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AN INVESTIGATION ON THE EMISSIONS IN A TORCH CHAMBER S.I.E.

BY
M.M.Avad, A.A.Desoky and A.Abdelkhalek

دراسة عن ملوثات البيئة المنبعثة من محرك اشعال بالشرارة ذو نافورة لهب

الخلاصة – يشتمل هذا البحث على الدراسة العملية والنظرية لمحرك اشعال بالشرارة ذو نافورة لهب لاظهار تأثير بعض العوامل المتعلقة بالتصميم والتشغيل على ملوثات البيئة المنبعثة من المحرك وقد أجريت الدراسة العملية على محرك دويتز - ديزل معدل ليعمل كمحرك اشعال بالشرارة ذو نافورة لهب . وقد تم دراسة تأثير عاملين رئيسيين من عوامل التصميم الهندسي وهما الشكل الهندسي للفوهة الموصلة من الغرفة المساعدة وغرفة الاحتراق الرئيسية ونقطة بدء الاشعال داخل غرفة الاحتراق المساعدة وعوامل التشغيل التي درست هي نسبة الهواء الى الوقود وسرعة المحرك وفتحة الخانق للمغذي . وقد أجريت التجارب المعملية باستخدام ثلاث مواضع للاشعال وفوهتين بين غرفتي الاحتراق . وقد تم أيضا تطوير محاكاة رياضية لعمليات الانضغاط والاحتراق والتمدد لمحرك اشعال بالشرارة ذو نافورة لهب . وتم في هذا النموذج دراسة تأثير نسبة الهواء الى الوقود على ملوثات البيئة المنبعثة من هذا المحرك وقد أستخدمت بعض النتائج العملية التي تم الحصول عليها على نفس المحرك كمدخلات للنموذج الرياضي . وقد أظهرت النتائج العملية أن تصميم الفوهة الموصلة بين غرفتي الاحتراق له تأثير كبير على ملوثات البيئة المنبعثة من محركات الاشعال بالشرارة ذو نافورة اللهب . وكذلك أظهرت النتائج أنه كلما استخدم شمعة احتراق أطول داخل غرفة الاحتراق المساعدة كلما قلت ملوثات البيئة في نواتج الاحتراق . وقد اظهرت النتائج العملية أن أفضل الحالات هو استخدام فوهة توصيل لامسة معرفة مع شمعة الاحتراق الممتدة . وقد اظهرت النتائج النظرية الاتفاق في الاتجاه مع النتائج العملية .

ABSTRACT

Experimental and theoretical investigations were made on a torch chamber spark ignition engine to determine the effect of some design and operational parameters on engine emissions. The experimental study was performed using a modified Deutz-Diesel engine developed to be a torch chamber spark ignition engine. During this investigation, two main design parameters were investigated namely, the connecting passage shape and the flame initiation point. In this work three ignition point location and two different connecting passage shapes were investigated over a wide range of mixture strength, engine speed and throttle opening. A mathematical model of compression, combustion and expansion processes of the torch chamber spark ignition engine has also been developed. This model aims at estimating the effect of mixture strength on the NO and CO rates of formation in this engine. In developing the model, experimental data obtained on the same engine are used. The experimental results reveals that, the configuration of the connecting passage has a great effect on emissions from the torch chamber S.I.E. It also reveals that, as the spark plug is extended inside the combustion chamber, engine emissions decrease. The optimum case is reached when using a convergent divergent passage coupled with the extended spark plug. The trends obtained from the mathematical simulation are in a good agreement with the experimental one.

1. INTRODUCTION

Air pollution is one of the great problems that appeared in the last few years. Internal combustion engines are considered to be one of the main sources of pollutants, especially the conventional S.I.E. The major air pollutants found in the exhaust of the conventional S.I.E. are, NOx, CO, and HC. Many investigators try to find out a new combustion technique that minimizes exhaust emissions and gives a good fuel economy. This can be done if the new combustion technique achieves the following conditions.

1-Higher compression ratio without detonation.
2-Higher combustion efficiency.
3-Extended lean misfire limit.
4-Low combustion temperature so that, NOx formation may not increase and maintaining low CO and HC.

It is known that operating the engine at higher compression ratio, will increase the tendency to detonate. We are confronted with the problem that, operating the engine at lean mixture will decrease combustion rate due to relatively low temperature and increased cycle variations. Engines operating at high compression ratios with lean mixtures are inherently likely to lead to increased NOx emissions associated with the higher oxygen available and higher mean temperature levels[1].

To overcome the difficulties associated with lean burning in S.I.E., it is necessary to increase combustion rates and improve ignition performance. This can be done by improving and enhancing the degree of turbulence within the combustion chamber at the combustion time[2]. It is necessary to strengthes the mixture near the ignition point using the technique of torch chamber engine.

Combustion characteristics and engine performance of torch chamber S.I.E. was studied by Abas[3]. He investigated the effects of connecting passage shape and spark gap projection on engine performance and combustion characteristics. He found that using the convergent divergent passage coupled with the extended spark gap projection will improve engine performance. He found also that the lean misfire limit (LML) of stable operation of the engine is dependent on the connecting passage shape and less sensitive to the location of the ignition point.

Flame photographs have been recorded by Sinnamon and Cole[4] during their investigation of divided chamber jet ignition engine. The photographs have revealed that, the torch jet which ruches from the prechamber causes a large increase in turbulence intensity and in the rate of mixing in the main chamber. The analysis indicates that, this torch jet induced turbulence is responsible for a large increase in the initial mass rate of burning and, as a result, shortening of burn interval.

Many investigations have been done by Adams[5,6] on the torch chamber S.I.E. to study the generating of combustion turbulence, optimum torch chamber volume and orifice diameter. He also computed the combustion wave thickness, turbulence characteristics time and the effect of nozzle orientation on combustion rate, engine performance and engine emissions. A prechamber, named the turbulence generated pot (TGP), was developed by Noguchi et al[7] to extend the effective LML and to increase flame propagation velocity. They found that these parameters are depending on the configuration of the TGP and also on the spark plug location. They also proved that when the jet flame from the TGP creates a strong turbulence in the main chamber, the result will be a high flame speed and a reduction in NOx emissions.

