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Modeling and Evaluation of Combustion Process of a Three-valve Stratified Charge Engine

By

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Abstract: This papaer describes the development of a mathematical model for the compression, combustion and expansion process of a stratified charge engine, consisting of an auxilary combustion chamber with an inlet valve and a main combustion chamber with both the inlet and exhaust valves.
After calculating the mixture formation at the end of compression process, a simple combustion model was developed to compute the gas temperature, gas pressure and the rate of formation of NO and CO at each crank angle, using the basic energy equation and reaction kinetics for both the auxilary and main chambers. The above calculations were also extended for the expansion process.
The evaluation of the model was carried out by comparing the computed and experimental data. A satisfactory correlation was observed between them.

INTRODUCTION

The subject of reducing emissions from automobiles has received considerable attention from several researchers. This is primarily due to the stringent standards imposed on the emission of nitric oxide, carbon monoxide and unburnt hydrocarbons. To meet these standars, the three-valve stratified charge engine was developed [*1*].

This engine has both the combustion chambers, auxiliary and main, in order to keep the stratified charge geometrically, for the purpose of achieving good fuel consumption and clearing the emission regulations, so that this design has three valves, with one intake valve eath for the auxiliary and main chambers, and an exhaust valve fitted in the main chamber.

During the intake process, a rich mixture is introduced through the small intake valve by an auxiliary carburettor, and a lean mixture through an intake valve by a main carburettor. In this process, the rich mixture from the auxiliary chamber flows into the main chamber through the torch opening and mixes with the lean mixture in the main chamber. The reverse flow of lean mixture from the main chamber during the compression process causes a considerable reduction in the richness of the mixture present in the auxiliary chamber and also generates turbulence. Such a rich mixture (nearing stoichiometry) assures ignition and the turbulence accelerates the process of combustion in auxiliary chamber. As this process continues, a jet of burning gases

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flow through the torch opening into the main chamber. This jet ignites the lean mixture and enhances the combustion in main chamber. Experimental results have shown that this type of engine successfully meets the present emission standards [*1 and 2*].

Some experiments were conducted with a three valve carbureted prechamber stratified engine [*3 and 4*]. However, from the results the exact relationship is not determined between the engine performance and the geometrical or the operating factors. In an earlier paper [*1*], a mixture formation model was proposed to explain these relations, but considerable difficulty was experienced in applying that model.

Over the past few years, the models of thermodynamic processes in an engine have been found to be a useful tool in the analysis and design of engines. Several simulation models have been developed for a three-valve stratified charge engine [*5 and 6*], but their conclusions were not compared with the experimental data. In order to help the understanding and interpreting of experimental data obtained from a three-valve stratified charge engine, a detailed simulation model on a crankangle by crankangle basis has been presented [*7 and 8*]. In these papers, the calculated results were compared with the experimental values obtained under only a certain engine operating condition.

The primary purpose of our work is to develop a combustion model for predicting the emission of NO and CO, as well as the pressure- and temperature-time diagrams for both the auxiliary and main chambers. With this idea, a mathematical model was developed for the compression, combustion and expansion processes of a three-valve stratified charge engine. These predicted diagrams are compared with the measured data to verify the proposed model, and furthermore the estimated exhaust emissions (NO and CO) are examined by comparing with the measured values in order to identify the effect of torch opening area on the emissions.

Model Assumptions

The present model, which starts from the inlet valve closing, predicts the fuel air ratio in the main and auxiliary chambers during the compression process. The combustion is assumed to be initiated by spark ignition in the auxiliary chamber. The onset of combustion occurs the in main chamber, when the flame reaches the torch opening of auxiliary chamber. The expansion process starts at the end of combustion process and is completed at the opening of exhaust valve.

The following assumptions are made in developing the model.

- i The unburnt mixture at any instant consists of fuel air mixture and residual gas.
- ii The unburnt and burnt gases are assumed to be an ideal one.
- iii Both the auxiliary and main chambers are treated as two different thermodynamic systems, each having its own gas composition.
- iv Each chamber is assumed to be homogeneous before and after combustion.
- v The residual gas fraction is assumed for each chamber at the beginning of the compression process.
- vi During the combustion process an unburnt and a burnt zone are assumed to be present in both the chambers.

vii Instantaneous mixing is assumed for each zone. Hence each zone is under uniform temperature and composition.
viii Gas pressure is assumed uniform in each chamber during compression, combustion and expansion processes.
ix Quasi-steady and isentropic flow is assumed across the torch opening.
x Heat transfer in each chamber is estimated by using Woschni's relation [*9*] and assuming the wall temperature of each surface as constant.
xi The burning rate is claculated by assuming the duration of combustion with a constant flame velocity, details of which will be described later.
xii Nitric oxide emissions are predicted by extended Zeldovich kinetic scheme [*10*].

Fundamental Relations of the Model

Compression Process: Both in the auxiliary and main chambers, the constituents consist of air fuel mixture and residual gas, which is calculated by using a measured scavenging efficiency (η_s) for the auxiliary chamber.

