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Analysis on Emission Characteristics of Extended Expansion Lean Burn Spark Ignition Engine

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Abstract- In the development of internal combustion engines, there has been a continuous effort to reduce fuel consumption and exhaust emissions. Improved fuel efficiency with reduced exhaust gas emissions is one of the major challenges that engineers and scientists in the automotive industry and are facing. Also in recent years there has been great concern that, the internal combustion engine is predominantly responsible for atmospheric pollution, which is detrimental to human health and environmental damage. Consequently, research engineers have been striving to reduce the quantity of pollutants emitted from exhaust system without sacrificing power and fuel consumption. Further, the load controls in these engines are performed through throttling, which is mainly responsible for poor part load efficiency. In SI engines, the compression ratio is restricted by the combustion process, but the expansion ratio can be extended. To achieve lean combustion the following modification were done, combustion chamber was modified to enhance and swirl and squish, copper as a catalyst was coated on the cylinder head and piston crown, and high energy transistorized coil ignition (TCI) system was used to ignite lean mixture.

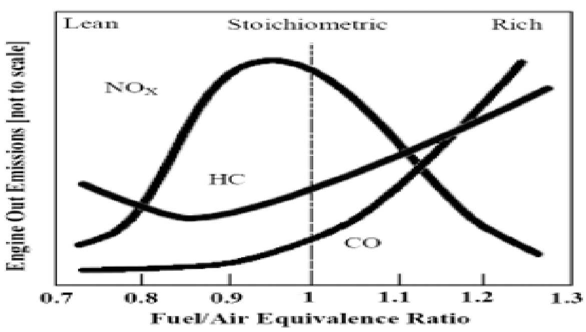
Keywords- TCI, CR, EEE and UBHC etc.

I. INTRODUCTION

The incomplete burning of the air-fuel mixture in the combustion chamber produces pollutants. The major pollutants emitted from the exhaust due to incomplete combustion are, unburnt hydrocarbons (UBHC), oxides of nitrogen (NOX) and highly poisonous carbon monoxide (CO). If however combustion is 100 % complete the only products being expelled from the exhaust would be water vapour, which is harmless, and carbon dioxide, which is inert gas and, as such, it is not directly harmful to humans. Lean burn technology burn the air-fuel mixture completely or almost completely, in an efficient manner, so that fuel consumption is reduced and the level of pollutants are with limits. Hence lean combustion is a preferred concept for reducing exhaust emissions for meeting stringent emission standards.

1. Lean Burn Combustion Engine

In practice use of homogeneous lean burn mixtures pose many problems, such as, lower flame propagation, occurrence of misfire, low mixture distribution quality in multi-cylinder engines and high amount of unburnt HC in the exhaust.



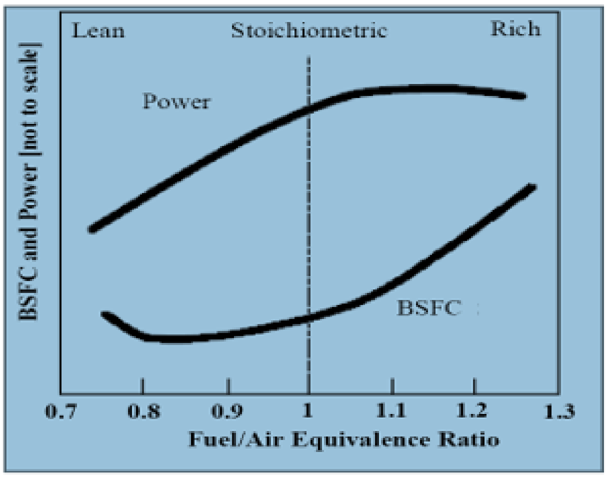


Fig 1 Effect of Air-fuel Ratio on Power, Fuel Consumption, and Emissions.

2. Extended Expansion Engine

Today many automotive manufacturers have focused their research activities on this problem. The major task is either to eliminate the throttle valve or to use it as open as possible regardless of the load conditions. Practical methods to increase the efficiency at part load are given below as subtitles in Figure 2 (Osman Akin Kutlar et al 2005).

The cycle conceived by the British engineer James Atkinson offers a solution to this two-fold problem in which the load is controlled without pumping losses, and that the compression ratio is differentiated from the expansion ratio. Based on this principle, for each load level, the combustion chamber volume is designed to maintain the compression ratio at the maximum value below the knock limit, and to obtain the most effective expansion ratio.