The present work aims to study the torch chamber as a means of controlling engine exhaust emissions. This work investigates the effect of flame initiation point, connecting passage shape and other engine operating parameters that affect engine exhaust emissions. The experimental work is directed to clarify the effect of the above mentioned parameters on the engine emissions. A mathematical model has also been developed to study the effect of equivalence ratio on engine emissione. A comparison between calculated and measured NOx and CO trends are made and discussed. The results presented in this paper and more detaileld of the experimental procedure are from reference[8].

2-EXPERIMENTAL SETUP AND PROCEDURES

The experiments were carried out at the Army Research Center. Fig.(1) shows a schematic diagram for the experimental setup. It was built to verify the possibilities of variation of connecting passage shape, spark gap projection and engine operating conditions. The setup also verified accurate measurements of engine data and exhaust emissions. The engine used is a modified Deutz-Diesel engine. It is a four stroke, four cylinder, overhead valves, air cooled with a 11 cm. bore, 14 cm. stroke and with a swirl chamber in the cylinder head. It is modified to be a torch chamber S.I.E. with compression ratio of 9:1.

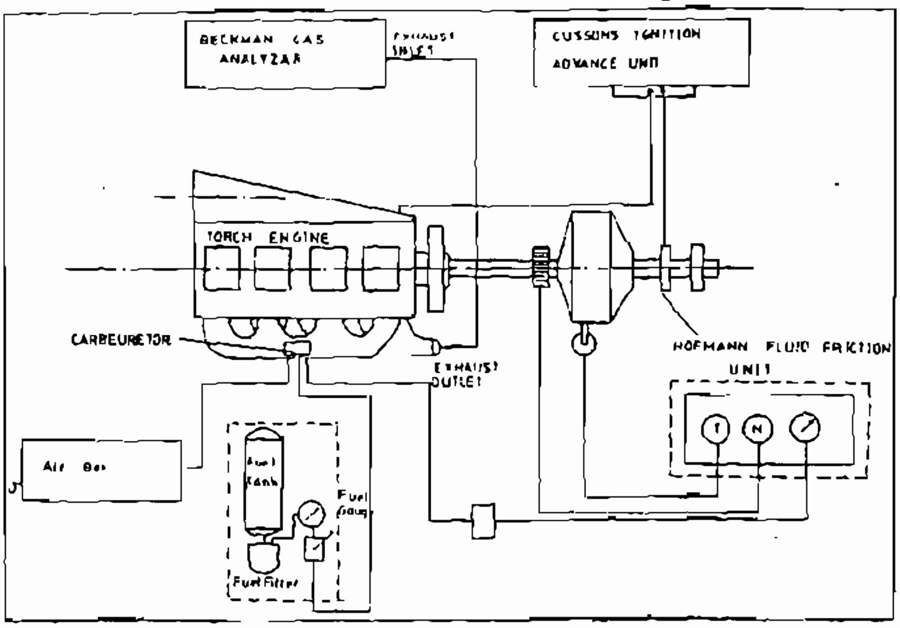


Fig. (1) Schematic Diagram Of The Experimental Set-up

The intake manifold was modified to achieve uniform mixture distribution between cylinders. So a new one was provided to the engine having a central position for the carburetor and it was adopted to be fitted with an electric heating for the charge and controlling the charge temperature to be 30 oC±2. A conventional carburetor was selected according to the engine air flow requirement at extreme condition. An adjustable needle valve was fitted to the original main-jet nozzle to control manually the mixture strength.

The engine was modified to be a S.I.E. A conventional spark ignition system was used. The distributor used is of contact breaker type and provided with a centrifugal and vacuum spark advance which assists rapid setting of optimum ignition timing. Three spark plugs with different electrode lengths were used. The electrode lengths used were 3.5, 8 and 13 mm. The third one was developed for this work.

The modified cylinder head is shown in Fig.(2).It consists of unscavenged spherical shaped swirl chamber casted in the cylinder head with 28.22 cc volume which represents 34.7% of the clearance volume (after modification). It is connected to the main chamber with a cylindrical straight passage of9mm.diameter, 20 mm. length and 30o inclination to the piston crown. It is fitted with a spark plug in the original location of the heating plug. Two different connecting passage shapes were used, the original cylindrical one and a convergent divergent nozzle with a 9 mm. throat and 19o divergent angle. Fig.(3) shows the configurations of the two connecting passage used.

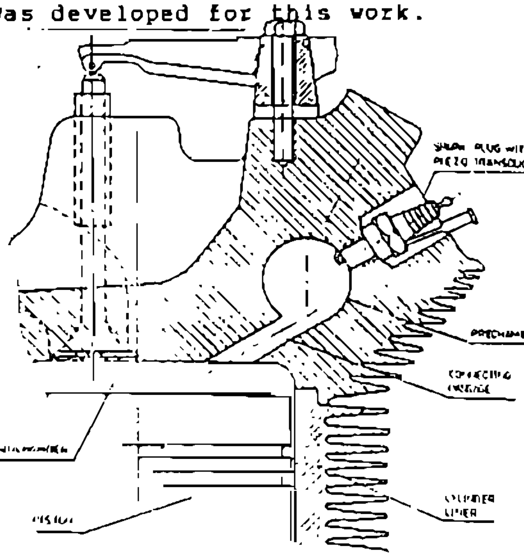


Fig. (2) Modified Engine Cylinder Head

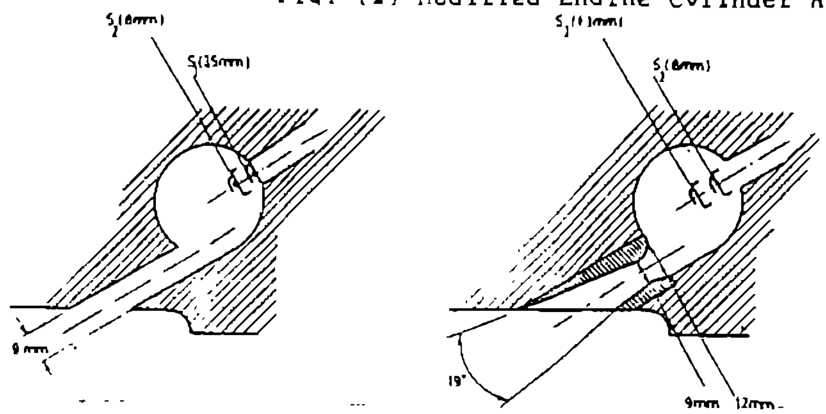


Fig. (3) Prechamber Combination To Be Tested

Combination A . Straight connection passage with spark gap location. S_1
Combination B . Straight connection passage with spark gap location S_2
Combination C . Convergent divergent passage with spark gap location S_2
Combination D . Convergent divergent passage with spark gap location S_3