The auxiliary and main chambers are treated as open systems. For an open system, the basic energy equation can be written as

$$\dot{U}=-P\cdot\dot{V}+\sum\dot{Q}_i+h\cdot\dot{M}_t \tag{1}$$

The first term on the right hand side can be obtained from the cylinder volume (V) as a function of crankangle. The second term can be estimated from the product of heat transfer coefficient (H), area of the concerned surface (A) and the temperature difference between the gas and wall. H is estimated from Woschni's relation [*9*].

$$H=110\,(P\cdot W)^{0.8}/(d^{0.2}T^{0.53}) \tag{2}$$

The heat transfer equation becomes

$$\dot{Q}=H\cdot A\cdot(T-T_w) \tag{3}$$

The surface area of the auxiliary chamber does not vary with respect to time. The main chamber is divided into 3 surfaces. They are cylinder head, piston and sleeve. The wall temperature is assumed constant throughout the calculation.

The third term on the right hand side of equation (1), deals with the enthalpy transfer across the system. This can be obtained by computing the mass flow through the torch opening.

$$\dot{M}_t=A_{ef}P_1\cdot\sqrt{\frac{\phi_1 g}{R_1\cdot T_1}} \tag{4}$$

where

$$\phi_1=\frac{2K}{K-1}\left[\left(\frac{P_2}{P_1}\right)^{2/k}-\left(\frac{P_2}{P_1}\right)^{(K+1)/K}\right] \tag{5}$$

The effective opening area A_{ef} includes a constant discharge coefficient of the torch opening, whose value is 0.6. The ratio of specific heats K is assumed constant for air and fuel. It is computed based on the mass fraction for the mixture.

The term on the left hand side of expression (1) indicates the rate of change of internal energy ($\dot{U}$) with respect to time. The internal energy can be expressed as a function of temperature, pressure and equivalence ratio. Since the equivalence ratio can be computed based on the mass flow from the main chamber to the auxiliary chamber, expression (1) can be solved to obtain the temperature at any instant. The procedure for solving the basic energy equation is described in reference [*11*].

COMBUSTION PROCESS: The combustion process is analysed by developing a simple combustion model.

Simple Combustion Model: The combustion process begins in the auxiliary chamber when the spark plug is fired. In the present analysis the combustion process is assumed to be present in 4 phases as shown in Fig. 1.

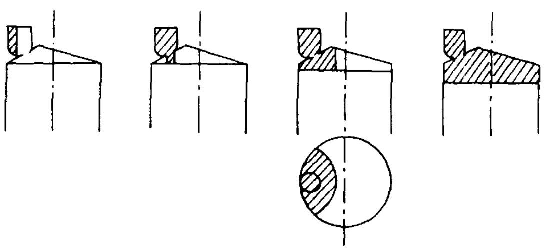

FIG. 1. Four Phases of Combustion

In the first phase, the flame is assumed to propagate in the auxiliary chamber in a cylindrical fashion and only unburnt mixture flows through the torch opening from the auxiliary chamber to the main chamber, until the flame reaches the torch opening. In this period, the auxiliary chamber has two zones comprised of the unburnt and burnt. The second phase shows the completion of combustion in the auxiliary chamber. Near the end of combustion, the burning gases (torch flame) enter the main chamber through the torch opening and ignite the lean mixture in it. A cylindrical flame front is also assumed to propagate on either side of the torch opening in the main chamber, parallel to the axis of the cylinder. Phase 3 shows the completion of combustion in the smaller space of the main chamber while it continues in the other larger space.

The expansion process is assumed to start from phase 4, and continues until the exhaust valve opens. In phases 3 and 4, only burnt gas flows through the torch opening.

Mathematical Formulation: During the combustion process there are two zones present in the auxiliary and main chambers. Hence the basic energy equation for the burnt zone in each chamber can be written as

$$\dot{\overline{u_b \cdot M_b}} = -P \cdot \dot{V}_b + \sum \dot{Q}_{bi} + h_c \cdot \dot{M}_c + h_t \cdot \dot{M}_t$$

Since the internal energy (u_b) can be expressed as a function of temperature, pressure and equivalence ratio, the above expression becomes

$$u_b \cdot \dot{M}_b + M_b \left[\frac{\partial u_b}{\partial T_b} \cdot \frac{dT_b}{dt} + \frac{\partial u_b}{\partial P} \cdot \frac{dP}{dt} + \frac{\partial u_b}{\partial F} \cdot \frac{dF}{dt} \right]$$
$$= -P \cdot \dot{V}_b + \sum \dot{Q}_{bi} + h_c \cdot \dot{M}_c + h_t \cdot \dot{M}_t \qquad (6)$$

The characteristic gas equation can be logarithmically differentiated to obtain the next relation

$$\dot{T}_b = T_b[\dot{P}/P + \dot{V}_b/V_b - \dot{M}_b/M_b] \qquad (7)$$

Equations for the unburnt side (8) and (9) can be written as similar to equations (6) and (7).

