughout the cycle and between cycles.

The instantaneous velocity and acceleration of the piston were also used in another subroutine for calculations of kinetic energy and friction losses. The instantaneous velocity was approximated by the rate of change of two incremental piston positions:

$$\text{AVGVEL} = (Y3 - Y2) / \text{TIME}$$

$$\text{TIME} = (\text{INCREMENT} / 360) / (\text{RPM} * 60)$$

where:

AVGVEL=average instantaneous piston velocity [in./sec.]

Y3=the instantaneous piston pin height [inches]

Y2=the value of Y3 prior to the last incremental movement [inches]

TIME=duration of the incremental movement [seconds]

INCREMENT=the incremental change in crankshaft rotation [degrees]

RPM=rotational velocity of the crankshaft [rotations per minute]

The instantaneous acceleration was approximated by the rate of change in velocity over three incremental piston positions:

$$\text{AVGACC} = ((Y2 - Y1) / \text{TIME} + (Y3 - Y2) / \text{TIME}) / \text{TIME}$$

rearranging:

$$\text{AVGACC} = (Y3 - 2 * Y2 + Y1) / (\text{TIME}^2)$$

where:

Y3=the instantaneous piston pin height [inches]

Y2=the value of Y3 prior to the last incremental movement [inches]

Y1=the value of Y3 prior to the last two incremental movements [inches]

The use of previous and subsequent piston pin heights would have yielded a better approximation for instantaneous acceleration than using two previous values, but the BASIC program retains previously calculated values more efficiently than it can project subsequent values. The value of AVGACC lagged the rest of the model by one increment, but this was not significant in the final outcome. The kinetic energy of the piston had only an instantaneous effect on shaft output. The net effect over the whole cycle was zero, because the inertia of a moving mass in a steady cycle cannot generate or consume energy.

The variable-stroke engine (see figure 4) has extra links which complicate calculation of the piston position.

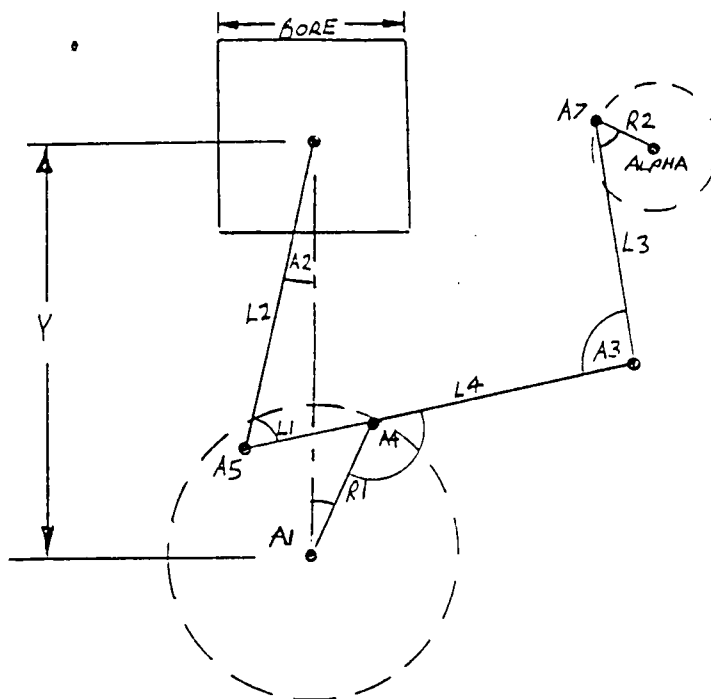


Figure 4. Variable-Stroke Engine Dimensional Parameters

The values for bore, displacement, and compression ratio were the same as those used in the conventional engine model. It follows that the clearance volume was also identical.

The linkage lengths were assigned non-dimensional values by trial and error to obtain a mechanism which generated the desired continuously-variable stroke action. The motion of the piston was calculated to find the maximum travel during the compression stroke. The linkage lengths were then given dimensions based on the actual length of the compression stroke.

The details of the piston position calculation are shown in the BASIC program in Appendix D., page 126, lines 970 through 1290, and are not reproduced here because of their geometric complexity. The approach was similar to that used in the conventional engine model, except that the bearing center lines were assigned Cartesian coordinates to help track their relative positions. Calculations of AVGVEL and AVGACC followed the same techniques used for the conventional engine model.

Intake

Movement of the piston during the intake stroke lowers the pressure in the cylinder below atmospheric pressure. The pressure differential draws the mixture

through the intake valve, which restricts the flow in accordance with the instantaneous position of the valve. The restriction was approximated as an orifice in this model.

Subsonic flow of a compressible fluid through an orifice has the following functional relationship (ref. 7, pg. 3.8.2-4):

$$W = K * Y * A * C * \text{SQR}((P1 - P2) * \text{DENSITY} * GC)$$

where:

W=mixture flow rate [lbm/sec.]

K=proportionality constant

Y=fluid net expansion factor

A=orifice area [sq.inches]

C=orifice discharge coefficient

P1=upstream pressure [psia]

P2=downstream pressure [psia]

DENSITY=upstream gas density [lbm/cu.inch]

GC=unit conversion factor [in. lbm/lbf sec.²]

The quantity of gas, in pound-moles, which flows through an orifice during an incremental time is the product of the flow rate and the duration:

$$\text{FLOW} = W * \text{TIME} / \text{MW}$$

where:

FLOW=the flow quantity [moles]

TIME=incremental duration [sec.]

MW=gas molecular weight [lbm/mole]

The upstream gas density was calculated from the ideal gas equation of state:

$$\text{DENSITY} = P_1 * MW / R_0 / T / 778.16 / 12$$

where:

R₀=universal gas constant [BTU/mole °R]

T=upstream gas temperature [°R]

unit conversions:

778.16 ft.-lbf./BTU

12 in./ft.

combining equations:

$$\text{FLOW} = \text{TIME} / \text{MW} * K * Y * A * C * \text{SQR}((P_1 - P_2) * P_1 * GC * MW / R_0 / T)$$

rearranging:

$$\text{FLOW} = (K * Y * A * C * \text{SQR}(GC / R_0)) * \text{TIME} * \text{SQR}((P_1 - P_2) * P_1 / MW / T)$$

A constant was defined to collect the terms which are independent of temperature, pressure, and the gas molecular weight:

$$\text{CONST} = K * Y * A * C * \text{SQR}(\text{GC} / \text{R0})$$

where:

CONST=orifice flow constant

K=proportionality constant

Y=fluid net expansion factor

A=orifice area

C=orifice discharge coefficient

GC=unit conversion factor

R0=universal gas constant

The fluid net expansion factor Y and the orifice discharge coefficient C are not actually independent of the temperature, pressure, and gas molecular weight. The dependence is complex; fortunately, the values of Y and C were nearly constant over the range of conditions experienced in this model, so it was acceptable to approximate them as independent constants. Substitution of the orifice flow constant into the flow equation:

$$\text{FLOW} = \text{CONST} * \text{TIME} * \text{SQR}((\text{P1} - \text{P2}) * \text{P1} / \text{MW} / \text{T})$$

This formula shows that the moles which flow through the valve increase with a larger orifice flow constant, longer flow duration, and greater pressure differential. These dependencies are intuitively obvious. Less obvious is the dependence of the flow quantity on the upstream

pressure and the inverse relationship to molecular weight and upstream temperature. These terms define the density of the upstream gas, and the resulting relationship to flow rate is consistent with observation.

Choked flow through the valve occurs when the pressure ratio across the valve exceeds a critical value determined by the ratio of gas specific heats (ref. 7, pg. 3.3.9-1):

$$\text{CRITICAL} = (2/(K+1))^{(K/(K-1))}$$

$$K = C_P / C_V$$

where:

CRITICAL = the downstream to upstream pressure ratio which causes sonic flow through the orifice

K = the ratio of gas specific heats

C_P = intake mixture specific heat at constant pressure
[BTU/mole °R]

C_V = intake mixture specific heat at constant volume
[BTU/mole °R]

If a choked flow condition was encountered at the intake valve, the pressure differential corresponding to the critical pressure ratio was substituted for the actual pressure differential in the equation for FLOW.

An orifice flow constant (CONST) is heavily dependent on the size and shape of the orifice. The value will be zero when the intake valve is closed, but the value when

the valve is open depends on the valve geometry and the instantaneous positions of the camshaft and lifter. Actual calculation of CONST is complex and was not the concern of this model. Instead of calculating the orifice flow constant for a given intake valve design, the value of CONST was assumed to vary as a function of crankshaft angle. (One could theoretically design a valve and actuator which would physically provide these values.):

If $\text{ANGLE} < \text{OPEN}$, then $\text{CONST} = 0$.

If $\text{OPEN} \leq \text{ANGLE} \leq \text{CLOSE}$, then:

$\text{CONST} = \text{MAX} * (1 + \sin((\text{ANGLE} - \text{OPEN}) / (\text{CLOSE} - \text{OPEN}) * \pi * 2 - \pi / 2)) / 2$.

If $\text{CLOSE} < \text{ANGLE}$, then $\text{CONST} = 0$.

where:

CONST=orifice flow constant

MAX=value of CONST with fully-open valve

ANGLE=instantaneous crankshaft angle [degrees]

OPEN=crankshaft angle where the intake valve starts opening [degrees]

CLOSE=crankshaft angle where the intake valve closes [degrees]

This function satisfies the following boundary conditions:

If $\text{ANGLE} = \text{OPEN}$, then $\text{CONST} = 0$.

If $\text{ANGLE} = \text{CLOSE}$, then $\text{CONST} = 0$.

If $ANGLE = (OPEN + CLOSE) / 2$, then $CONST = MAX$.

The sine nature of the function models the behavior of a physical mass accelerating and decelerating without discontinuities with respect to time.

Ideally, the valve would open fully at the start of the intake stroke, and it would close instantaneously at the beginning of the compression stroke. Design limitations require the opening to start before the start of the intake stroke to maximize the volumetric efficiency. This complicates the model because $CLOSE < OPEN$ (in the model the value of $ANGLE$ was zero at the piston top-dead center position before the intake stroke and 720 degrees at the end of the exhaust stroke). Redefinition of the ranges and the corrections to the $CONST$ calculation are as follows:

If $ANGLE < CLOSE$, then replace $(ANGLE - OPEN) / (CLOSE - OPEN)$ with $(ANGLE + 720 - OPEN) / (CLOSE + 720 - OPEN)$.

If $CLOSE < ANGLE < OPEN$, then $CONST = 0$.

If $OPEN < ANGLE$, then replace $(ANGLE - OPEN) / (CLOSE - OPEN)$ with $(ANGLE - OPEN) / (CLOSE + 720 - OPEN)$.

The following substitutions were made in the flow equation for the intake valve:

FLOW=INFLOW
 CONST=INCONST
 P1=ATMPRESS
 P2=PRESS
 MW=MWI
 T=ATMTEMP
 CRITICAL=ICRITICAL
 CP=ICP
 CV=ICV
 K=IK
 MAX=INMAX
 OPEN=INOPEN
 CLOSE=INCLOSE

The resulting flow through the intake valve became the following:

$$\text{INFLOW} = \text{INCONST} * \text{TIME} * \text{SQR}((\text{ATMPRESS} - \text{PRESS}) * \text{ATMPRESS} / \text{MWI} / \text{ATMTEMP})$$

$$\text{INCONST} = \text{INMAX} * (1 + \text{SIN}((\text{ANGLE} - \text{INOPEN}) / (\text{INCLOSE} - \text{INOPEN}) * \text{PI} * 2 - \text{PI} / 2)) / 2$$

where:

INFLOW=the flow quantity through the intake valve
[moles]

INCONST=orifice flow constant of the intake valve

TIME=incremental duration [sec.]
ATMPRESS=atmospheric (upstream) pressure [psia]
PRESS=cylinder (downstream) pressure [psia]
MWI=intake mixture molecular weight [lbm/mole]
ATMTEMP=atmospheric (upstream) mixture temperature [$^{\circ}$ R]
INMAX=value of INCONST with the intake valve fully-open
ANGLE=instantaneous crankshaft angle [degrees]
INOPEN=crankshaft angle where intake valve starts
opening [degrees]
INCLOSE=crankshaft angle where intake valve is closed
[degrees]

When the intake valve starts opening before the start of the intake stroke, the pressure in the cylinder is above the atmospheric pressure. The gases in the cylinder, which are the burned products of combustion, escape through the intake valve into the intake manifold. A small quantity of reverse flow is tolerated because the early start of valve opening yields a higher volumetric efficiency over the total duration of the intake stroke.

The quantity of the reverse flow was calculated by the same equation as the intake flow, except that the upstream conditions were the conditions in the cylinder:

$$\text{REVFLOW} = \text{INCONST} * \text{TIME} * \text{SQR}((\text{PRESS} - \text{ATMPRESS}) * \text{PRESS} / \text{MW} / \text{TEMP})$$

where:

REVFLOW=the reverse flow quantity [moles]

INCONST=the intake valve flow constant

PRESS=cylinder (upstream) pressure [psia]

ATMPRESS=atmospheric (downstream) pressure [psia]

TIME=incremental duration [sec.]

MW=cylinder gas mixture molecular weight [lbm/mole]

TEMP=cylinder gas mixture temperature [$^{\circ}\text{R}$]

The gas which reverse-flowed through the intake valve expanded in the intake manifold to atmospheric pressure. Assuming that the valve throttled the flow adiabatically, the temperature of the gas in the manifold was still equal to TEMP. In an actual manifold the reverse-flow gas would mix with the fresh in-coming charge, but in this model the reverse-flow gas was treated as isolated.

The quantity of reverse-flow gas is NSUMR [moles]. When the pressure in the cylinder descended below atmospheric pressure, the NSUMR gas returned to the cylinder through the intake valve before fresh charge was ingested, until the quantity of NSUMR was depleted. The flow rate of returning gas mixture was defined by the same equation which characterized INFLOW and REVFLOW:

$$\text{RETFLOW} = \text{INCONST} * \text{TIME} * \text{SQR}((\text{ATMPRESS} - \text{PRESS}) * \text{ATMPRESS} / \text{MW} / \text{TEMP})$$

where:

RETFLOW=the returning flow quantity [moles]

INCONST=orifice flow constant of the intake valve

TIME=incremental duration [sec.]

ATMPRESS=atmospheric (upstream) pressure [psia]

PRESS=cylinder (downstream) pressure [psia]

MW=cylinder gas mixture molecular weight [lbm/mole]

TEMP=cylinder gas mixture temperature [$^{\circ}\text{R}$]

Exhaust

Pressure in the cylinder and movement of the piston during the exhaust stroke pushes the burned gas mixture through the exhaust valve, which restricts the flow. The exhaust restriction was approximated as an orifice using the same assumptions as for the intake valve. The equation previously derived for the flow through the intake valve is as follows:

$$\text{FLOW} = \text{CONST} * \text{TIME} * \text{SQR}((\text{P1} - \text{P2}) * \text{P1} / \text{MW} / \text{T})$$

where:

FLOW=the flow quantity [moles]

CONST=orifice flow constant

TIME=incremental duration [sec.]

P1=upstream pressure [psia]

P2=downstream pressure [psia]

MW=gas molecular weight [lbm/mole]

T=upstream gas temperature [$^{\circ}$ R]

The value of CONST for the exhaust valve varied as a function of crankshaft angle:

If $\text{ANGLE} < \text{OPEN}$, then $\text{CONST} = 0$.

If $\text{OPEN} \leq \text{ANGLE} \leq \text{CLOSE}$, then:

$\text{CONST} = \text{MAX} * (1 + \sin((\text{ANGLE} - \text{OPEN}) / (\text{CLOSE} - \text{OPEN}) * \pi * 2 - \pi / 2)) / 2$.

If $\text{CLOSE} < \text{ANGLE}$, then $\text{CONST} = 0$.

where:

CONST=orifice flow constant

MAX=value of CONST with fully-open valve

ANGLE=instantaneous crankshaft angle [degrees]

OPEN=angle where valve starts opening [degrees]

CLOSE=angle where valve is closed [degrees]

Ideally, the exhaust valve would open fully at the start of the exhaust stroke, and it would close instantaneously at the beginning of the intake stroke. Usually the valve timing is set to close the exhaust valve shortly after the start of the intake stroke to maximize the discharge of the burned gas, allowing more fresh charge ingestion during the intake stroke. This complicated the model because $\text{CLOSE} < \text{OPEN}$, and the ANGLE ranges of

application were no longer valid. Redefinition of the ranges and the corrections to the CONST calculation were as follows:

If $\text{ANGLE} < \text{CLOSE}$, then replace $(\text{ANGLE} - \text{OPEN}) / (\text{CLOSE} - \text{OPEN})$ with $(\text{ANGLE} + 720 - \text{OPEN}) / (\text{CLOSE} + 720 - \text{OPEN})$.

If $\text{CLOSE} < \text{ANGLE} < \text{OPEN}$, then $\text{CONST} = 0$.

If $\text{OPEN} < \text{ANGLE}$, then replace $(\text{ANGLE} - \text{OPEN}) / (\text{CLOSE} - \text{OPEN})$ with $(\text{ANGLE} - \text{OPEN}) / (\text{CLOSE} + 720 - \text{OPEN})$.

The following substitutions were made in the flow equations for the exhaust valve:

$\text{FLOW} = \text{EXFLOW}$

$\text{CONST} = \text{EXCONST}$

$P1 = \text{PRESS}$

$P2 = \text{ATMPRESS}$

$\text{MW} = \text{MW}$

$T = \text{TEMP}$

$\text{MAX} = \text{EXMAX}$

$\text{OPEN} = \text{EXOPEN}$

$\text{CLOSE} = \text{EXCLOSE}$

The resulting flow through the exhaust valve was as follows:

$\text{EXFLOW} = \text{EXCONST} * \text{TIME} * \text{SQR}((\text{PRESS} - \text{ATMPRESS}) * \text{PRESS} / \text{MW} / \text{TEMP})$

$$\text{EXCONST} = \text{EXMAX} * (1 + \sin((\text{ANGLE} - \text{EXOPEN}) / (\text{EXCLOSE} - \text{EXOPEN}) * \pi * 2 - \pi / 2)) / 2$$

where:

EXFLOW=the flow quantity through the exhaust valve
[moles]

EXCONST=orifice flow constant of the exhaust valve

TIME=incremental duration [sec.]

ATMPRESS=atmospheric (downstream) pressure [psia]

PRESS=cylinder (upstream) pressure [psia]

MW=cylinder gas mixture molecular weight [lbm/mole]

TEMP=cylinder gas mixture temperature [$^{\circ}\text{R}$]

EXMAX=value of EXCONST with fully-open valve

ANGLE=instantaneous crankshaft angle [degrees]

EXOPEN=crankshaft angle where exhaust valve starts
opening [degrees]

EXCLOSE=crankshaft angle where exhaust valve is closed
[degrees]

Mass Flow

The quantity of gas in the cylinder was increased by the amount entering through the intake valve and was decreased by the amount exiting through the exhaust valve. The mole fraction of the constituents was determined by the total number of moles of each gas in the cylinder before the incremental mass flow and the quantity and relative composition of the flowing gases.

The mole fractions of the INFLOW gas have been defined by the assumption of a stoichiometric mixture with a theoretical mixture of nitrogen to oxygen in dry air. Gasoline was represented in this model by C7H17 (ref. 4, pg. 152):

$$NC7H17 = NC7H17 + XC7H17I * INFLOW$$

$$NN2 = NN2 + XN2I * INFLOW$$

$$NO2 = NO2 + XO2I * INFLOW$$

$$NCO2 = NCO2 + XCO2I * INFLOW$$

$$NH2O = NH2O + XH2OI * INFLOW$$

where:

NC7H17 = number of moles C7H17 in the cylinder

NN2 = number of moles N2 in the cylinder

NO2 = number of moles O2 in the cylinder

NCO2 = number of moles CO2 in the cylinder

NH2O = number of moles H2O in the cylinder

XC7H17I = theoretical mole fraction of C7H17

XN2I = theoretical mole fraction of N2

XO2I = theoretical mole fraction of O2

XCO2I = theoretical mole fraction of CO2

XH2OI = theoretical mole fraction of H2O

INFLOW = total moles of fresh intake gas

The composition of the EXFLOW, REVFLOW, and RETFLOW gases was identical to the composition of the mixture in the cylinder.

$$NC7H17 = NC7H17 + XC7H17 * (RETFLOW - EXFLOW - REVFLOW)$$

$$NN2 = NN2 + XN2 * (RETFLOW - EXFLOW - REVFLOW)$$

$$NO2 = NO2 + XO2 * (RETFLOW - EXFLOW - REVFLOW)$$

$$NCO2 = NCO2 + XCO2 * (RETFLOW - EXFLOW - REVFLOW)$$

$$NH2O = NH2O + XH2O * (RETFLOW - EXFLOW - REVFLOW)$$

where:

XC7H17 = mole fraction of C7H17 in the cylinder

XN2 = mole fraction of N2 in the cylinder

XO2 = mole fraction of O2 in the cylinder

XCO2 = mole fraction of CO2 in the cylinder

XH2O = mole fraction of H2O in the cylinder

RETFLOW = total moles returning through the intake valve

EXFLOW = total moles flowing through the exhaust valve

REVFLOW = total moles escaping through the intake valve

The total moles of gas mixture in the cylinder was the sum of all the constituent gases. The mole fraction of each constituent gas was related to this sum.

$$NSUM = NC7H17 + NN2 + NO2 + NCO2 + NH2O$$

$$XC7H17 = NC7H17 / NSUM$$

$$XN2 = NN2 / NSUM$$

$$XO_2 = NO_2 / NSUM$$

$$XC_{O_2} = NC_{O_2} / NSUM$$

$$XH_2O = NH_2O / NSUM$$

where:

NSUM = total moles of gas mixture in the cylinder

The molecular weight of the gas mixture in the cylinder was a function of molecular weight of the constituent gases and their relative concentrations.

$$MW = MWC_7H_{17} * XC_7H_{17} + MWN_2 * XN_2 + MW_{O_2} * XO_2 + MWC_{O_2} * XC_{O_2} + MWH_2O * XH_2O$$

where:

MW = the equivalent molecular weight of the gas mixture [lbm/mole]

MWC_{7H17} = molecular weight of C_{7H17} [lbm/mole]

MWN₂ = molecular weight of N₂ [lbm/mole]

MW_{O2} = molecular weight of O₂ [lbm/mole]

MWC_{O2} = molecular weight of CO₂ [lbm/mole]

MWH_{2O} = molecular weight of H_{2O} [lbm/mole]

Specific Heat

Calculation of the temperature of the gas mixture in the cylinder was dependent on the magnitude of the energy in the mixture, the quantity of the mixture, and the specific heat capacity of the mixture. The specific heat varied as a function of temperature and composition.