In a conventional Spark Ignition (SI) engine, the Compression Ratio (CR) is equal to the Expansion Ratio (ER). Further, the load controls in these engines are performed through throttling, which is mainly responsible for poor part load efficiency. In these engines, to increase the cycle efficiency, one has to increase either the compression ratio or expansion ratio or both. In SI engines, the compression ratio is restricted by the combustion process, but the expansion ratio can be extended. The engine with higher expansion ratio than compression ratio is referred to as Extended Expansion Engine (EEE).

This simulation is limited to only in-cylinder processes. For simplicity, the combustion chamber was assumed to be cylindrical shape. In this work, all cylinders of a multiple-cylinder engine are assumed to be identical, and assumed to follow the same thermodynamic process, and to operate with identical conditions. Overall results for a multiple cylinder engine are obtained by multiplying the results from the single-cylinder analysis by the number of cylinders.

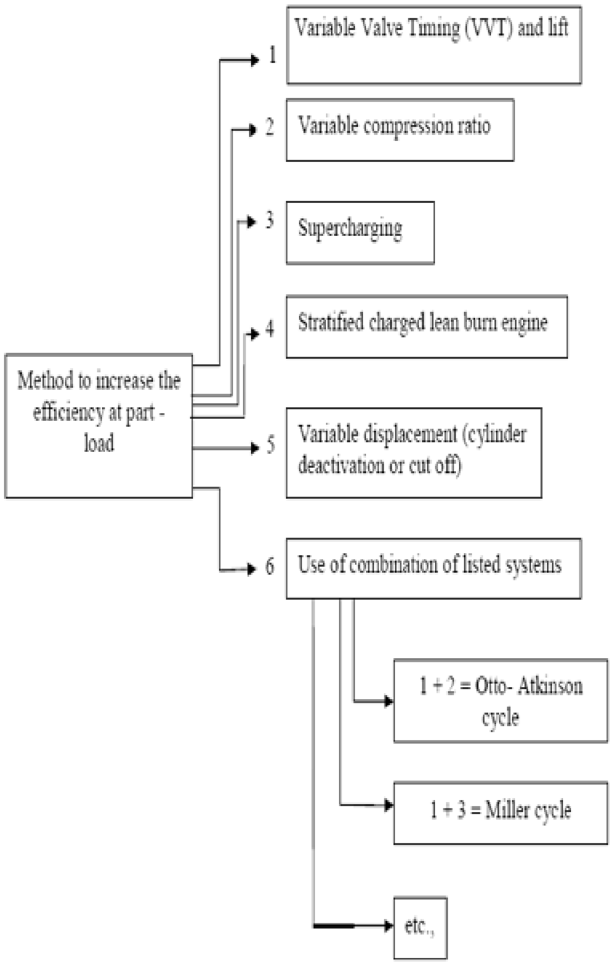


Fig 2 Practical Methods to Increase the Efficiency of SI Engine at Part Load (Osman Akin Kutlar et al 2005).

II. LITERATURE REVIEW

Kodah et al (2000) describes a simple analysis for the prediction of pressure within a spark ignition engine. This is done by modeling the combustion process using the Wiebe function approach, which is an exponential function in the form

$$y = 1 - e^{-ax^{m}}$$

to calculate the rate of fuel burned. By careful selection of a and m, any spark ignition engine with any combustion chamber shape and any specified dimensions can be assessed by this model. Validity of this model has been tested by comparing the model results with those obtained from running the engine under the same operating conditions.

Table 1 Summary of Lean Burn Engine Literature.

Author and Year	Modifications done to achieve lean burn	Remarks
Increasing swirl, squish and turbulence		
Souich Matsushita et al (1985)	Helical port with swirl control valve and programmed sequential fuel injection and feed back control of air fuel ratio.	20% fuel economy, also gives high performance and reduces emissions.
Chau et al (1988)	Four – valve head with a centralized spark plug, a two-valve head with a helical swirl intake port and two-valve head with straight intake port.	Best mixing was found in the four – valve head, which requires the least spark advance and was able to burn the leanest mixture.
Yuhiko Kiyota et al (1992)	Barrel-stratification.	Extremely lean conditions (maximum of A/F ratio 32), indicated thermal efficiency improved by 15% and reduction in NO_x emission achieved.