The derivative of total mass in each chamber at any instant can be expressed as

$$\dot{M}=\dot{M}_b+\dot{M}_u \tag{10}$$

The derivative of total volume in each chamber can be written as

$$\dot{V}=\dot{V}_b+\dot{V}_u \tag{11}$$

These are the six major equations that should be solved in the above combustion model. The unknowns in these equations are $\dot{P}$, $\dot{T}_b$, $\dot{T}_u$, $\dot{M}_b$, $\dot{M}_u$, $\dot{V}_b$ and $\dot{V}_u$. These unknowns are calculated for both the auxiliary and main chambers. The above equations can be solved simultaneously only if $\dot{V}_b$ is estimated.

In the present analysis, a volume burning rate ($\dot{V}_b$) is estimated by using a following simpler burning model. At the onset of spark, a finite delay period is preset, in which there is no appreciable rise in pressure and temperature, because the flame propagates very slowly in the initial (0 to 10%) and final (90 to 100%) periods. Therefore the model consists of three straight lines, and in these periods, the rate of the flame propagation is assumed as one third of that between 10% and 90% of the flame travel as shown in Fig. 2. In this model, the durations of combustion in both the chambers

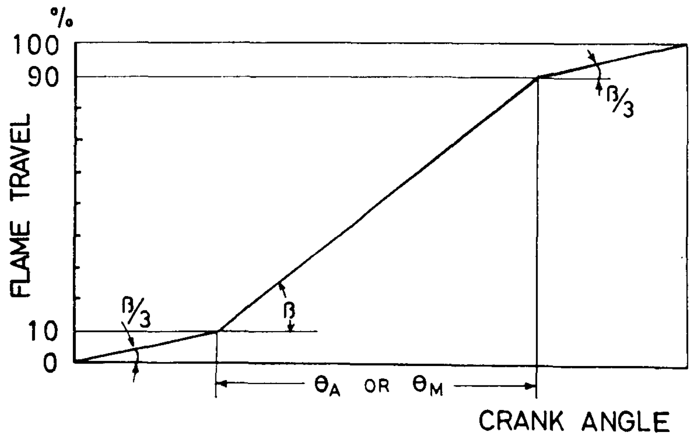

FIG. 2. Simple Burning Model

are the time between 10% and 90% of flame travel, and assumed as 15° crankangle duration for the auxiliary chamber and 30° for the main chamber respectively.

Reaction Kinetics: The extended Zeldovich mechanism is used for the nitric oxide relations. Basically the formation of rate equation is similar to Lavoie et al. [*10*], except an additional term is introduced to take into account for the flux of nitric oxide $d[NO]/dt|_t$, due to mass flow through the torch opening. The relations for extended Zeldovich mechanism are

$$N+NO \underset{}{\overset{K_1}{\rightleftarrows}} N_2+O; \qquad K_1=1.55\times10^7\times e^{-500/RT} \quad m^3/mole\text{-}s$$

$$O_2+N \overset{K_2}{\rightleftarrows} NO+O; \qquad K_2=1.33\times10^4\times e^{-7080/RT} \quad m^3/mole\text{-}s$$

$$N+OH \overset{K_3}{\rightleftarrows} NO+H; \qquad K_3=4.2\times10^7 \quad m^3/mole\text{-}s$$

The rate equation can be written as

$$\frac{d[NO]}{dt}=\frac{2(1-\beta^2)}{V}\left[\frac{R_1}{1+\beta\cdot S}\right]+\frac{d[NO]}{dt}\bigg|_t \tag{12}$$

where

$$\beta=[NO]/[NO]_e;\ S=R_1/(R_2+R_3)$$
$$R_1=K_1[N]_e[NO]_e;\ R_2=K_2[O_2]_e[N]_e;\ R_3=K_3[N]_e[OH]_e$$

Since the temperature, pressure and mass are known, equation (12) can be solved to obtain the nitric oxide present in the auxiliary and main chambers.

The CO is estimated from the relation given by Westenberg [12] by adding another term to take into account the flux of CO flowing through the torch $d[CO]/dt|_t$.

$$-\frac{d[CO]}{dt}=K_1[OH][CO]+\frac{d[CO]}{dt}\bigg|_t \tag{13}$$

where

$$K_1=3\times 10^5\times e^{-600/RT}\quad \mathrm{m^3/mole\text{-}s}$$

Based on the equilibrium composition of [OH], the amount of CO present in the auxiliary and main chambers is estimated at each crankangle by solving equation (13). When the flame reaches the other end, combustion is assumed to be over and expansion process is assumed to start.

Expansion Process: The analysis of this process is similar to the compression process, except the constituents in both the chambers contain only burnt gases. In addition to the temperature, pressure and mass flow rate through the torch, the rate of formation of NO and CO is also computed in the early part of expansion process. Since the temperature of the gas is fairly high, dissociation is taken into account. The expansion process ends when the exhaust valve opens.