Empirical relationships provide the following fourth order polynomials for calculating the specific heats for pure gases with temperatures between 300 °K and 1000 °K (ref. 5, pg. 32).

For C7H17:

$$CP/R0 = (4.06) + (.061)*T - (1.88E-6)*T^2 + (0)*T^3 + (0)*T^4$$

For CO2:

$$CP/R0 = (2.4008) + (8.74E-3)*T - (6.61E-6)*T^2 + (2E-9)*T^3 + (6.33E-16)*T^4$$

For H2O:

$$CP/R0 = (4.0701) - (.00111)*T + (4.15E-6)*T^2 - (2.96E-9)*T^3 + (8.07E-13)*T^4$$

For N2:

$$CP/R0 = (3.6748) - (.00121)*T + (2.32E-6)*T^2 - (6.32E-10)*T^3 - (2.26E-13)*T^4$$

For O2:

$$CP/R0 = (3.6256) - (.00188)*T + (7.06E-6)*T^2 - (6.76E-9)*T^3 + (2.16E-12)*T^4$$

At temperatures between 1000 °K and 5000 °K, the equations are as follows.

For C7H17:

$$CP/R0 = (4.06) + (.061)*T - (1.88E-6)*T^2 + (0)*T^3 + (0)*T^4$$

For CO2:

$$CP/R0 = (2.4008) + (8.74E-3)*T - (6.61E-6)*T^2 + (2E-9)*T^3 + (6.33E-16)*T^4$$

For H2O:

$$CP/R0 = (4.0701) - (.00111)*T + (4.15E-6)*T^2 - (2.96E-9)*T^3 + (8.07E-13)*T^4$$

For N2:

$$CP/R0 = (3.6748) - (.00121)*T + (2.32E-6)*T^2 - (6.32E-10)*T^3 - (2.26E-13)*T^4$$

For O2:

$$CP/R0 = (3.6256) - (.00188)*T + (7.06E-6)*T^2 - (6.76E-9)*T^3 + (2.16E-12)*T^4$$

The model used temperature in units of degrees Rankine rather than Kelvin, but the preceding formulas require Kelvin degrees, so the following conversion was made:

$$KELV = TEMP * 5 / 9$$

where:

KELV = temperature in degrees Kelvin

TEMP = temperature in degrees Rankine

The first set of equations applies for gas temperatures between 540 °R and 1800 °R; the second set applies for temperatures between 1800 °R and 9000 °R. CP and R0 can have any consistent units. The units in the model were BTU/mole °R.

The specific heat of an ideal gas mixture is a function of the specific heats of the individual gases and their mole fractions in the mixture.

$$CP = CPC7H17 * XC7H17 + CPCO2 * XCO2 + CPH2O *XH2O + CPN2 *XN2 + CPO2 *XO2$$

where:

CP=specific heat of the mixture [BTU/mole °R]

CPC7H17=specific heat of C7H17 [BTU/mole °R]

CPCO2=specific heat of CO2 [BTU/mole °R]

CPH2O=specific heat of H2O [BTU/mole °R]

CPN2=specific heat of N2 [BTU/mole °R]

CPO2=specific heat of O2 [BTU/mole °R]

The coefficients of the individual gas specific heat equations can be combined in the previous equation by combining those coefficients which are multiplied by the same power of temperature (T or KELV).

$$CP/R0 = C0 + C1 * KELV + C2 * KELV^2 + C3 * KELV^3 + C4 * KELV^4$$

where:

R0=universal gas constant [1.98588 BTU/mole °R]

C0=sum of constant coefficients weighted by mole fraction for each individual gas

C1=sum of first order coefficients weighted by mole fraction for each individual gas [1/°K]

C2=sum of second order coefficients weighted by mole fraction for each individual gas [1/°K^2]

C3=sum of third order coefficients weighted by mole fraction for each individual gas [1/°K^3]

C4=sum of fourth order coefficients weighted by mole fraction for each individual gas [1/°K^4]

The constant volume specific heat is the difference between the constant pressure specific heat and the universal gas constant.

$$CV = CP - R_0$$

where:

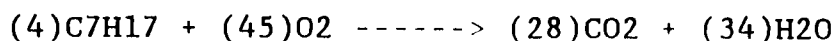
CV=constant volume specific heat [BTU/mole °R]

CP=constant pressure specific heat [BTU/mole °R]

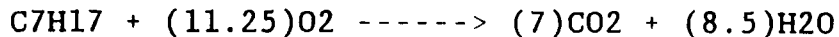
Combustion

Combustion of the fuel in the cylinder alters the composition of the mixture and converts chemical energy into internal energy. Combustion is not instantaneous, but is distributed over several degrees of crankshaft rotation.

The stoichiometric chemical equation for combustion of gasoline may be represented as follows:



or:

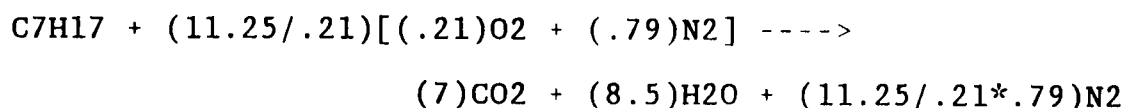


Every 12.25 moles of reactants yields 15.5 moles of combustion products.

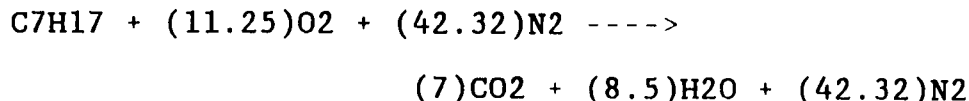
Simplifications were made in choosing this chemical equation for combustion of gasoline. Gasoline is a blend of different hydrocarbons and additives. C_7H_{17} is the

equivalent chemical formula for gasoline used in this model. High temperature dissociation of gases and other chemical processes occur in a spark ignition engine, but they were not considered in this model.

The oxygen which reacts with the gasoline is obtained from air. Assuming that air is composed of 21 percent molecular oxygen and 79 percent molecular nitrogen, the following chemical equation describes the theoretical combustion of gasoline in air:



or:



Assuming that the carburetor of the model engine provides the stoichiometric mixture of air and gasoline, the intake mixture will contain 11.25 moles of oxygen and 42.32 moles of nitrogen for every mole of gasoline. The air-fuel mass ratio of this mixture is found by multiplying by the molecular weights:

$$\text{AF} = (11.25 * \text{MW}_{\text{O}_2} + 42.32 * \text{MW}_{\text{N}_2}) / \text{MWC}_7\text{H}_{17}$$

$$\text{AF} = (11.25 * 32 + 42.32 * 28) / 101$$

$$\text{AF} = 15.3$$

where:

AF=air-fuel mass ratio

MW_{O2}=molecular weight of oxygen [lbm/mole]

MW_{N2}=molecular weight of nitrogen [lbm/mole]

MW_{C7H17}=molecular weight of gasoline [lbm/mole]

The theoretical mixture entered the cylinder through the intake valve and mixed with the gas which was already in the cylinder from the previous cycle. The mixture then contained not only the gasoline and oxygen reactants, but it also contained gaseous nitrogen and the products of combustion, gaseous water and carbon dioxide. The ratio of gasoline to oxygen was unaffected by the presence of the other gases and remained constant throughout the cycle. The quantity of water and carbon dioxide in the cylinder increased as combustion proceeded.

The energy released by combustion was in direct proportion to the quantity of gasoline which reacted during an incremental step of the cycle. Assuming that the exhaust water is in a gaseous state, and that the gasoline and air are initially unmixed, the heat of combustion is given in the literature as 44.51 kilojoules per gram of gasoline (ref. 4, pg. 152). Converting units:

$$44.51 \text{ [kJ/g]} * 453.6 * 101 / 1.055 = 1932856 \text{ [BTU/mole]}$$

unit conversions:

453.6 grams/lbm

101 lbm/mole (C7H17)

1.055 kJ/BTU

The rate of combustion in an engine cylinder will be influenced by the mixture temperature and pressure, the geometry of the cylinder, and the composition and distribution of the mixture in the cylinder. Analysis of these effects is very complex and was not critical in this model. The rate of combustion was approximated by a cosine relationship to the crankshaft angle (ref. 4, pg. 80):

$$X = (1 - \cos((\text{ANGLE} - \text{SPARKANGLE}) * \pi / \text{BURNANGLE})) / 2$$

where:

X = the fraction of reactants which are burned

ANGLE = the instantaneous crankshaft angle [degrees]

SPARKANGLE = the crankshaft angle where combustion begins
[degrees]

BURNANGLE = the total duration of combustion [degrees]

The duration of combustion (BURNANGLE) was estimated by an empirical relationship (ref. 2, page 110):

$$\text{BURNANGLE} = 37 + (\text{RPM} / 120)$$

where:

RPM = rotational velocity of the crankshaft
[rotations/minute]

The value of X varied from zero, for crankshaft angles less than SPARKANGLE, to one, for angles greater than the sum of SPARKANGLE and BURNANGLE. The quantity of unburned fuel remaining in the cylinder at any instant during the combustion process was given by the following equation:

$$\text{REMFUEL} = \text{FUEL} * (1 - X)$$

where:

REMFUEL=remaining quantity of unburned fuel [moles]

FUEL=quantity of fuel prior to start of combustion
[moles]

X=the fraction of reactants which are burned

The quantity of fuel burned during an incremental rotation of the crankshaft was the difference between the quantity of fuel present at the start of the increment and the quantity remaining after the increment:

$$\text{BURN} = \text{NC7H17} - \text{REMFUEL}$$

where:

BURN=quantity of fuel burned during increment [moles]

NC7H17=quantity of fuel prior to increment [moles]

combining equations:

$$\text{BURN} = \text{NC7H17} - \text{FUEL} * (1 - X)$$

$$\text{BURN} = \text{NC7H17} - \text{FUEL} * (1 + \cos((\text{ANGLE} - \text{SPARKANGLE}) * \pi / \text{BURNANGLE})) / 2$$

The value of BURN in the model was limited so that it did not exceed the quantity of reactants present in the cylinder at the start of the incremental combustion.

The energy released by combustion during the increment was the product of the quantity of fuel burned and the heat of combustion:

$$Q_{\text{COMB}} = \text{BURN} * 1932856 \text{ [BTU/mole]}$$

At crankshaft angles less than SPARKANGLE and angles greater than the sum of SPARKANGLE and BURNANGLE, the value of QCOMB was zero.

The quantity of reactants and products changed in proportion to the amount of fuel burned and in accordance with the stoichiometric combustion equation:

$$\text{NC7H17} = \text{NC7H17} - \text{BURN}$$

$$\text{NO2} = \text{NO2} - 11.25 * \text{BURN}$$

$$\text{NCO2} = \text{NCO2} + 7 * \text{BURN}$$

$$\text{NH2O} = \text{NH2O} + 8.5 * \text{BURN}$$

$$\text{NSUM} = \text{NC7H17} + \text{NN2} + \text{NO2} + \text{NCO2} + \text{NH2O}$$

where:

$$\text{NSUM} = \text{the quantity of gas in the cylinder [moles]}$$

The value of NSUM increased during combustion because there are more moles of products than reactants in the chemical reaction.

Heat Transfer

The cylinder walls are exposed to the gas mixture inside the cylinder and a cooling water jacket around the outside of the cylinder. Thermal heat transfer occurs continuously at both of these interfaces. In addition, sliding friction of the piston (and piston rings) against the cylinder walls generates thermal energy which was dissipated by the cylinder walls in the model.

Temperature gradients in the cylinder walls were disregarded in the model, so a single value of temperature, WALLTEMP, described the temperature of the entire wall. This temperature changed in relation to the energy transferred to and from the wall:

$$\text{WALLTEMP} = \text{WALLTEMP} + (\text{QWALLS} + \text{SLIDE} - \text{QCOOLANT}) / (\text{METALCP} * \text{WALLMASS})$$

where:

WALLTEMP = the cylinder wall temperature [$^{\circ}\text{R}$]

QWALLS = thermal energy transferred from the gas mixture
[BTU]

SLIDE = thermal energy generated by friction of the
piston on the walls of the cylinder [BTU]

QCOOLANT = thermal energy transferred to the coolant [BTU]

METALCP = thermal specific heat capacity of the wall
material [BTU/lbm $^{\circ}\text{R}$]

WALLMASS = total metallic mass of the cylinder walls [lbm]

The values of METALCP and WALLMASS were determined based on assumptions of the engine size and material. The value for SLIDE was calculated for incremental movements of the piston when the model calculated the frictional losses of the engine (see the Thermodynamics and Shaft Output section, page 49).

The heat transferred to the cylinder walls from the gas mixture, QWALLS, is due to convection by the movement of the gas mixture over the wall surfaces and to radiation from the high-temperature gases in the cylinder.

Convective heat transfer is characterized by the following equation (ref. 9, pg. 309):

$$QWALLS = TIME * GASAREA * HGAS * (TEMP - WALLTEMP)$$

where:

TIME=time for incremental crankshaft rotation [sec.]

GASAREA=area of cylinder wall exposed to convective flow
[square inches]

HGAS=convective heat transfer coefficient
[BTU/sec. sq.in. °R]

TEMP=the gas mixture temperature [°R]

The convective heat transfer coefficient, HGAS, was calculated from the characteristics of the gas mixture near the surface of the cylinder walls. It is common practice

to assume that the temperature of the gas near the walls is the average temperature of the bulk gas temperature and the wall temperature:

$$\text{GASFILMTEMP} = (\text{TEMP} + \text{WALLTEMP}) / 2$$

Density of the gas mixture was calculated from the ideal gas equation of state:

$$\text{GASDENS} = \text{PRESS} * \text{MW} / \text{GASFILMTEMP} / \text{R0} / 12 / 778.16$$

where:

GASDENS=the gas mixture (film) density [lbm/cu.in.]

PRESS=the gas mixture pressure [psia]

MW=the gas mixture molecular weight [lbm/mole]

R0=the universal gas constant [BTU/mole °R]

The average velocity of the gas flowing over the wall surfaces is affected by the velocity of the piston, the shape of the cylinder volume, the design of the valves, and the characteristics of combustion. In this model, the gas velocity was chosen to be the piston velocity plus a constant which represented the other effects. The value of this constant was selected by numerical trial and error to yield a total heat transfer rate which approximated empirical results.

Convective heat transfer coefficients for turbulent flow can be calculated with the following equation (ref. 9, pg. 431):

$$H = .023 * C_p * G * ((U/D_h/G)^{.2}) / (Pr^{.67})$$

where:

H=convective coefficient [BTU/hour sq.ft. °R]

C_p=fluid specific heat at constant pressure
[BTU/lbm °R]

G=fluid mass velocity (density*velocity)
[lbm/hour sq.ft.]

U=fluid viscosity [lbm/ft. hour]

D_h=hydraulic diameter [ft.]

Pr=fluid Prandtl number

substitute properties of the gas film:

$$HGAS = GASFACTOR * GASDENS * GASVEL * (1/12 / BORE / GASDENS / GASVEL)^{.2}$$

where:

$$GASFACTOR = .023 * C_p * (U^{.2}) / (Pr^{.67}) \\ [(BTU/lbm \text{ } ^\circ R) * (lbm/ft. \text{ sec.})^{.2}]$$

GASDENS=the gas mixture (film) density [lbm/cu.in.]

GASVEL=average gas velocity over the wall surface
[in./sec.]

BORE=cylinder bore diameter (chosen for hydraulic diameter) [in.]

The value of the lumped parameter GASFACTOR was determined by substituting the values for C_p, U, and Pr for the gas at the film temperature. The following quadratic

equation curve fits tabulated data for air at atmospheric pressure, and it was used in the model for the cylinder gas mixture (which is assumed to be mostly air):

$$\text{GASFACTOR} = -4.36\text{E-}11 * \text{GASFILMTEMP}^2 + 2.89\text{E-}07 * \text{GASFILMTEMP} + .000578$$

Radiation from the gas mixture to the cylinder walls is characterized by the Stefan-Boltzmann law (ref. 9, pg. 223):

$$\text{QRAD} = \text{E} * \text{SB} * ((\text{TEMP}^4) - (\text{WALLTEMP}^4))$$

where:

QRAD=radiation heat transfer [BTU/sec. sq.in.]

E=emissivity of gas mixture

SB=Stefan-Boltzmann coefficient [3.305E-15
BTU/sec. sq.in. °R⁴]

TEMP=the gas mixture temperature [°R]

WALLTEMP=the cylinder wall temperature [°R]

Determination of the value of emissivity of a high-temperature gas is beyond the scope of this study. A value was selected which approximates an average emissivity for carbon-dioxide at high temperature (ref. 6, pg. 199):

$$\text{E} = .05$$

The temperature of the cylinder wall never gets hot enough for the last term in the equation to become significant, so radiation heat transfer rate was approximated by the following:

$$Q_{RAD} = .05 * 3.305E-15 * (TEMP^4)$$

Convective heat transfer between the cylinder wall and the liquid coolant was calculated by the following:

$$Q_{COOLANT} = TIME * COOLANTAREA * H_{COOLANT} * (WALLTEMP - COOLANTTEMP)$$

where:

$Q_{COOLANT}$ = thermal energy transferred to the coolant [BTU]

$TIME$ = time for incremental crankshaft rotation [sec.]

$COOLANTAREA$ = area of wall exposed to coolant flow [sq.in.]

$H_{COOLANT}$ = convective heat transfer coefficient
[BTU/sec. sq.in. °R]

$WALLTEMP$ = the cylinder wall temperature [°R]

$COOLANTTEMP$ = the coolant temperature [°R]

The convective heat transfer coefficient, $H_{COOLANT}$, was calculated by the same equation used for the gas mixture:

$$H = .023 * C_p * G * ((U/D_h/G)^{.2}) / (Pr^{.67})$$

substitute values for the coolant film:

$$HCOOLANT=COOLANTFACTOR*((COOLANTVEL)^.8)*((1/12/HYDIAM)^.2)$$

where:

$$COOLANTFACTOR=.023*Cp*(density^.8)*(U^.2)/(Pr^.67)$$

COOLANTVEL=the average coolant velocity [in./sec.]

HYDIAM=coolant flow hydraulic diameter (2*annular height) [in.]

The following quadratic equation curve fits tabulated data for water at atmospheric pressure, and it was used in the model for the coolant:

$$COOLANTFACTOR=-9.3E-09*COOLANTFILMTEMP^2+2.08E-05*COOLANTFILMTEMP-.0069$$

Thermodynamics and Shaft Output

Shaft horsepower output from the engine was determined by calculating the energy balance of the system. First an energy balance was done for the gas in the cylinder. Then an energy balance was done for the piston and linkage. After all energy storage and dissipation was accounted for, the remaining energy was delivered to the output shaft and resulted in the brake horsepower output of the engine.

Gas Mixture Enthalpy and Work

An energy balance on the gas in the cylinder must account for the energy flowing with the gases through the intake and exhaust valves, the energy released as a result of combustion in the cylinder, the energy transfer in the form of heat to the cylinder walls, and the work due to expansion or compression of the gas in the cylinder.

Energy entering through the intake and exhaust valves is the enthalpy of the INFLOW, EXFLOW, REVFLOW, and RETFLOW gases. The kinetic energy of the flowing gases was ignored in this analysis, because it was assumed that the gas velocity was small in the intake manifold and inside the control volume of the cylinder (the higher flow velocities through the valves are outside the control volume).

Enthalpy (H) is defined as follows:

$$H=N*CP*(T-ATMTEMP)$$

where:

N=moles of gas

CP=the specific heat of the gas (constant pressure)

T=the gas temperature

ATMTEMP=atmospheric (reference) temperature

The combined enthalpies of the INFLOW and RETFLOW gases entering the cylinder through the intake valve were assigned to the variable INENTHALPY [BTU]. Since the

temperature of the INFLOW gas was ATMTEMP, and the temperature of the RETFLOW gas was TEMP, the resulting equation for INENTHALPY became:

$$\text{INENTHALPY} = \text{INFLOW} * \text{ICP} * (\text{ATMTEMP} - \text{ATMTEMP}) + \text{RETFLOW} * \text{CP} * (\text{TEMP} - \text{ATMTEMP})$$

which reduces to:

$$\text{INENTHALPY} = \text{RETFLOW} * \text{CP} * (\text{TEMP} - \text{ATMTEMP})$$

The combined enthalpies of the EXFLOW gas exhausting through the exhaust valve and the REVFLOW gas escaping through the intake valve were assigned to the variable EXENTHALPY [BTU]. Both the EXFLOW and REVFLOW gases came from the gas in the cylinder at temperature TEMP.

$$\text{EXENTHALPY} = \text{EXFLOW} * \text{CP} * (\text{TEMP} - \text{ATMTEMP}) + \text{REVFLOW} * \text{CP} * (\text{TEMP} - \text{ATMTEMP})$$

which reduces to:

$$\text{EXENTHALPY} = (\text{EXFLOW} + \text{REVFLOW}) * \text{CP} * (\text{TEMP} - \text{ATMTEMP})$$

The WORK [BTU] done by the gas in the cylinder on the boundary as a result of expansion or compression was defined as the product of the pressure on the piston and the change in volume resulting from movement of the piston.

$$\text{WORK} = \text{PRESS} * \text{AREA} * (\text{Y2} - \text{Y3}) / 12 / 778.16$$

where:

PRESS = the gas pressure [psia]

AREA=the cross-sectional area of the cylinder [sq.in.]

Y2=the height of the piston before movement [in.]

Y3=the height of the piston after movement [in.]

The equation for work assumed that the pressure PRESS was constant during incremental movement of the piston. Initially, the value of PRESS was chosen as the pressure of the gas in the cylinder before the incremental movement. During an expansion stroke (pressure decreasing) the value calculated for the WORK done by the gas was artificially high. During a compression stroke (pressure increasing) the value calculated for WORK was negative and the magnitude was artificially small. The result would be a cumulative error which tended to deplete the energy content of the gas mixture. The correction for this error is described later in this section.