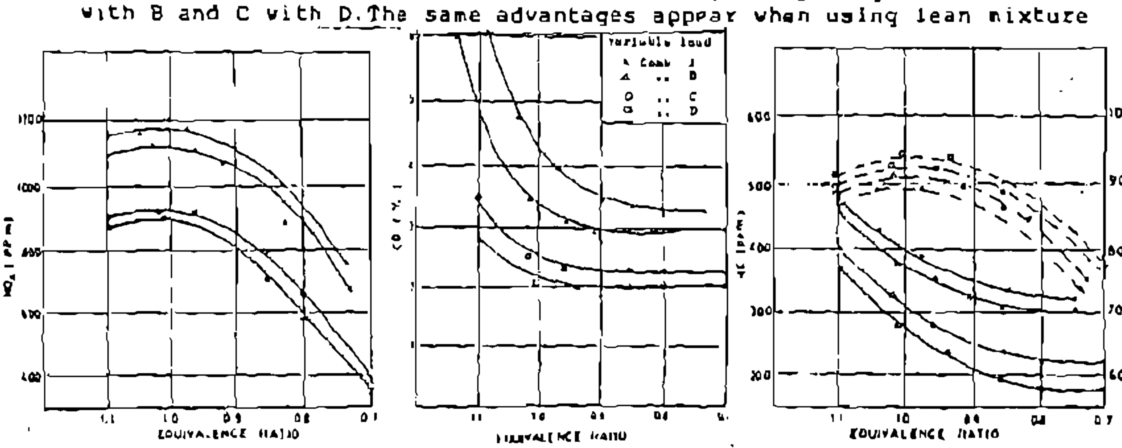
Engine speed was monitored directly from a magnetic pick-up in connection with a 60 tooth gear wheel fixed on the dynamometer input shaft. The engine torque was measured by a fluid friction dynamometer having maximum capacity of 715 Kp.m and maximum speed of 6000 rpm. The percentage of throttle opening was displayed by means of analogue indicator. The exhaust gas temperature was measured using a chrom-alumel thermocouple fixed on the exhaust manifold, attached to analogue indicator. Engine exhaust emissions were measured using Beckman gas analyzers. The system is suitable for analyzing hot exhaust gases. The gases which can be analyzed are CO, CO_2, NOx, HC, and O_2. The HC emission is analyzed at 180 oC (dew point =170oC).

To cover the experimental investigations, the mixture strength were changed from rich (ϕ=1.1) to lean misfire limit. Engine speed was changed from 1000 to 2200 rpm. The tests were classified into two series. In the first series of tests, straight passage was used coupled with the cylinder head arrangements A and B, Fig.(3). While in the second series of tests the convergent divergent nozzle was used coupled with the arrangements C and D, Fig.(3).

3.EXPERIMENTAL RESULTS AND DISCUSSIONS

The results of the tests are collected and displayed graphically as a four families of curves. For each combination A,B,C&D, (Fig.(3), engine emissions of NOx, CO and HC are meaaured and recorded for different operating conditions. Shown in these families of curves are the effect of equivalence ratio, bmep, engine speed and throttle opening on engine emissions. The test engine was adopted for optimum ignition advance all-over the tests.

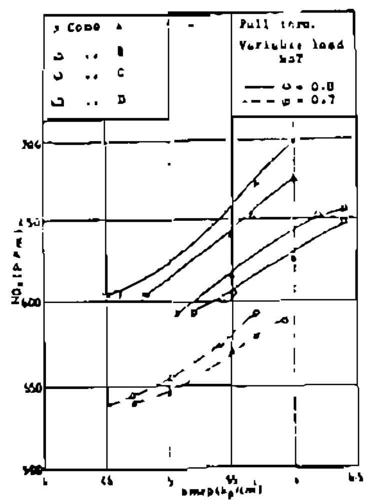
Shown in Figs.(4)through(6) are comparative curves of engine emissions of NOx, CO and HC as a function of equivalence ratio at full throttle, variable load and engine speed of 1000 rpm. It is clear from these figures that the increase of spark gap projection tends to decrease engine emissions of NOx, CO and HC, comparing combination A with B and C with D.The same advantages appear when using lean mixture

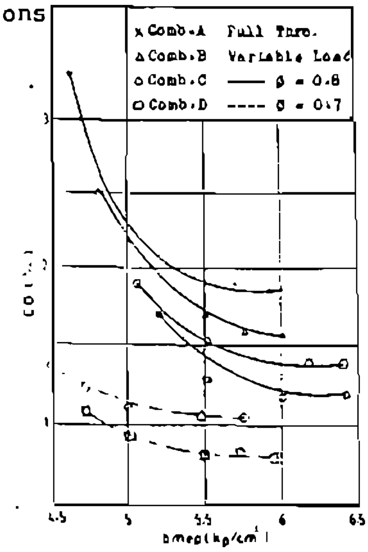


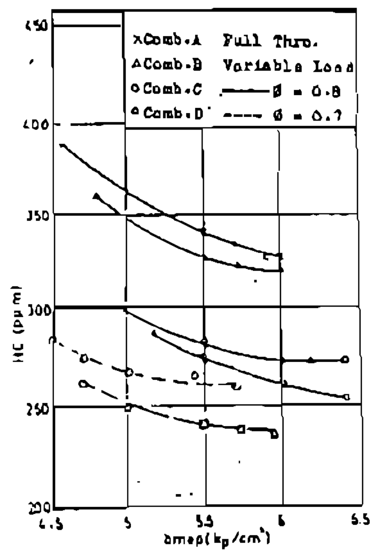
Figs. (4)through(6) Engine Emissions As a Function Of Equivalence Ratio At Engine Speed 1000 rpm.