Computer Program and Input Data

To start with, the input data such as engine specifications which include bore, stroke, connecting rod length and auxiliary chamber dimensions as shown in Table 1 are read. The initial pressure and temperature in both the chambers are calculated at inlet valve closing point by assuming the residual gas fraction in the auxiliary and main chambers as 18% and 10% respectively.

The computations are started at inlet valve closing point by calculating the mass flow through the torch with an increment of 1° crankangle. In the compression period, the basic energy equation is solved to obtain temperature and later pressure with the characteristic gas equation in both the chambers. In addition to it the equivalence ratio is estimated in the auxiliary chamber. When the combustion starts, the burning rate is estimated and the six equations are solved to obtain the pressure and temperature of unburnt and burnt gas in both the chambers. Using reaction kinetics, the concentration of NO and CO is also determined.

During the expansion process, the basic energy equation is solved to determine the pressure and temperature of gas in both the chambers as well as NO and CO. At

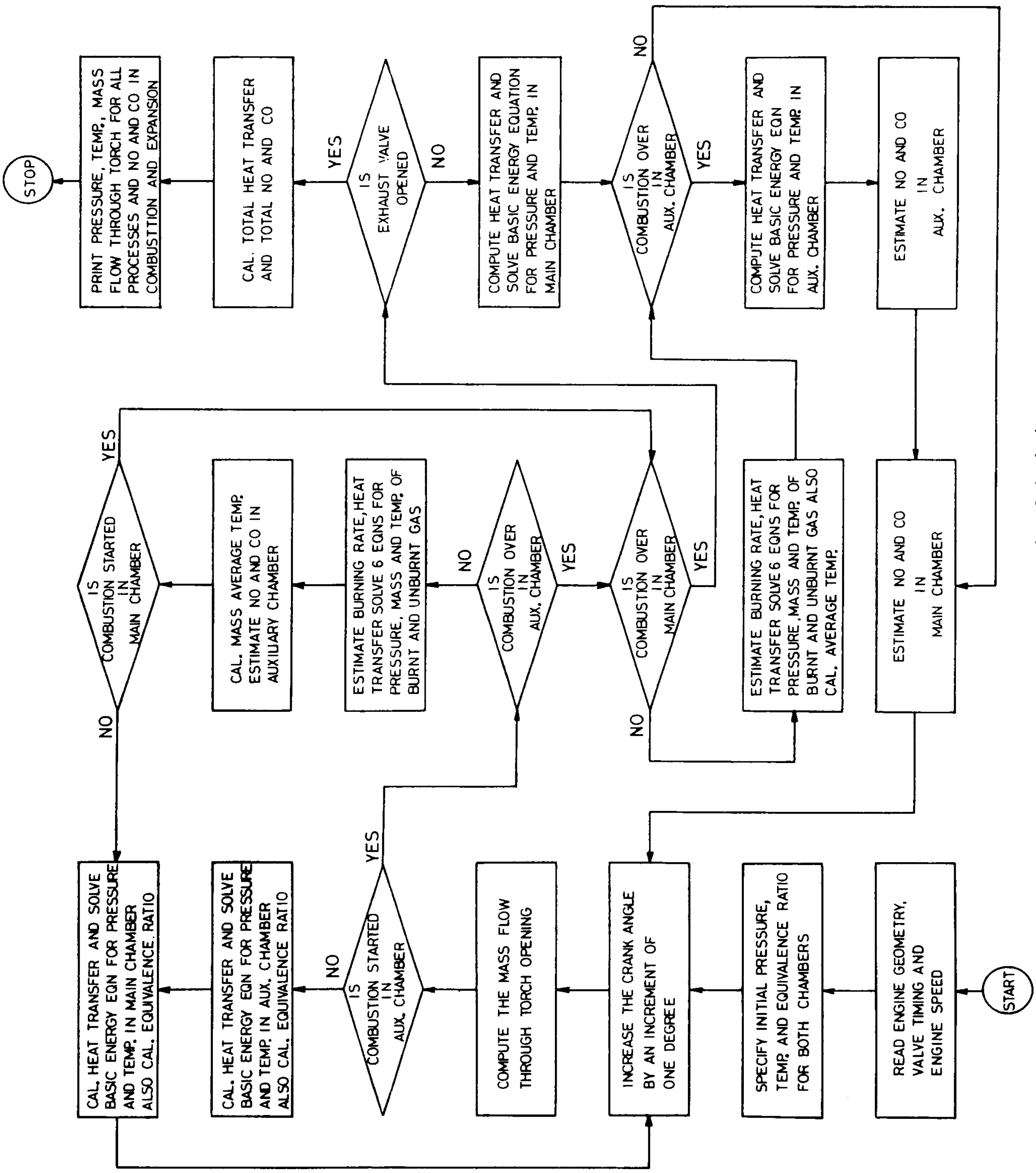

FIG. 3. Flow Chart for Calculation

the time of exhaust valve opening, total emissions of NO and CO are estimated. The flow chart for calculating the thermodynamic cycle is shown in Fig. 3.

