THEORETICAL AND EXPERIMENTAL STUDIES ON THE COMBUSTION OF SYNTHETIC FUELS IN SPARK IGNITION ENGINES.

A.A. Deenky and S.H. El-Feam
Department of Mechanical Engineering,
Faculty of Engineering, Mansoura University,
El-Mansoura, Egypt.

ABSTRACT

A theoretical and experimental investigation on the performance and emission of spark ignition engine have been made using gasoline, alcohol and gasoline-alcohol mixture as fuels. A thermodynamic model has been developed with superimposed pseudo-kinetic NO and CO formation. This has been used to compute estimated performance and emission data for a spark ignition engine. The model has also been used to compare the prediction performance and emission data when operating the engine with methanol, ethanol and gasoline fuels.

A series of an experimental tests were carried out on a single cylinder spark ignition engine to confirm the theoretical results of the mathematical model. Comparison of results indicates that alcohol seems to be a viable alternative fuel to gasoline with respect to efficient, operational and emission considerations.

INTRODUCTION

The incentives for studying the combustion of alternative fuels in engines are, some fuel or fuel additive may solve the problem of emissions, knock, derivability and efficiency. Also an alternative fuel may help conserve the world's petroleum supplies as well as being available when those supplies are exhausted. Transportation is unique among the energy consuming sectors of the economy of industrialized nations, because it is totally dependant on one source of fuel crude oil. It has therefore imperative to investigate the consequences of using non-petroleum fuels for transportation application.

Alcohol has been proposed as a clean-burning synthetic fuel, that could have many applications, and its advantages have been widely acclaimed [1,2]. It can be obtained either from the distillation of wood products or, more efficiently, by combining hydrogen and carbon monoxide which are available from coal gasification processes. Alcohol has a research octane number of 106 [3] compared to 90:106 normal gasoline, and a blending octane value of 110. Thus a 70/30 mixture of methanol with 90 octane gasoline gives a blend fuel with an octane number of 98. It could therefore be used as an effective substitute for tetraethyl lead as an anti-knock agent.

Information in the literature is plentiful for automotive use of methanol; however, few systematic studies exist on combustion and emission properties of an engine fueled with the higher alcohols and alcohol-gasoline blends [4,5]. The objective of this study is to obtain more quantitative information on the combustion and emission
properties of spark ignition engines using gasoline, methanol, ethanol, methanol-gasoline blends and ethanol-gasoline blends fuels.

THEORETICAL ANALYSIS

The availability of a mathematical model would obviously be most helpful in the further development of the spark ignition engine. A simple equilibrium cycle model with superimposed "pseudo kinetic" and ED formulation has been developed. This model is used to compare likely engine performance with gasoline, methanol and ethanol fueling. The cycle synthesis is concerned with the progressive step by step calculations of the physical and thermodynamic states of the charge in the combustion chamber at succeeding points around the relevant parts of the engine cycle.

The compression and expansion processes are considered isentropic, and the combustion process is assumed to occur instantaneously at constant volume at TDC. The working fluid is assumed to be a homogeneous fuel-air mixture in chemical equilibrium at the appropriate temperature and pressure at each point in the cycle. An elementary treatment of the exhaust and induction processes is included and provision for a degree of exhaust gas residual is incorporated. The assumed thermodynamic cycle is illustrated in Fig. 1.

![Diagram of the assumed thermodynamic cycle](image)

**Fig. 1** The assumed thermodynamic cycle.

1. Thermodynamic Cycle Analysis

During the compression process, the fuel-air mixture is compressed isentropically from "1" to "2". Initially the working fluid prior to compression is assumed to consist only of fuel vapor, air at atmospheric pressure and a temperature of 298 K. At the end of one complete cycle, recycled residual gas is included and the cycle computation is repeated. The cycle is completed at point "6" which is corresponding to point "8". The process is considered in an incremental manner, a 10 degree crank angle increment being adopted. The working fluid composition is considered to be constant during the compression process. The thermodynamic properties (enthalpy, entropy, and volumetric specific heat) were evaluated from sixth order polynomial equations [5]. The value of specific heat ratio is assumed constant during each crank angle increment and is
updated at the end of each increment to allow for the change in mixture pressure and temperature. During each increment in compression, an isentropic change of state due to piston motion is assumed. At this stage in the development of the model the working fluid during the compression process is assumed to be completely vaporized. This assumption can be improved later by assuming some of the fuel to be in the liquid state.

