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NITROGEN OXIDE CONTROL WITH DELAYED-MIXING, STRATIFIED-CHARGE ENGINE CONCEPT

U.S. ENVIRONMENTAL PROTECTION AGENCY
Office of Air and Waste Management
Office of Mobile Source Air Pollution Control
Emission Control Technology Division
Ann Arbor, Michigan 48105

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NITROGEN OXIDE CONTROL WITH DELAYED-MIXING, STRATIFIED-CHARGE ENGINE CONCEPT

by

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ABSTRACT

The purpose of this study is to explore methods of controlling the nitrogen oxide emissions from internal combustion engines. From computer calculations, the delayed mixing stratified charge engine concept was selected. In the delayed mixing stratified charge engine concept, combustion is initiated and completed in a fuel-rich region, then air is mixed into those rich products. A study of existing engines shows that some operational stratified charge engines limit nitrogen oxide emissions in a manner similar to the delayed mixing concept. A single cylinder engine was modified to include an air injection valve. When air was injected after rich combustion, the nitrogen oxide emissions were lower, the hydrocarbon emissions were lower, the carbon monoxide emissions were about the same and the efficiencies were lower than for homogeneous operation at the same overall fuel-air ratio.

PREFACE

The material presented in this report is essentially the Ph.D. Dissertation of L. W. Evers. Much of the support for the study was provided by the U. S. Environmental Protection Agency. The theoretical study given in Chapter II was presented as a Society of Automotive Engineers (SAE) paper (741172) at the International Stratified Charge Engine Conference in Troy, Michigan (1974). Other results are to be presented in a SAE paper in February 1977.

The thesis was written in the conventional British units. Appendix I has been added to the report which includes the conversion to SI Units.

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NOMENCLATURE

a_c	shape factor "a" of the Weibe combustion function
a_e	shape factor "a" of the Weibe expansion function
A_2	air or air with recirculated exhaust gases
ATDC	after top dead center
BTDC	before top dead center
CFM	cubic feet per minute
CFR	Cooperative Fuel Research Engine
CO	carbon monoxide
D	delayed mixing stratified charge engine
EGR	recirculated exhaust gas
F	equivalence ratio, fuel to air ratio divided by the stoichiometric fuel to air ratio
F_E	equivalence ratio of the combustion products at the time of exhaust or the overall equivalence ratio
F_P	equivalence ratio products
H	homogeneous engine
HC	hydrocarbon
I	instantaneous mixing stratified charge engine
IEE	indicated enthalpy efficiency - percent
IMEP	indicated mean effective pressure - psi
ISNO	indicated specific nitric oxide - g/ihp-hr
L:A-P	delayed mixing stratified charge engine, see Chapter III, C

L;P-A	one member of the limited mixing stratified charge engine group, see Chapter III, C
MBT	maximum advance for best torque
m_c	shape factor "m" of the Weibe combustion function
m_e	shape factor "m" of the Weibe expansion function
NO	nitric oxide
NO_x	nitrogen oxide, all of the oxides of nitrogen
P	combined mixed products of system 1 and system 2
P_1	product system 1
P_2	product system 2
P_E	combined products at exhaust conditions
P_i	initial pressure of the reactants
ppm	part per million
PROCO	Ford Programmed Combustion Engine
R_1	reactant system 1
R_2	reactant system 2
R;P-A	instantaneous mixing stratified charge engine, see Chapter III, C
RPM	revolutions per minute
t	time
t_c	time of combustion
t_e	time of expansion
T_{iR}	initial temperature of reactants
TCCS	Texaco Controlled Combustion System Engine
v(t)	volume function
v_{cl}	clearance volume of combustion chamber

$V_d(t)$	displaced volume at time "t"
$V_T(t)$	total volume of combustion chamber at time "t"
ϕ_o	overall equivalence ratio
ϕ_c	carbureted equivalence ratio
θ_{si}	start of air injection in crank angle degrees
θ_{sp}	spark timing in crank angle degrees

CHAPTER I

INTRODUCTION

The oxides of nitrogen emissions from vehicle internal combustion engines are one of the causes of photochemical-smog. Smog is produced by the interaction of nitrogen oxide, hydrocarbons, air and sunlight. In an attempt to prevent this form of air pollution, the Federal Government has set standards for the emissions from motor vehicles. Nitrogen oxide emissions of conventional spark ignition engines are difficult to deal with because they tend to be large at high efficiency and high power operating conditions.

Our general approach to the problem was to study engine processes, doing theoretical calculations to determine if the nitrogen oxide emissions could be reduced while maintaining satisfactory engine performance The most promising concept was then selected for experimental evaluation.

A computer program approximating conventional homogeneous engines was used to provide a better understanding of nitric oxide emissions. The computer program was also used to determine how unusual operating conditions influence nitric oxide emissions. A second computer program was used to study nitric oxide emissions of stratified charge engines. From these studies it appeared that the delayed mixing stratified charge engine concept had the most potential for limiting nitric oxide emissions and maintaining engine performance.

The delayed mixing stratified charge engine concept consists of having all of the fuel concentrated in a rich region where combustion is initiated. After combustion of the rich mixture is completed, the lean region consisting of air or air with recirculated exhaust gas is gradually and thoroughly mixed into the rich products of combustion.

When the nitrogen oxide emissions of various types of internal combustion engines were examined, it was apparent that the delayed mixing concept was part of a unique group of engines. This group of engines uses similar principles of operation and has similar trends in the nitrogen oxide emissions. The nitrogen oxide emissions of this group tend to be lower than other engines for operation with near stoichiometric fuel-air ratios. The other group of engines is usually operated lean to limit nitrogen oxide emissions.

The experimental program is intended to simulate the combustion processes of the delayed mixing stratified charge engine concept. The experimental engine was not intended to represent a practical engine configuration but was, instead, a means for testing the delayed mixing concept. A single cylinder (CFR) engine was modified to include an air injection valve which could inject air at the desired time in the engine cycle. The experimental engine was operated at a variety of conditions which provided the results necessary to evaluate the delayed mixing concept.

SUMMARY

The theoretical studies of engine power, fuel economy and emissions shown in Chapter II indicate that the delayed mixing stratified charge engine concept is capable of limiting the nitric oxide emissions while having reasonable efficiency and specific power. The delayed mixing process consists of rich combustion followed by air being mixed into the rich products. During the initial phase of rich combustion little nitric oxide is formed due to the lack of oxygen in the charge. When the air is mixed into the rich products of combustion, two factors can be used to limit nitric oxide. First, if the final overall fuel-air ratio is kept near stoichiometric, the nitric oxide will be low because the mixture is rich for most of its kinetic history. However, if the overall fuel-air ratio is made lean, the product mixture will have both the high temperatures and oxygen required for nitric oxide formation. With delayed mixing and a lean overall fuel-air mixture, the nitric oxide emissions can be very high. The second factor which can be used to limit nitric oxide formation is the time at which expansion cooling stops the nitric oxide kinetics. If the end of the active nitric oxide kinetic period were to occur before the air is added to the rich products, the final nitric oxide concentration would be

that associated with rich combustion. Likewise, as the end of the active nitric oxide kinetic period is extended into the mixing process, the final nitric oxide concentration will be more affected by the mixing process.

It is shown in Chapter III that some real engines have low nitrogen oxide emissions when operated near stoichiometric fuel-air ratios. Those engines and the delayed mixing concept have similar combustion processes. This group has been defined as the limited mixing stratified charge engine group because the products of combustion are by some means temporarily prevented from mixing with the rest of the charge. The mixing process which follows combustion is characterized by the products mixing into the air, or by the air mixing into the products, or by some intermixing process. Engines are judged to be part of the limited mixing group by examining the influence of fuel-air ratio on the nitrogen oxide emissions and by examining the combustion processes. The Ford Programmed Combustion (PROCO) Engine, the Newhall Engine and the divided chamber diesel engine are included in the limited mixing stratified charge engine group.

The delayed mixing stratified charge engine concept was simulated by a single cylinder (CFR) engine with provisions for air injection. A rich charge is drawn into the combustion chamber through the normal inlet valve.

The equivalence ratio of this charge is called the carbureted equivalence ratio. At the desired time in the cycle the air is injected which reduces the equivalence ratio to the value called the overall equivalence ratio. Details of the experimental engine, the entire test facility and instrumentation are presented in Chapter IV.

The experimental engine was operated with air injection occurring after combustion to simulate the delayed mixing processes. The experimental results are presented in Chapter V. A comparison can be made of the emissions for homogeneous operation and for delayed mixing operation at the same overall equivalence ratios. The delayed mixing operation is much lower in nitrogen oxide emissions, lower in hydrocarbon emissions and about the same in carbon monoxide emissions. From the hydrocarbon and carbon monoxide emissions, it can be concluded that the combustion process is completed. The magnitude of the nitrogen oxide emissions from the delayed mixing operation is similar to that of homogeneous operation if its overall equivalence ratio were equal to the carbureted equivalence ratio of the delayed mixing engine. Apparently, little additional nitrogen oxide is formed due to air injection, because the completion of the combustion occurs late in the expansion when the temperatures are relatively low. Results with a large diameter or with a small diameter air injection nozzle were similar.

The indicated enthalpy efficiency of delayed mixing operation is very low compared to that of homogeneous operation at the same overall equivalence ratio. Since the completion of combustion occurs late in the expansion little additional work can be obtained. Attempts were made to inject the air during the combustion process in order to extract more work and find the tradeoff between nitrogen oxide emissions and efficiency. These experiments resulted in the discovery that air injection during combustion would disrupt the combustion process. In a sense the air injection would tend to blow out the flame. As a result the hydrocarbon emissions increased, the carbon monoxide emissions increased, the efficiency remained low and the nitrogen oxide increased. Other tests were run with air injection before combustion. In these tests, all the emissions were similar to those of homogeneous operation at the same overall equivalence ratio. The efficiency approached that of homogeneous operation at the same overall equivalence ratio. Efficiency was less than with homogeneous operation because of the greater heat transfer rates due to motion of the charge caused by air injection. The motion of the charge also resulted in very stable and rapid combustion.

Another factor which reduces the efficiency farther is the work required to compress the injected air. In the test facility the air is compressed separately and injected as desired. The measured power and corresponding measured efficiency do not include the work required to compress the injected air. Estimates of the air injection compression power are made and subtracted from the measured power to obtain the corrected power and corresponding corrected efficiency. Generally the measured efficiency is slightly greater than homogeneous operation at the carbureted equivalence ratio and the corrected efficiency is lower.

In order to evaluate the potential of the delayed mixing concept, it is necessary to know if the low efficiency is an inherent result of the combustion process or if it is a result of the means used to simulate the combustion process. In the Newhall engine, the PROCO engine and the divided chamber diesels the combustion process, which is similar to the delayed mixing concept, is achieved through the use of fuel injection. Thus, it would seem possible to devise a delayed mixing engine that would use fuel injection instead of air injection to obtain stratification. The resulting efficiency would not include an external compressor. Much of the efficiency of the experimental engine can be attributed to the air injection which is the means selected to simulate the delayed mixing process.

A certain amount of inefficiency is due to the combustion process itself. The fact that the combustion is delayed will mean that it will occur later in the expansion and consequently be less efficient. The Newhall engine, El-Messir (1973), has a lower efficiency than a corresponding homogeneous engine. The divided chamber diesel, Obert (1968), has a lower indicated enthalpy efficiency than a corresponding open chamber diesel.

The calculation of both the efficiency and the IMEP depend upon the power of the engine. For the same fuel flow rate, the efficiency and the IMEP will be proportional to each other. A comparison can be made between operation with air injection after combustion and homogeneous operation at the same mass flow rate of fuel and the same total mass of air. The low efficiency associated with air injection after combustion results in correspondingly low IMEP.

The measured IMEP of the experimental engine simulating delayed mixing was found to be somewhat greater than that of homogeneous operation. The reason for the greater IMEP is that the carbureted air flow rate was maintained constant with air injection adding to the total mass of the charge. In a sense the air injection is supercharging the engine.

In summary, the delayed mixing stratified charge engine concept is an effective method of controlling the nitrogen oxide emissions. The carbon monoxide emissions are equivalent to those of homogeneous engines and the hydrocarbons are lower. The disadvantage of the delayed mixing concept is its low efficiency inherent in the combustion process.

CONCLUSIONS

1. The computer models of homogeneous and stratified charge engines indicated that the delayed mixing stratified charge engine concept could be used to limit nitric oxide emissions, while having reasonable specific power and efficiency.
2. In general engines can be divided into two groups representing markedly different responses of nitrogen oxide emissions to changes in the overall fuel-air ratio. The first group is characterized by producing the maximum nitrogen oxide (either ppm or g/kw-hr) near stoichiometric fuel-air ratios or slightly lean. The second group, which includes the delayed mixing concept, is characterized by producing the maximum nitrogen oxide emissions at leaner fuel-air ratios and being less sensitive to changes in the fuel-air ratio.
3. Experimental simulation of the delayed mixing stratified charge engine concept has shown that nitrogen oxides are much lower than for a corresponding homogeneous engine. The emissions of hydrocarbons are lower than for a corresponding homogeneous engine, while the carbon monoxide emissions are about the same. Efficiency is much lower than for a similar homogeneous engine.

4. The difference between the predictions of reasonable efficiency and the low efficiency found with the simulated delayed mixing engine is due to the late time in the expansion when the combustion process is completed and due to external work required to compress the injection air.
5. Air injection resulted in increased motion of the charge which increased heat transfer rates.
6. Air injection during combustion disturbed the combustion process which resulted in greater hydrocarbon emissions, greater carbon monoxide emissions, and lower efficiency.
7. Air injection before combustion resulted in emissions similar to those for a corresponding homogeneous operation and efficiencies slightly below the corresponding homogeneous operation. The combustion process was found to be very rapid with reduced cycle to cycle peak pressure variations.
8. The major disadvantage of the delayed mixing stratified charge engine concept is its inherent low efficiency resulting from the extended combustion process. Attempts at reducing the combustion period by injecting air earlier resulted in disturbing the rich combustion process without increasing the efficiency.

CHAPTER II

A THEORETICAL STUDY OF NITRIC OXIDE EMISSIONS FROM INTERNAL COMBUSTION ENGINES

A. INTRODUCTION

It is the objective of this study to establish a concept for an internal combustion engine that would limit the nitric oxide emissions with minimal sacrifice in specific power and efficiency. Initially the work was aimed at understanding the formation of nitric oxide. Of the various computer models which were developed, the most significant are the homogeneous and stratified charge engine models. A paper which describes these engine models entitled "A Search for a Low Nitric Oxide Engine" was presented at the Central States Section of the Combustion Institute in March, 1974, (unpublished) and at the International Stratified Charge Engine Conference in October, 1974, SAE 741172, Evers, Myers and Uyehara (1974). Excerpts from the paper are presented to describe the nitric oxide emissions from homogeneous and stratified charge engines.

A large number of combustion and expansion configurations and processes were studied where exhaust NO_x concentrations might vary by several orders of magnitude. Because of the number of calculations needed short computational times were judged more important than a high degree of precision. It was further felt that after identification of low NO_x configurations and processes which also had reasonable efficiency and specific power, more detailed and more precise simulations could be run if desired. Consequently, a simplified computer program was developed.

In developing the simplified computer program it was deliberately kept flexible and not identified with a particular engine configuration. This resulted in the arbitrary specification of a number of input items. However, in general, when specifying input items a specific engine "configuration" is typically in mind.

B. NITRIC OXIDE KINETICS

The kinetics of nitric oxide have been used by many investigators in predicting the nitric oxide concentration in the products of combustion. The method requires the selection of appropriate elementary chemical reactions, the corresponding reaction rates, and a group of simplifying assumptions.

Newhall and Shahed (1971) showed that the two elementary chemical equations A and B (called the Zeldovich mechanism) satisfactorily predicted the nitric oxide concentration of fuel-lean mixtures but somewhat underestimated the concentration of fuel-rich mixtures.

$$N_2 + O \rightleftharpoons NO + N \qquad (A)$$

$$O_2 + N \rightleftharpoons NO + O \qquad (B)$$

The tests were performed in a closed constant volume cylindrical combustion chamber. In order to perform the kinetic calculations, Newhall assumed that all of the species in the combustion products except nitric oxide were at equilibrium concentrations.

The Zeldovich mechanism is used for all the kinetic calculations presented in this chapter. The equations are easily handledin computer calculations. For more information about the method used to calculate nitric oxide concentrations see Appendix A. Additional equations can be added to the Zeldovich mechanism in hopes of improving the accuracy of kinetic calculations. Since we are looking for large changes in nitric oxide levels, the Zeldovich mechanism is sufficient.

Lavoie et al. (1970) used an expanded set of elementary chemical reactions which added equations C through F to the Zeldovich mechanism.

$$N + OH \rightleftharpoons NO + H \qquad (C)$$

$$H + N_2O \rightleftharpoons N_2 + OH \qquad (D)$$

$$O + N_2O \rightleftharpoons N_2 + O_2 \qquad (E)$$

$$O + N_2O \rightleftharpoons NO + NO \qquad (F)$$

They also showed that the N and N_2O concentrations exist nearer a steady state value than an equilibrium value.

Thus they assumed that N and N_2O exist at the steady state concentrations and all the other species, except NO, exist at equilibrium concentrations. A computer model based on this kinetics mechanism was found to be in agreement with internal combustion engine test results. The comparison was made for fuel-lean mixtures, No comparisons were offered for fuel-rich mixtures. Lavoie pointed out that equations D, E and F are only important for lean mixtures at low temperatures.

A thorough review of nitric oxide kinetics done by De Soete (1975) concluded that equations C and D had little effect for equivalence ratios less than 1.5 and temperatures greater than 3240°R (1800°K). DeSoto's choice of the most acceptable set of kinetics equations are the Zeldovich mechanism (equations A and B) plus equations E and F.

Fenimore (1971) showed that for fuel-rich hydrocarbon flames the Zeldovich mechanism did not predict the nitric oxide formed in the flame. He suggested that nitric oxide formed in the flame was due to a different kinetic mechanism such as equations G and H. The nitric oxide generated by that mechanism is called "prompt NO".

$$CH + N_2 \rightleftarrows HCN + N \qquad (G)$$

$$C_2 + N_2 \rightleftarrows 2CN \qquad (H)$$

The assumption of equilibrium concentration of atomic oxygen in the flame was investigated by Iverach et al.

(1973). They measured the nitric oxide concentration in a rich premixed flame and back calculated the Zeldovich mechanism to obtain the atomic oxygen concentration required to produce that quantity of nitric oxide. Through other measurements they were able to determine the actual atomic oxygen present in the flame. A comparison shows that the atomic oxygen concentration required by the Zeldovich mechanism is orders of magnitude greater than the actual concentration for hydrocarbon flames and about equal for CO/H_2 flames. They concluded that prompt NO was formed by a different kinetic mechanism as suggested by Fenimore. Their choice for the mechanism is equation G and I.

$$C + N_2 \rightleftharpoons CN + N \qquad \text{(I)}$$

By only using the Zeldovich mechanism in our calculations we expect to somewhat underestimate the nitric oxide concentrations for fuel-rich combustion.

C. COMPUTER MODELS

Since the nitric oxide concentration is not markedly affected by the intake, compression,or exhaust processes, they were not included in the computer program in the interest of reducing the computer time. Calculations of specific power and efficiency required estimation of the work of these processes. The intake and exhaust work was assumed to be zero. An isentropic compression to the state specified at the beginning of combustion was used to estimate the work of compression. It was further decided not to include heat transfer in the computer programs. Neglecting the heat transfer shortens the computer time while resulting in greater pressures, temperatures and an overestimate of the nitric oxide concentration. In essence, the approach was to treat the gases as being in one or more closed adiabatic containers of uniform pressure with specified initial conditions and an arbitrarily specified expansion and combustion process.

1. One System Model

The one system model is intended to describe homogeneous combustion similar to a conventional homogeneous charge spark ignition engine, but in a general manner. To describe the regions within the one-system model we considered two approaches. In both approaches the reactants were assumed to be one homogeneous region. The mixed products approach assumes the products to be one homogeneous region with constant properties throughout, including pressure and temperature. As new products of combustion are formed they are instantaneously mixed with existing products which can change the average properties of the product region. The unmixed products approach assumes the product region to be made up of a number of small subregions each with its own individual properties. Each subregion could have a different temperature but they all have the same pressure. As products of combustion are formed they are grouped into the small subregions which maintain their own identity and do not mix with the other products.

An advantage of the unmixed products approach is that temperature gradients and their influence on nitric oxide formation can be shown in the product region. Blumberg and Kummer (1971) compared the nitric oxide formed by the mixed and unmixed products approach. They found the ratio of nitric oxide predicted by the mixed products approach to nitric oxide predicted by the unmixed products approach to vary by only ± 17% over a range of equivalence ratios from 0.7 to 1.3.

The mixed products approach was selected for these calculations because it was sufficiently accurate and required less computer time. The computer time for the mixed products approach is shorter since it is only necessary to keep track of two regions, products and reactants. With the unmixed products approach it is necessary to keep track of the reactants plus the many small subregions of products.

The thermodynamic equations and computer programs used to describe the one-system model are given in Appendix B.

2. Two System Model

The two system model is used to represent nonhomogeneous combustion as occurs in stratified charge engines. The program consists of two one-system models each capable of handling two regions, reactants and products. As is shown in Figure 1, initially both system 1 and system 2 are homogeneous reactants.

In the general case, reactants in both systems would be converted to products but not necessarily at the same rate. After some interval to be specified (called the time of combination) the two systems are combined to form a single system having three regions as shown in Figure 1. The reactants remaining in system 1 or 2 at the time of combination are maintained intact but the products from both system 1 and 2 are immediately mixed. See Appendix C for details of the two system model and computer program.

All calculations reported herein used the special cases of either delayed or instantaneous mixing models as shown in Figure 1. In both the delayed and instantaneous mixing models the reactants in system 2(A_2) are either air or a homogeneous mixture of air plus exhaust gas recirculation (EGR) of exhaust composition. For delayed mixing the combination of system 1 and system 2 occurred immediately but A_2 did not start to mix with

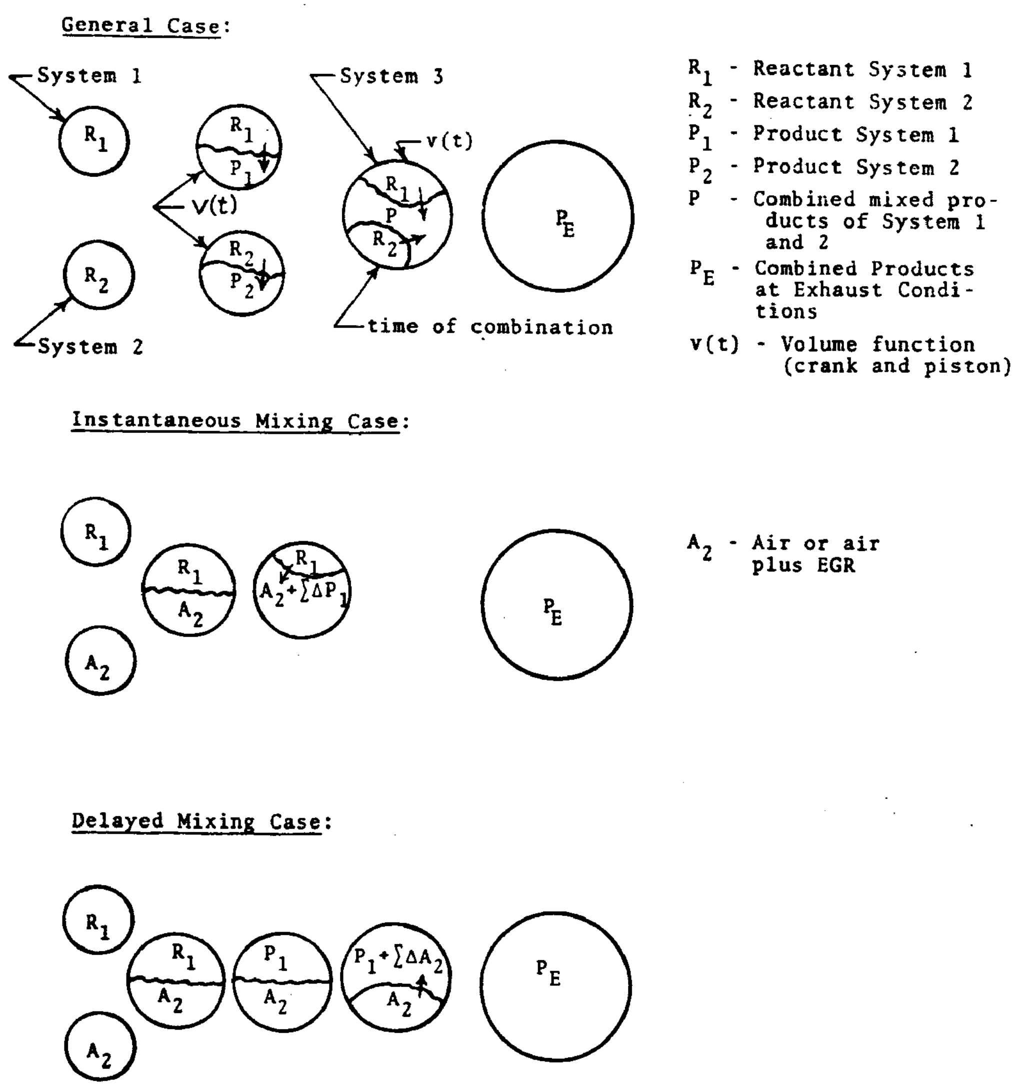


Figure 1 Two System Model of the Stratified Charge Engine Configurations

products P_1 until system 1 was entirely burnt. Then A_2 is mixed with P_1 at a specified rate until complete mixing occurs and only products having the exhaust air-fuel ratio exist in the system.

In instantaneous mixing, combination occurs immediately; R_1 is converted to P_1 at a specified rate; and P_1 is instantaneously mixed with A_2. To illustrate, if the rich products of system 1 are mixed with A_2 (let A_2 be air), the composition of $[A_2 + \Sigma\Delta P_1]$ (see Figure 1) will change continuously from that of air to that of exhaust products as R_1 is burnt. This is in contrast with delayed mixing where, if A_2 is air and R_1 is a rich mixture, the composition of $[P_1 + \Sigma\Delta A_2]$ will gradually change from rich products to products having exhaust composition.

D. POTENTIAL ENGINE CONFIGURATION

1. Homogeneous Charge-Conventional SI Engine

As indicated previously, when specifying initial and other conditions necessary to run the program, one tends to have a specific engine configuration in mind. The first configuration which was studied primarily for reference purposes was basically the conventional spark ignition engine with a wide open throttle. In order to use the program for this configuration, the program requires that the pressure, temperature and composition at the start of combustion, the rate or extent of combustion and a function to describe the system volume be specified.

The Weibe (1956) function was selected to represent both the extent of combustion and the system volume. Figure 2 illustrates use of the Weibe function to describe the extent of combustion. Three variables are required to define a specific combustion process. For the calculations presented the time of combustion (t_c) was varied while m_c equaled two and a_c equaled five. Also shown in Figure 2 is a cosine burning law which sometimes is used to represent combustion, Heywood, et al. (1973).

The expansion function also uses the Weibe function to describe the displacement volume (see Figure 3). The total volume of the system is equal to the displacement volume plus the clearance volume. A clearance volume was selected which is equivalent to a compression ratio of nine. Since the expansion function shows an increasing volume from the clearance volume to the maximum volume this is analogous to a crank and piston at top dead center and at bottom dead center. Both the time of expansion (t_e) and m_e were varied while a_e was held constant at five.

The initial condition of the one-system model can be defined by its pressure, temperature, fuel-to-air equivalence ratio, fuel composition and the amount of recirculated exhaust gas. The fuel was taken to be C_8H_{16}. It was assumed that the fuel-to-air equivalence ratio for the

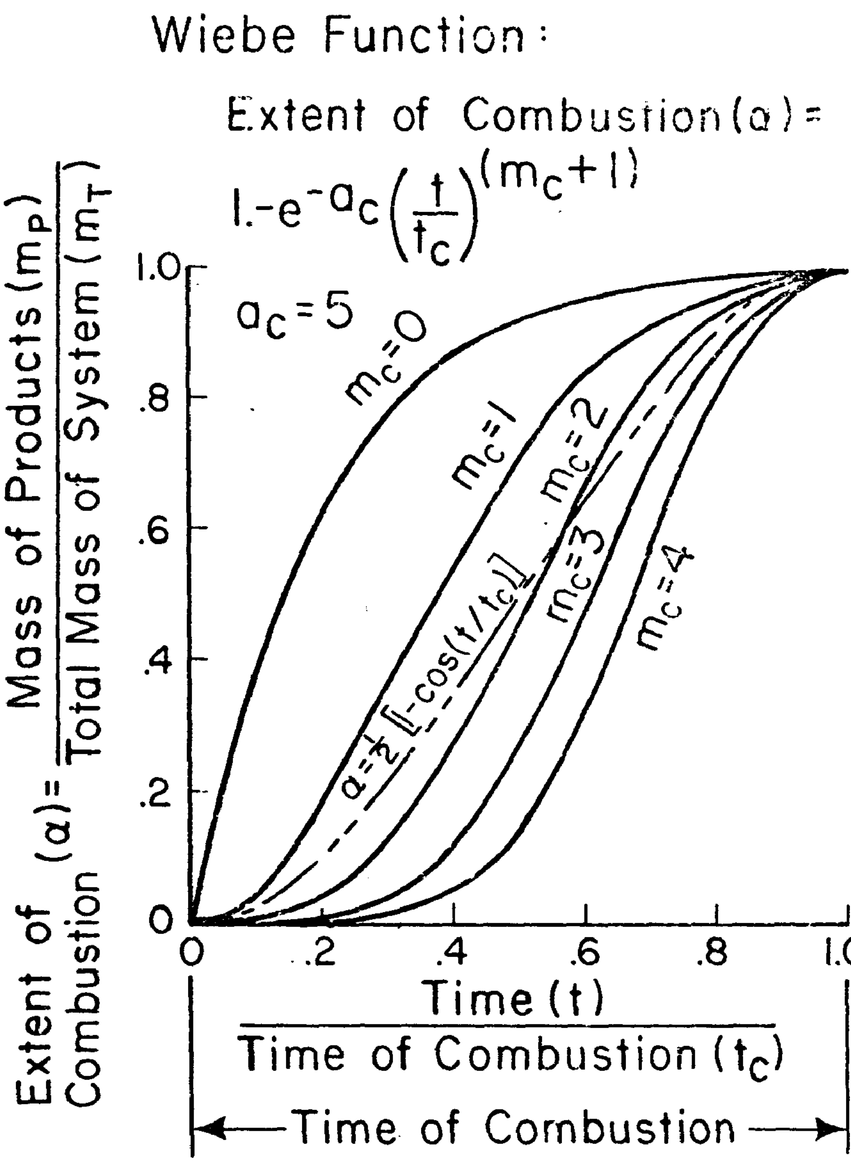


Figure 2 The Combustion Function for the One-System and Two-System Models

$$\text{Total Vol.}\left[V_T(t)\right] = \text{Clearance Vol.}\,(V_{cl}) + \text{Displaced Vol.}\left[V_d(t)\right]$$

$$\frac{V_T(t)}{V_d(t_e)} = \frac{V_{cl}}{V_d(t_e)} + \frac{V_d(t)}{V_d(t_e)}$$

Where:

$$\frac{V_{cl}}{Vd(t_e)} = \frac{1}{\text{Compression Ratio} - 1}$$

$$\frac{Vd(t)}{Vd(t_e)} = 1. - e^{-a_e}\left(\frac{t}{t_e}\right)^{(m_e+1)}$$

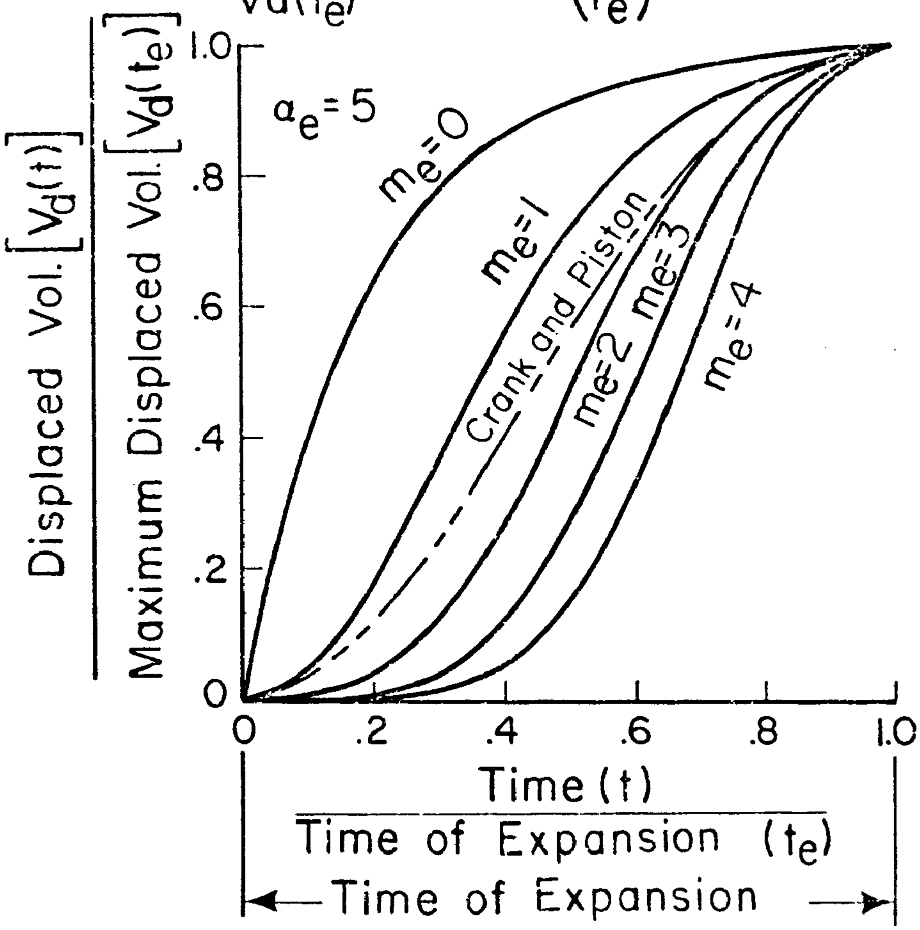


Figure 3 The Expansion Function for the One-System Model (Weibe) and the Two-System Model (Crank and Piston)

recirculated exhaust gas is the same as that of the reactants.

Three parameters were selected to characterize the performance of an engine. These are indicated specific nitric oxide (ISNO) which relates the pollution level to engine power, indicated enthalpy efficiency (IEE) which measures the utilization of the fuel, and indicated mean effective pressure (IMEP) which is related to specific power output and thus relative engine size. These terms are defined in Appendix D which also shows how the compression work was calculated. See Appendix I for SI units.

One set of calculations represented a study of seven independent variables. Four of these variables were the initial conditions, two of these variables were t_e and m_e from the expansion function and one variable was t_c from the combustion function. Each of seven variables was assigned a high and a low level. All possible combination of these variables were calculated ($2^7 = 128$ runs).

From the seven variable-two level calculations, it is possible to establish that the four major variables influencing nitric oxide levels are the fuel-to-air equivalence ratio, the initial temperature of the reactants, the exhaust gas recirculation, and the time of expansion. Consequently, another parametric set of calculations was

1. As discussed by Lauck et al. (1962), the indicated enthalpy efficiency is numerically equal to thermal efficiency which is more commonly but incorrectly used.

performed with these four variables. The values of the variables used in these calculations are shown in Table I. A total of 135 calculations were made to represent all possible combinations.

The results of the calculations are presented in Figures 4 and 5. In each curve shown in Figure 4, indicated specific nitric oxide is plotted versus indicated mean effective pressure for the fifteen combinations of equivalence ratio and fraction of exhaust gas recirculation. Each of the nine curves represent one of the nine combinations of the three values of time of expansion and the three values of initial temperature of the reactants.

The first observation is that the nine indicated specific nitric oxide curves are very similar in shape to each other. Also, not unexpectedly, only at high equivalence ratios (rich mixtures) or low equivalence ratios (lean mixtures) does the indicated specific nitric oxide reach desirably low levels. There does not exist some fortuitous combination of variables which excludes the influence of equivalence ratio. Exhaust gas recirculation in combination with either rich or lean mixtures contributes greatly to the reduction of nitric oxide.

The addition of exhaust gas recirculation or cooler initial temperatures results in lower product temperatures and consequently lower nitric oxide concentrations.

TABLE I

Values of the Parameters for the Calculations

Show in Figures 4 and 5

Parameters	Values
Equivalence Ratio (F)	.6, .8, 1.0, 1.2, 1.4
Initial Temperature of the Reactants (T_{iR})	800.°R, 1200.°R, 1400.°R (444.°K), (667.°K), (778°K)
Exhaust Gas Recirculation (EGR)	.0, .1, .2
Initial Pressure (P_i)	250. psi (172400. N/m^2)
Time of Expansion (t_e)-sec	.005, .025, .05 (∿6000 rpm) (∿1200 rpm) (∿ 600 rpm)
Expansion Shape (m_e)	1.
(a_e)	5.
Time of Combustion (t_c)-sec	.05 t_e
Combustion Shape (m_c)	2.
(a_c)	5.

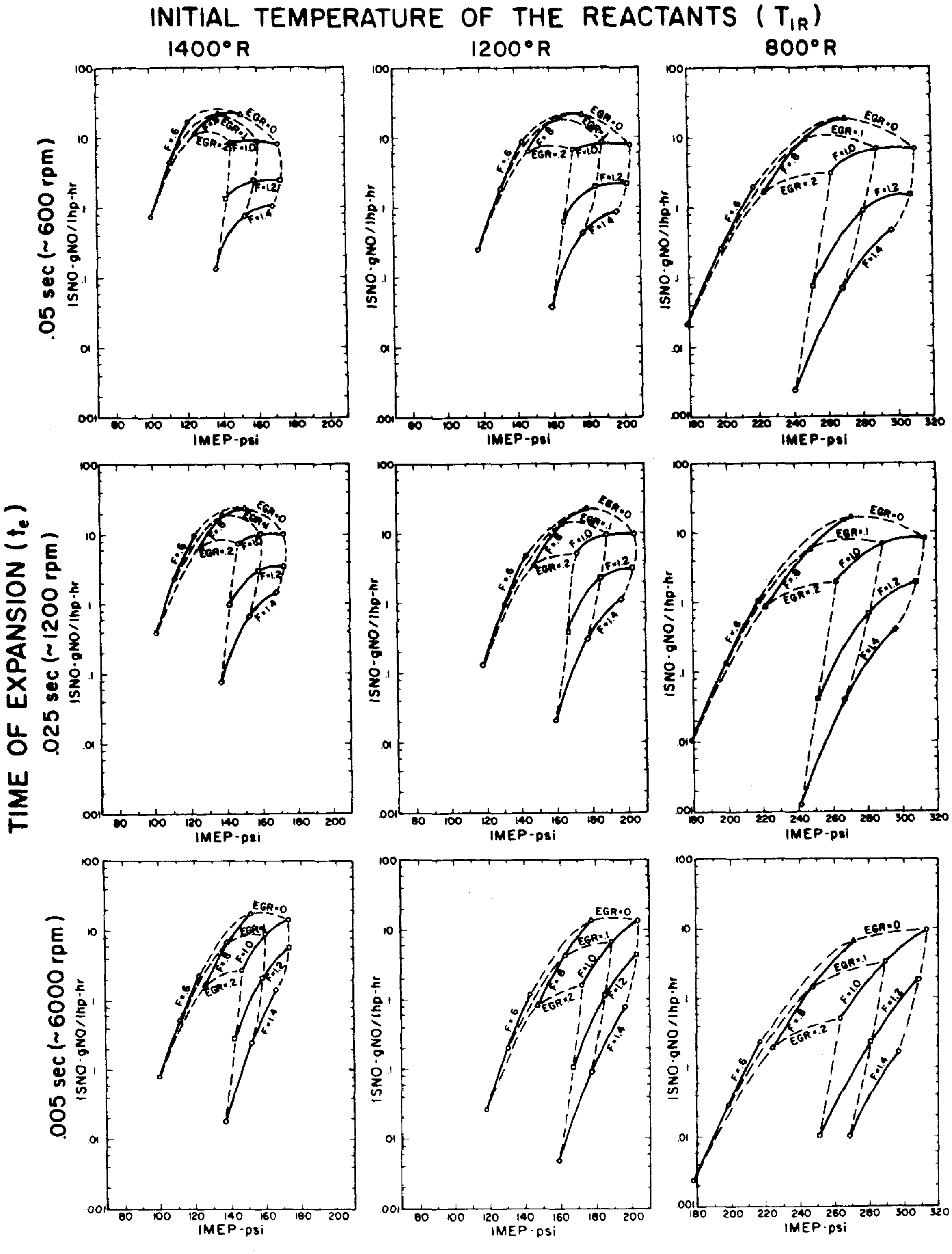


Figure 4. Indicated specific nitric oxide (ISNO) versus indicated mean effective pressure (IMEP) for homogeneous charge engine configuration.

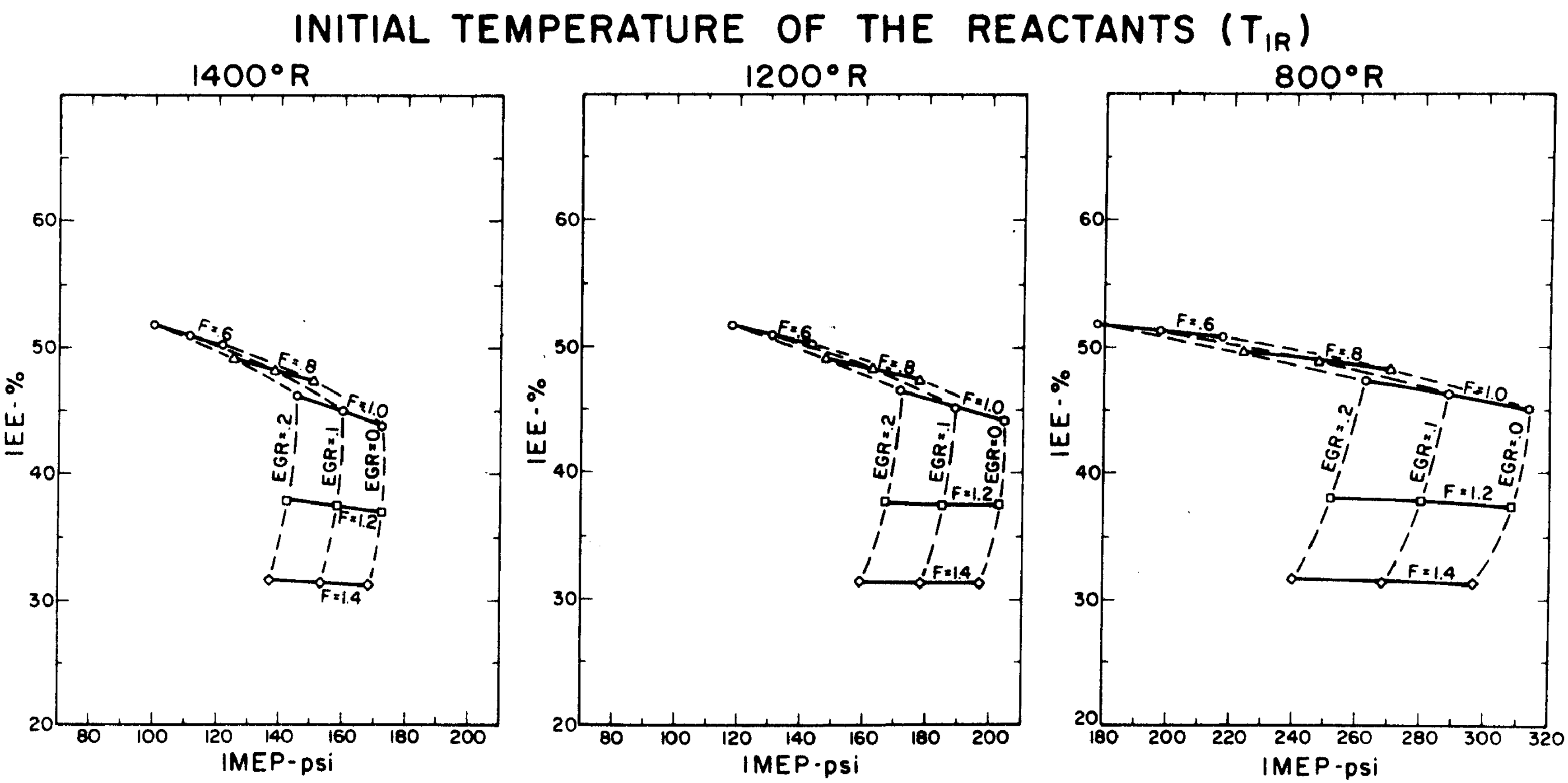


Figure 5. Indicated enthalpy efficiency (IEE) versus indicated mean effective pressure (IMEP) for homogeneous charge engine configuration.

Peak specific nitric oxide levels occur just to the lean side of stoichiometric except for high engine speeds. The peak nitric oxide levels occur in the lean region because of the greater oxygen concentration and high associated temperatures. As the mixtures become leaner, the flame temperatures decrease substantially and so do the nitric oxide rate constants and the amount of nitric oxide formed. However, note the change in the effect of EGR (at F = 1.0, for example) from the lower right curves hand to the upper left hand curves in Figure 4.

The indicated mean effective pressure is changed by those factors which influence the specific volume of the charge or the amount of fuel. Reduced temperature, higher fuel air ratios in lean region, or less exhaust gas recirculation all result in a greater indicated mean effective pressure. The magnitude of the IMEP is greater than conventional spark ignition engines because the initial conditions resulted in a larger mass of charge. Calculations with a smaller charge have the same trends.

In the computer model, a decrease in the time of expansion results in lower nitric oxide concentrations because less time is available for nitric oxide formation. In the model we have assumed no heat transfer. In an actual engine, the amount of heat transfer also depends on the time of expansion (or cycle time) and a decrease in the time of expansion (decreased cycle time) would cause decreased heat transfer which will tend to increase the level of nitric oxide concentration. Actual nitric oxide concentrations would reflect a balance of these effects. Since any engine coupled to the wheels by conventional transmissions would operate over a range of expansion times, it would be difficult to take advantage of possible benefits of rapid expansion to limit nitric oxide formation.

The indicated enthalpy efficiency curves, Figure 5, have well known shapes. Changes in indicated mean effective pressure stretch the curves. The indicated enthalpy efficiency is seen to be mainly a function of equivalence ratio with a small change due to exhaust gas recirculation. At high equivalence ratios the efficiency drops due to unburnt fuel. As the mixture becomes leaner than stoichiometric the efficiency increases because of the lower temperatures and the accompanying reduction in specific heats and dissociation.

2. Stratified Charge Engine

Generally, in a stratified charge engine having the objective of reducing NO_x emissions, combustion takes place in a rich region and the rich products and air are combined in a prescribed way to give a stoichiometric (or leaner) exhaust mixture. The two cases - delayed and instantaneous mixing - described under the two-system model represent two extremes of mixing and were used to study stratified charge engine performance. Crank and piston expansion was used for the two-system model because of the minor effect of changes in the expansion function had on NO.

The instantaneous mixing model is somewhat similar to an engine having a prechamber in which the rich reactants are burned. The products leave the prechamber and are rapidly mixed with the air in the main chamber. In the instantaneous mixing model the mixing would be instantaneous, the pressure in both chambers would be the same and the volumes of both the prechamber and the main chamber would change. In the delayed mixing model the air would be in the prechamber. After combustion, air would gradually be forced out of the prechamber and into the products by changes in the volumes of the main chamber and prechamber.

Let us define the time of heat release[2] to include both the energy release due to rich combustion and the energy release when additional air is supplied to the products of rich combustion. Note that heat release does not include energy release due to changing chemical equilibrium. The first computations that were run compared the effects of changing the time of heat release on delayed and instantaneous mixing (see Figure 6), when the exhaust products are stoichiometric and the overall exhaust gas recirculation is twenty percent. The relative shapes of the

2. The term heat release is used in a descriptive sense, and refers to the analogy between combustion and heat addition processes.