The convergent divergent passage has the same advantages, comparing combination B with D. Combination D satisfies the lower engine emissions all-over equivalence ratios examined. The relative gains obtained when using combination D instead of A has been noticed to increase as the mixture becomes leaner. It is also clear from the results that using convergent divergent passage makes it possible to run the engine at relatively lean mixture, ϕ=.7 with less exhaust emissions.

Figs.(7)through(9) show the effects of bmep on engine emissions when using lean mixtures,ϕ=.8 and 0.7. It is clear from these figures that as bmep increases NOx increases while CO and HC decreases. It can be also seen that further extension of spark gap projection, keeping the bmep constant, lead to decrease engine emissions. The use of convergent divergent passage has the same advantages. Also use of combination D gives the lower values of engine emissions for all values of bmep tested. The result also show that running the engine with convergent divergent passage at leaner mixture ϕ= 7 lead to a further decrease in engine emissions



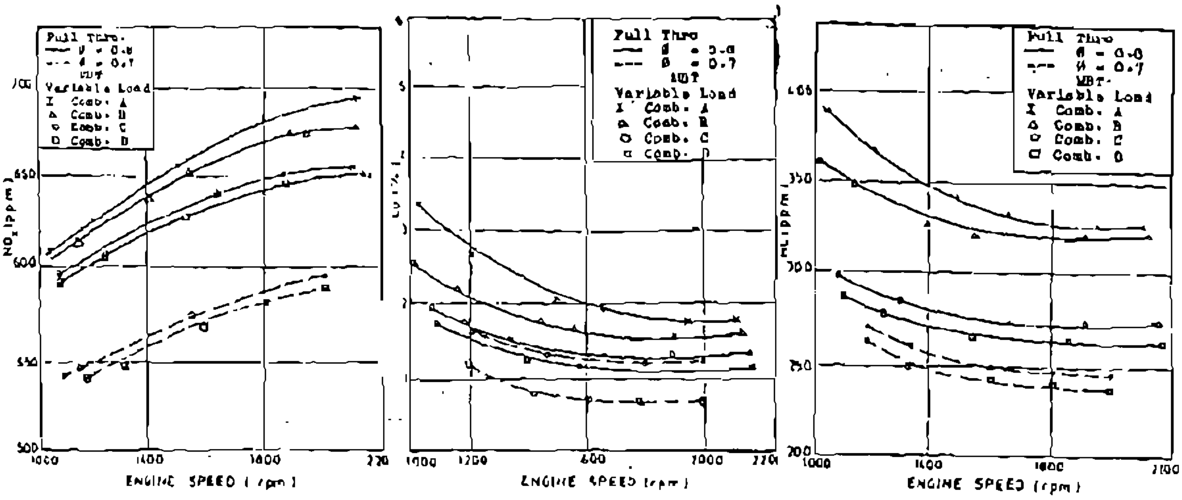




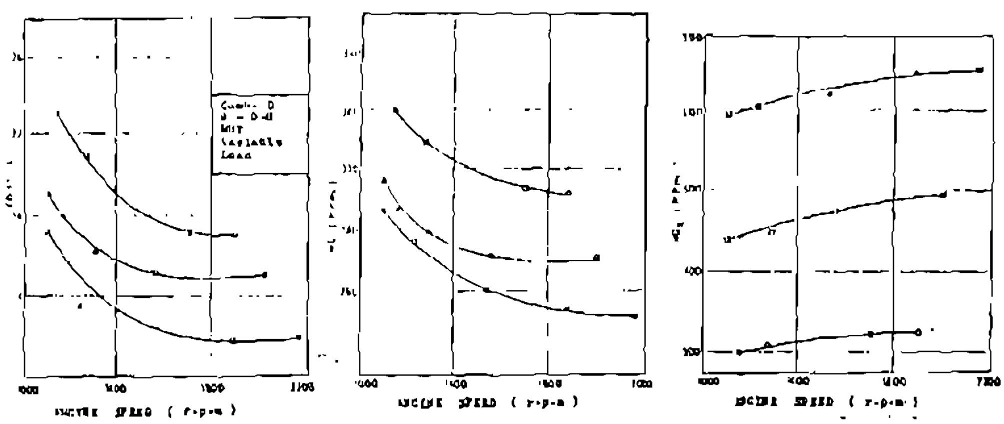
Figs. (7)through(9) Engine Emissions As a Function Of bmep

The influence of engine speed on exhaust emissions is shown in Figs.(10)through(12). These relations are recorded at full throttle, variable load and at equivalence ratios of 0.8 and 0.7. It can be seen from these figures that as engine speed increases NOx emission increases while CO and HC decrease. It is also noticed that further expanding of the spark gap projection inside the torch chamber, keeping engine speed constant, will decrease engine emissions. The use of convergent divergent passage gives less engine emissions. The relative gain obtained when using combination D instead of A increases as engine speed increases.

Figs.(13)through(15) show the influence of throttle opening on engine emissions. Combination O is used with full,3/4,and 1/2 throttle



Figs. (10)through(12) Engine Emissions As a Function Of Engine Speed At Full Throttle Opening



Figs. (13)through(15) Engine Emissions As A Function Of Throttle Opening

opening and 0.8 equivalence ratio with variable engine speed and load. Fig.(13) show that as the throttle opening is decreased, NOx emission decreases, while Figs(14)and(15) show that as the throttle opening is decreased the HC and CO emissions increase.