EXPERIMENTS

Test Engine: The basic construction of the three valve stratified charge engine used in the present analysis is shown in Fig. 4, and the specifications are given in Table 1. As shown in the figure the combustion chamber is divided into main and auxiliary ones. The later chamber is formed by a hot cup with a torch opening and is fitted with a small auxiliary intake valve. The test engine is controlled mainly by the output from

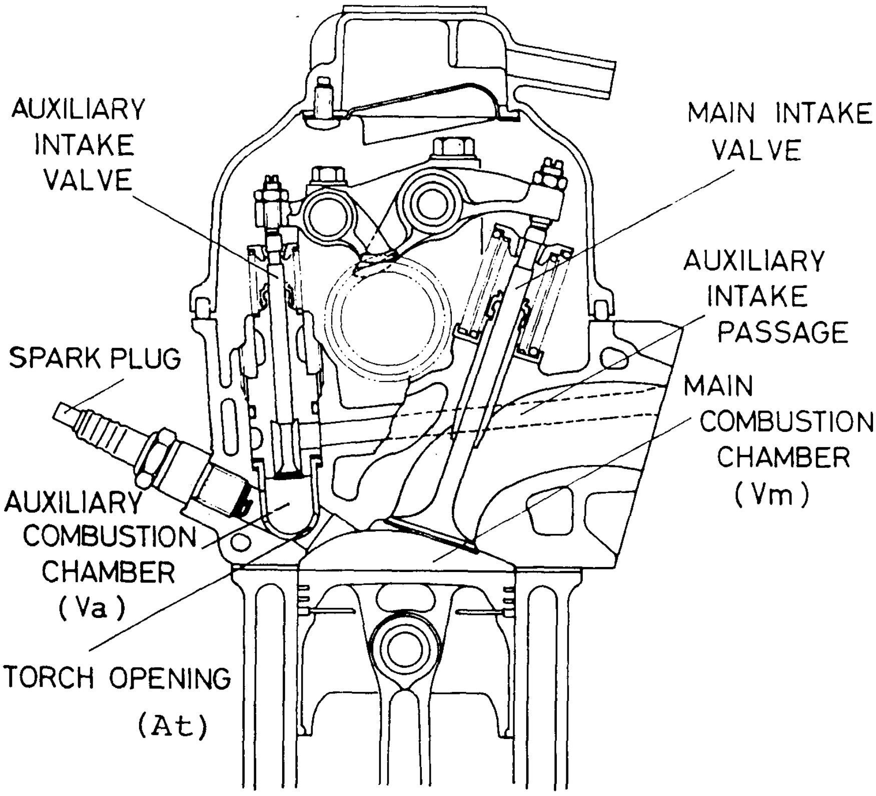

FIG. 4. Engine Cross Section

TABLE 1. Specifications of Test Engine

Bore × Stroke (mm)		74×86.5		
Displacement volume (V_d) (cc)		372 cc×4 cylinders		
Compression ratio (s)		8.0		
Auxiliary chamber volume (V_a) (cc)		4.65		
Torch opening area ratio (A_t/V_a) (1/cm)		1)	2)	3)
		0.027,	0.090,	0.159
Valve timing	IN open	10° ATDC		
Main chamber	IN close	30° ABDC		
(1 mm lift)	EX open	30° BBDC		
	EX close	10° BTDC		
Auxiliary chamber	IN open	30° ATDC		
(0.6 mm lift)	IN close	10° BBDC		

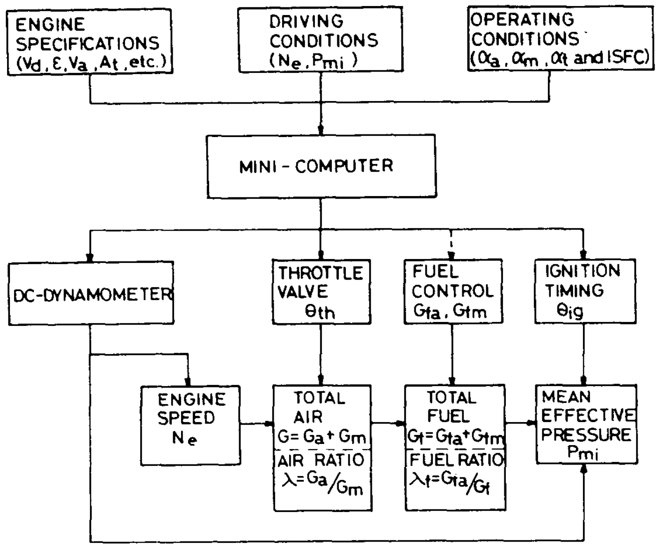

FIG. 5. Block Diagram for Engine Operation

TABLE 2. Engine Operating Conditions

Engine Speed	$N=3\,000$ rpm
Volumetric Efficiency	$\eta_v \simeq 50\%$
Total Air-Fuel Ratio	$\alpha_T \simeq 20$
Indicated Mean Effective Pressure	$P_{mi}=3.43$ kg/cm^2
Indicated Specific Fuel Consumption	ISFC$\simeq$240 gr/IPS-h

a minicomputer to meet the value of P_{mi} with the presumed one. Fig. 5 shows the block diagram for controlling engine operations, which are tabulated in Table 2.