The compressed mixture at temperature T2 and pressure P2 is assumed to burn adiabatically at constant volume. In this analysis, the products of combustion are assumed in chemical equilibrium and comprise CO, CO2, O2, H2, H2O, OH, H, O, NO and N2.

To determine the number of moles of products of combustion, dissociation reactions are considered. In the present study there are six such independent reactions, those used are:

\[
\begin{align*}
\text{CO2} & \xrightarrow{k_1} \text{CO} + \frac{1}{2} \text{O}_2 \\
\text{H}_2\text{O} & \xrightarrow{k_2} \text{H}_2 + \frac{1}{2} \text{O}_2 \\
\text{H}_2\text{O} & \xrightarrow{k_3} \frac{1}{2} \text{H}_2 + \frac{1}{2} \text{O}_2 \\
\frac{1}{2} \text{H}_2 & \xrightarrow{k_4} \text{H} \\
\frac{1}{2} \text{O}_2 & \xrightarrow{k_5} 0 \\
\frac{1}{2} \text{O}_2 + \frac{1}{2} \text{N}_2 & \xrightarrow{k_6} \text{NO}
\end{align*}
\]

The expansion of the products of combustion from point "b" to point "a" (Fig. 1) is again treated in an incremental manner. A single expansion increment comprises two steps; these are:
1) An isentropic change of state due to piston motion during the increment.
2) Calculation of the revised equilibrium composition at the new pressure and temperature at the end of the increment.

The exhaust process is represented from "a" to "d" (Fig. 1). The process is treated in two stages. The first stage corresponds to the period from "a" to "b", which represents the event where the exhaust valve opens and the products of combustion expelled into the atmosphere. The process is irreversible and is accompanied by an increase
in entropy. However, the condition at point "g" can be found by allowing the burned gas remaining in the cylinder to expand isentropically, and do work in pushing out the expelled portion. The temperatures, pressures and entropy at point "g" then correspond to the condition at point "h". The process 1 → 2 is computationally treated in two steps. In the first of these steps, the gases remaining in the cylinder at the end of expansion are allowed to expand isentropically to an intermediate pressure $P_{m1}$, where $P_{m1} = \frac{1}{2} P_0$. Then in the second step the gases expand from $P_{m1}$ down to the exhaust pressure, $P_3 = 1$ atm. The second stage of the exhaust process corresponds to step 1 → 2 in Fig. 1. During this period, it is assumed that the plenum is returned to TDC, with expulsion of most of the residual gas remaining in the cylinder at BDC.

The induction process corresponds to the period 7 → 8 in Fig. 1, the current stage in the model's development equal exhaust and induction pressure have been assumed. At point "f" the intake valve opens and fresh completely vaporized charge is induced and mixed with the residual gases remaining in the cylinder from the previous cycle. The pressure is considered to remain constant throughout this process and it is assumed that no chemical reaction occurs.

The complete computation cycle outlined above is then repeated using the values obtained for point "b" as the new initial point in the cycle. The whole procedure is repeated until correspondence between conditions at point "i" and point "h" is obtained in successive cycles.

1. Performance Calculations

Computations have been performed for three fuels (CRBNA, DHON, and CYNON), for a compression ratio in the range 6 to 16 and for equivalence ratios in the range 0.6 to 1.6.

Applying the first law of thermodynamics, for a closed system, the net work done of the cycle is obtained by the following relation

$$W = \frac{1}{4} \left( u_1 - u_0 \right) \times \frac{1}{4} \left( u_1 - u_0 \right) \text{ J/mole, charge}$$

The indicated mean effective pressure is determined by dividing the net work done by the displacement volume. The indicated thermal efficiency is the ratio of the net work done to the energy content of the fuel supplied.