The results described above demonstrate that the engine exhaust emissions decrease as the spark gap projection extends inside the torch chamber. This may be explained as follows; the spark gap being extended in the torch chamber, the amount of unburned mixture escaping from the torch chamber into the main chamber during the early stage of combustion are reduced. Using the expanded spark plug will initiate the flame kernel approximately in central position, thus a spherical flame development may be obtained. This may decrease the combustion duration and consequently decrease the unburned mixture escaping via the connecting passage. Moreover, moving the ignition point towards the center of the combustion chamber, the residual gas around the spark gap are well scavenged, this assists good ignitability of the mixture and extends the LML. This may be due to the expected rise in temperature of the center electrode tip. This is induced by the elongated heat rejection path. Thus, it may be assumed that the LML was extended due to the decrease in the quenching effect on the flame kernel.

The effect of the geometrical shape of the connecting passage is clarified comparing the results obtained when using combination B and C. These results conclude that the use of convergent divergent passage instead of the straight one improves the engine emissions. This may be due to improved turbulent motion generated by the flow through the nozzle. This result agrees with the result obtained by Shapiro[9]. Abas[3] in his experiments on the same engine showed that the use of the convergent divergent nozzle extends the LML, enhance turbulent generation and achieve faster burning rate. A survey to these results reveals that, the use of convergent divergent passage coupled with the extended spark gap projection will decrease combustion duration, decrease combustion temperature and enhance turbulent generation. These will decrease engine exhaust emissions of NOx, CO and HC.

The effects of operating parameters examined may be summarized as follows, it is possible to run the torch chamber S.I.E. at relatively lean mixture with gains in fuel economy and engine emissions. This advantage may be due to,the increased turbulence created with the torch chamber to compensate for the lower burning velocity of the lean mixture. The combustion of lean mixtures will result in low combustion temperature, which may not support NOx formation while maintaining lower CO and HC emissions. The influence of engine speed on emissions can be explained as follows; As the engine speed increases, the mixture turbulence is increased and this will increase the burning rate and good mixing of the fuel and air. Consequently a reduction in HC and CO emissions may results. Also as the engine speed is increases, the peak cylinder pressure increases with a corresponding increase in peak temperature. This high temperature coupled with its duration are conductive to NOx formation. The influence of throttle opening is that, as the throttle opening is decreased NOx decreases while HC and CO increase. This may be due to the decrease in bmep and combustion temperature as the throttle opening is decreased. Also the turbulence motion and the combustion rate decreases. These will tend to decrease NOx and increase CO while give a chance to quench HC.

4-THEORETICAL ANALYSIS

The primary purpose of the present work is to develop a simple combustion model for predicting the pressure and temperature vs crankangle diagrams and the NO and CO emissions rate of formation. The analysis is made by means of the pressure vs crankangle diagram, experimental rate of combustion and optimum ignition timing which are measured on the same engine as an input data to the model. In this study the effect of equivalence ratio on NO and CO rates of formation are examined. The basic kinetic, energy and chemical equilibrium equations are taken into account.

The torch chamber S.I.E. is treated as a two thermodynamic subsystems (pre- and main chambers systems) coupled by mutual mass flow. The working fluid prior to ignition is considered to be a non reacting homogeneous mixture of fuel vapor plus air. The compression process is assumed to be adiabatic. Initially the mixture is assumed to be at atmospheric pressure and temperature and a volume of 1496.46 cc. (the cylinder volume). The compression begins at BDC and is treated in an incremental manner, a 2 degree crankangle increment is adopted. Before the end of this process, the combustion is assumed to start at a specified MBT spark timing. The function of this process is to determine the initial conditions in both systems before combustion.

The combustion process starts when the spark plug is fired at MBT spark timing. The process is treated in an incremental manner, a 2 degree crankangle increment being adopted. Each successive increment in crankangle comprises four steps, as shown in Fig.(16). These four steps are;

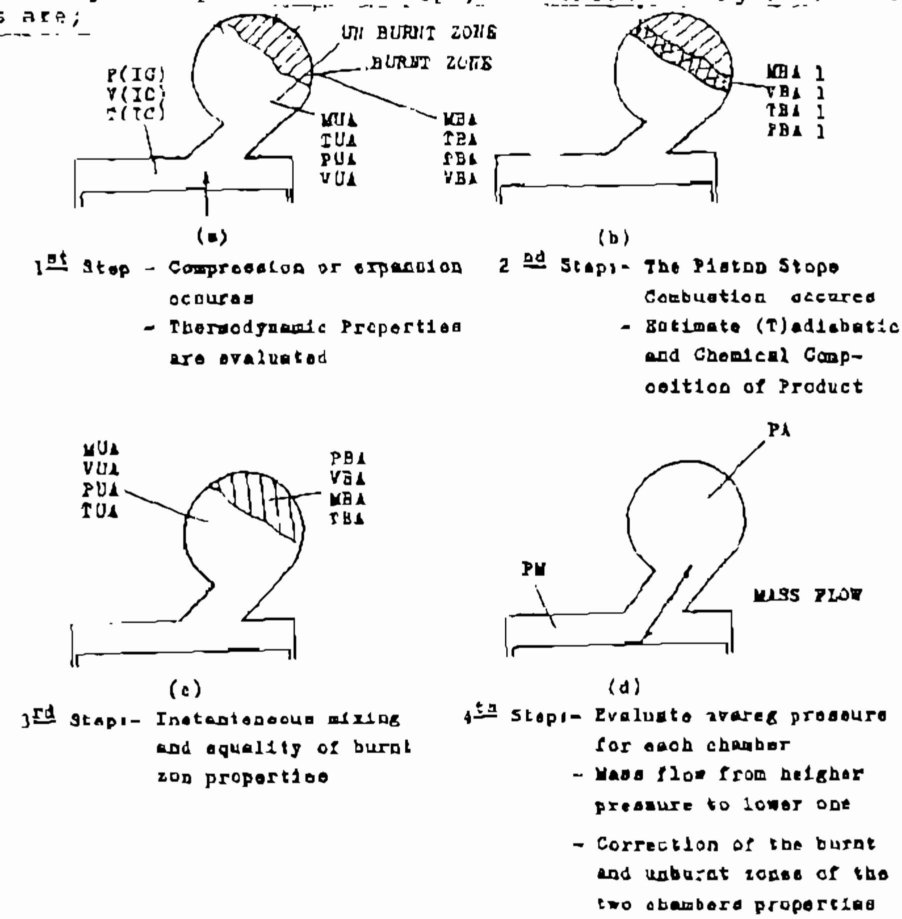


Fig. (16) Analysis Of Combustion Process where;
MUA, TUA, PUA and VUA are mass, temperature, pressure and volume of unburnt gas in the prechamber, MBA, TBA, PBA and VBA are mass temperature, pressure and volume of burnt gas,