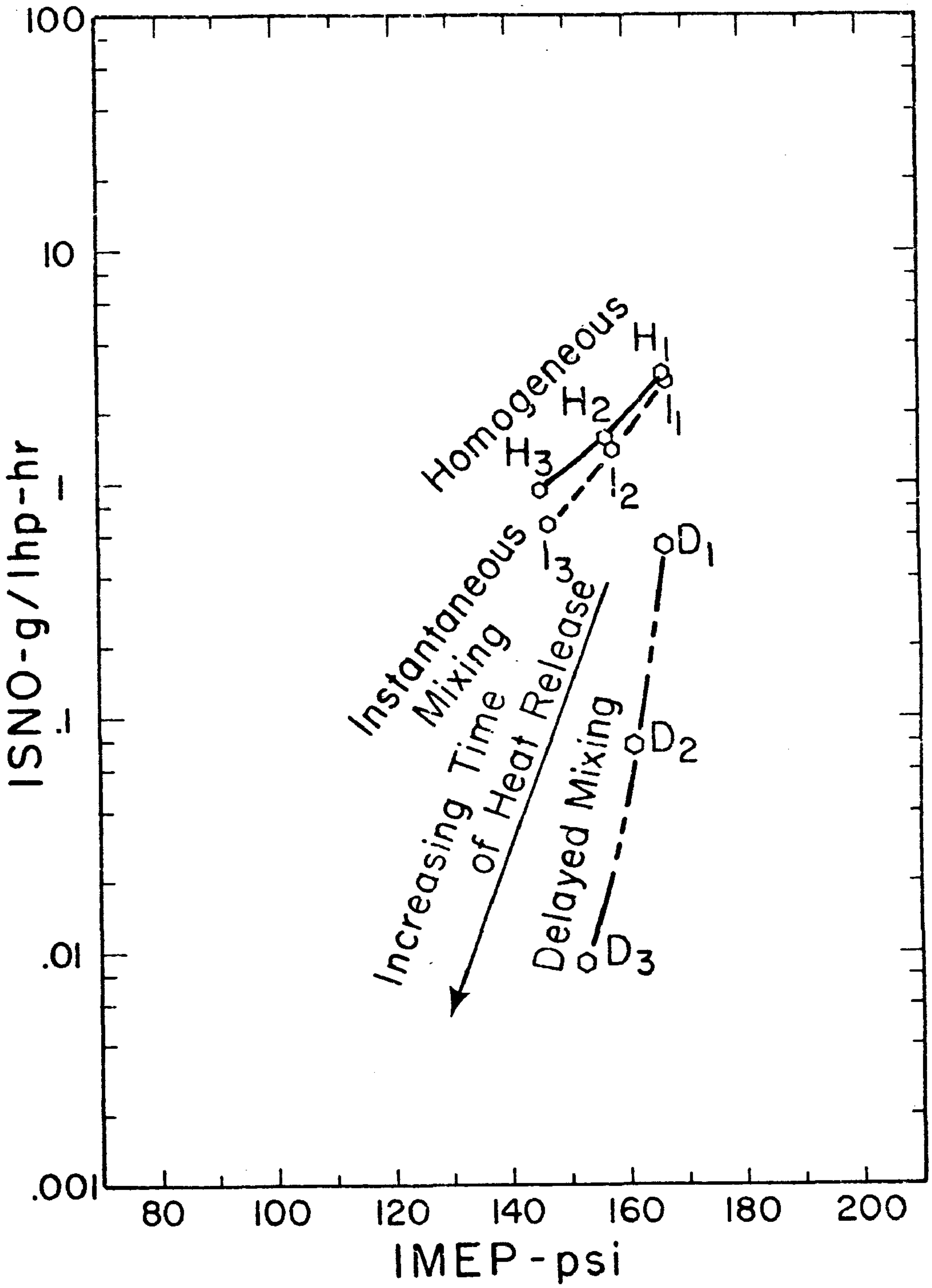


Figure 6 The Influence of the Times of Heat Release for a Homogeneous Charge Engine Operating with F = 1.0 and EGR = .2 and the Delayed and Instantaneous Mixing Stratified Charge Engine Configuration Operating with $F_{exhaust} = 1.0$ and $EGR_{overall} = .2$

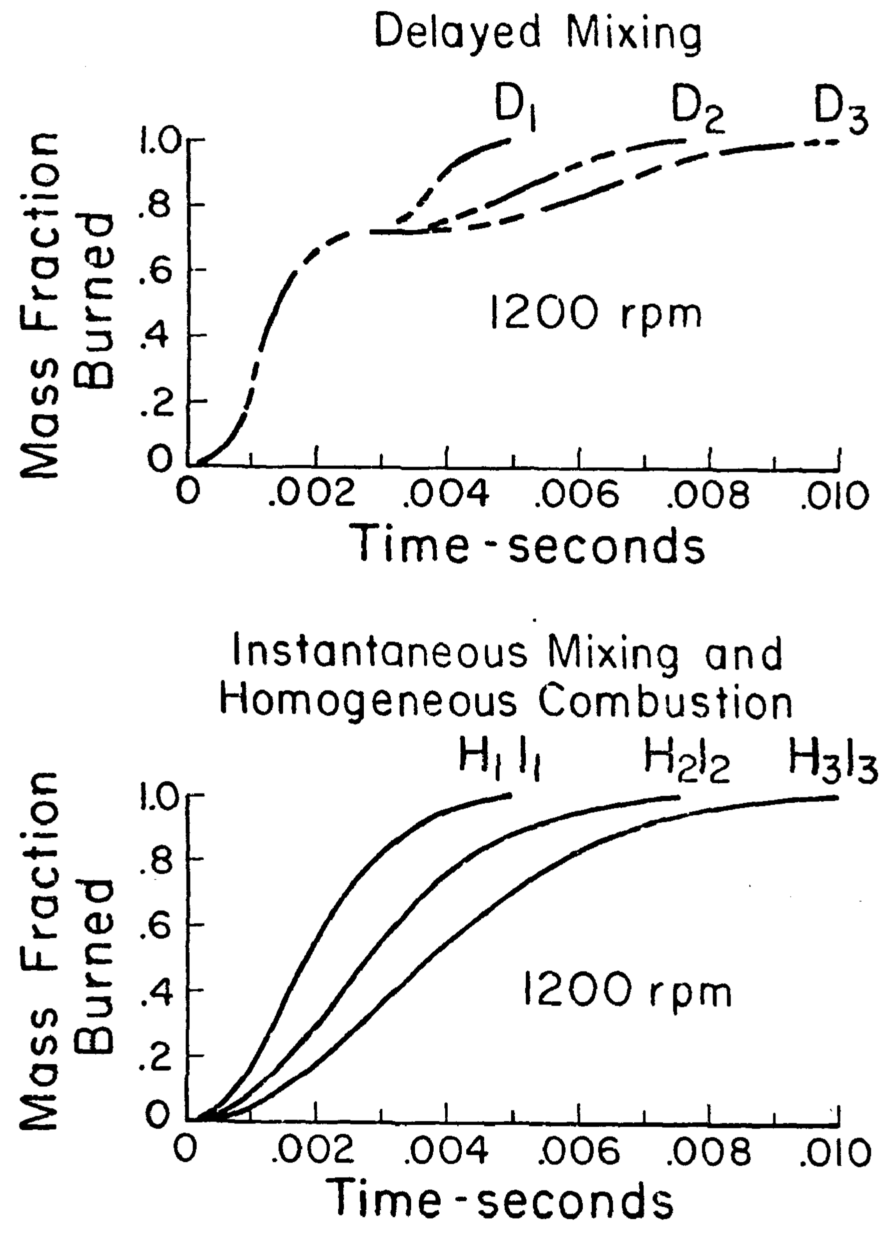


Figure 7 A Comparison of the Heat Release for Instantaneous Mixing, Delayed Mixing and Homogeneous Combustion Corresponding to Figure 6.

heat release curves are shown in Figure 7. It should be noted that, as the duration of heat release was increased, the major amount of combustion for instantaneous mixing occurred considerably later than that for delayed mixing. This occurred because, in the delayed case, only the period of mixing and not the duration of the rich combustion was changed.

In spite of the relatively delayed combustion for the instantaneous mixing case, it shows significantly higher NO_x for the same IMEP. The reasons for this unexpected result is that for instantaneous mixing the product mixture $[A_2 + \Sigma \Delta P_1]$ is always in the lean region starting with essentially air passing through the relatively high NO_x formation zone of around 0.8 equivalence ratio and finally reaching a F = 1.0. For delayed mixing the product mixture $[P_1 + \Sigma \Delta A_2]$ is on the rich side starting with F = 1.4 and ending with F = 1.0.

The next set of computations were run with a constant time of heat release but the overall EGR and overall fuel-air ratio were varied. Recirculated exhaust gas was assigned the composition corresponding to the final overall equivalence ratio. Heat release patterns for both delayed and instantaneous mixing are shown on the top of Figure 8 and corresponds to D_2 and I_2 in Figure 7. Also included in Figure 8, on the bottom, is the variation of

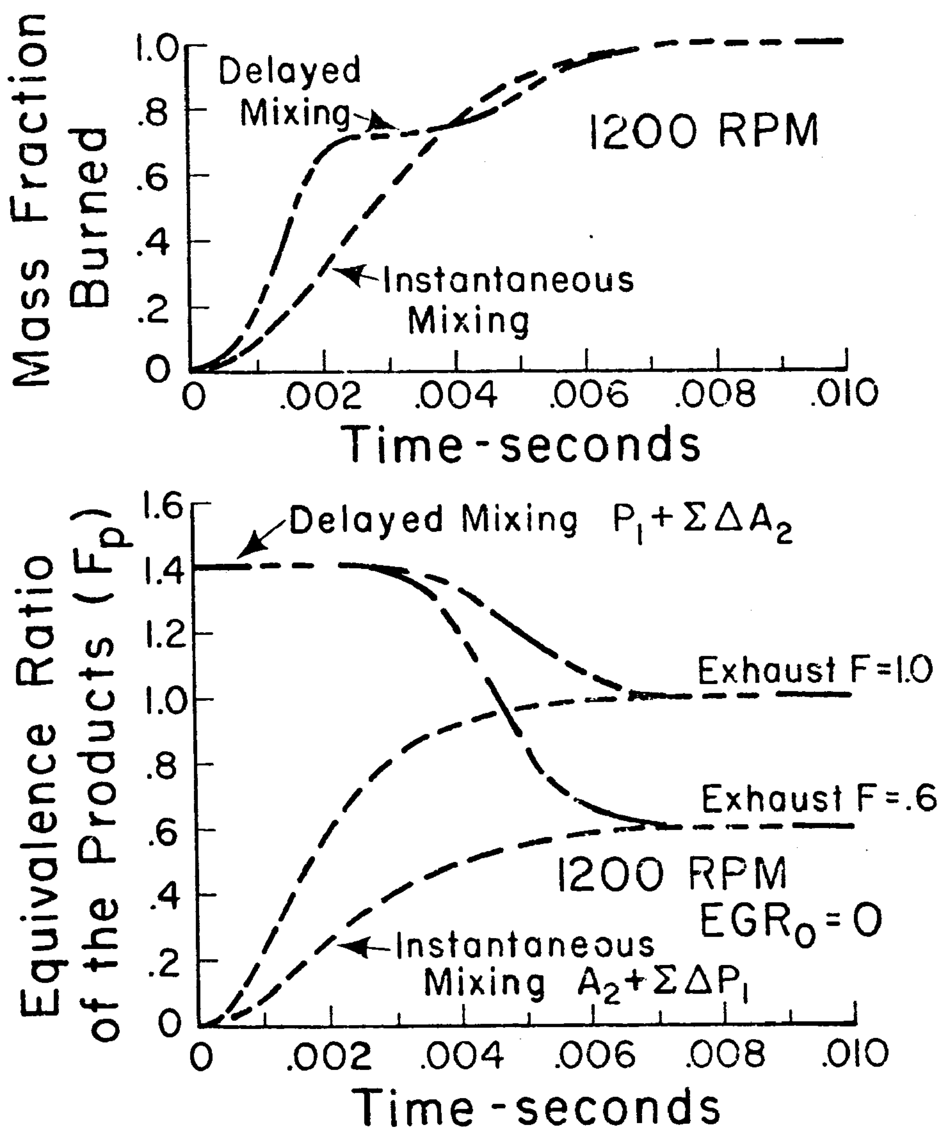


Figure 8 The Mass Fraction Burned and the Product Mixture Equivalence Ratio (for Two of the Nine Conditions) Corresponding to Figure 9 and Figure 10

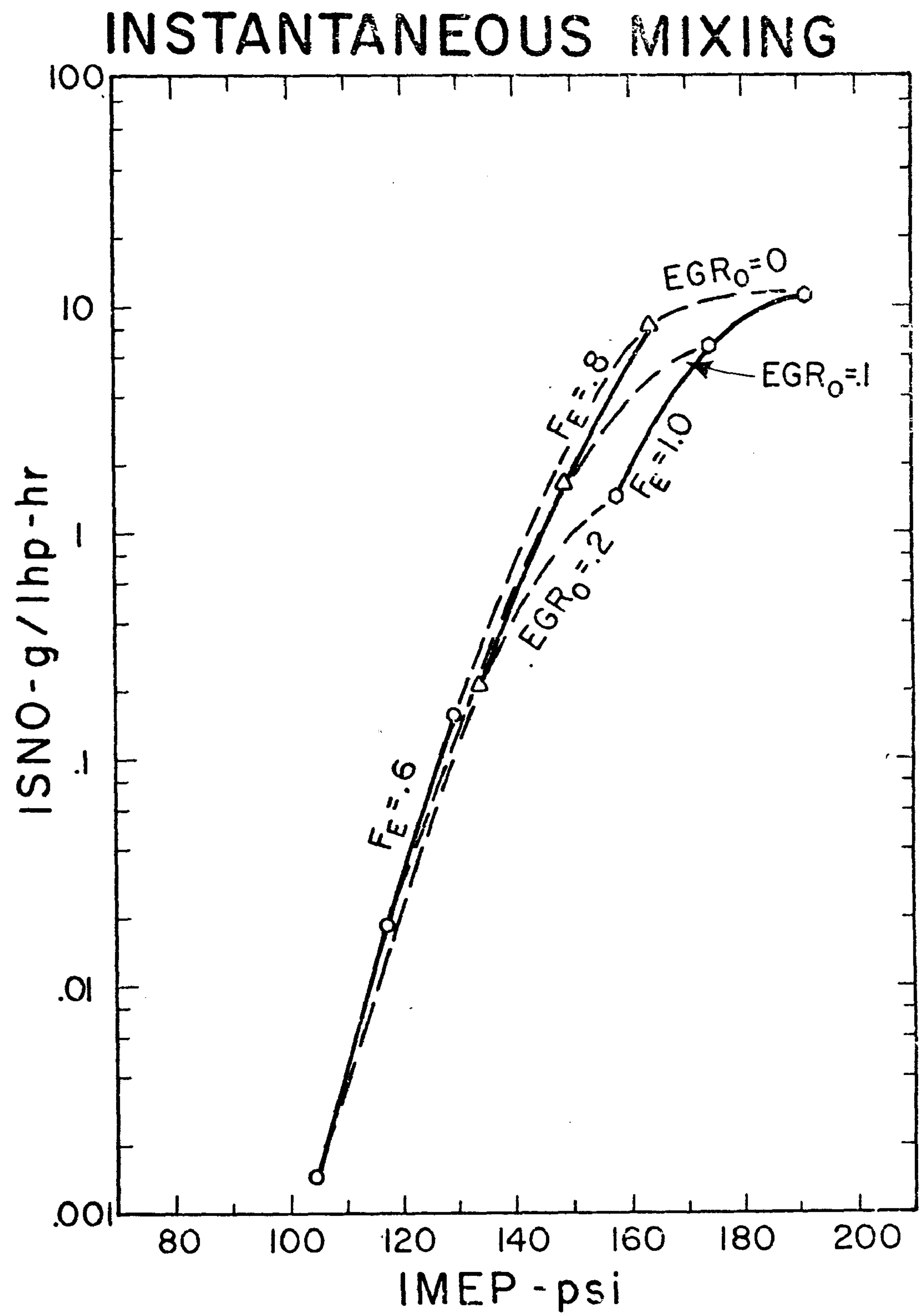


Figure 9 Indicated Specific Nitric Oxide for Instantaneous Mixing Stratified Charge Engine Configuration

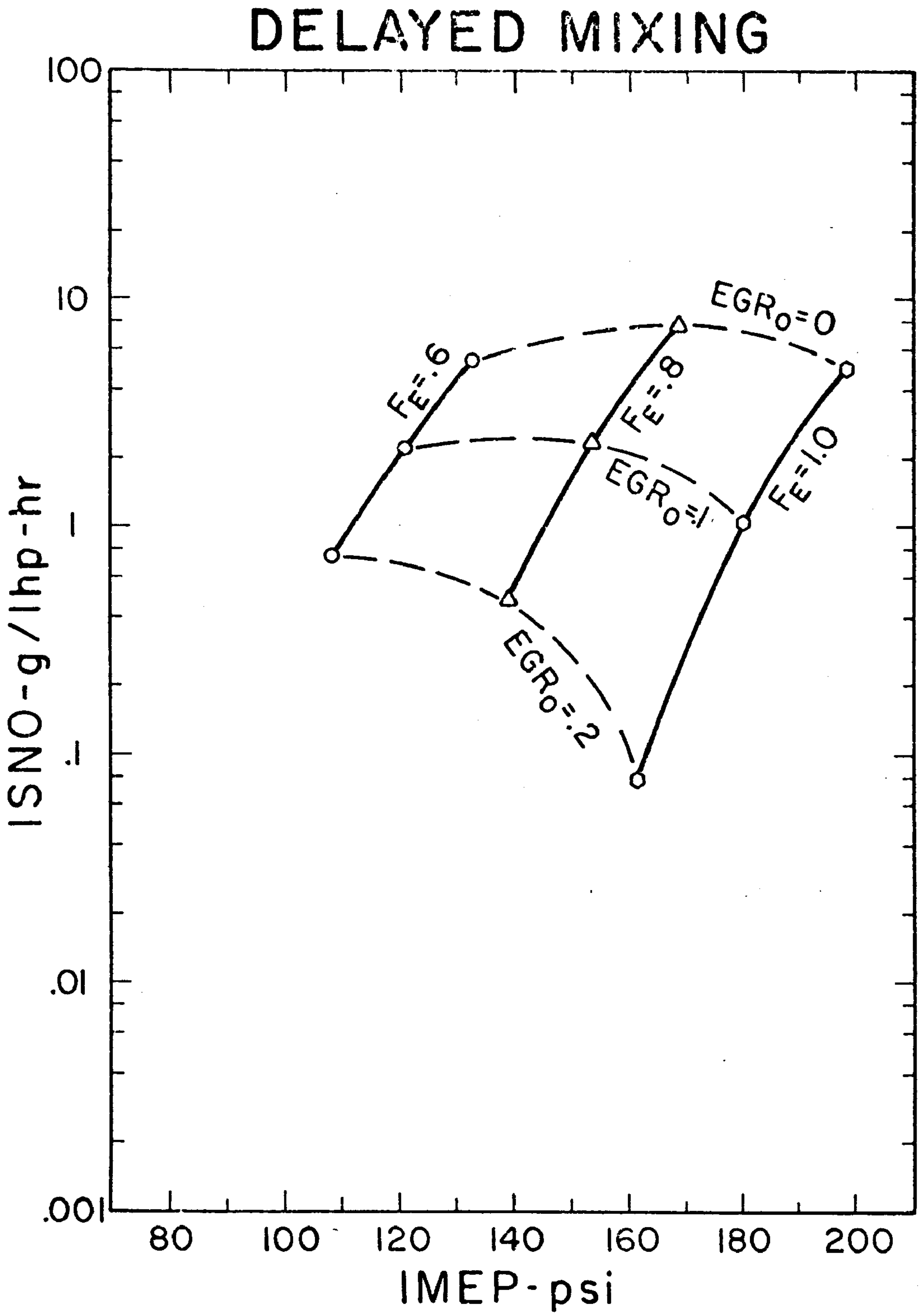


Figure 10 Indicated Specific Nitric Oxide for Delayed Mixing Stratified Charge Engine Configuration

the product mixture equivalence ratio for delayed and instantaneous mixing at two different exhaust equivalence ratios (is also called overall equivalence ratio) with no EGR. The resulting computed indicated specific nitric oxide for instantaneous mixing is shown in Figure 9 while ISNO for delayed mixing is shown in Figure 10.

The trends shown in Figure 9 for instantaneous mixing are obviously different from those for delayed mixing shown in Figure 10. These differences can readily be explained by recalling that, in general, the maximum nitric oxide is formed when the equivalence ratio is around F = .8. Thus, if the equivalence ratio of the product mixture is required to pass through equivalence ratios near F = .8, the nitric oxide levels will be high. If the F = .8 region can be avoided at least until expansion has cooled the products, the nitric oxide levels will be low. For example (as can be seen in the lower part of Figure 8), with an exhaust equivalence ratio of 0.6, the product mixture $[A_2 + \Sigma\Delta P_1]$ for instantaneous mixing never passes through a fuel-air ratio having rapid NO_x formation rates, i.e., it never exceeds F = 0.6. By contrast, for delayed mixing with an exhaust equivalence ratio of 0.6, the product mixture $[P_1 + \Sigma\Delta A_2]$ passes through the fuel-air ratio having rapid NO_x formation, i.e., F = 0.8 region prior to reaching its final exhaust value of F = .6. Consequently, as shown in Figure 10, exhaust NO_x concentrations with delayed mixing increase with leaner exhaust mixtures rather than decreasing as in the case of instantaneous mixing, the exact trend depending upon the amount of EGR. This effect is further accentuated by the rich fuel-air mixture being diluted with the recirculated exhaust gases having a lean equivalence ratio corresponding to the exhaust equivalence ratio. The resulting products have an equivalence ratio that is closer to stoichiometric than the rich air-fuel mixture. It should be clear, however, that regardless of how rich the fuel-air mixture is made $[P_1 + \Sigma\Delta A_1]$ must pass through the rapid NO_x formation region, if the exhaust is to have lean products composition.

In order to obtain a direct comparison between instantaneous mixing and delayed mixing, the data presented in Figures 9 and 10 are replotted by equivalence ratio and EGR as shown in Figures 11 and 12, respectively. Also included in Figures 11 and 12 is a homogeneous combustion case which corresponds to the same time of combustion and has a heat release characteristic identical to that of instantaneous mixing.

Looking first at Figure 11 for the exhaust equivalence ratio of F = 1.0, delayed mixing has the lowest ISNO and, by a slight amount, the greatest IMEP. The slightly greater IMEP is due to the earlier heat release of delayed mixing.

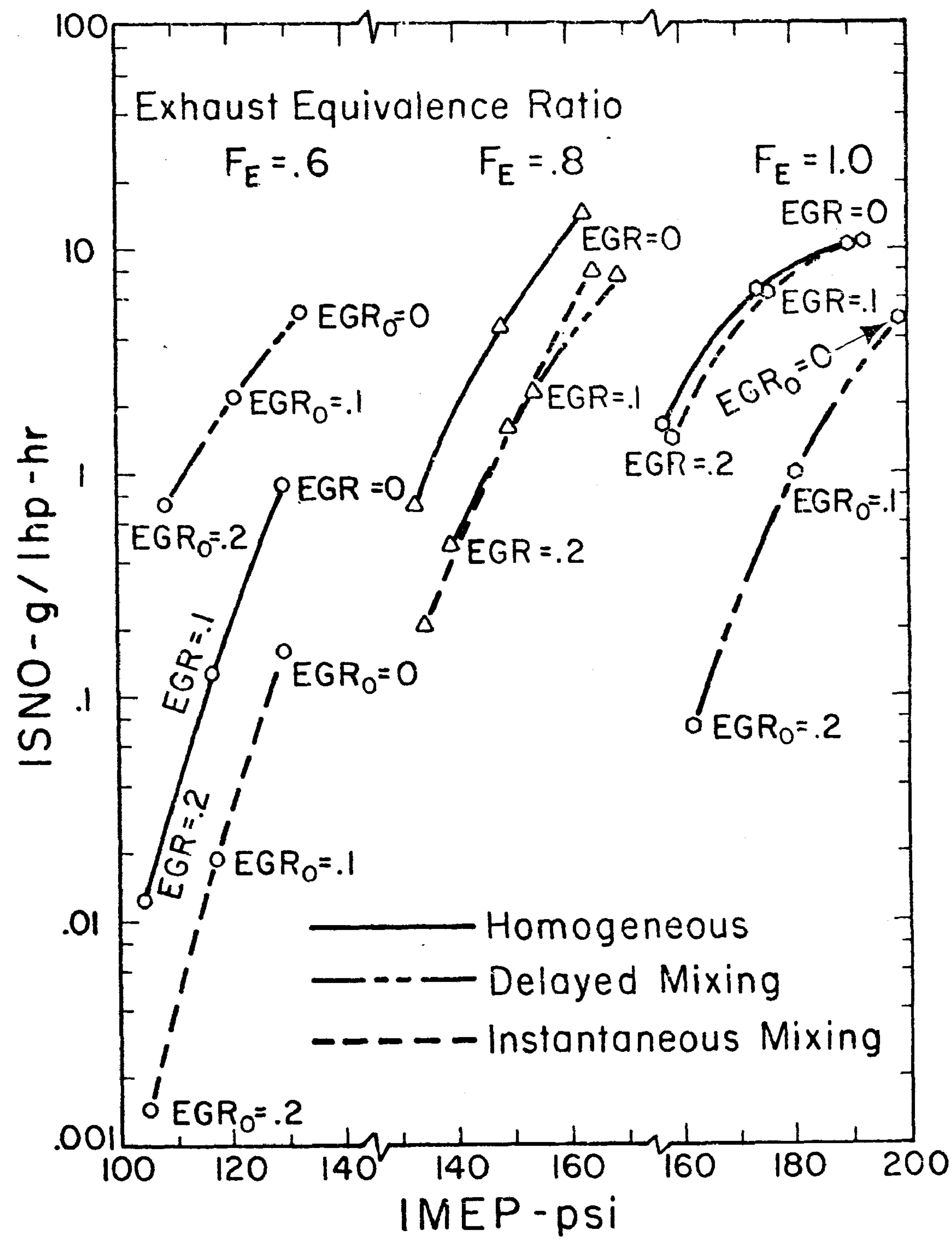


Figure 11 A Comparison Between Delayed Mixing, Instantaneous Mixing and Homogeneous Combustion by Constant Equivalence Ratio

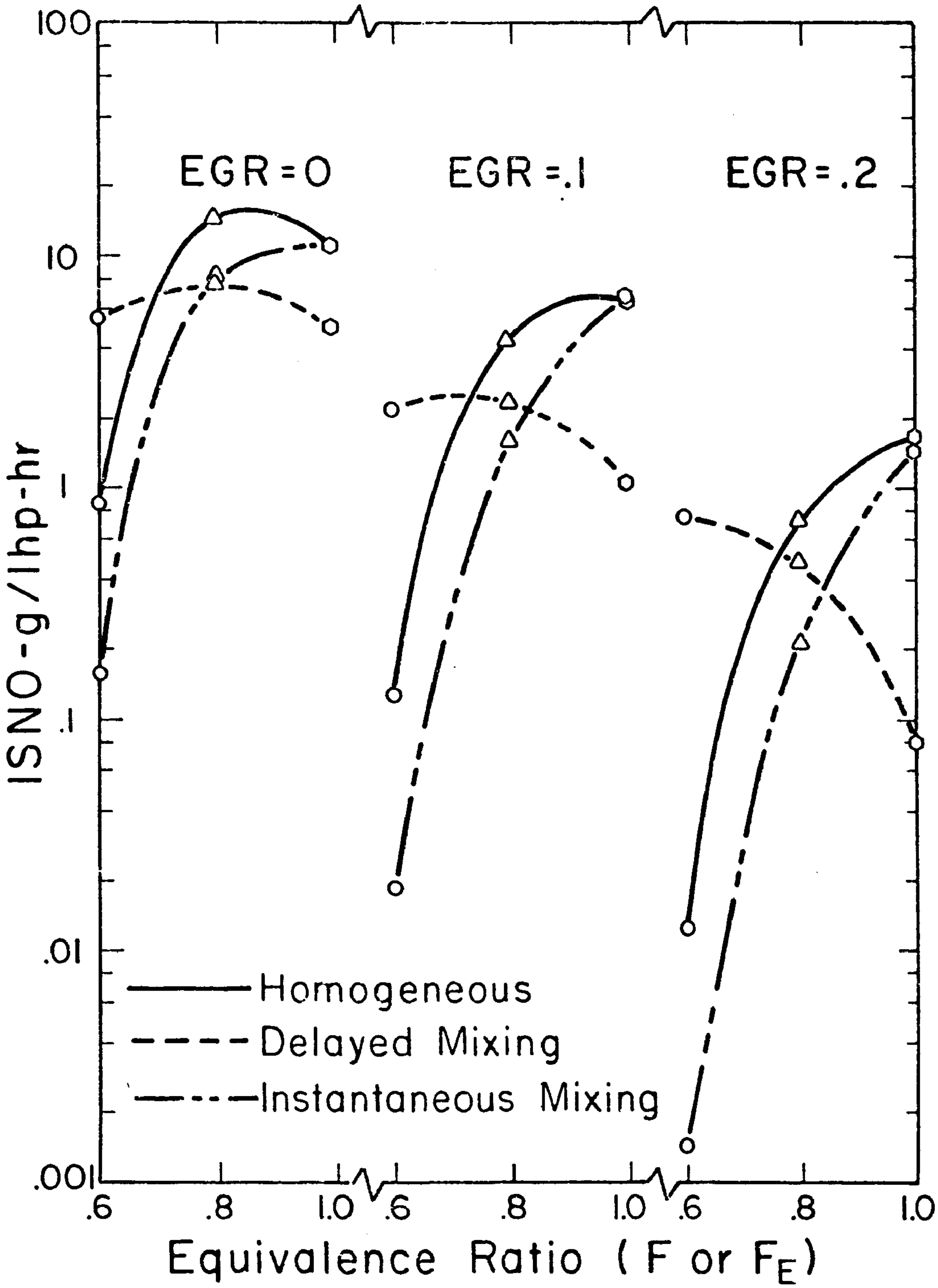


Figure 12 A Comparison Between Delayed Mixing, Instantaneous Mixing and Homogeneous Combustion by Constant EGR

The fact that the IMEP is slightly greater for the instantaneous mixing than for the homogeneous case is due to lower temperatures of the product mixture for instantaneous mixing and therefore less dissociation although the heat release curves are identical. At an exhaust equivalence ratio of F = 0.8, the delayed mixing and instantaneous mixing predict about the same results since they are both required to finally arrive at the high nitric oxide formation region. From an NO_x standpoint, instantaneous mixing is by far the most desirable at an exhaust equivalence ratio of F = 0.6, and even homogeneous combustion is better than the delayed mixing, since it is not required to pass through the region of F = 0.8. In summary, at stoichiometric mixtures, delayed mixing gives lower NO_x for about the same IMEP than instantaneous mixing. However, as the exhaust mixture is made leaner the situation reverses until, for an exhaust equivalence ratio of F = 0.6, the instantaneous mixing gives NO_x several orders of magnitude lower than delayed mixing. This effect is shown more dramatically in Figure 12 where the difference in trends with equivalence ratio is clearly illustrated. It can also be seen in Figure 12 that all three curves have different shapes at different values of EGR. For example, with zero EGR the delayed mixing curve is nearly horizontal while with 0.2 EGR it has a large negative slope. Note that the shape of the homogeneous curve is also changed significantly.

A comparison of the efficiencies of delayed mixing, instantaneous mixing,and homogeneous combustion is given in Figure 13. The instantaneous mixing and homogeneous combustion cases show about the same variation because they both have identical heat release functions, but the efficiency is slightly higher for the instantaneous mixing because of lower specific heats and less dissociation. The delayed mixing is somewhat more efficient because more of its heat release comes earlier in the expansion.

Figures 14 and 15 can be viewed as the potential engine combustion configurations for an arbitrarily specified maximum permissible value of ISNO. The logical choices would be those configurations having the highest IMEP and efficiency, i.e., the ones nearest the upper right hand corner. Associated with each point is a group of symbols which specify delayed mixing (D), instantaneous mixing (I), or homogeneous combustion (H) engine configurations. The equivalence ratio (F) for homogeneous combustion or the average overall exhaust equivalence ratio (F_E) for stratified charge engine configurations is included. Finally the amount of recirculated exhaust gas (EGR) for homogeneous combustion or the average overall amount of recirculated exhaust gas (EGR_0) for stratified charge engines is also shown.

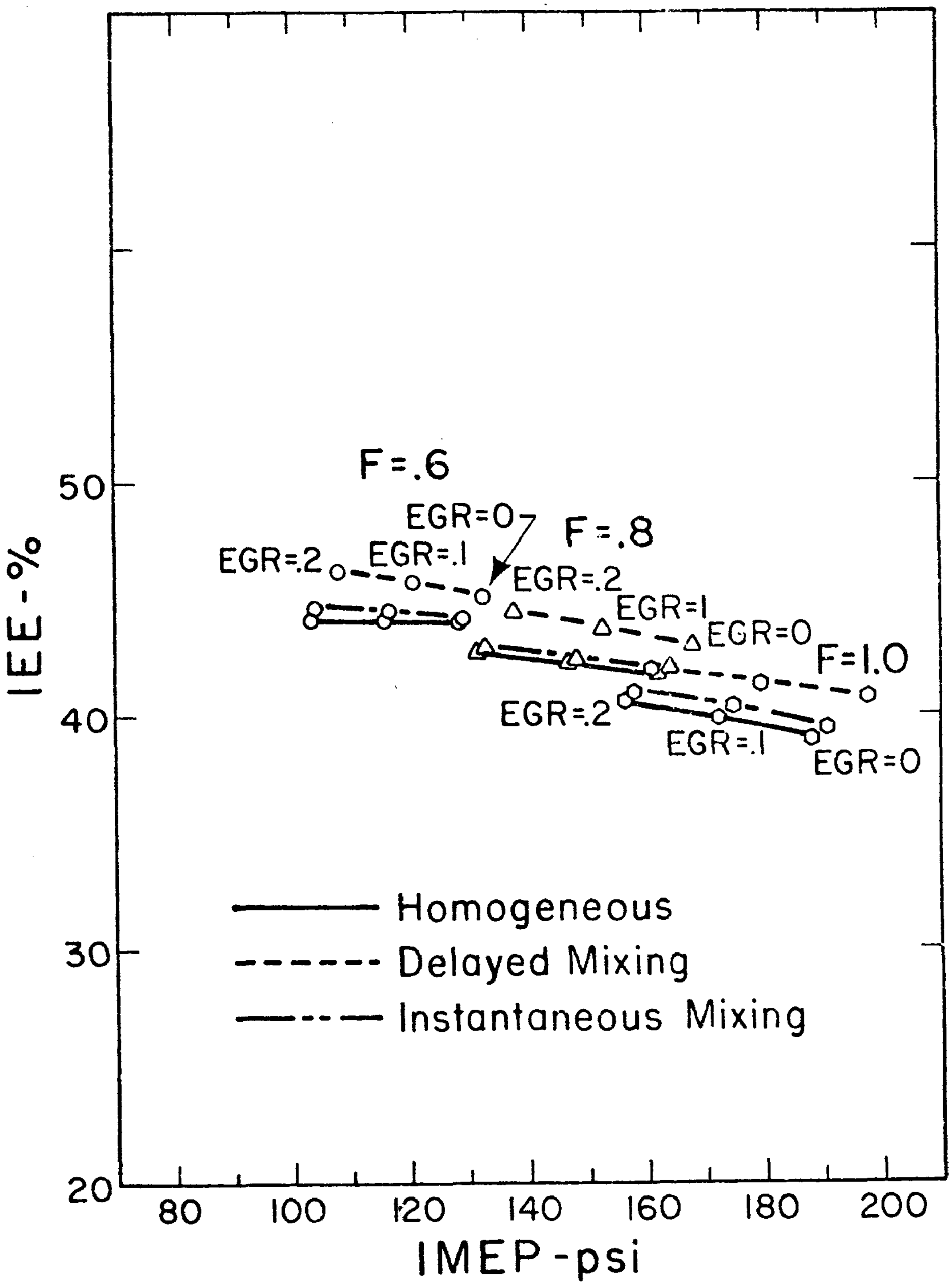


Figure 13 A Comparison of the Indicated Enthalpy Efficiency of Delayed Mixing, Instantaneous Mixing and Homogeneous Combustion

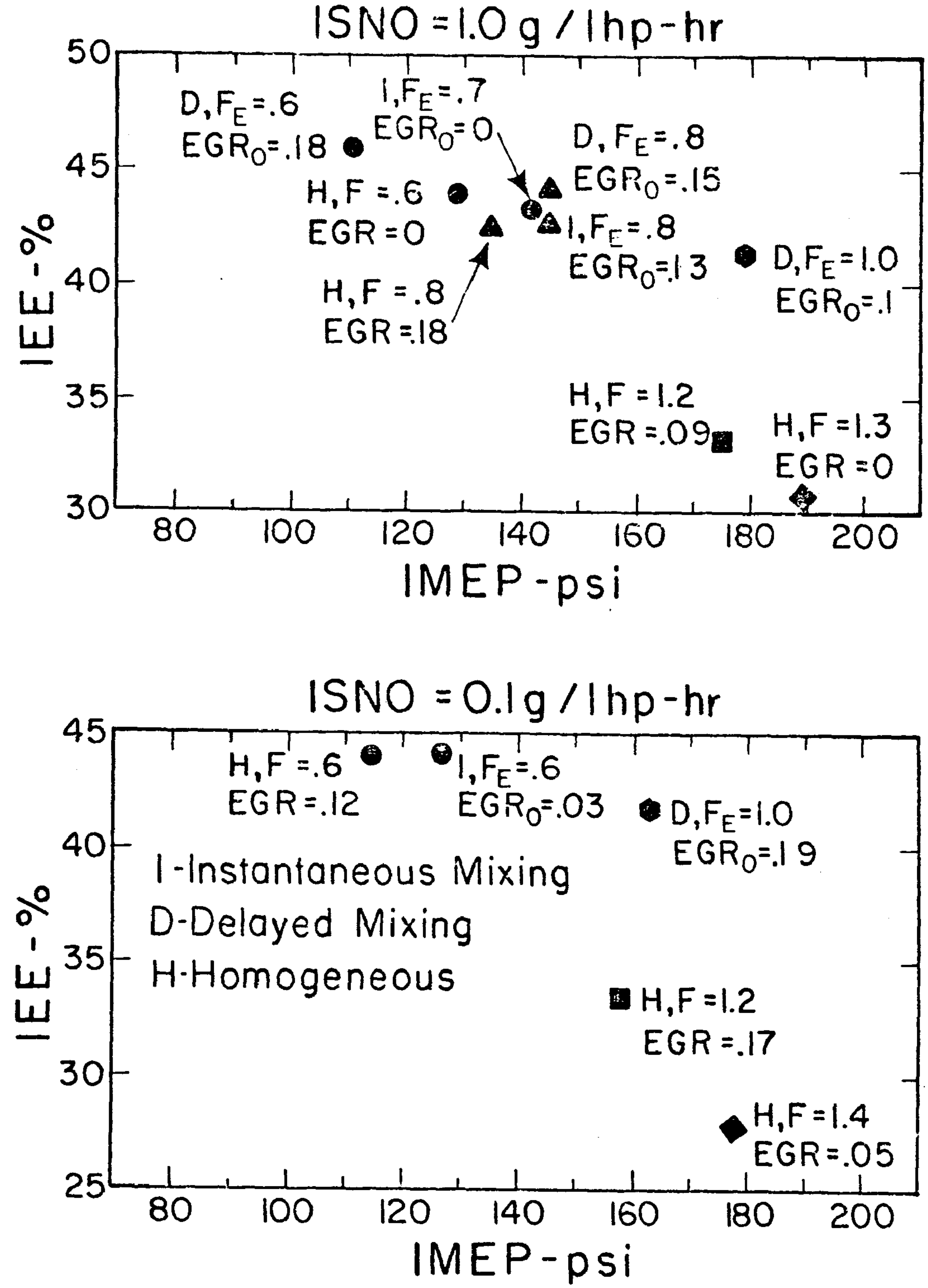


Figure 14 The Engine Combustion Configuration Meeting an Indicated Specific Nitric Oxide Level of 1.0 and 0.1 g/I hp-hr

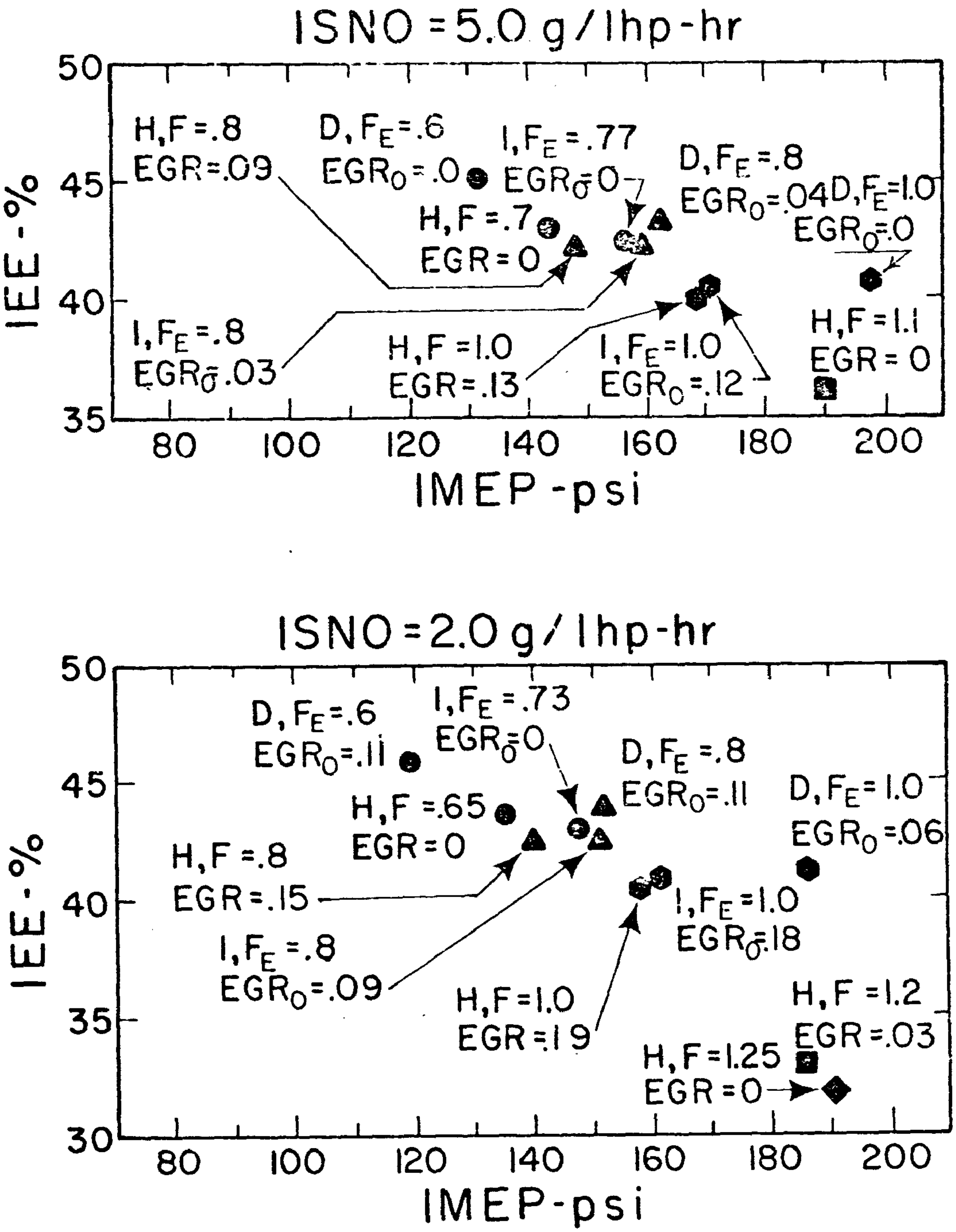


Figure 15 The Engine Combustion Configuration Meeting an indicated Specific Nitric Oxide Level of 5.0 and 2.0 g/Ihp-hr

At the very low value of ISNO = 0.1, only a few of the combustion configurations can be used, as seen in the lower part of Figure 14. The delayed mixing looks attractive because of its reasonably high IMEP and efficiency. Both lean homogeneous combustion and lean instantaneous mixing have slightly greater efficiencies but substantially lower IMEP's. In contrast, the rich homogeneous combustion for the H, F = 1.4, EGR = .05 combustion configuration has a slightly larger IMEP but a considerably lower efficiency. These results emphasize the basic limitation of the homogeneous charge engine with regards to the formation of nitric oxide. When operated rich, the efficiency is low and when operated lean, the IMEP is low.

As the limit of ISNO is raised to 1.0, 2.0 and 5.0 as shown in the top of Figure 14 and in Figure 15, the choice of configurations increases. The stoichiometric delayed mixing stratified charge engine configuration still appears to offer the best compromise of efficiency and IMEP in comparison with homogeneous combustion configuration. At these levels the very lean delayed mixing case (D, F_E = .6) appears to be a poor choice because of its very low IMEP. Note that points representing an equivalence ratio of 0.8 cluster in the center of the figures. For this 0.8 value, delayed mixing is slightly better than instantaneous mixing which in turn is slightly better than homogeneous combustion.

Homogeneous combustion and instantaneous mixing cases with an overall stoichiometric mixture become permissible at ISNO = 2.0 and 5.0. Their efficiency is about the same as the delayed mixing stoichiometric case but their IMEP is less.

All of the above comparisons were made at conditions analogous to wide open throttle. Comparisons at lighter loads depend upon the type of load control used, i.e., throttling versus overall fuel-air equivalence ratio control. Note that as the overall equivalence ratio is changed, the ratio of the volume of the fuel-air mixture to the volume of air must change. When the equivalence ratio is reduced for the instantaneous mixing case the ISNO is reduced along with the IMEP as shown in Figure 9. If the instantaneous mixing stratified charge engine were to be controlled by changing the overall equivalence ratio, it would have the advantage of reduced emissions at reduced powers. In contrast, the delayed mixing stratified charge engine controlled by changing equivalence ratio would have small changes in ISNO with no exhaust gas recirculation (See Figure 10). However when EGR = .2 the ISNO would increase as power (equivalence ratio) decreased. At low ISNO levels the delayed mixing engine configuration would have to operate near stoichiometric with considerable EGR. Consequently, throttling would be the only feasible method of controlling engine power under these conditions.

In summary, the delayed mixing stratified charge engine configuration operated with an overall mixture near stoichiometric is the best compromise of efficiency and IMEP at moderate to low levels of ISNO. However, the method of load control may depend upon the level of ISNO permitted.

E. CONCLUSIONS

1. The conventional homogeneous charge engine, by operating either very fuel-rich or very fuel-lean, can limit the nitric oxide emissions.

2. Exhaust gas recirculation is an effective way of reducing the nitric oxide emissions from either a homogeneous or stratified charge engine.

3. In the delayed mixing stratified charge concept (rich combustion followed by the addition of air to the rich products), extending the period during which the air is mixed into the rich products greatly reduces the nitric oxide level and also reduces the power.

4. For the delayed mixing stratified charge concept, the nitric oxide levels are low when the exhaust gas recirculation is high and the exhaust equivalence ratio is near stoichiometric.

5. For the instantaneous mixing stratified charge concept (products of rich combustion as they are formed are instantaneously mixed with the air and product mixture), the nitric oxide levels are low when the exhaust gas recirculation is high and the exhaust equivalence ratio is fuel-lean.

6. If an arbitrary level of nitric oxide were specified, the various engine configurations would compare as follows:

Configuration	Exhaust Eq. Ratio	Efficiency	IMEP
Homogeneous Charge	Rich	Low	High
Homogeneous Charge	Lean	High	Low
Instantaneous Mixing	Lean	High	Low
Delayed Mixing	Stoich.	High	High

7. The delayed mixing stratified charge engine concept has considerable promise because of its reasonably high efficiency and IMEP associated with low nitric oxide levels.

CHAPTER III

A DISCUSSION OF NITROGEN OXIDE EMISSIONS FROM VARIOUS TYPES OF INTERNAL COMBUSTION ENGINES

A. INTRODUCTION

Various types of internal combustion engines will be discussed and compared on the basis of their nitrogen oxide emissions versus overall fuel-air ratio. It will be shown that, in general, engines can be divided into two groups with markedly different responses of nitrogen oxide emissions with changes in the overall fuel-air ratio.

The first group is characterized by producing the most nitrogen oxide emissions (either ppm or g/hp-hr) near stoichiometric fuel-air ratios or slightly fuel lean. This group consists of homogeneous charge engines, rapid mixing stratified charge engines, three-valve stratified charge engines and open chamber diesels. In the first group the fuel is distributed throughout the air before combustion or the products of rich combustion are rapidly mixed with the lean ratios.

The second group is characterized by producing the most nitrogen oxide emissions (either ppm or g/hp-hr) at leaner fuel-air ratios and being less sensitive to changes in the fuel-air ratio. This group consists of the limited mixing stratified charge engine and the divided chamber diesels. In the second group of engines mixing is temporarily limited between the rich products of combustion and the lean region.

The two different groups can be explained on the basis of the fundamental concepts of nitric oxide chemical kinetics. Nitrogen oxides are formed in the products of combustion because these are formed at a sufficiently high temperature and have oxygen available. The nitric oxide chemical kinetics are slowed down by cooling associated with the expansion process of an engine. Thus nitric oxide active chemical kinetic period starts with combustion and ends during expansion. The availability of oxygen is primarily a function of the local fuel-air ratio (or equivalence ratio). The temperature of a local region of products is a function of the flame temperature, the amount of compression or expansion, the heat transfer, mixing with other regions, and chemical reaction due to the mixing of regions with different chemical composition. The fuel-air ratio is important to the temperature of the products because it influences the flame temperature and the chemical reaction resulting from mixing. It is not surprising that the fuel-air ratio is a significant parameter since it affects both the available oxygen and the product temperature. The pressure also affects the nitric oxide formation rate to a small extent and since the pressure effect is small it will not be included in the discussion.

The final concentration of nitric oxide is primarily a result of the temperature and equivalence ratio history of the products of combustion.

B. COMBUSTION CONCEPTS OF STRATIFIED CHARGE ENGINES

In many stratified charge engines a fuel injection system is used to form a rich combustion region. The lean region is initially composed of residual product from the previous cycle plus the fresh charge of air or air with EGR. The difference between the various types of these stratified charge engines is in the processes used to complete the combustion when the rich products are mixed with the lean region. Conceptually these engines can be divided into two categories, those which rapidly mix the rich product of combustion with the lean region and those which limit the mixing of the rich products with the lean region.

The difference between simplified examples of the rapid mixing and limited mixing stratified charge engines is illustrated in Figure 16. The generalized combustion schematics shown in Figure 16 assume that the combustion chamber can be divided into a few regions of uniform properties. The percentage of the mass in the various regions is shown as a function of the percentage of the time of heat release.

Figure 16a represents rapid mixing of the products of rich combustion (P_1). As soon as a small element is

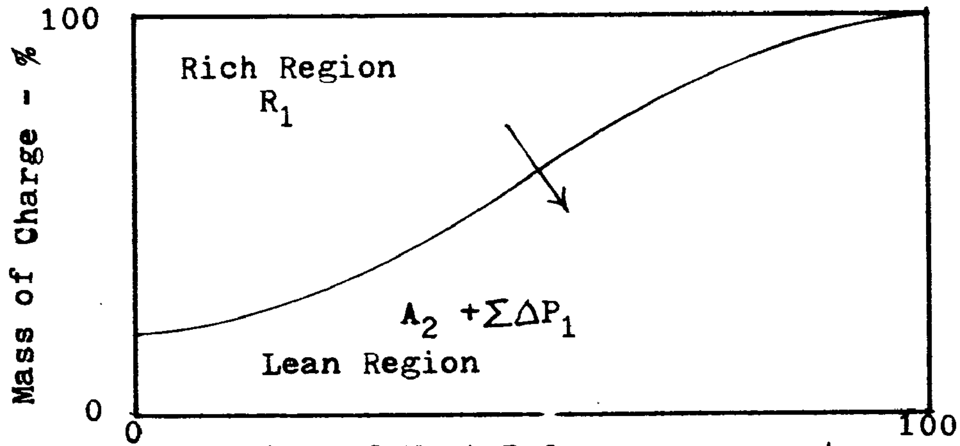


Figure 16a, Simplified Rapid Mixing Combustion Process (Instantaneous Mixing, Chapter II)(R;P-A)

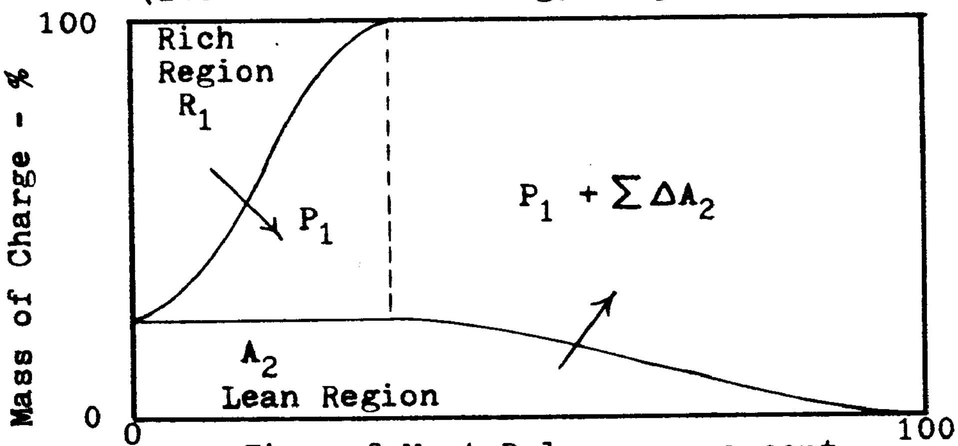


Figure 16b, Simplified Limited Mixing Combustion Process (Delayed Mixing, Chapter II)(L;A-P)

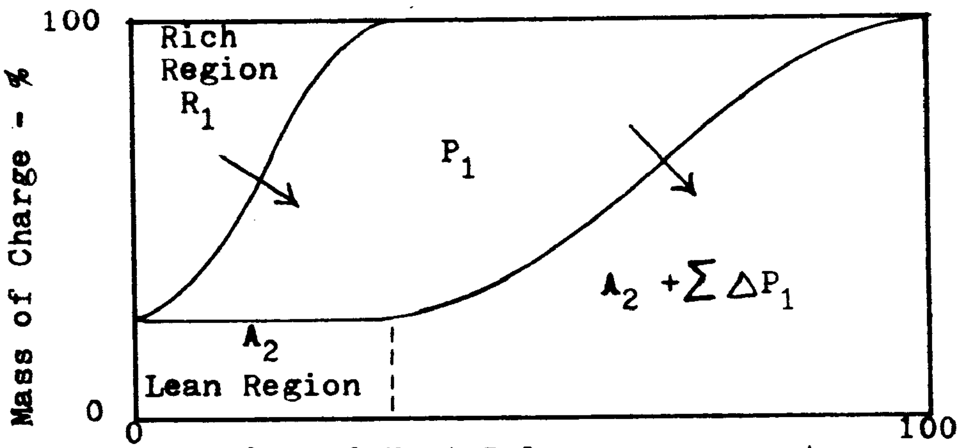


Figure 16c, Simplified Limited Mixing Combustion Process (L;P-A)

Figure 16 Simplified Combustion Process Schematics burnt it is thoroughly mixed throughout the lean product mixture to give at any instant $A_2 + \Sigma\Delta P_1$. Initially the product mixture consists of residual products plus air or air with EGR (A_2). This process is symbolized by R;P-A for the rapid mixing of the rich products into the lean product mixture (or lean region). The combustion processes shown in Figure 16a are similar to the theoretical instantaneous mixing concept discussed in Chapter II and represent a simplification of the rapid mixing group of stratified charge engines.