To achieve better output and fuel economy VVT, Mitsubishi Innovative Valve Time and Electronic Control (MIVEC) was developed. There is three-mode changeover

- Deactivate both intake and exhaust valves
- Select low-speed cam with moderate lifts and short durations
- Select high-speed cam with high lifts and long durations

III. THEORETICAL ANALYSIS

1. Modeling of Extended Expansion Lean Burn

The following assumptions are made in the development of model to suit the practical conditions.

- The charge in the cylinder at any instant consists of fuel-air mixture and residual gases.
- Ideal gas equation is valid
- At any instant during combustion, the cylinder volume consists of burnt and unburnt volumes separated by a thin flame front.
- The pressure in the unburnt and burnt zones are assumed to be uniform at a given crank angle.

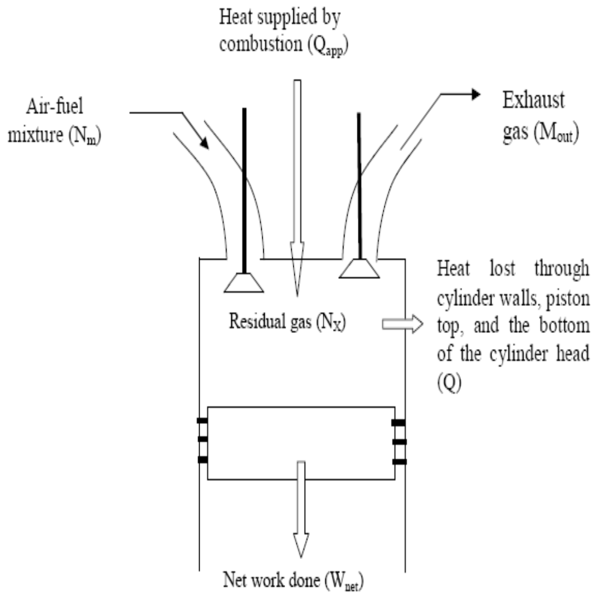


Fig 3 Schematic Diagram of Thermodynamic Modeling Process.

2. Compression Process

During the first iteration, pressure and temperature at the start compression was assumed and the residual gas fraction (NX) is assumed to be zero. At the end first iteration, the parameters are calculated and the iteration is repeated until the intake temperature and pressure converges. The fuel vapour and air in the engine at the beginning of compression stroke is given by

$$N_m = \frac{p_1 v_{ivc}}{RT_1}$$

where p1 and T1 pressure and temperature of charge during start of compression in bar and K respectively, R is universal gas constant of charge in kJ/ kmole K and V_{ivc} volume of charge at inlet valve closing timing in m3.

3. Combustion Process

The total heat input is given by

$Q_{tot} = Q_{app} + Q$

Qapp is apparent heat release from combustion in kW and Q is the convective heat transfer to the cylinder in kW

4. Friction Calculations

Empirical relation (Ganesan 1999) have been used to calculate the power absorbed in friction and the various expressions are as given below.

Mean effective pressure (MEP) lost to overcome friction due to gas pressure behind rings.

$$F_{mep1} = 0.42*(p_a - p_m)*\frac{S}{B^2}*(0.0888r + 0.182r^{1.33-0.394C_m/100})*10$$

5. Equilibrium Calculation of Species

Twelve species were considered in the calculations of combustion product concentrations (Benson and Whitehouse 1979; Olikara and Borman.

$$ae\,(C_nH_m) + \frac{ae}{\phi}(O_2 + \frac{78}{21}N_2 + \frac{1}{21}Ar) \longrightarrow X_{CO}\,CO + X_{CO2}\,CO_2 + X_{H20}\,H_2O$$

$$+ X_{H2}\,H_2 + X_{OH}\,OH + X_H\,H + X_{N2}N_2 + X_{NO}\,NO + X_N\,N + X_{O2}\,O_2 + X_0\,O + X_{Ar}\,Ar$$

IV. DESCRIPTION OF COMPUTER PROGRAM

1. Functions

This function calculates the mean molar specific heat of the gases at

$$C_V\,(T) = (B - R) + \frac{C}{T} \qquad kJ\ kmol^{-1}K^{-1}$$

2. Subroutines

This subroutine calculates the volume of the cylinder for each crank angle using the following relation

$$V(\theta) = V_{disp}\left[\frac{r}{(r-1)} - \frac{(1-\cos\theta)}{2} + \frac{L}{S} - \frac{1}{2}\sqrt{\left(\frac{2L}{S}\right)^2 - \sin^2\theta}\right]$$

This subroutine calculates the combustion product concentrations for one particular state (Pi, Ti). The concentration of twelve species were calculated for given pressure and temperature. Here estimated concentration of twelve species checked for balance if they do not balance with a stipulated accuracy a Newton-Raphson adjustment is made and calculations repeated. For peak pressure and temperature the equilibrium value of CO emission stored in emission data file. This subroutine calculates UBHC concentrations using the following empirical relation.