1-Compression or expansion occurs as the piston moves up or down-ward. Thermodynamic properties of the gases in both systems are evaluated.
2-At the spark timing, combustion occurs in a specified fraction of the mixture. The adiabatic flame temperature and the chemical equilibrium of the product species of CO, CO_2, O_2, H_2, H_2O, OH, H, O, N and N_2 are calculated.
3-Instantaneous mixing of the total burnt zone is allowed and average pressure and temperature are evaluated.
4-An aqui-steady mass flow through the connecting passage as a result of the pressure difference between the two systems is allowed, then a correction is made for the thermodynamic properties.

The new adiabatic flame temperature of the burnt zone is calculated and the rate of NO and CO formation is estimated. This process ends when all the mass of the reactant are burned early in the expansion process.

The expansion process is assumed to be adiabatic and is treated in an incremental manner, a 2 degree crankangle increment is again adopted. The process starts at TDC and ends at the time of exhaust valve opening. During this process and after the combustion ends, each crankangle increment comprises two steps. These two steps are, expansion due to piston motion, followed by mass flow through the torch opening. The rate of NO and CO formations are predicted taking into account the flux transfer through the torch opening. The combustion gases are assumed to be in chemical equilibrium at the appropriate pressure and temperature.

The kinetic mechanism proposed for NO calculation is the extended Zeldovish mechanism which comprises the following reactions[10];

$$N_2 + O \rightleftharpoons NO + N \quad K1=1.4*10^{14}\exp(-37900/RT) \text{ cc/mol.sec} \tag{1}$$

$$O_2 + N \rightleftharpoons NO + O \quad K2=6.4*10^{7}\exp(-3140/RT) \text{ cc/mol.sec} \tag{2}$$

$$N + OH \rightleftharpoons NO + H \quad K3=4*10^{13} \text{ cc/mol.sec} \tag{3}$$

The rate of NO formation is predicted using the following relation taking into account the flux due to mass flow through the torch opening;

$$\frac{d(NO)}{dt} = 2*\frac{MNO}{\rho} * \frac{(1-B^{2})*R1}{(1-K*B)} + \left.\frac{d(NO)}{dt}\right|_t \tag{4}$$

Where; MNO is the molecular weight of NO, ρ is the gas density=PV/RT, B=NO/NOe, K=R1/(R2+R3), [d(NO)/dt]t flux of NO through torch opening, R1=K1[O]e[N2]e, R2=K2[NO]e[O]e and R3=K3[NO]e[H]e

Using the temperature, pressure and equilibrium values of O, H_2, O_2, OH and H at the end of each crankangle increment, kinetic NO formation is calculated by integrating of equation(4). The initial value for NO at spark timing is assumed to be zero.

To calculate the kinetic CO formation, it is assumed that at spark timing all the carbon in fuel is converted to CO which is subsequently oxidized to CO_2 during the combustion process via the following reaction[11];

$$CO + OH \rightleftharpoons CO2 + H \quad K4=3*10**5*exp(-600/RT)\ cc/mol.sec \quad (5)$$

and the CO formation rate is calculated by integrating of the following relation;

$$\frac{d(CO)}{dt} = K[OH][H] + \left.\frac{d(CO)}{dt}\right|_t \quad (6)$$

The product temperature, pressure and the concentration of the species are assumed to remain as predicted from the full equilibrium model. Thus by using the carbon conservation equation;

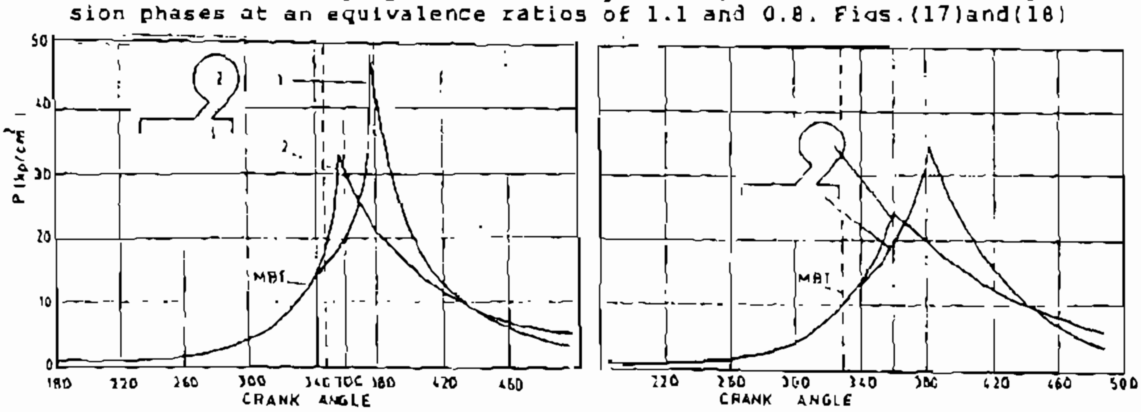
$$(CO) + (CO2) = (CO)e + (CO2)e \quad (7)$$

The CO levels through out the combustion and expansion processes are computed.

5-THEORETICAL RESULTS AND DISCUSSIONS

The computed results include, NO and CO rates of formation, pressure, temperature and mass transfer history through the torch opening for both pre and main chambers. The program was executed for four values of equivalence ratio namely 1.1, 1.0, 0.9 and 0.8 and an engine speed of 1500 rpm. The results were collected and displayed as two sets of curves. In the first set histories of pressure, temperature, mass flow through torch opening and NO & CO vs. crankangle are presented. The second set of curves are a comparison between the computed and measured values of NO & CO emissions.