The energy balance of the gas in the cylinder required that the energy stored in the enthalpy of the gas (ENTHALPY) must be equal to the initial ENTHALPY [BTU] plus the energy added to the volume minus energy lost from the volume. Energy released by combustion of the charge in the cylinder (QCOMB) was treated simply as energy added to the volume.

$$\text{ENTHALPY} = \text{ENTHALPY} + \text{INENTHALPY} - \text{EXENTHALPY} - \text{QWALLS} + \text{QCOMB} - \text{WORK}$$

where:

ENTHALPY=gas enthalpy in the cylinder [BTU]

INENTHALPY=enthalpy of INFLOW and RETFLOW gas [BTU]

EXENTHALPY=enthalpy of EXFLOW and REVFLOW gas [BTU]

QWALLS=heat transferred to the cylinder walls [BTU]

QCOMB=energy released by combustion [BTU]

WORK=expansion work done by gas on piston [BTU]

Note that the values of QWALLS and WORK will be negative in some circumstances.

The enthalpy of a gas was shown previously to be:

$$H = N * CP * (T - \text{ATMTEMP})$$

Solving this equation for the gas temperature T yields the following:

$$T = H / N / CP + \text{ATMTEMP}$$

Substitute the values $H = \text{ENTHALPY}$, $T = \text{TEMP}$, and $N = \text{NSUM}$ for the gas in the cylinder:

$$\text{TEMP} = \text{ENTHALPY} / \text{NSUM} / \text{CP} + \text{ATMTEMP}$$

where:

TEMP=the gas temperature [$^{\circ}\text{R}$]

ENTHALPY=the gas enthalpy [BTU]

NSUM=the quantity of gas [moles]

CP=the gas specific heat [BTU/mole °R]

ATMTEMP=atmospheric temperature [°R]

The pressure PRESS [psia] of the gas in the cylinder was found from the ideal gas equation of state:

$$\text{PRESS} = \text{NSUM} * \text{R}_0 * 12 * 778.16 * \text{TEMP} / \text{VOLUME}$$

The values calculated for ENTHALPY, TEMP, and PRESS were dependent on the value calculated for WORK. As previously discussed, the value of WORK was based on the value of PRESS before the piston movement, which would result in a cumulative error. The cumulative error was reduced by averaging the value of PRESS before the movement with the value of PRESS after the movement (result of last calculation). The average pressure was used to recalculate WORK, ENTHALPY, TEMP, and PRESS. This iteration, coupled with a sufficiently small value of INCREMENT, satisfactorily eliminated the cumulative error. This was verified by observing the cyclic variation in gas temperature with simple piston-cylinder motion without mass flow, friction, heat transfer, or combustion. Without the correction, the temperature would increase with every cycle.

The WORK done by the gas mixture on the piston was stored as kinetic energy in the piston and engine mechanism, dissipated as friction on the cylinder walls and engine mechanism, and converted to torque in the engine output shaft. Some of the WORK was used to "push" the piston against atmospheric pressure.

The work [BTU] done by the piston on the atmosphere was the product of atmospheric pressure on the piston and the change in volume resulting from movement of the piston.

$$\text{ATMWORK} = \text{ATMPRESS} * \text{AREA} * (\text{Y2} - \text{Y3}) / 12 / 778.16$$

where:

ATMPRESS=atmospheric pressure [psia]

AREA=the cross-sectional area of the cylinder [sq.in.]

Y2=the height of the piston before movement [in.]

Y3=the height of the piston after movement [in.]

Kinetic Energy

The energy stored kinetically in the piston and engine mechanism depends on the mass and mass distribution of the engine components and their rotational and translational velocities. Calculation of this energy is very complex; fortunately, a detailed accounting of this energy term was not crucial to the value of this model. At constant RPM, the kinetic energy stored in rotating masses (ie. the crankshaft and flywheel) is constant, and it does

not affect the calculation of the engine output power. The reciprocating motion of the piston in the cylinder results in an oscillating storage of kinetic energy which can be adequately described by analysis of the piston motion with an appropriate assumption of the piston mass. The connecting rod "big end" is part of the rotating mass, whereas the piston pin end is part of the translating piston mass. The kinetic energy of the remainder of the connecting rod, which both rotates and translates, was not calculated in this model.

The change in kinetic energy of the piston, DELKE [BTU], was estimated by calculation of the change in piston velocity. Acceleration of the piston, AVGACC, and the mass of the piston, PISTONMASS, defined the force required to change the speed of the piston translation. The product of this force and the movement distance is equivalent to the change in kinetic energy. For accuracy, the movement distance used in the calculation was half of the two previous incremental movements, because the AVGACC value was based on the change in velocity between those two movements.

$$\text{DELKE} = \text{PISTONMASS} * \text{AVGACC} * (\text{Y3} - \text{Y1}) / 2 / \text{GC} / 12 / 778.16$$

where:

PISTONMASS=the mass of the piston and pin [lbm]

AVGACC=piston acceleration [in./sec.²]

Y3=height of the piston after incremental movement
[inches]

Y1=height of the piston before the last two
incremental movements [inches]

Mechanical Friction

Movement of the piston in the cylinder is resisted by friction. Where there is an oil film between the piston and the cylinder walls, the friction is a result of viscous shear of the oil film. Some of the oil film is burned away during combustion, resulting in sliding of the piston rings, ring lands, and skirt against the cylinder walls (ref. 12, pg. 83).

Calculation of piston friction in this model was based on sliding surfaces without an oil film. The frictional force was directly proportional to the normal force between the surfaces and was independent of the sliding velocity. Assuming pure Coulomb friction, the coefficient of sliding friction (FRICTION) was held constant.

The energy lost to friction of the piston sliding on the walls of the cylinder, SLIDE [BTU], was proportional to the normal force of the piston against the walls. The normal force is a function of the angle of the connecting rod and the piston force arising from pressure differential and inertia.

$$\text{FORCE} = \text{AREA} * (\text{PRESS} - \text{ATMPRESS}) + \text{PISTONMASS} * \text{AVGACC} / \text{GC}$$

where:

FORCE=piston force tangential to cylinder wall [lbf]

AREA=cross-sectional area of the cylinder [sq.in.]

PRESS=the gas pressure [psia]

ATMPRESS=atmospheric pressure [psia]

PISTONMASS=the mass of the piston and pin [lbm]

AVGACC=piston acceleration [in./sec.²]

The normal component of force between the piston and the cylinder walls is equal to the product of FORCE and the tangent of the connecting rod angle. The energy absorbed by action of the frictional force through the incremental movement of the piston was calculated as follows:

$$\text{SLIDE} = \text{ABS}(\text{FRICTION} * \text{FORCE} * (\text{Y2} - \text{Y3}) * \text{TAN}(\text{A2})) / 12 / 778.16$$

where:

SLIDE=energy absorbed by friction of piston [BTU]

FRICTION=the coulomb frictional coefficient

FORCE=piston force tangential to cylinder wall [lbf]

Y2=the height of the piston before movement [in.]

Y3=the height of the piston after movement [in.]

TAN(A2)=the tangent of the connecting rod angle relative
to the sliding surfaces

An absolute value was required in the preceding calculation because the values of FORCE, Y2-Y3, AND TAN(A2) will be negative in some circumstances, but the energy required to overcome sliding friction is always a positive quantity.

The friction of the piston rings on the cylinder wall is treated in the same manner as the piston friction, except the normal force is a function of the spring tension of the rings and the pressure differential between the cylinder and the crankcase:

$$\text{RINGSLIDE} = \text{ABS}(\text{FRICTION} * \text{RINGFORCE} * (\text{Y2} - \text{Y3})) / 12 / 778.16$$
$$\text{RINGFORCE} = \text{SPRING} + \text{RINGAREA} * (\text{PRESS} - \text{ATMPRESS})$$

where:

RINGSLIDE=energy dissipated by friction of piston rings
[BTU]

RINGFORCE=piston ring force normal to cylinder wall [lbf]

SPRING=the normal force of the rings on the cylinder
wall due to spring tension [lbf]

RINGAREA=the effective unbalanced ring area [sq. in.]

Friction in the bearings of the engine mechanism is a result of viscous shear of oil in the bearing clearances. The frictional forces are proportional to the oil viscosity, bearing geometry, and rotational velocity.

Viscous drag force, F [lbf], on a flat plate moving over an oil film opposes the velocity (ref. 7, pg. 3.15-2):

$$F = \text{VISCOSITY} * \text{AREA} * \text{VEL} / H / GC$$

where:

F = the viscous drag force [lbf]

VISCOSITY = viscosity of the oil [lbm/in. sec.]

AREA = area of the plate in contact with oil [sq.in.]

VEL = velocity of the plate [in./sec.]

H = thickness of the oil film [in.]

Viscous drag of a rotating journal bearing with diameter much larger than the thickness of the annular clearance is approximately equivalent to the drag on a flat plate. The plate velocity is replaced with the tangential velocity at the nominal bearing radius.

$$\text{VEL} = A * (D/2) / \text{TIME}$$

where:

VEL = tangential velocity [in./sec.]

A = incremental angular bearing rotation [radians]

$D/2$ = nominal radius of the bearing [in.]

TIME=time required for the incremental rotation [sec.]

The energy required to overcome the viscous drag is the product of the drag force (F) and the distance through which the force acts.

$$\text{ROTATE} = F * A * (D/2) / 12 / 778.16$$

where:

ROTATE=the energy lost to viscous drag during an incremental rotation [BTU]

substituting for F:

$$\text{ROTATE} = (\text{VISCOSITY} * \text{AREA} * \text{VEL} / H / GC) * A * (D/2) / 12 / 778.16$$

substituting for VEL:

$$\text{ROTATE} = \text{VISCOSITY} * \text{AREA} * (A * (D/2) / \text{TIME}) / H / GC * A * (D/2) / 12 / 778.16$$

simplifying:

$$\text{ROTATE} = \text{VISCOSITY} * \text{AREA} * (A * D/2)^2 / \text{TIME} / H / GC / 12 / 778.16$$

The energy required to overcome viscous friction in the bearing must be a positive quantity. Note that all the terms in the previous equation are always positive except for the value of the incremental angular bearing rotation (A) which changes sign when the direction of rotation is reversed. Since this value is squared in the equation, the value of ROTATE will always be positive.

Limitations to the applicability of the previous equation to the model of the engine mechanism can be

identified. The equation assumes constant oil viscosity. The viscous dissipation of energy in a journal bearing causes the oil to heat up, thereby lowering the viscosity of the oil. The computer model assumed that the oil had the same viscosity in each bearing.

The thickness of the oil film will vary around the circumference of the bearing as a function of radial load. The clearance will be reduced along one half of the circumference, but it will be increased along the opposite half. The resultant local effects on viscous drag tend to offset each other (ref. 12, pg. 93), so in the model it was assumed that the clearance was constant, and viscous drag was independent of radial load.

The viscous drag equation is derived from the behavior of steady, fully-developed flow. The direction of rotation changes for some bearings in the engine mechanism, so the velocity gradient is not always (and may never be) fully-developed. Solution of this complex fluid mechanics problem was not incorporated in the computer model; however, bearings subject to reciprocating motion would tend to have shorter tangential displacements ($A \cdot D/2$) than those with constant rotational velocities. Since the $A \cdot D/2$ term is squared in the ROTATE energy equation, reciprocating bearings will have a smaller contribution to the total value of ROTATE. It follows that errors

resulting from unsteady rotation will not have a substantial effect on the viscous loss calculation results.

Shaft Output

Work that the gas mixture does on the piston during incremental movement pushes the piston against atmospheric pressure, adds kinetic energy to the piston mass, is dissipated as sliding and viscous friction, and powers the output shaft.

$$\text{WORK} = \text{ATMWORK} + \text{DELKE} + \text{SLIDE} + \text{ROTATE} + \text{SHAFT}$$

solving for output shaft energy:

$$\text{SHAFT} = \text{WORK} - \text{ATMWORK} - \text{DELKE} - \text{SLIDE} - \text{ROTATE}$$

where:

SHAFT=the output shaft energy [BTU]

WORK=expansion work done by the gas on the piston
[BTU]

ATMWORK=work done by the piston on the atmosphere [BTU]

DELKE=change in kinetic energy of the piston [BTU]

SLIDE=energy absorbed by friction of piston [BTU]

ROTATE=the energy lost to viscous drag [BTU]

The rate of SHAFT energy is the instantaneous net output power of the engine.

$$\text{BHP} = (\text{SHAFT} / 2544.48) / (\text{TIME} / 3600)$$

where:

BHP=instantaneous engine output power [horsepower]

SHAFT=the output shaft energy [BTU]

TIME=time required for incremental rotation [sec.]

unit conversions:

2544.48 BTU/horsepower/hour

3600 seconds/hour

Summation of the instantaneous net output power over the complete four-stroke cycle yields the net horsepower output of the engine.

$$\text{SUMBHP} = \text{SUMBHP} + \text{BHP}$$

$$\text{CYCLEHP} = \text{SUMBHP} / (\text{ANGLE} / \text{INCREMENT} - 1)$$

where:

SUMBHP=the sum of instantaneous BHP [horsepower]

CYCLEHP=the net power output of the cycle [horsepower]

ANGLE=the angular position of the crankshaft [degrees]

INCREMENT=the size of the incremental crankshaft rotation [degrees]

CHAPTER III

EXECUTING THE PROGRAMS

Program Input Values

Values were assumed for the engine parameters which corresponded to a cylinder in a General Motors 305 cubic inch V8 engine. This established the following sizing parameters for both the conventional and variable-stroke engine models:

BORE (cylinder diameter)=3.735 inches

PISTONMASS (reciprocating mass)=3 lbm

DISPL (displacement, on compression stroke)=38.125 cu.in.

COMPRESS (compression ratio)=8.6

The sizing criteria for comparison of the two engines was based on equivalent volumes in both cylinders at the start of the compression stroke and at the end of the compression stroke. Thus compression ratios were equal, and the volumes at the onset of combustion were equal. The total travel length of the piston was greater in the variable-stroke engine during the expansion, exhaust, and intake strokes.

Internal heat transfer area was based on these parameters, assuming that the volume was cylindrical with

height varying according to piston position. The cylinder wall and external (coolant) heat transfer parameters were assumed as follows:

COOLANTTEMP (bulk coolant temperature)=650 °R

COOLANTVEL (mean coolant velocity)=25 inches/sec.

HYDIAM (hydraulic diameter of coolant passages)=.5 inches

WALLTHK (thickness of cylinder walls and head)=.25 inches

METALDENSITY (density of wall and head material)=.28
lbm/cubic inch

METALCP (specific heat capacity)=.11 BTU/lbm °R

All journal bearing diameters were selected arbitrarily. Bearing lengths were 0.5 times the bearing diameters, and bearing clearances were set at 0.001 times the bearing diameters in accordance with published recommendations (ref. 14, vol. 2, page 516).

The value for INMAX (intake valve sizing coefficient) was chosen as .12 by trial and error to get typical values for volumetric efficiency in the conventional engine model. This value was also used for EXMAX (exhaust valve sizing coefficient), and the same values were used in the variable-stroke engine model.

FRICITION (Coulomb friction coefficient) was set at .05, which is typical for lubricated machined steel surfaces. The oil viscosity in the journal bearings was

set at .00068 lbm/sec. in., which corresponds to SAE 10W-30 oil at 650 °R.

The incremental step size chosen for evaluation of parameters in the model was five degrees of crankshaft rotation. This value represented a trade-off between accuracy and computer solution time.

Valve timing (INOPEN, INCLOSE, EXOPEN, and EXCLOSE) and ignition timing (SPARKANGLE) were optimized for maximum engine horsepower output by a sectioning method (ref. 3, page 219). Both engine models were optimized for 3500 rpm operation, and the results were verified by perturbation of the angles plus and minus one incremental step. Maximum performance was indicated when perturbations of the valve timing and ignition timing parameters would not yield an increase in output horsepower.

The sectioning technique could not entirely rule out the possibility that the maximum performance found was only a local maximum for the set of timing parameters. Random perturbations with larger step sizes did not indicate multimodal relationships of these parameters to horsepower output.

Linkage lengths in the variable-stroke engine model were selected by trial and error to provide the desired piston motion. The direction of the second shaft rotation was arbitrarily selected to be opposite the rotation of the

crankshaft. Another complete set of linkage length values would be obtained for rotation of both shafts in the same direction. The attempt was to drive the clearance volume to zero at the end of the exhaust stroke, but values very close to zero were avoided because of an adverse effect on the stability of the model. Practical considerations in an actual engine would limit the minimum clearance volume to that required for valve operation (assuming conventional valve design).

With all of the input variables determined, the engine simulations were run through several cycles to stabilize the output values. The first cycle started with the stoichiometric mixture in the cylinder at atmospheric temperature and pressure. Successive cycles included the effects of the residual gas. The starting cylinder wall temperature was selected by inspection to expedite the "warm up" operation. The engine simulations were at steady-state operation when the conditions at the start of a cycle were the same as the conditions at the end of the cycle. Five complete four-stroke cycles were required to achieve reasonably steady-state operation.

Results

The results from both engine models are shown in Table I. The values are total quantities for one complete four-stroke cycle of an eight cylinder engine. The variable-stroke engine delivered 12% more power to the output shaft, because of its higher volumetric efficiency. Both engine models lost about the same percentages of the total energy to heat transfer and to the exhaust flow, with the majority of the total energy going to the exhaust. The variable-stroke engine had more energy dissipated in viscous friction, but the actual quantity was still small.

TABLE I. PROGRAM RESULTS

Parameter Name	Conventional V8 Engine	Variable-Stroke V8 Engine
Output Power (HP)	169.95	190.88
Volumetric efficiency (%)	81.24	91.92
Combustion Energy (BTU/cycle)	14.65	16.57
Heat Transfer (BTU/cycle)	0.44	0.46
Sliding Friction (BTU/cycle)	0.22	0.22
Viscous Friction (BTU/cycle)	0.06	0.14
Shaft Output (BTU/cycle)	4.12	4.63
Exhaust Energy (BTU/cycle)	9.80	11.11

The conventional engine was slightly more efficient. The ratio of shaft output energy to combustion energy in the conventional engine was 28.1% compared to 27.9% for the variable-stroke engine. This difference may be the result of the viscous friction or it may be an arbitrary outcome of error propagation in the computer models.

The following sections discuss values of several output parameters. The data is shown graphically in figures 5 through 17 (see Appendix A) for expeditious comparison of the performance of the two engine models. All values in these figures indicate the instantaneous values of the parameters during the (five degree) incremental crankshaft rotations.

Cylinder Volume

Figure 5 plots data describing the cylinder volume for the conventional and the variable-stroke engine models. For convenience, the plot is divided into four sections representing the four-stroke cycle. These divisions are shown at the crankshaft angles where the piston of the conventional engine is at top-dead-center (TDC) or bottom-dead-center (BDC) even though the four engine processes overlap these limits.

Variation of the cylinder volume with rotation of the crankshaft is similar to a sine wave. The conventional

engine curve shows a slightly higher rate of change near TDC (minimum volume) than at BDC (maximum volume). This is an expected result of the conventional engine geometry. The variable-stroke engine curve shows how the maximum and minimum values change during the cycle. Both curves are continuous and have volumes at the start of the cycle which match those at the end of the cycle.

At the start of the intake stroke, the volume in the variable-stroke engine is smaller than the volume in the conventional engine. The volumes are identical at the end of the intake stroke and at the end of the compression stroke, satisfying the engine sizing criteria for comparison of the two engine models.

The variable-stroke engine has a larger volume at the end of the expansion stroke to try to get more work from expansion of the combustion products. The volume is smaller at the end of the exhaust stroke in the variable-stroke engine to more completely purge the exhaust gases from the cylinder in preparation for the next intake cycle.

Intake Flow Rate

Figure 6 shows how the flow rate through the intake valve varies with angular rotation of the crankshaft. The valve timing of both engine models was set at an optimum opening angle of 25 before TDC, but the closing of the

variable-stroke engine intake valve occurred 15 degrees after the conventional engine intake valve (at 275 degrees and 260 degrees, respectively). The variable-stroke engine appears to use this extra open duration to ingest more fresh charge.

Both engines show slight negative intake flow values, indicating the occurrence of reverse flow through the intake valves. This result would not appear to have a significant effect on the overall performance of either engine.

The intake flow continues past BDC because there is still a partial vacuum in the cylinder even as the piston starts rising on the compression stroke. The restriction of flow by the intake valve prevents the attainment of atmospheric pressure in the cylinder at BDC.

Exhaust Flow Rate

The exhaust valve opening of both engine models begins at the crankshaft angle of 440 degrees, and the exhaust flow starts during the latter half of the expansion stroke (see figure 7). The heated combustion products must be released as quickly as possible to minimize the work required as the piston rises on the exhaust stroke.

The conventional engine exhaust valve closes at 60 degrees and the variable-stroke engine valve closes at 40

degrees after TDC of the intake stroke. The figure shows that the exhaust process continues slightly past TDC, but the valve overlap does not appear to be significant at these angles. The primary advantage to late closing of the intake valve is to maximize the size of the opening area over the duration of the exhaust stroke.

Mixture Quantity

The curves on figure 8 clearly show how the gases are pumped through the engine. The upward slope is the intake process, and the downward ramp is the exhaust of the combustion products. The small rise in molar quantity near the start of the expansion stroke signals the combustion process, because the chemical reaction yields more moles of products than the moles of reactants.

The variable-stroke engine has a smaller quantity of gas in the cylinder at the start of the intake stroke because the volume in the cylinder is nearly zero at the start of the cycle. The peak total quantity of gas is higher in the variable-stroke engine, indicating a higher volumetric efficiency.

Combustion Energy

Burning of the reactants in both engines releases energy at a rate characterized by the sine wave constraint

(see figure 9). The burning duration for both models is about 66 degrees of crankshaft rotation.

Combustion in the variable-stroke engine occurs a few crankshaft degrees after the conventional engine. Ignition timing was optimized at 335 degrees and 325 degrees respectively.

The variable-stroke engine has more energy released by combustion because the higher volumetric efficiency provided more fuel for combustion.

Mixture Temperature

The mixture temperature decreases during the intake stroke as the cooler fresh charge mixes with the higher temperature residual gas from the previous cycle. The variable-stroke engine mixture cools more rapidly because the quantity of residual gas is much smaller (see figure 10).

The temperature rises as the compression stroke adds work to the gas, and the temperature jumps rapidly at the onset of combustion. Even though the variable-stroke engine had more energy released by combustion, the temperature peak is not as high. This could be the result of the difference in ignition timing, the cooler intake stroke temperatures, and the larger quantity of gas in the variable-stroke cylinder to absorb the combustion energy.

The mixture temperatures drop somewhat as the expansion stroke delivers work to the output shaft and as the exhaust products are allowed to escape through the exhaust valve. The temperatures increase again as the piston rises on the exhaust stroke, adding energy to the system while the quantity of gas in the cylinder is diminishing. The variable-stroke engine gas temperature reaches a high extreme because the volume of gas at the end of the exhaust stroke is nearly zero. This would not appear to have a significant effect on the total overall performance of the engine, but it could have major implications on the choice of component materials.