Instruments: The gas pressure in both the chambers are measured by two piezo type pressure transducers (Kistler) respectively. The temperature-time diagrams for both the chambers are obtained with an infra-red radiometer (Huggins). The radiometer is calibrated using a black body furnace and the output obtained from the radiometer shows good agreement with the calibration curve. The frequency response of the radiometer was found to be about 100 kHz as results of the calibration tests. The exhaust gases are sampled continuously from the exhaust port. The gas analysis is carried out by the following analysers: non-dispersive infra-red analyser [NDIR] for CO and CO_2, chemiluminescence analyser [CLA] for NO and NO_x.

Data Proceesing: The time varying pressure and temperature of gas in both the combustion chambers are obtained with the electric pulse signals of ignition timing and crankangle (1° interval), and recorded into a data recorder, by monitoring all the output on an oscilloscope. From 100 cycles of the transient pressure and temperature stored in the magnetic disk of the minicomputer, the average values of i) gas pressure and temperature crankangle diagrams, ii) the maximum gas pressure and temperature, iii) the pressure and temperature gradient and iv) the heat release rate, together with their standard deviations are calculated.

RESULTS AND CONSIDERATIONS

The data obtained from the model is compared with the experimental results, to examine the adequacy of the model for the analysis of a three valve stratified charge engine for three different torch opening areas. For the comparison with the measured temperature curve, it is used the mass mean value (T_m) of the computed temperature, as obtained by the following equation,

$$T_m=(M_u \cdot T_u+M_b \cdot T_b)/(M_u+M_b) \tag{14}$$

Pressure and Temperature Diagrams: In all these cases, it can be seen that there is a good correlation between the computed and measured gas pressure in both the chambers during the compression period, as shown in Fig. 6, although the measured pressure are always lower than the calculated one, in the early period of compression. At the time of combustion, the computed pressure is slightly higher than the experimental value, which is the average of 100 cycles.

In the same diagrams, the computed gas temperature in the auxiliary chamber can be seen to follow closely the experimental data during both the compression and expansion processes. There is a slight difference in the lower range than 200°C during the compression stroke, which may be mainly cuased by the inherent characteristics of the infra-red radiometer.

For the main chamber, although the computed and measured temperature qualitatively agree with each other, some discrepancies can be observed near the maximum gas temperature. In particular, the measured gas temperature for the case 2 is remarkably lower than the calculated one on the range of maximum temperature, and furthermore both the measured and calculated temperature curves cross each other near the B.D.C.

These discrepancies are assumed to be caused by the following reasons. First reason is that the computed temperature is the mass average one as shown in euqation (14), while the experimental data is measured at a local measuring point and is an average of 100 cycles of combustion. The second one comes from less reliability of the combustion model in which the volume burning rate ($\dot{V}_b$) is estimatee by a simple assumption of burning velocity, and such factors affecting the combustion process as jet action, mixing diffusion and heat transfer between the torch flame and the lean mixture in the main chamber are all neglected.

Equivalence Ratio and Mass Flow: The computed value of equivalence ratio in the auxiliary chamber during the compression process is shown in Fig. 7. It may be noted that the upward motion of piston increases the pressure in the main chamber resulting in a mass flow through the torch opening, which is illustrated in the same figure and thereby decreases the richness of the mixture in the auxiliary chamber as shown in the figure.

Exhaust Emissions: The NO and CO calculated for both the chambers are plotted in Fig. 8 for the case 2. The total emissions obtained from the model are compared with the experimental data, in Fig. 9, for three different torch openings. In all these cases it may be observed that the computed values of NO is higher than the experimental data. This could be due to the reason that the computed gas temperature is always higher than the measured one. The same figure shows that the values of NO