3. Emission Calculation

The kinetic mechanism proposed for NO calculation is the extended Zeldovich mechanism, which is comprised of the following reactions:

$$\text{N}_2 + \text{O} \xrightarrow{K=7} \text{NO} + \text{A} \quad \text{...(7)}$$
The differential equation governing the formation of NO is:

\[
\frac{d[NO]}{dt} = k_7[O_2][NO] + k_8[O_2][N] - k_9[NO][O] = \frac{k_7[O_2][NO] + k_8[O_2][N] - k_9[NO][O]}{dt}. \quad \text{(10)}
\]

Taking the temperature, pressure and equilibrium values of O, N2, O2, OH and N at the end of each crankangle increment during the expansion process, kinetic NO formation is computed using Euler integration method for eq. (10). The initial value for NO concentration at TDC is assumed to be zero. In the present study, it is assumed that at TDC all carbon content of the fuel is instantaneously converted to CO which is subsequently oxidized to CO2 during the expansion process via the following reaction:

\[
CO + OH \xrightarrow{k_{10}} CO_2 + H \quad \text{(11)}
\]

Thus the rate equation for CO is simply as follows:

\[
\frac{d[CO]}{dt} = -k_{10}[CO][OH] - k_{10}[H][CO] \quad \text{(12)}
\]

The CO level through the expansion stroke is computed from eq. (12) and by solving the carbon conservation equation:

\[
[CO] + [CO_2] = [CO_2] + [CO] \quad \text{(13)}
\]

EXPERIMENTAL STUDY

The engine utilized in this work was a single cylinder air cooled standard spark ignition engine. It was equipped with standard throttle and carburetor to monitor the exhaust temperature. A needle valve was used to precisely control the fuel rate. Since the engine was operated at wide open throttle and different throttle positions in these tests, adjustment of this valve was the means for obtaining different air fuel ratios. Spark advance was adjusted in all tests for maximum power, while compression ratio was kept constant at 6.5:1.

Air flow rate to the engine was determined with the aid of an air box, calibrated orifice and manometer. The power output of the engine was measured by a dc dynamometer unit coupled directly to the engine shaft. This dynamometer is also used in mooring tests to determine the friction horsepower. Measurements of the engine speed were performed with a speed indicator.
The fuels used in experiments were gasoline, methanol, ethanol, a 10 % methanol-90 % gasoline and a 50 % ethanol-50 % gasoline. To avoid problems of cold starting with alcohol fuels, a dual fuel system was used. The engine was started on gasoline and then switched over to the alcohol fuels when warmed up.

RESULTS AND DISCUSSIONS

Computations have been performed for three fuels (CD16, CD50, and CM50) for different compression ratios and different equivalence ratios. Shown in Figs. 2 through 7 are some of the more important computed results of the performance and emission levels.

Figure 2 shows the effects of fuel type and equivalence ratio on theoretical peak cycle temperature and pressure. The cycle peak pressure and temperature vary with equivalence ratio. Because of the dissociation and the influence of different specific heats of CO and CO2, the maximum peak pressure and temperature occur with rich

![Fig. 2](image1)

**Fig. 2** Effect of fuel type and equivalence ratio on calculated peak pressure and peak temperature.

![Fig. 3](image2)

**Fig. 3** Effect of fuel type and equivalence ratio on calculated relative peak pressure and peak temperature.
Fig. 4 Effect of fuel type and equivalence ratio on calculated work done and specific energy consumption.

Fig. 5 Effect of fuel type and equivalence ratio on calculated relative work done and specific energy consumption.

mixtures. The diagrams also show that the fuel composition can influence the calculated peak pressure and peak temperature. Alcohol has a lower proportion of oxygen in the fuel molecule than isooctane. This leads to a lower peak pressure and peak temperature for alcohol-s fuel than isooctane. More ready comparison of the results shown in Fig. 3 can be made if normalized with isooctane as the datum.

Shown in Fig. 4 are the effects of fuel type and equivalence ratio on calculated net output per unit mass of charge and indicated specific energy consumption such a quantity is importance when comparing, since the cost of producing fuel is generally reported as cost per energy unit. For these reasons we define the indicated specific energy consumption as the rate of energy consumed divided by the indicated power. As a less dissociation occurs with alcohol, resulting in a lower ratio of moles product-moles reactant, there is a slightly higher net output per unit mass of charge and slightly lower indicated specific energy consumption for alcohol fuels than isooctane.