Mixture Pressure

The pressure in the cylinders of both engines drop below atmospheric pressure during the intake stroke, pulling fresh charge through the intake valve. The pressures rise as the mixtures are compressed, and they jump rapidly when combustion begins (see figure 11).

The mixture pressure in an engine is directly related to the power output of the engine. The timing of the pressure peak must be optimized based on the position of the linkage to convert the pressure into torque of the output shaft. Both engines appear to reach the same peak pressure, but the variable-stroke engine timing lags the

conventional engine by a few degrees of crankshaft rotation.

The pressures decrease during the expansion and exhaust strokes, although there is a very slight increase in pressure towards the end of the exhaust stroke as the piston rises and the exhaust valve begins closing.

Work On Piston

The work imparted to the piston by the gas mixture in the cylinder is positive during the downward movement of the piston (see figure 12). However downward movement of the piston against atmospheric pressure on the other side requires work from the piston or engine mechanism (see figure 13). The signs of the work quantities reverse when the piston rises in the cylinder. The net result with the pressure changes during the engine cycle is that the piston movement consumes work during the intake, compression, and exhaust strokes and delivers work to the connecting rod and crankshaft during the expansion stroke.

Both engines consume nearly the same quantities of work during three strokes, but the variable-stroke engine gets more work from the expansion stroke. Since the peak pressures were nearly the same, the movement of the piston in the variable-stroke engine must be greater near the timing of the pressure peak.

Heat Transfer Rate

The high temperature residual gas in the cylinder at the start of the intake stroke loses heat to the cylinder wall. As the cooler intake charge enters the cylinder, the direction of heat transfer changes, until the heating of the mixture by compression and combustion raises the gas mixture temperature above the wall temperature (see figure 14).

Most of the heat transfer occurs during the expansion stroke because of the large temperature differential between the gas mixture and the cylinder wall. The variable-stroke engine has a higher peak heat flux. The rate of heat transfer drops at the end of the expansion stroke because the piston velocity is momentarily zero at the BDC position.

Mechanical Friction

Sliding friction of the pistons and piston rings on the cylinder walls of both engines is a complex function of the piston velocity, the pressure differential across the piston, and the angle of the connecting rod. The variable-stroke linkage appears to be more efficient during the intake and expansion strokes, but has higher frictional energy dissipation during the compression and exhaust strokes (see figure 15).

The variable-stroke engine consumes more energy than the conventional engine in the viscous friction associated with the mechanism bearings (see figure 16). The value of the friction calculated in this model is small compared to the overall performance of the engines.

The discontinuities in the variable-stroke friction curve are due to the programming technique used for calculation of the bearing rotations. The instantaneous character of the discontinuities makes their effect on the overall performance insignificant.

Shaft Output

Figure 17 shows the instantaneous values of energy delivered from or consumed by the engines over their operating cycles. The dominant influences on the shapes of these curves are the pressure differentials across the pistons and the acceleration/deceleration of the piston mass near the TDC and BDC positions.

Local differences between the outputs of the two engines are easily observed, but the net difference over the cycle is not readily apparent in this figure.

CHAPTER IV

CONCLUSIONS

The results obtained from the computer models indicate that the variable-stroke engine produced a measurable gain in output horsepower compared to a conventional four-stroke internal combustion engine. The gain was a direct result of an increase in the volumetric efficiency of the variable-stroke engine. The small clearance volume at the end of the exhaust stroke more completely purged the combustion products from the cylinder and facilitated a greater ingestion of fresh charge on the intake stroke.

The long expansion stroke of the variable-stroke engine model did not produce a measurable increase in output power. By the end of the expansion stroke, the pressure differential across the piston was too small to contribute much useful work.

The overall efficiencies of both engines were nearly identical. The efficiency of the variable-stroke engine model was slightly smaller because of mechanical friction in the mechanism, but the difference was not significant.

The use of the variable-stroke mechanism in an engine design would involve a trade-off in the size, cost, and complexity of the mechanism verses the performance gain

attainable for a given engine size. Fuel economy would not be significantly affected.

There are several areas which warrant additional study:

1. Refine the constants used in the models based on a more thorough calibration of the fixed-stroke model with actual engine data.
2. Design the mechanism components with enough detail to identify their sizes and costs.
3. Examine different arrangements of the mechanism (linkage lengths and directions of rotation) to quantify their overall performance effects.
4. Perform an analysis of the sensitivity of the engine model performance to the values of the input parameters.
5. Analyze the vibration and engine unbalance caused by the mechanism.
6. Explore the performance of the engine model with supercharging and turbocharging.
7. Generate a performance curve for the engine model over a range of engine speeds.
8. Investigate more thoroughly the effects of cylinder clearance volume on engine volumetric efficiency.

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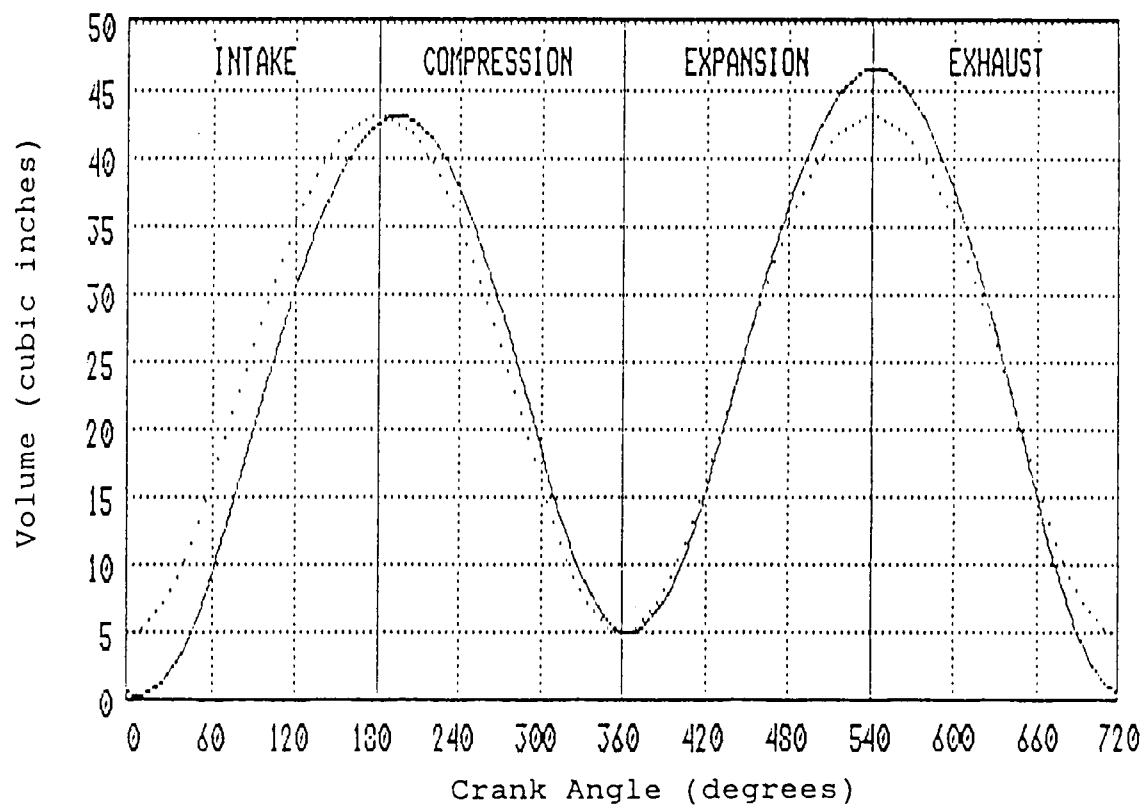
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APPENDIXES

APPENDIX A.

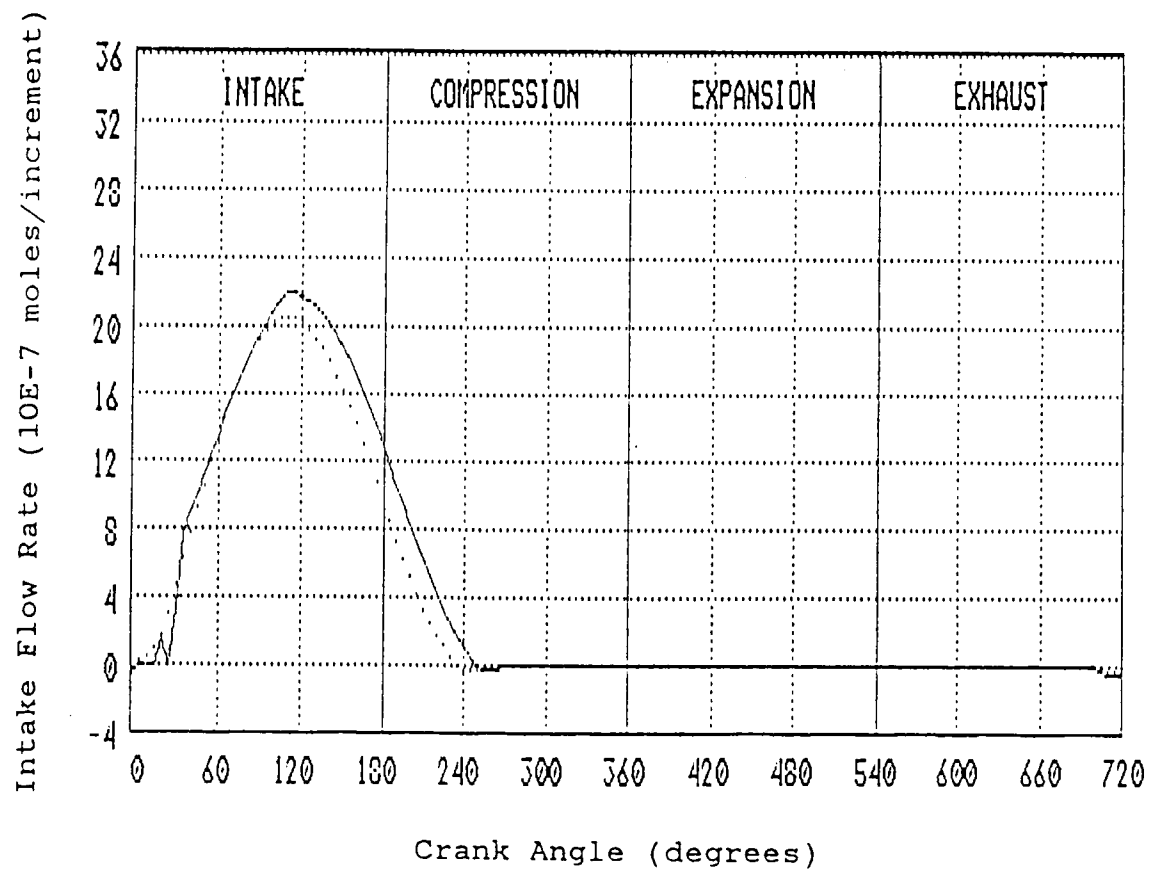
FIGURES



Conventional - - - - -

Variable-Stroke ———

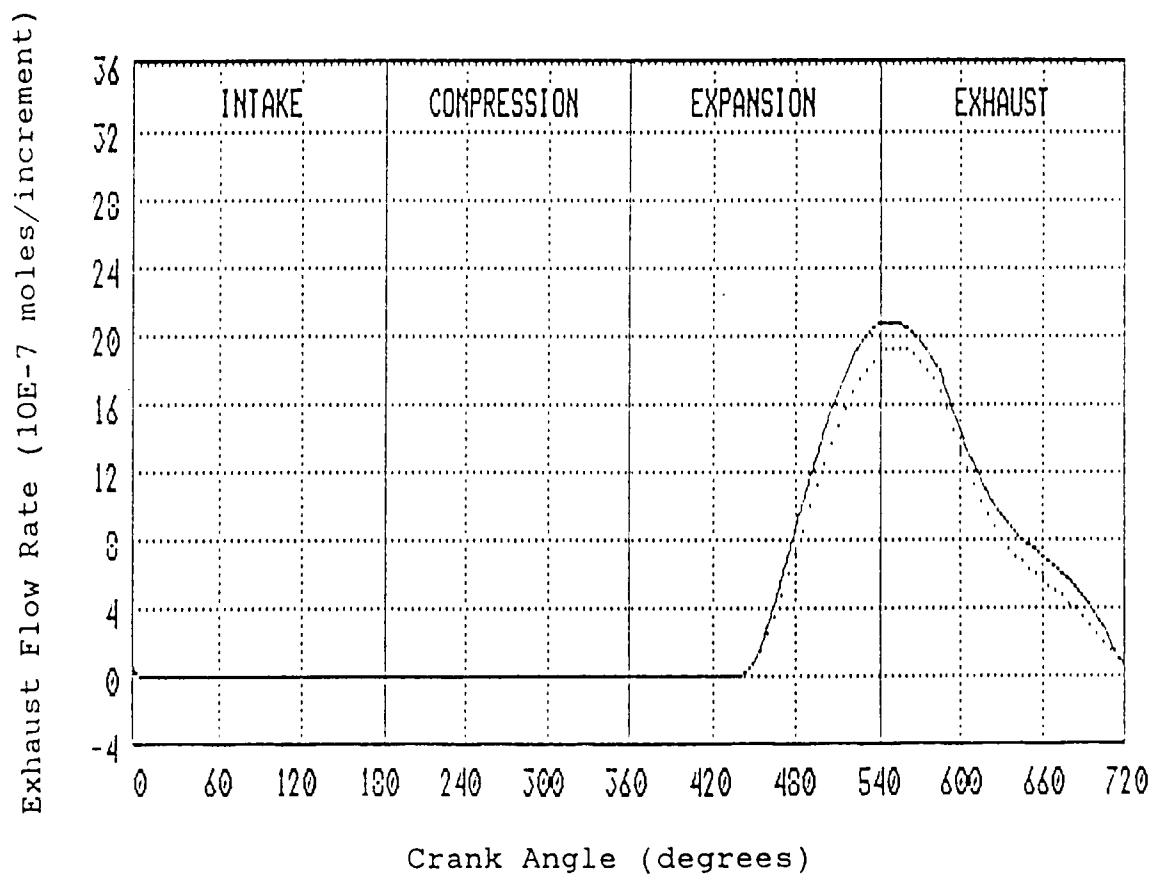
Figure 5. Cylinder Volume vs Crank Angle



Conventional - - - - -

Variable-Stroke ———

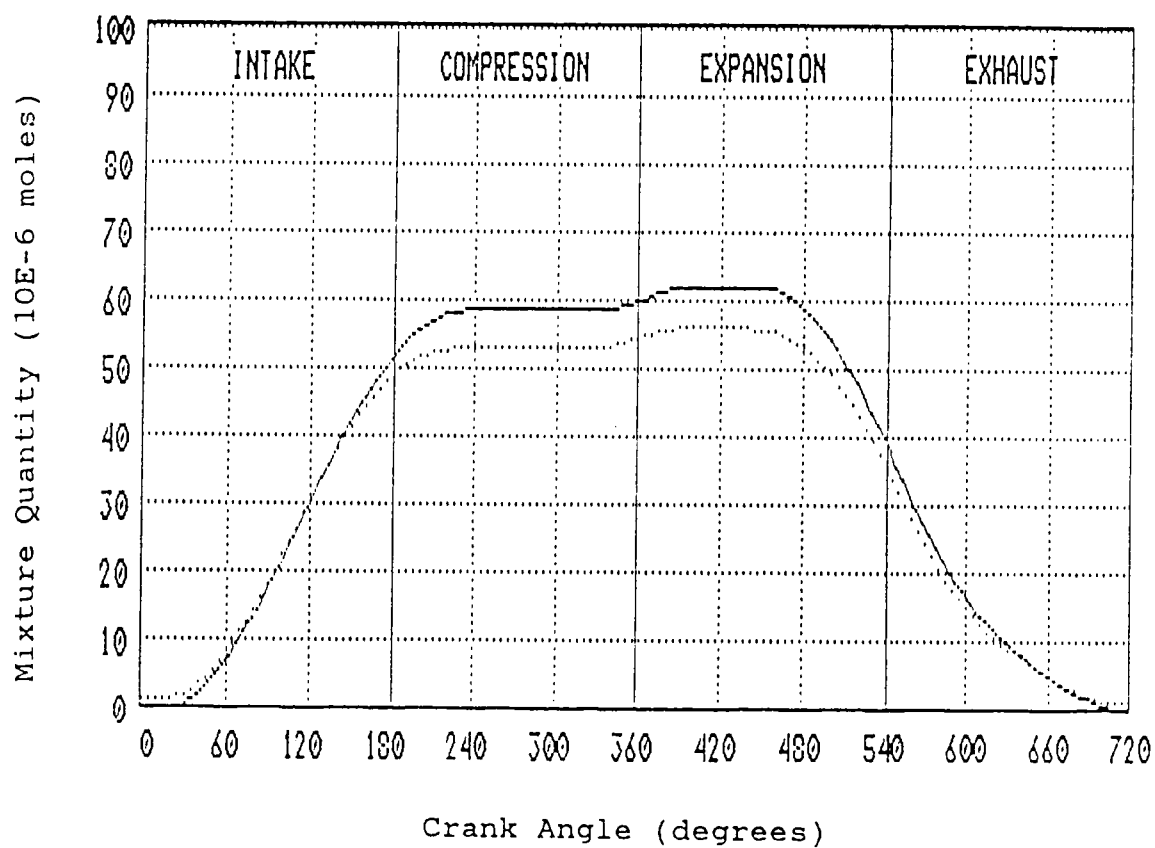
Figure 6. Intake Flow Rate vs Crank Angle



Conventional

Variable-Stroke ———

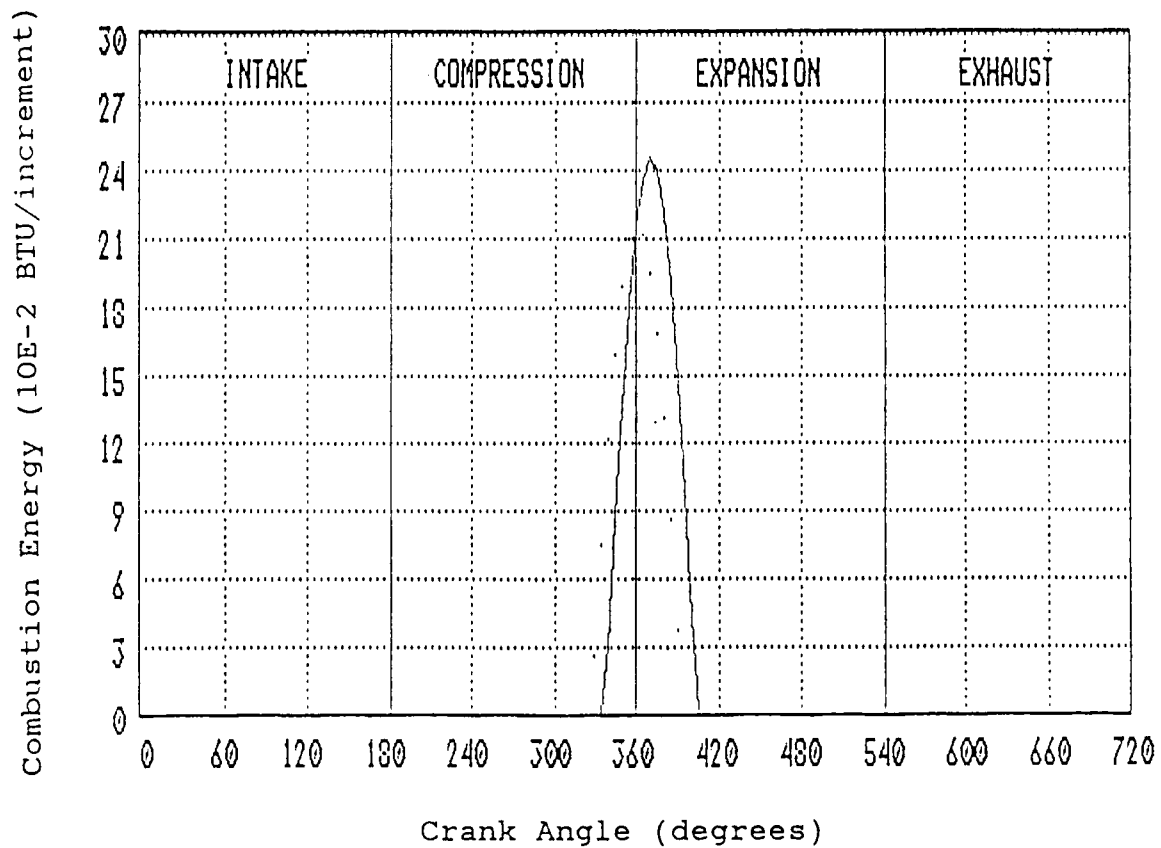
Figure 7. Exhaust Flow Rate vs Crank Angle