The combustion schematics presented in Figure 16b and Figure 16c represent two simplified versions of the limited mixing stratified charge engine group. Both versions start with a period of rich combustion with no mixing. If the lean region (A_2) is mixed into the rich products ($P_1 + \Sigma\Delta A_2$) the process is symbolized by L;A-P. This combustion process is similar to the delayed mixing concept discussed in Chapter II. The delayed mixing concept represents one simplified version of the limited mixing stratified charge engine group. If the rich products (P_1) are mixed into the lean region ($A_2 + \Sigma\Delta P_1$) the process is symbolized by L;P-A. This version of the limited mixing stratified charge engines was not discussed in Chapter II and cannot be simulated by the computer program.

The combustion schematics are presented as an aid to describing the more complicated combustion of real engines. The Texaco Controlled-Combustion System (TCCS) Engine is an operational version of a rapid mixing stratified charge (R;P-A) engine as are some open chamber diesels. The Ford Programmed Combustion (PROCO) Engine, the Newhall Engine and some divided chamber diesels are operational examples of limited mixing stratified charge engines (L;P-A).

It should be pointed out that the three-valve stratified charge engine which normally uses carburetion to distribute the fuel throughout the charge does not fit into the categories of rapid mixing or limited mixing stratified charge engines. Instead it is similar to a homogeneous engine with the small prechamber functioning as a method of charge ignition.

C. DEFINITIONS OF SOME TYPES OF INTERNAL COMBUSTION ENGINES

Listed below are definitions of some of the various types of internal combustion engines as used in this paper.

Internal Combustion Engine: An internal combustion engine is an engine in which the products of combustion are used directly as the working fluid, Obert (1968).

Homogeneous Charge Engine: A homogeneous charge engine is an internal combustion engine in which the design is intended to distribute all of the constitutents of the charge uniformly throughout the combustion chamber.

<u>Stratified Charge Engine</u>: A stratified charge engine is an internal combustion engine in which the design is intended to distribute the constituents of the charge nonuniformly throughout the combustion chamber. For example, the fuel can be arranged into fuel-lean and fuel-rich regions; the fuel-lean region can be air.

<u>Diesel Engine</u>: A diesel engine is a stratified charge engine which uses the high temperature of the compressed air to ignite the injected fuel.

<u>Open Chamber Engine</u>: An open chamber engine is an internal combustion engine which has a single combustion chamber.

<u>Divided Chamber Engine</u>: A divided chamber engine is an internal combustion engine in which the clearance volume is divided into two (or more) connected chambers. The chamber in which combustion is initiated is called the precombustion chamber or prechamber; the other chamber in which expansion occurs is called the main chamber.

<u>Three-Valve Stratified Charge Engine</u>: A three-valve stratified charge engine is a divided chamber spark ignition stratified charge engine which uses a third valve to admit a fuel-rich mixture near the spark plug in the small prechamber and uses the normal inlet valve to admit a fuel-lean mixture into the larger main chamber.

Rapid Mixing Stratified Charge Engine Group: The rapid mixing stratified charge engines are a group of stratified charge engines in which all of the fuel passes through the rich region and the products formed during the combustion initiated in the rich region are rapidly mixed with the lean region.

Instantaneous Mixing Stratified Charge Engine (R;P-A): An instantaneous mixing stratified charge engine is a theoretical version of the rapid mixing stratified charge engine group, in which the products formed during rich-region combustion are instantaneously mixed throughout the lean region.

Limited Mixing Stratified Charge Engine Group: The limited mixing stratified charge engines are a group of stratified charge engines in which combustion is started in the fuel-rich region, then by some means the mixing is temporarily limited between the rich products and the lean region. The lean region can be mixed into the rich products, the rich products can be mixed into the lean region or the rich products and the lean region can be mixed into each other.

Delayed Mixing Stratified Charge Engine (L;A-P): A delayed mixing stratified charge engine is a theoretical version of a limited mixing stratified charge engine group. Combustion is initiated and completed in the rich region and then the lean region is mixed into the rich products.

D. HOMOGENEOUS ENGINES

Most automobiles and many light and medium duty trucks are powered by homogeneous charge spark ignition engines which have open combustion chambers. Considerable research has centered on reducing NO_x emissions from these engines. Although many factors influence NO_x emissions fuel-air ratio is the most significant. Huls (1966) measured the emission from a homogeneous charge engine. A typical curve from his thesis is reproduced in Figure 17 and shows the variation of emissions with air-fuel ratio. The maximum NO_x emissions occur at a slightly lean fuel-air mixture. At these conditions the two important factors in nitric oxide formation, high temperature and availability of oxygen, are in optimum proportions to produce maximum NO_x emissions. When richer fuel-air mixtures are used the lack of oxygen limits nitric oxide formation. When leaner fuel-air mixtures are used the lower temperatures limit nitric oxide formation.

The NO_x emissions of homogeneous engines can be reduced by operating either very rich or very lean. Rich operation results in higher levels of hydrocarbons and carbon monoxide plus lower efficiencies. For these reasons very rich operation is not desirable. A lean burning

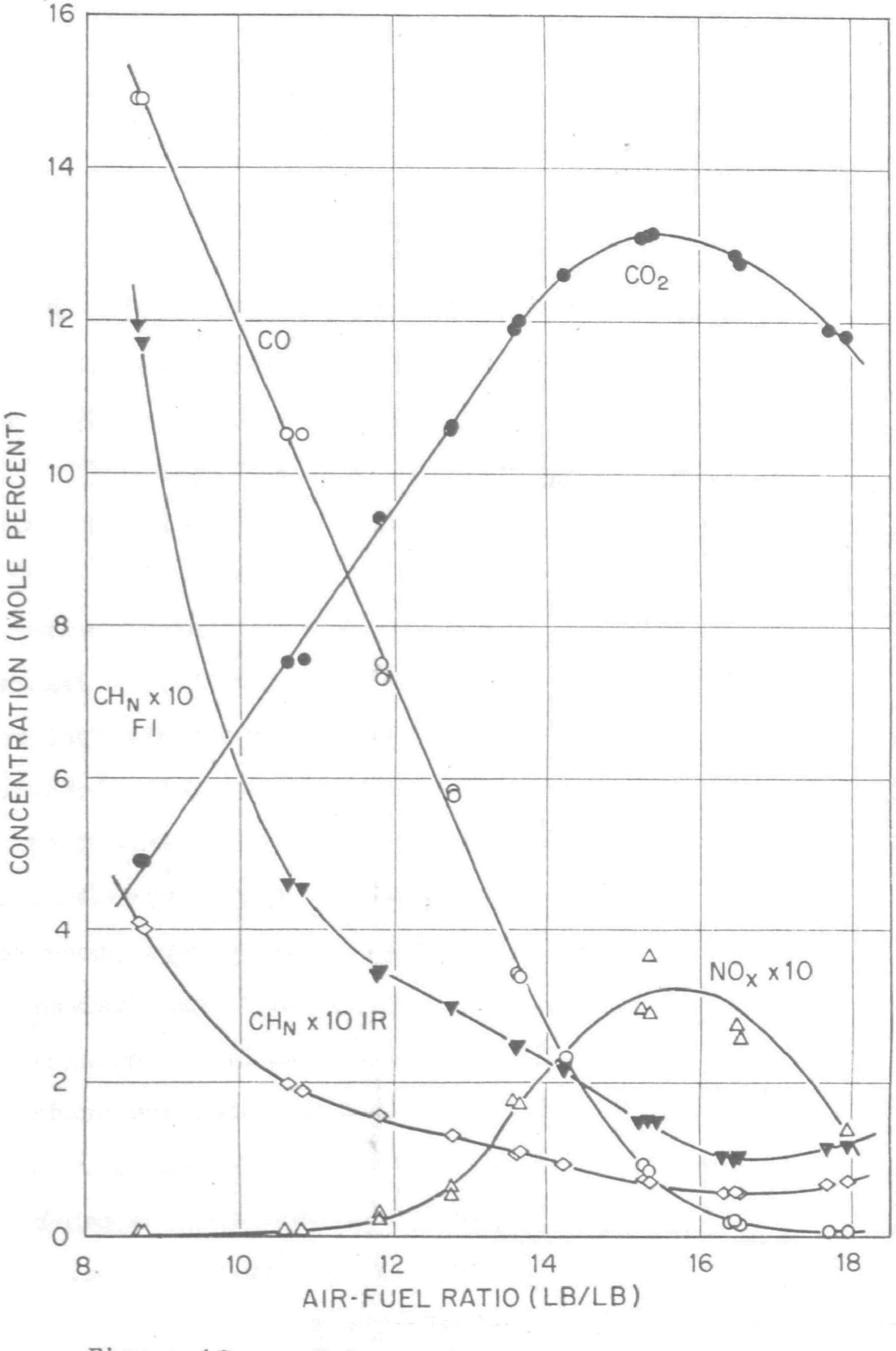


Figure 17 Exhaust Composition versus Air-Fuel Ratio for 15° BTDC Spark Timing From Huls, T. A. (1966)

homogeneous engine has the advantage of lower hydrocarbons, lower carbon monoxide and higher efficiency but has the disadvantage of lower specific power, lower flame speed and difficulty with ignition. When attempts are made to operate even leaner a point is reached called the lean limit where the engine will start to misfire.

Recent developments in spark timing control and carburetion described by John (1975) and by Adams (1976) have allowed homogeneous engines to operate at lower fuel-air ratios. The "lean-burning engines" have lower nitrogen oxide emissions, lower carbon monoxide emissions, lower hydrocarbon emissions (except at very lean operation) and higher efficiency due to more complete combustion. The disadvantage is their lower specific power.

E. THREE-VALVE STRATIFIED CHARGE ENGINES

The three-valve stratified charge engine is the most investigated form of the divided chamber spark ignition stratified charge engine. It has a long history with the initial patent being issued to H. R. Ricardo in 1918. A thorough presentation of evolvement, analysis and progression of three-valve stratified charge engines is presented by Turkish (1974).

A fuel-rich charge is drawn into a small prechamber through the third valve and a lean charge is drawn into the main combustion chamber through the normal inlet valve.

Combustion is initiated by the spark plug located in the prechamber. As pressure builds up in the prechamber it forces high temperature rich products of combustion out through an orifice into the main combustion chamber. The hot products initiate combustion of the lean mixture in the main chamber. Nitrogen oxide produced in the prechamber is low because of its rich charge and its small percentage of the total charge. The low temperature associated with lean combustion of the main chamber charge limits the main chambers nitrogen oxide contribution.

The three-valve engine could be regarded as a version of a homogeneous engine since most of the charge is in the main chamber and the charge in the main chamber is roughly homogeneous.

The Honda CVCC engine, a three-valve stratified charge engine, clearly demonstrated the potential for the low NO_x emissions by meeting the original 1975 United States and 1975 Japanese emissions standards, Tasuku Date et al. (1974). Honda's approach has been to optimize experimentally and analytically the prechamber volume, the orifice size and the fuel distribution as is described by Yasuo Shizuo et al. (1974).

The essential aspect of performance of all three-valve stratified charge engines, that accounts for the low NO_x emissions, is the ability to operate at very lean overall fuel-air mixtures in the main chamber. When a three-valve engine is compared to a similar conventional homogeneous engine as was done by Tasuku Date et al. (1974) of Honda, by Davis et al. (1974) of General Motors, by Purins (1974) of Ford and by Yasuo Sakai (1974) of Nissan the results are essentially the same. Davis states, "At lean overall mixtures, both the Jet Ignition Stratified Charge Engine and the conventional spark ignition engines have similar HC, CO, and NO_x emission characteristics when overall air-fuel ratio is changed." Tasuku Date, Yasuo Sakai and Purins presented curves which showed a direct comparison between their type of three-valve engines and a similar homogeneous engine. These results of Tusuku Date and Yasuo Sakai are reproduced in Figures 18 and 19. The lowest nitrogen oxide emissions correspond to leanest operation not normally obtainable by conventional homogeneous engines.

From Figures 18 and 19 it is apparent that the peak nitrogen oxide emissions occur at approximately the same overall air-fuel ratio for both the homogeneous and three-valve engines. The nitrogen oxide peaks coincide because most of the charge is in the main chamber and is somewhat homogeneous. The small amount of charge in the prechamber mixes rapidly with the main chamber charge and represents a convenient method of ignition.

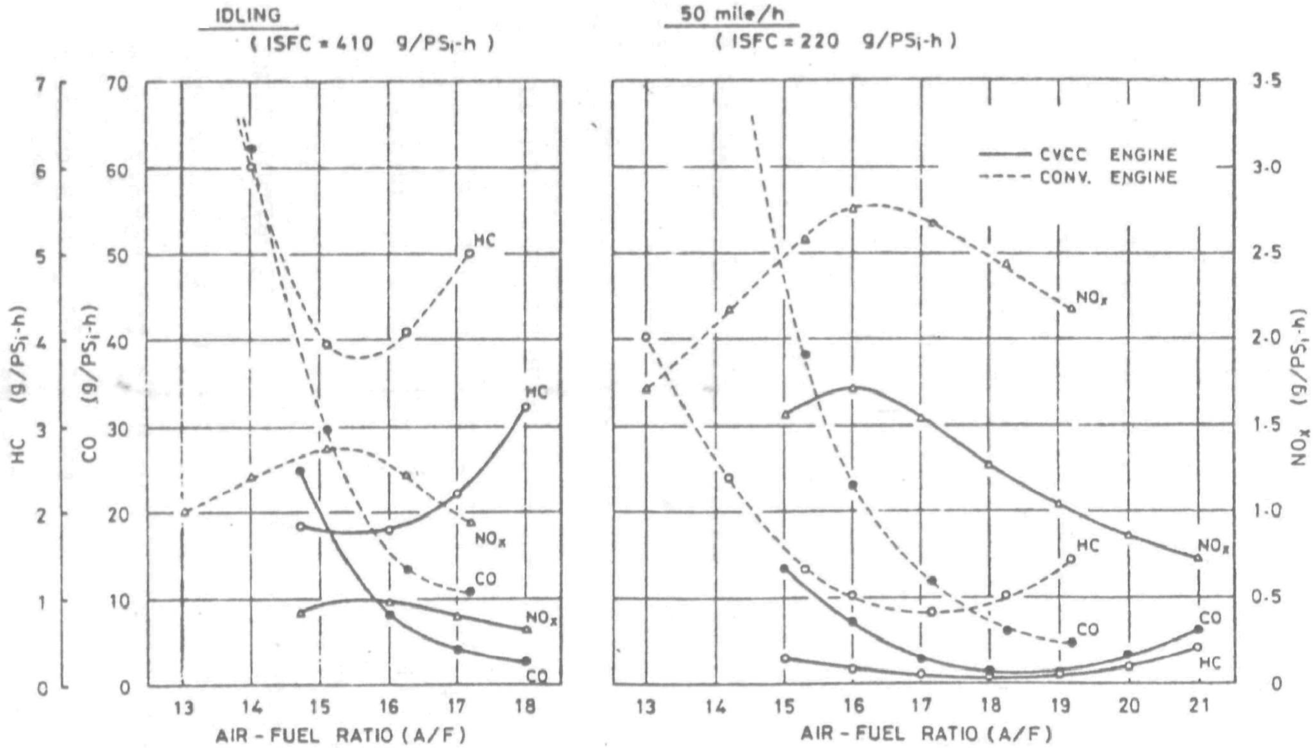


Figure 18 Comparison of Exhaust Emission of CVCC Engine with Conventional Engine From Tasuku Date et al. (1974)

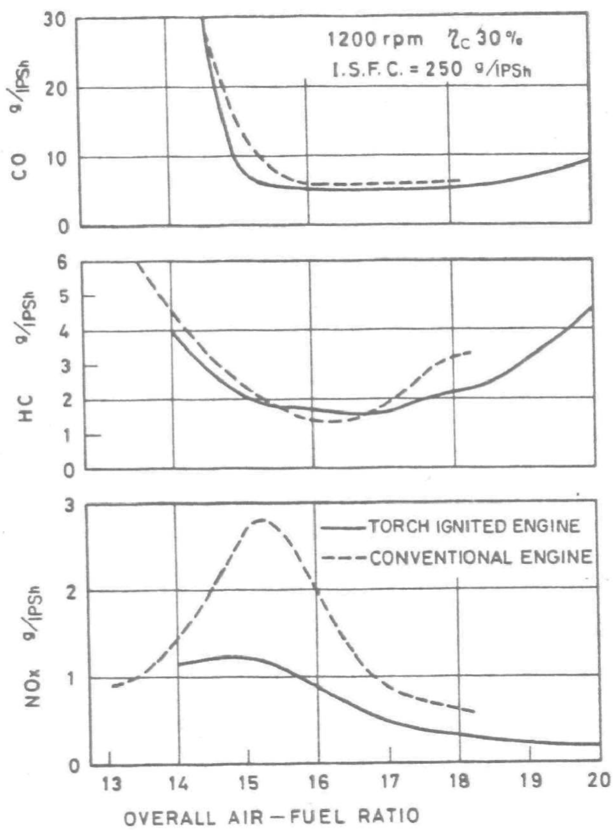


Figure 19 Comparisons of Exhaust Emissions at Constant Indicated Specific Fuel Consumption
From Yasuo Sakai (1974)

F. RAPID MIXING STRATIFIED CHARGE ENGINES

1. Instantaneous Mixing Stratified Charge Engine Concept

The instantaneous mixing stratified charge engine concept described in Chapter II is a simplified theoretical version of the rapid mixing stratified charge engine group. It has the most nitric oxide emissions near stoichiometric fuel-air ratios as seen in Figure 12. The shape of the nitric oxide curve is similar to the homogeneous curve. The combustion process is similar to that described in Figure 16a, R;P-A.

The lean region is initially air or air with EGR (A_2) and it is at compression temperature. As combustion occurs the rich products which are formed are instantaneously mixed throughout the product region. The result of combustion is that the product mixture fuel-ratio and temperature increase toward those of the corresponding homogeneous case. The nitric oxide formed by the instantaneous mixing case will, as usual, depend on the time available, the product mixture temperature history and the product mixture fuel-air ratio history.

When the final mixture is stoichiometric the instantaneous mixing engine has nearly the same nitric oxide emissions as the homogeneous engine. Under these conditions some of the temperature history is sufficiently high to increase the nitric oxide formation rate and to allow the instantaneous mixing engine to form about the same amount of nitric oxide as the homogeneous engine. This process is assisted by the fact that the equivalence ratio of the product mixture passes through the rapid nitric oxide forming region, which is slightly fuel-lean.

For the other conditions shown in Figure 12 the instantaneous mixing engine produces less nitric oxide than the homogeneous engine. Under these conditions the temperature and the equivalence ratio of the instantaneous engine is always less than that of the homogeneous engine as is the resulting nitric oxide formation rate. The lower temperatures and leaner product mixture does not allow the instantaneous engine's nitric oxide concentration to reach the level of the homogeneous engine before the cooling associated with expansion stops the nitric oxide kinetics.

2. Texaco Controlled-Combustion System (TCCS)

Texaco Inc. has developed an open combustion chamber stratified charge engine which uses the Texaco Controlled-Combustion System (TCCS). In this engine air is caused to swirl in the combustion chamber. Fuel is injected into the swirling air and immediately ignited. A flame front is established that burns the fuel as fast as it is injected. The rich products of combustion are rapidly cooled by the excess air. The NO_x emissions as reported by Mitchell et al. (1972) are presented in Figure 20. His curves show the variation of NO_x emissions with changes in the engine IMEP. Although fuel-air ratios are not plotted directly the lower IMEP corresponds to leaner overall fuel-air ratios. Mitchell states, "Fuel combustion under rich mixture conditions limits formation of oxides of nitrogen in the flame zone. A quick quench of the burned gases by the cold excess air precludes further NO_x formation in these post-flame gases." This description along with the generally increasing nitrogen oxide concentration with IMEP implies that the TCCS engine could be considered a rapid mixing stratified charge engine. The rapid mixing stratified charge engine would be expected to have increasing nitric oxide emissions with increasing IMEP as does its theoretical counterpart the instantaneous mixing engine. See Figure 9 for the calculated results of the instantaneous engine.

G. LIMITED MIXING STRATIFIED CHARGE ENGINES

1. Delayed Mixing Stratified Charge Engine Concept

The delayed mixing concept of stratified charge engines is a theoretical version of the limited mixing stratified charge engine group which corresponds to the combustion process described in Figure 16b, L;A-P.

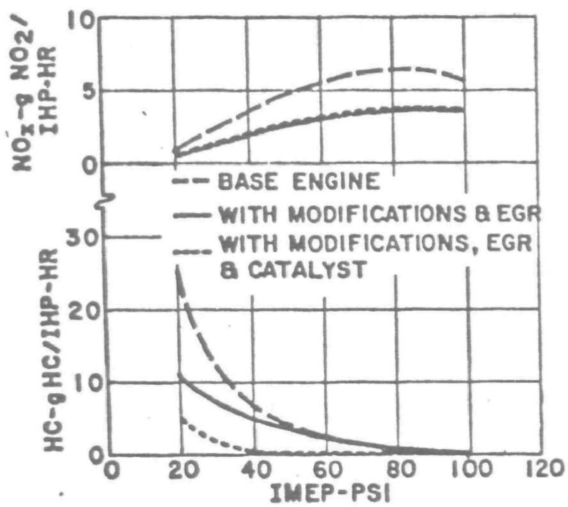


Figure 20 Single Cylinder Evaluations HC plus NO$_x$ Emissions From Mitchell et al. (1972)

All of the fuel is in the rich region which burns first. During the rich combustion little nitric oxide is formed because of the lack of oxygen even though the temperature of the rich products is quite high. When the rich combustion is finished, the lean region (air plus EGR) is added incrementally to the rich products. The incremental addition of the air continues the combustion process and keeps the product mixture temperature high. If the overall air-fuel ratio is lean the excess air will cause a reduction in the temperature of the product mixture.

The amount of nitrogen oxide which is formed with an overall stoichiometric mixture is less for the delayed mixing engine than the corresponding homogeneous engine burning a stoichiometric mixture as shown in Figure 12. The difference is primarily caused by the equivalence ratio of the delayed mixing engine being richer than that of the homogeneous engine. The product temperature for both engines would be similar. Expansion stops the nitric oxide kinetics before the nitric oxide concentration of the delayed mixing engine reached that of the homogeneous engine.

When the overall mixture of a delayed mixing engine is very lean the nitric oxide emissions from the delayed mixing engines are much larger than the instantaneous engine using a correspondingly lean mixture. The delayed mixing product mixture starts at a high temperature and is provided with abundance of oxygen which results in the high nitric oxide concentration. The expansion cools the product mixture and stops the nitric oxide kinetics before the reverse reactions can reduce the concentration. In contrast the homogeneous mixture is always at a low temperature which limits the nitric oxide formation.

If the delay period were to be extended to the end of the active nitric oxide kinetic period the nitric oxide emissions would correspond to that of a homogeneous engine operating at the rich equivalence ratio. In general, the delay period is not that long and some nitric oxide is formed due to mixing of the lean region with the rich products.

2. Ford Programmed Combustion (PROCO) Engine

Ford Motor Company has developed a type of open chamber stratified charge engine which is called the Ford Programmed Combustion (PROCO) Engine. Stratification is established by causing the air in the combustion chamber to have a high swirl and to spray the fuel into the central region of the swirl. A spark plug located near the fuel spray cone ignites the rich region. Lavoie and Blumberg (1973) of Ford compared the nitric oxide emissions from the PROCO engine to that of a similar homogeneous engine. Their results are shown in Figure 21. They found that near overall stoichiometric fuel-air

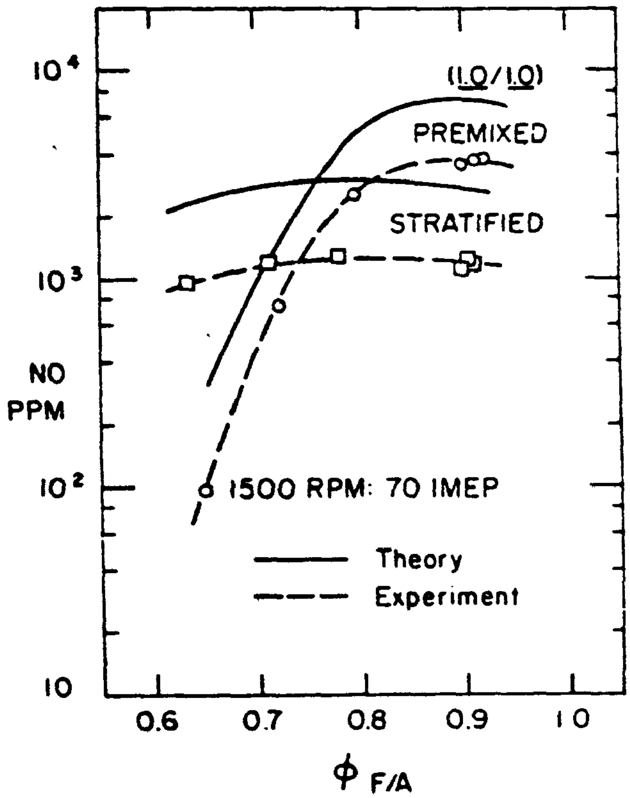


Figure 21 Comparison of Premixed and Stratified NO Concentrations As a Function of Equivalence Ratio at 0% EGR. From Lavoie and Blumberg (1973)

mixtures, the PROCO engine had less nitric oxide emissions but that for very lean mixtures, the PROCO engine had greater nitric oxide emissions than the homogeneous engine. Blumberg (1973) of Ford has been able to predict the same trends in nitric oxide emissions by use of a computer model of the PROCO engine. The theoretical results are also shown in Figure 21. One essential feature of the model is that the stratification of the charge remains throughout combustion and expansion until the charge temperature is sufficiently low to prevent changes in the nitric oxide concentration. In the model the richest element, in the center, burns first and combustion progresses to the leanest element. When the overall equivalence ratio is made leaner each element likewise becomes leaner. Since the first elements to burn adds the most to the nitric oxide concentration and the first elements are made leaner (still rich but nearer stoichiometric), they tend to increase the nitric oxide concentration. The last elements to burn which are lean and become leaner, tend to reduce the nitric oxide concentration. The net result is that the nitric oxide concentration is nearly independent of fuel-air ratio as seen in Figure 21.

An interesting comparison can be made between the delayed mixing engine with zero EGR shown in Figure 12 and the PROCO engine also with zero EGR shown in Figure 21. Both show very little change in nitric oxide concentration with equivalence ratio. The swirl stratification model and the delayed mixing model both have a delay in the mixing of the rich product region. In the Ford swirl stratification model, the delay extends until the nitric oxide kinetics have been stopped by expansion. In the delayed mixing model the delay lasts only part way through the active nitric oxide kinetic period. In the actual PROCO engine some mixing is likely to occur between the various elements during the active nitric oxide kinetic period and yet the measured nitric oxide concentration is also nearly independent of equivalence ratio. The delay before the rich products start to mix and the limited amount of mixing that does occur before expansion cooling stops the nitric oxide kinetics are the main reasons for the shape of the nitric oxide emissions curve.

3. Newhall Engine

A version of the divided chamber spark ignition stratified charge engine has been built and tested by Newhall and El-Messiri (1973). The fuel is injected into a large prechamber and only air is brought into the main chamber. The nitrogen oxide emissions shown in Figure 22 have a maximum at about an overall equivalence ratio of .7. If the rich products leaving the

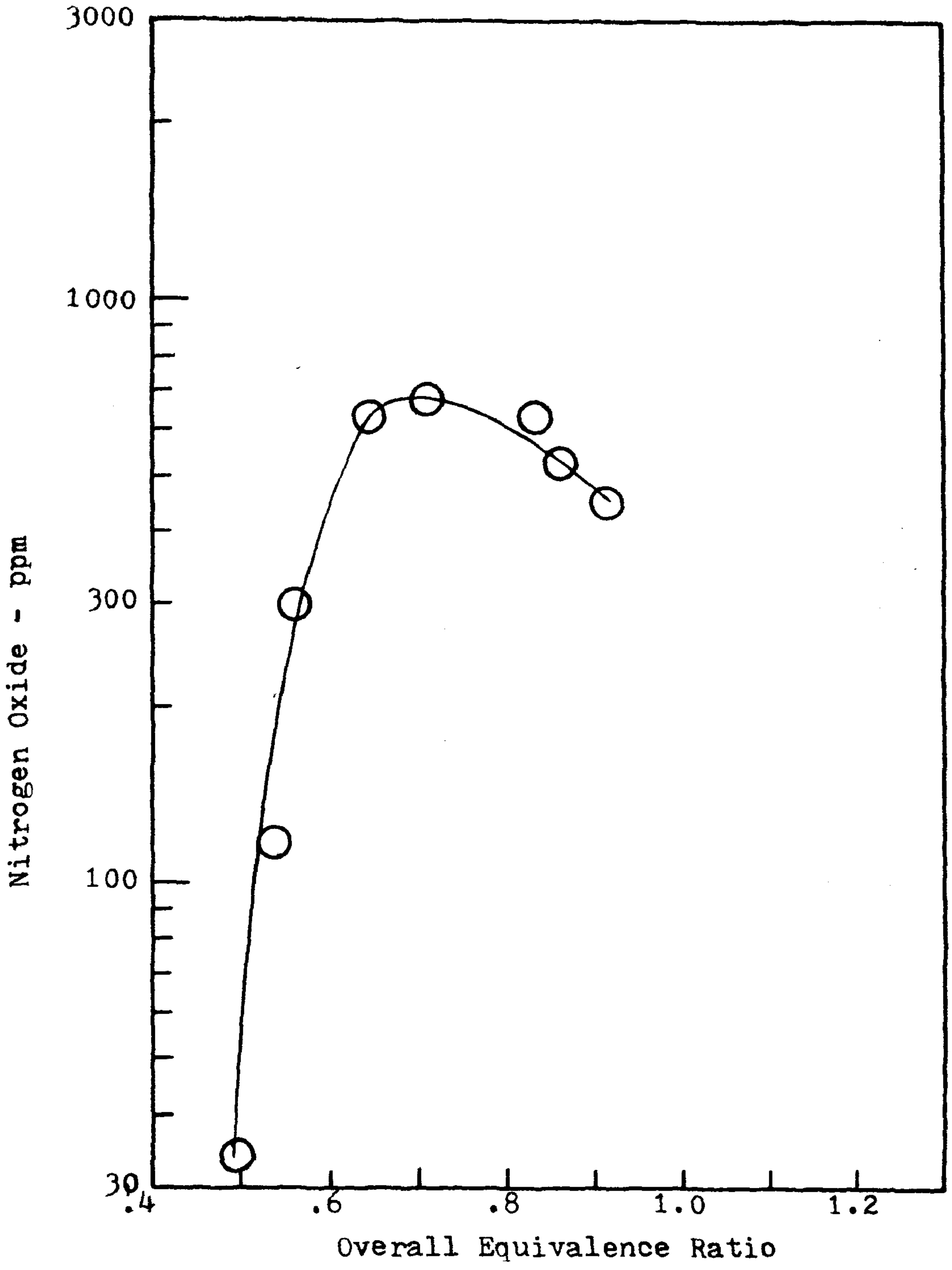


Figure 22 Nitrogen Oxide Concentration versus Overall Equivalence Ratio for Wide Open Throttle and 65% Prechamber Volume, From Ingham(1976)

prechamber are rapidly mixed with the air in the main chamber as is done in the instantaneous model the nitric oxide maximum would occur near stoichiometric, probably not leaner than an overall equivalence ratio of .8. Suppose all of the nitric oxide were to be formed in the prechamber and the mixing were delayed until after the active nitric oxide kinetic period. If the maximum nitric oxide occurred at a prechamber equivalence ratio of .85, the overall equivalence ratio would be about .55 (.85 x .65 = .5525). Apparently a limited amount of mixing and nitric oxide formation does occur in the main combustion chamber which accounts for the nitric oxide maximum occurring between the instantaneous mixing concept and the other extreme of all of the nitric oxide being formed in the prechamber.

The Newhall Engine could be included in the L;P-A group shown in Figure 16c. The rich products of combustion probably remain in the combustion chamber for a short period and when they emerge, they mix rapidly with the lean mixture.

H. COMPRESSION IGNITION STRATIFIED CHARGE ENGINE

Although diesel engines are stratified charge engine they are not usually grouped with spark ignition stratified charge engines because of the large differences in the combustion processes. Certainly the diesel combustion process is considerably more complex than many spark ignition stratified charge engines. However, the chemical reactions and chemical kinetics of nitric oxide apply equally as well to diesel engines as they do to spark ignition stratified charge engines, as they do to any form of combustion with air. For example, Tuteja (1972) showed that the Zeldovich mechanism could be used to calculate the nitric oxide concentration in diffusion flames with reasonable accuracy. Many diesel engine models based on the nitric oxide chemical reactions and chemical kinetics have satisfactorily predicted nitric oxide emissions. Some examples are Cakir (1974), Shahed (1973) Nightingale (1975) and Khan (1973).

Most of the nitric oxide is formed in the regions of hot products of combustion, some is formed in the combustion zone and none is formed in the cool reactants. Nightingale (1975) of Ricardo and Co. demonstrated this point by taking samples inside of a diesel combustion chamber at various times and locations during the combustion process. His results, shown in Figure 23, are for a point in the center of the fuel spray. As combustion proceeds, the hydrocarbons drop to a low level and the CO_2 increases to a higher level. The CO peak is due to the richness of the region being sampled. After combustion

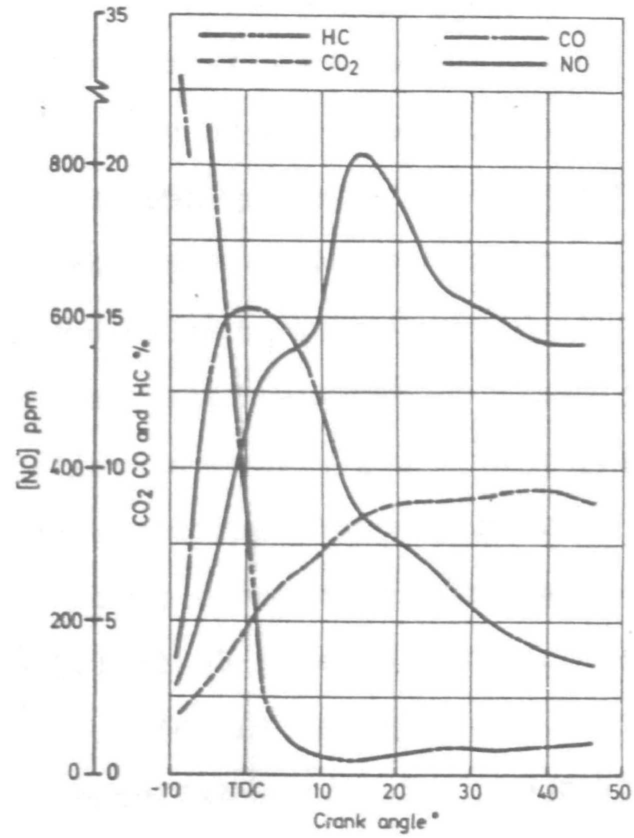


Figure 23 Gaseous Composition in the Center of the Fuel Spray As a Function of Crank Angle From Nightingale, D.R. (1975)

is completed, the nitric oxide level continues to increase until it peaks. This is evidence which shows that much of the nitric oxide is formed in the products of combustion as would be expected from the chemical kinetics.

Nightingale also presented results showing the nitric oxide concentration versus equivalence ratio for all samples taken later than 5^{o} ATDC. At an angle of 5^{o} ATDC most of the combustion should be complete. His results are given in Figure 24a. In general, the results indicate that samples of products having an equivalence ratio near .9 will have greater nitric oxide concentrations than either richer or leaner samples. A similar curve was obtained by Rhee (1976) with samples in the lean region and is presented in Figure 24b. The magnitude of the nitrogen oxide emissions reported by Nightingale is much lower because of the engine modification necessary for photographing the combustion process. The nitric oxide maximum occurs at about the same equivalence ratio as the nitric oxide maximum for homogeneous combustion. The similarity in the results is because both types of combustion produce most of the nitric oxide in the combustion products by means of the same chemical reactions and chemical kinetics, even though the combustion processes used to form the products are very different.

A study of Pischinger (1972) compared the nitrogen oxide emissions from open chamber (direct injection)

Figure 24a

Experimental In-Cylinder NO concentration as a Function of Equivalence Ratio, From Nightingale (1975)

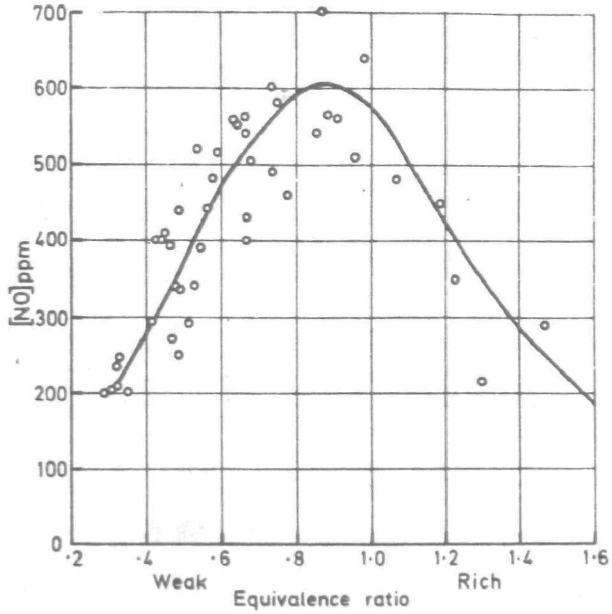


Figure 24b

Nitrogen Oxide Concentration Versus Equivalence Ratio For Samples Taken From 15-20° ATDC, From Rhee (1976)

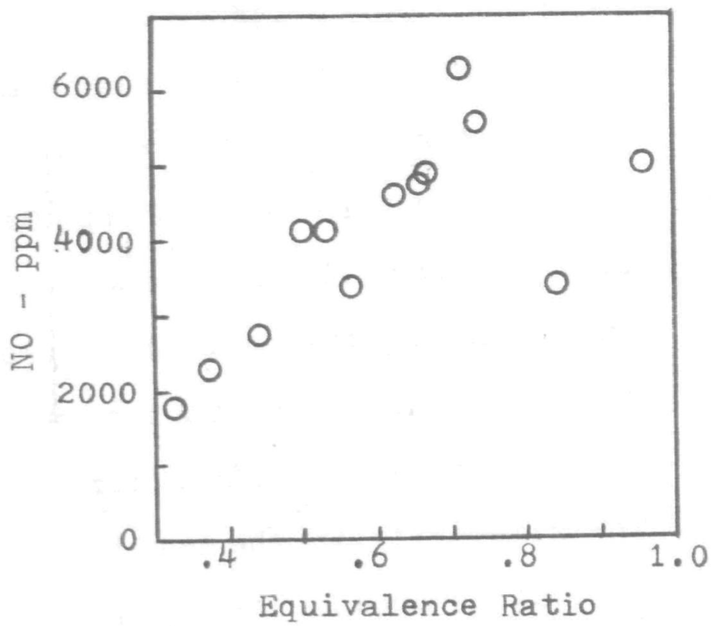


Figure 24 Nitric Oxide Concentration Versus Equivalence Ratio For Samples taken From Within a Direct Injection Diesel After Combustion

diesels and divided chamber (indirect injection) diesels. His results are given in Figure 25. The open chamber diesel has increasing nitrogen oxide levels with increasing BMEP while the divided chamber diesel has a maximum nitrogen oxide level at about the center of its operating range. Monaghan et al. (1974) and others have found similar results in comparing the open chamber and divided chamber diesels.

1. Open Chamber Diesel Engines

The open chamber (direct injection) diesel has a nitrogen oxide emissions curve that is similar to the characteristic shape of the rapid mixing stratified charge engine group, as shown in Figure 25. The explanation for the increasing nitrogen oxide concentration with increasing overall fuel-air ratio depends on how the combustion process is visualized. Shahed (1973) and Nightingale (1975) present combustion models in which the combustion is assumed to occur at stoichiometric conditions and no mixing occurs between the products formed at various times. With their models the increase in the overall fuel-air ratio results in more products at stoichiometric conditions and consequently more nitrogen oxide. In Khan (1973) combustion model, air is entrained in a vaporized fuel jet with fuel and air mixing occurring within the jet. The rate of combustion is

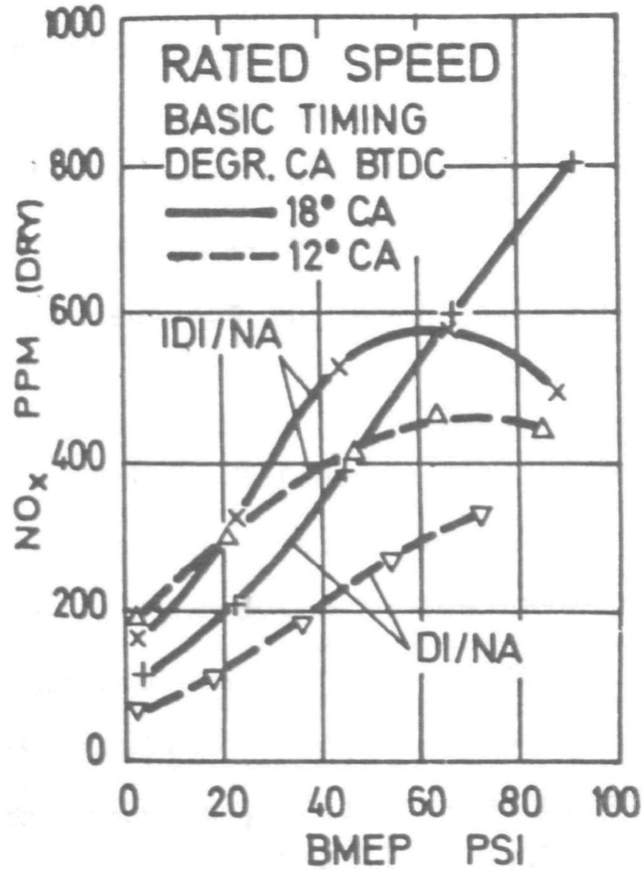


Figure 25 NO_x Concentration, Naturally Aspirated Direct-injection and Prechamber Engines From Pischinger, R. (1976)

determined by the mixing process. In this model, the changes in the equivalence ratio of the products will influence the nitric oxide concentration. Khan's model predicts a nitric oxide peak at high overall fuel-air ratios whereas Shahed's and Nightingale's models do not. See Figure 26. The existence of a nitrogen oxide maximum at high fuel-air ratios is shown experimentally by Nightingale (1975), in Figure 26b. The smoke limit usually prevents diesels from operating at high overall fuel-air ratios where the nitrogen oxide maximum occurs.

If Khan's model is used to visualize the combustion processes, then it would be possible to include the open chamber diesel in the rapid mixing stratified charge engine group. The nitrogen oxide emission curve from the open chamber diesel also indicates that it should be in the rapid mixing group.

2. Divided Chamber Diesel Engines

In the divided chamber diesel engines, the fuel is injected into the prechamber where combustion begins. The pressure in the prechamber increases and forces the product mixture out into the main chamber. Typically the design of the combustion chambers promote swirl in the main chamber. As is shown in Figure 25, the nitrogen oxide concentration has a maximum which does not occur for the open chamber diesel normal operating range. The nitrogen oxide maximum can be explained by postulating

Figure 26a
Calculated (dashed line) and Experimental (solid line) Exhaust NO versus Engine Fueling
From Khan et al. (1973)

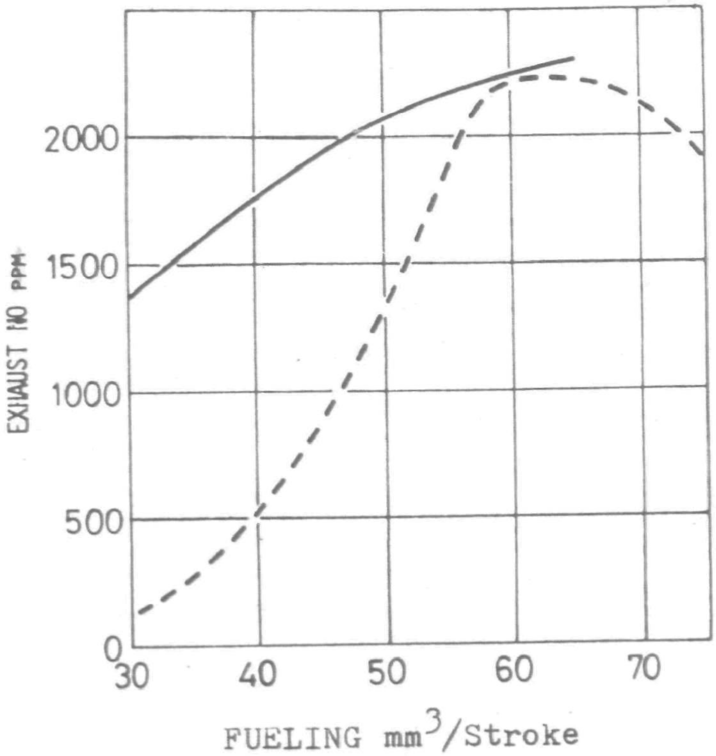


Figure 26b
The Effect of Varying Engine Overall Air/Fuel Ratio (Engine Load) on Exhaust NO Concentration
From Nightingale, D. R. (1975)

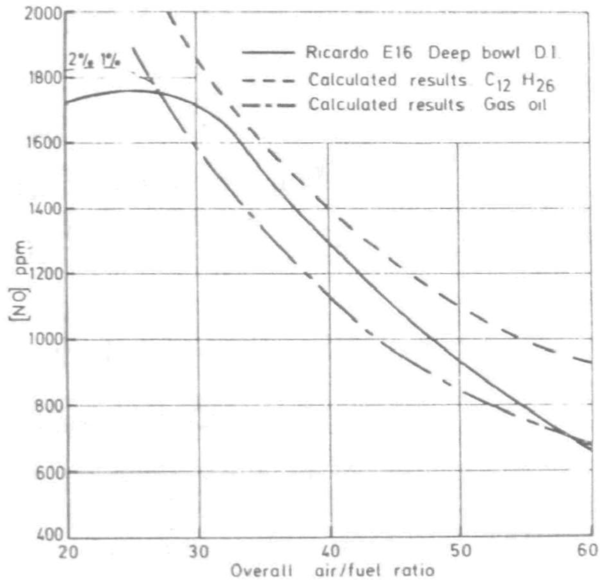


Figure 26 Comparisons Between Nitric Oxide Emissions from Models and Measured Results

that the product mixture entering the main chamber has limited mixing with the main chamber air.

The mixture in the prechamber will always be richer than the overall mixture because only a fraction of the air is in the prechamber. At high powers (BMEP) the prechamber contains a rich fuel-air mixture. The richness of the mixture would account for low nitrogen oxide concentrations in the prechamber. Other factors could limit the nitrogen oxide formation in the main chamber. For example, it is possible that the secondary swirl caused by the products entering the main chamber from the prechamber could result in fluid rotation with the high temperature (low density) products in the center and the low temperature (high density) air on the sides. This stratification could account for some of the delay in the mixing process. Possibly the products mixture enters the main chamber too late in the nitric oxide's active kinetic period to result in more nitrogen oxide. The increase in heat transfer due to the swirl could reduce the temperatures and consequently limit nitrogen oxide concentrations.

Because of the complexity of the combustion and mixing processes, it is difficult to attribute the shape of the nitrogen oxide curve to any one factor. However, it seems reasonable to group the divided chamber engine with the other limited mixing stratified charge engine because it has the same general shape of nitrogen oxide emissions, the prechamber limits mixing,and the swirl in the main chamber also limits mixing.

CHAPTER IV

THE TEST FACILITY AND INSTRUMENTATION

A. INTRODUCTION

In order to study the delayed mixing concept of stratified charge engines, it was first necessary to decide whether the test facility should simulate a practical engine or simply test the concept. There are many problems associated with designing a practical engine such as keeping the rich charge and the air separated before mixing, finding an effective way of mixing the air with the rich products, and determining the most appropriate values for the main variables. Because of the problems associated with the design of a practical engine it was decided not to attempt the simulation of a practical delayed mixing engine.