V. EXPERIMENTAL IVESTIGATIONS

1. Experimental set-up

Developed to conduct tests on a four-stroke, single cylinder extended expansion lean burn engine operating on gasoline. Necessary instrumentations were provided to evaluate the performance and emissions of engine at different operating conditions. This chapter discusses modifications done to operate the engine as extended expansion lean burn engine, experimental setup and instruments used for the work.

2. Test Engine

The main reason for choosing this engine is that, it could operate safely at compression ratios, which normally SI engine cannot withstand. A provision was made to mount a piezoelectric pressure transducer flush with 82 the cylinder head surface to measure the cylinder pressure. Also piezoelectric pressure sensors are mounted in the intake and exhaust to measure manifold pressure.

Table 2 Specification of the Engine.

Type	Four stroke, water-cooled CI engine modified to run in SI mode
Make	KIRLOSKAR
Number of Cylinder	One
Bore X Stroke	80mm X 110 mm
Displacement volume	552.92cc
Compression ratio	16.5
Connecting rod length	230 mm
Rated Power (original diesel engine)	5BHP@1500rpm
Valve Timing	
Inlet Valve Opening (IVO)	13°bTDC
Inlet Valve Closing (IVC)	30°aBDC
Exhaust Valve Opening (EVO)	20°bBDC
Exhaust Valve Closing (EVC)	14°aTDC

3. Modification for Changing Compression Ratio (CR)

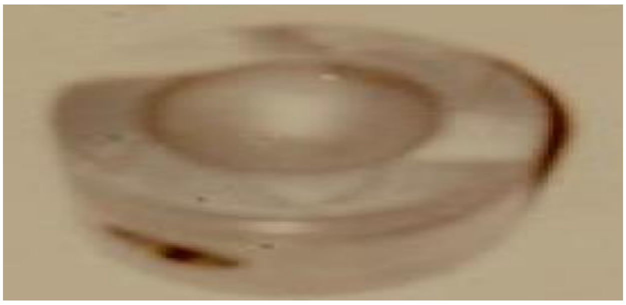


Fig. 4 Enhanced Swirl Combustion Chamber.

The principle of extended expansion is based on modified-Atkinson Cycle which has a larger Expansion Ratio (ER) than Compression Ratio (CR), unlike a conventional Otto-cycle, where CR is equal to ER. In Atkinson Cycle expansion process is extended until the pressure is atmospheric (Heywood 1988).

The ER is decided by the geometry of the engine. In conventional engine as mentioned earlier, ER is equal to CR. Effective compression ratio is called as Compression Ratio (CR) unless otherwise mentioned. CR is the ratio of volume at IVC to volume at TDC i.e.,

$$CR = \frac{V_{at\,IVC}}{V_{TDC}}$$

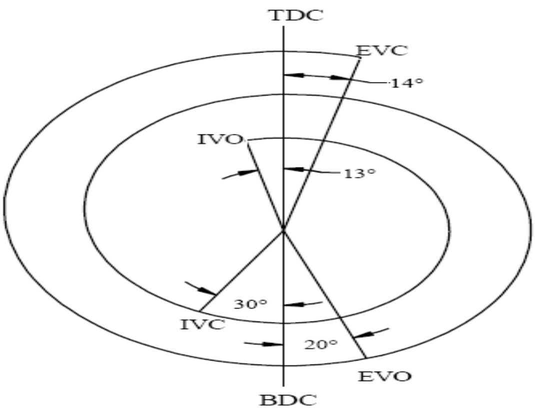


Fig 5 Valve Timing Diagram for ER/CR Ratio 1 (Conventional Engine).

4. Induction System

CI Engine intake manifold was modified to fit suitable carburetor. A side draught carburetor was used in the modified engine. To adjust the air-fuel ratio arrangement has been made to adjust the main needle holder to vary its position at any throttle position. Since delay of IVC increases, the quantity of charge pushed back also increases, thus lesser amount of charge will be retained inside the engine cylinder.