Conventional - - - - -

Variable-Stroke ———

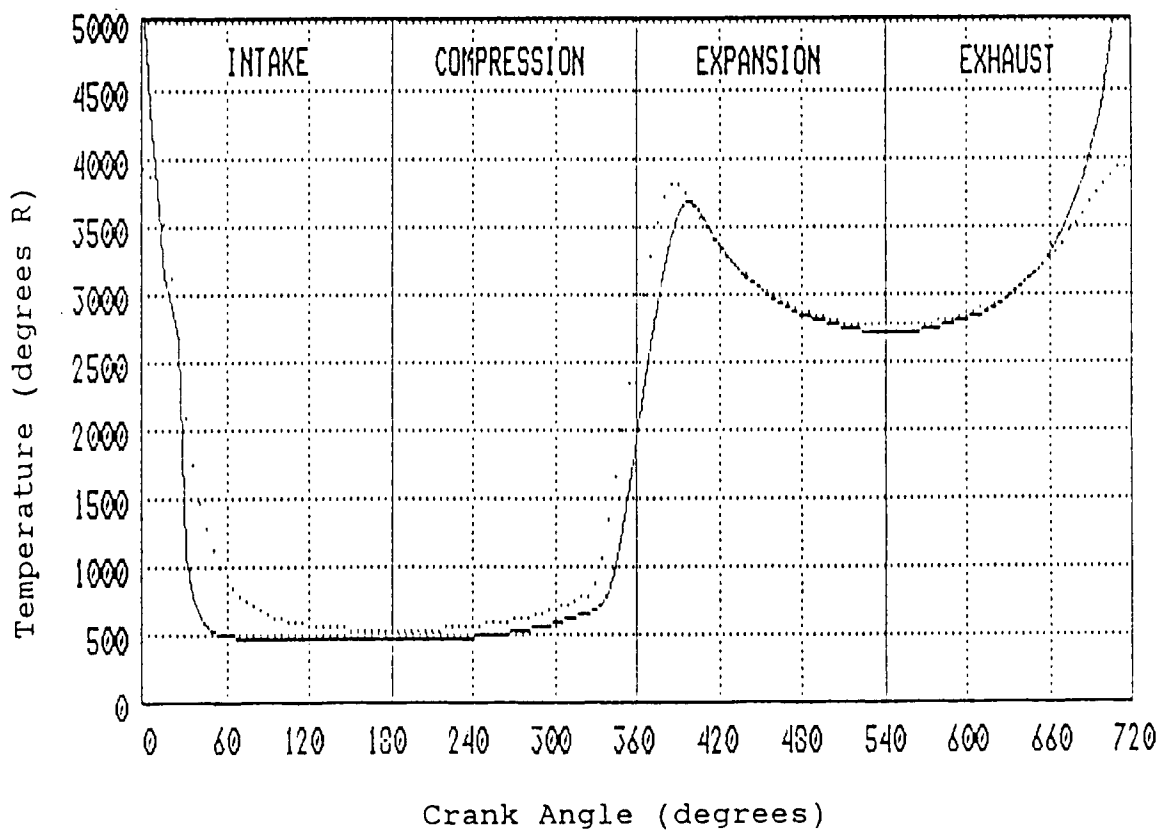
Figure 8. Mixture Quantity vs Crank Angle



Conventional - - - - -

Variable-Stroke ———

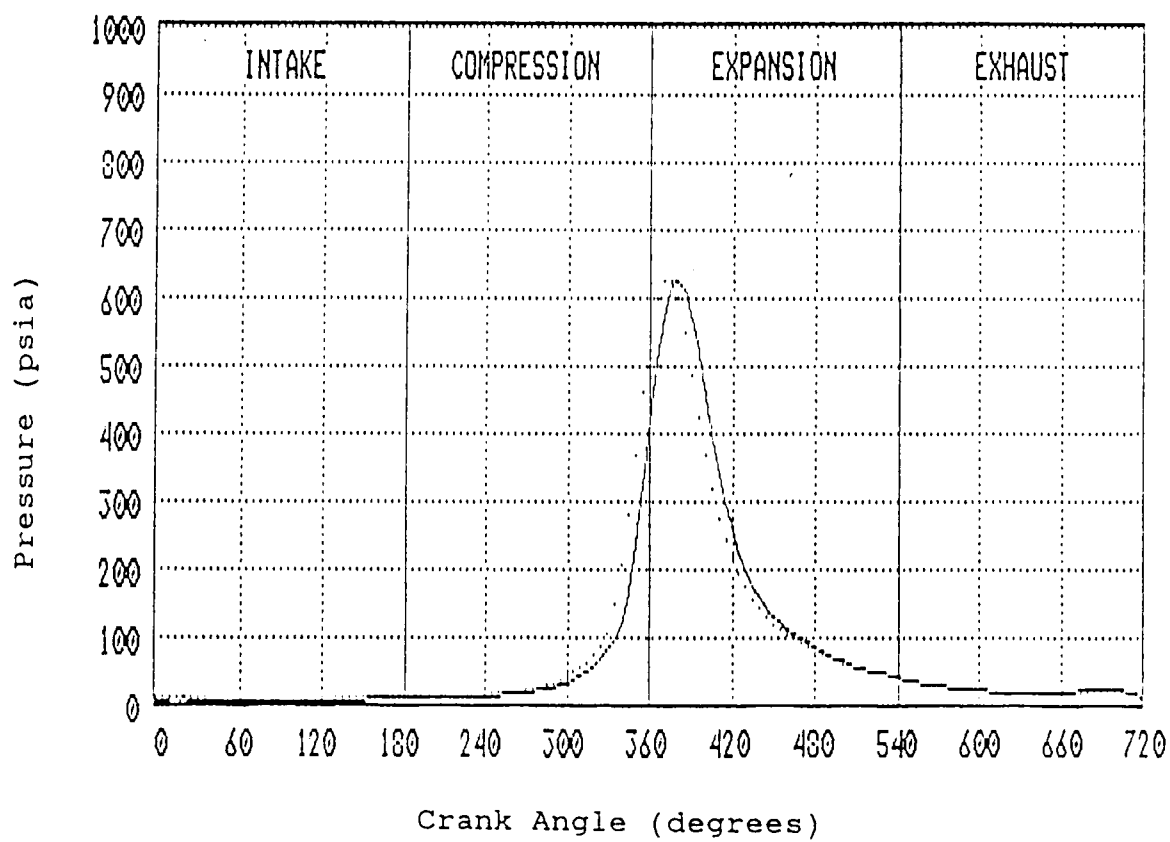
Figure 9. Combustion Energy vs Crank Angle



Conventional -

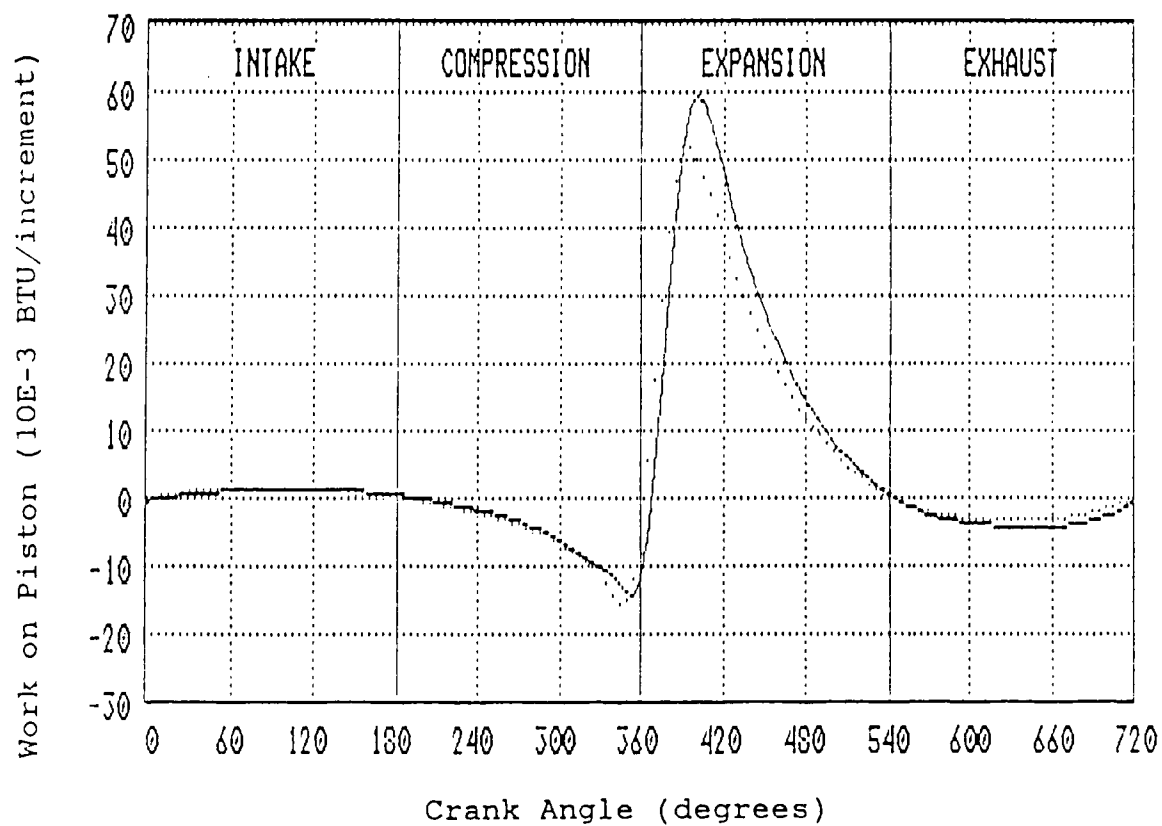
Variable-Stroke ———

Figure 10. Mixture Temperature vs Crank Angle



Conventional - - - - -
 Variable-Stroke ———

Figure 11. Mixture Pressure vs Crank Angle



Conventional - - - -

Variable-Stroke ———

Figure 12. Work On Piston vs Crank Angle

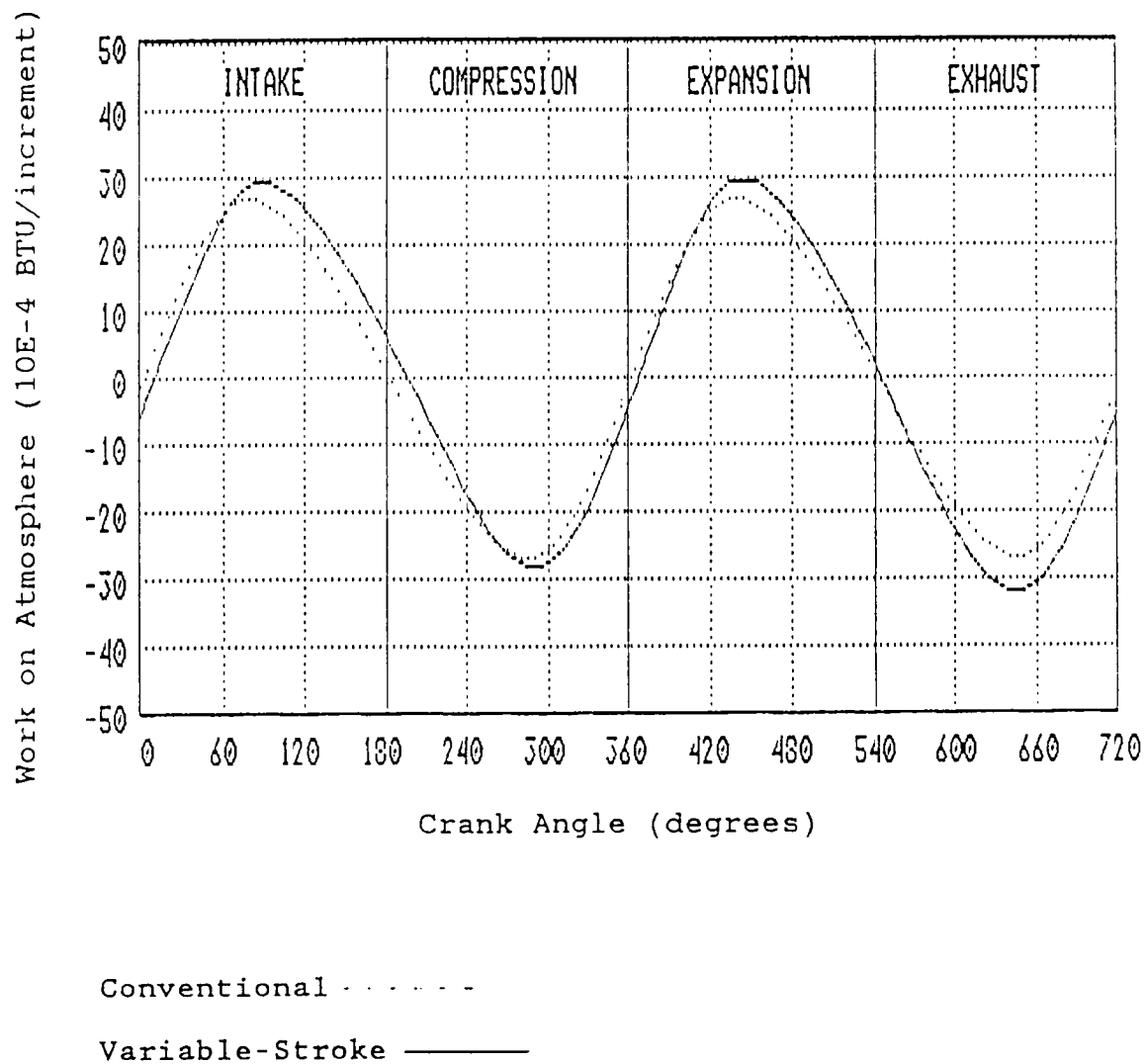
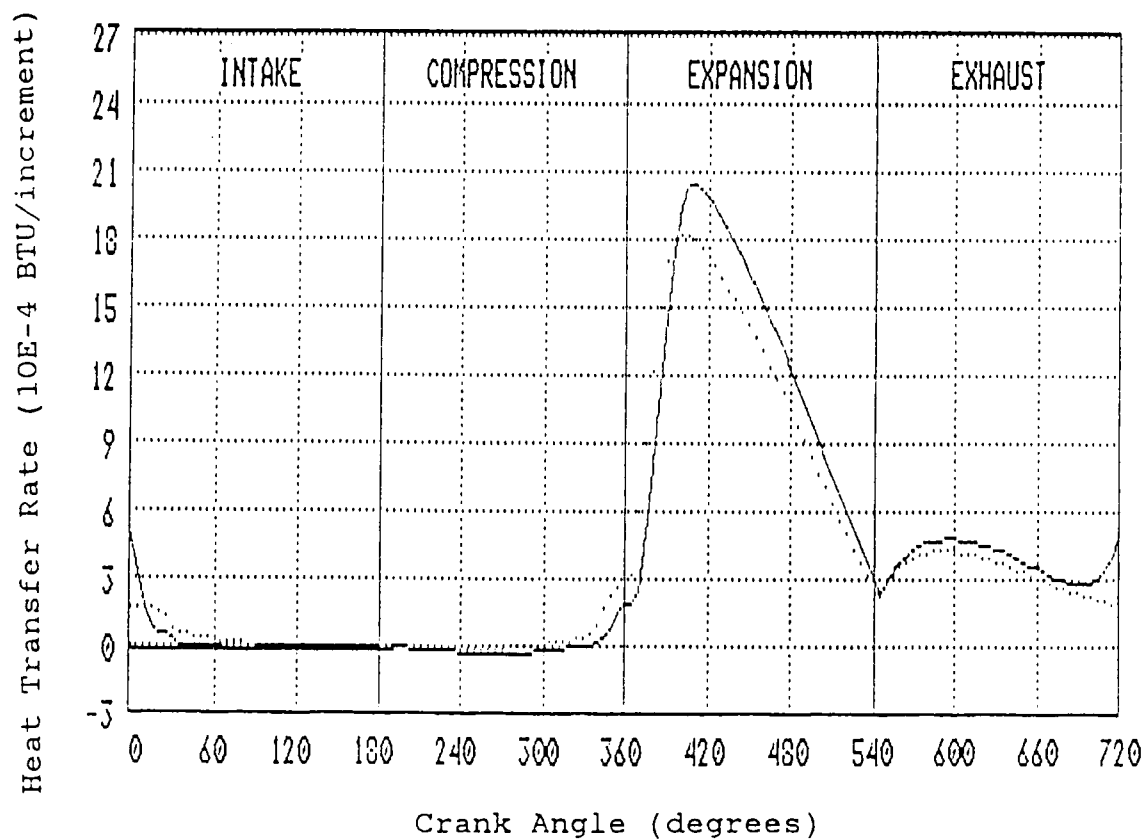


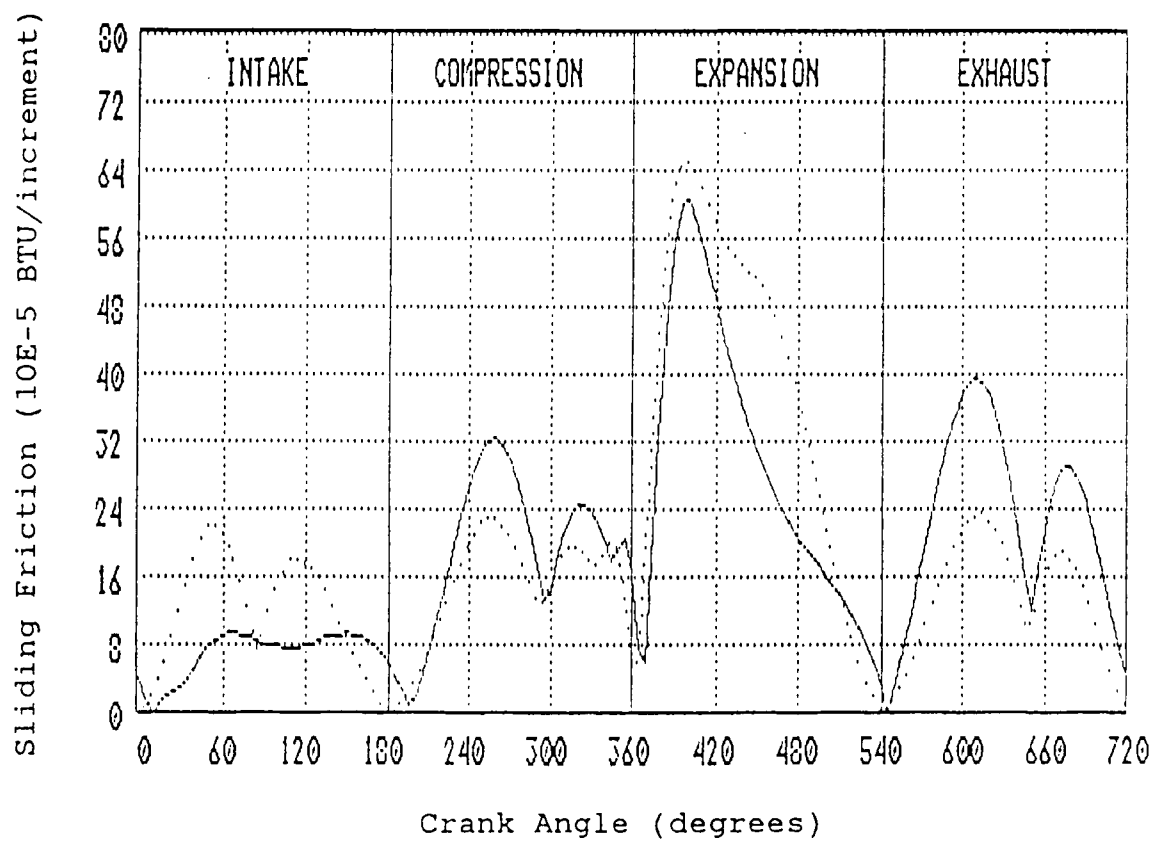
Figure 13. Work On Atmosphere vs Crank Angle



Conventional

Variable-Stroke ———

Figure 14. Heat Transfer Rate vs Crank Angle



Conventional - - - - -

Variable-Stroke ———

Figure 15. Sliding Friction vs Crank Angle

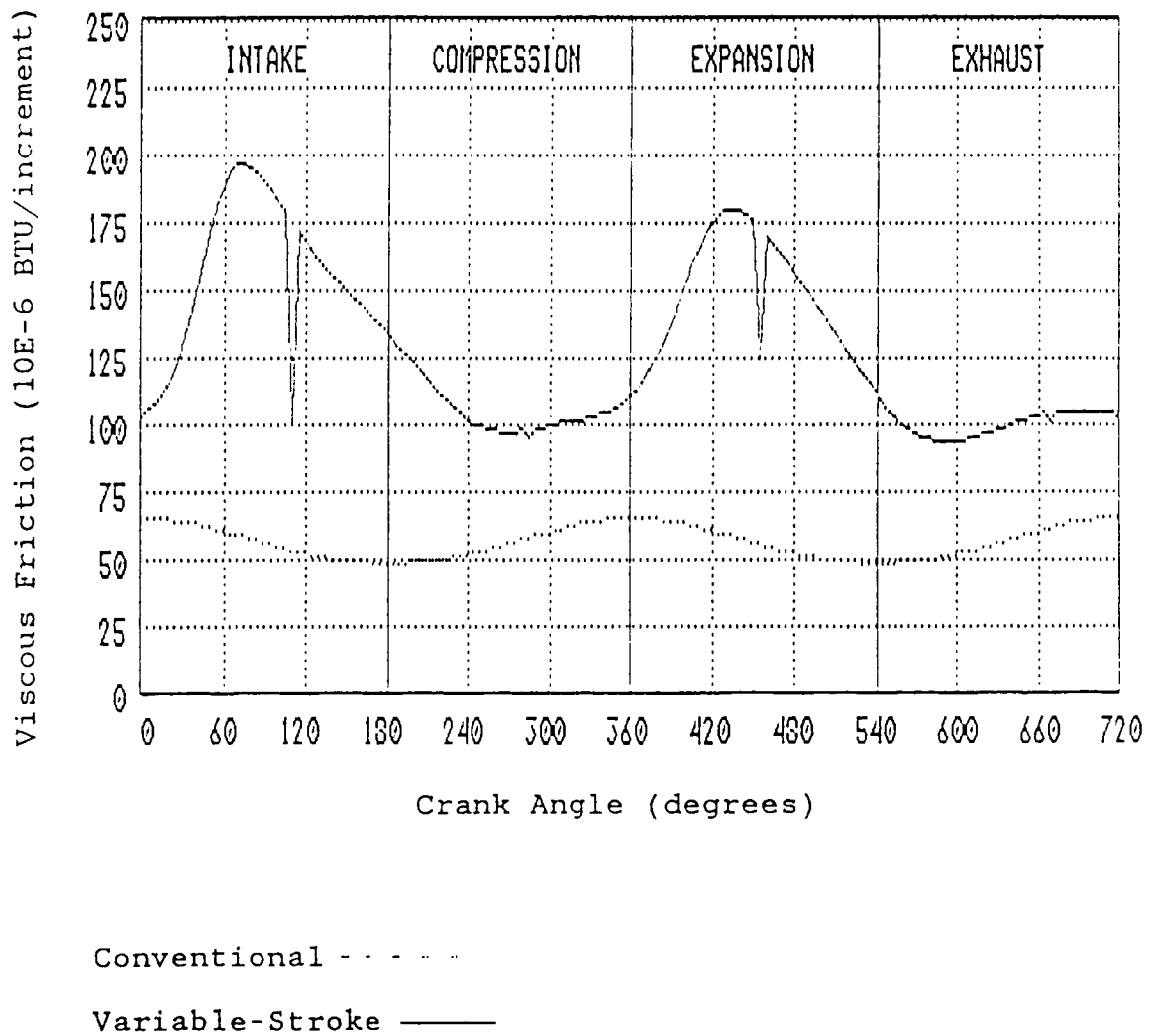
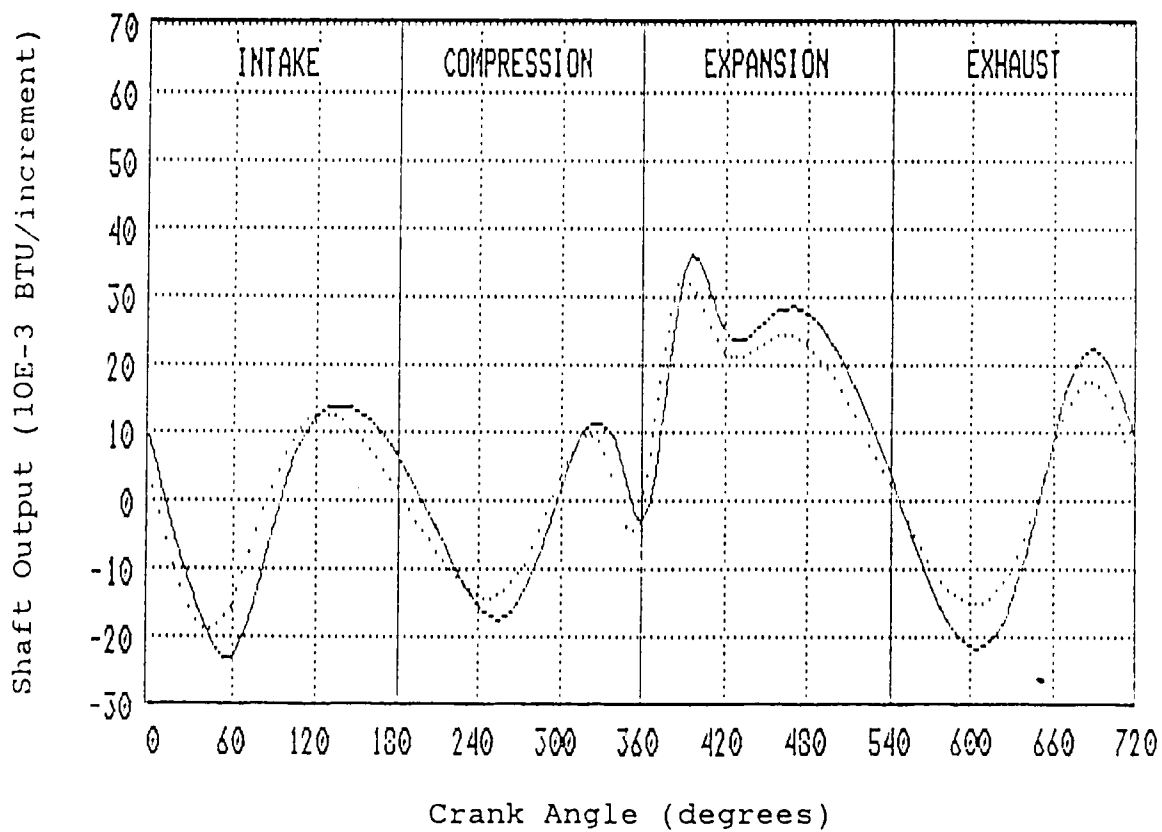


Figure 16. Viscous Friction vs Crank Angle



Conventional - - - - -

Variable-Stroke —

Figure 17. Shaft Output vs Crank Angle

APPENDIX B.