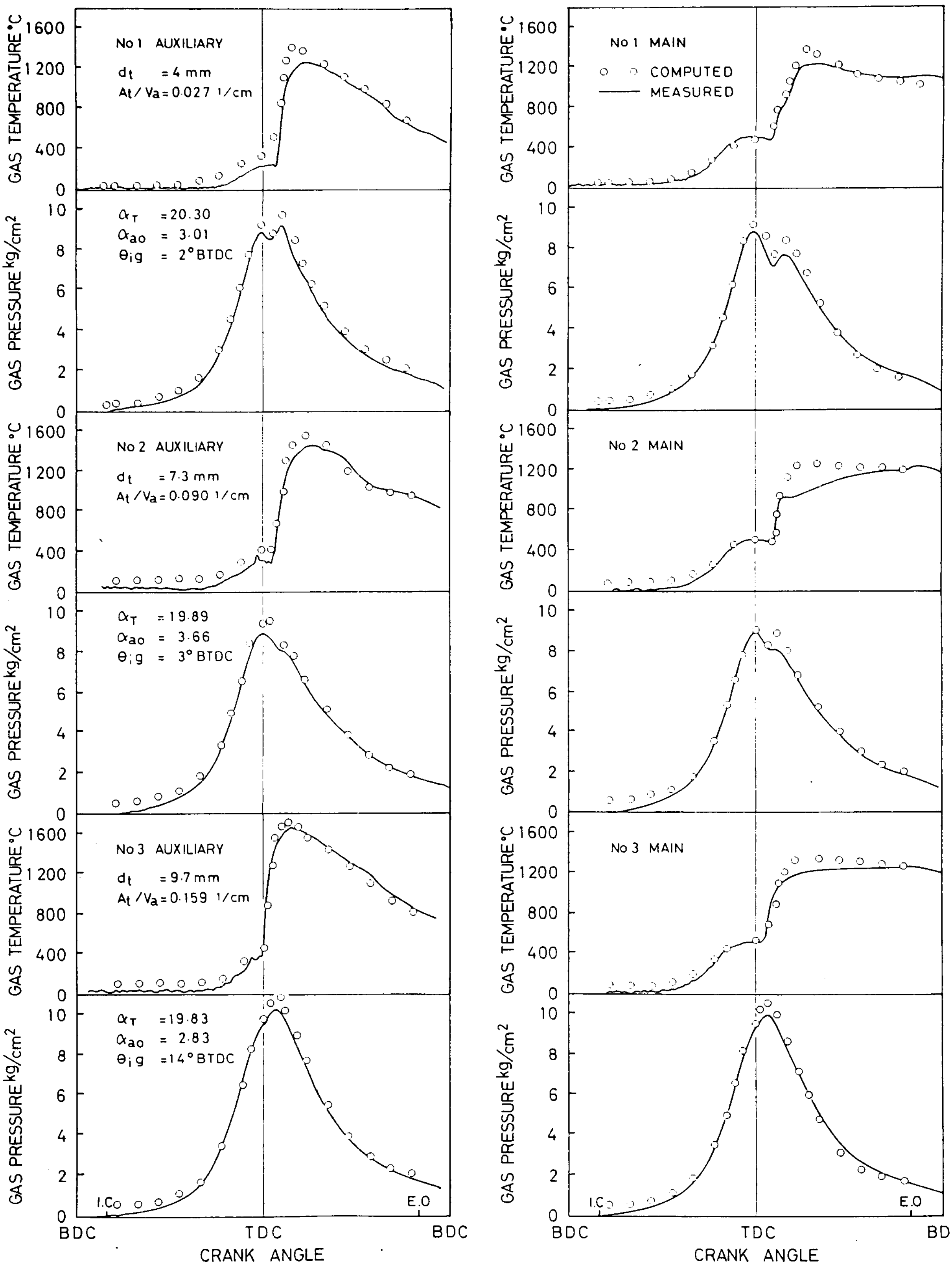

FIG. 6. Computed and Measured Temperature- and Pressure-crankangle Diagrams

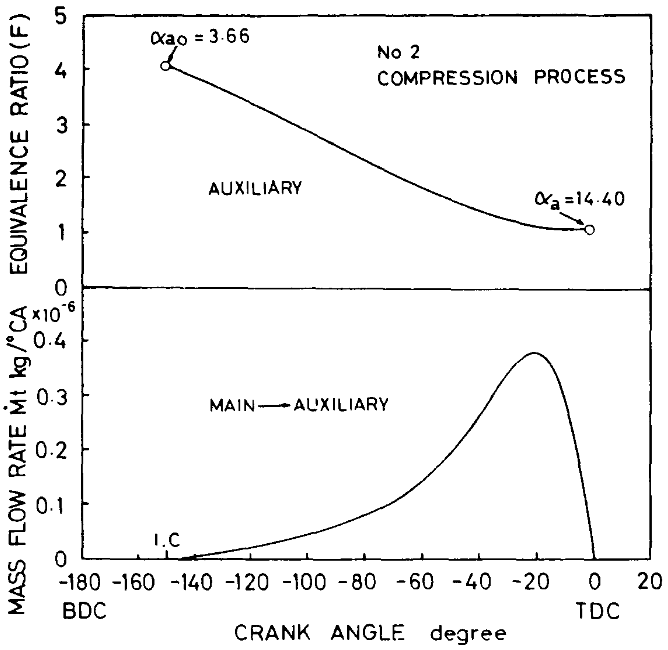

FIG. 7. Computed Equivalence Ratio in the Auxiliary Chamber and Mass Flow Rate through Torch Opening