This result can be easily shown in Fig. 5, when the results are normalized with the results obtained for isooctane as a datum.
Figure 6 illustrates the influence of fuel type and equivalence ratio on calculated kinetic NO and CO mass emission levels (gm/lawh). In order to include the range of equivalence ratios to be considered the NO and CO mass emissions levels have been plotted on a log scale. These diagrams show that the peak NO levels occur near the lean side of stoichiometric because of the greater oxygen concentration and the associated high temperature. For lean mixture, the rate of NO formation rises due to the attendant drop in the adiabatic flame temperature. For a rich mixture, the rate decreases due to oxygen deficiency and falling adiabatic flame temperature. One other observation is that the nitric oxide concentration is mainly dependent on the equivalence ratio(0). The effect of fuel type is less important. However, alcohols produces slightly less nitric oxide than isocetane with both lean and slightly above stoichiometric. This is because of the lower adiabatic flame temperature for alcohols as shown in Fig. 7.

Figure 7 Relative calculated CO and NO emission levels for gasoline, methanol and ethanol fuels.
Also shown in Fig. 6 the computed CO levels from the kinetic mechanism as a function of equivalence ratio. The diagram demonstrates that for fuel rich mixtures, the CO concentration increases steadily with increasing Φ as the amount of excess fuel increases. With fuel lean mixture (0.7 < Φ < 1.0), CO concentration varies little with Φ. Also, it can be seen that the effects of fuel type on CO concentration is less important. However, alcohol fuels produce a slightly lower CO level as a result of a lower ratio of carbon in the fuel molecules, and its more favorable dissociation properties. This can be shown in Fig. 7, where the results were normalized to iso-octane results as a datum.

**Fig. 8. Measured brake thermal efficiency for used fuels, at different equivalence ratios.**

**Fig. 9. Measured brake thermal efficiency for used fuels, at different speeds.**

**Fig. 10. Measured brake thermal efficiency for used fuels, at different speeds.**
Figures 8-12 show the experimental results using gasoline, methanol, gasoline-methanol, and gasoline-ethanol blends as fueling fluids. Fig. 8 shows the relation between the engine brake thermal efficiency and fuel air equivalence ratio at constant speed of 1500 RPM. It can be noted that the efficiency is higher for alcohol fuels rather than gasoline and gasoline-ethanol blends. Figs. 9 and 10 also represent the engine brake thermal efficiency versus engine brake horsepower and engine speed respectively. The brake thermal efficiency for methanol and ethanol is higher than for gasoline. This is attributed to the lower heating values of these fuels than gasoline.

Measured levels of CO and NO in the engine exhaust gases when using gasoline, methanol, ethanol, gasoline-methanol, and gasoline-ethanol blend fuels are shown in Fig. 11 and Fig. 12 respectively. From Fig. 11 it can be noted that CO concentration levels are higher than that for alcohol fuels. Also, Fig. 12 shows that for lean mixture (equivalence ratio < 1), NO concentration levels are higher for gasoline than that for alcohol and gasoline-ethanol blends fuels.

**Fig. 11 Measured CO concentration levels for different fuels.**

**Fig. 12 Measured NO concentration levels for different fuels.**

**CONCLUSION**

On the basis of the theoretical and experimental results, the following conclusions can be offered:

1. The fuels which are still regarded today as unconventional will become increasingly important in the future, such fuels to which relatively well suited to the automotive sector are alcohol and gasoline-ethanol blends.

2. Alcohol as a motor fuel provides improved power output, specific energy consumption and higher thermal efficiency relative to gasoline.

3. The concentration levels of CO and NO emissions in the exhaust gas are reduced with all alcohol and gasoline-alcohol blends and fuels.
MACCHIATURE

CR = Compression ratio
T/A = Fuel to air mass ratio
Pn = Maximum pressure in atm.
SE = Specific energy consumption (kJ/kw-h)
TDC = Top dead center
TV = Maximum temperature
V = Volume
k = Forward rate constant for reaction \( 1 \) (cm\(^2\)/mole, s)

No. = No. of moles of \( O_2 \) in mixture
IV = Internal volume in cm\(^3\)
U = Net work done per cycle
\( \eta \) = Brake thermal efficiency
\( \phi \) = Equivalence ratio, (actual /A ratio in stoichiometric /A ratio)

REFERENCES