NOMENCLATURE AND SYMBOLS

TABLE II. SYMBOLS USED IN THE ENGINE MODEL PROGRAMS

Symbol	Definition	Units
A1	crank angle (from vertical)	radians
A1OLD	previous value of A1	radians
A2	angle of connecting rod from vertical	radians
A2OLD	previous value of A2	radians
A3	crankpin angle	radians
A3OLD	previous value of A3	radians
A4	angle in the variable-stroke engine mechanism	radians
A4OLD	previous value of A4	radians
A5	angle in the variable-stroke engine mechanism	radians
A5OLD	previous value of A5	radians
A7	angle in the variable-stroke engine mechanism	radians
A7OLD	previous value of A7	radians
ALPHA	angle of second shaft in the variable-stroke engine mechanism	radians
ALPHA0	initial value of ALPHA	radians
ANGLE	crank angle	degrees
AREA	cross-sectional area of cylinder bore	sq. inches
AREA1	journal bearing area	sq. inches
AREA2	journal bearing area	sq. inches

TABLE II. (CONTINUED)

Symbol	Definition	Units
AREA3	journal bearing area	sq. inches
AREA4	journal bearing area	sq. inches
AREA5	journal bearing area	sq. inches
AREA6	journal bearing area	sq. inches
AREA7	journal bearing area	sq. inches
ATMPRESS	atmospheric pressure	psia
ATMTEMP	atmospheric temperature	degrees R
ATMWORK	piston work on atmosphere	BTU
AVGACC	average acceleration of the piston over increment	inches per sec. squared
AVGVEL	average velocity of piston over incremental angle	in. per sec.
BETA	angle in the variable-stroke engine mechanism	radians
BHP	instantaneous output power	horsepower
BORE	diameter of cylinder bore	inches
BURN	amount of fuel burned during incremental angle	moles
BURNANGLE	total duration of combustion	degrees
C0	variable used for calc. of gas mixture specific heat	none
C1	variable used for calc. of gas mixture specific heat	1/deg K
C2	variable used for calc. of gas mixture specific heat	1/deg K squared

TABLE II. (CONTINUED)

Symbol	Definition	Units
C3	variable used for calc. of gas mixture specific heat	1/deg K cubed
C4	variable used for calc. of gas mixture specific heat	1/deg K to 4th power
CLRVOL	clearance volume in cylinder with piston at TDC	cubic inches
COMPRESS	compression ratio	none
CONST1	angle in the variable-stroke engine mechanism	radians
CONST2	angle in the variable-stroke engine mechanism	radians
COOLANTAREA	surface area of cylinder wall exposed to coolant flow	sq. inches
COOLANTFACTOR	curve-fit temperature function of combined coolant properties	(see Heat Transfer Discussion)
COOLANTFILMTEMP	film temperature used for evaluation of coolant heat transfer properties	deg R
COOLANTTEMP	bulk temperature of coolant	deg R
COOLANTVEL	average velocity of coolant over heat transfer surfaces	in. per sec.
CP	gas mixture heat capacity (constant pressure)	BTU per mole degree R
CRITICAL	critical pressure ratio for choked flow	none
CV	gas mixture heat capacity (constant pressure)	BTU per mole degree R
CYCLE	number of four-stroke cycles	none

TABLE II. (CONTINUED)

Symbol	Definition	Units
CYCLEHP	net horsepower delivered to crankshaft during cycle	horsepower
D1	journal bearing diameter	inches
D2	journal bearing diameter	inches
D3	journal bearing diameter	inches
D4	journal bearing diameter	inches
D5	journal bearing diameter	inches
D6	journal bearing diameter	inches
D7	journal bearing diameter	inches
DELKE	change in kinetic energy	BTU
DIRECTION	relative rotation direction of the variable-stroke engine shafts	none
DISPL	swept volume of piston on compression stroke	cubic inches
E2	intermediate value for gas enthalpy	BTU
EC	distance in variable-stroke mechanism solution	inches
ENTHALPY	enthalpy of gas mixture in cylinder	BTU
EXCLOSE	crank angle (0 to 720) where exhaust valve closes	degrees
EXCONST	exhaust valve flow constant at a given crank angle	(see Exhaust Discussion)
EXENTHALPY	enthalpy leaving cylinder during incremental angle	BTU

TABLE II. (CONTINUED)

Symbol	Definition	Units
EXFLOW	amount of gas mixture moles exhausted during increment	(see Exhaust Discussion)
EXMAX	flow constant with fully open exhaust valve	(see Exhaust Discussion)
EXOPEN	crank angle (0 to 720) where exhaust valve begins opening	degrees
FORCE	downward force on piston from inertia and pressure differential	lbf
FRICITION	Coulomb sliding friction coefficient	none
FUEL	total amount of fuel in cylinder immediately before ignition	moles
GAMMA	angle in the variable-stroke engine mechanism	radians
GASAREA	surface area of cylinder exposed to gas mixture	sq. inches
GASDENS	density of the gas mixture	lbm per cubic inch
GASFACTOR	curve-fit temperature function of combined gas mixture properties	(see Heat Transfer Discussion)
GASFILMTEMP	temperature used for calc. of gas mixture heat transfer properties	deg R
GASVEL	nominal velocity of gas mixture over heat transfer surfaces	feet per second

TABLE II. (CONTINUED)

Symbol	Definition	Units
GC	English system unit conversion factor	in. lbm per lbf sec. sq.
H	distance in variable-stroke mechanism solution	inches
H1	journal bearing radial clearance	inches
H2	journal bearing radial clearance	inches
H3	journal bearing radial clearance	inches
H4	journal bearing radial clearance	inches
H5	journal bearing radial clearance	inches
H6	journal bearing radial clearance	inches
H7	journal bearing radial clearance	inches
HCOOLANT	heat transfer coefficient of coolant on cylinder wall	BTU per sec. sq. in. °R
HGAS	heat transfer coefficient of gas mixture on cylinder wall	BTU per sec. sq. in. °R
HYDIAM	hydraulic diameter of coolant jacket	inches
ICP	intake gas mixture heat capacity (constant pressure)	BTU per mole degree R

TABLE II. (CONTINUED)

Symbol	Definition	Units
ICRITICAL	critical pressure ratio of intake mixture	none
ICV	intake gas mixture heat capacity (constant volume)	BTU per mole degree R
IDEALC7H17	quantity of C7H17 in cylinder with 100% volumetric efficiency	moles
IK	ratio of specific heats of gas mixture	none
INCLOSE	crank angle (0 to 720) where intake valve closes	degrees
INCONST	intake valve flow constant at a given crank angle	(see Intake Discussion)
INCREMENT	crank angle step at which all variables are evaluated	degrees
INENTHALPY	enthalpy entering cylinder during incremental angle	BTU
INFLOW	amount of gas mixture ingested during increment	moles
INMAX	flow constant with fully-open intake valve	(see Intake Discussion)
INOPEN	crank angle (0 to 720) where intake valve begins opening	degrees
K	ratio of specific heats of gas mixture	none
KELV	Kelvin temperature of the gas mixture	degrees K
L1	variable-stroke engine linkage length	inches

TABLE II. (CONTINUED)

Symbol	Definition	Units
L2	variable-stroke engine linkage length	inches
L3	variable-stroke engine linkage length	inches
L4	variable-stroke engine linkage length	inches
M	distance in variable-stroke mechanism solution	inches
MAX2AT	variable-stroke crank angle at TDC	degrees
MAXAT	variable-stroke crank angle at TDC	degrees
MAXC7H17	peak quantity of C7H17 in the cylinder	moles
METALCP	heat capacity of the cylinder wall material	BTU per lbm degree R
METALDENSITY	density of the cylinder wall material	lbm per cubic inch
MIN2AT	variable-stroke crank angle at BDC	degrees
MINAT	variable-stroke crank angle at BDC	degrees
MW	molecular weight of the gas mixture	lbm per mole
MWC7H17	molecular weight of C7H17 (gasoline)	lbm per mole
MWC02	molecular weight of CO2 (carbon-dioxide)	lbm per mole

TABLE II. (CONTINUED)

Symbol	Definition	Units
MWH2O	molecular weight of H2O (water vapor)	lbm per mole
MWI	molecular weight of the intake gas mixture	lbm per mole
MWN2	molecular weight of N2 (nitrogen)	lbm per mole
MW02	molecular weight of O2 (oxygen)	lbm per mole
NC7H17	amount of C7H17 in the gas mixture	moles
NC7H17I	amount of C7H17 in the intake gas mixture	moles
NC02	amount of CO2 in the gas mixture	moles
NC02I	amount of CO2 in the intake gas mixture	moles
NH2O	amount of H2O in the gas mixture	moles
NH2OI	amount of H2O in the intake gas mixture	moles
NN2	amount of N2 in the gas mixture	moles
NN2I	amount of N2 in the intake gas mixture	moles
NO2	amount of O2 in the gas mixture	moles
NO2I	amount of O2 in the intake gas mixture	moles

TABLE II. (CONTINUED)

Symbol	Definition	Units
NSUM	amount of gas mixture in the cylinder	moles
NSUMR	amount of gas mixture in intake manifold due to reverse flow	moles
OUTDIAM	cylindrical diameter of coolant jacket	inches
P2	intermediate value used in mixture pressure calculation	psia
PI	geometric constant	none
PISTONMASS	mass of moving piston including rings and pin	lbm
PRESS	pressure of gas mixture in the cylinder	psia
QCOMB	energy released by burning fuel over an incremental angle	BTU
QCOOLANT	heat energy transferred to the coolant over an increment	BTU
QWALLS	heat energy transferred from the gas mixture to the cylinder walls	BTU
R	specific gas constant of the gas mixture in the cylinder	BTU per mole degree R
R0	universal gas constant	BTU per mole degree R
R1	variable-stroke engine linkage length	inches
R2	variable-stroke engine linkage length	inches

TABLE II. (CONTINUED)

Symbol	Definition	Units
RETFLOW	amount of reverse-flow gas mixture ingested during an incremental angle	moles
REVFLOW	amount of gas mixture which reverse flows through intake valve during increment	moles
RODLENGTH	length of connecting rod between bearing center lines	inches
ROTATE	energy dissipated by viscous drag	BTU
RPM	rotational speed of crankshaft	rotations per min.
S	distance in variable-stroke mechanism solution	inches
SFAC	variable-stroke mechanism scaling factor	none
SHAFT	instantaneous shaft output energy	BTU
SLIDE	energy dissipated by sliding friction	BTU
SPARKANGLE	crank angle (0 to 720) where combustion begins	degrees
STROKE	total vertical travel of the piston in the cylinder	inches
SUMBHP	sum of BHP over the full 720-degree cycle	horsepower
SUMDELKE	sum of DELKE over the full 720-degree cycle	BTU
SUMEXENTHALPY	sum of EXENTHALPY over the full 720-degree cycle	BTU

TABLE II. (CONTINUED)

Symbol	Definition	Units
SUMINENTHALPY	sum of INENTHALPY over the full 720-degree cycle	BTU
SUMQCOMB	sum of QCOMB over the full 720-degree cycle	BTU
SUMQWALLS	sum of QWALLS over the full 720-degree cycle	BTU
SUMROTATE	sum of ROTATE over the full 720-degree cycle	BTU
SUMSHAFT	sum of SHAFT over the full 720-degree cycle	BTU
SUMSLIDE	sum of SLIDE over the full 720-degree cycle	BTU
T2	intermediate value of gas mixture temperature	deg R
TEMP	temperature of the gas mixture in the cylinder	deg R
THROW	the crankpin offset from the crank center line	inches
TIME	duration of each crank angle increment	seconds
TORQUE	instantaneous torque output to crankshaft	foot lbf
V1	variable used in calculation of viscous friction	cubic inches
V2	variable used in calculation of viscous friction	cubic inches
V3	variable used in calculation of viscous friction	cubic inches

TABLE II. (CONTINUED)

Symbol	Definition	Units
VISCOSITY	oil viscosity	lbm per inch second
VOLUME	volume of the gas mixture in the cylinder	cubic inches
W2	intermediate value for WORK	BTU
WALLHT	cylindrical height of the coolant jacket	inches
WALLMASS	mass of walls between gas mixture and coolant jacket	lbm
WALLTEMP	nominal temp. of cylinder walls during increment	deg R
WALLTHK	thickness of walls between the gas mixture and coolant jacket	inches
WORK	pressure-volume work of the gas mixture on the piston	BTU
XB	variable-stroke mechanism bearing coordinate	inches
XC	variable-stroke mechanism bearing coordinate	inches
XC7H17	mole fraction of C7H17 in the gas mixture	none
XC7H17I	mole fraction of C7H17 in the intake gas mixture	none
XCO2	mole fraction of CO2 in the gas mixture	none
XD	variable-stroke mechanism bearing coordinate	inches

TABLE II. (CONTINUED)

Symbol	Definition	Units
XE	variable-stroke mechanism bearing coordinate	inches
XF	variable-stroke mechanism bearing coordinate	inches
XG	variable-stroke mechanism bearing coordinate	inches
XH2O	mole fraction of H2O in the gas mixture	none
XN2	mole fraction of N2 in the gas mixture	none
XN2I	mole fraction of N2 in the intake gas mixture	none
XO2	mole fraction of O2 in the gas mixture	none
XO2I	mole fraction of O2 in the intake gas mixture	none
Y1	the value of Y3 prior to the last two incremental piston movements	inches
Y2	the value of Y3 prior to the last incremental movement	inches
Y3	the instantaneous piston pin height	inches
YA	variable-stroke mechanism bearing coordinate	inches
YAMAX	a maximum value of YA	inches
YAMAX2	a maximum value of YA	inches
YAMIN	a minimum value of YA	inches

TABLE II. (CONCLUDED)

Symbol	Definition	Units
YAMIN2	a minimum value of YA	inches
YB	variable-stroke mechanism bearing coordinate	inches
YC	variable-stroke mechanism bearing coordinate	inches
YD	variable-stroke mechanism bearing coordinate	inches
YE	variable-stroke mechanism bearing coordinate	inches
YF	variable-stroke mechanism bearing coordinate	inches
YG	variable-stroke mechanism bearing coordinate	inches

APPENDIX C.

PROGRAM FOR CONVENTIONAL ENGINE

```

10 REM ***** CONVENT.BAS (CONVENTIONAL ENGINE MODEL) *****
15 REM *****
20 OPEN "C:MAIN1.PRN" FOR OUTPUT AS #1
22 OPEN "C:MAIN2.PRN" FOR OUTPUT AS #2
24 OPEN "C:MAIN3.PRN" FOR OUTPUT AS #3
30 PI=3.141593 : R0=1.98588 :REM BTU/MOLE-R
40 DEF FNASIN(A)=ATN(A/SQR(1-A*A))
50 DEF FNACOS(A)=-ATN(A/SQR(-A*A+1))+PI/2
60 GC=12*32.1739 : REM IN-LBM/LBF-SEC^2

80 REM ***** CONVENTIONAL ENGINE CHARACTERISTICS *****
85 REM *****
90 READ RODLENGTH, BORE, DISPL, COMPRESS, RPM, INCREMENT
100 DATA 8, 3.735, 38.125, 8.6, 3500, 5
110 READ PISTONMASS, D1, D2, D3, FRICTION
120 DATA 3, 2.5, 1.5, 2, .05
130 READ AREA1, AREA2, AREA3, H1, H2, H3, VISCOSITY
140 DATA 9.8, 3.5, 6.3, .0025/2, .0015/2, .0020/2, .00068
150 READ INMAX, INOPEN, INCLOSE, EXMAX, EXOPEN, EXCLOSE
160 DATA .12, 695, 260, .12, 440, 60
170 READ SPARKANGLE
180 DATA 325
190 READ COOLANTTEMP, COOLANTVEL, HYDIAM, WALLTHK, METALDENSITY, METALCP
200 DATA 650, 25, .5, .25, .28, .11
210 AREA=BORE^2*PI/4 : STROKE=DISPL/AREA : THROW=STROKE/2
220 CLRVOL=DISPL/(COMPRESS-1)
225 BURNANGLE=37+RPM/120
230 OUTDIAM=BORE+2*WALLTHK
240 WALLHT=(DISPL+CLRVOL)/AREA
250 COOLANTAREA=(OUTDIAM)^2*PI/4+(OUTDIAM)*PI*WALLHT
260 WALLMASS=(AREA*WALLTHK+(OUTDIAM^2-BORE^2)*PI/4*WALLHT)*METALDENSITY
270 TIME=INCREMENT/RPM*60/360 : REM SECONDS

290 REM ***** INLET CONDITIONS *****
295 REM *****
300 ATPRESS=14.696:ATMTEMP=537
310 READ NC7H17I, NN2I, NO2I, NCO2I, NH2OI
320 DATA 1, 42.32, 11.25, 0, 0 : REM THEORETICAL BALANCE
330 XC7H17I=NC7H17I/(NC7H17I+NN2I+NO2I)
340 XN2I=NN2I/(NC7H17I+NN2I+NO2I)
350 XO2I=NO2I/(NC7H17I+NN2I+NO2I)
360 READ MWC7H17, MWN2, MWO2, MWCO2, MWH2O : REM MOLECULAR WEIGHTS
370 DATA 101, 28, 32, 44, 18
380 MWI=XC7H17I*MWC7H17+XN2I*MWN2+XO2I*MWO2
385 IDEALC7H17=XC7H17I*ATMPRESS*(DISPL+CLRVOL)/R0/ATMTEMP/12/778.16

```

```

400 REM ***** STARTING CONDITIONS *****
405 REM *****
410 NC02=0:NH20=0:XC7H17=XC7H17I:XN2=XN2I:XO2=XO2I
420 MW=XC7H17*MWC7H17+XN2*MWN2+XO2*MWO2:R=R0/MW
430 PRESS=ATMPRESS:TEMP=ATMTEMP
440 VOLUME=CLRVOL
450 NSUM=PRESS*VOLUME/R0/TEMP/12/778.16
460 NC7H17=XC7H17*NSUM:NN2=XN2*NSUM:NO2=XO2*NSUM
470 GOSUB 1590 : REM SPECIFIC HEATS
480 ICV=CV:ICP=CP:IK=ICP/ICV:ICRITICAL=(2/(IK+1))^(IK/(IK-1))
485 ENTHALPY=0
490 WALLTEMP=654 : REM DEGREES RANKINE
500 Y1=RODLENGTH+THROW : Y2=Y1 : Y3=Y2

520 REM ***** MAIN PROGRAM *****
525 REM *****
530 FOR CYCLE=1 TO 5
540     PRINT "CYCLE=";CYCLE
550     CYCLEHP=0:SUMBHP=0:SUMQCOMB=0:SUMQWALLS=0:SUMSLIDE=0
555     SUMROTATE=0:SUMSHAFT=0:SUMINENTHALPY=0:SUMEXENTHALPY=0:SUMDELKE=0
560     FOR ANGLE=0 TO 720-INCREMENT STEP INCREMENT
570         GOSUB 750 : REM PISTON POSITION
580         GOSUB 900 : REM INTAKE
590         GOSUB 1200 : REM EXHAUST
600         GOSUB 1470 : REM MASS FLOW
610         GOSUB 1590 : REM SPECIFIC HEATS
620         GOSUB 1770 : REM COMBUSTION
630         GOSUB 1910 : REM HEAT TRANSFER
650         GOSUB 2062 : REM THERMODYNAMICS AND SHAFT OUTPUT
652         SUMQCOMB=SUMQCOMB+QCOMB:SUMQWALLS=SUMQWALLS+QWALLS
654         SUMSLIDE=SUMSLIDE+SLIDE:SUMROTATE=SUMROTATE+ROTATE
656         SUMSHAFT=SUMSHAFT+SHAFT:SUMINENTHALPY=SUMINENTHALPY+INENTHALPY
658         SUMDELKE=SUMDELKE+DELKE:SUMEXENTHALPY=SUMEXENTHALPY+EXENTHALPY
660         IF CYCLE<5 THEN GOTO 705
670         REM OUTLIST ANGLE AND CALCULATED VALUES:
680         PRINT#1,USING "#####.##";ANGLE,
682         PRINT#1,USING "  +#.###^" ;VOLUME,PRESS,TEMP,WORK,ATMWORK
685         PRINT#2,USING "#####.##";ANGLE,
690         PRINT#2,USING "  +#.###^" ;NSUM,QCOMB,QWALLS,INENTHALPY,_
            EXENTHALPY,ENTHALPY
695         PRINT#3,USING "#####.##";ANGLE,
700         PRINT#3,USING "  +#.###^" ;SLIDE,ROTATE,SHAFT,AVGVEL,_
            AVGACC,DELKE
705     NEXT ANGLE
706     P$="###.###"
710     PRINT " V8 MOTORHP=",;PRINT USING P$;CYCLEHP*8
711     PRINT "  VOL.EFF.=",;PRINT USING P$;MAXC7H17/IDEALC7H17*100
712     PRINT "    QCOMB=",;PRINT USING P$;SUMQCOMB
713     PRINT "    QWALLS=",;PRINT USING P$;SUMQWALLS