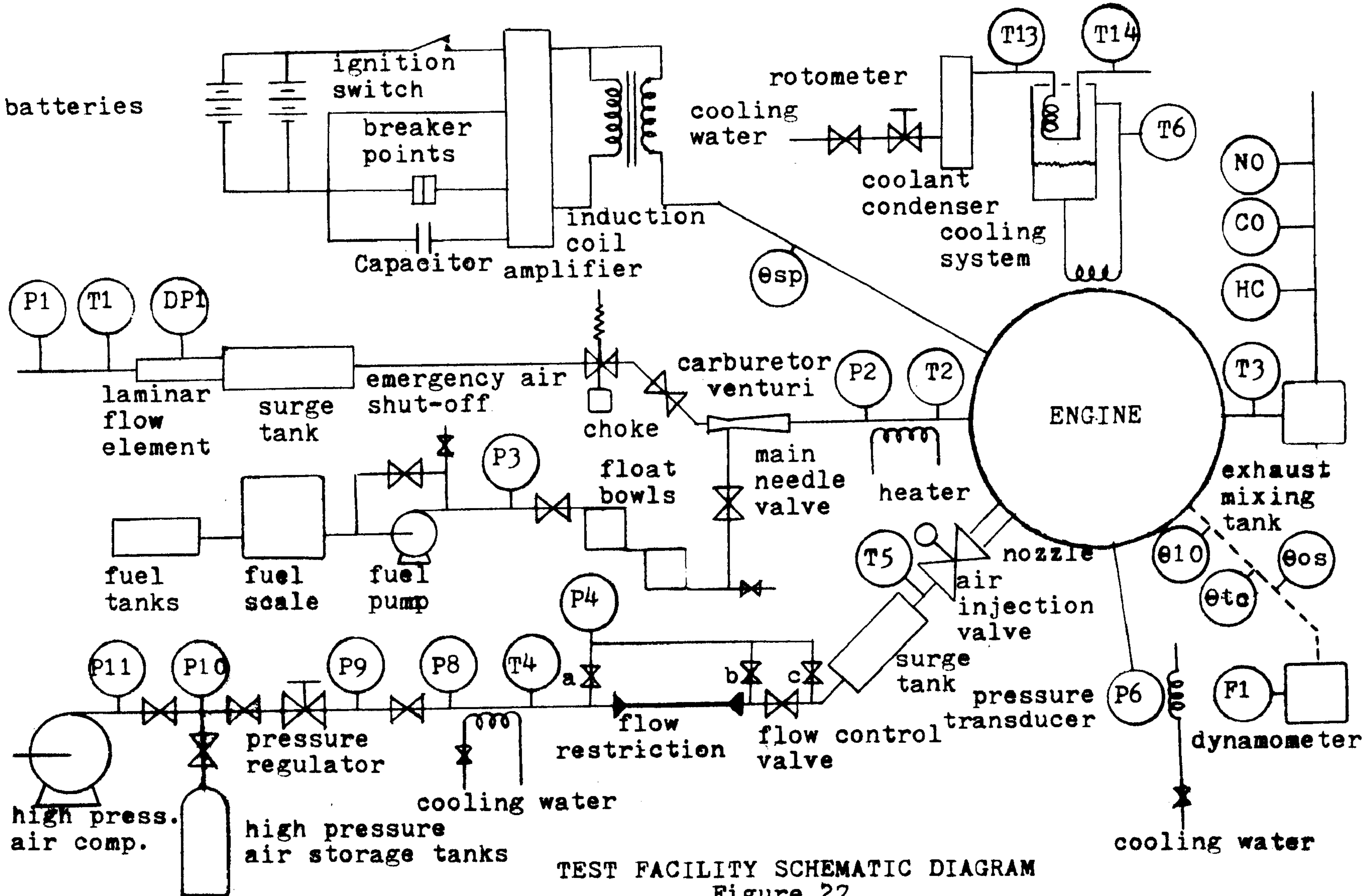
The alternative to designing a practical delayed mixing engine is to experimentally simulate the combustion processes and study the main parameters. This is the approach taken here. The test facility consists of a single cylinder CFR engine fitted with an adjustable carburetor and an adjustable air injection system. The test facility simulates delayed mixing combustion by admitting a rich charge to the combustion chamber, burning the charge and injecting the air when combustion is essentially complete.

A schematic diagram of the test facility is presented in Figure 27.

B. ENGINE

An ASTM-CFR (Fuel Research Engine, 4-1957) single cylinder engine with variable compression ratio was used for the experimental program. Three radial access ports plus the spark plug hole were in the head assembly. One port directly across the cylinder from the spark plug was used for the air injection nozzle. The two other ports are located at 45^{o} either side of the spark plug. One of these ports houses the pressure transducer. Prior to testing, the engine was disassembled, cleaned and rebuilt.

When the compression ratio was adjusted, the influence of the nozzle, inlet and exhaust valve penetration, and spark plug on the clearance volume was included in the calculations. The procedure as deduced from the ASTM Manual (1971), is to place a small block, .625 inches (15.9 mm) high, in the combustion chamber positioned between the valves while the piston is at top dead center. Next the head is cranked down until it stops. A micrometer mounted between the variable head assembly and the stationary base of the engine is adjusted to .362 inches (9.19 mm). The compression ratio can be calculated by means of the following equation.



TEST FACILITY SCHEMATIC DIAGRAM
Figure 27

$$\text{Compression Ratio} = \frac{4.5^{*} + 0.3 + \text{micrometer reading}}{0.3 + \text{micrometer reading}}$$

In this equation the 0.3 inches corresponds to a physical distance of .263 (.625 - .362 = .263) and 4.5 corresponds to the engine stroke. The value of .3 is used in place of .263 to compensate for a slightly larger cylinder diameter in the clearance volume. Corrections for the volume added by the spark plug is equivalent to an additional .004 inches, the valve penetration reduces the volume equivalent to .0253 inches and the air injection nozzle increases the volume in accordance with the size of nozzle used. The correction for the 5/16 inch nozzle is equivalent to an increase of .0156 inches. The compression ratio equation for this nozzle is as follows.

$$\text{Compression ratio} = \frac{4.5 + .294 + \text{micrometer reading}}{.294 + \text{micrometer reading}}$$

The compression ratio equation for the .0935 inch nozzle is as follows.

$$\text{Compression ratio} = \frac{4.5 + .281 + \text{micrometer reading}}{.281 + \text{micrometer reading}}$$

Connected to one end of the engine crank shaft is a dynamometer and connected to the other end is a sprocket used to drive the cam-operated air injection valve. The angular location of the sprocket teeth are sensed by a

*All the numbers on this page are in inches, to convert to millimeter multiply by 25.4.

magnetic pick-up for use as timing marks on the oscilloscope display. Another magnetic pick-up is used to indicate top dead center; a third triggers the sweep of the oscilloscope; and a fourth magnetic pick-up senses engine speed for the emergency shut-down system.

C. DYNAMOMETER

A 50 horsepower (37285 watt) General Electric dynamometer serial number 2127382 is used in conjunction with the control system. The dynamometer system keeps the engine speed constant at the desired level for either motored or powered operation. A 24.75 inch (6278.7 mm) diameter scale calibrated in .2 lb_f (.89N) increments is used to measure a force which can be converted to engine power by the relationship below.

$$\text{Horsepower} = \frac{\text{Force Read on Scale x RPM}}{4000.}$$

When the engine is producing power, the scale force is an indicator of brake horsepower based on this conversion. When the engine is being motored with the ignition turned off the scale force indicates the approximate frictional horsepower. The sum of the powered and motored horsepower is approximately the indicated horsepower.

Also connected to the dynamometer is the sensor for a tachometer. The engine RPM is displayed on one of the control panels.

D. FUEL SYSTEM

The fuel system is shown as part of the test facility schematic presented in Figure 27. Fuel is stored in outboard engine fuel tanks. The tanks are connected to an automatic fuel weighing system. In the fuel weighing system a time is measured which corresponds to a specified mass of fuel; in our tests twenty grams. A fuel pump draws the fuel from the scale and delivers it to the float bowls. The float bowls maintain a constant level by means of the float bowl valve. Two float bowls in series are used to increase the stability of the fuel flow rate to the engine. Fuel leaves through the bottom of the second float bowl, passes through the main needle valve which is a 1/4 inch micrometer valve and enters the venturi of the CFR carburetor where fuel mixes with the air.

A number of calibration runs were performed on the fuel system by extracting fuel from between the needle valve and the carburetor. A small valve was installed in that line for the purpose of controlling fuel flow rate. During the test period the fuel was collected in a beaker while the fuel scale was activated as often as possible. A comparison is shown below of the time interval corresponding to the flow of twenty grams of fuel as obtained from the fuel scale and calculated from the accumulated fuel.

Time Interval -seconds			
Fuel Scale			
Average	Std. Dev.	Accumulated Fuel	Time Interval Ratio
64.02	.53	65.18	.9822
65.51	1.26	65.93	.9936
66.42	.634	66.74	.9952
66.15	.721	66.34	.9971

Although the scale is seen to be quite accurate one known source of error can be calculated is due to the buoyancy associated with the tubing used to draw the fuel from the fuel beaker located on the scale. Since the fuel level in the beaker is lowered during the weighing process the buoyancy force exerted on the tubing and the beaker is reduced. The reduction in the buoyancy appears to the scale as removal of fuel. Thus the scale would indicate more fuel removed for the beaker than was actually removed or the time interval indicated by the fuel scale for the removal of twenty gram of fuel is shorter than the correct value. For our system the error is equivalent of .0922 grams. The ratio of the time interval indicated by the fuel scale to the time interval corrected for buoyancy is .9954. This is approximately the ratio of the fuel scale time interval to that deduced from the accumulated fuel. If the fuel scale were to be corrected for buoyancy of the fuel removing tube the accuracy would be greater. However, it is sufficiently accurate without the buoyancy correction.

E. CARBURETED AIR SYSTEM

The air that flows through the carburetor, called carbureted air, is drawn into a surge tank through a laminar flow element as indicated in Figure 27. The surge tank reduces the air flow fluctuations and establishes a nearly constant pressure differential across the laminar flow element. An inclined manometer is used to measure the pressure differential. The laminar flow element was calibrated by motoring the engine and measuring the air entering the laminar flow element with a large bellows meter. The flow of air through the laminar flow element can be described by the linear relationship shown below.

$$\text{Volumetric Flow Rate (CFM)}^{*} = 5.964 \times \Delta P(\text{inches of } H_2O)$$

A choke is located in the carbureted air line to restrict the air flow for part load operation. An emergency air shut off valve is also located in the air line which closes in the event of an over speed accident.

The air enters the venturi section of a CFR carburetor and mixes with the fuel. Between the carburetor and the engine is a section of piping which is heated with a strip heater. A mercury manometer is used to measure the inlet fuel-air mixture pressure and various

*See Appendix I for conversion to SI units.

temperatures are measured by thermocouples connected to the multipoint recorder.

F. INJECTION AIR SYSTEM

The high pressures associated with combustion require that the injection air be compressed to high pressures. A high pressure air compressor is used to fill high pressure air storage tanks as shown in Figure 27. During engine operation the air is drawn from the storage tanks and passes through a pressure regulator which controls the entrance pressure to the flow restriction. The flow restriction is used to measure the air flow rate. Following the flow restriction is a needle valve which is used to control the flow of air. A surge tank near the air injection valve damps out flow fluctuation caused by the periodic opening of the air injection valve. After leaving the air injection valve the air flows through a nozzle which controls how the air enters the combustion chamber. More details about the main components of the air injection system are given in the following sections.

1. High Pressure Air Compressor System

A four stage Joy Model 15-H air compressor capable of reaching pressures of 3500 psi (24130 KP_a) is used to supply the high pressure air. Five air storage tanks rated at 2300 psi (15860 KP_a) are used to store the air. The tanks provide sufficient capacity for normal operation of a few hours. Typically the air pressure is reduced to about 1200 psi before entering the engine test cell. The pressure regulator used is pneumatically controlled.

2. High Pressure Air Flow Measurement System

A major problem associated with the air injection system is the method of measuring the air flow rate. At the high pressures used, a laminar flow element would have to be very small in diameter and very long to provide sufficient pressure drop for accurate measurement. There is also the question of high pressure requirements of the pressure differential measuring device.

The approach used was to include in the line a substantial flow restriction. The flow restriction consists of a .054 inch (1.37 mm) diameter tube with three .017 inch (0.43 mm) diameter wires in it. Within the four foot flow restriction the flow would be turbulent. For fully developed turbulent flow through a nearly constant temperature tube a computer program was developed which correlated the flow through the flow restriction to the inlet pressure, inlet temperature and the outlet pressure. Numerous calibration runs were made under various flow conditions. The measured flow was compared to the computer program predicted flow rate. A very slight adjustment was made in a constant in the friction factor equations in order to establish agreement between the measured and calculated results. The deviation from the calculated values is usually less than 2 psi (13.97 KP_a). The details of the computer program and the calibration of the flow restrictions are presented in Appendix E.

3. Air Injection Valve

The air injection valve had to be specially designed because of its special requirements for large flow rates and short open periods. The essential features of the valve are that it opens and closes in 40 crank angle degrees and it can let in a mass of air about equal to the carbureted charge at wide open throttle. A sketch of the valve is shown in Appendix F. Included in Appendix G is a computer program which describes the flow of air through the valve and the dynamic forces resulting from the cam engaging the cam follower.

G. PRESSURE TRANSDUCER SYSTEM

An AVL type KQD 250C pressure transducer is mounted in one of the access ports in the head assembly. This type of pressure transducer is water cooled and has been found to be quite accurate by Lancaster (1975). The pressure transducer was prepared, as recommended by Lancaster, with a General Electric RTV 560 (synthetic rubber) coating to reduce the heat transfer. The pressure transducer signal goes to a Kistler charge amplifier model 566, S1N848. The signal from the charge amplifier is displayed on the oscilloscope.

H. OSCILLOSCOPE

A Tecktronix Type 35A oscilloscope is used to display the cylinder pressure signal and the timing marks. The timing marks indicate ten crank angle degree increments with the top dead center pulse being greater in amplitude. The oscilloscope has a single beam. In order to show both the pressure signal and the timing marks it is necessary to use a special plug-in unit which will either alternately display the two signals or simultaneously display both signals by alternately displaying small increment of each (chopped). The oscilloscope was checked-out and calibrated before it was put in service.

The engine speed can be obtained from the oscilloscope display of the timing marks. From the oscilloscope it is possible to determine the elapsed time between top dead center pulses. This elapsed time can be converted into engine RPM. The engine speed was determined by this method.

I. MULTIPOINT TEMPERATURE RECORDER

A Leeds and Northrup multipoint temperature recorder serial number E79-52108-1-1 is used to record the various temperatures. The thermocouples are chromel-alumel, Type

K. The multipoint recorder has the capability of recording 24 different temperatures. Twelve of the temperatures can be in the range of 20°F to 250°F (266.3°K to 394°K) and the other twelve can be in the range of 200°F to 2000°F (366.3°K to 1366°K). The numbers indicated on the test facility schematic diagram (Figure 27) correspond to the numbers printed by the multipoint recorder.

J. EMISSIONS CART SYSTEM

The emissions cart system is composed of two parts the cart containing the instrumentation and the cart containing the calibration and purge gas cylinders. The instrumentation cart is set-up to measure nitrogen oxides (NO and NO_x), carbon monoxide and hydrocarbons. There are valves on the instrumentation cart that are used to select a purge gas (nitrogen), room air, the appropriate calibration gas or a sample from the engine exhaust gases. The sample is run through an ice bath and filter before it goes to the instrumentation. Pumps on the instrumentation cart cause the sample to flow from the exhaust pipe and through the instrumentation. Details of each of the instruments is given below.

1. Nitrogen Oxide Measurement

The nitrogen oxide measurement is made by a chemiluminescent NO-NO_x gas analyzer model 10A manufactured by Thermo Electron Corporation. The instrument has seven ranges for 0-10 to 0-1000 PPM, its sensitivity is .1 PPM and can measure either NO or NO_x.

The chemiluminescent nitrogen oxide analyzer uses the photons generated when nitric oxide and ozone are reacted to count the number of nitric oxide molecules. A photomultiplier tube is optically filtered so that it will only respond to the nitric oxide photons. The rate of flow of sample through the reaction cell is maintained constant so that the output signal will be proportional to nitric oxide. When the nitrogen oxides (NO_x) are to be measured the sample is passed through a thermoconverter which converts the various oxide of nitrogen to nitric oxide before the sample goes to the reaction chamber.

2. Carbon Monoxide Measurement

The carbon monoxide is measured by a nondispersive infrared detector (Infrared Analyzer - IRISA) manufactured by Beckman. The instrument consist of two parts; a unit containing the sensing elements and a unit containing the electronics and meter. A nondispersive infrared detector detects carbon monoxide by comparing the amount of infrared radiation absorbed by the sample and a reference gas. Infrared radiation is passed through a reference cell and also through a cell containing the sample gas. After the radiation passes through the cells it goes to two cells containing carbon monoxide. If the sample contains carbon monoxide it will absorb some of the energy of the infrared radiation that is at the carbon monoxide absorption frequencies. Thus the infrared radiation leaving the sample cell will be deficient in energy associated with the carbon monoxide absorption bands. Since the detection cells are filled with carbon monoxide they too would absorb energy in the absorption band of carbon monoxide. The detection cell associated with the sample cell will receive less energy in the absorption bands when more carbon monoxide is in the sample cell. The detection cell associated with the reference cell will receive the same amount of energy in the carbon monoxide absorption bands. The differences in the energy absorbed by the two detection cells results in a pressure differential which is measured by the deflection of a diaphragm between the two cells.

3. Hydrocarbon Measurement

The hydrocarbon measurement is made by a Beckman Hydrocarbon Analyzer, model 109A. The instrument operates on the principle of flame-ionization. A hydrogen flame is established that burns with a negligible number of ions. The sample gases are introduced into the reactants of the flame. If the sample gases have hydrocarbons present they will form ions when they burn. The ions are collected and result in a signal proportional to the concentration of hydrocarbons in the sample. It is necessary to maintain constant flow rates to assure the accuracy of the measurements.

CHAPTER V

THE EXPERIMENTAL PROGRAM AND TEST RESULTS

A. INTRODUCTION

The objective of the experimental program is to determine if the advantages of the delayed mixing concept shown by the theoretical study presented in Chapter II can be realized in an experimental engine. The experimental program is also intended to show the influence of major input variables on the emissions, efficiency and power.

A CFR engine was modified to include an air injection valve which can inject air at various times in the cycle or not at all. Typically a rich charge would be drawn into the combustion chamber. The charge is ignited before top dead center. Air is then injected into the combustion chamber after the rich mixture has finished burning. The combination of the rich mixture combustion followed by air injection is intended to represent the delayed mixing concept as symbolized by L;A-P. The delay between the combustion of the rich mixture and mixing is controlled by the timing of the spark and the air injection. Since it takes a finite amount of time for the air to mix and react with the rich products, this represents a difference from the theoretical concept which assumes that no time is required for the mixing.

The engine input variables were examined to determine the best way to control and measure them, and to decide which factors to vary and which to hold constant.

In addition to the measurement of the nitrogen oxide emissions, the carbon monoxide and hydrocarbon emissions were measured. The engine power and efficiency were also determined. A section is devoted to a discussion of the theoretical trends expected in these results.

The experimental program is divided into two sections. The first set of tests were performed with a large diameter air injection nozzle and the second set of tests were performed with a small diameter air injection nozzle. Within each set of data the major variables studied were the overall air-fuel ratio, the timing of the start of air injection and, to a lesser extent, the carbureted air-fuel ratio. As the experimental program proceeded, problems were recognized and fixed. For this reason some of the data is more accurate than others.

In general, the data obtained from each experiment is punched on computer cards and then processed by computer programs to obtain the results in a more convenient form. The computer programs used to reduce the data are given in Appendix H.

B. ENGINE INPUT VARIABLES

Listed below are the important engine input variables which influence the performance of the delayed mixing stratified charge test engine. These input variables fall into two categories, those that can be changed while the engine is in operation and those that require a mechanical modification to the test facility. The first group will be called operational variables and the second group will be called design variables.

1. Operational Variables

1) Fuel Flow Rate: The fuel flow rate is controlled by the main needle valve of the carburetor and is measured by the fuel scale.

2) Carbureted Air Flow Rate: The carbureted air flow rate is controlled by the choke and engine speed. A laminar flow element and an inclined manometer are used to measure the flow rate.

3) Injected Air Flow Rate: The injected air flow rate is controlled with the flow control valve and determined by the upstream temperature, the upstream pressure and downstream pressure of the flow restriction.

4) Start of Air Injection: The start of air injection is adjusted by selecting the desired relative position of the drive sprocket mounted on the crank shaft and the driven sprocket on the cam valve before the drive chain

is installed. The position of the crank shaft is measured on the flywheel and the position of the air injection valve is measured by a protractor mounted on the driven sprocket of the air injection valve.

5) Spark Timing: The spark timing is controlled by rotation of the housing containing the points and is measured by either an induction pick-up displayed on the oscilloscope or by a timing light.

6) Compression Ratio: The compression ratio is controlled by moving the engine head assembly up or down with a crank. A micrometer measures the position of the engine head, from that measurement the compression ratio can be determined,

7) Engine Speed: The engine speed is controlled by the dynamometer system and is measured by both a tachometer and the time between top dead center pulses on the oscilloscope.

8) Engine Inlet Pipe Pressure: The inlet pipe pressure is controlled by the choke and is measured by a manometer.

9) Engine Inlet Pipe Temperature: The inlet pipe is heated with a strip heater and its temperature is measured by a thermocouple.

10) Injection Air Temperature: The injection air temperature can be controlled by a heat exchanger and is measured by a thermocouple.

2. Design Variables

1) Period of Air Injection: The period of air injection can be changed by changing the cam design, but was not changed.

2) Nozzle Geometry: The nozzle diameter can be changed to change the injection velocity and the nozzle direction can be changed by drilling the nozzle at an angle to the nozzle center line.

3) Shrouded Valve: A shrouded valve can be used to induce swirl in the combustion chamber, but was not used.

3. Equivalence Ratios as Alternate Operational Variables

It is customary to discuss engine performance in terms of fuel-air (or air-fuel) ratio or equivalence ratio because the performance is usually a strong function of these variables. In general, equivalence ratio will be used. The set of three variables fuel flow rate, carbureted air flow rate and injection air flow rate will be replaced by an alternate set of three variables carbureted equivalence ratio, overall equivalence ratio and carbureted air flow rate. It is necessary to also include the carbureted air flow rate in the second set in order to specify the amount of charge.

4. Specified Operational Variables

Since there are ten operational variables it would be impractical to methodically test all combinations. If two levels of each variable were tested 1024 (2^{10} = 1024) runs would be required. It is necessary to hold many of the variables roughly constant and examine those of greatest interest. Listed below are the variables which were usually held constant and a justification for each.

Engine Speed: The engine speed was kept at about 800 RPM. This was done because of concern over the possibility of an air injection valve failure. The dynamic forces on the cam follower of the air injection valve increase with speed. At 800 RPM (13.3 rev/sec) the valve sounds fine and should have a long life based on calculation by the air injection valve computer program. When lower speed operation was tried the coupling between the dynamometer and the engine failed. It has been replaced by a larger coupling and has operated well at 800 RPM (13.3 rev/sec).

Carbureted Air Flow Rate: For nearly all of the runs the carbureted air flow rate was held constant at a level equivalent to a wide open choke and throttle at 800 RPM (13.3 rev/sec).

With the engine speed held constant it would be possible to change the mass flow rate of carbureted air by adjusting the choke, but it would also be necessary to readjust the fuel flow rate at each point because the change in inlet pipe pressure affects fuel flow rate. It takes some time to establish a constant fuel flow rate because the fuel scale does not provide an instantaneous measure of fuel flow rate, only an integrated average value. Several readings must be taken at each fuel flow setting to determine if the flow is steady and at the desired flow rate.

The advantage of a constant maximum carbureted air flow rate is that once a constant fuel flow rate has been established the carbureted equivalence ratio will be constant. It is then possible to run a series of tests at various overall equivalence ratios by changing only the rate of air injection.

With a constant carbureted air flow rate it is possible to calculate the fuel flow rate desired for a specific carbureted equivalence ratio and the air injection flow rate desired for a specific overall equivalence ratio. A number of tables and curves were made to assist in reaching a specific operating condition.

Carbureted Equivalence Ratio: In general, the carbureted equivalence ratio was adjusted to approximately 1.4. At this condition only about 65 PPM of nitrogen oxide are produced by the carbureted mixture. This value of equivalence ratio also corresponds to that used in the theoretical model.

Engine Inlet Pressure: The engine inlet pressure is related to the carbureted air flow rate which is held constant, thus the engine inlet pressure will also be constant.

Compression Ratio: At compression ratios much in excess of seven the engine will knock. It was concluded that the engine should be operated at a compression ratio of seven.

Spark Timing: In the large diameter nozzle tests, the spark was arbitrarily set. Later it was concluded that the spark should be set at the maximum advance for best torque (MBT). The spark timing at the MBT represents the most power and best efficiency that can be obtained at that operating condition.

C. A DISCUSSION OF THE THEORETICAL TRENDS EXPECTED IN THE EXPERIMENTAL RESULTS

1. Nitrogen Oxide Emissions

Both Chapter II and Chapter III have dealt with the trends in nitrogen oxide emissions. In summary, the nitrogen oxide emissions for a delayed stratified charge engine can be expected to increase with leaner overall equivalence ratios, with leaner carbureted equivalence ratios, with earlier air injection and with advanced spark timing.

2. Carbon Monoxide Emissions

In engines the most significant factors affecting the formation of carbon monoxide (CO) are the fuel-air ratio and the CO chemical kinetics. When a fuel-rich mixture is burned CO is formed instead of CO_2 because of the lack of oxygen. Thus the richer the fuel-rich mixture the greater the concentration of CO in the exhaust.

Carbon monoxide is also found in the exhaust of engines when operated with fuel-lean mixtures. The exhaust gas concentration is determined by the fuel-air ratio, the temperatures, the pressures and the chemical kinetic history. In the combustion chamber the equilibrium concentration of CO is high after combustion and decreases during expansion. The actual CO concentration rapidly approaches the high equilibrium value while temperatures are high and attempts to follow the decreasing equilibrium values during expansion. The chemical kinetics of CO freezes during the expansion with the resulting CO concentration lying between that of the equilibrium value at combustion conditions and the equilibrium value at exhaust conditions. Newhall (1969) has described this behavior of CO in great detail.

In the experimental engine the amount of CO found in the exhaust will depend upon the mixing and reaction of the injected air with the rich products of combustion. If the injected air does not mix and react with rich products, large carbon monoxide emissions would result similar to those expected for homogeneous operation at the carbureted fuel-air ratio. If the injected air does mix and react with the rich products, small carbon monoxide emissions would result similar to those expected for homogeneous operation at the overall fuel-air ratio.

The carbon monoxide emissions for air injection operation before combustion will also depend upon the mixing process. The more rapid the mixing the closer the level of carbon monoxide emissions will be to the homogeneous operation at the overall fuel-air ratio.

With air injection the amount of charge is greater because the injected air adds to the constant maximum carbureted air intake. Since the charge is greater the cylinder pressure will also be greater. The greater cylinder pressure will not influence the equilibrium concentration of CO but it may change the CO kinetics by changing the concentration of other species which are pressure sensitive. It would be necessary to perform a kinetics computer calculation to determine the pressure effect on CO concentration.

3. Hydrocarbon Emissions

Daniel and Wentworth (1962) showed that the primary source of hydrocarbon emissions for conventional four-stroke spark ignition engines was due to the quenching of the flame as it approached the cool combustion chamber wall. They also showed that oxidation of the hydrocarbon emissions could occur in the exhaust pipe if sufficient oxygen and high enough temperatures are present.

After combustion is completed an envelope of unburned fuel-air mixture lies around the combustion chamber surface. The amount of unburned hydrocarbon contained within the envelope depends of the volume of the envelope and the fuel density within the envelope. The volume of the envelope is changed by changes in the quench distance, piston motion and air motion. The envelope also includes any crevices that are not penetrated by the flame front, such as between piston and the cylinder.

The thickness of the quench layer is a function of pressure, temperature and fuel-air ratio. Increases in the temperature and pressure tend to decrease the quench layer thickness. The effect of fuel-air ratio on the quench layer thickness is complicated by its affect on the pressure and temperature. In general, the quench distance reaches a minimum thickness at a slightly rich fuel-air ratio. The amount of hydrocarbons in the quench layer will also depend on the fuel-air ratio. Hydrocarbon emissions are found to decrease as the fuel-air mixture is made leaner until they become low and relatively constant for lean fuel-air mixtures. But, if the fuel-air mixture is made sufficiently lean to result in misfire, the hydrocarbon level will increase due to the unburned fuel.

When air is injected into the cylinder before combustion, there are many factors which will affect the hydrocarbon level and it is difficult to conclude how these factors will sum to influence the total hydrocarbon emissions of early injection. The increased pressure forces more fuel-air mixture into the crevices favoring higher hydrocarbon emissions, but also tends to reduce the quench layer thickness favoring lower hydrocarbon emissions. The increased density of the quench envelope would tend to increase the hydrocarbons. When air is injected during the compression stroke, it is unlikely that it will be able to penetrate the boundary layers and crevices before combustion. Consequently, the fuel-air mixture in the quench envelope will be fuel-rich, which would tend to increase the hydrocarbon emissions. It is difficult to predict how the total hydrocarbon emissions of the early air injection operation will compare to homogeneous operation at the same overall fuel-air ratio.

When air is injected after combustion, the quench envelope has already been formed under conditions of the rich carbureted charge. The greater fuel-air concentration will tend to increase the hydrocarbon emissions. In contrast, the injection of air will increase the velocities in the combustion chamber which will reduce the boundary layer thickness and tend to reduce the hydrocarbon emissions. Again it is difficult to determine how these factors affect the hydrocarbon emissions of the late injection operation as compared to homogeneous operation at the same overall fuel-air ratio.

4. Engine Power

In a conventional homogeneous engine the specific power depends on the engine speed, spark timing, fuel-air ratio of the charge, amount of charge, compression ratio and a number of other minor variables. Engine speed and compression ratio have been held constant for most of the tests. Spark timing has either been held constant or adjusted for the maximum advance for best torque operating condition. For most operating conditions the amount of carbureted charge was also held constant at the maximum value. Thus, the fuel-air ratio is the most significant variable remaining.

When the homogeneous engine is operated fuel-rich, power is limited by the amount of air available for combustion. Since the carbureted charge is constant the amount of power is expected to be nearly constant. At very rich fuel-air ratios, the fuel will displace some of the air causing a slight reduction in power. When the homogeneous engine is operated fuel-lean, the power is limited by the amount of fuel available for combustion. As the fuel-air ratio becomes leaner the power will decrease.

Engine operation with air injection results in a larger charge since the carbureted charge is constant and the injected air will add to the total charge. The extent of the increase in power associated with the larger charge will depend of the total amount of charge, the air injection timing, the carbureted fuel-air and the overall fuel-air ratios.

The air which is injected into the combustion chamber must be compressed to the appropriate pressure. The work required to compress the air will depend on the process used. If a method of compressing the air in the cylinder along with the rich charge and by the combustion process can be developed, then external compressors will not be required. The work of compressing the air would be similar to that of compressing the charge. In the experimental engine system, the air is compressed separately to greater than injection pressure and stored in gas bottles until it is injected. The work used to drive the compressor is not a reasonable estimate of the work required to compress the air for injection. It has been assumed that the air compression work is best represented by an isentropic compression for room temperature and pressure to injection pressure as measured in the accumulator before the air injection valve.

5. Engine Efficiency

The fuel-air ratio significantly influences efficiency for both homogeneous operation and air injection operation. With air injection operation two fuel-air ratios have to be considered, that of the carbureted charge and that of the overall charge including the injected air. The efficiency for homogeneous charge engines is expected to drop sharply for fuel-rich mixtures because of the lack of oxygen to complete the combustion process. Likewise making a fuel-lean mixture leaner results in better efficiency because the combustion process goes nearer to completion.

During air injection operation the efficiency associated with the combustion of the rich charge would correspond to the efficiency of the homogeneous operation at the same rich fuel-air ratio. The addition of air to the rich charge will tend to complete the combustion process and increase the efficiency over that of the carbureted charge alone. When the air enters the combustion chamber early in the expansion it can do useful work through expansion even if it does not react. Increasing the amount of air injected will make the overall mixture leaner and further increase the efficiency. The efficiency of the delayed mixing engine will fall somewhere between the efficiency of the homogeneous operation corresponding to the carbureted fuel-air ratio and to the overall fuel-air ratio.

If the air injection occurs late in the expansion process, little additional work can be extracted because the amount of expansion which remains is small and insufficient time may remain during the expansion to mix and react the air with the rich products. On the other hand if air is injected early, before combustion, it will have time to form a more nearly homogeneous mixture at the overall fuel-air ratio. Under this form of operation the efficiency should approach that of a corresponding homogeneous operation at the same overall fuel-air ratio. The efficiency of air injection operation is expected to be between that of homogeneous operation with the carbureted fuel-air ratio and with the overall fuel-air ratio.

Spark timing is used to adjust for maximum engine torque. When the engine is operating at the maximum torque it is also operating at the maximum efficiency because the power output is greatest for a constant amount of fuel input. With the spark too far advanced the compression work will be done against combustion pressure and with the spark too far retarded the pressure peak will occur later in the expansion. In either operating condition the efficiency will be less than maximum.

Another factor which affects the efficiency of the engine is the heat transfered to the combustion chamber walls. The convective heat transfer will be increased by greater velocities in the combustion chamber. When air is injected the velocities will be greater because of the kinetic energy of the air jet. The efficiency of air injection operation could be lower due to more heat transfer than corresponding homogeneous operation.

Two measures of efficiency will be used in the presentation of results. The measured indicated enthalpy efficiency is based on the brake horsepower of the engine plus the mounted horsepower. The corrected indicated enthalpy efficiency is based on the brake horsepower plus the motored horsepower less an estimated power required to compress the injected air.

D. LARGE DIAMETER NOZZLE TEST RESULTS AND DISCUSSION

1. Check-Out

Much of the initial operation was intended to check-out the system and to determine how best to control the

the system. Some mechanical problems with the air injection valve were discovered and corrected. The coupling between the engine and the dynamometer was too small and had to be replaced. The carburetor float bowl was first replaced by a larger one and later both float bowls were placed in series to stabilize the fuel flow rate. It was necessary to replace the fuel pump, one was purchased which incorporated a pressure regulator. The new fuel pump greatly improved the stability of the fuel flow to the carburetor. Improvements were made to the fuel scale to improve its accuracy. Although the repairs and maintenance were time consuming they did not represent significant technical problems.

2. Test Conditions for Large Diameter Nozzle

The large diameter nozzle has a diameter of .3125 inches (7.938 mm) and is directed perpendicular to the center line of the cylinder and into the clearance volume. A flow restriction consisting of a .0935 inch (2.37 mn) hole in a disc was placed between the air injection valve seat and the nozzle. The flow restriction caused an increase in the air injection accumulator pressure to prevent the flow of combustion product into the air injection valve. The maximum velocity through the flow restriction is sonic velocity which resulted in the maximum velocity from the nozzle of about one hundred feet per second.

The engine was operated near 800 RPM with the maximum carbureted air flow rate. The compression ratio was held constant at seven. The spark timing was held constant at 10° BTDC. Both the injection air temperature and the carbureted air temperature were uncontrolled. These temperatures were usually above room temperature due to higher temperature of the engine.

The magnitude of the other variables which changed during the testing are shown in the figures.

All testing with the large diameter nozzle was done with air injection after the start of combustion.

3. Summary of Results for Large Diameter Nozzle

1) The nitrogen oxide emissions for operation with air injection are much lower than homogeneous engine operation at the same overall air-fuel ratio.

2) The carbon monoxide emissions for operation with air injection are essentially the same as homogeneous engine operation at the same overall air-fuel ratio.

3) The hydrocarbon emissions for operation with air injection are lower than homogeneous engine operation at the same overall air-fuel ratio.

4) The measured indicated enthalpy efficiency for operation with air injection is lower than homogeneous engine operation at the same overall air-fuel ratio.

5) The corrected indicated enthalpy efficiency for operation with air injection is much lower than homogeneous engine operation at the same overall air-fuel ratio.

4. Nitrogen Oxide Emissions for Large Diameter Nozzle

A set of NO_x emission results for the low velocity nozzle is shown in Figure 28. The homogeneous curve was obtained by operating the engine without air injection at various overall equivalence ratios. At very rich operation the nitric oxide concentrations are very low. As the overall equivalence ratio becomes leaner the nitric oxide increases until it peaks at about $\phi_o = .95$. At this equivalence ratio the engine combustion becomes unstable which in part accounts for reduced NO_x at lower overall equivalence ratios. The lower flame temperatures associated with fuel lean operation also reduce the NO_x emissions. An NO_x peak with slightly fuel-lean mixtures is typical of homogeneous engine operation.

Also shown in Figure 28 is a set of results with air injection having a carbureted equivalence ratio (ϕ_c) of about 1.4 and at various air injection timings. With the start of air injection at 42^o ATDC (crank angle degrees), very little additional NO_x is formed due to an injection. Advancing the air injection to start at 22^o ATDC increases the NO_x slightly. With the air injection

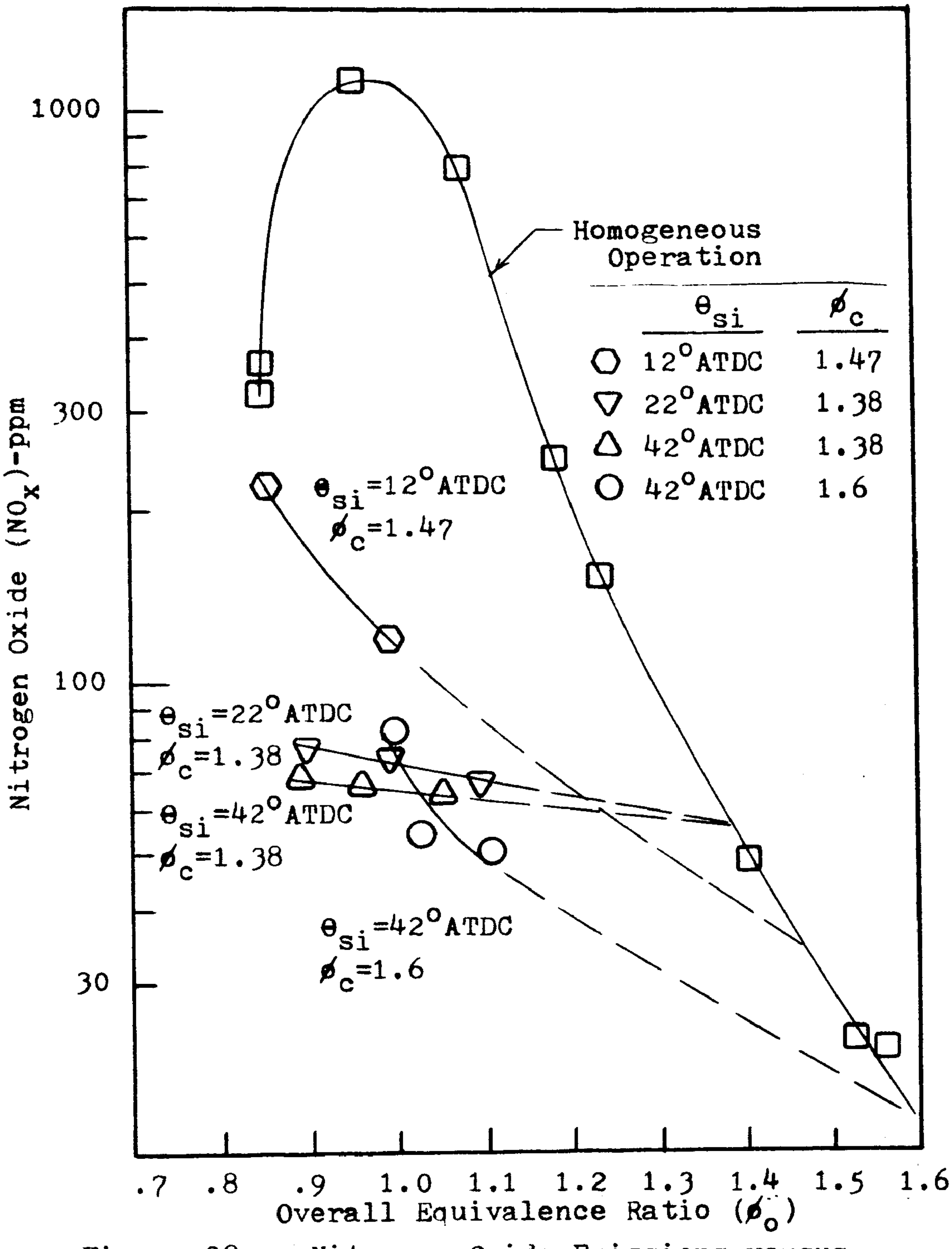


Figure 28 Nitrogen Oxide Emissions versus Overall Equivalence Ratio for Various Air Injection Timings With a Large Diameter Nozzle

starting at 12° ATDC, the amount of NO_x formed increases markedly. When the air is injected earlier, it is injected into higher temperature products. Thus, in the process of mixing and reacting with the rich products, it is more likely to produce additional NO_x emissions.

For each of the air injection timings, the curves were connected to the homogeneous operating point corresponding to their carbureted equivalence ratio. The point on the homogeneous curve represents the amount of NO_x which is formed during the period of rich combustion. The injection of air will reduce the overall equivalence ratio and result in additional NO_x formation. It is possible to operate at different points along the curve by changing the amount of air which is injected.

One set of data was obtained at a carbureted equivalence ratio of about 1.6 and with air injection starting at 42° ATDC. Although the NO_x formed by the carbureted charge is lower than that formed by $\phi_c = 1.4$ the exhaust NO_x is about the same. The slope of the curve is much steeper than the corresponding curve at $\phi_c = 1.4$ with air injection starting at 42° ATDC. The difference in the shape of the curves can be explained by considering the differences in the amounts of air which must be injected. To reach an overall stoichiometric equivalence ratio, it is necessary to inject 40% more air into the combustion chamber if the carbureted equivalence ratio is 1.4 but 60% more air if the carbureted equivalence ratio is 1.6. Since the total charge will be substantially greater, the cylinder pressure will be greater. The relative increase in pressure will compress the products of combustion and increase their temperature. The higher temperature will result in more rapid nitric oxide formation.

5. Carbon Monoxide for Large Diameter Nozzle

Presented in Figure 29 is a plot of the exhaust gas carbon monoxide versus overall equivalence ratio. The homogeneous operating points, indicated by squares, are connected by the solid line. The other points, representing air injection operation, are not connected because they are very near the homogeneous curve.

These results show that the injected air has sufficient time to mix and react with the products of the rich charge. The combustion is completed to the same degree as homogeneous operation with the same overall equivalence ratio. Under these air injection operating conditions, the exhaust emissions of carbon monoxide are mainly a function of overall equivalence ratio.

6. Hydrocarbon Emissions Large Diameter Emissions

The hydrocarbon emissions are presented in Figure 30. For operation with air injection, the hydrocarbon level is substantially below that of the homogeneous

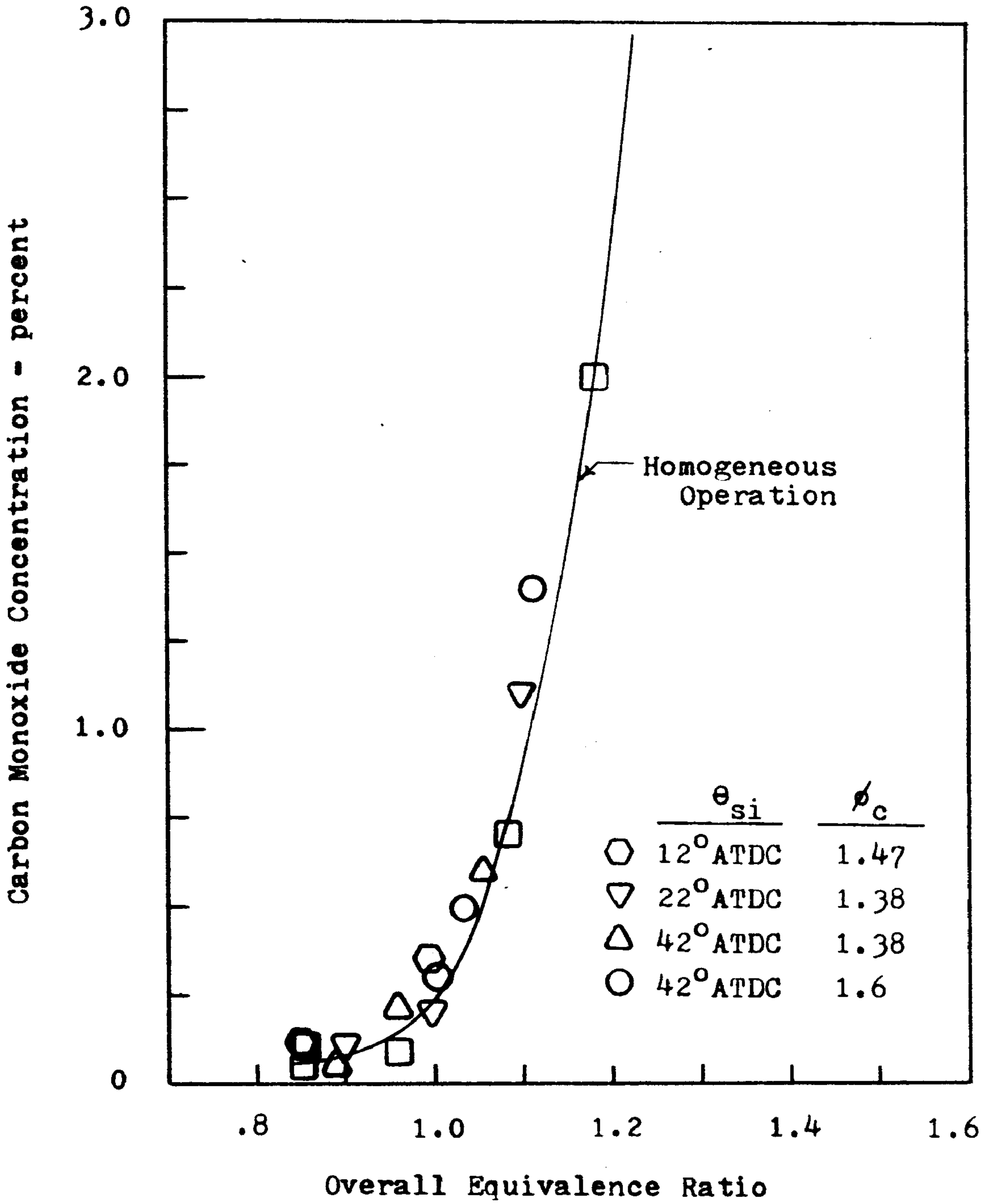


Figure 29 Carbon Monoxide Emissions versus Overall Equivalence Ratio for Various Air Injection Timings With a Large Diameter Nozzle

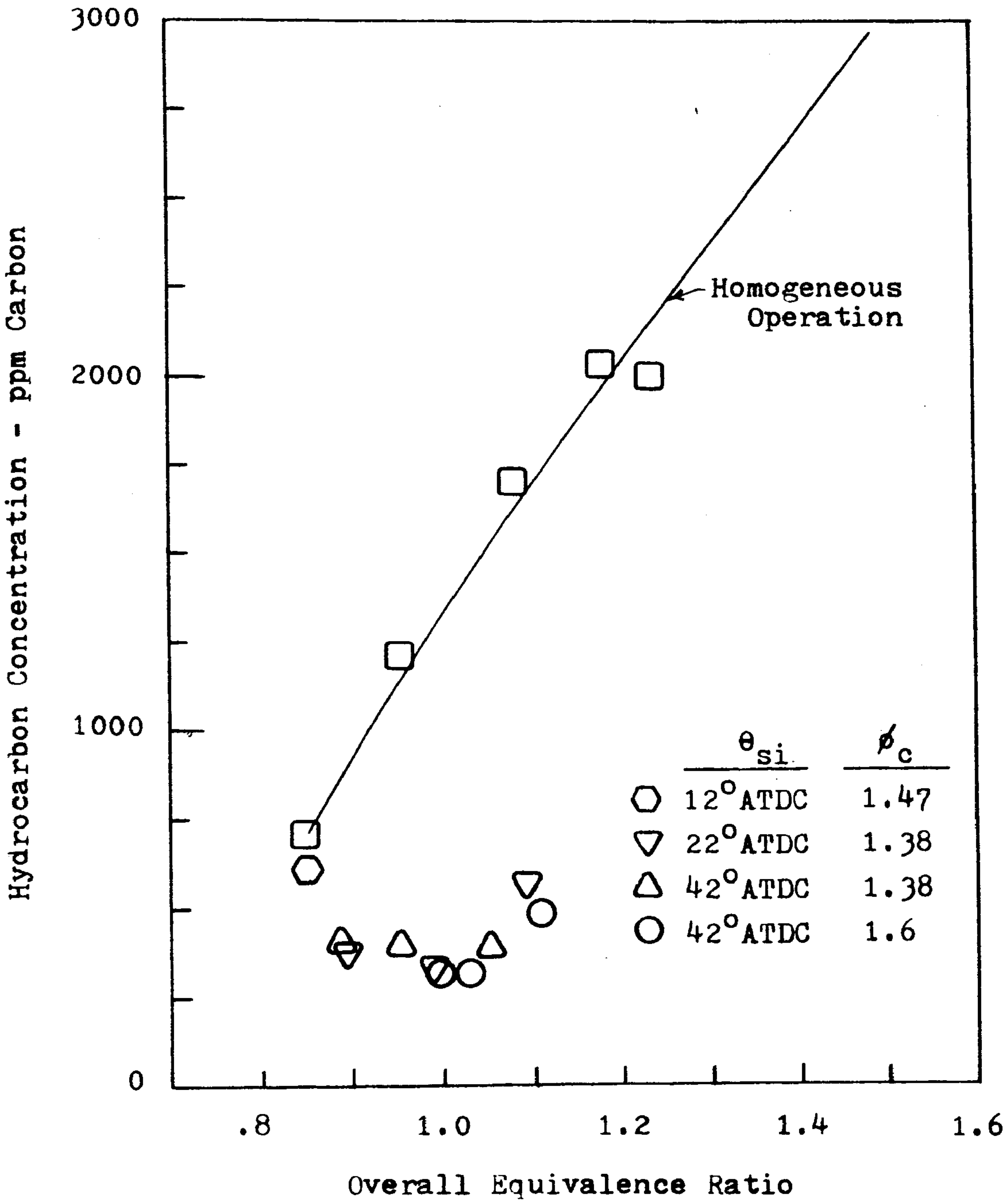


Figure 30 Hydrocarbon Emissions versus Overall Equivalence Ratio for Various Air Injection Timings With a Large Diameter Nozzle

operation with the same overall equivalence ratio. The most likely explanation for the lower hydrocarbons is that the turbulence and general increase in the motion of the charge due to air injection reduces the thickness of the quench layer.

7. Engine Efficiency for Large Diameter Nozzle

Because the injection air is compressed separately from the engine the measured output power does not include the work required to compress the injection air. An estimate of the work required to compress the injected air is made and subtracted from the measured work. As a result the efficiency can be presented as measured values or corrected values. In Figure 31 both the measured and corrected indicated enthalpy efficiencies are presented.