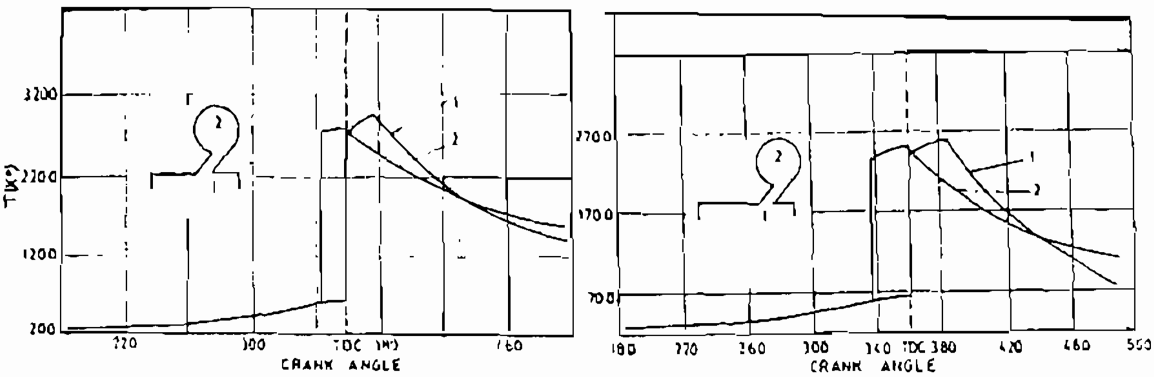
Shown in Figs.(17)through(26) are computational curves of pressure, temperature, mass flow and NO & CO emissions vs. crankangle. The results are displayed for the compression, combustion and expansion phases at an equivalence ratios of 1.1 and 0.8. Figs.(17)and(18)



Figs(17)and(18)Pressure History of Auxiliary and Main Chambers at Equivalence Ratios of 1.1 and .8 Respectively.

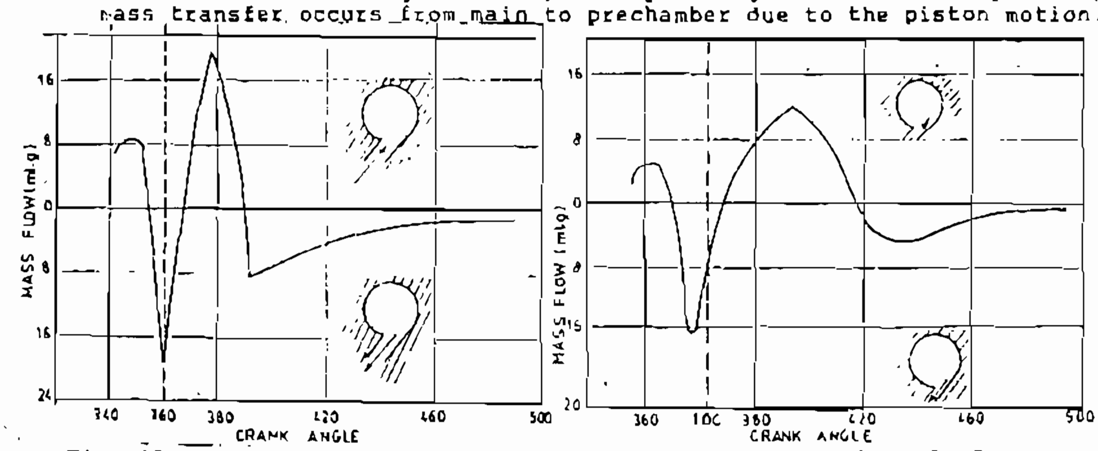
show the pressure histories for both chambers. It is clear from these figures that, as the mixture becomes leaner, the peak pressure in both chambers decreases, and the peak pressure in the main chamber is greater than that of the prechamber. It is also noticed that the rate of pressure decrease during the expansion process is faster in the main chamber than that in the prechamber.

Figs.(19)and(20) show the temperature vs. crankangle diagrams for both chambers at equivalence ratios of 1.1 and 0.8. It can be seen from these figures that, the temperature levels are always higher in the main chamber than that in the torch chamber. Also, the maximum gas temperature through the cycle is reached in the main chamber. The results conclude also that, the rate of temperature decrease in the main chamber is more rapid than that in the prechamber.



Figs.(19)and(20)Temperature History for Auxiliary and Main Chambers at Equivalence Ratios of 1.1 and .8 Respectively.

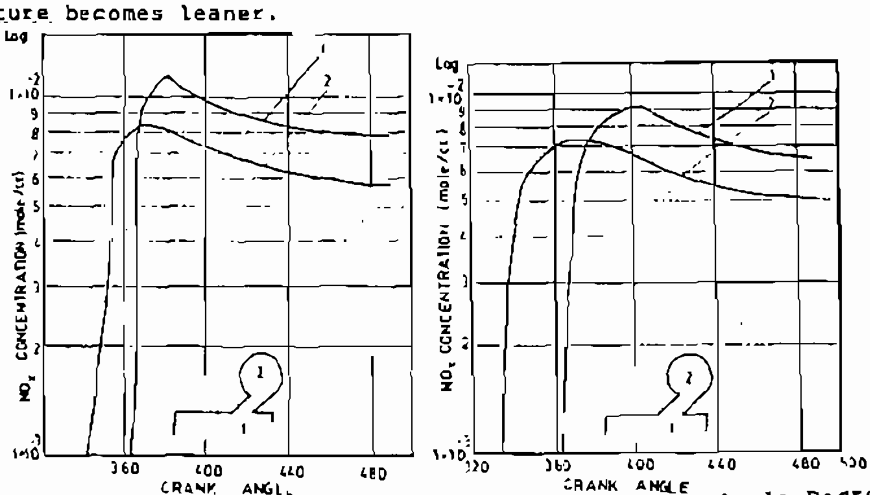
A history of the mass transfer through the torch opening at equivalence ratios of 1.1 and 0.8 is shown in Figs.(21)and(22). It is clear from these figures that, early during the combustion process, mass transfer occurs from main to prechamber due to the piston motion.