```

```

714 PRINT " SLIDE=", :PRINT USING P$,SUMSLIDE
715 PRINT " ROTATE=", :PRINT USING P$,SUMROTATE
716 PRINT " SHAFT=", :PRINT USING P$,SUMSHAFT
717 PRINT " DELKE=", :PRINT USING P$,SUMDELKE
718 PRINT " INENTHALPY=", :PRINT USING P$,SUMINENTHALPY
719 PRINT " EXENTHALPY=", :PRINT USING P$,SUMEXENTHALPY
721 PRINT " TOTAL IN=", :PRINT USING P$,SUMQCOMB+SUMINENTHALPY
722 PRINT " TOTAL OUT=", :PRINT USING P$,SUMQWALLS+SUMSLIDE+SUMROTATE+_
    SUMSHAFT+SUMEXENTHALPY
725 NEXT CYCLE
730 CLOSE :PRINT "NORMAL TERMINATION":STOP

750 REM ***** PISTON POSITION SUBROUTINE *****
755 REM *****
760 A1OLD=A1 : A2OLD=A2 : A3OLD=A3
770 IF ANGLE=0 THEN A1OLD=-INCREMENT*PI/180
780 A1=ANGLE*PI/180
790 A2=FNASIN(THROW*SIN(A1)/RODLENGTH)
800 IF COS(A1)>.9999 THEN A3=PI : GOTO 830
810 IF COS(A1)<-.9999 THEN A3=0 : GOTO 830
820 A3=PI-FNACOS(COS(A1))-ABS(A2)
830 Y1=Y2 : Y2=Y3
840 Y3=THROW*COS(A1)+RODLENGTH*COS(A2)
850 AVGVEL=(Y3-Y2)/TIME
860 AVGACC=(Y3-2*Y2+Y1)/TIME^2
870 VOLUME=CLRVL+(THROW+RODLENGTH-Y3)*AREA
880 RETURN

900 REM ***** INTAKE SUBROUTINE *****
905 REM *****
910 INFLOW=0:REVFLOW=0:RETFLOW=0:INCONST=0
920 IF INCLOSE<INOPEN GOTO 960
930 IF ANGLE<=INOPEN THEN RETURN
940 IF ANGLE>=INCLOSE THEN RETURN
950 INCONST=INMAX/2*(SIN((2*(ANGLE-INOPEN)/(INCLOSE-INOPEN)-.5)*PI)+1):GOTO 990
960 IF ANGLE<=INCLOSE THEN INCONST=INMAX/2*(SIN((2*(ANGLE-INOPEN+720)/_
    (INCLOSE-INOPEN+720)-.5)*PI)+1):GOTO 990
970 IF ANGLE<=INOPEN THEN RETURN
980 INCONST=INMAX/2*(SIN((2*(ANGLE-INOPEN)/(INCLOSE+720-INOPEN)-.5)*PI)+1)
990 IF ATPRESS<PRESS THEN GOTO 1150 : REM REVERSE FLOW THROUGH INTAKE VALVE
995 DELPRESS=ATMPRESS-PRESS
996 IF ATPRESS*ICRITICAL>PRESS THEN DELPRESS=ATMPRESS*(1-ICRITICAL):REM CHOKED
1000 INFLOW=INCONST*TIME*SQR(DELPRESS*ATMPRESS/ATMTEMP/MWI):REM MOLES
1010 P2=(NSUM+INFLOW)*R0*12*778.16*TEMP/VOLUME
1020 IF ATPRESS<(P2+PRESS)/2 THEN INFLOW=0 : GOTO 1040 : REM AVOID INSTABILITY
1025 DELPRESS=ATMPRESS-(P2+PRESS)/2
1026 IF ATPRESS*ICRITICAL>(P2+PRESS)/2 THEN DELPRESS=ATMPRESS*(1-ICRITICAL)
1030 INFLOW=INCONST*TIME*SQR(DELPRESS*ATMPRESS/ATMTEMP/MWI)
1040 IF NSUM<=0 THEN RETURN

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```

1060 RETFLOW=INCONST*TIME*SQR((ATMPRESS-PRESS)*ATMPRESS/TEMP/MW)
1070 P2=(NSUM+RETFLOW)*R0*12*778.16*TEMP/VOLUME
1080 IF (P2+PRESS)/2 > ATMPRESS THEN RETFLOW=0 : RETURN : REM AVOID INSTABILITY
1090 RETFLOW=INCONST*TIME*SQR((ATMPRESS-(P2+PRESS)/2)*ATMPRESS/TEMP/MW)
1100 IF RETFLOW>NSUMR THEN RETFLOW=NSUMR
1110 NSUMR=NSUMR-RETFLOW
1120 INFLOW=INFLOW-RETFLOW : REM BACKFLOW IS REINGESTED BEFORE FRESH CHARGE
1130 IF INFLOW < 0 THEN INFLOW=0
1140 RETURN
1150 REVFLOW=INCONST*TIME*SQR((PRESS-ATMPRESS)*PRESS/TEMP/MW)
1160 P2=(NSUM-REVFLOW)*R0*12*778.16*TEMP/VOLUME
1170 IF (P2+PRESS)/2 < ATMPRESS THEN REVFLOW=0 : RETURN : REM AVOID INSTABILITY
1180 REVFLOW=INCONST*TIME*SQR(((P2+PRESS)/2-ATMPRESS)*(P2+PRESS)/2/TEMP/MW)
1190 NSUMR=NSUMR+REVFLOW : RETURN

1200 REM ***** EXHAUST SUBROUTINE *****
1205 REM *****
1210 EXFLOW=0:EXCONST=0
1220 IF EXCLOSE<EXOPEN GOTO 1260
1230 IF ANGLE<=EXOPEN THEN RETURN
1240 IF ANGLE>=EXCLOSE THEN RETURN
1250 EXCONST=EXMAX/2*(SIN((2*(ANGLE-EXOPEN)/(EXCLOSE-EXOPEN)-.5)*PI)+1)
1255 GOTO 1290
1260 IF ANGLE<=EXCLOSE THEN EXCONST=EXMAX/2*(SIN((2*(ANGLE-EXOPEN+720)/_
    (EXCLOSE-EXOPEN+720)-.5)*PI)+1):GOTO 1290
1270 IF ANGLE<=EXOPEN THEN RETURN
1280 EXCONST=EXMAX/2*(SIN((2*(ANGLE-EXOPEN)/(EXCLOSE+720-EXOPEN)-.5)*PI)+1)
1290 IF PRESS<ATMPRESS THEN RETURN
1292 K=CP/CV : CRITICAL=(2/(K+1))^(K/(K-1))
1294 DELPRESS=PRESS-ATMPRESS
1296 IF PRESS*CRITICAL>ATMPRESS THEN DELPRESS=PRESS*(1-CRITICAL) : REM CHOKED
1300 EXFLOW=EXCONST*TIME*SQR(DELPRESS*PRESS/TEMP/MW):REM MOLES
1310 P2=(NSUM-EXFLOW)*R0*12*778.16*TEMP/VOLUME
1320 IF (P2+PRESS)/2<ATMPRESS THEN EXFLOW=0:RETURN : REM AVOID INSTABILITY
1325 DELPRESS=(P2+PRESS)/2-ATMPRESS
1326 IF (P2+PRESS)/2*CRITICAL>ATMPRESS THEN DELPRESS=(P2+PRESS)/2*(1-CRITICAL)
1330 EXFLOW=EXCONST*TIME*SQR(DELPRESS*(P2+PRESS)/2/TEMP/MW)
1340 RETURN

1470 REM ***** MASS FLOW SUBROUTINE *****
1475 REM *****
1480 NC7H17=NC7H17+XC7H17I*INFLOW-XC7H17*(EXFLOW+REVFLOW-RETFLOW)
1490 NN2=NN2+XN2I*INFLOW-XN2*(EXFLOW+REVFLOW-RETFLOW)
1500 NO2=NO2+XO2I*INFLOW-XO2*(EXFLOW+REVFLOW-RETFLOW)
1510 NCO2=NCO2+XCO2I*INFLOW-XCO2*(EXFLOW+REVFLOW-RETFLOW)
1520 NH2O=NH2O+XH2OI*INFLOW-XH2O*(EXFLOW+REVFLOW-RETFLOW)
1530 NSUM=NC7H17+NN2+NO2+NCO2+NH2O : REM MOLES
1540 XC7H17=NC7H17/NSUM:XN2=NN2/NSUM:XO2=NO2/NSUM:XCO2=NCO2/NSUM:XH2O=NH2O/NSUM
1550 MW=XC7H17*MWNC7H17+XN2*MWNN2+XO2*MWNO2+XCO2*MWNCO2+XH2O*MWNH2O

```

```

1560 R=R0/MM : IF ANGLE=INCLOSE THEN MAXC7H17=NC7H17
1570 RETURN

1590 REM ***** SPECIFIC HEAT SUBROUTINE *****
1595 REM *****
1600 IF TEMP>1800 GOTO 1670
1610 C0=4.06*XC7H17+2.4008*XC02+4.0701*XH2O+3.6748*XN2+3.6256*X02
1620 C1=.061*XC7H17+8.740001E-03*XC02-.00111*XH2O-.00121*XN2-.00188*X02
1630 C2=-1.88E-06*XC7H17-6.61E-06*XC02+4.15E-06*XH2O+2.32E-06*XN2+7.06E-06*X02
1640 C3=2E-09*XC02-2.96E-09*XH2O-6.32E-10*XN2-6.76E-09*X02
1650 C4=6.33E-16*XC02+8.07E-13*XH2O-2.26E-13*XN2+2.16E-12*X02
1660 GOTO 1720
1670 C0=4.06*XC7H17+4.46*XC02+2.716*XH2O+2.896*XN2+3.62*X02
1680 C1=.061*XC7H17+.0031*XC02+.002945*XH2O+.001515*XN2+.000736*X02
1690 C2=-1.88E-06*XC7H17-1.24E-06*XC02-8.02E-07*XH2O-5.72E-07*XN2-1.965E-07*X02
1700 C3=2.27E-10*XC02+1.02E-10*XH2O+9.98E-11*XN2+3.62E-11*X02
1710 C4=-1.55E-14*XC02-4.847E-15*XH2O-6.52E-15*XN2-2.89E-15*X02
1720 KELV=TEMP*5/9
1730 CP=(C0+C1*KELV+C2*KELV^2+C3*KELV^3+C4*KELV^4)*R0 : REM BTU/MOLE-R
1740 CV=CP-R0 : REM BTU/MOLE-R
1750 RETURN
1770 REM ***** COMBUSTION SUBROUTINE *****
1775 REM *****
1780 IF ANGLE<SPARKANGLE THEN FUEL=NC7H17 : RETURN
1790 IF ANGLE>SPARKANGLE+BURNANGLE THEN QCOMB=0:RETURN
1800 BURN=NC7H17-FUEL*(1+COS((ANGLE-SPARKANGLE)*PI/BURNANGLE))/2 : REM MOLES
1810 IF NC7H17<BURN THEN BURN=NC7H17
1820 IF NO2<11.25*BURN THEN BURN=NO2/11.25
1830 NC7H17=NC7H17-BURN
1840 NO2=NO2-11.25*BURN
1850 NCO2=NCO2+7*BURN
1860 NH2O=NH2O+8.5*BURN
1870 NSUM=NC7H17+NN2+NO2+NCO2+NH2O
1880 QCOMB=BURN*1932856 : REM BTU
1890 RETURN

1910 REM ***** HEAT TRANSFER SUBROUTINE *****
1915 REM *****
1920 GASFILMTEMP=(TEMP+WALLTEMP)/2
1930 GASENS=PRESS*MM/GASFILMTEMP/R0/12/778.16 : REM LBM/CU-IN
1940 GASVEL=ABS(AVGVEL)+50:REM IN/SEC (PISTON VEL+50)
1950 GASFACTOR=-4.36226E-11*GASFILMTEMP^2+2.89E-07*GASFILMTEMP+.0005783
1960 HGAS=GASFACTOR*GASENS*GASVEL*(1/12/BORE/GASENS/GASVEL)^.2
1970 GASAREA=AREA+4*VOLUME/BORE
1980 QWALLS=TIME*GASAREA*(HGAS*(TEMP-WALLTEMP)+3.305E-15*.05*TEMP^4): REM BTU
2000 COOLANTFILMTEMP=(WALLTEMP+COOLANTTEMP)/2
2010 COOLANTFACTOR=-9.3E-09*COOLANTFILMTEMP^2+2.0788E-05*COOLANTFILMTEMP-.0069
2020 HCOOLANT=COOLANTFACTOR*(COOLANTVEL)^.8*(1/12/HYDIAM)^.2
2030 QCOOLANT=HCOOLANT*TIME*COOLANTAREA*(WALLTEMP-COOLANTTEMP) : REM BTU

```

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2050 WALLTEMP=WALLTEMP+((QWALLS+SLIDE-QCOOLANT)/(METALCP*WALLMASS)
2055 RETURN

2062 REM ***** THERMODYNAMICS AND SHAFT OUTPUT SUBROUTINE *****
2061 REM *****
2064 INENTHALPY=RETFLOW*CP*(TEMP-ATMTEMP) : REM BTU
2065 EXENTHALPY=(EXFLOW+REVFLOW)*CP*(TEMP-ATMTEMP) : REM BTU
2066 REM DETERMINE AVERAGE PRESSURE OVER INCREMENT:
2067 W2=AREA*PRESS*(Y2-Y3)/12/778.16 : REM BTU
2068 E2=ENTHALPY+INENTHALPY-EXENTHALPY-QWALLS+QCOMB-W2 : REM BTU
2069 T2=E2/NSUM/CP+ATMTEMP : REM DEGREES R
2070 P2=NSUM*R0*12*778.16*T2/VOLUME : REM PSIA
2071 PRESS=(PRESS+P2)/2
2072 REM DETERMINE EXPANSION WORK AND RESULTING STATE:
2073 WORK=AREA*PRESS*(Y2-Y3)/12/778.16 : REM BTU
2074 ENTHALPY=ENTHALPY+INENTHALPY-EXENTHALPY-QWALLS+QCOMB-WORK : REM BTU
2075 TEMP=ENTHALPY/NSUM/CP+ATMTEMP : REM DEGREES R
2076 PRESS=NSUM*R0*12*778.16*TEMP/VOLUME : REM PSIA
2077 REM CALCULATE ENERGY DISSIPATION:
2078 ATMWORK=ATMPRESS*AREA*(Y2-Y3)/12/778.16 : REM BTU
2080 DELKE=PISTONMASS*AVGACC*(Y3-Y1)/2/GC/12/778.16 : REM BTU
2090 FORCE=AREA*(PRESS-ATMPRESS)+PISTONMASS*AVGACC/GC
2105 SLIDE=ABS(FRICTION*(2*(PRESS-ATMPRESS)+100)*(Y2-Y3)/12/778.16) : REM BTU
2110 SLIDE=SLIDE+ABS(FRICTION*FORCE*TAN(A2)*(Y2-Y3)/12/778.16) : REM BTU
2120 ROTATE=((VISCOSITY/4/GC/TEMP/12/778.16)*((AREA1*D1^2*(A1-A1OLD)^2/H1)+_
(AREA2*D2^2*(A2-A2OLD)^2/H2)+(AREA3*D3^2*(A3-A3OLD)^2/H3)) : REM BTU
2130 SHAFT=WORK-ATMWORK-DELKE-SLIDE-ROTATE : REM BTU
2135 BHP=(SHAFT/2544.48)/(TIME/3600) : REM INSTANTANEOUS HORSEPOWER
2140 TORQUE=BHP*TIME*550 : REM FT-LB
2150 SUMBHP=SUMBHP+BHP : REM HORSEPOWER
2160 CYCLEHP=SUMBHP/(ANGLE/INCREMENT+1) : REM OUTPUT HORSEPOWER
2170 RETURN
2200 END

```

APPENDIX D.

PROGRAM FOR VARIABLE-STROKE ENGINE

```

10 REM ***** V-STROKE.BAS (VARIABLE-STROKE ENGINE MODEL) *****
15 REM *****
20 OPEN "C:VSMAN1.PRN" FOR OUTPUT AS #1
22 OPEN "C:VSMAN2.PRN" FOR OUTPUT AS #2
24 OPEN "C:VSMAN3.PRN" FOR OUTPUT AS #3
30 PI=3.141593:R0=1.98588:REM BTU/MOLE-R
40 DEF FNASIN(A)=ATN(A/SQR(1-A*A))
50 DEF FNACOS(A)=-ATN(A/SQR(-A*A+1))+PI/2
60 GC=12*32.1739 : REM IN-LBM/LBF-SEC^2

80 REM ***** VARIABLE STROKE ENGINE CHARACTERISTICS *****
81 REM *****
82 READ BORE, DISPL, COMPRESS, RPM, INCREMENT
84 DATA 3.735, 38.125, 8.6, 3500, 5
86 AREA=BORE^2*PI/4
88 TIME=INCREMENT/RPM*60/360: REM SECONDS
90 READ L1, L2, L3, L4, R1, R2
92 DATA 1.4, 10, 5.2, 2.8, 1.38, .6 : REM NON-DIMENSIONAL
94 READ XE, YE, XF, YF
96 DATA 0, 0, 3, 4.4 : REM NON-DIMENSIONAL
98 READ ALPHA0,DIRECTION
99 DATA -.005, -1
100 REM FIND EXTREME PISTON POSITIONS
110 YAMAX=0 : YAMAX2=0 : YAMIN=L1+L2+L3+L4+R1+R2 : YAMIN2=YAMIN
120 FOR ANGLE=0 TO 720-INCREMENT STEP INCREMENT
130 GOSUB 970 : REM PISTON POSITION
140 IF YA>YAMAX THEN YAMAX=YA : MAXAT=ANGLE
150 IF YA<YAMIN THEN YAMIN=YA : MINAT=ANGLE
160 NEXT ANGLE
170 FOR ANGLE=0 TO 720-INCREMENT STEP INCREMENT
180 GOSUB 970 : REM PISTON POSITION
190 IF ABS(MAXAT-ANGLE)<180 GOTO 210
195 IF ABS(720+MAXAT-ANGLE)<180 GOTO 210
200 IF YA>YAMAX2 THEN YAMAX2=YA : MAX2AT=ANGLE
210 IF ABS(MINAT-ANGLE)<180 GOTO 230
215 IF ABS(720+MINAT-ANGLE)<180 GOTO 230
220 IF YA<YAMIN2 THEN YAMIN2=YA : MIN2AT=ANGLE
230 NEXT ANGLE
250 SFAC=DISPL/((YAMAX2-YAMIN2)*AREA) : REM SCALE FACTOR
260 L1=L1*SFAC : L2=L2*SFAC : L3=L3*SFAC : L4=L4*SFAC : R1=R1*SFAC
270 R2=R2*SFAC : XE=XE*SFAC : YE=YE*SFAC : XF=XF*SFAC : YF=YF*SFAC
275 YAMAX=YAMAX*SFAC : YAMIN=YAMIN*SFAC
276 YAMAX2=YAMAX2*SFAC : YAMIN2=YAMIN2*SFAC
280 CLRVOL=DISPL/(COMPRESS-1)
290 IF (YAMAX2+CLRVOL/AREA)<YAMAX THEN PRINT "GEOMETRY ERROR" : STOP
296 IF MINAT<MAX2AT THEN PRINT "CHANGE MAX/MIN POINTS" : STOP
298 IF MAX2AT<MIN2AT THEN PRINT "CHANGE MIN/MAX POINTS" : STOP

```

```

360 READ PISTONMASS, FRICTION, VISCOSITY
370 DATA 3, .05, .00068
380 READ D1, D2, D3, D4, D5, D6, D7
390 DATA 2.5, 1.5, 1.0, 2.0, 2.0, 1.0, 1.0
400 READ AREA1, AREA2, AREA3, AREA4, AREA5, AREA6, AREA7
410 DATA 9.8, 3.5, 1.6, 6.3, 6.3, 1.6, 1.6
420 READ H1, H2, H3, H4, H5, H6, H7
430 DATA .00125, .00075, .00050, .00100, .00100, .00050, .00050
440 READ INMAX, INOPEN, INCLOSE, EXMAX, EXOPEN, EXCLOSE
450 DATA .12, 695, 275, .12, 440, 40
460 READ SPARKANGLE
470 DATA 335
475 BURNANGLE=37+RPM/120
480 READ COOLANTTEMP, COOLANTVEL, HYDIAM, WALLTHK, METALDENSITY, METALCP
485 DATA 650, 25, .5, .25, .28, .11
487 OUTDIAM=BORE+2*WALLTHK
490 WALLHT=YAMAX-YAMIN
492 COOLANTAREA=(OUTDIAM)^2*PI/4+(OUTDIAM)*PI*WALLHT
494 WALLMASS=(AREA*WALLTHK+(OUTDIAM^2-BORE^2)*PI/4*WALLHT)*METALDENSITY