The homogeneous efficiencies reach a maximum at an overall equivalence ratio of .95. The efficiency is lower for leaner operation because the engine experiences partial misfire. The homogeneous efficiency is lower at greater equivalence ratios because of the incomplete combustion of the fuel.

Engine operation with air injection shows a general increase in measured efficiencies with leaner equivalence ratios. If no air were injected, measured efficiency would be equal to that of the homogeneous operation at

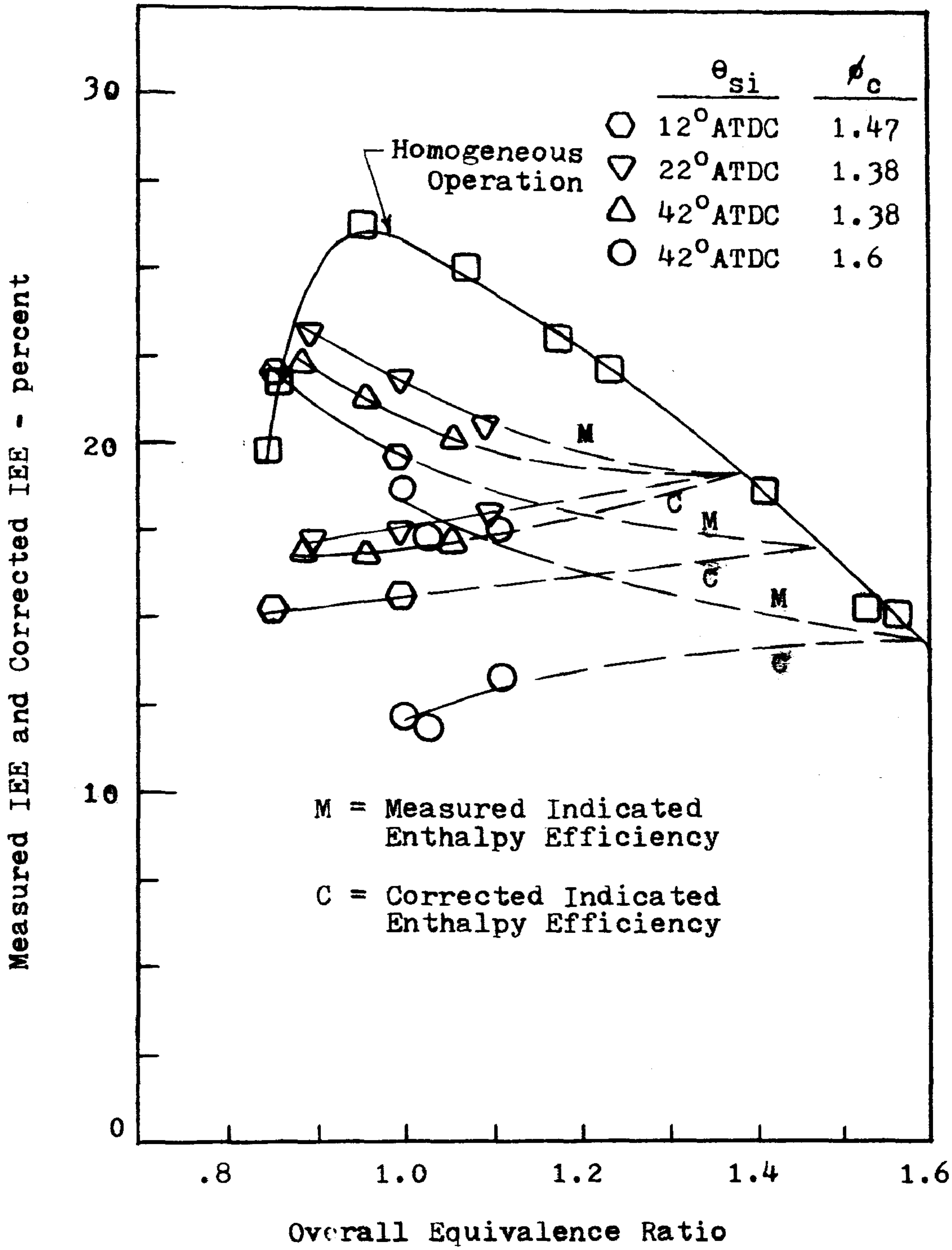


Figure 31 Measured and Corrected Indicated Enthalpy Efficiency versus Overall Equivalence Ratio for Various Air Injection Timings With a Large Diameter Nozzle

the carbureted equivalence ratio. As the amount of injected air increases the measured efficiency increases and the overall equivalence ratio becomes leaner. The addition of high pressure air during the expansion stroke will result in more measured work. Also, when the air mixes with the rich products, it will tend to finish the combustion process, causing higher pressures and an increase in measured work. Although the measured efficiency is increased with air injection, it is less than homogeneous operation at the same overall equivalence ratio. The lower measured efficiency is due to combustion being completed late in the expansion stroke. The later in the expansion stroke the combustion occurs the smaller will be the amount of expansion work.

When the measured work is corrected by subtracting the estimated work of compressing the injected air, the resulting corrected efficiency tends to decrease with increasing amounts of air injection. Apparently the increased amount of work produced by the injected air is less than work required to compress the air. A large compression ratio is required to bring atmospheric air to injection pressure and only a small expansion ratio is available in the engine. If the air did not react in the combustion chamber and only expanded, the lower expansion ratio would limit the work extracted from the air to something less than the compression work.

8. Discussion of Results for Large Diameter Nozzle

The low values of carbon monoxide and hydrocarbons lead to the conclusion that the injected air does mix and react to complete the combustion process. However, the measured indicated enthalpy efficiency shows only slightly more output power as a result of air injection. It appears that the combustion occurs too late in the expansion to contribute much to output power. Either the mixing is too slow or the air is injected too late in the expansion. It was decided that the mixing should be increased by changing the nozzle to a smaller diameter to obtain a higher velocity for more rapid mixing.

The large diameter nozzle was picked initially because of the lower pressure drop across it. In a practical engine it would be desirable to have the minimum pressure drop across the nozzle. The engine is going to have to provide the work required to inject or mix the air with the rich products of combustion.

E. SMALL DIAMETER NOZZLE TEST RESULTS AND DISCUSSION

1. Test Conditions for Small Diameter Nozzle

The small diameter nozzle has a diameter of .0935 inches (2.37 mm) and is directed at an angle of 45 degrees from a perpendicular to the center line of the cylinder in a plane parallel to the piston's top surface. The diameter was selected to be equal to that of the flow restriction placed between the nozzle and the air injection valve seat for the large diameter nozzle tests. In this way, flow through the valve would be similar for both types of nozzles.

The engine was operated near 800 RPM with the maximum carbureted air flow rate. Compression ratio was maintained at seven. The spark timing was varied in attempts to operate at the maximum advance for best timing, but these conditions were not always obtained. The injection air temperature was uncontrolled and was above room temperature due to temperature of the engine. The carbureted fuel-air mixture was heated for some of the test conditions by a strip heater wrapped around the inlet pipe.

For the small diameter nozzle, some of the operation was with air injection before combustion. These points were run to show the entire range of air-injection timings.

The values for important variables which changed during the tests are shown in the figures or in Table II.

2. Summary of Results for Small Diameter Nozzle

1) The nitrogen oxide emissions for engine operation with air injection usually increase with leaner overall equivalence ratios.

2) The nitrogen oxide emissions for engine operation with air injection when compared to homogeneous operation at the same overall equivalence ratio are much lower with air injection after combustion, are about the same with air injection before and rapidly increase with earlier air injection during combustion.

3) The carbon monoxide emissions for engine operation with air injection are usually the same as those of homogeneous operation at the same overall equivalence ratio, but greater for air injection during combustion, very early in the compression, or very late in the expansion.

4) The hydrocarbon emissions for engine operation with air injection are usually lower than homogeneous operation at the same overall equivalence ratio if leaner than 1.15, but greater for air injection during combustion or or very early in the compression.

5) The measured indicated enthalpy efficiency for engine operation with air injection is always less than homogeneous operation at the same overall equivalence ratio. The measured efficiency for air injection operation is low with air injection after combustion, approaches the homogeneous operation with air injection before combustion and is generally low with air injection during combustion.

6) The corrected enthalpy efficiency for engine operation with air injection is always much lower than homogeneous operation at the same overall equivalence ratio.

7) The measured indicated mean effective pressure for engine operation with air injection is higher with lean operation and with early air injection. Air injection after combustion has low IMEP. With air injection during combustion the measured IMEP is at a minimum.

8) When air is injected before combustion the combustion process is rapid, the cycle to cycle pressure variations are small, and the peak pressure is greater than the corresponding homogeneous operation.

9) When air is injected very early in the compression stroke (128° BTDC) the combustion process will result in knock.

10) Air injection can cause up to 40% more heat transferred to the cooling water on the basis of BTU per pound of charge.

3. Nitrogen Oxide Emissions for Small Diameter Nozzle

The nitrogen oxide emissions associated with homogeneous operation are shown in Figure 32. The data represents a band rather than a single line because of the variation in the spark timing, same operation was with

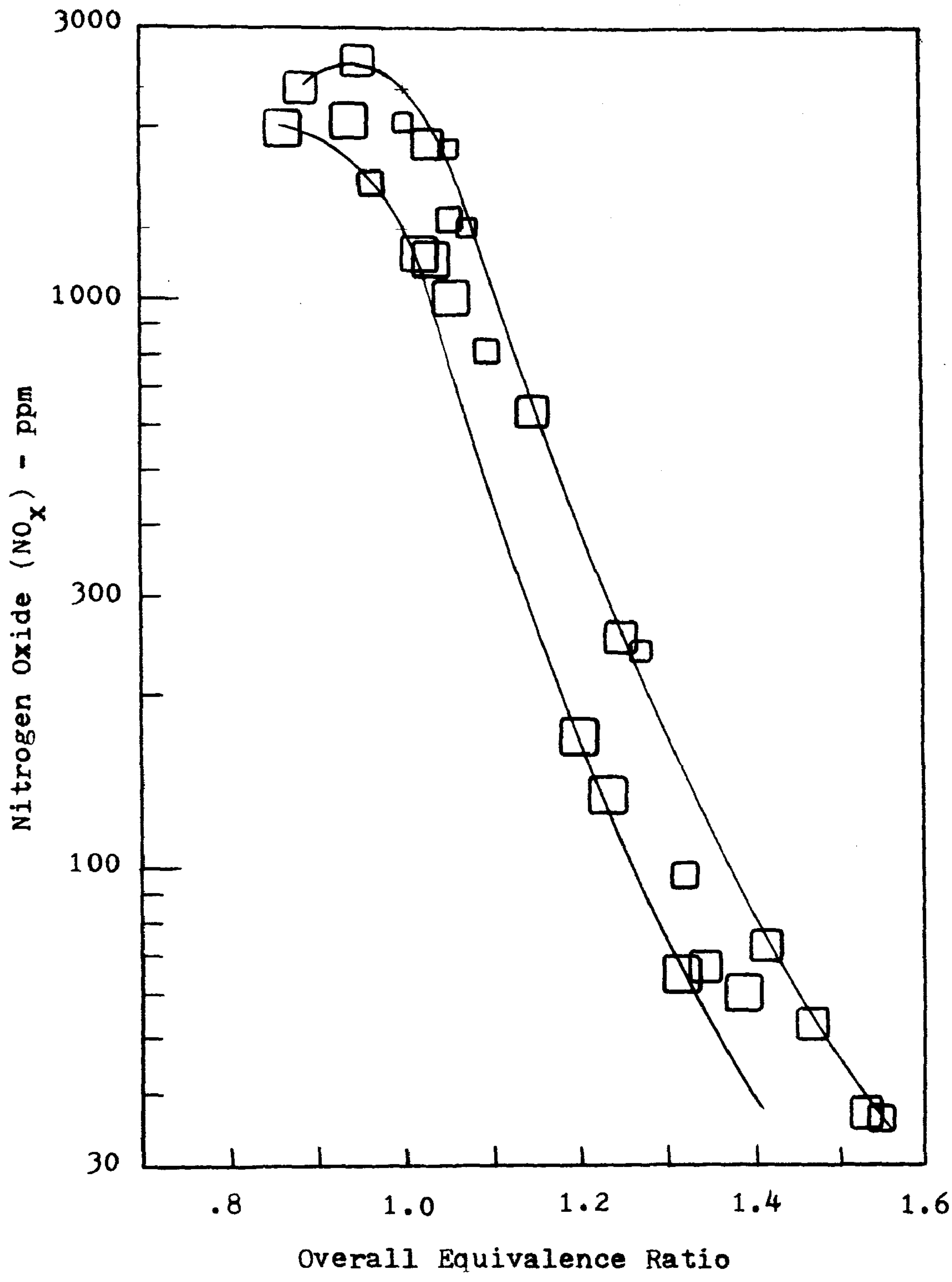


Figure 32 Nitrogen Oxide Emissions for Various Homogeneous Operating Conditions

engine inlet pipe heated, the accumulation of small measurement errors, and the small variation in operating conditions. The maximum nitrogen oxide emissions occur at a slightly lean equivalence ratio as would be expected. The engine was not operated much leaner than the maximum NO_x because of the start of misfiring.

In Figure 33 a few typical air injection operating conditions are shown along with the band of homogeneous operation as a function of overall equivalence ratio. This same set of data will be shown for the other results versus overall equivalence ratio. In general, starting air-injection earlier in the cycle results in large nitrogen oxide emissions. With air injection considerably before combustion as represented by the start of air injection at 38^{o} BTDC and 88^{o} BTDC the nitrogen oxide emissions correspond to the levels expected from homogeneous operation at the same overall equivalence ratio. When air injection is very late in the cycle (92^{o} ATDC) leaner equivalence ratios cause a reduction in the nitrogen oxide concentration. The air injection occurs when the temperatures are so low that little additional nitrogen oxide is formed and the air reduces the concentration by dilution.

In order to better show the influence of the timing of air injection on nitrogen oxide and other results,

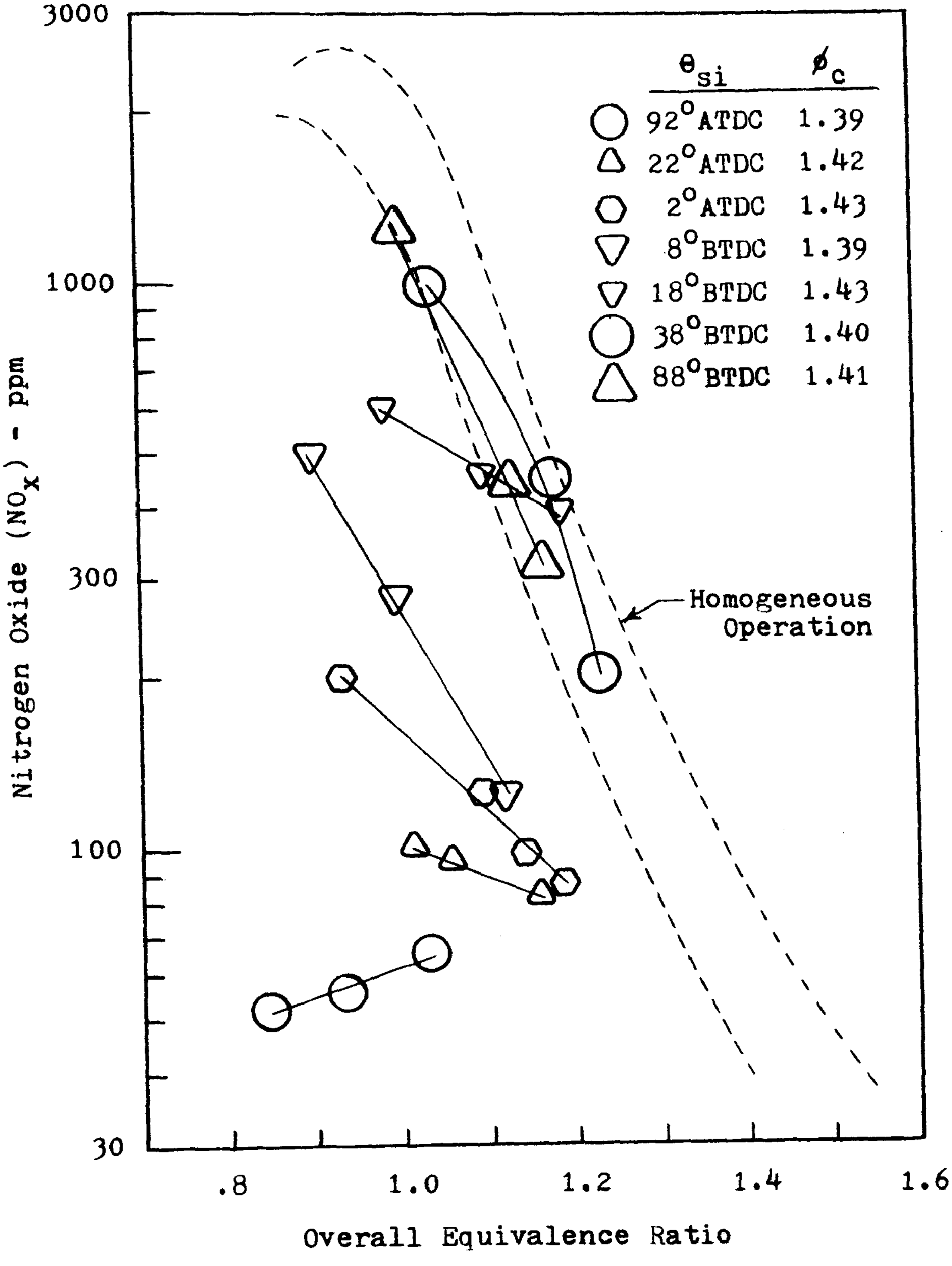


Figure 33 Nitrogen Oxide Emissions versus Overall Equivalence Ratio for Selected Air Injection Operating Conditions With a Small Diameter Nozzle

they will be plotted versus start of air injection. Since various operating conditions were run at one air injection timing the points shown will correspond to an overall stoichiometric mixture. The magnitude of any specific point was obtained by plotting the variable versus overall equivalence ratio, fitting a curve to the data and noting the intersection of that curve with a stoichiometric mixture. A summary of the extrapolated stoichiometric results is shown in Table II. Generally the carbureted equivalence ratio was about 1.4. A carbureted equivalence ratio is estimated for each point in Table II. Also shown in Table II is the spark timing and start of air injection.

The nitrogen oxide emissions versus the start of air injection are presented in Figure 34. Generally NO_x is low for air injection after combustion and high for air injection before combustion. When the air injection occurs near top dead center during combustion, the nitrogen oxide emissions are very sensitive to changes in the air injection timing and to changes in the other variables. The spread in the nitrogen oxide emission at 2^o ATDC and 8^o BTDC attest to the sensitivity of this region.

It was particularly difficult to adjust the spark for the best torque with air injection near top dead

TABLE II

Summary of Results for Air Injection Operation Extrapolated to a Stoichiometric Fuel-Air Mixture

Symbol	θ_{si} (deg.)	θ_{sp} (deg.)	ϕ_c	NO_x (ppm)	CO (%)	HC (ppm-C)	Meas. IEE (%)	Meas. IMEP (psi)
○	92	-26	1.39	59	2.6	1150	21.2	165
△	22	-30	1.42	100	.5	720	24.5	133
○	22	-30	1.60	97	.55	950	22.8	132
△	2	-20	1.44	95	>3.0	1850	22.5	120
○	2	-20	1.38	115	3.0	1800	21.5	113
⬡	2	-30	1.43	160	.45	780	24.5	130
◇	2	-40	1.35	150	.40	750	24.2	120
⬡	-8	-10	1.36	630	1.4	1300	24.2	122
▽	-8	-10	1.39	260	>3.0	2030	21.6	111
⬡	-8	-20	1.40	285	1.5	1100	23.4	120
◇	-8	-22	1.42	185	2.4	1550	22.3	120
▽	-18	-30	1.43	580	1.0	700	22.7	123
◇	-28	+10	1.44	1350	.92	1150	21.1	137
◇	-28	-6	1.24	1250	1.3	1250	26.0	128
○	-38	-8	1.40	1200	.6	640	24.8	129
⬡	-48	0	1.39	1650	.85	1220	25.8	136
△	-68	-6	1.42	1250	1.3	1000	25.0	132
△	-78	-5	1.45	1200	.5	1350	26.5	142
△	-88	-10	1.41	1200	.5	630	25.1	134
▽	-128	-6	1.42	260	2.6	1900	22.3	128

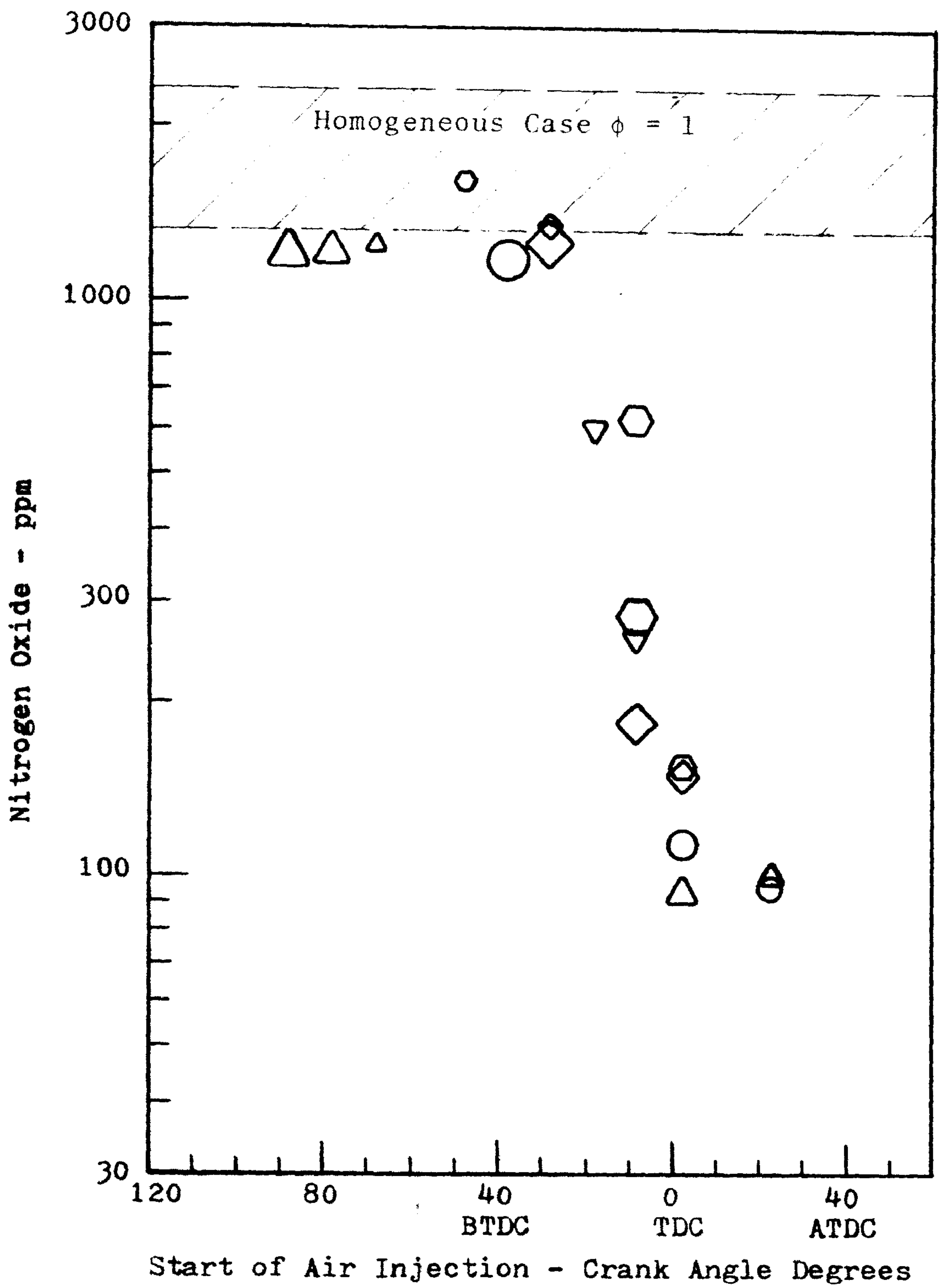


Figure 34 Nitrogen Oxide Emissions versus Start of Air Injection for Overall Stoichiometric Equivalence Ratio With a Small Diameter Nozzle

center. When the air is injected before combustion the best timing is about 6^{o} BTDC probably because of the greater turbulence. When the air is injected after combustion the best spark timing is about 25^{o} BTDC. Also changes in the spark timing will affect the timing of the pressure peak in the cylinder, which in turn influences the injection air pressure. The injection air pressure takes some time to stabilize. Thus, if one were to adjust the spark timing by merely observing the engine torque it would be easy to select the wrong timing. Observation of the cylinder pressure was used to determine the spark timing near top dead center. This method requires judgment as to the most appropriate position of the pressure peak with respect to top dead center. The adjustment of spark timing by observing the cylinder pressure is better than observing the engine torque, but not exact.

The spread in the nitrogen oxide data with air injection starting at 2^{o} ATDC can be explained mainly by the spark timing. The two points with the greatest nitrogen oxide emission have spark timings of 30^{o} BTDC and 40^{o} BTDC; from the pressure trace the spark is too advanced. The two other points have spark timings of 20^{o} BTDC; from the pressure trace the spark is too retarded. The advanced timing causes higher pressures and temperature which result in more nitrogen oxide emissions. The retarded timing causes lower pressures and temperatures which result in low nitrogen oxide emissions.

The two points with 20° BTDC spark timing with start of air injection at 2° ATDC differ in their carbureted equivalence ratio. The low nitrogen oxide point also corresponds to the richer carbureted equivalence ratio. Since the richer carbureted equivalence ratio would produce less nitrogen oxide before air injection it is reasonable to expect a lower final nitrogen oxide emission.

The same type of reasoning can be used to explain the spread in the nitrogen oxide data points with air injection at 8° BTDC. The more advanced timing results in greater NO_x emissions and richer carbureted equivalence ratio result in lower NO_x emissions.

When air is injected before combustion the nitrogen oxide concentration corresponds to that of homogeneous operation with the same overall equivalence ratio. Apparently the injected air mixes quickly to form a somewhat homogeneous mixture at the overall equivalence ratio before combustion.

4. Carbon Monoxide Emissions for Small Diameter Nozzle

The carbon monoxide emissions for homogeneous operation are presented in Figure 35. A curve drawn through

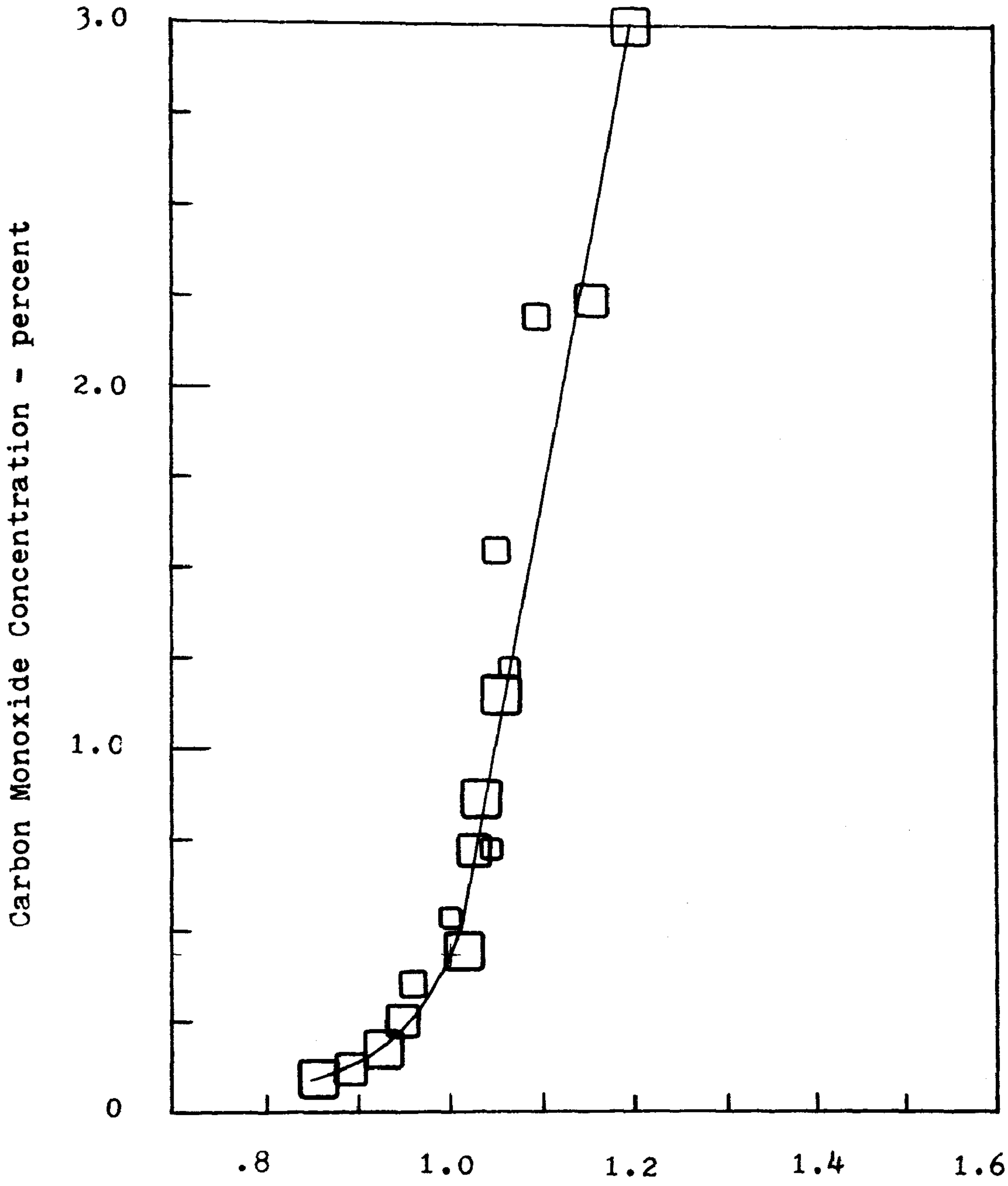


Figure 35 Carbon Monoxide Emissions versus Overall Equivalence Ratio for Homogeneous Operation

the data points describes the homogeneous carbon monoxide emissions. For rich operation the carbon monoxide levels are high while for lean operation the carbon monoxide emissions are low.

The carbon monoxide emissions for the typical set of air injection operating conditions are presented in Figure 36 along with the dotted line corresponding to the homogeneous operation. All of the data points fall near the homogeneous curve except for operation with the start of air injection at 8° BTDC and 92° ATDC. High values of carbon monoxide are expected with very late air injection (92° ATDC) because of the short time available for combustion and because of the low temperatures

Many points with air injection starting near top dead center have greater carbon monoxide emissions than homogeneous operation as seen in Figure 37. It is possible that air being injected during the period of combustion, as is the case for these points, can quench and disturb the combustion process. Probably the regions of incomplete combustion do not mix and as a result appear in the exhaust.

5. Hydrocarbon Emissions for Small Diameter Nozzle

The hydrocarbon emissions for homogeneous operation are shown in Figure 38. Two factors account for the spread in hydrocarbon emissions. Part way through the

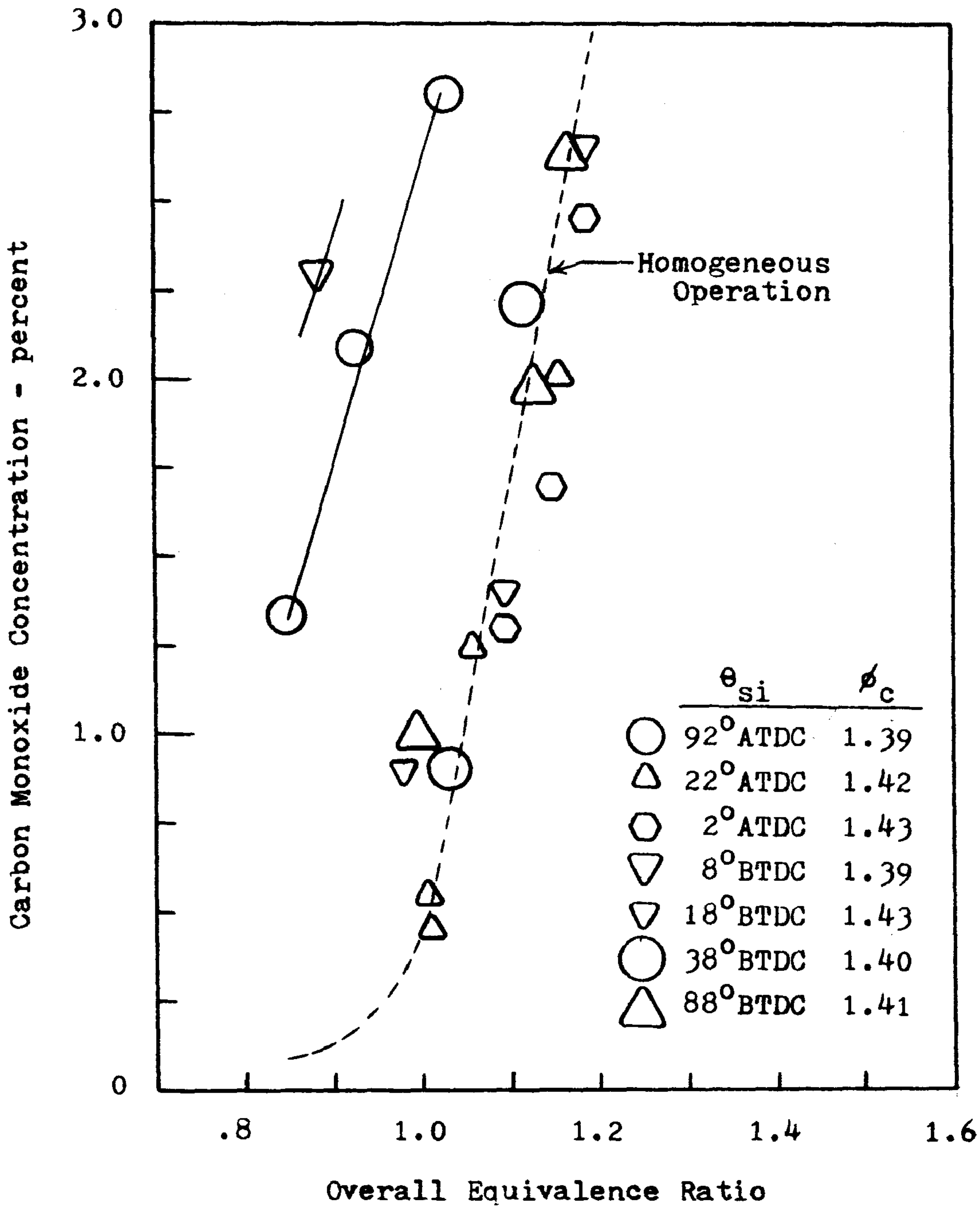


Figure 36 Carbon Monoxide Emissions versus Overall Equivalence Ratio for Air Injection Operation With a Small Diameter Nozzle

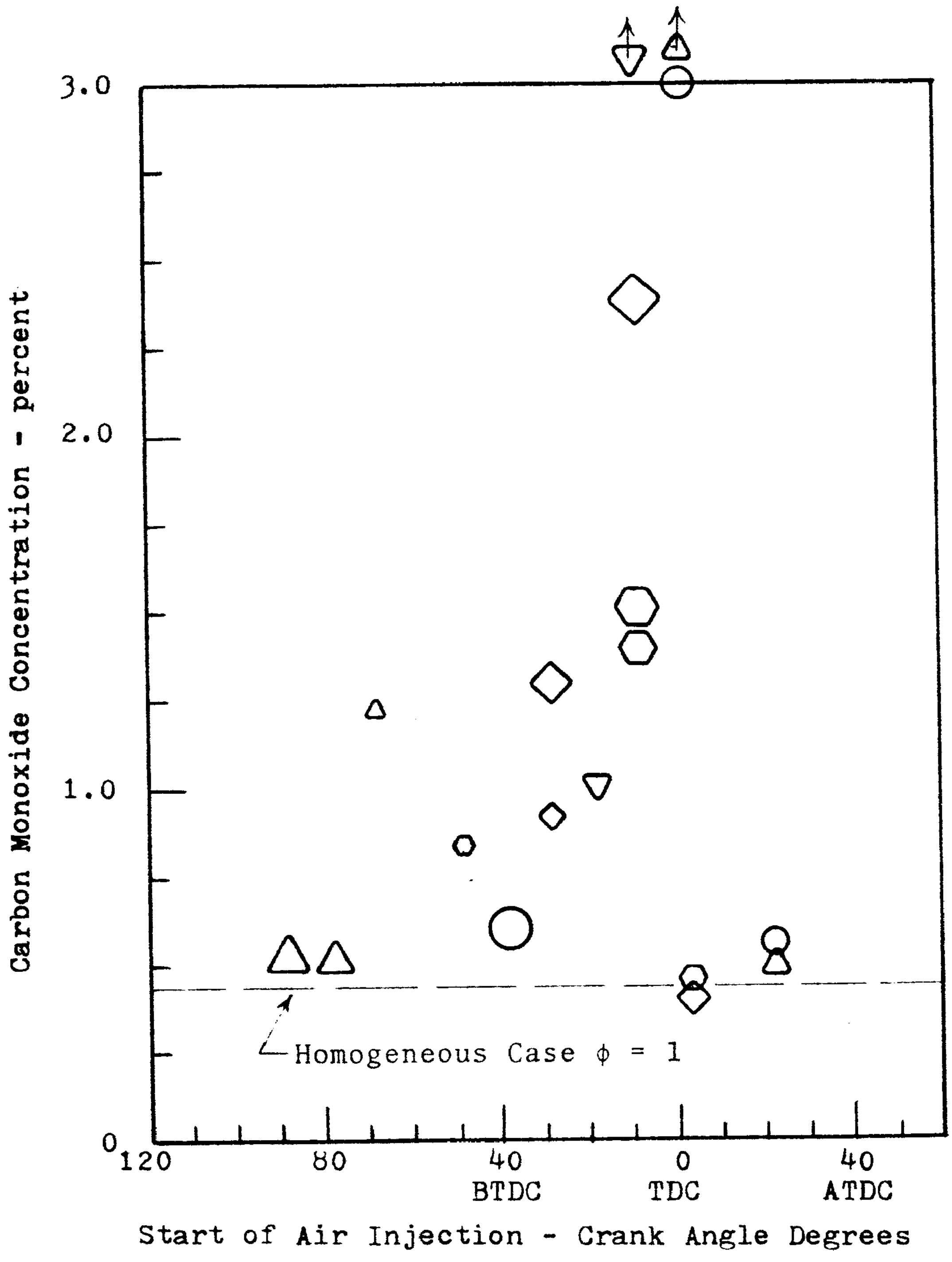


Figure 37 Carbon Monoxide Emissions Versus Start of Air Injection for Overall Stoichiometric Equivalence Ratio With a Small Diameter Nozzle

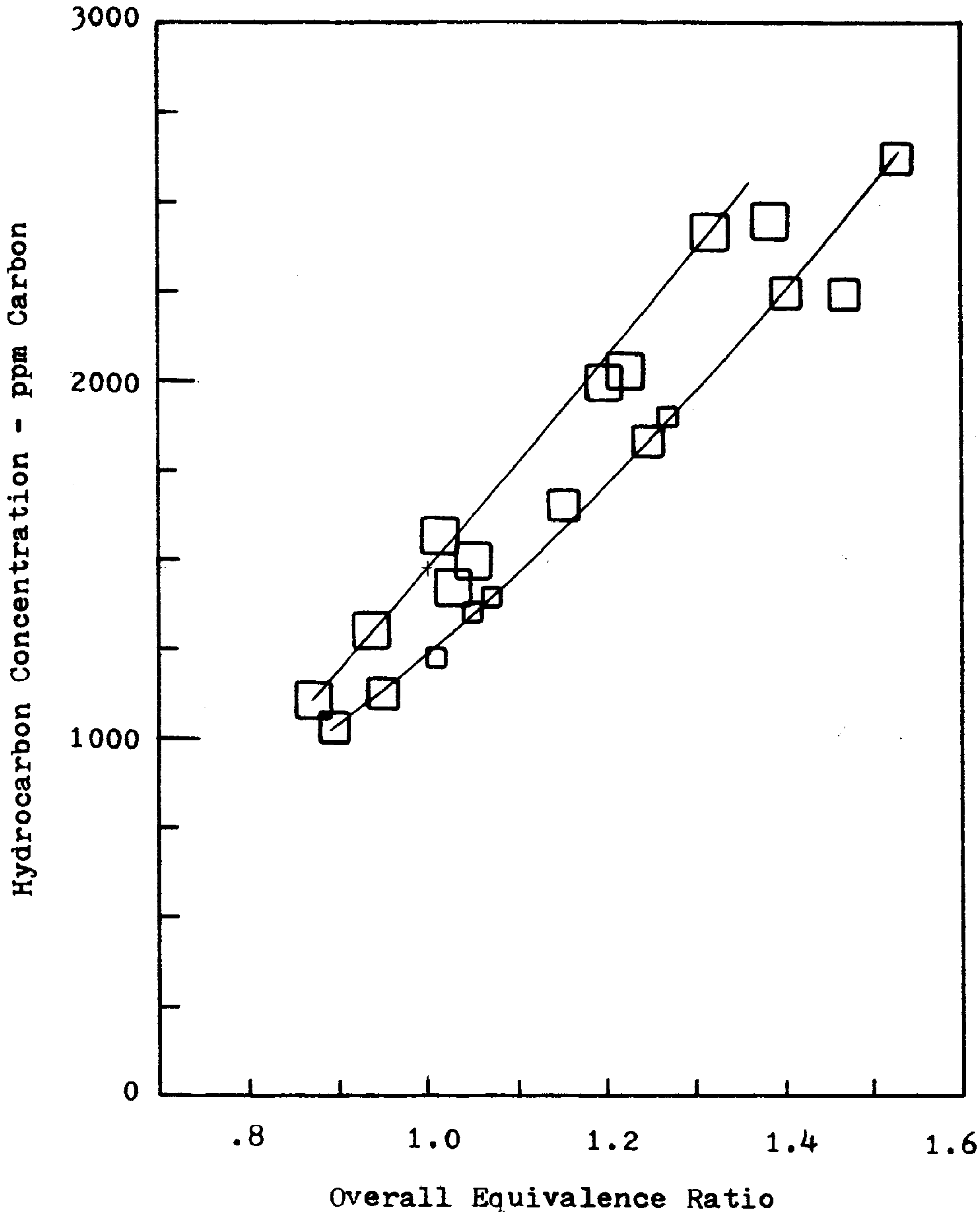


Figure 38 Hydrocarbon Emissions versus Overall Equivalence Ratio for Homogeneous Operation

testing program it was found that the wall of the engine inlet pipe was wetted by fuel and there existed the possibility of liquid fuel entering the inlet valve. For this reason the walls of the inlet manifold were heated. As a result of the heater the hydrocarbon emissions dropped. Another factor is the improvement of the calibration procedure of the hydrocarbon analyzer. The hydrocarbon analyzer is sensitive to the flow rate of sample and calibration gas. These two flow rates should be the same. A pressure regulator in the instrument is used to adjust the scale reading to correspond to the hydrocarbon concentration of calibration gas. It was found that changes in the calibration gas pressure supplied to the instrument would change the calibration gas flow rate. A procedure was devised to measure the calibration gas flow rate with the carbon monoxide flow meter and to set the flow rate equal to the sample flow rate.

The hydrocarbon emissions for the typical set of air injection timings are shown in Figure 39 along with the dotted lines representing the band of homogeneous operating results. With the exception of air injection starting at 8° BTDC the data shows lower hydrocarbon emissions for air injection operation with overall equivalence ratios leaner than 1.15. With air injection starting very late in the expansion (92° ATDC), the

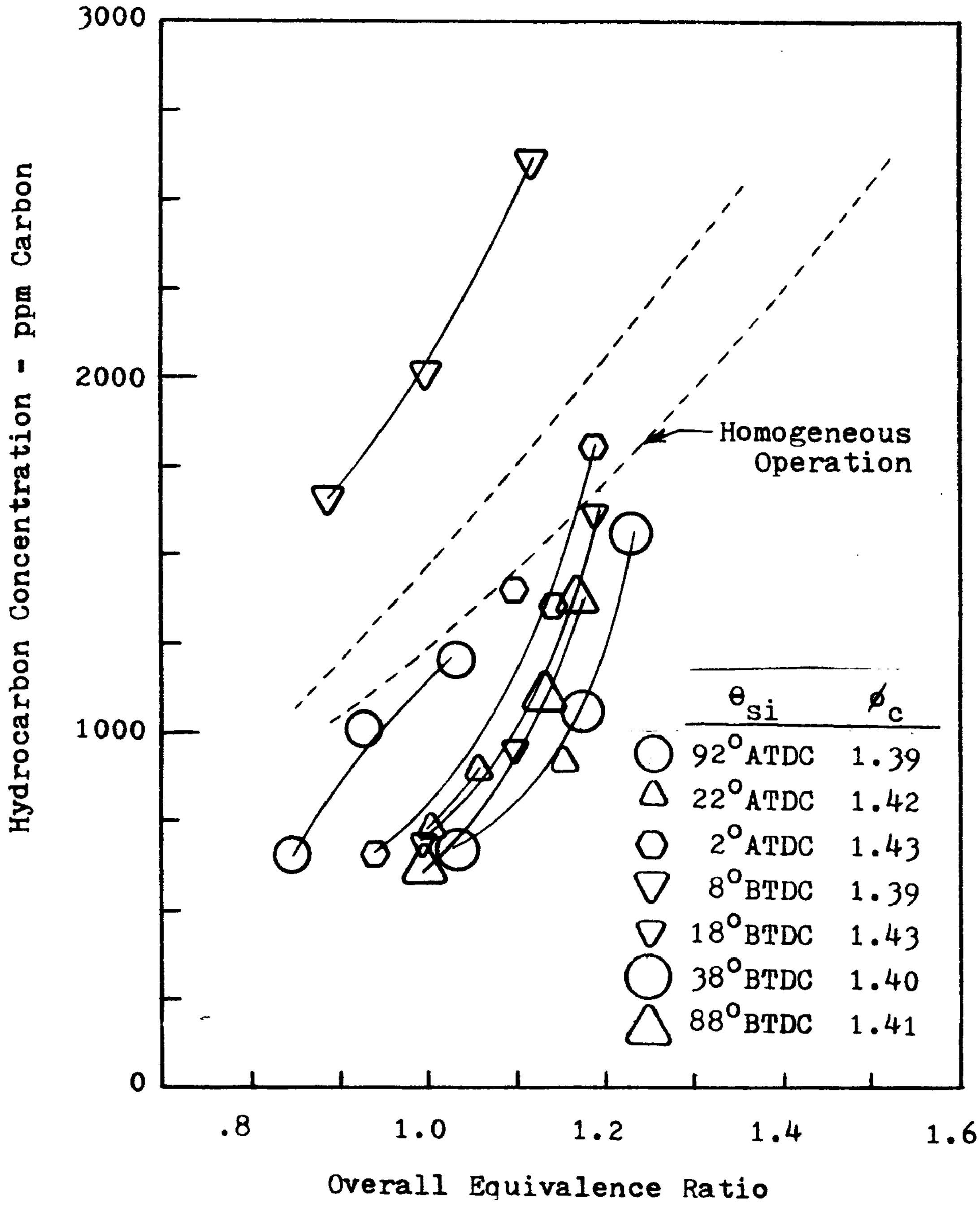


Figure 39 Hydrocarbon Emissions versus Overall Equivalence Ratio for Air Injection Operation with a Small Diameter Nozzle

hydrocarbons are nearer the homogeneous operation. The greater turbulence in the chamber reduces the hydrocarbon envelope and consequently the hydrocarbon emissions.

Air injection starting near top dead center results in greater hydrocarbon emissions as shown in Figure 40. The disturbance of the combustion process by air injection would account for the greater hydrocarbons and would be consistent with the greater carbon monoxide emissions found for the same injection timing.

6. Engine Efficiency for Small Diameter Nozzle

Shown in Figure 41 is the measured indicated enthalpy efficiency as a function of the overall equivalence ratio for the homogeneous operation. The efficiency decreases with richer equivalence ratios. The spread in the data is primarily due to variations in the timing and the accuracy of reading the dynamometer force. The low speed of the engine and the cycle to cycle variations of the engine resulted in a pulsating output torque. The needle on the dynamometer scale would vibrate requiring an estimate of its average value.

The measured indicated enthalpy efficiency for the typical set of air injection data is shown in Figure 42 with the homogeneous operation shown by dotted lines. Earlier air injection as well as leaner overall operation tend to increase the measured indicated enthalpy efficiency.