Figs.(21)and(22)Mass Flow Through Torch Opening vs Crank Angle Degrees at Equivalence Ratios of 1.1 and .8 Respectively.

As the pressure in the prechamber rises due to combustion and exceeds the pressure in the main chamber a reverse flow through the torch opening occurs. As the combustion in the main chamber starts, the rate of mass transfer from pre to main chamber decreases. As the pressure in the main chamber exceeds the pressure in the prechamber, mass transfer from main to prechamber. During expansion process, as the pressure in the main chamber decreases due to piston motion, the rate of mass transfer decreases.

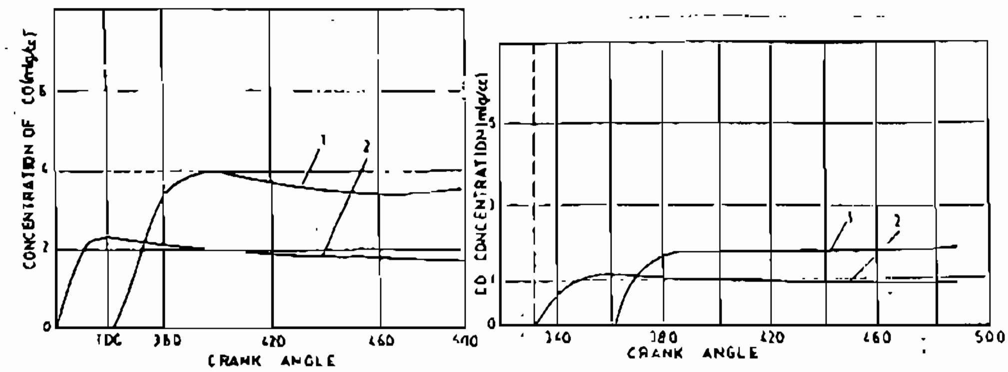
Figs.(23)and(24) illustrates the NO rate of formation in both chambers at equivalence ratios of 1.1 and 0.8. It is clear that in early combustion the NO rate of formation is greater in the main chamber than that in the prechamber. It is also noticed that the formation of NO is freezes later during the expansion process at a temperatures of 1950K and 1500K at equivalence ratios of 1.1 and 0.8 respectively. It can also be seen that, at the end of the expansion process, NO slightly decreases in the prechamber and increases in the main chamber. The results concluded also that the NO levels decreases as the mixture becomes leaner.



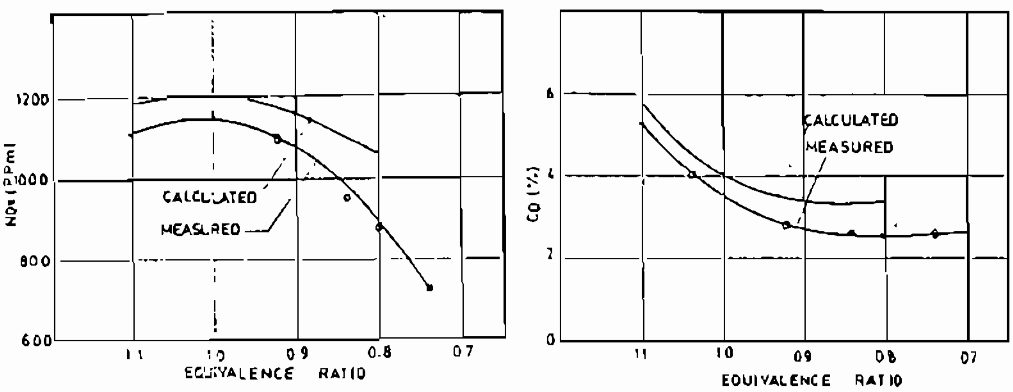
Figs.(23)and(24)Kinetic NO Rate of Formation vs Crank Angle Degrees at Equivalence Ratios of 1.1 and .8 Respectively.

Figs.(25)and(26) illustrates the history of CO formation. It is clear from these figures that the rate of CO formation in the main chamber is greater than that in the prechamber. It is also noticed that the CO formed in the case of rich mixture is greater than that of the lean mixture.

In Figs.(27)and(28) a comparison between the measured and calculated values of NO and CO emissions are made. From these curves it can be concluded that, the calculated value of NO is higher than the measured one. It is also noticed that the difference between the calculated and measured values increases at lean mixture. This may be due to neglecting the temperature gradient through the burned gases. The same trends are observed for the calculated and measured values of CO.



Figs.(25)and(26)Kinetic CO Rate of Formation vs Crank Angle Degrees at Equivalence Ratios of 1.1 and .8 Respectively.



Figs.(27)and(28)Comparison of Computed and Measured NO and CO Respectively vs Equivalence Ratio.

The theoretical results show that, the greatest part of the emission is generated in the main chamber. This may be due to the higher temperature levels in the main chamber. Also, the frees of NO occurs during the expansion process where the temperature is low. This may be due to the small effect of temperature in calculating reaction constant K1,K2 and K3. The slight increase and decrease in NO and CO in the main and prechamber respectively can be attributed to the flow through the orifice from the prechamber. The emission trends obtained from the model at the four equivalence ratios examined here agree with the measured values.

6-CONCLUSIONS

The experimental results give the following conclusions;

1-The use of convergent divergent passage and extended spark gap projection inside the prechamber decreases the engine emissions allover the range of experiments examined.

2-These advantages are still gained at part load with the cylinder head combination D.

3-The connecting passage shape has a great effect on LML of stable operation, while the LML is less sensitive to ignition point location.

Studying the calculated rates of emissions formation reveal that,

1-The main chamber is responsible for the formation of the greatest part of engine emissions.

2-The equivalence ratio effects the NO and CO emission rate of formation.

NOMENCLATURES

BDC	bottom dead center
bmep	brake mean effective pressure
C	degree centigrate
cc	cubic centimeter
K_1	reaction constant
LML	lean misfire limit
MBT	maximum advance for best torque
rpm	revolution per minute
S.I.E	spark ignition engine
TDC	top dead center
ϕ	equivalence ratio

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