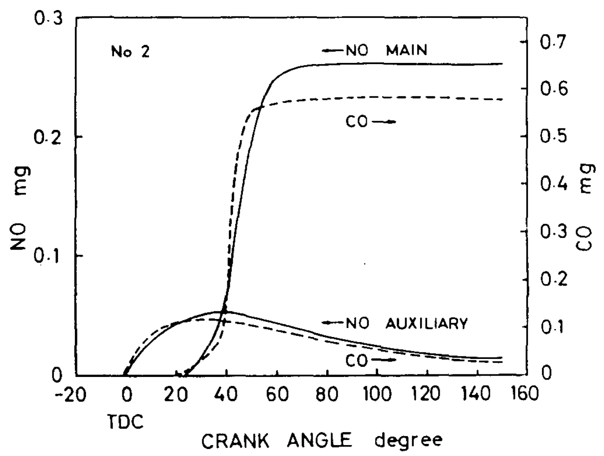

FIG. 8. Computed Emissions of NO & CO in both the Chambers

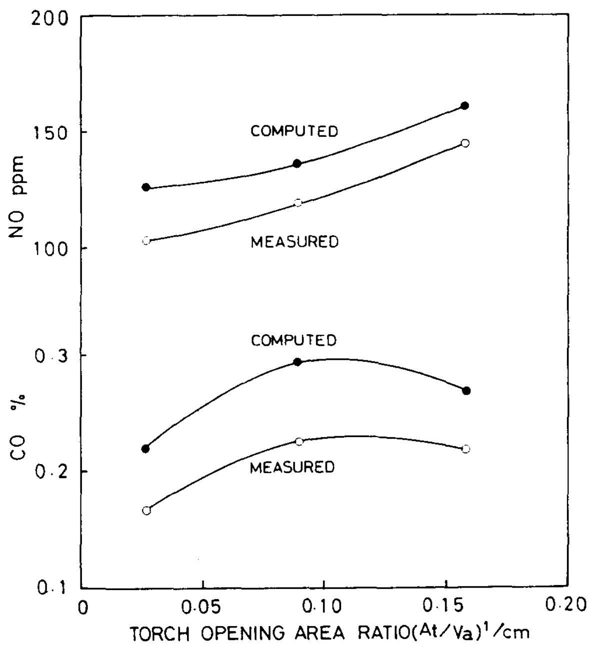

FIG. 9. Computed and Measured Total Emissions of NO & CO for Difference Torch Openings

formed increases as the torch opening area increases. This may be caused mainly by the higher peak temperature for larger torch opening area.

In the same figure, it may be noticed that the computed CO is also higher than the measured one. Since the exhaust gas is sampled just after the port, the CO from the cylinder may continue to undergo oxidation in the exhaust port, thereby causing such a reduction in the concentration.

CONCLUSIONS

The modeling of a three valve stratified charge engine has been described, with particular reference to the combustion process.

1) A satisfactory agreement has been observed between the computed and measured pressure and temperature time diagrams for both the chambers, except a few discrepancies in the temperature diagrams.

2) The eqivalence ratio in the auxiliary chamber is computed during the compression process by calculating the mass flow through the torch opening.

3) The NO and CO predicted from the model is in close agreement with the measured values, in particular the trend of both the curves plotted against the torch opening area is similar to each other.

Based on these considerations, it can be concluded that the developed model may be available for analysing the performance of a three valve stratified charge engine. However, the model seems to be inadequate for the case 2 and needs to be improved by taking into considerations such as the mixture formation before ignition and the jet action of torch flame in the main combustion chamber.

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SYMBOLS

A	= Area of any surface	mt
d	= Bore	m
F	= Equivalence Ratio	
g	= Acceleration due to gravity	m/sec^2
G	= Total air	kg
h	= Specific Enthalpy	kcal/kg
H	= Convective heat transfer coefficient	kcal/hr-m^2 °K
K	= Ratio of Specific heats	
M	= Mass	kg
N	= Engine speed	RPM
P	= Gas pressure	kg/cm^2
Q	= Heat transfer	kcal
R	= Gas constant	kcal/kg °K
t	= Time	s
T	= Gas temperature	°K
u	= Specific internal energy	kcal/kg
U	Internal energy	kcal
V	= Cylinder Volume	m^3
W	= W_c+W_n, Woschni's coefficient	m/s
θ	= Crankangle	
α	= Air fuel ratio	
ε	= Compression ratio	
λ	= G_a/G_m, Air ratio between both chambers	

SUBSCRIPTS

a	= Auxiliary chamber	s	= Spark point
b	= Burnt side	t	= Torch
c	= Combustion	th	= Throttle
e	= Equilibrium condition	T	= Total
ef	= Effective condition	u	= Unburnt side
f	= Fuel	w	= Wall
i	= Index for different surfaces	0	= Initial condition
ig	= Ignition	1	= Upstream condition
m	= Main chamber	2	= Downstream condition
mi	= Indicated mean effective condition		

SUPERSCRIPT

Dot = Derivative with respect to time

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