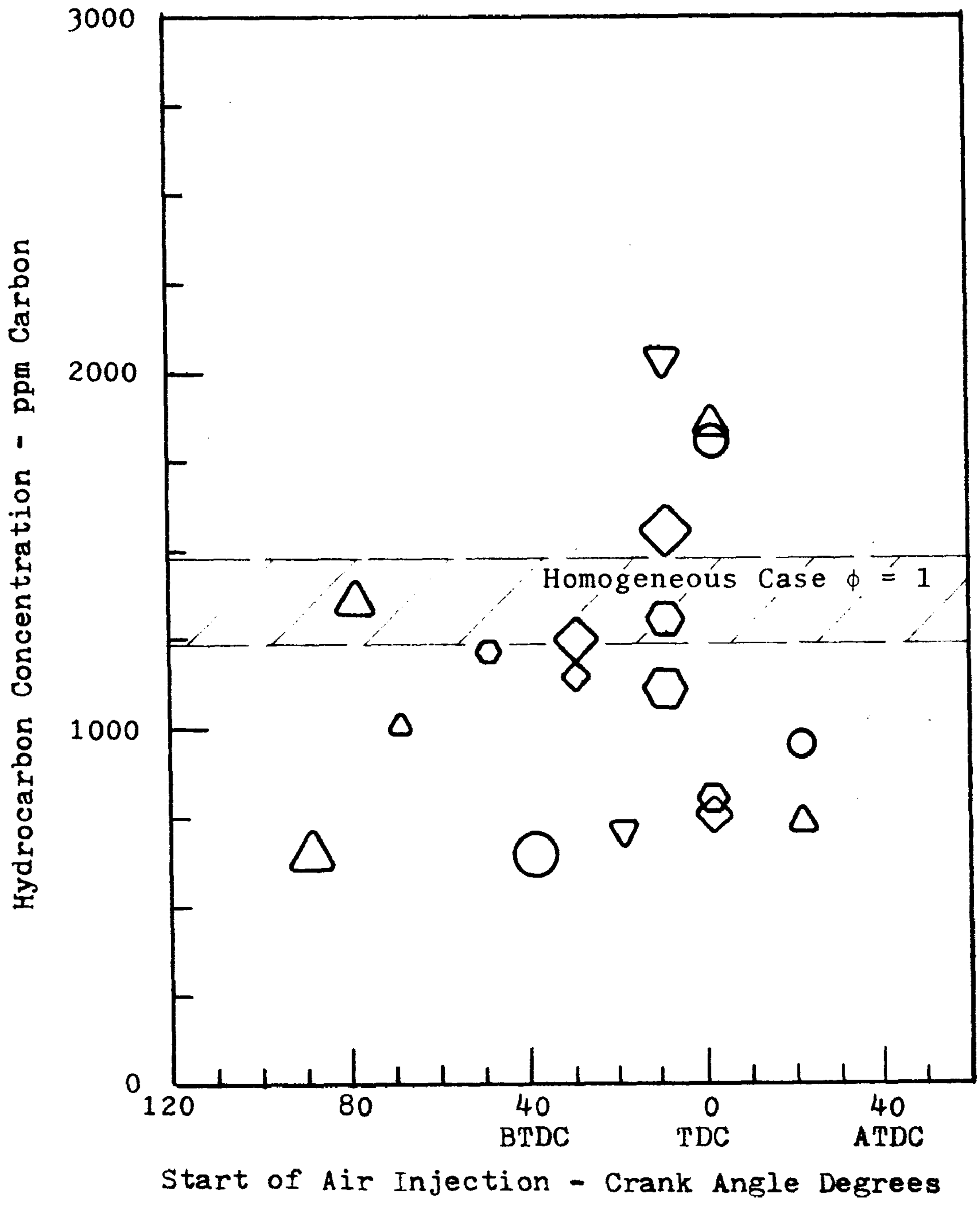


Figure 40 Hydrocarbon Emissions versus Start of Air Injection for Overall Stoichiometric Equivalence Ratio With a Small Diameter Nozzle

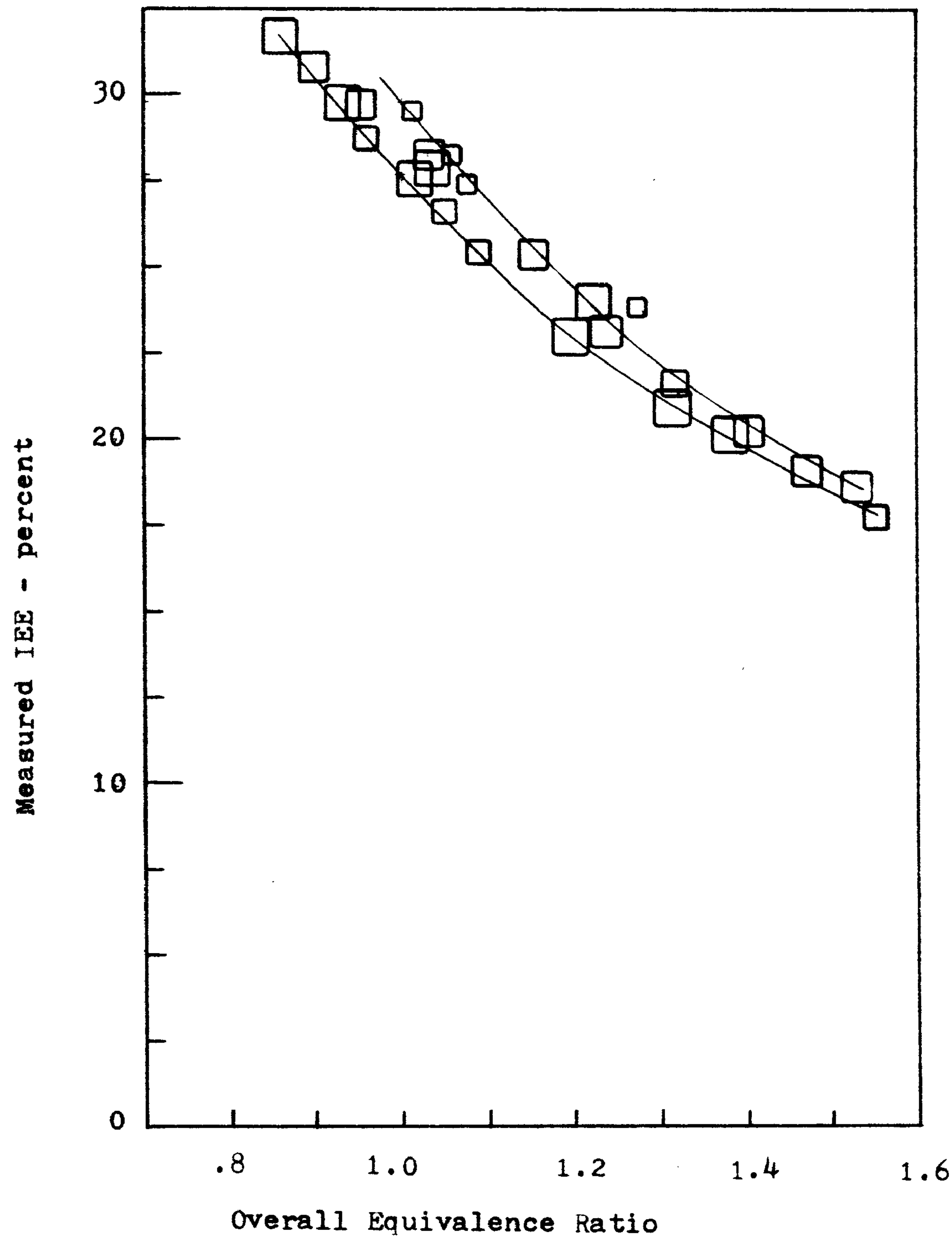


Figure 41 Measured Indicated Enthalpy Efficiency versus Overall Equivalence Ratio for Homogeneous Operation

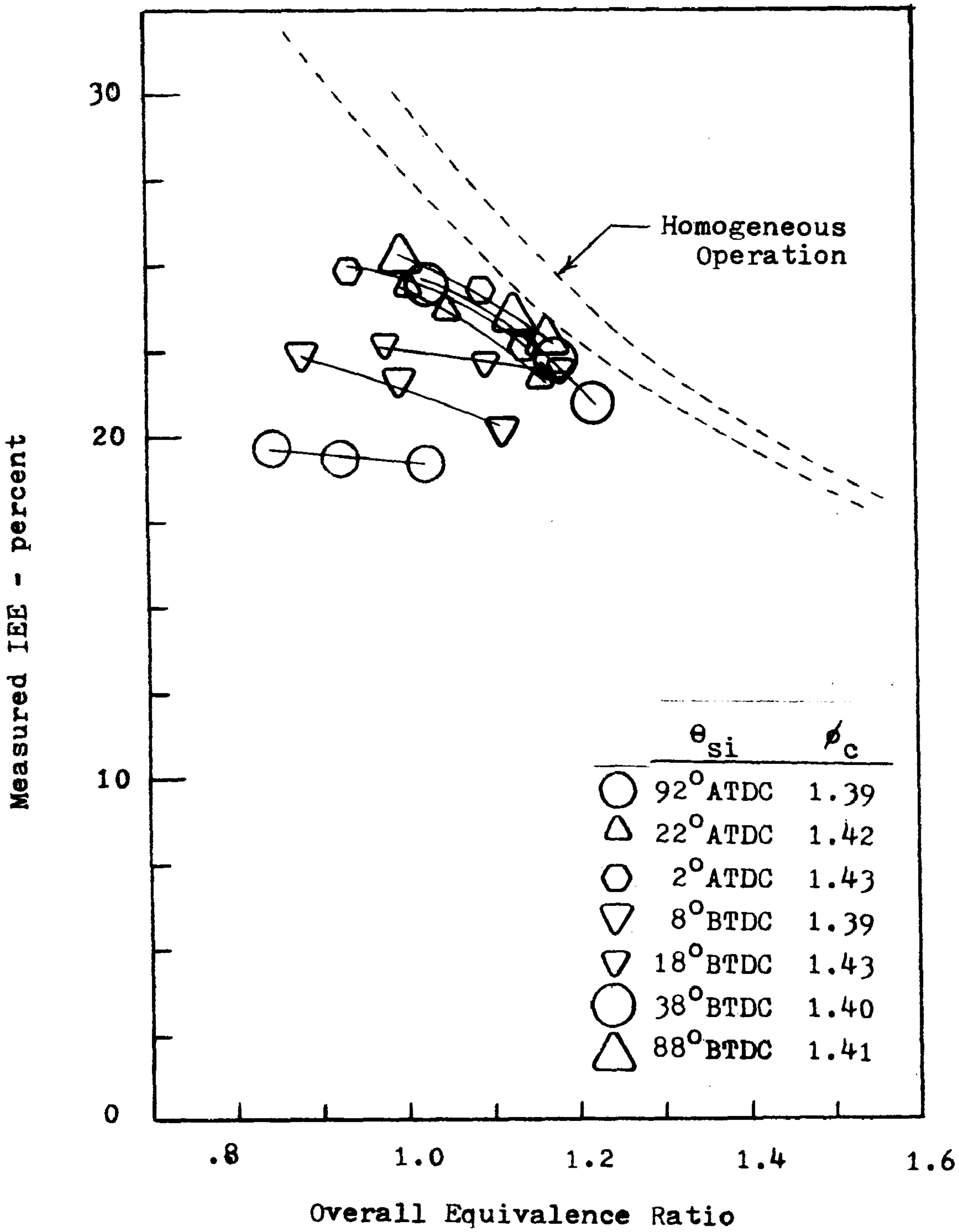


Figure 42 Measured Indicated Enthalpy Efficiency versus Overall Equivalence Ratio for Air Injection Operation With a Small Diameter Nozzle

Note that with injection before combustion, the measured efficiency is less than the corresponding homogeneous operation. The difference in efficiencies is probably due to greater heat transfer caused air injection induced charge motion. An increase of 40% in the heat transferred to the cooling water on the basis of BTU per pound charge was measured for operation with air injection.

In Figure 43 the corrected indicated enthalpy efficiency is shown as a function of overall equivalence ratio. With injection before combustion the corrected efficiency is about equal to the efficiency of the homogeneous operation with the carbureted equivalence ratio. The estimated work required to compress the air is a substantial percentage of the engine output.

The measured indicated enthalpy efficiency is plotted versus the start of air injection in Figure 44. The greater measured efficiencies occur when the air is injected into the combustion chamber before ignition. The measured efficiency is larger because the injected air has time to mix with the rich charge before combustion thus allowing more fuel to burn nearer top dead center. When air injection occurs well into the expansion stroke little additional work can be obtained from the completion of combustion.

With the start of air injection at 2^{o} ATDC the spread in the efficiency can be attributed to the different spark

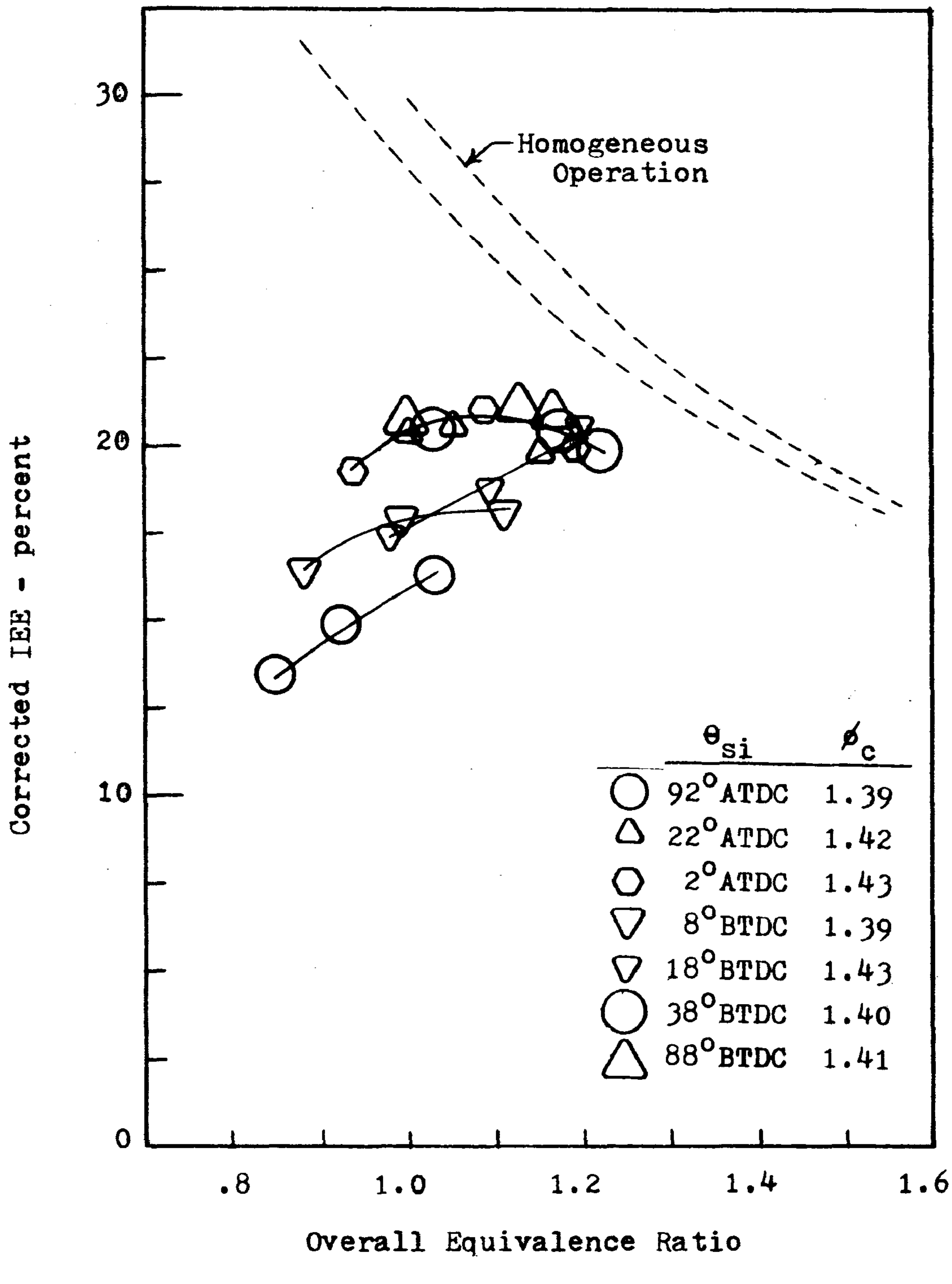


Figure 43 Corrected Indicated Enthalpy Efficiency versus Overall Equivalence Ratio for Air Injection Operation With a Small Diameter Nozzle

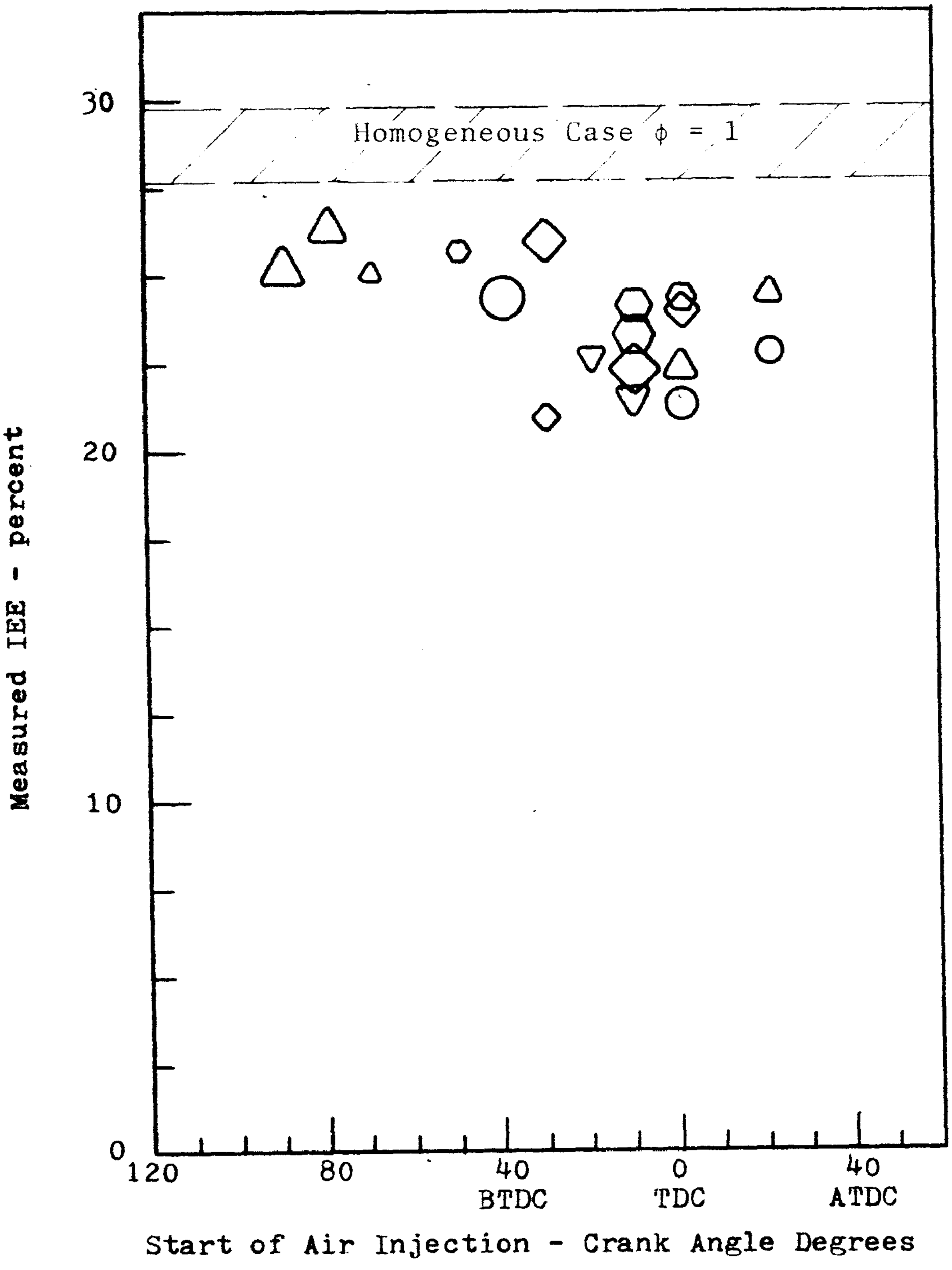


Figure 44 Measured Indicated Enthalpy Efficiency versus Start of Air Injection for Stoichiometric Equivalence Ratio With a Small Diameter Nozzle

timing. Spark timing of 30° BTDC and 40° BTDC have greater efficiency than the two points at 20° BTDC. The four points corresponding to air injection starting at 8° BTDC have a spread that can not be explained entirely by spark timing. The highest and lowest efficiency points both have a spark timing of 10° BTDC. The difference between the two points is due to the carbureted equivalence ratio. The higher efficiency point has the leaner carbureted equivalence ratio. It is reasonable to expect that leaner carbureted equivalence ratios would result in greater efficiencies.

The measured indicated enthalpy efficiencies are quite low for air injection during the combustion process. This result is consistent with the high CO and hydrocarbons for the same injection timing. If the injection of air disturbs the combustion process and cause incomplete combustion less energy will be available to do work.

7. Engine Power for Small Diameter Nozzle

The measured indicated mean effective pressure (IMEP) will be used as a measure of the specific power of the engine. Presented in Figure 45 is a plot of IMEP versus overall equivalence ratio for homogeneous operation. The IMEP is nearly constant in the rich region and decreases in the lean region with leaner operation. The

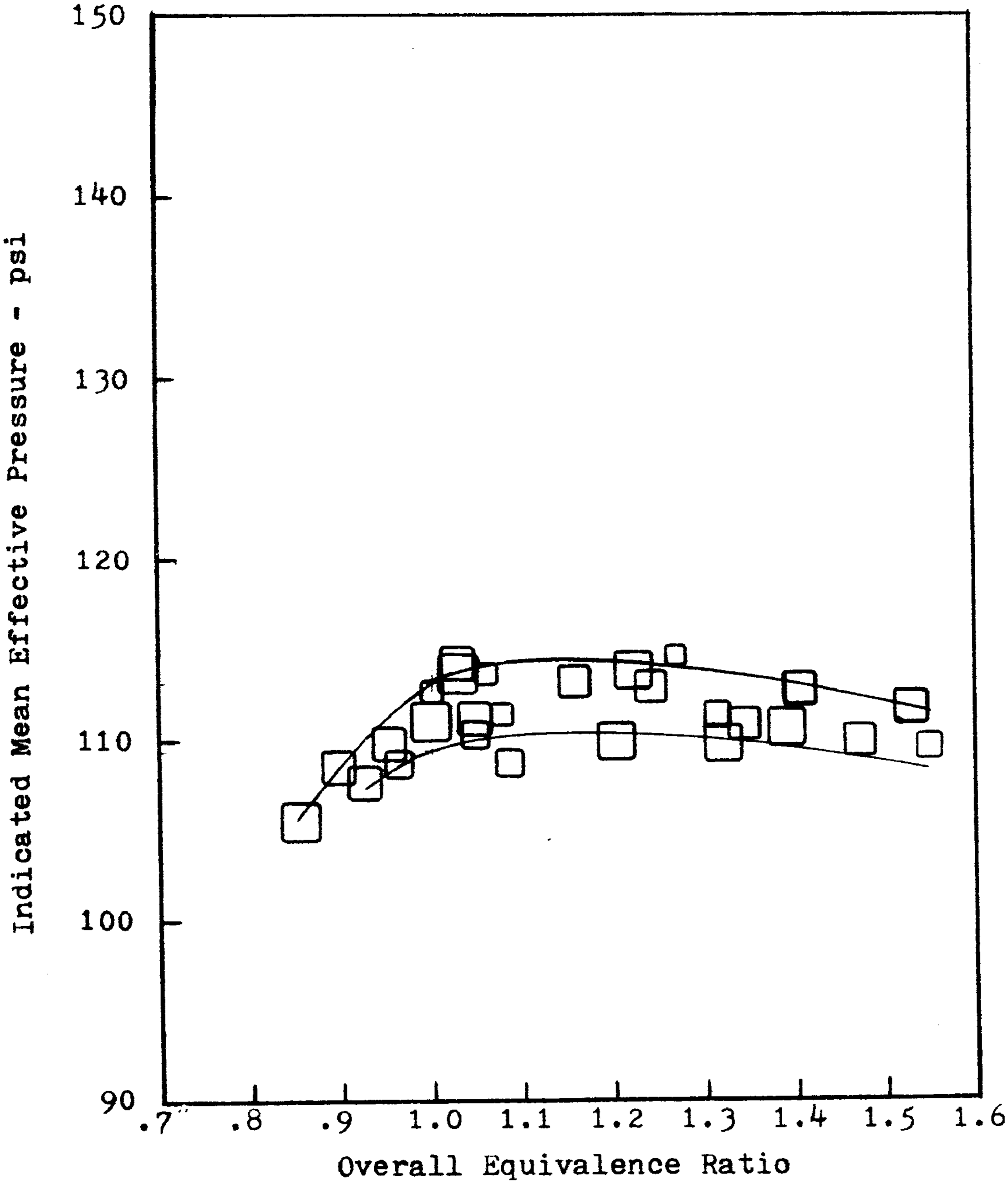


Figure 45 Indicated Mean Effective Pressure versus Overall Equivalence Ratio for Homogeneous Operation

homogeneous data represents a band of results mainly because of the difficulty in setting the MBT spark timing and the fluctuations in the dynamometer torque reading.

Presented in Figure 46 is the typical set of air injection data versus the overall equivalence ratio. The measured IMEP is seen to increase with leaner overall equivalence ratios. This trend reflects the increasing amount of injected air. In a sense the injected air supercharges the engine. The influence of the timing of air injection is given in Figure 47. A minimum in the measured IMEP occurs near top dead center when the air is being injected during combustion. The air injection apparently disturbs the combustion process causing incomplete combustion with reduced power.

When the IMEP is corrected by subtracting the estimated work required to compress the injected air the magnitude of the corrected IMEP is equal to or less than the corresponding homogeneous operation. Air injection before combustion results in the largest corrected IMEP.

8. Start of Air Injection at 128° BTDC

The results obtained with air injection starting at 128° BTDC will be presented separately because they deviate from the pattern of the other data. By referring to Table II it is possible to compare the result of the 128° BTDC air injection timing with the result of other

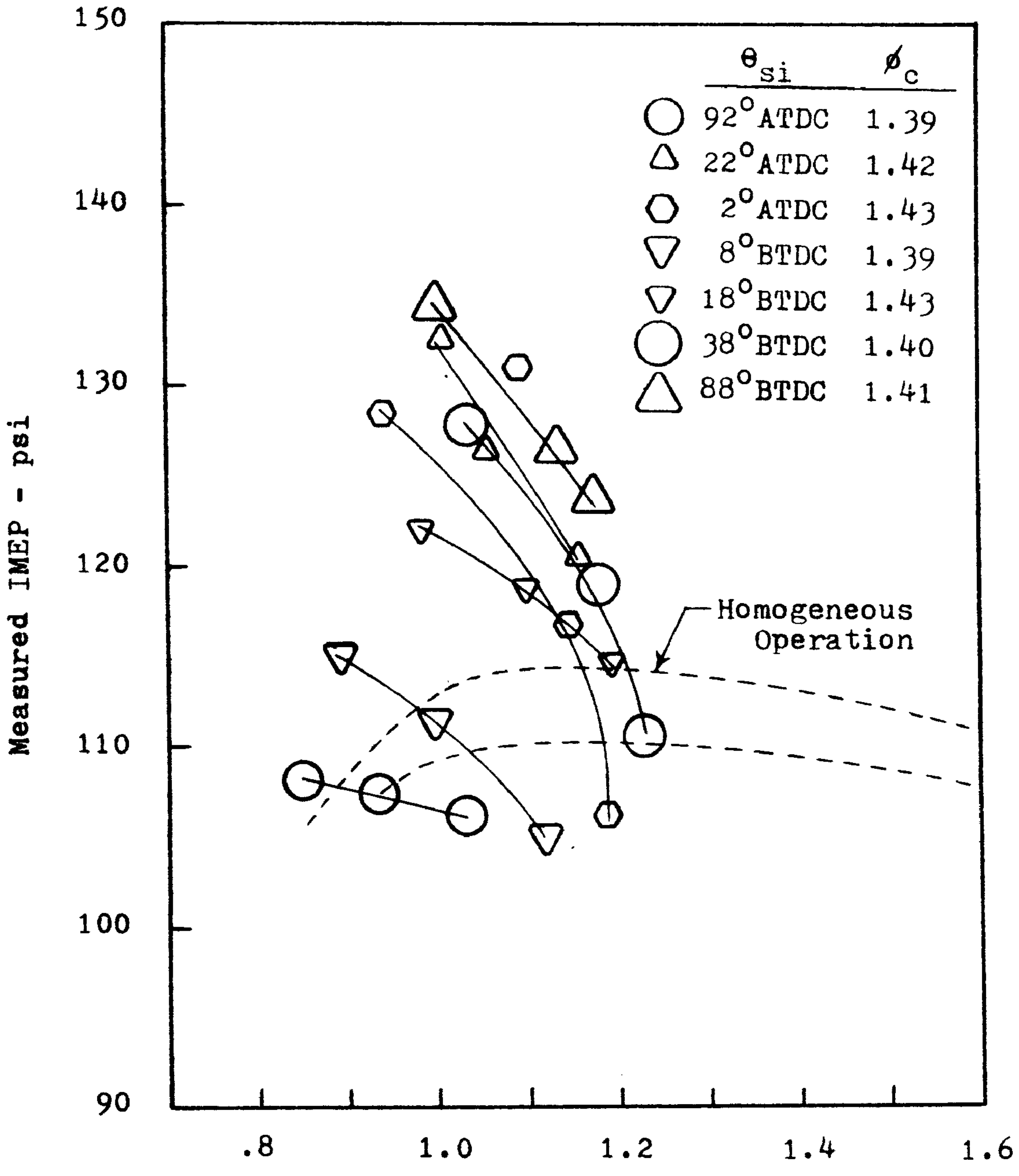


Figure 46 Measured Indicated Enthalpy Efficiency Versus Start of Air Injection for Overall Stoichiometric Equivalence Ratio With a Small Diameter Nozzle

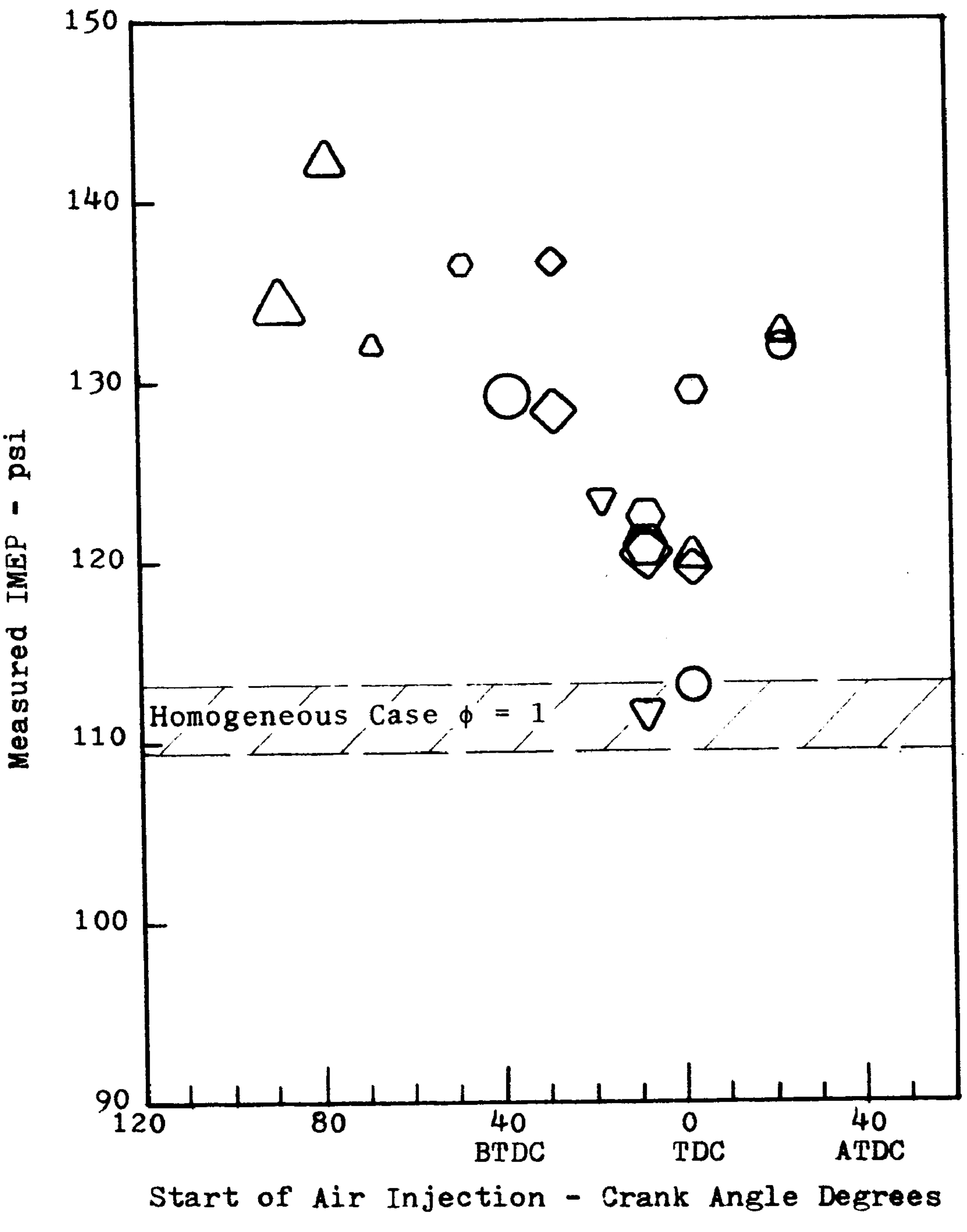


Figure 47 Measured Indicated Mean Effective Pressure Versus Start of Air Injection for Overall Stoichiometric Equivalence Ratio With a Small Diameter Nozzle

early air injection timings. The 128° BTDC air injection timing has less nitrogen oxide emissions, greater carbon monoxide emissions, greater hydrocarbon emissions and lower efficiency. Knock was also observed for the 128° BTDC air injection timing and not in the other air injection timings.

A second set of data was obtained at the 128° BTDC air injection timing with the nozzle rotated 90° so that the air jet is directed down into the cylinder and at an angle of 45° to the cylinder center line. For the rotated air jet, the nitrogen oxide, carbon monoxide and hydrocarbon emissions approach the values of other early air injection. The pressure trace indicates knock, which accounts for the low efficiency.

The probable reason for the difference in the results is that when the air is injected in the plane of the piston surface a stratification results with air on top and rich fuel mixture below. By rotating the nozzle down into the cylinder the stratification is prevented.

The knock for both cases is probably due to the extra compression of the charge when it is injected very early because of the large cylinder volume at the time of air injection.

9. The Effect of Air Injection on Combustion

One interesting discovery was the effect of air injection on the combustion process. The time interval between the spark and peak pressure was reduced from about 40 crank angle degrees for homogeneous operation to 15 crank angle degrees with early air injection. As a result of the more rapid combustion, the MBT spark timing was changed from about 25° BTDC homogeneous operation to about 6° BTDC. Also the cycle to cycle peak pressure variations were nearly eliminated. Presented in Figure 48 is a set of pressure versus time curves which show typical homogeneous operation, air injection during combustion, and air injection before combustion. The marks along the bottom of the pressure trace represent 10° crank angle degree increments with the large mark at top dead center.

The most probable explanation for the rapid and consistent combustion is that the turbulence resulting from air injection causes increased flame velocities. Lancaster et al. (1976) have shown that the flame velocity increases with greater turbulence in the combustion chamber of spark ignition homogeneously charged engines. The kinetic energy of the injected air is large because of its high velocity. Much of the kinetic energy will be converted to random turbulent motion. The influence of the turbulence can be seen when the air is injected during the combustion process as shown in Figure 48b. A

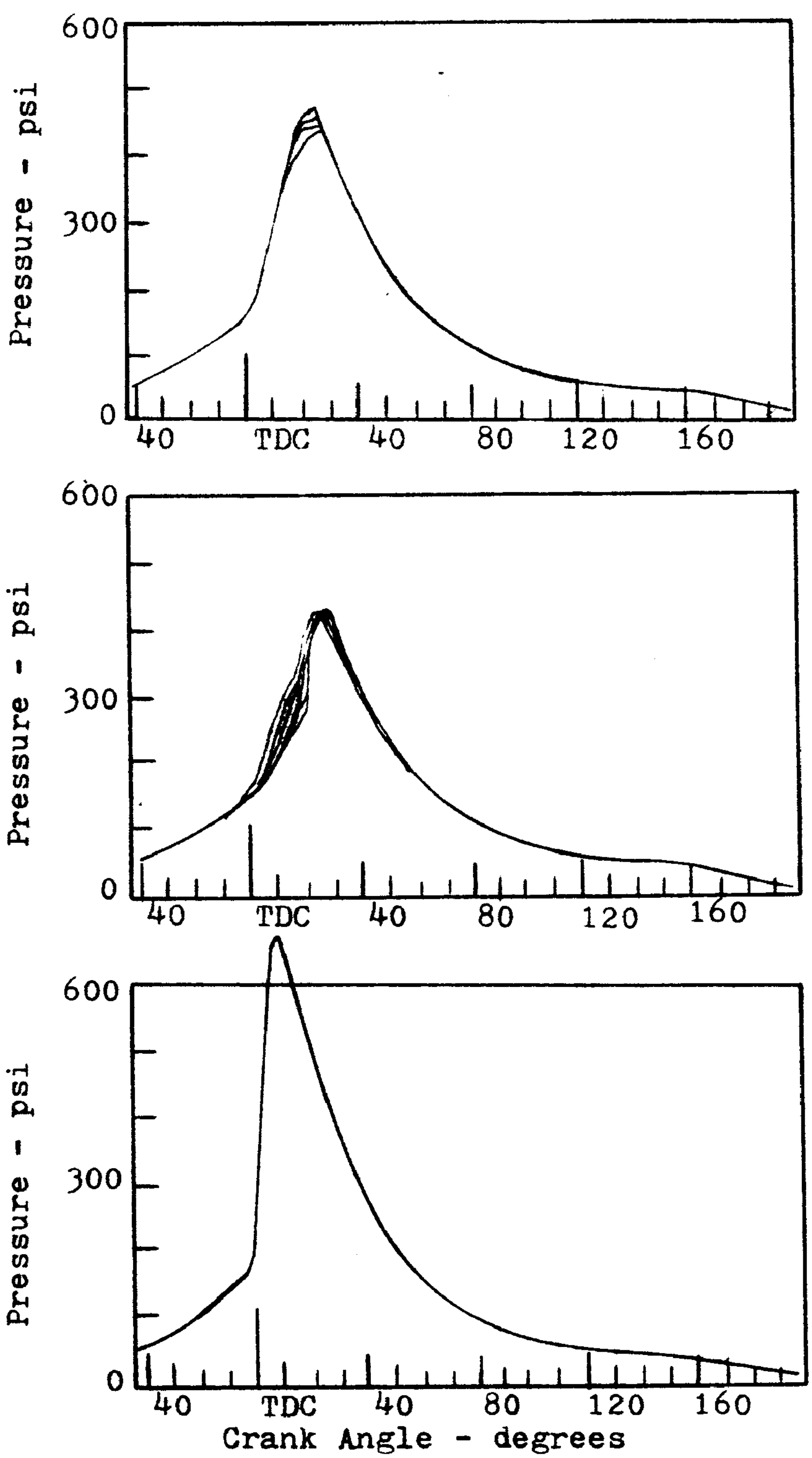


Figure 48a

Homogeneous Operation
$\phi_o = 1.20$
$\theta_{sp} = 20^o$BTDC

Figure 48b

Air Injection During Comb.
$\phi_o = 1.10$
$\phi_c = 1.45$
$\theta_{si} = 2^o$BTDC
$\theta_{sp} = 20^o$BTDC

Figure 48c

Air Injection Before Comb.
$\phi_o = 1.17$
$\phi_c = 1.40$
$\theta_{si} = 38^o$BTDC
$\theta_{sp} = 8^o$BTDC

Figure 48 Cylinder Pressure Versus Crank Angle for Homogenous Operation, Air Injection During Combustion and Air Injection Before Combustion definite increase in the rate of combustion is observed as a result of air injection.

10. Discussion of Results for Small Diameter Nozzle

It is apparent that air injection after combustion can reduce the nitrogen oxide and hydrocarbon emissions. Carbon monoxide emissions are at essentially the same level for either operation with air injection or homogeneously. However, when the timing of air injection is advanced nearer the combustion period to extract more work, all the emissions increase and the efficiency decreases. It seems very unlikely that the delayed mixing concept will be able to limit emissions while operating with homogeneous mixture efficiencies. The measured efficiencies are low with air injection because of the incomplete combustion when the air is injected during combustion and because of the increased heat transfer due to charge motion. When the work required to compress the injected air is charged against the cycle the efficiency is very low. Likewise the corrected IMEP will also be low.

The air injection before combustion resulted in very rapid and consistent combustion. Even with the desirable combustion, the measured efficiency was less than a corresponding homogeneous operation probably because of the additional heat transfer associated with the air injection induced charge motion.

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Appendix A

METHOD USED TO CALCULATE NITRIC OXIDE

The rate of formation of nitric oxide is described by the following set of equations.

Rate Constants:

$$N_2 + O \rightarrow NO + N \quad ; \quad K_{1c} = 7 \times 10^{13}\ exp(-75{,}000/RT)^*$$

$$NO + N \rightarrow N_2 + O \quad ; \quad K_{1b} = 1.55 \times 10^{13}$$

$$O_2 + N \rightarrow NO + O \quad ; \quad K_{2f} = 13.3 \times 10^{9}T\ exp(-7{,}080/RT)$$

$$NO + O \rightarrow O_2 + N \quad ; \quad K_{2b} = 3.2 \times 10^{9}T\ exp(-39{,}100/RT)$$

Rate Equations:

$$\frac{d[NO]}{dt} = K_{1f}[N_2][O] - K_{1b}[NO][O] + K_{2f}[O_2][N] - K_{2b}[NO][O]$$

or

$$\frac{d[NO]}{dt} = A - B[NO] \quad ; \quad A = K_{1f}[N_2][O] + K_{2f}[O_2][N]$$

$$B = K_{1b}[N] + K_{2b}[O]$$

For the case of constant temperature, pressure and equivalence ratio, A and B are constant. The solution to this first order differential with constant coefficients is given below.

$$[NO]_2 = \frac{A}{B} + ([NO]_1 - \frac{A}{B})\ e^{-B(t_2 - t_1)} \quad ; \quad t = time$$

---

*degrees Kelvin

This equation was used to predict the nitric oxide formation by evaluating A and B at an average value of temperature, pressure and equivalence ratio for a time step.

APPENDIX B

ONE SYSTEM COMPUTER MODEL

The following set of equations can be written for the one system model when it consists of two regions, reactants and products as seen in Fig. B1.

Conservation of Energy:

$$\dot{\overline{m_R u_R}} = -p\dot{V}_R - h_R \dot{m}_P$$

$$\dot{\overline{m_P u_P}} = -p\dot{V}_P + h_R \dot{m}_P$$

Conservation of Mass:

$$m_T = m_R + m_P$$

Conservation of Volume:

$$V_T = V_R + V_P$$

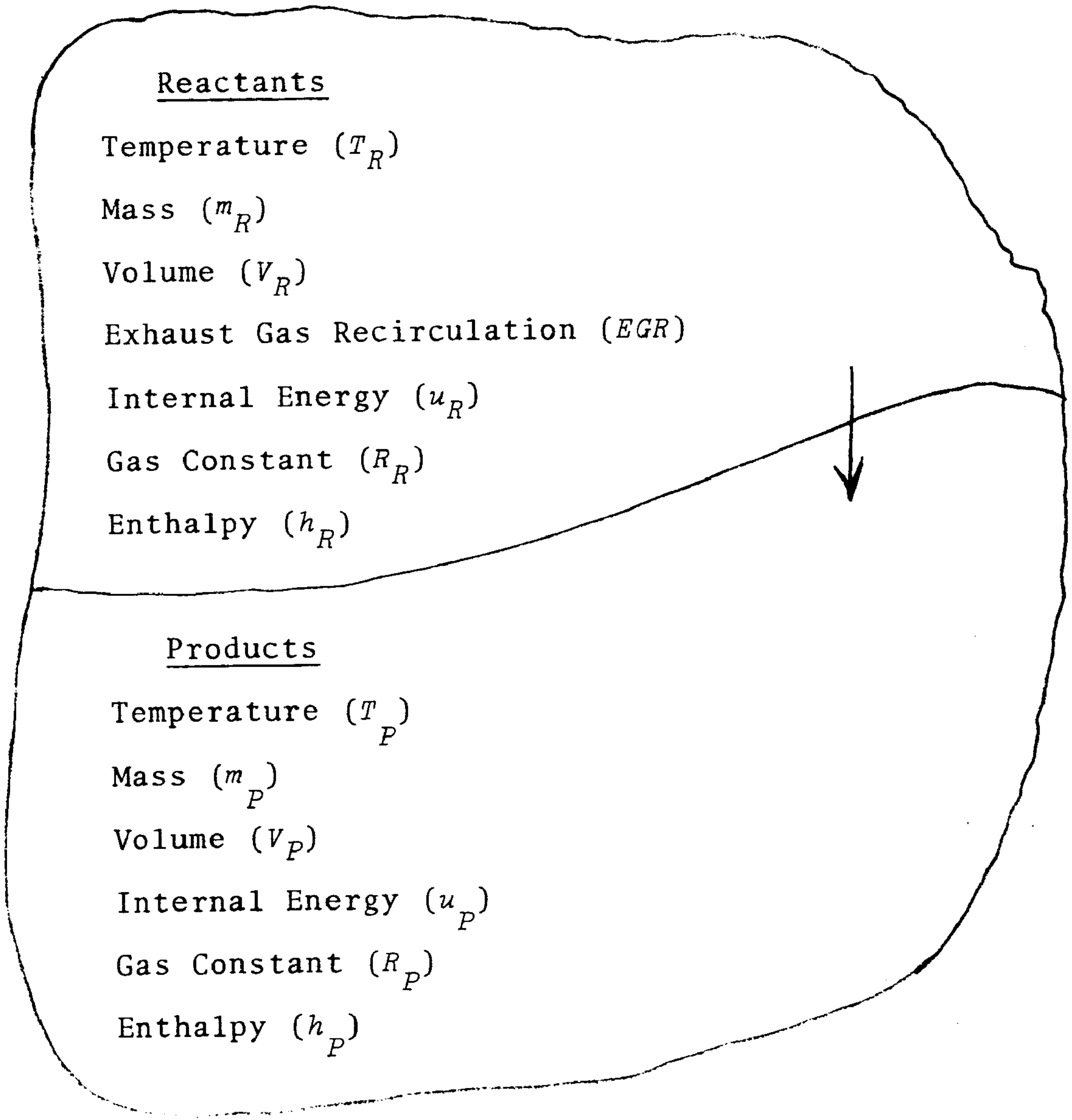
Ideal Gas Relationships:

$$pV_R = m_R R_R T_R$$

$$pV_P = m_P R_P T_P$$

Also the internal energy, its rate of change with the temperature, and the gas constant for the reactants can be obtained from equations describing these properties as functions of pressure, temperature, fuel equivalence ratio, and exhaust gas recirculation. The internal energy of the products and the gas constant, their rates of change with respect to pressure, temperature, and equivalence ratio are obtained from equations describing these properties as functions of pressure, temperature, fuel and equivalence ratio.

The two region equations can be solved for the following rate equations.



System

Pressure (p)

Fuel to Air Equivalence Ratio (F)

Volume (V_T or v_T)

Rate of Change of Volume ($\dot{V}_T$ or $\dot{v}_T$)

Fraction of the system mass which is products (α)

Rate of combustion ($\dot{m}_P$ or $\dot{\alpha}$)

Figure A1. One System Thermodynamic Model

$$\dot{p} = \frac{-p\dot{v}_T + \dot{\alpha}\left[R_P T_P - R_R T_R + (h_R - h_P)\left(T_P \frac{\partial R_P}{\partial T} + R_P\right)\Big/\left(\frac{\partial u_P}{\partial T} + T_P \frac{\partial R_P}{\partial T} + R_P\right)\right]}{v_T - \frac{(1-\alpha) R_R{}^2 T_R}{p\left(\frac{\partial u_R}{\partial T} + R_R\right)} - \alpha\left(T_P \frac{\partial R_P}{\partial p} + \frac{\left(T_P \frac{\partial R_P}{\partial T} + R_P\right)\left(\frac{T_P R_P}{p} - T_P \frac{\partial R_P}{\partial p} - \frac{\partial u_P}{\partial p}\right)}{\left(\frac{\partial u_P}{\partial T} + T_P \frac{\partial R_P}{\partial T} + R_P\right)}\right)}.$$

$$\dot{T}_R = \frac{R_R T_R \dot{p}}{p\left(\frac{\partial u_R}{\partial T} + R_R\right)}$$

$$\dot{T}_P = \frac{\dot{\alpha}(h_R - h_P) + \dot{p}\alpha\left(\frac{T_P R_P}{p} - T_P \frac{\partial R_P}{\partial p} - \frac{\partial u_P}{\partial p}\right)}{\alpha\left(\frac{\partial u_P}{\partial T} + T_P \frac{\partial R_P}{\partial T} + R_P\right)}$$

where: $\alpha = \frac{m_P}{m_T}$

Main Computer Program: The computer program uses a modified Euler method to predict the temperature of the reactants and the products. The pressure at the new point is determined from the temperatures at the new point, the gas constants and the total specific volume at the new point.

Computer Program Input:

NUR: The number of different sets of engine operating conditions to be calculated.

TITLE: A one line title for each set of operating conditions, this could be the run identification number.

P: The initial pressure in psia.

PHE: The equivalence ratio of the charge.

TR: The initial temperature of the reactants in degrees rankine.

EGR: The fraction of the total mass which is recirculated exhaust gas.

TOE: The time of expansion in seconds.

XXA: A shape factor of the Wiebe function describing expansion, see Figure 3.

XXM: A shape factor of the Wiebe function describing expansion, see Figure 3.

TOC: The time of combustion in seconds.

XA: The shape factor of the Weibe function describing combustion, see Figure 2.

XM: A shape factor of the Weibe function describing combustion, see Figure 2.

Function DELT(s): This function assigns the magnitude of the time step depending on the time (s) in the expansion.

Function VOL(s): The volume function assigns a system volume depending upon the time in the expansion. It uses the Weibe function.

Function ALF(s): This function describes the mass fraction burnt by a Weibe function dependent on the time in the expansion.

Subroutine FENERG: The FENERG subroutine is used to calculate the properties of the reactants.

Subroutine EQFLT( ): This subroutine is used to calculate the equilibrium flame temperature.

Subroutine NO ( ): The NO subroutine is used to calculate the nitric oxide formed during a time step.

Subroutine EQBM: This subroutine calculates the equilibrium concentrations of the various species in the products of combustion.