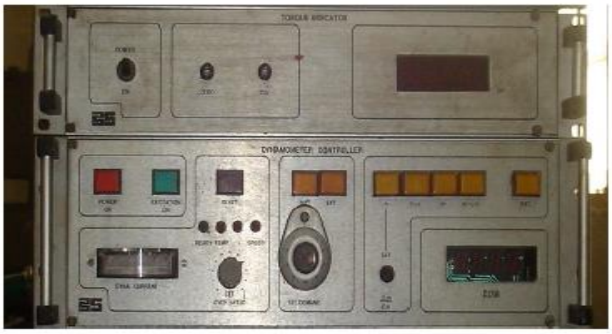


Fig 6 Dynamometer Controller and Torque Indicator

VI. RESULTS AND DISCUSSION

1. Effect of Compression Ratio on Lean Limit

Experimental values of brake power is plotted for different CRs to indicate the lean limit in terms of brake power. Figure 6.1 shows the effect of compression ratio on lean limit of EEE with ER/CR ratio of 1.5. The brake power output decreases as the mixture becomes lean. With very lean mixtures power output falls suddenly due to misfiring. The air-fuel ratio at which this event occurs is called the Lean Misfire Limit (LML).

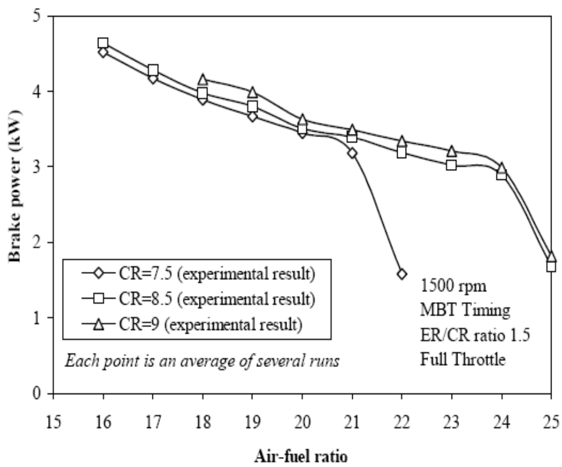


Fig 7 Effect of Compression Ratio on Brake Power Indicating Lean Limit.

2. Effect of Extended Expansion

Simulation results are in good agreement with experimental results. The percentage variation between the simulation and experimental values are between 3 to 12%. This deviation is due to theoretical assumptions made in the simulation procedure.

The result obtained in the present study is also compared with conventional extended expansion engine (i.e. with out applying lean burn concepts) studied by Ganesan (1998). Test engine used by the other researcher was single cylinder, vertical, air-cooled, four stroke, spark-ignition engine and which was tested at 1500 rpm, air-fuel ratio 18 with full-throttle conditions and with effective compression ratio 8.

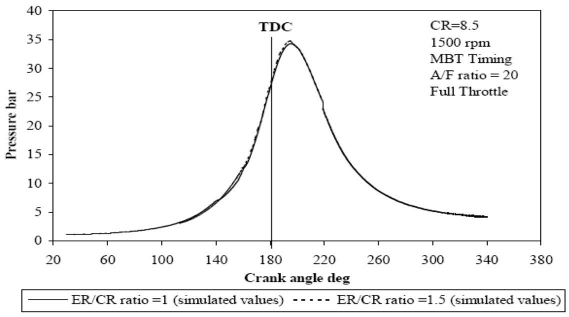


Fig 8 Pressure variations with respect to crank angle for conventional and extended expansion engine.

CONCLUSIONS

- CO is emission relatively high in EEE engine. For a CR of 8.5, compared to the base engine, engine with ER/CR ratio 1.5 shows, about 23.56% increase in CO emission. But the numerical value of CO emission is negligibly low compared to when engine operated with rich mixture.
- ER/CR ratio of 1.5 is found to be optimum for the extended engine considered to give maximum brake thermal efficiency and minimum BSFC.
- The effect of increase in compression ratio of EEE with ER/CR ratio 1.5 are, i.e.

when compression ratio increases from 7.5 to 8.5, 3.76% improvement in the brake power, 6.76% improvement in brake thermal efficiency, 6.36% reduction in BSFC and the percentage increase of NOX emission, UBHC emission and CO emission are respectively about 3.59%, 3 .77% and 9.91%. But compare to the base engine the percentage increase in emission are low in EEE.

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