500 REM ***** INLET CONDITIONS *****
505 REM *****
510 ATPRESS=14.696:ATMTEMP=537
520 READ NC7H17I, NN2I, NO2I, NCO2I, NH2OI
530 DATA 1, 42.32, 11.25, 0, 0 : REM THEORETICAL BALANCE
540 XC7H17I=NC7H17I/(NC7H17I+NN2I+NO2I)
550 XN2I=NN2I/(NC7H17I+NN2I+NO2I)
560 XO2I=NO2I/(NC7H17I+NN2I+NO2I)
570 READ MNC7H17, MNN2, MNO2, MNCO2, MNH2O : REM MOLECULAR WEIGHTS
580 DATA 101, 28, 32, 44, 18
590 MWI=XC7H17I*MNC7H17+XN2I*MNN2+XO2I*MNO2
595 IDEALC7H17=XC7H17I*ATMPRESS*(DISPL+CLRVL)/RO/ATMTEMP/12/778.16

610 REM ***** STARTING CONDITIONS *****
615 REM *****
620 NCO2=0 : NH2O=0 : XC7H17=XC7H17I : XN2=XN2I : XO2=XO2I
630 MW=XC7H17I*MNC7H17+XN2I*MNN2+XO2I*MNO2 : R=RO/MW
640 PRESS=ATMPRESS : TEMP=ATMTEMP
650 ANGLE=0 : GOSUB 970
660 Y3=YA : Y2=YA : Y1=YA
670 NSUM=PRESS*VOLUME/RO/TEMP/12/778.16
680 NC7H17=XC7H17I*NSUM : NN2=XN2I*NSUM : NO2=XO2I*NSUM
690 GOSUB 1880 : REM SPECIFIC HEATS
700 ICV=CV:ICP=CP:IK=ICP/ICV:ICRITICAL=(2/(IK+1))^(IK/(IK-1))
710 ENTHALPY=0
720 WALLTEMP=660

```

```

740 REM ***** MAIN PROGRAM *****
745 REM *****
750 FOR CYCLE=1 TO 3
760     PRINT "CYCLE=";CYCLE
762     CYCLEHP=0:SUMBHP=0:SUMQCOMB=0:SUMQWALLS=0:SUMSLIDE=0
764     SUMROTATE=0:SUMSHAFT=0:SUMINENTHALPY=0:SUMEXENTHALPY=0:SUMDELKE=0
780     FOR ANGLE=0 TO 720-INCREMENT STEP INCREMENT
790         GOSUB 970 : REM PISTON POSITION
800         GOSUB 1310 : REM INTAKE
810         GOSUB 1600 : REM EXHAUST
820         GOSUB 1760 : REM MASS FLOW
830         GOSUB 1880 : REM SPECIFIC HEATS
840         GOSUB 2060 : REM COMBUSTION
850         GOSUB 2200 : REM HEAT TRANSFER
860         GOSUB 2370 : REM THERMODYNAMICS AND SHAFT OUTPUT
862         SUMQCOMB=SUMQCOMB+QCOMB : SUMQWALLS=SUMQWALLS+QWALLS
864         SUMSLIDE=SUMSLIDE+SLIDE : SUMROTATE=SUMROTATE+ROTATE
866         SUMSHAFT=SUMSHAFT+SHAFT : SUMINENTHALPY=SUMINENTHALPY+_
            INENTHALPY
868         SUMDELKE=SUMDELKE+DELKE : SUMEXENTHALPY=SUMEXENTHALPY+_
            EXENTHALPY
870         IF CYCLE<3 GOTO 920
880         REM OUTLIST ANGLE AND CALCULATED VALUES:
900         PRINT#1,USING "#####.##";ANGLE,
901         PRINT#1,USING "  +#.###^";VOLUME,PRESS,TEMP,WORK,ATMWORK
905         PRINT#2,USING "#####.##";ANGLE,
910         PRINT#2,USING "  +#.###^";NSUM,QCOMB,QWALLS,INENTHALPY,_
            EXENTHALPY,ENTHALPY
912         PRINT#3,USING "#####.##";ANGLE,
915         PRINT#3,USING "  +#.###^";SLIDE,ROTATE,SHAFT,AVGVEL,_
            AVGACC,DELKE
920     NEXT ANGLE
922     P$="###.###"
924     PRINT " V8 MOTORHP=",:PRINT USING P$;CYCLEHP*8
926     PRINT "   VOL.EFF.=", :PRINT USING P$;MAXC7H17/IDEALC7H17*100
928     PRINT "     QCOMB=", :PRINT USING P$;SUMQCOMB
930     PRINT "     QWALLS=", :PRINT USING P$;SUMQWALLS
932     PRINT "     SLIDE=", :PRINT USING P$;SUMSLIDE
934     PRINT "     ROTATE=", :PRINT USING P$;SUMROTATE
936     PRINT "     SHAFT=", :PRINT USING P$;SUMSHAFT
938     PRINT "     DELKE=", :PRINT USING P$;SUMDELKE
940     PRINT " INENTHALPY=", :PRINT USING P$;SUMINENTHALPY
942     PRINT " EXENTHALPY=", :PRINT USING P$;SUMEXENTHALPY
946     PRINT "   TOTAL IN=", :PRINT USING P$;SUMQCOMB+SUMINENTHALPY
948     PRINT "   TOTAL OUT=", :PRINT USING P$;SUMQWALLS+SUMSLIDE+SUMROTATE+_
        SUMSHAFT+SUMEXENTHALPY
950 NEXT CYCLE
952 CLOSE :PRINT "NORMAL TERMINATION":STOP

```

```

970 REM ***** PISTON POSITION SUBROUTINE *****
975 REM *****
980 A1=( 90+ANGLE )*PI/180
990 XD=XE-R1*COS( A1 )
1000 YD=YE+R1*SIN( A1 )
1010 ALPHA=A1/2*DIRECTION+ALPHA0
1020 XG=XF-R2*COS( ALPHA )
1030 YG=YF+R2*SIN( ALPHA )
1040 S=SQR((XG-XD)^2+(YG-YD)^2)
1050 M=(S*S+L4*L4-L3*L3)/2/L4
1060 H=SQR(S*S-M*M)
1070 IF H/S=1 THEN CONST1=PI/2:GOTO 1090
1080 CONST1=FNASIN(H/S):IF M<0 THEN CONST1=PI-CONST1
1090 CONST2=FNASIN((XG-XD)/S)
1100 XC=XD+L4*SIN(CONST1+CONST2)
1110 YC=YD+L4*COS(CONST1+CONST2)
1120 XB=XD-(XC-XD)*(L1/L4)
1130 YB=YD-(YC-YD)*(L1/L4)
1140 YA=YB+SQR(L2*L2-(XA-XB)^2)
1150 VOLUME=CLRVOL+(YAMAX2-YA)*AREA
1160 Y1=Y2 : Y2=Y3 : Y3=YA
1170 AVGVEL=(Y3-Y2)/TIME
1180 AVGACC=(Y3-2*Y2+Y1)/TIME^2
1190 A2OLD=A2 : A2=FNASIN(XB/L2)
1200 BETA=FNASIN((YB-YC)/(L1+L4)) : GAMMA=FNASIN((XG-XC)/L3)
1210 A3OLD=A3 : A3=PI/2-BETA+GAMMA
1220 EC=SQR((XC-XE)^2+(YC-YE)^2)
1230 A4OLD=A4
1240 IF ABS(EC-R1-L4)<.00001 THEN A4=PI : GOTO 1270
1250 IF ABS(EC+R1-L4)<.00001 THEN A4=0 : GOTO 1270
1260 A4=FNACOS((L4*L4+R1*R1-EC*EC)/2/R1/L4)
1270 A5OLD=A5 : A5=PI/2+A2+BETA
1280 A7OLD=A7 : A7=PI/2-ALPHA+GAMMA : IF ANGLE=0 THEN A7OLD=A7
1290 RETURN

1310 REM ***** INTAKE SUBROUTINE *****
1315 REM *****
1320 INFLOW=0:REVFLOW=0:RETFLOW=0:INCONST=0
1330 IF INCLOSE<INOPEN GOTO 1370
1340 IF ANGLE<=INOPEN THEN RETURN
1350 IF ANGLE>=INCLOSE THEN RETURN
1360 INCONST=INMAX/2*(SIN((2*(ANGLE-INOPEN)/(INCLOSE-INOPEN)-.5)*PI)+1)
1365 GOTO 1400
1370 IF ANGLE<=INCLOSE THEN INCONST=INMAX/2*(SIN((2*(ANGLE-INOPEN+720)/_
(INCLOSE-INOPEN+720)-.5)*PI)+1):GOTO 1400
1380 IF ANGLE<=INOPEN THEN RETURN
1390 INCONST=INMAX/2*(SIN((2*(ANGLE-INOPEN)/(INCLOSE+720-INOPEN)-.5)*PI)+1)
1400 IF ATPRESS<PRESS THEN GOTO 1550 : REM REVERSE FLOW THROUGH INTAKE VALVE
1402 DELPRESS=ATMPRESS-PRESS

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1404 IF ATMPRESS*ICRITICAL>PRESS THEN DELPRESS=ATMPRESS*(1-ICRITICAL)
1410 INFLOW=INCONST*TIME*SQR(DELPRESS*ATMPRESS/ATMTEMP/MMI):REM MOLES
1420 P2=(NSUM+INFLOW)*R0*12*778.16*TEMP/VOLUME
1430 IF ATMPRESS<(P2+PRESS)/2 THEN INFLOW=0 : GOTO 1450 : REM AVOID INSTABILITY
1435 DELPRESS=ATMPRESS-(P2+PRESS)/2
1436 IF ATMPRESS*ICRITICAL>(P2+PRESS)/2 THEN DELPRESS=ATMPRESS*(1-ICRITICAL)
1440 INFLOW=INCONST*TIME*SQR(DELPRESS*ATMPRESS/ATMTEMP/MMI)
1450 IF NSUMR<=0 THEN RETURN
1460 RETFLOW=INCONST*TIME*SQR((ATMPRESS-PRESS)*ATMPRESS/TEMP/MM)
1470 P2=(NSUM+RETFLOW)*R0*12*778.16*TEMP/VOLUME
1480 IF (P2+PRESS)/2 > ATMPRESS THEN RETFLOW=0 : RETURN : REM AVOID INSTABILITY
1490 RETFLOW=INCONST*TIME*SQR((ATMPRESS-(P2+PRESS)/2)*ATMPRESS/TEMP/MM)
1500 IF RETFLOW>NSUMR THEN RETFLOW=0:REM NSUMR
1510 NSUMR=NSUMR-RETFLOW
1520 INFLOW=INFLOW-RETFLOW : REM BACKFLOW IS REINGESTED BEFORE FRESH CHARGE
1530 IF INFLOW < 0 THEN INFLOW=0
1540 RETURN
1550 REVFLOW=INCONST*TIME*SQR((PRESS-ATMPRESS)*PRESS/TEMP/MM)
1560 P2=(NSUM-RETFLOW)*R0*12*778.16*TEMP/VOLUME
1570 IF (P2+PRESS)/2 < ATMPRESS THEN REVFLOW=0 : RETURN : REM AVOID INSTABILITY
1580 REVFLOW=INCONST*TIME*SQR(((P2+PRESS)/2-ATMPRESS)*(P2+PRESS)/2/TEMP/MM)
1590 NSUMR=NSUMR+REVFLOW
1595 RETURN

1600 REM ***** EXHAUST SUBROUTINE *****
1605 REM *****
1610 EXFLOW=0:EXCONST=0
1620 IF EXCLOSE<EXOPEN GOTO 1660
1630 IF ANGLE<=EXOPEN THEN RETURN
1640 IF ANGLE>=EXCLOSE THEN RETURN
1650 EXCONST=EXMAX/2*(SIN((2*(ANGLE-EXOPEN)/(EXCLOSE-EXOPEN)-.5)*PI)+1)
1655 GOTO 1690
1660 IF ANGLE<=EXCLOSE THEN EXCONST=EXMAX/2*(SIN((2*(ANGLE-EXOPEN+720)/_
    (EXCLOSE-EXOPEN+720)-.5)*PI)+1):GOTO 1690
1670 IF ANGLE<=EXOPEN THEN RETURN
1680 EXCONST=EXMAX/2*(SIN((2*(ANGLE-EXOPEN)/(EXCLOSE+720-EXOPEN)-.5)*PI)+1)
1690 IF PRESS<ATMPRESS THEN RETURN
1692 K=CP/CV : CRITICAL=(2/(K+1))^(K/(K-1)) : DELPRESS=PRESS-ATMPRESS
1696 IF PRESS*CRITICAL>ATMPRESS THEN DELPRESS=PRESS*(1-CRITICAL) : REM CHOKED
1700 EXFLOW=EXCONST*TIME*SQR(DELPRESS*PRESS/TEMP/MM):REM MOLES
1710 P2=(NSUM-EXFLOW)*R0*12*778.16*TEMP/VOLUME
1720 IF (P2+PRESS)/2<ATMPRESS THEN EXFLOW=0:RETURN : REM AVOID INSTABILITY
1725 DELPRESS=(P2+PRESS)/2-ATMPRESS
1726 IF (P2+PRESS)/2*CRITICAL>ATMPRESS THEN DELPRESS=(P2+PRESS)/2*(1-CRITICAL)
1730 EXFLOW=EXCONST*TIME*SQR(DELPRESS*(P2+PRESS)/2/TEMP/MM)
1740 RETURN

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1760 REM ***** MASS FLOW SUBROUTINE *****
1765 REM *****
1770 NC7H17=NC7H17+XC7H17I*INFLOW-XC7H17*(EXFLOW+REVFLOW-RETFLOW)
1780 NN2=NN2+XN2I*INFLOW-XN2*(EXFLOW+REVFLOW-RETFLOW)
1790 NO2=NO2+XO2I*INFLOW-XO2*(EXFLOW+REVFLOW-RETFLOW)
1800 NCO2=NCO2+XCO2I*INFLOW-XCO2*(EXFLOW+REVFLOW-RETFLOW)
1810 NH2O=NH2O+XH2OI*INFLOW-XH2O*(EXFLOW+REVFLOW-RETFLOW)
1820 NSUM=NC7H17+NN2+NO2+NCO2+NH2O : REM MOLES
1830 XC7H17=NC7H17/NSUM:XN2=NN2/NSUM:XO2=NO2/NSUM:XCO2=NCO2/NSUM:XH2O=NH2O/NSUM
1840 MM=XC7H17*MMNC7H17+XN2*MMN2+XO2*MMNO2+XCO2*MMNCO2+XH2O*MMNH2O
1850 R=R0/MM : IF ANGLE=INCLOSE THEN MAXC7H17=NC7H17
1860 RETURN

1880 REM ***** SPECIFIC HEAT SUBROUTINE *****
1885 REM *****
1890 IF TEMP>1800 GOTO 1960
1900 C0=4.06*XC7H17+2.4008*XCO2+4.0701*XH2O+3.6748*XN2+3.6256*XO2
1910 C1=.061*XC7H17+8.740001E-03*XCO2-.00111*XH2O-.00121*XN2-.00188*XO2
1920 C2=-1.88E-06*XC7H17-6.61E-06*XCO2+4.15E-06*XH2O+2.32E-06*XN2+7.06E-06*XO2
1930 C3=2E-09*XCO2-2.96E-09*XH2O-6.32E-10*XN2-6.76E-09*XO2
1940 C4=6.33E-16*XCO2+8.07E-13*XH2O-2.26E-13*XN2+2.16E-12*XO2
1950 GOTO 2010
1960 C0=4.06*XC7H17+4.46*XCO2+2.716*XH2O+2.896*XN2+3.62*XO2
1970 C1=.061*XC7H17+.0031*XCO2+.002945*XH2O+.001515*XN2+.000736*XO2
1980 C2=-1.88E-06*XC7H17-1.24E-06*XCO2-8.02E-07*XH2O-5.72E-07*XN2-1.965E-07*XO2
1990 C3=2.27E-10*XCO2+1.02E-10*XH2O+9.98E-11*XN2+3.62E-11*XO2
2000 C4=-1.55E-14*XCO2-4.847E-15*XH2O-6.52E-15*XN2-2.89E-15*XO2
2010 KELV=TEMP*5/9
2020 CP=(C0+C1*KELV+C2*KELV^2+C3*KELV^3+C4*KELV^4)*R0 : REM BTU/MOLE-R
2030 CV=CP-R0 : REM BTU/MOLE-R
2040 RETURN

2060 REM ***** COMBUSTION SUBROUTINE *****
2065 REM *****
2070 IF ANGLE<SPARKANGLE THEN FUEL=NC7H17 : RETURN
2080 IF ANGLE>SPARKANGLE+BURNANGLE THEN QCOMB=0:RETURN
2090 BURN=NC7H17-FUEL*(1+COS((ANGLE-SPARKANGLE)*PI/BURNANGLE))/2 : REM MOLES
2100 IF NC7H17<BURN THEN BURN=NC7H17
2110 IF NO2<11.25*BURN THEN BURN=NO2/11.25
2115 IF BURN<1E-10 THEN RETURN
2120 NC7H17=NC7H17-BURN
2130 NO2=NO2-11.25*BURN
2140 NCO2=NCO2+7*BURN
2150 NH2O=NH2O+8.5*BURN
2160 NSUM=NC7H17+NN2+NO2+NCO2+NH2O
2170 QCOMB=BURN*1932856 : REM BTU
2180 RETURN

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2200 REM ***** HEAT TRANSFER SUBROUTINE *****
2205 REM *****
2210 GASFILMTEMP=(TEMP+WALLTEMP)/2
2220 GASDENS=PRESS*MM/GASFILMTEMP/R0/12/778.16 : REM LBM/CU-IN
2230 GASVEL=ABS(AVGVEL)+50:REM IN/SEC (PISTON VEL+50)
2240 GASFACTOR=-4.36226E-11*GASFILMTEMP^2+2.89E-07*GASFILMTEMP+.0005783
2250 HGAS=GASFACTOR*GASDENS*GASVEL*(1/12/BORE/GASDENS/GASVEL)^.2
2260 GASAREA=AREA+4*VOLUME/BORE
2270 QWALLS=TIME*GASAREA*(HGAS*(TEMP-WALLTEMP)+3.305E-15*.05*TEMP^4): REM BTU
2290 COOLANTFILMTEMP=(WALLTEMP+COOLANTTEMP)/2
2300 COOLANTFACTOR=-9.3E-09*COOLANTFILMTEMP^2+2.0788E-05*COOLANTFILMTEMP-.0069
2310 HCOOLANT=COOLANTFACTOR*(COOLANTVEL)^.8*(1/12/HYDIAM)^.2
2320 QCOOLANT=HCOOLANT*TIME*COOLANTAREA*(WALLTEMP-COOLANTTEMP) : REM BTU
2340 WALLTEMP=WALLTEMP+(QWALLS+SLIDE-QCOOLANT)/(METALCP*WALLMASS)
2350 RETURN
2370 REM ***** THERMODYNAMICS AND SHAFT OUTPUT SUBROUTINE *****
2380 REM *****
2390 INENTHALPY=RETFLOW*CP*(TEMP-ATMTEMP) : REM BTU
2400 EXENTHALPY=(EXFLOW+REVFLOW)*CP*(TEMP-ATMTEMP) : REM BTU
2410 REM DETERMINE AVERAGE PRESSURE OVER INCREMENT:
2420 W2=AREA*PRESS*(Y2-Y3)/12/778.16 : REM BTU
2430 E2=ENTHALPY+INENTHALPY-EXENTHALPY-QWALLS+QCOMB-W2 : REM BTU
2440 T2=E2/NSUM/CP+ATMTEMP : REM DEGREES R
2450 P2=NSUM*R0*12*778.16*T2/VOLUME : REM PSIA
2460 PRESS=(PRESS+P2)/2
2470 REM DETERMINE EXPANSION WORK AND RESULTING STATE:
2480 WORK=AREA*PRESS*(Y2-Y3)/12/778.16 : REM BTU
2490 ENTHALPY=ENTHALPY+INENTHALPY-EXENTHALPY-QWALLS+QCOMB-WORK : REM BTU
2500 TEMP=ENTHALPY/NSUM/CP+ATMTEMP : REM DEGREES R
2510 PRESS=NSUM*R0*12*778.16*TEMP/VOLUME : REM PSIA
2520 REM CALCULATE ENERGY DISSIPATION:
2530 ATMWORK=ATMPRESS*AREA*(Y2-Y3)/12/778.16 : REM BTU
2540 DELKE=PISTONMASS*AVGACC*(Y3-Y1)/2/GC/12/778.16 : REM BTU
2550 FORCE=AREA*(PRESS-ATMPRESS)+PISTONMASS*AVGACC/GC
2560 SLIDE=ABS(FRICTION*(2*(PRESS-ATMPRESS)+100)*(Y2-Y3)/12/778.16) : REM BTU
2570 SLIDE=SLIDE+ABS(FRICTION*FORCE*(Y2-Y3)*TAN(A2)/12/778.16) : REM BTU
2580 V1=(AREA1*D1^2*(INCREMENT*PI/180)^2/H1)+(AREA2*D2^2*(A2-A2OLD)^2/H2)+_
(AREA3*D3^2*(A3-A3OLD)^2/H3)
2590 V2=(AREA4*D4^2*(A4-A4OLD)^2/H4)+(AREA5*D5^2*(A5-A5OLD)^2/H5)+_
(AREA6*D6^2*(INCREMENT*PI/360)^2/H6)
2600 V3=(AREA7*D7^2*(A7-A7OLD)^2/H7)
2610 ROTATE=(VISCOSITY/4/GC/TIME/12/778.16)*(V1+V2+V3) : REM BTU
2620 SHAFT=WORK-ATMWORK-DELKE-SLIDE-ROTATE : REM BTU
2630 BHP=(SHAFT/2544.48)/(TIME/3600) : REM INSTANTANEOUS HORSEPOWER
2640 TORQUE=BHP*TIME*550 : REM FT-LB
2650 SUMBHP=SUMBHP+BHP : REM HORSEPOWER
2660 CYCLEHP=SUMBHP/(ANGLE/INCREMENT+1) : REM OUTPUT HORSEPOWER
2670 RETURN
2680 END

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VITA

Burt Randall Jennings was born in Alamogordo, New Mexico on May 30, 1959. He attended various elementary schools around the country and in West Germany. He graduated from Hephzibah High School in Hephzibah, Georgia in June 1977. The following September he entered the Georgia Institute of Technology, and in September 1981 he received his Bachelor's degree in Mechanical Engineering.

He was employed by Sverdrup Technology Incorporated at the Arnold Engineering Development Center, Tennessee in September 1981 and began study towards a Master's degree in the following year at the University of Tennessee Space Institute.

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