Subroutine ENERGY: The ENERGY subroutine determines the properties of the products of combustion. This subroutine was developed by G. L. Borman (1964) and can be found in his thesis.

```
45    FORMAT('  TWO REGION VARIABLE VOLUME COMBUSTION - MAY 11,1973')
      DIMENSION TITLE(24)
      COMMON/VE/TOE,XXA,XXM,C
      COMMON/A/TOC,XA,XM,J
      DATA CAT/8./,HAT/16./,OAT/0./,FAPC/.5/,QHV/19157./,NCFR/0/,KLU/1/
      C=.125
      WRITE(6,45)
      READ,NUR
15    READ(5,35)TITLE
      READ,P,PHE,TR,EGR
      READ,TOE,XXA,XXM
      READ,TOC ,XA,XM
      WRITE(6,35)TITLE
35    FORMAT(24A3)
      WRITE(6,40)PHE,EGR,TOE,XXM,XXA,TOC,XM,XA
40    FORMAT('  EQUIVALENCE RATIO=',F5.2,' EXHAUST GAS RECULATION RATE='
     1,F5.3/'  EXPANSION- TIME=',E10.4,'   XXM=',F4.1,'   XXA=',F4.1,
     2'  COMBUSTION- TIME=',E10.4,'   XM='F4.1,'   XA=',F4.1/)
      WRITE(6,26)
26    FORMAT('       TIME                NO-PPM       TEMP PROD        TEMP REACT
     1          PRESSURE       FRACTION BURNED     VOLUME      WORK FACTOR')
      ZZZ=(2.*CAT+HAT/2.-OAT)/2.
      FAS=(CAT*12.01+HAT*1.008+OAT*16.)/(ZZZ*32.+ZZZ*3.76*28.011)
      CALL EQFIT(TP,P,TR,PHE,QHV,FAS,FAPC,EGR,CAT,HAT,OAT)
      CALL FENERG(P,TR,PHE,UR,DUTR,RR,QHV,FAS,FAPC,EGR,CAT,HAT,OAT)
      VO=RR*TR/P
      VGO=VOL(0.)
      VG=VGO
      S=0.
      AL=0.
      WF=0.
      XNOP=0.
      J=1
      PO=P
      TO=TR
```

```
      GO TO 33
60    DT=DELT(S)
      S=S+DT
      ALL=AL
      AL=ALF(S)
      RMB=(AL-ALL)/DT
      VGG=VG
      VG=VOL(S)
      PP=P
      TVOL=VD*VG/VGD
      DTVOL=VD*(VG-VGG)/(VGD*DT)
      GO TO (110,120),J
110   CALL ENERGY(P,TP,PHE,FAS,UP,FAPC,RP,DUP,DLT,DUF,DRP,DRT,DRF)
      CALL FENERG(P,TR,PHE,UR,DUTR,RR,QHV,FAS,FAPC,EGR,CAT,HAT,UAT)
      HR=UR+RR*TR
      HP=UP+RP*TP
      ZR=RR/(DUTR+RR)
      ZP=(TP*DRT+RP)/(DUT+TP*DRT+RP)
      DP=(-P*DTVOL                +RMB*(RP*TP-RR*TR+ZP*(HR-HP)))/(TVOL-
     1AL*TP*DRP -ZR*(1.-AL)*RR*TR/P-ZP*AL*(RP*TP/P-TP*DRP -DUP ))
      DTR=(     (1.-AL)*RR*TR*DP/P)/((1.-AL)*(DUTR+RR))
      DTP=(     RMB*(HR-HP)+DP*AL*(RP*TP/P-TP*DRP -DUP ))/(AL*(DUT +TP
     1*DRT +RP))
      TRP=TR
      TPP=TP
      DTRR=DTR
      DTPP=DTP
      TR=TR+DTR*DT
      TP=TP+DTP*DT
      P=((1.-AL)*RR*TR+AL*RP*TP)/TVOL
      CALL ENERGY(P,TP,PHE,FAS,UP,FAPC,RP,DUP,DLT,DUF,DRP,DRT,DRF)
      CALL FENERG(P,TR,PHE,UR,DUTR,RR,QHV,FAS,FAPC,EGR,CAT,HAT,UAT)
      HR=UR+RR*TR
      HP=UP+RP*TP
      ZR=RR/(DUTR+RR)
```

```
      ZP=(TP*DRT+RP)/(DUT+TP*DRT+RP)
      DP=(-P*DTVOL                 +RMB*(RP*TP-RR*TR+ZP*(HR-HP)))/(TVOL-
     1AL*TP*DRP -ZR*(1.-AL)*RR*TR/P-ZP*AL*(RP*TP/P-TP*DRP -DUP ))
      DTR=(     (1.-AL)*RR*TR*DP/P)/((1.-AL)*(DUTR+RR))
      DTP=(     RMB*(HR-HP)+DP*AL*(RP*TP/P-TP*DRP -DUP ))/(AL*(DUT +TP
     1*DRT +RP))
      TR=TRR+(DTR+DTRR)*DT/2.
      TP=TPP+(DTP+DTPP)*DT/2.
      P=((1.-AL)*RR*TR+AL*RP*TP)/TVOL
      GO TO 130
120   CALL ENERGY(P,TP,PHE,FAS,UP,FAPC,RP,DUP,DUT,DUF,DRP,DRT,DRF)
      TR=0.
      ZP=(TP*DRT+RP)/(DUT+TP*DRT+RP)
      DP=(-P*DTVOL        )/(TVOL-TP*DRP -ZP*(RP*TP/P-TP*DRP-DUP ))
      DTP=(     DP*(RP*TP/P-TP*DRP -DUP ))/(DUT +TP*DRT +RP)
      TPP=TP
      DTPP=DTP
      TP=TP+DTP*DT
      P=RP*TP/TVOL
      CALL ENERGY(P,TP,PHE,FAS,UP,FAPC,RP,DUP,DUT,DUF,DRP,DRT,DRF)
      ZP=(TP*DRT+RP)/(DUT+TP*DRT+RP)
      DP=(-P*DTVOL        )/(TVOL-TP*DRP -ZP*(RP*TP/P-TP*DRP-DUP ))
      DTP=(     DP*(RP*TP/P-TP*DRP -DUP ))/(DUT +TP*DRT +RP)
      TP=TPP+(DTP+DTPP)*DT/2.
      P=RP*TP/TVOL
130   XNOPP=XNOP
      CALL NO(XNOP,XNOPP,PP,P,TPP,TP,PHE,ALL,AL,DT,CAT,HAT,DAT,IERR,NOER
     1,KLO)
      IF(IERR.NE.0) GO TO 50
      WF=WF+(PP+P)*(VG-VGG)/2.
33    WRITE(6,25)S,XNOP,TP,TR,P,AL,VG,WF
25    FORMAT(8F15.5)
      IF(S-TOE)60,50,50
50    ETA=3.15E-6*XNOP*P*(1.+C)/WF
      PUNCH 55,TITLE,PO,PHE,TO,EGR,TOE,XXM,XXA,TOC,XM,XA,XNOP,ETA
```

```
55    FORMAT(24A3/5E14.5/5E14.5/2E14.5)
      WRITE(6,65)ETA
65    FORMAT('  NITRIC OXIDE/WORK=',E15.5///)
      NUR=NUR-1
      IF(NUR)59,59,150
59    STOP
      END
      FUNCTION DELT(S)
      DIMENSION B(7)
      DATA B/1.E-5,2.E-5,5.E-5,1.E-4,5.E-4,1.E-3,2.E-3/
      I=0
      IF(S-5.9E-5)1,2,10
10    IF(S-1.9E-4)2,3,20
20    IF(S-.99E-3)3,4,30
30    IF(S-1.9E-3)4,5,40
40    IF(S-5.9E-3)5,6,50
50    IF(S-.99E-2)6,7,7
7     I=I+1
6     I=I+1
5     I=I+1
4     I=I+1
3     I=I+1
2     I=I+1
1     I=I+1
      DELT=B(I)
      RETURN
      END
```

```
      FUNCTION VOL(S)
      COMMON/V1/TOE,XXA,XXM,C
      IF(S)10,10,20
20    VOL=C-EXP(-XXA*((S/TOE)**(XXM+1.)))+1.
      GO TO 30
10    VOL=C
30    CONTINUE
      RETURN
      END

      FUNCTION ALF(S)
      COMMON /A/TOC,XA,XM,J
      IF(S-TOC)10,20,20
10    ALF=1.-EXP(-XA*((S/TOC)**(XM+1.)))
      GO TO 30
20    ALF=1.
      J=2
30    CONTINUE
      RETURN
      END
```

```
      SUBROUTINE FENERG(P,TR,PHE,UR,DUTR,RR,QHV,FAS,FAPC,EGR,CAT,HAT
     1,OAT)
      HFV=QHV-19186.+((((1.5757E-15*TR-5.4331E-12)*TR-4.5082E-08)*TR
     1+3.6285E-04)*TR+3.4765E-02)*TR+15.791+155.07
      DHFVT=(((7.8785E-15*TR-2.1732E-11)*TR-1.3510E-07)*TR+7.2570E-04)
     1*TR+3.4765E-02
      RF=1.9859/(CAT*12.011+HAT*1.008+OAT*16.)
      UF=HFV-RF*TR
      DUTF=DHFVT-RF
      UA=((((6.3156E-17*TR-9.3532E-13)*TR+3.9016E-09)*TR+5.1979E-06)
     1*TR+.16528)*TR
      DUTA=(((3.1578E-16*TR-3.74528E-12)*TR+1.17048E-08)*TR
     1+1.03958E-05)*TR+.16528
      CALL ENERGY(P,TF,PHE,FAS,UE,FAPC,RE,DUPE,DUTE,DUFE,DRPE,DRTE,DRFE)
      UR=(1.-EGR)*(PHE*FAS*UF+UA)/(PHE*FAS+1.)+EGR*UE
      DUTR=(1.-EGR)*(PHE*FAS*DUTF+DUTA)/(PHE*FAS+1.)+EGR*DUTE
      RR=(1.-EGR)*(PHE*FAS*RF+.06857)+EGR*RE
      RETURN
      END

      SUBROUTINE FDFLT(TP,P,TR,PHE,QHV,FAS,FAPC,EGR,CAT,HAT,OAT)
      DATA DHE/.2/
      CALL FENERG(P,TR,PHE,UR,DUTR,RR,QHV,FAS,FAPC,EGR,CAT,HAT,OAT)
      HR=UR+RR*TR
      TP=TR+3000.
  110 CALL ENERGY(P,TP,PHE,FAS,UP,FAPC,RP,DUP,DUT,DUF,DRP,DRT,DRF)
      HP=UP+RP*TP
      DH=HR-HP
   30 IF(ABS(DH)-DHE)10,10,20
   20 TP=TP+DH/(DUT+RP+DRT*TP)
      GO TO 110
   10 CONTINUE
      RETURN
      END
```

```
      SUBROUTINE NO(XNOP,XNOPP,PP,PF,TPP,TP,PHE,ALL,AL,DT,CAT,HAT,OAT,
     1IERR,NOFR,KLU)
      COMMON/BLOC/C   ,H   ,O   ,F   ,T  ,P    ,XH,XO,XN,XH2,XOH,XCO,XNO,XO2,
     1XH2O,XCO2,XN2,XAR, X13,I    ,N    ,K
      DATA V/1.98588/
      XNOCC=XNOPP*PP/(1.E+6*14.7*82.0567*TPP/1.8)
      C=CAT
      H=HAT
      O=OAT
      F=PHE
      T=(TPP+TP)/2.
      P=(PP+PF)/2.
      TEE=T/1.8
      CC=P/(14.7*82.0567*TEE)
      N=NOFR
      K=KLU
      CALL EQBM
      IERR=I
      IF(I.NE.0) GO TO 99
      A1=XN2*XO*7.E13*EXP(-7.5E4/(V*TEE))*CC*CC
      A2=XO2*XN*13.3E9*TEE*EXP(-7080./(V*TEE))*CC*CC
      B1=XN*1.55E13*CC
      B2=XO*3.2E9*TEE*EXP(-39.1E3/(V*TEE))*CC
      A=A1+A2
      B=B1+B2
      XNOC=(ALL/AL)*(XNOCC*TPP*PF/(TP*PP)+(A/B-XNOCC)*(1.-EXP(-B*DT)))
     1 +(1.-ALL/AL)*A*DT/2.
      XNOP=XNOC*1.E+6*14.7*82.0567*TP/(1.8*PF)
      GO TO 10
99    WRITE(6,100)IERR
100   FORMAT('  SUBROUTINE EQBM FAILED - ERROR CODE IS',I4)
10    CONTINUE
      RETURN
      END
```

```
      SUBROUTINE EQBM
C
C     DEVELOPED BY   CHERIAN OLIKARA       VERSION   NOV 1972
C     UNIVERSITY OF WISCONSIN, MADISON
C
      COMMON /BLOC/ AN,AM,AL,PHI,T,P,X1,X2,X3,X4,X5,X6,X7,X8,X9,X10,
     1X11,X12,X13,IERR,NOFR,KL0
      DIMENSION A(4,4),B(4)
      DATA JF/0/
C
C     ** SECTION 100 CALCULATES THE CONSTANTS USED IN THE SUBROUTINE.
C
      R0=(AN+0.25*AM-0.5*AL)/PHI
      R=R0+AL/2.0
      R1=R0*3.731137
      R2=R0*0.0444126
      IF(R.GT.0.5*AN) GO TO 110
      IERR=1
      GO TO 710
  110 D1=AM/AN
      D2=2.0*R/AN
      D3=2.0*R1/AN
      D4=R2/AN
      IF(NOFR-1) 115,112,115
  112 ANOM=X7/X13
      D5=ANOM/AN
C
C     THE EQUILIBRIUM CONSTANTS WERE CURVE FITTED (LEAST SQUARES) IN THE
C     RANGE 800 TO 3800 DEG K (1440 TO 6840 DEG R) FROM DATA IN JANAF
C     THERMOCHEMICAL TABLES SECOND EDITION (1970) .
C
  115 SQP=SQRT(P/14.7)
      TA=0.005*T/9.0
      ALTA=ALOG(TA)
```

```
      TAIN=1.0/TA
      TASQ=TA*TA
      C1=(10.0**(0.462001*ALTA-11.2170*TAIN+2.66703-0.100425*TA
     1+0.448836E-2*TASQ))/SQP
      C2=(10.0**(0.273274*ALTA-12.9757*TAIN+3.22140-0.540085E-1*TA
     2+0.182832E-2*TASQ))/SQP
      C3=(10.0**(0.342035*ALTA-24.6170*TAIN+3.16160-0.775319E-1*TA
     3+0.480024E-2*TASQ))/SQP
      C5=(10.0**(-0.132424*ALTA-2.13142*TAIN+0.852672+0.348813E-1*TA
     5-0.305775E-2*TASQ))
      C7=(10.0**(0.492411E-1*ALTA-4.69007*TAIN+0.643123-0.154083E-1*TA
     7-0.694746E-4*TASQ))
      C9=(10.0**(-0.735196*ALTA+12.4351*TAIN-2.61191+0.254389*TA
     9-0.161547E-1*TASQ))*SQP
      C10=(10.0**(-0.638530E-1*ALTA+14.8294*TAIN-4.75849+0.162629*TA
     1-0.123693E-1*TASQ))*SQP
C
C     ** SECTION 200 DECIDES WHETHER OR NOT TO MAKE A NEW ESTIMATE
C     OF X4,X6,X8 AND X11.
C
      IF(KLO-1) 305,205,410
  205 IF(JF.EQ.0) GO TO 305
      IF(PHI-PHIPR) 305,210,305
  210 IF(ABS(T/TPR-1.0).GT.0.2) GO TO 305
      IF(ABS(P/PPR-1.0).GT.0.2) GO TO 305
      GO TO 410
C
C     ** SECTION 300 CAN MAKE AN INITIAL ESTIMATE OF X4,X6,X8 AND X11.
C
  305 IF(PHI.GT.1.0) GO TO 310
      PAR=1.0/(R+R1+R2+0.25*AM)
      GO TO 315
  310 PAR=1.0/(R1+R2+AN+0.5*AM)
  315 FUN1=2.0*AN*C10
      FUN2=0.5*AN*C9
```

```
      FUN3=2.0/PAR
      FUN4=2.0*R
      DX=1.0
  320 SQDX=SQRT(DX)
      FDX=(FUN1*SQDX+AN)/(1.0+C10*SQDX)+FUN2*SQDX/(1.0+C9*SQDX)
     1+FUN3*DX-FUN4
      IF(FDX) 325,330,335
  335 DX=0.1*DX
      IF(DX.GE.1.0E-20) GO TO 320
      IERR=2
      GO TO 710
  325 IND=1
  327 SQDX=SQRT(DX)
      FDX=(FUN1*SQDX+AN)/(1.0+C10*SQDX)+FUN2*SQDX/(1.0+C9*SQDX)
     1+FUN3*DX-FUN4
      DDX=0.25*FUN1/(SQDX*(1.0+C10*SQDX)**2)+0.5*FUN2/(SQDX*(1.0+C9
     1*SQDX)**2)+FUN3
      RAT=FDX/DDX
      DX=DX-RAT
      IF(ABS(RAT/DX).LE.1.0E-2) GO TO 330
      IND=IND+1
      IF(IND.LE.20) GO TO 327
  330 SQDX=SQRT(DX)
      X4=0.5*AN*PAR/(1.0+C9*SQDX)
      X6=AN*PAR/(1.0+C10*SQDX)
      X8=DX
      X11=R1*PAR
C
C     ** SECTION 400 CALCULATES THE ELEMENTS OF THE MATRIX OF LINEARISED
C     EQUATIONS.
C
  410 IND=1
      T76=0.0
      T79=0.0
      T711=0.0
```

```
455 CONTINUE
    SQX4=SQRT(X4)
    SQX8=SQRT(X8)
    SQX11=SQRT(X11)
    X1=C1*SQX4
    X2=C2*SQX8
    X3=C3*SQX11
    X5=C5*SQX4*SQX8
    X9=C9*X4*SQX8
    X10=C10*X6*SQX8
    T14=0.5*C1/SQX4
    T28=0.5*C2/SQX8
    T311=0.5*C3/SQX11
    T54=0.5*C5*SQX8/SQX4
    T58=0.5*C5*SQX4/SQX8
    T94=C9*SQX8
    T98=0.5*C9*X4/SQX8
    T106=C10*SQX8
    T108=0.5*C10*X6/SQX8
    IF(NDFS-1) 460,465,470
460 X7=C7*SQX11*SQX8
    T78=0.5*C7*SQX11/SQX8
    T711=0.5*C7*SQX8/SQX11
    GO TO 470
465 X7=D5*(X5+X10)
    T76=D5*(1.0+C10*SQX8)
    T78=0.5*D5*C10*X6/SQX8
470 A(1,1)=T14+2.0+T54+2.0*T94
    A(1,2)=-D1*(1.0+T106)
    A(1,3)=(T58+2.0*T98)-D1*T108
    A(1,4)=0.0
    A(2,1)=T54+T94
    A(2,2)=(1.0+T76+2.0*T106)-D2*(1.0+T106)
    A(2,3)=(T28+T58+T78+2.0+T98+2.0*T108)-D2*T108
    A(2,4)=T711
```

```
      A(3,1)=0.0
      A(3,2)=T76-D3*(1.0+T106)
      A(3,3)=T78-D3*T108
      A(3,4)=T311+T711+2.0
      A(4,1)=T14+1.0+T54+T94
      A(4,2)=1.0+T76+T106+D4*(1.0+T106)
      A(4,3)=T28+T58+T78+1.0+T98+T108+D4*T108
      A(4,4)=T311+T711+1.0
      B(1)  =-(X1+2.0*X4+X5+2.0*X9)+D1*(X6+X10)
      B(2)  =-(X2+X5+X6+X7+2.0*X8+X9+2.0*X10)+D2*(X6+X10)
      B(3)  =-(X3+X7+2.0*X11)+D3*(X6+X10)
      B(4)  =-(X1+X2+X3+X4+X5+X6+X7+X8+X9+X10+X11+D4*(X6+X10))+1.0
C
C     ** SECTION 500 SOLVES THE MATRIX EQUATION BY GAUSSIAN ELIMINATION
C     USING MAXIMUM PIVOT POINT STRATEGY.
C
      DO 505 K=1,3
      KP1=K+1
      BIG=ABS(A(K,K))
      IF(BIG.GE.1.0E-10) GO TO 520
      IBIG=K
      DO 510 I=KP1,4
      IF(ABS(A(I,K)).LE.BIG) GO TO 510
      BIG=ABS(A(I,K))
      IBIG=I
  510 CONTINUE
      IF(BIG.GE.1.0E-15) GO TO 512
      IERR=3
      GO TO 710
  512 IF(IBIG.EQ.K) GO TO 520
      DO 515 J=K,4
      TEMP=A(K,J)
      A(K,J)=A(IBIG,J)
      A(IBIG,J)=TEMP
  515 CONTINUE
```

```
      TEMP=B(K)
      B(K)=B(IBIG)
      B(IBIG)=TEMP
  520 DO 525 I=KP1,4
      TERM=A(I,K)/A(K,K)
      DO 530 J=KP1,4
      A(I,J)=A(I,J)-A(K,J)*TERM
  530 CONTINUE
      B(I)=B(I)-B(K)*TERM
  525 CONTINUE
  505 CONTINUE
      IF(ABS(A(4,4)).GT.1.0E-15) GO TO 550
      IERR=3
      GO TO 710
  550 S4=B(4)/A(4,4)
      S3=(B(3)-A(3,4)*S4)/A(3,3)
      S2=(B(2)-A(2,3)*S3-A(2,4)*S4)/A(2,2)
      S1=(B(1)-A(1,2)*S2-A(1,3)*S3-A(1,4)*S4)/A(1,1)
C
C     ** SECTION 600 CORRECTS THE ESTIMATES OF THE VARIABLES AND
C     CHECKS ACCURACY.
C
  602 NCK=0
      X4=X4+S1
      IF(ABS(S1/X4 ).GT.1.0E-5) NCK=1
      X6=X6+S2
      IF(ABS(S2/X6 ).GT.1.0E-5) NCK=1
      X8=X8+S3
      IF(ABS(S3/X8 ).GT.1.0E-5) NCK=1
      X11=X11+S4
      IF(ABS(S4/X11).GT.1.0E-5) NCK=1
      IF((X4.GT.0.0).AND.(X8.GT.0.0).AND.(X11.GT.0.0)) GO TO 620
      IERR=4
      GO TO 710
  620 IF(NCK.EQ.0) GO TO 625
```

```
      IF(IND.LT.25) GO TO 622
      IERR=5
      GO TO 710
  622 IND=IND+1
      GO TO 455
  625 IF(X6.GE.0.0) GO TO 702
      IERR=4
      GO TO 710
C
C     ** SECTION 700 CALCULATES THE MOLFRACTIONS AND RETURNS WITH IERR=0
C     TO THE CALLING PROGRAM OR IF AN ANSWER WASNOT OBTAINED RETURNS
C     WITH IERR=(THE ERROR CODE)
C
  702 IERR=0
      JF=1
      PHIPR=PHI
      TPR=T
      PPR=P
      SQX4=SQRT(X4)
      SQX8=SQRT(X8)
      SQX11=SQRT(X11)
      X1=C1*SQX4
      X2=C2*SQX8
      X3=C3*SQX11
      X5=C5*SQX4*SQX8
      X9=C9*X4*SQX8
      X10=C10*X6*SQX8
      X13=(X6+X10)/AN
      X12=R2*X13
      IF(NOFR-1) 715,720,725
  715 X7=C7*SQX11*SQX8
      GO TO 725
  720 X7=ANOM*X13
  725 RETURN
  710 JF=0
      RETURN
      END
```

APPENDIX C

TWO SYSTEM COMPUTER MODEL

The following set of equations can be written for the two system model having three regions, lean reactants, rich reactants and combined products as seen in Fig. C1.

Conservation of Energy:

$$\dot{\overline{m_{LR}u_{LR}}} = - p\dot{V}_{LR} - h_{LR}\dot{m}_{LP}$$

$$\dot{\overline{m_{RR}u_{RR}}} = - p\dot{V}_{RR} - h_{RR}\dot{m}_{RP}$$

$$\dot{\overline{m_{P}u_{P}}} = - p\dot{V}_{P} + h_{LR}\dot{m}_{LP} + h_{RR}\dot{m}_{RP}$$

Conservation of Mass:

$$m_T = m_{LR} + m_{RR} + m_P$$

Conservation of Volume:

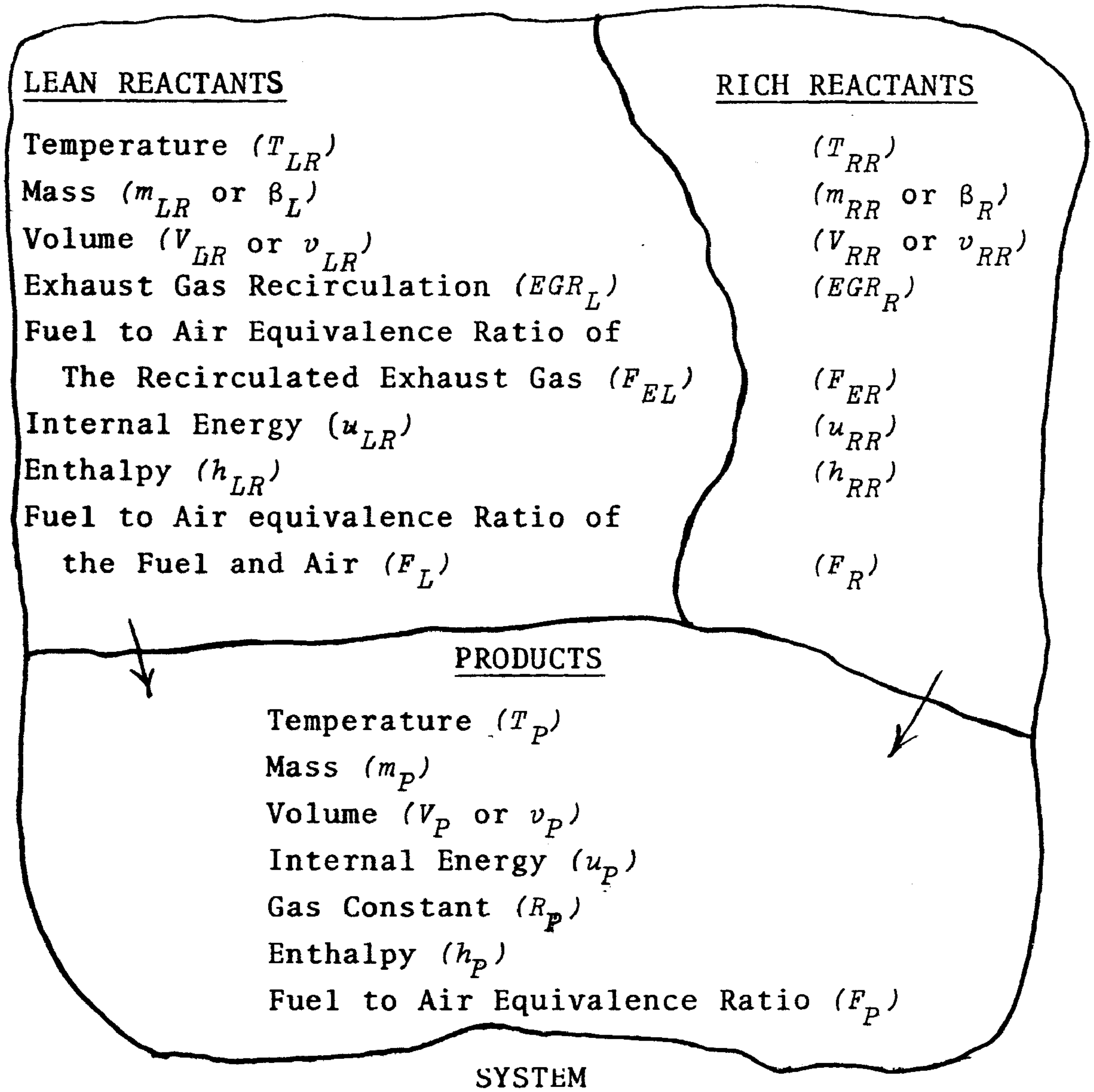
$$V_T = V_{LR} + V_{RR} + V_P$$

Ideal Gas Relationships:

$$pV_{LR} = m_{LR}R_{LR}T_{LR}$$

$$pV_{RR} = m_{RR}R_{RR}T_{RR}$$

$$pV_P = m_PR_PT_P$$



SYSTEM

Pressure $((p)$

Volume $(V_T$ or $v_T)$

Rate of Change of Volume $(\dot{V}_T$ or $\dot{v}_T)$

Rate of Combustion $(\dot{m}_P$ or $\dot{\alpha}_P)$

Fraction of the System which is Product (α_P)

Fraction of the System which is Lean Reactants (β_L)

Fraction of the System which is Rich Reactants (β_R)

Figure C1

Two System Thermodynamic Model With Three Regions

The internal energies and gas constants will be handled as discussed in the one system model.

The rate equations for the two system model having three regions are as follows.

$$\dot{p} = \left(- p\dot{v}_T + \dot{\beta}_L (R_{LR}T_{LR} - R_P T_P) + \dot{\beta}_R (R_{RR}T_{RR} - R_P T_P) + \frac{[-\dot{\beta}_L (h_{LR} - h_P) - \dot{\beta}_R (h_{RR} - h_P)](T_P \frac{\partial R_P}{\partial T} + R_P)}{(\frac{\partial u_P}{\partial T} + T_P \frac{\partial R_P}{\partial T} + R_P)} + \dot{F}\alpha_P \left[T_P \frac{\partial R_P}{\partial F} - \frac{(T_P \frac{\partial R_P}{\partial F} + \frac{\partial u_P}{\partial F})(T_P \frac{\partial R_P}{\partial T} + R_P)}{(\frac{\partial u_P}{\partial T} + T_P \frac{\partial R_P}{\partial T} + R_P)} \right] \right) \Big/$$

$$\left(v_T - \frac{\beta_L R_{LR}^2 T_{LR}}{p(\frac{\partial u_{LR}}{\partial T} + R_{LR})} - \frac{\beta_R R_{RR}^2 T_{RR}}{p(\frac{\partial u_{RR}}{\partial T} + R_{RR})} - \alpha_P \left[T_P \frac{\partial R_P}{\partial p} + \frac{(\frac{R_P T_P}{p} - T_P \frac{\partial R_P}{\partial p} - \frac{\partial u_P}{\partial p})(T_P \frac{\partial R_P}{\partial T} + R_P)}{(\frac{\partial u_P}{\partial T} + T_P \frac{\partial R_P}{\partial T} + R_P)} \right] \right)$$

$$\dot{T}_{LR} = \frac{R_{LR} T_{LR} \dot{p}}{(\frac{\partial u_{LR}}{\partial T} + R_{LR})}$$

$$\dot{T}_{RR} = \frac{R_{RR} T_{RR} \dot{p}}{p(\frac{\partial u_{RR}}{\partial T} + R_{RR})}$$

$$\dot{T}_P = \frac{-\dot{\beta}_L(h_{LR}-h_P)-\dot{\beta}_R(h_{RR}-h_P)+\alpha_P\dot{p}\left(\frac{R_P T_P}{p} - T_P\frac{\partial R_P}{\partial p} - \frac{\partial u_P}{\partial p}\right)-\alpha_P\dot{F}_P\left(T_P\frac{\partial R_P}{\partial F} + \frac{\partial u_P}{\partial F}\right)}{\alpha_P\left(\frac{\partial u_P}{\partial T} + T_P\frac{\partial R_P}{\partial T} + R_P\right)}$$

Where:

$$\alpha_P = \frac{m_P}{m_T}, \qquad \beta_L = \frac{m_{LR}}{m_T}, \qquad \beta_R = \frac{m_{RR}}{m_T}$$

Main Computer Program: The two system model consist of a main program and eight subprograms. The main program keeps track of combustion, expansion, and mixing processes. It also handles the program input and outputs. The subprograms are called as needed to perform their specialized tasks. Listed below are the input terms as an aid to understanding the program:

Computer Program Inputs:

NUR:	The number of different sets of engine operating conditions to be calculated.
TITLE:	A one line title for each set of calculations, such as run number.
TCM:	The time of combination of the two product systems (seconds).
GAM:	The initial fraction of the mass in the lean system.
CL:	The time between the initiation of combustion of the two systems (seconds).
THDO:	Crank angle at which combustion and the computer program begins (degrees).
RPM:	The engine speed (revolutions per minute).
C:	Clearance volume divided by the displaced volume.
XLR:	The length of the connecting rod divided by the radius of the crank shaft.

TOCL:	Time of combustion for the lean system (seconds).
TOCR:	Time of combustion for the rich system (seconds).
XML:	Shape parameter "m" of the combustion function for the lean system.
XMR:	Shape parameter "m" of the combustion function for the rich system.
XAL:	Shape parameter "a" of the combustion function for the lean system.
XAR:	Shape parameter "a" of the combustion function for the rich system.
PL:	Initial pressure of the lean system (psia).
PR:	Initial pressure of the rich system (psia).
FL:	Equivalence ratio of the lean system.
FR:	Equivalence ratio of the rich system.
TLR:	Initial temperature of the lean system reactants (degrees Rankine).
TRR:	Initial temperature of the rich system reactants (degrees Rankine).
EGRL:	Mass fraction of recirculated exhaust gases in the lean system.
EGRR:	Mass fraction of recirculated exhaust gases in the rich system.
FEL:	The equivalence ratio of the recirculated exhaust gases in the lean system.

FER: The equivalence ratio of the recirculated exhaust gases in the rich system.

Subroutine RPT ( ): The RPT subroutine is used to calculate the reactant to products process for a time step. This subroutine is capable of handling two regions of reactants and one region of products. The output of RPT is the new properties at the end of the time step. By setting the fraction of one of the reactant regions to zero the subroutine will handle two regions, one product region and one reactant region.

Subroutine (PRDMIX( ): The PRDMIX subroutine is used at the time of combination when the two product regions are combined into a single one. This subroutine determines the properties of the combined product region.

Subroutine PRESEQ ( ): After the products are combined the pressures in the two reactant regions and the combined product region are probable different. PRESEQ subroutine adjust the volumes of each region until they all have the same pressures. The regions expand or contract isentropically.

Subroutine EQFLT ( ): EQFLT is used to calculate the equilibrium flame temperature.

Subroutine NO ( ):	The NO subroutine is used to calculate the nitric oxide formed during a time step.
Subroutine PRD ( ):	The PRD subroutine calculates the change in properties for a time step when combustion is completed and only products remain.
Subroutine RENERG ( ):	This subroutine calculates the properties of the reactants.
Function DELT ( ):	This function assigns the magnitude of the time step depending on the time in the expansion.
Subroutine EQBM:	The EQBM subroutine calculates the equilibrium concentration of the various species in the products of combustion. EQBM is shown in Appendix B. This subroutine was developed by Cherian Olikara.
Subroutine ENERGY:	The ENERGY subroutine determines the properties of the products of combustion. This subroutine was developed by G. L. Borman (1964) and can be found in his thesis.

The computer program follows.

```
      DIMENSION TITLE(24)
      DATA CAT/8./,HAT/16./,OAT/0./,FAPC/.5/,QHV/19157./,NOFR/0/,KLO/1/
45    FORMAT('    STRATIFIED CHARGE ENGINE SIMULATION - JAN 22,1974'/)
      WRITE(6,45)
      READ,NUR
150   READ(5,35)TITLE
      WRITE(6,35)TITLE
35    FORMAT(24A3)
      READ,TOM,GAM,CL
      READ,THDO,RPM,C,XLR
      READ,TOCL,XML,XAL
      READ,TOCR,XMR,XAR
      READ,PL,FL,TLR,EGRL,FEL
      READ,PR,FR,TRR,EGRR,FER
      WRITE(6,15)RPM,TOM,CL,GAM
15    FORMAT(' RPM=',F8.0,'   TIME WHEN MIXING OCCURS=',E12.5,
     1'  COMBUSTION LAG=',E12.5,'  MASS FRACTION OF LEAN=',F5.3)
      WRITE(6,25)FL,EGRL,FEL,TOCL,XML,XAL
      WRITE(6,85)FR,EGRR,FER,TOCR,XMR,XAR
25    FORMAT(' LEAN-EQUIV RATIO=',F5.2,'  EXHAUST GAS RECIR=',F5.3,
     1'  EXHAUST GAS EQUIV RATIO=',F5.2,
     2'   COMBUSTION - TIME=',E10.4,'   XML=' ,F4.1,'  XAL=',F4.1)
85    FORMAT(' RICH-EQUIV RATIO=',F5.2,'  EXHAUST GAS RECIR=',F5.3,
     1'  EXHAUST GAS EQUIV RATIO=',F5.2,
     2'   COMBUSTION - TIME=',E10.4,'   XMR=' ,F4.1,'  XAR=',F4.1)
      WRITE(6,55)
55    FORMAT('    TIME            CRANK ANGLE        VOLUME     NO-PPM
     1 TEMP PROD  PHE PRD   TEMP REACT    PRESSURE    FRACTION BURNED
     2   WORK')
27    ZZZ=(2.*CAT+HAT/2.-OAT)/2.
      FAS=(CAT*12.01+HAT*1.008+OAT*16.)/(ZZZ*32.+ZZZ*3.76*28.011)
      FMWT=CAT*12.011+HAT*1.008+OAT*16.
      THD=THDO
      THE=THD*.017453
      IF(THE)16,17,16
```

```
17      VG=C
        GO TO 18
16      VG =C+.5-COS(THE)/2.+XLR*(1.-SQRT(1.-((SIN(THE)/XLR)**2 )))/2.
18      DFR=0.
        S=0.
        ALLS=0.
        ALRS=0.
        WFL=0.
        WFR=0.
        XNOPL=0.
        XNOPR=0.
        RMBL=0.
        RMBR=0.
        JJ=1
        IL=1
        IR=1
        IF(TOM)30,30,32
30      JJ=4
        CCCL=CL
        CL=0.
32      IF(CL)200,200,300
200     CALL EQFLT(TLP,PL,TLR,FL,QHV,FAS,FAPC,EGRL,FEL,FMWT)
        CALL RENERG(PL,TLR,FL,EGRL,FEL,ULR,DUTLR,RLR,QHV,FAS,FAPC,FMWT)
        VOLS=RLR*TLR/PL
        TVL=VOLS
        VGOL=VG
        ULRB=ULR
        XKL=(DUTLR+RLR)/DUTLR
        WCL=TLR*RLR*(1.-((1.+C)/VG)**(1.-XKL))/(1.-XKL)
        GAML=1.
        FLAA=EGRL*FEL/(1.+FEL*FAS)+(1.-EGRL)*FL/(1.+FL*FAS)
        FLP=FLAA/(1.-FAS*FLAA)
        IF(JJ.EQ.1)CCL=0.
        GO TO 201
202     DT=DELT(S)
```

```
      IF(S.GE.TOM) GO TO 400
      S=S+DT
      THD=S*RPM*6.+THDO
      THE=THD*.017453
      VG =C+.5-COS(THE)/2.+XLR*(1.-SQRT(1.-((SIN(THE)/XLR)**2 )))/2.
      TVLL=TVL
      TVL=VOLS*VG/VGOL
      DTVL=(TVL-TVLL)/ DT
      PLL=PL
      TLPP=TLP
      GO TO (215,211,210),IL
215   ALLSS=ALLS
      ALLS=1.-EXP(-XAL*(((S-CCL)/TOCL)**(XML+1.)))
      RMBL=(ALLS-ALLSS)/DT
      IF(S-CCL.GE.TOCL) IL=2
      CALL RTP(PL,GAML,TLR,TRR,TLP,FL,FR,FLP,DFP,ALLSS,ALRSS,RMBL,RMBR
     1,EGRL,FEL,EGRR,FER,TVLL,DTVL,FAS,FAPC,QHV,FMWT,DT)
      GO TO 235
211   ALLS=1.
      ALLSS=1.
      RMBL=0.
      TLR=0.
      IL=3
210   CALL PRD(PL,TLP,FLP,DTVL,TVLL,FAS,FAPC,DT)
235   XNOPLL=XNOPL
      IF(FLP.LT.1.E-6) GO TO 242
      CALL NO(XNOPL,XNOPLL,PLL,PL,TLPP,TLP,FLP,ALLSS,ALLS,DT,CAT,HAT,OAT
     1,IERR,NOFR,KLO)
      IF(IERR.NE.0) GO TO 50
242   WFL=WFL+(PLL+PL)*(TVL-TVLL)/2.
201   WRITE(6,65)S,THD,VG,XNOPL,TLP,FLP,TLR,PL,ALLS,WFL
65    FORMAT(5E14.5,F6.3,4E14.5)
250   GO TO (203,314,275,300),JJ
275   JJ=2
      GO TO 202
```

```
203   IF(S-ABS(CL))202,204,204
204   JJ=2
300   CALL EQFLT(TRP,PR,TRR,FR,QHV,FAS,FAPC,EGRR,FER,FMWT)
      CALL RENERG(PR,TRR,FR,EGRR,FER,URR,DUTRR,RRR,QHV,FAS,FAPC,FMWT)
      VORS=RRR*TRR/PR
      TVR=VORS
      VGOR=VG
      URRB=URR
      XKR=(DUTRR+RRR)/DUTRR
      WCR=TRR*RRR*(1.-((1.+C)/VG)**(1.-XKR))/(1.-XKR)
      GAMR=0.
      FRAA=EGRR*FER/(1.+FER*FAS)+(1.-EGRR)*FR/(1.+FR*FAS)
      FRP=FRAA/(1.-FAS*FRAA)
      IF(JJ.EQ.1)CCR=0.
      IF(JJ.EQ.2)CCR=ABS(CL)
      GO TO 301
302   DT=DELT(S)
      IF(S.GE.TOM) GO TO 400
      S=S+DT
      THD=S*RPM*6.+THDO
      THE=THD*.017453
      VG =C+.5-COS(THE)/2.+XLR*(1.-SQRT(1.-((SIN(THE)/XLR)**2 )))/2.
314   TVRR=TVR
      TVR=VORS*VG/VGOR
      DTVR=(TVR-TVRR)/DT
      PRR=PR
      TRPP=TRP
      GO TO (315,311,310),IR
315   ALRSS=ALRS
      ALRS=1.-EXP(-XAR*(((S-CCR)/TOCR)**(XMR+1.)))
      RMBR=(ALRS-ALRSS)/DT
      IF(S-CCR.GE.TOCR) IR=2
      CALL RTP(PR,GAMR,TLR,TRR,TRP,FL,FR,FRP,DFP,ALLSS,ALRSS,RMBL,RMBR
     1,EGRL,FEL,EGRR,FER,TVRR,DTVR,FAS,FAPC,QHV,FMWT,CT)
      GO TO 335
```

```
311   ALRS=1.
      ALRSS=1.
      RMAR=0.
      TRR=0.
      IR=3
310   CALL PRD(PR,TRP,FRP,DTVR,TVRR,FAS,FAPC,DT)
335   XNOPRR=XNOPR
      IF(FRP.LT.1.E-6) GO TO 342
      CALL NO(XNOPR,XNOPRR,PRR,PR,TRPP,TRP,FRP,ALRSS,ALRS,DT,CAT,HAT,DAT
     1,IERR,NOFR,KLO)
      IF(IERR.NE.0) GO TO 50
342   WFR=WFR+(PRR+PR)*(TVR-TVRR)/2.
301   WRITE(6,65)S,THD,VG,XNOPR,TRP,FRP,TRR,PR,ALRS,WFR
      GO TO (303,202,202,305),JJ
303   IF(S-ABS(CL))302,304,304
304   JJ=3
      GO TO 200
305   P=PR
      FP=FRP
      TP=TRP
      WFL=0.
      WFR=0.
      WF=0.
      XNOP=0.
      ALP=0.
      CCL=CCCL
      VGO=VG
      TVOL=GAM*VOLS+(1.-GAM)*VORS
      VO=TVOL
      GO TO 401
400   CALL PRDMIX(PS,TPS,FP,PL,TLP,FLP,PR,TRP,FRP,ALLS,ALRS,
     1GAM,FAS,FAPC,QHV,FMWT,XNOPL,XNOPR,XNOP)
405   CALL PRESEQ(P,TP,PS,TPS,FP,TLR,PL, FL ,TRR,PR,FR ,EGRL,FEL
     1,EGRR,FER,ALLS,ALRS,GAM,TVOL,FAS,FAPC,QHV,FMWT)
      VGO=VG
```

```
      VO=TVOL
      ALP=ALLS*GAM+ALRS*(1.-GAM)
      WF=GAM*WFL+(1.-GAM)*WFR
      WRITE(6,75)
75    FORMAT('  THE PRODUCTS ARE MIXED AT THIS TIME')
      GO TO 401
402   DT=DELT(S-TOM)
      S=S+DT
      THD=S*RPM*6.+THDO
      THE=THD*.017453
      VG =C+.5-COS(THE)/2.+XLR*(1.-SQRT(1.-((SIN(THE)/XLR)**2 )))/2.
      TVOLL=TVOL
      TVOL=VO*VG/VGO
      DTVOL=(TVOL-TVOLL)/DT
      PP=P
      TPP=TP
414   GO TO (442,450,450,410),IL
442   ALLSS=ALLS
      IF(S-CCL)450,450,443
443   ALLS=1.-EXP(-XAL*(((S-CCL)/TOCL)**(XML+1.)))
      RMBL=(ALLS-ALLSS)/DT
      IF(S-CCL.GE.TOCL) IL=2
450   GO TO (452,455,455),IR
452   ALRSS=ALRS
      ALRS=1.-EXP(-XAR*(((S-CCR)/TOCR)**(XMR+1.)))
      RMBR=(ALRS-ALRSS)/DT
      IF(S-CCR.GE.TOCR) IR=2
455   ALPP=ALP
      ALP=ALLS*GAM+ALRS*(1.-GAM)
      FPP=FP
      FPAA= (FLP*ALLS*GAM/(1.+FLP*FAS)+FRP*ALRS*(1.-GAM)/(1.+FRP*FAS))
     1/ALP
      FP=FPAA/(1.-FAS*FPAA)
      AFP=(FPP+FP)/2.
      DFP=(FP-FPP)/DT
```

```
      CALL RTP(P,GAM,TLR,TRR,TP,FL,FR,FPP,DFP,ALLSS,ALRSS,RMBL,RMBR,EGRL
     1,FEL,EGRR,FER,TVOLL,DTVOL,FAS,FAPC,QUV,FMWT,DT)
420   GO TO (412,411,412),IL
411   ALLS=1.
      ALLSS=1.
      RMBL=0.
      TLR=0.
      IL=3
412   GO TO (417,413,417),IR
413   ALRS=1.
      ALRSS=1.
      RMBR=0.
      TRR=0.
      IR=3
417   IF(IL-3)435,480,480
480   IF(IR-3)435,481,481
481   IL=4
      AFP=FP
      DFP=0.
      ALP=1.0
      ALPP=1.0
      GO TO 435
410   CALL PRD(P,TP ,FP,DTVOL,TVOLL,FAS,FAPC,DT)
435   XNOPP=XNOP
      CALL NO(XNOP,XNOPP,PP,P,TPP,TP,AFP,ALPP,ALP,DT,CAT,HAT,DAT,IERR,
     1NOER,KLD)
      IF(IERR.NE.0) GO TO 50
      WF=WF+(PP+P)*(TVOL-TVOLL)/2.
401   WRITE(6,65)S,THD,VG,XNOP,TP,FP,TLR,P,ALLS,WF
      GO TO (408,408,408,80),IL
408   WRITE(6,95)TRR,ALRS
95    FORMAT(76X,E14.5,14X,E14.5)
80    IF(THD-180.)402,50,50
50    CONTINUE
      CALL ENERGY(P,TP,FP,FAS,UP,FAPC,RP,DUP,DUT,DUF,DRP,DRT,DRF)
```

```
      UR=GAM*ULRB+(1.-GAM)*URRB
      UB=UP-UR+WF
      VRF=GAM*VOLS*VG/VGOL+(1.-GAM)*VORS*VG/VGOR
      VPF=RP*TP/P
      VB=VPF-VRF
      WC=GAM*WCL+(1.-GAM)*WCR
      WN=WF+WC
      GMNO=.0068541*RP*XNOP
      SNO=GMNO*2544.4/WN
      EFF=WN/(QHV*(GAM*(1.-EGRL)*FL*FAS/(1.+FL*FAS)+(1.-GAM)*(1.-EGRR)
     1*FR*FAS/(1.+FR*FAS)))
      EP=WN*(1.+C)/VPF
      WRITE(6,105)UP,UR,WF,UB
105   FORMAT('   ENERGY BALANCE -  UP=',E12.5,'   UR=',E12.5,
     1'    WORK=',E12.5,'    SUM=',E12.5)
      WRITE(6,106)VPF,VRF,VB
106   FORMAT('   SPECIFIC VOLUME BALANCE AT FINAL CONDITIONS-  VP='
     1,E12.5,'   VR=',E12.5,'   DIFFERENCE=',E12.5)
      WRITE(6,107)WC,EFF,EP,SNO
107   FORMAT('   PERFORMANCE PARARETERS -   ESTIMATED COMPRESSION WORK='
     1,E12.5/'   ENTHALPY EFFIEIENCY=',E12.5,'    IMEP=',E12.5,
     2'    ISNO=',E12.5//)
      PUNCH 108,UB,VB,EFF,EP,SNO
108   FORMAT(5E14.5)
      NUR=NUR-1
      IF(NUR)59,59,150
59    STOP
      END
      SUBROUTINE RTP(P ,GAM ,TLR,TRR,TP,FL,FR,FPP,DFP,ALLSS,ALRSS,RMBL
     1,RMBR,EGRL,FEL,EGRR,FER,TVOLL,DTVOL,FAS,FAPC,QHV,FMWT,DT)
      FFL=1.
      FFR=1.
      IF(GAM.LE.1.E-4) FFL=0.
      IF(GAM.GE..9999) FFR=0.
      CALL ENERGY(P,TP,FPP,FAS,UP,FAPC,RP,DUP,DLT,DUF,DRP,DRT,DRF)
```

```
      CALL RENERG(P,TLR,FL,EGRL,FEL,ULR,DUTLR,RLR,QHV,FAS,FAPC,FMWT)
      CALL RENERG(P,TRR,FR,EGRR,FER,URR,DUTRR,RRR,QHV,FAS,FAPC,FMWT)
      ALP=ALLSS*GAM+ALRSS*(1.-GAM)
      HP=UP+RP*TP
      HLR=ULR+RLR*TLR
      HRR=URR+RRR*TRR
      ZL=RLR*TLR/(P*(DUTLR+RLR))
      ZR=RRR*TRR/(P*(DUTRR+RRR))
      AA=(TP*DRT+RP)/(DUT+TP*DRT+RP)
      BL=(1.-ALLSS)*GAM*RLR
      BR=(1.-ALRSS)*(1.-GAM)*RRR
      H=-RMBL*GAM*(RLR*TLR-RP*TP)-RMBR*(1.-GAM)*(RRR*TRR-RP*TP)-P*DTVOL
      G=TVOLL-ALP*TP*DFP
      D=ALP*(RP*TP/P-TP*DRP-DUP)
      F=ALP*(DUT+TP*DRT+RP)
      C=RMBL*GAM*(HLR-HP)+RMBR*(1.-GAM)*(HRR-HP)
      E=DUF+TP*DRF
      DP=(H+C*AA+DFP*ALP*(TP*DRF-E*AA))/(G-ZL*BL-ZR*BR-D*AA)
      DTLR=ZL*DP*FFL
      DTRR=ZR*DP*FFR
      IF(ALP)10,10,20
10    DTP=D*DP-E*DFP*ALP
      GO TO 30
20    DTP=(C+D*DP-E*ALP*DFP)/F
30    TRRR=TRR
      TLRR=TLR
      DTRRR=DTRR
      DTLRR=DTLR
      DTPP=DTP
      TLR=TLR+DTLR*DT
      TRR=TRR+DTRR*DT
      TP=TP+DTP*DT
      ALLS=ALLSS+RMBL*DT
      ALRS=ALRSS+RMBR*DT
      FP=FPP+DFP*DT
```

```
 TVOL=TVOLL+DTVOL*DT
 ALP=ALLS*GAM+ALRS*(1.-GAM)
 P=((1.-ALLS)*GAM*RLR*TLR+(1.-ALRS)*(1.-GAM)*RRR*TRR+ALP*RP*TP)/
1TVOL
 CALL ENERGY(P,TP,FP,FAS,UP,FAPC,RP,DUP,DUT,DUF,DRP,DRT,DRF)
 CALL RENERG(P,TLR,FL,EGRL,FEL,ULR,DUTLR,RLR,QHV,FAS,FAPC,FMWT)
 CALL RENERG(P,TRR,FR,EGRR,FER,URR,DUTRR,RRR,QHV,FAS,FAPC,FMWT)
 HP=UP+RP*TP
 HLR=ULR+RLR*TLR
 HRR=URR+RRR*TRR
 ZL=RLR*TLR/(P*(DUTLR+RLR))
 ZR=RRR*TRR/(P*(DUTRR+RRR))
 AA=(TP*DRT+RP)/(DUT+TP*DRT+RP)
 BL=(1.-ALLS)*GAM*RLR
 BR=(1.-ALRS)*(1.-GAM)*RRR
 H=-RMBL*GAM*(RLR*TLR-RP*TP)-RMBR*(1.-GAM)*(RRR*TRR-RP*TP)-P*DTVOL
 G=TVOL-ALP*TP*DRP
 D=ALP*(RP*TP/P-TP*DRP-DUP)
 F=ALP*(DUT+TP*DRT+RP)
 C=RMBL*GAM*(HLR-HP)+RMBR*(1.-GAM)*(HRR-HP)
 E=DUF+TP*DRF
 DP=(H+C*AA+DFP*ALP*(TP*DRF-E*AA))/(G-ZL*BL-ZR*BR-D*AA)
 DTLR=ZL*DP*FFL
 DTRR=ZR*DP*FFR
 DTP=(C+D*DP-E*ALP*DFP)/F
 TLR=TLRR+(DTLR+DTLRR)*DT/2.
 TRR=TRRR+(DTRR+DTRRR)*DT/2.
 TP=TPP+(DTP+DTPP )*DT/2.
 P=((1.-ALLS)*GAM*RLR*TLR+(1.-ALRS)*(1.-GAM)*RRR*TRR+ALP*RP*TP)/
1TVOL
 RETURN
 END
 SUBROUTINE PRDMIX(PS,TPS,FP,PL,TLP,FLP,PR,TRP,FRP,ALLS,ALRS,
1GAM,FAS,FAPC,QHV,FMWT,XNOPL,XNOPR,XNOP)
 CALL ENERGY(PL,TLP,FLP,FAS,ULP,FAPC,RLP,DUP,DUT,DUF,DRP,DRT,DRF)
```

```
      CALL ENERGY(PR,TRP,FRP,FAS,URP,FAPC,RRP,DLP,DUT,DUF,DRP,DRT,DRF)
      XNOPS=(ALLS*GAM*XNOPL*RLP+ALRS*(1.-GAM)*XNOPR*RRP)
      DUL=.1
      ALP=ALLS*GAM+ALRS*(1.-GAM)
      BL=ALLS*GAM/ALP
      BR=ALRS*(1.-GAM)/ALP
      VP=BL*RLP*TLP/PL+BR*RRP*TRP/PR
      FPAA= (FLP*ALLS*GAM/(1.+FLP*FAS)+FRP*ALRS*(1.-GAM)/(1.+FRP*FAS))
     1/ALP
      FP=FPAA/(1.-FAS*FPAA)
      RP=BL*RLP+BR*RRP
      UPA=BL*ULP+BR*URP
      TPP=BL*TLP+BR*TRP
      P=RP*TPP/VP
      CALL ENERGY(P,TPP,FP,FAS,UP,FAPC,RP,DUP,DLP,DUF,DRP,DRT,DRF)
      UPP=UP
      TP=TPP+100.
30    P=RP*TP/VP
      CALL ENERGY(P,TP,FP,FAS,UP,FAPC,RP,DUP,DUT,DUF,DRP,DRT,DRF)
      DU=UP-UPA
      IF(ABS(DU)-DUL)10,10,20
20    T=TP
      TP=TP-DU*(TP-TPP)/(UP-UPP)
      TPP=T
      UPP=UP
      GO TO  30
10    TPS=TP
      PS=P
      XNOP=XNOPS/(ALP*RP)
      RETURN
      END
      SUBROUTINE PRESEQ(P,TP,PS,TPS,FP,TLP,PL, FL ,TRP,PR,FR ,EGRL,FEL
     1,EGRR,FER,ALLS,ALRS,GAM,TVOL,FAS,FAPC,QHV,FMWT)
      CALL ENERGY(PS,TPS,FP,FAS,UPS,FAPC,RP ,DUP,DUT,DUF,DRP,DRT,DRF)
      CALL RENERG(PL,TLP,FL,EGRL,FEL,ULR,DUTLR,RLP,QHV,FAS,FAPC,FMWT)
```

```
      CALL RENERG(PR,TRR,FR,EGRR,FER,URR,DUTRR,RRR,QHV,FAS,FAPC,FMWT)
      ERRL=.1
      ALP=ALLS*GAM+ALRS*(1.-GAM)
      CL=(1.-ALLS)*GAM
      CR=(1.-ALRS)*(1.-GAM)
      URA=CL*ULR+CR*URR
      TVOL=CL*RLR*TLR/PL+CR*RRR*TRR/PR+ALP*RP*TPS/PS
      RLRS=RLR
      RRRS=RRR
      RPS=RP
      TLRS=TLR
      TRRS=TRR
      XKL=(DUTLR+RLR)/DUTLR
      XKR=(DUTRR+RRR)/DUTRR
      PP=PS
      P=PS
      TLR=TLRS*((P*RLRS/(PL*RLR))**((XKL-1.)/XKL))
      TRR=TRRS*((P*RRRS /(PR*RRR))**((XKR-1.)/XKR))
      TP=TPS
      CALL RENERG(P ,TLR,FL,EGRL,FEL,ULR,DUTLR,RLR,QHV,FAS,FAPC,FMWT)
      CALL RENERG(P ,TRR,FR,EGRR,FER,URR,DUTRR,RRR,QHV,FAS,FAPC,FMWT)
      ERRW=CL*ULR+CR*URR-URA
      P=PS-10.
30    TLR=TLRS*((P*RLRS/(PL*RLR))**((XKL-1.)/XKL))
      TRR=TRRS*((P*RRRS /(PR*RRR))**((XKR-1.)/XKR))
      TP=(P*TVOL-CL*RLR*TLR-CR*RRR*TRR)/(RP*ALP)
      CALL RENERG(P,TLR,FL,EGRL,FEL,ULR,DUTLR,RLR,QHV,FAS,FAPC,FMWT)
      CALL RENERG(P,TRR,FR,EGRR,FER,URR,DUTRR,RRR,QHV,FAS,FAPC,FMWT)
      CALL ENERGY(P,TP,FP,FAS,UP,FAPC,RP,DUP,DUT,DUF,DRP,DRT,DRF)
      ERW=ALP*(UP-UPS)+CL*ULR+CR*URR-URA
      IF(ABS(ERW)-ERRL)10,10,20
20    PPP=P
      P=P-ERW*(P-PP)/(ERW-ERRW)
      PP=PPP
      ERRW=ERW
```

```
      GO TO 30
10    CONTINUE
      RETURN
      END
      SUBROUTINE EQFLT(TP,P,TR,PHE,QHV,FAS,FAPC,EGR,FE,FMWT)
      DATA DHE/.2/
      CALL RENERG(P,TR,PHE,EGR,FE,UR,DUTR,RR,QHV,FAS,FAPC,FMWT)
      HR=UR+RR*TR
      FA=EGR*FE/(1.+FE*FAS)+(1.-EGR)*PHE/(1+PHE*FAS)
      FP=FA/(1.-FAS*FA)
      TP=TR+3000.
110   CALL ENERGY(P,TP,FP ,FAS,UP,FAPC,RP,DUP,DUT,DUF,DRP,DRT,DRF)
      HP=UP+RP*TP
      DH=HR-HP
30    IF(ABS(DH)-DHE)10,10,20
20    TP=TP+DH/(DUT+RP+DRT*TP)
      GO TO 110
10    CONTINUE
      RETURN
      END
      SUBROUTINE NL(XNOP,XNOPP,PP,PF,TPP,TP,PHE,ALL,AL,DT,CAT,HAT,OAT,
     1IERR,NOFR,KLO)
      COMMON/BLOC/C    ,H   ,O   ,F   ,T  ,P    ,XH,XO,XN,XH2,XOH,XCO,XNO,XO2,
     1XH2O,XCO2,XN2,XAR, X13,I     ,N     ,K
      DATA V/1.98588/
      C=CAT
      H=HAT
      O=OAT
      F=PHE
      T=(TPP+TP)/2.
      P=(PP+PF)/2.
      TEE=T/1.8
      CC=P/(14.7*82.0567*TEE)
      N=NOFR
      K=KLO
```

```
      XNOCC=XNOPP*PP/(1.E+6*14.7*82.0567*TPP/1.8)
      CALL EQBM
      IERR=I
      IF(I.NE.0) GO TO 99
      A1=XN2*XO*7.E13*EXP(-7.5E4/(V*TEE))*CC*CC
      A2=XO2*XN*13.3E9*TEE*EXP(-7080./(V*TEE))*CC*CC
      B1=XN*1.55E13*CC
      B2=XO*3.2E9*TEE*EXP(-39.1E3/(V*TEE))*CC
      A=A1+A2
      B=B1+B2
      XNOC=(ALL/AL)*(XNOCC*TPP*PF/(TP*PP)+(A/B-XNOCC)*(1.-EXP(-B*DT)))
     1 +(1.-ALL/AL)*A*DT/2.
      XNOP=XNOC*1.E+6*14.7*82.0567*TP/(1.8*PF)
      GO TO 10
   99 WRITE(6,100)IERR
  100 FORMAT('  SUBROUTINE EQBM FAILED - ERROR CODE IS',I4)
   10 CONTINUE
      RETURN
      END
      SUBROUTINE PRD(P,TP,FP,DTVOL,TVOLL,FAS,FAPC,DT)
      CALL ENERGY(P,TP,FP ,FAS,UP,FAPC,RP,DUP,DUT,DUF,DRP,DRT,DRF)
      ZP=(TP*DRT+RP)/(DUT+TP*DRT+RP)
      DP=(-P*DTVOL        )/(TVOLL-TP*DRP -ZP*(RP*TP/P-TP*DRP-DUP ))
      TVOL=TVOLL+DTVOL*DT
      DTP=(     DP*(RP*TP/P-TP*DRP -DUP ))/(DUT +TP*DRT +RP)
      TPP=TP
      DTPP=DTP
      TP=TP+DTP*DT
      P=RP*TP/TVOL
      CALL ENERGY(P,TP,FP ,FAS,UP,FAPC,RP,DUP,DUT,DUF,DRP,DRT,DRF)
      ZP=(TP*DRT+RP)/(DUT+TP*DRT+RP)
      DP=(-P*DTVOL        )/(TVOL-TP*DRP -ZP*(RP*TP/P-TP*DRP-DUP ))
      DTP=(     DP*(RP*TP/P-TP*DRP -DUP ))/(DUT +TP*DRT +RP)
      TP=TPP+(DTP+DTPP)*DT/2.
      P=RP*TP/TVOL
```

```
      RETURN
      END
      SUBROUTINE RENERG(P,TR,PHE,EGR,FE,UR,DUTR,RR,QHV,FAS,FAPC,FMWT)
      HFV=QHV-19186.+(((((1.5757E-15*TR-5.4331E-12)*TR-4.5082E-08)*TR
     1+3.6285E-04)*TR+3.4765E-02)*TR+15.791+155.07
      DHFVT=(((7.8785E-15*TR-2.1732E-11)*TR-1.3510E-07)*TR+7.2570E-04)
     1*TR+3.4765E-02
      RF=1.9859/FMWT
      UF=HFV-RF*TR
      DUTF=DHFVT-RF
      UA=((((6.3156E-17*TR-9.3632E-13)*TR+3.9016E-09)*TR+5.1979E-06)
     1*TR+.16528)*TR
      DUTA=(((3.1578E-16*TR-3.7452E-12)*TR+1.1704E-08)*TR
     1+1.03958E-05)*TR+.16528
      CALL ENERGY(P,TR, FE,FAS,UE,FAPC,RE,DUPE,DUTE,DUFE,DRPE,DRTE,DRFE)
10    UR=(1.-EGR)*(PHE*FAS*UF+UA)/(PHE*FAS+1.)+EGR*UE
      DUTR=(1.-EGR)*(PHE*FAS*DUTF+DUTA)/(PHE*FAS+1.)+EGR*DUTE
      RR=(1.-EGR)*(PHE*FAS*RF+.06857)/(PHE*FAS+1.)+EGR*RE
      RETURN
      END
      FUNCTION DELT(S)
      DIMENSION B(3)
      DATA B/5.E-5,1.E-4,1.E-3/
      I=1
      IF(S.GE.0.99E-3) I=I+1
      IF(S.GE.0.99E-2) I=I+1
      DELT=B(I)
      RETURN
      END
```

APPENDIX D
DEFINITIONS OF THE DEFORMANCE PARAMETERS USED IN CHAPTER II

Indicate Specific Nitric Oxide (ISNO)

$$\text{ISNO} = \frac{grams\ of\ NO(g/lb_m)\ 2544.4(Btu/hr\text{-}hr}{(W_{exp}-W_{comp})\ (\frac{Btu}{lb_m})}$$

The work required for exhaust and intake is assumed to be zero.

$$\text{Compression Work} = \frac{T_2 R(1-CR^{(1-k)})}{(1-k)\ \text{x}\ 778} = T_2\ \text{x}\ F^*$$

CR = compression ratio

*The term F was evaluated slightly differently for the one system and two system models.

Indicated Mean Effective Pressure (IMEP)

$$\text{IMEP} = \frac{(W_{exp}-W_{comp})(\frac{\text{Btu}}{\text{lbm}})\ \text{x}\ 788\ (\frac{\text{ft-lb}}{\text{Btu}})}{(v_1-v_2)(\frac{\text{ft}^3}{\text{lb}_m})\ \text{x}\ 144\ (\frac{\text{in}^2}{\text{ft}^2})}$$

Indicated Ethalpy Efficiency (IEE)

$$\text{IEE} = \frac{(w_{exp}-w_{comp})(\frac{\text{Btu}}{\text{lb}_{total\ mix}})}{\text{Lower Heat of Comb.}\ (\frac{\text{BTU}}{\text{lb}_{fuel}})\dfrac{\text{FA}(\frac{\text{lb}_{fuel}}{\text{lb}_{air}})(1-\text{EGR})(\frac{\text{lb}_{FA_{mix}}}{\text{lb}_{total\ mix}})}{(1+FA)(\frac{\text{lb}_{FA_{mix}}}{\text{lb}_{air}})}}$$

APPENDIX E

AIR INJECTION FLOW RATE MEASUREMENT

The flow rate of high pressure air going to the air injection valve is measured by means of a flow restriction. The flow within the flow restriction is turbulent and is assumed to be isothermal. The heavy brass walls of the flow restriction and its very small cross sectional area will assist in making the flow more isothermal. A computer program was written which calculates the flow rate as a function of the inlet pressure, inlet temperature and the exit pressure. This computer program is incorporated into the data reductions computer programs.

When the computer program was compared to the calibration data it was found that the friction factor equation had to be adjusted in order to obtain a fit. Shown in Figure E1 is a plot of the actual and calculated exit pressure of the flow restriction versus the actual pressure drop across the flow restriction. Only a few points fall outside of an error of two psi. The smallest divisions on the pressure gage is one psi.

The subroutine called FRI calculates the flow rate from the measured inlet temperature, inlet pressure and outlet pressure. The program uses a trial and error method to determine the flow necessary to have a calculated outlet pressure equal to the measured outlet pressure. Many of the equations used were obtained from Chapman (1971), the friction factor equivalence were obtained from Kays (1966).

The subroutine FRI follows.

```
      SUBROUTINE FRI(PATM,TO,PO,PBO,FR)
      DATA GC/32.1739/,R/53.36/,XK/1.399/,VES/12.4E-6/
     1,RH/4.066E-4/,XL/3.92/,A/1.112E-5/,Z/.105/,Y/.25/,Q/.2/
      TO=TO+459.69
      PO=(PO+PATM)*144.
      PBO=(PBO+PATM)*144.
      FRR=0.
      PB=PO
 60   FRA=FR/A
      CON=PO*SQRT(XK*GC/(R*TO))
      XMA=FRA/CON
 20   FRAA=CON*XMA/((1.+(XK-1.)*XMA*XMA/2.)**((XK+1.)/(2.*(XK-1.))))
      EER=(FRA-FRAA)/FRA
      IF(ABS(EER)-1.E-4)30,30,10
 10   XMA=(FRA-FRAA)*XMA/FRAA+XMA
      GO TO 20
 30   VFA=1.+(XK-1.)*XMA*XMA/2.
      TA=TO/VFA
      PA=PO/(VFA**(XK/(XK-1.)))
      CA=SQRT(XK*R*TA*GC)
      VA=XMA*CA
      REA=VA*4.*RH*PA/(R*TA*VES)
      IF(REA-3.E4)31,31,32
 32   E=Z*((3.E4)**(Q-Y))
      F=E/(REA**Q)
      GO TO 33
 31   F=Z/(REA**Y)
 33   CONTINUE
      DFA=1./(XK*XMA*XMA)+ALOG(XMA*XMA*XK)
      DFB=DFA-F*XL/RH
      IF(DFB)61,61,62
 62   XMB=.1
 40   DFBB=1./(XK*XMB*XMB)+ALOG(XMB*XMB*XK)
      ERR=(DFB-DFBB)/DFB
      IF(ABS(ERR)-1.E-4)50,50,80
```

```
80    XMB=XMB+(DFB-DFBB)*(XMA-XMB)/(DFA-DFBB)
      GO TO 40
50    PBB=PB
      PB=PA*XMA/XMB
      ERR=PB-PBO
      IF(ABS(ERR)-.1)55,55,70
70    FRS=FR
100   FR=FR-ERR*(FR-FRR)/(PB-PBB)
      FRR=FRS
      GO TO 60
61    FR=.8*A*CA*PBO /(R*TA)
      PRINT,FR
55    CONTINUE
      RETURN
      END
```

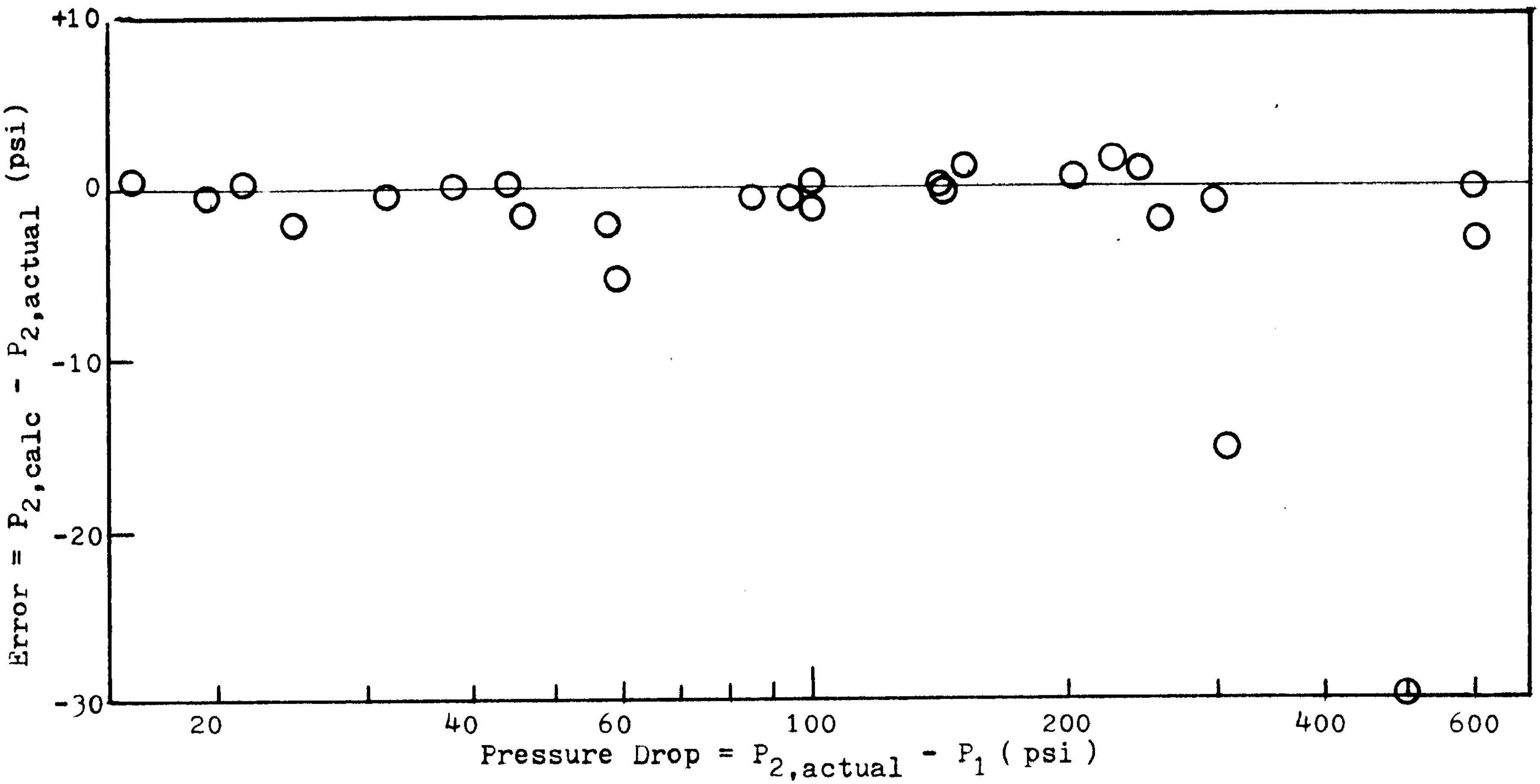


Figure E1 Comparison of Measured and Calculated Flow Restriction Pressures for the Same Flow Rate

APPENDIX F

MECHANICAL DESIGN OF AIR INJECTION VALVE

The air injection valve is a spring loaded and is activated by a cam engaging the cam follower. A sketch of the air injection valve is shown in Figure F1. The cam follower is mounted in a holder which can slide back and forth to open and close the valve. The spring is used to keep the valve closed. Downstream of the valve seat is a flow restriction which prevents back flow of high pressure and temperature products of combustion. The large diameter nozzle is shown in the sketch, it screws into the engine.

The cam is mounted on a disc that is connected to a shaft which in turn is mounted on two bearings. The drive sprocket is keyed to the shaft. The shaft is perpendicular to the center line of the valve steam. With the valve closed, the center line of the shaft is five inches from the center line of the cam follower.

A computerized milling machine was used to make the cam surface. An eight-power polynomial as described by Mabie et al. (1958) is used for the motion of the cam. A computer program is used to calculate the shape of the cam surface. The computer program for calculating the cam surface follows.

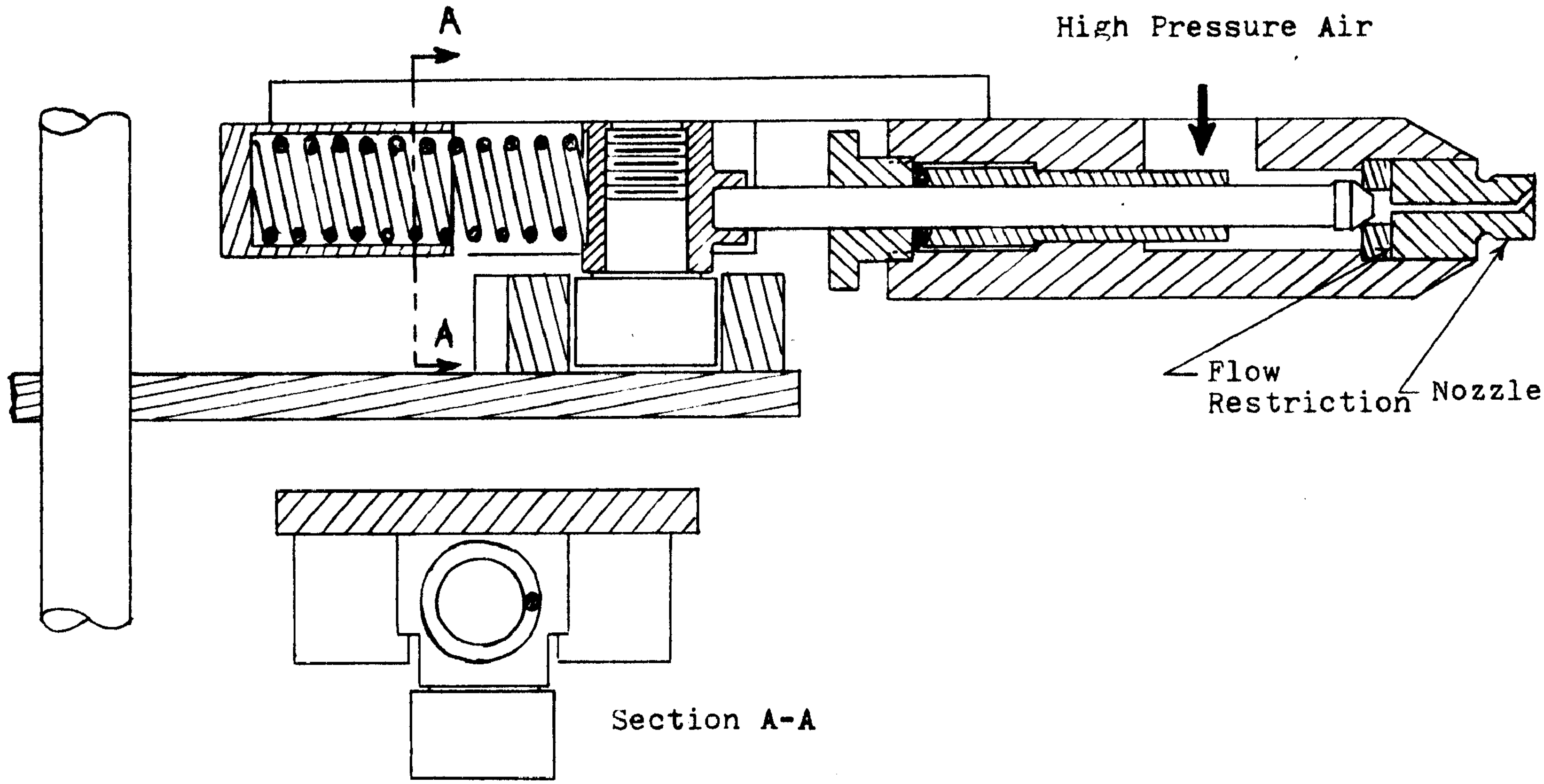


Figure F1 A Sketch of the Air Injection Valve

A second computer program is necessary to convert the shape of the cam to punched tape instruction for the milling machine. M. Fugelso very capably made the punch tape and operated the milling machine.

```
      XL=.125
      RO=5.
      R=.625
      C=15.
      T=1./(C*10.)
      A=C*3.1416/180.
      KK=C*10
      DO 10 K=1,KK
      BB=I*C
      B=BB*3.1416/180.
       D=6.09755*(T**3)-20.7304*(T**5)+26.73155*(T**6)-13.60965*(T**7)
       +2.56095*(T**8))*B
      DD=(XL/A)*(6.09755*3.*(T**2)-20.7304*5.*(T**4)+26.73155*6.*(T**5)
       -13.60965*7.*(T**6)+2.56095*8.*(T**7))
      RR=RO-D*XL
      AL=  ATAN( DD/RR)
      RP=SQRT(RR**2+R**2-2.*RR*R*COS(AL))
      BETA=ATAN(R*SIN(AL)/(RR-R*COS(AL)))
      GAM=3.1416-AL-BETA
      RQ=SQRT(RP**2+4.*(R**2)-4.*R*RP*COS(GAM))
      TP=(B-BETA)*180./3.1416
      TQ=TP+ATAN(4.*R*RP*SIN(GAM)/(RP**2+RQ**2-4.*R**2))*180./3.1416
      ALL=AL*180./3.1416
      BET=BETA*180./3.1416
      GA=GAM*180./3.1416
      PRINT,BB,RP,TP,RQ,TQ,RR,ALL,BET,GA
10    T=T+.1/C
      END
```

APPENDIX G

PERFORMANCE OF AIR INJECTION VALVE

A computer program was written which predicts the flow rate through the nozzle for various angular positions of the cam. The program also calculates the dynamic forces resulting from the cam engaging the cam follower. The flow is assumed to be isentropic. The flow is not allowed to be supersonic because neither the valve seat or the air injection nozzle have a converging-diverging shape.

The flow path through the valve starts with an isentropic compression to the valve opening area, then a sudden enlargement to the inside diameter of the valve seat and followed by an isentropic compression to the diameter of the nozzle. The subroutine ISENT handles isentropic compressions, the STEP subroutine handles sudden steps in the diameter and when the nozzle is near choked flow it is necessary to reverse the calculations of the step which is done by PETS.

The air injection valve performance computer program follows.

```
      DATA GC/32.1739/,R/53.36/,XK/1.399/,
     1Q/3.14159/,QQ/.0174532/,U/1./,SK/40./,FU/100./
      READ,NUR
      PRINT,NUR
      READ,DC
      PRINT,DC
      DC=DC/12.
      READ,RPM
      PRINT,RPM
      READ,PO,TO
      PRINT,PO,TO
      READ,POC,ALO
      PRINT,POC,ALO
      ALO=ALO*QQ
      DB=.3125/12.
      XLR=4.
      C=.166667
      VGO=C+.5-COS(ALO)/2.+XLR*(1.-SQRT(1.-((SIN(ALO)/XLR)**2)))/2.
 60   READ,XL,B,TH
      PRINT,XL,B,TH
      TO=TO+459.69
      PO=PO*144.
      AO=ALO+TH*2.*QQ
      SO=TH/(6.*RPM)
      VG=C+.5-COS(AO)/2.+XLR*(1.-SQRT(1.-((SIN(AO)/XLR)**2)))/2.
      PE=POC*((VGO/VG)**XK)
      PE=PE*144.
      T=TH/B
      IF(T-1.)11,11,12
 12   T=ABS(T-2.)
 11   CONTINUE
      D=6.09755*(T**3)-20.7804*(T**5)+26.73155*(T**6)-13.60965*(T**7)
     1 +2.56095*(T**8)
      ACC=(36.58530*T-415.308*(T**3)+801.9465*(T**4)
     1-571.6053*(T**5)+143.4132*(T**6))*XL/(B**2)
```

```
      DD=(XL/B)*(6.09755*3.*(T**2)-20.7804*5.*(T**4)+26.73155*6.*(T**5)
     1 -13.60965*7.*(T**6)+2.56095*8*(T**7))
      RR=5.-D*XL
      AL=ATAN(DD/(RQ*RR))
      XL=XL/12.
      XD=XL*D
      IF(XD-DB)39,38,37
38    AA=Q*DB*DB/2.8284
      GO TO 37
39    AA=Q*(XD*DB-XD*XD/2.)*.7071
37    POE=PO/PE
      AA=AA*.73
      AB=Q*DB*DB/4.
      AC=Q*DC*DC/4.
      RAB=AA/AB
      RBC=AB/AC
      RAC=RAB*RBC
      XM=SQRT((-1.+(POE**((XK-1.)/XK)))*2./(XK-1.))
      IF(1.-XM)41,42,42
41    XM=1.0
42    IF(RAC-1.)15,15,16
15    XMA=XM
      GO TO 17
16    XMA=XM/RAC
17    ERR=POE
      XMAA=0.
95    IF(RAB-1.)21,21,22
22    CALL ISENT(XMA,RAB,XMB,PAB,TAB,LA)
      GO TO 23
21    CALL STEP(XMA,RAB,XMB,PAB,TAB,LA)
23    CALL ISENT(XMB,RBC,XMC,PBC,TBC,LB)
      IF(LB-1)90,90,91
91    RBA=1./RAB
      IF(RAB-1.)93,92,92
93    CALL  PETS(XMA,RAB,XMB,PAB,TAB,LA)
```

```
      IF(LA-1 )90,90,97
97    CALL STEP(XMA,RAB,XMB,PAB,TAB,LA)
      CALL ISENT(XMB,RBC,XMC,PBC,TBC,LB)
      GO TO 90
92    CALL ISENT(XMB,RBA,XMA,PBA,TBA,LA)
96    PAB=1./PBA
      TAB=1./TBA
90    POA= (1.+(XK-1.)*XMA*XMA/2.)**(XK/(XK-1.) )
      TOA=1.+(XK-1.)*XMA*XMA/2.
      POEE=POA*PAB*PBC
      ERRR=ERR
      ERR=POE-POEE
      IF(ABS(ERR)-1.E-4)75,75,83
83    IF(LA+LB-3)80,87,87
87    IF(ERR)89,75,75
89    IF(LB-1)80,80,65
65    IF(ERR-ERRR)66,76,66
66    XMA=XMA-1.2*ERR*(XMA-XMAA)/(ERR-ERRR)
      GO TO 95
80    XMS=XMA
      IF(ERR-ERRR )88,76,88
88    XMA=XMA-ERR*(XMA-XMAA)/(ERR-ERRR)
      XMAA=XMS
      GO TO 95
76    PRINT,ERR,POE,POEE,XMA,XMB,XMC
75    CON=PO*SQRT(XK*GC/(R*TO))
20    FRAA=CON*XMA/((1.+(XK-1.)*XMA*XMA/2.)**((XK+1.)/(2.*(XK-1.))))
      FRA=FRAA*AA
      PO=PO/144.
      PE=PE/144.
      XD=XD*12.
      AA=AA*144.
      PA=PO/POA
      TA=TO/TOA
      IF(1.-XMA)77,77,78
```

```
77    PB=PE
      TB=0.
      PC=0.
      TC=0.
      FRB=0.
      FRC=0.
      GO TO 79
78    PB=PA/PAB
      TB=TA/TAB
      PC=PB/PBC
      TC=TB/TBC
      FRB=144.*PB*AB*XMB*SQRT(XK*GC/(R*TB))
      FRC=144.*PC*AC*XMC*SQRT(XK*GC/(R*TC))
79    FCR=FU+SK*XD+3.*W*RPM*RPM*ACC/GC-AB*PB*144.
      FCA=SIN(AL)*FCR/COS(AL)
      FCN=FCR/COS(AL)
      ALLL=AL/GQ
      TO=TO-459.69
      TA=TA-459.69
      TB=TB-459.69
      TC=TC-459.69
      PRINT,FCR,FCA,FCN,ALLL
      PRINT,XD,AA,PAB,PBC,PAC
      PRINT,XMA,XMB,XMC
      PRINT,FRA,FRB,FRC
      PRINT,SO
      PRINT,PO,PA,PB,PC,PE
      PRINT,TO,TA,TB,TC
      NUR=NUR-1
      IF(NUR)70,70,60
70    STOP
      END
```

```
      SUBROUTINE ISENT(XMA,RAB,XMB,PAB,TAB,L)
      DATA XK/1.399/
      L=1
      RAS=(((1.+(XK-1.)*XMA*XMA/2.)/((XK+1.)/2.))**((XK+1.)/(2.*(XK-1)))
     1)/XMA
      IF(RAS-RAB)61,61,60
61    XMB=1.
      CB=XMB/((1.+(XK-1.)*XMB*XMB/2.)**((XK+1.)/(2.*(XK-1.))))
      XMAA=0.
      ERR=1.
      XMA=CB
31    CA=(XMA*RAB)/((1.+(XK-1.)*XMA*XMA/2.)**((XK+1.)/(2.*(XK-1.))))
      ERRR=ERR
      ERR=CB-CA
      IF(ABS(ERR)-1.E-6)40,40,51
51    XMS=XMA
      IF(ERR-ERRR)100,43,100
100   XMA=XMA-ERR*(XMA-XMAA)/(ERR-ERRR)
      XMAA=XMS
      GO TO 31
60    CA=(XMA*RAB)/((1.+(XK-1.)*XMA*XMA/2.)**((XK+1.)/(2.*(XK-1.))))
      XMBB=0.
      ERR=1.
      XMB=CA
30    CB=XMB/((1.+(XK-1.)*XMB*XMB/2.)**((XK+1.)/(2.*(XK-1.))))
      ERRR=ERR
      ERR=CA-CB
      IF(ABS(ERR)-1.E-7)40,40,50
50    XMS=XMB
      IF(ERR-ERRR)101,43,101
101   XMB=XMB-ERR*(XMB-XMBB)/(ERR-ERRR)
      XMBB=XMS
      GO TO 30
43    PRINT,ERR,CA,CB,XMA,RAB,XMB
40    FBA=(1.+(XK-1.)*XMB*XMB/2.)/(1.+(XK-1.)*XMA*XMA/2.)
```

```
      PAB=XMB*SQRT(FBA)/(XMA*RAB)
      TAB=FBA
      IF(XMB-1.)70,71,71
71    L=2
70    CONTINUE
      RETURN
      END
      SUBROUTINE  STEP(XMA,RAB,XMB,PAB,TAB,L)
      DATA XK/1.399/
      L=1.
62    IF(XMA-1.)60,61,61
61    XMA=1.
      L=2
60    CA=XK*XMA*XMA+1./RAB
      XMB=1.
      CB=XMA*SQRT((1.+(XK-1.)*XMA*XMA/2.)/(1.+(XK-1.)*XMB*XMB/2.))*
     1(1.+XK*XMB*XMB)/XMB
      ERR=(CA-CB)/CA
      XMBB=XMB
      XMB=XMA*RAB
30    CB=XMA*SQRT((1.+(XK-1.)*XMA*XMA/2.)/(1.+(XK-1.)*XMB*XMB/2.))*
     1(1.+XK*XMB*XMB)/XMB
      ERRR=ERR
      ERR=(CA-CB)/CA
      IF(ABS(ERR)-1.E-5)40,40,50
50    XMS=XMB
      IF(ERR-ERRR)101,43,101
101   EF=-ERR*(XMB-XMBB)/((ERR-ERRR)*XMB)
102   IF(ABS(EF)-.1)100,100,90
90    EF=EF/2.
      GO TO 102
100   XMB=XMB+EF*XMB
      XMBB=XMS
```

```
      GO TO 30
43    PRINT,ERR,CA,CB,XMA,RAB,XMB
40    FBA=(1.+(XK-1.)*XMB*XMB/2.)/(1.+(XK-1.)*XMA*XMA/2.)
      PAB=XMB*SQRT(FBA)/(XMA*RAB)
      TAB=FBA
      RETURN
      END
      SUBROUTINE  PETS(XMA,RAB,XMB,PAB,TAB,L)
      DATA XK/1.399/
      L=1.
      CB=1.+(XK-1.)*XMB*XMB
      XMA=1.
      CA=XMB*SQRT((1.+(XK-1.)*XMB*XMB/2.)/(1.+(XK-1.)*XMA*XMA/2.))*
     1(XK*XMA*XMA+1./RAB)/XMA
      ERR=(CB-CA)/CB
      XMAA=XMA
      XMA=XMB/RAB
30    CA=XMB*SQRT((1.+(XK-1.)*XMB*XMB/2.)/(1.+(XK-1.)*XMA*XMA/2.))*
     1(XK*XMA*XMA+1./RAB)/XMA
      ERRR=ERR
      ERR=(CB-CA)/CB
      IF(ABS(ERR)-1.E-5)40,40,50
50    XMS=XMA
      IF(ERR-ERRR)101,43,101
101   EF=-ERR*(XMA-XMAA)/((ERR-ERRR)*XMA)
102   IF(ABS(EF)-.1)100,100,90
90    EF=EF/2.
      GO TO 102
100   XMA=XMA+EF*XMA
      XMAA=XMS
      GO TO 30
43    PRINT,ERR,CA,CB,XMA,RAB,XMB
40    IF(XMA-1.)60,61,61
61    XMA=1.
      L=2
```

```
60    FBA=(1.+(XK-1.)*XMB*XMB/2.)/(1.+(XK-1.)*XMA*XMA/2.)
      PAB=XMB*SQRT(FBA)/(XMA*RAB)
      TAB=FBA
      RETURN
      END
```

APPENDIX H

DATA REDUCTION COMPUTER PROGRAMS

The data obtained from each operating condition (run) is punched on computer cards to expedite the calculations. Two computer programs are available one with a short output in which the important results are on one line, the other longer output provides more detail. One important advantage of using a computer program to reduce the data is that, if changes are desired in the calculations, all of the calculations can easily be repeated.

The long output and then the short output computer programs follow. Note that the subroutine FRI called by the programs is included in Appendix E.

APPENDIX I

CONVERSIONS TO SI UNITS

Term	Units Used In Report	To Obtain SI Units	SI Units
Length	inches	x 25.4	milli~~meters~~ (mm)
Force	lb_f	x 4.448	Newton (N)
Mass	lb_m	÷ 2.2	Kilogram (kg)
Temperature	°R	÷ 1.8	Deg. Kelin (°K)
Energy	BTU	x 1055.	Joule (J)
Energy	hp-hr	x 2684519.	Joule or Watt-sec
Power	hp	x 745.7	Watt (W)
Pressure	psi	x 6895.	Pascal (Pa) (N/m^2)
IMEP	"	"	"
ISNO	g/ihp-hr	x .0003725	kg/J or kg/W-sec.
CFM	ft^3/min.	x. 0004719	m^3/sec.
RPM	rev./min.	÷ 60.	Hertz (Hz)(rev./sec.)

```
      DIMENSION TITLE(23)
      DATA R/53.36/,TA/459.69/,W/.04409 /,XK/1.395/,VC/37.33/
     1,FAS/.06628/
      WRITE(6,45)
45    FORMAT(//'  SUMMARY OF RESULTS  -F'//)
      READ,NUR
      PRINT,NUR
150   READ(5,35)TITLE
      WRITE(6,35)TITLE
35    FORMAT(3X,23A3)
      READ,T1,T2,T3,T4,T5
      WRITE(6,25)T1,T2,T3,T4,T5
25    FORMAT(' T1=',F7.1,'F   T2=',F7.1,'  T3=',F7.1,'  T4=',F7.1,
     1'  T5=',F7.1)
      READ,PATM,P2,P4A,P4B,P4C
      WRITE(6,15)PATM,P2,P4A,P4B,P4C
15    FORMAT(' PATM=',F7.3,'PSIA  P2=',F7.3,'IN HG  P4A=',F7.1,'PSIG
     1P4B=',F7.1,'  P4C=',F7.1)
      READ,HC,CO,XNO,XNOX
      WRITE(6,135)HC,CO,XNO,XNOX
135   FORMAT(' EMISSIONS- HC=',F7.0,'PPM  CO=',F7.2,'PERCENT  NO='
     1,F7.0,'PPM  NOX=',F7.0,'PPM')
      READ,DH1,DT1,FP,FM
      WRITE(6,55)DH1,DT1,FP,FM
55    FORMAT(' DH1=',F6.3,'INCHES  DT1=',F7.2,'SECONDS  FP=',F7.2,
     1'LBS   FM=',F7.2)
      READ,XN,C,S
      WRITE(6,65)XN,C,S
65    FORMAT(' N=',F6.2,'  C=',F6.2,'CM  S=',E10.3,'SEC/CM')
      READ,FRIA
      WRITE(6,175)FRIA
175   FORMAT(' FRIA=',F8.6,'LB/SEC')
      READ,RMR,T13,T14
      WRITE(6,185)RMR,T13,T14
  185 FORMAT(' RMR=',F7.3,'   T13=',F7.1,'=  T14=',F7.1/)
```

```
      IF(FRIA)11,11,12
12    CALL FRI(PATM,T4,P4A,P4B,FRIA)
11    FRF=W/DT1
      FRCA=PATM*144.*5.9643*DH1/(60.*R*(T1+TA))
      WRITE(6,75)FRF,FRCA,FRIA
75    FORMAT(' MASS FLOW RATE-  FUEL=',F9.7,'LBS/SEC  CARBURETED AIR='
     1,F8.6,'  INJECTED AIR=',F8.6)
      FRTA=FRCA+FRIA
      RIT=FRIA/FRTA
      RCT=FRCA/FRTA
      WRITE(6,85)RCT,RIT
85    FORMAT(' FRACTION OF AIR CARBURETED=',F7.5,'  FRACTION OF AIR INJE
     1CTED=',F7.5)
      ERO=FRF/(FRTA*FAS)
      ERC=FRF/(FRCA*FAS)
      WRITE(6,95)ERO,ERC
95    FORMAT(' EQUIVALENCE RATIO-  OVERALL=',F7.4,'  CARBURETED=',F7.4)
      RPM=XN*60./(C*S)
      WRITE(6,195)RPM
195   FORMAT(' RPM=',F8.2)
      HPM=FM*RPM/4000.
      HPB=FP*RPM/4000.
      P2=P2/2.036
      HPC=FRIA*((R*(T2+TA)*((((P4C+PATM)/(P2+PATM))**((XK-1.)/XK))-1.))/
     1(XK-1.)+P2*VD/12.)/(550.*.80)
      HPBC=HPB-HPC
      HPI=HPB+HPM
      HPIC=HPI-HPC
      WRITE(6,105)HPB,HPC,HPBC,HPM,HPI,HPIC
105   FORMAT(' HORSPOWER-  GROSS BRAKE=',F7.4,'HP  EST INJ AIR COMP=',
     1F7.4,'  NET BRAKE=',F7.4,'  MOTORED=',F7.4,'  GROSS IND=',
     2F7.4,'  NET IND=',F7.4)
      A=42.208/(20747*60.*FRF)
      EEB=HPB*A
      EEBC=HPBC*A
```

```
      EEI=HPI*A
      EEIC=HPIC*A
      WRITE(6,115)EEB,EEBC,EEI,EEIC
115   FORMAT(' ENTHALPY EFFICIENCY-  GROSS BRAKE=',F6.4,'  NET BRAKE='
     1,F6.4,'  GROSS INDICATED=',F6.4,'  NET INDICATED=',F6.4)
      B=21215.60/RPM
      BMEP=HPB*B
      BCMEP=HPBC*B
      CMEP=HPC*B
      XIMEP=HPI*B
      XICMP=HPIC*B
      WRITE(6,125)BMEP,BCMEP,CMEP,XIMEP,XICMP
125   FORMAT(' MEAN EFFECTIVE PRESSURE-  GROSS BRAKE=',F7.2,
     1'  NET BRAKE=',F7.2,'  COMPRESSION=',F7.2,'  GROSS IND=',F7.2,
     2'  NET IND=',F7.2)
      Y=(4.5+12.5*4.76/ERD-CO*.045)/(1.-CO/200.)
      D=453.59*Y*FRF*3600./114.232
      E=10.E-6*30.008*D
      BSNOX=XNOX*E/HPB
      BCSNX=XNOX*E/HPBC
      XISNX=XNOX*E/HPI
      XICSX=XNOX*E/HPIC
      WRITE(6,155)BSNOX,BCSNX,XISNX,XICSX
155   FORMAT(' SPECIFIC NOX-  GROSS BRAKE=',E12.5,'GM/(HP-HR)'
     1'  NET BRAKE=',E12.5,'  GROSS IND=',E12.5,'  NET IND=',E12.5)
      F=.01*28.011*D
      BSCO=CO*F/HPB
      BCSCO=CO*F/HPBC
      XISCO=CO*F/HPI
      XICSO=CO*F/HPIC
      WRITE(6,165)BSCO,BCSCO,XISCO,XICSO
165   FORMAT(' SPECIFIC CO-  GROSS BRAKE=',E12.5,'GM/(HP-HR)'
     1'  NET BRAKE=',E12.5,'  GROSS IND=',E12.5,'  NET IND=',E12.5)
      HTCWS=RMW*0.05667*(T13-T14)
      HTCWL=HTCWS/(FRTA+FRF)
```

```
      WRITE(6,205)HTCWS,HTCWL
  205 FORMAT(' HEAT TRANSFERRED TO COOLING WATER -',F8.1,' BTU/SEC',
     1F8.1,' BTU/LB'//)
      NUR=NUR-1
      IF(NUR)59,59,150
59    STOP
      DIMENSION TITLE(24)
      DATA R/53.36/,TA/459.69/,W/.24407 /,XK/1.399/,VE/37.33/
     1,FAS/.06628/
      WRITE(6,45)
45    FORMAT(//'  SUMMARY OF RESULTS  -E'//)
      WRITE(6,201)
201   FORMAT('  RUN  ERO   FFC    FLCA   SI-CA  SP-CA  PPM    IHP     CHP
     1   IEEG  IEEN  IMEPG IMEPN  NOX-PPM  NOX-SG  CO-PER  CO-SG  HC-PPM
     2')
      READ,NUR
      READ,N
150   READ(5,35)TITLE
35    FORMAT(24A3)
      READ,T1,T2,T3,T4,T5
      READ,PATM,P2,P4A,P4B,P4C
      READ,HC,CO,XNO,XNOX
      READ,DH1,DT1,FP,FM
      READ,XN,C,S
      READ,FRIA
      READ,RAR,T13,T14
      IF(FRIA)11,11,12
12    CALL FRI(PATM,T4,P4A,P4B,FRIA)
```

```
11      FRF=W/DT1
        FRCA=PATM*144.*5.9643*DH1/(60.*R*(T1+TA))
        FRTA=FRCA+FRIA
        ERO=FRF/(FRTA*FAS)
        ERC=FRF/(FRCA*FAS)
        RPM=XN*60./(C*S)
        HPM=FM*RPM/4000.
        HPB=FP*RPM/4000.
        P2=P2/2.036
        HPC=FRIA*((R*(T2+TA)*(((P4C+PATM)/(P2+PATM))**((XK-1.)/XK))-1.))/
       1(XK-1.)+RPM*(PATM-P2)*VD/1440.)/(550.*.8)
        HPI=HPB+HPM
        HPIC=HPI-HPC
        A=42.208/(20747*60.*FRF)
        EEI=HPI*A
        EEIC=HPIC*A
        B=21215.60/RPM
        XIMEP=HPI*B
        XICMP=HPIC*B
        Y=(4.5+12.5*4.76/ERO-CO*.045)/(1.-CO/200.)
        D=453.59*Y*FRF*3600./114.232
        E=10.E-6*30.008*D
        XISNX=XNOX*E/HPI
        F=.01*28.011*D
        XISCO=CO*F/HPI
        WRITE(6,200)N,ERO,ERC,FRCA,RPM,HPI,HPC,EEI,EEIC,XIMEP,XICMP
       1,XNOX,XISNX,CO,XISCO,HC
200     FORMAT(I5,2F6.3,F7.5,14X,F8.2,2F7.4,2F6.4,2F7.2,F6.0,F9.3,F7.3,F9.
       13,F7.0/)
        N= +1
        NUR=NUR-1
        IF(NUR)59,59,150
59      STOP
        END
```

TECHNICAL REPORT DATA
(Please read Instructions on the reverse before completing)

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16. ABSTRACT

The purpose of this study is to explore methods of controlling the nitrogen oxide emissions from internal combustion engines. From computer calculations, the delayed mixing stratified charge engine concept was selected. In the delayed mixing stratified charge engine concept, combustion is initiated and completed in a fuel-rich region, then air is mixed into those rich products. A study of existing engines shows that some operational stratified charge engines limit nitrogen oxide emissions in a manner similar to the delayed mixing concept. A single cylinder engine was modified to include an air injection valve. When air was injected after rich combustion, the nitrogen oxide emissions were lower, the hydrocarbon emissions were lower, the carbon monoxide emissions were about the same and the efficiencies were lower than for homogeneous operation at the same overall fuel-air ratio.

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS	b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
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