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

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***Abstract* - A thermodynamic modelling of extended expansion lean burn has been developed to predict the performance and emission characteristics. Experimentally a single cylinder four-stroke water-cooled diesel engine has been converted to operate as SI engine. To achieve lean combustion thefollowing 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.The top dwell of the intake cam were increased to delay intake valve closing. Four different top dwell were formed in the camshaft to achieve the intake valve closing timing 93º 113º 125º, 134º after BDC corresponding to the ER/CR ratio of 1.25, 1.5, 1.75 and 2 respectively. A side draught carburetor was used in the modified engine. 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. Considering the maximum brake thermal efficiency and minimum BSFC, ER/CR ratio 1.5 and air-fuel ratio about 20 is found to be optimum. When the lean limit extend from an air-fuel ratio 17 to 20, EEE with ER/CR ratio 1.5 shows about 16.55% improvement in brake thermal efficiency, 21.45% decrease in BSFC, 53% reduction in NOX emission, 15.39% increase in UBHC emission, 45.38% reduction in CO emission. The maximum brake thermal efficiency and volumetric efficiency was observed at 1400 rpm.**

***Keywords*- IVC, TCI,BSFC and NOX etc.**

I.INTRODUCTION

Raising the ER unfortunately raises the CR as well. However, by suitably modifying the Intake Valve Closure Timing (IVCT) it is practically possible to increase the ER alone without adversely increasing the effectiveCR. It is possible to close the intake valve late during the compression process or by early closing intake valve as tried by James H. Tuttle (1980; 1982). Late closing of the intake valve modifies the p-V diagram of Otto cycle into what is known as Otto-Atkinson cycle (David Luria and Yahada 1982) or modifiedAtkinson cycle (Georege R. Eakin 1988).

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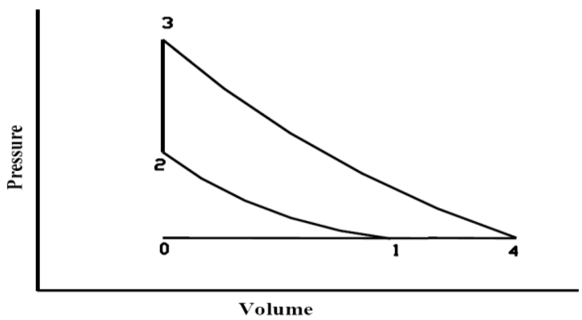

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Fig 1 Atkinson Cycle.

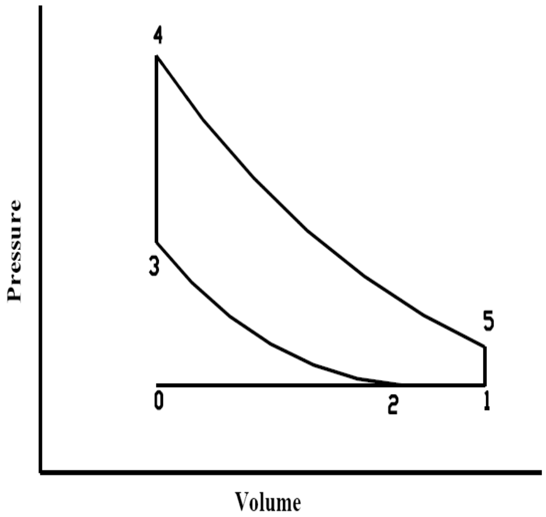

Fig 2 Modified Otto-Atkinson Cycle.

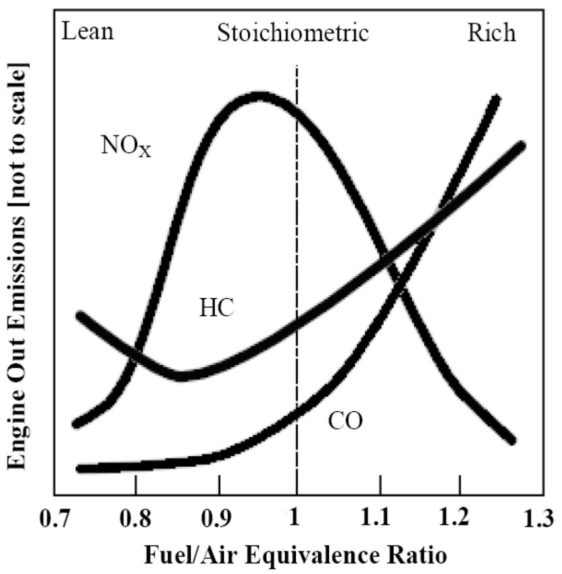

Fig 3 Effect of Air-fuel Ratio on Power, Fuel Consumption, and Emissions.

II. LITERATURE REVIEW

Fredrik Soderberg and Bengt Johansson (2001) developed cross flow cylinder head has been used for comparison between throttled and unthrottled operation using LIVC. Pressure measurements have been used forcalculations of indicated load and heat-release. Emission measurements had also been made. A model was used for estimating the amount of residual gases resulting from the different load strategies. Unthrottled operation usinglate intake valve closing resulted in lower pumping losses, but also in increased amounts of residual gases, using this cylinder head. This is due to the special design, with one intake valve and one exhaust valve per camshaft.

Late intake valve closing was achieved by phasing one of the camshafts, resulting in late exhaust valve closing as well. With very late phasing ,low load - the effective compression ratio was reduced. This, in combination with high amount of residual gases, resulted in a very unstable combustion. The main combustion, 10-90% heat-release, was quite fast for all load control strategies, indicating high turbulence. This could not be seen in the early combustion, however. An increase in fuel conversion efficiency could be achieved with up to 9%. The emissions of NOX were reduced by 4-5 g/kWh with LIVC for loads between 4-8 bar IMEP.

III. ANALYSIS

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.

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

Mean effective pressure absorbed in friction due to wall tension of rings.

$$F_{mep2} = 10*\frac{0.377*S*n_{pr}}{B^2}$$

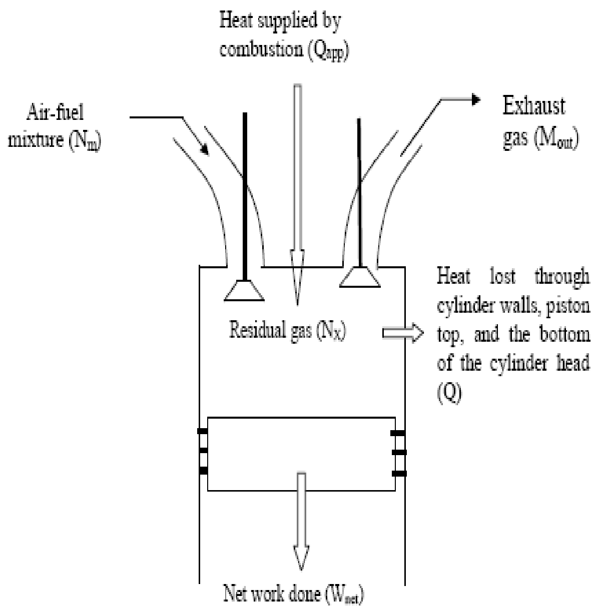

Fig 4 Schematic Diagram of Thermodynamic Modeling Process.

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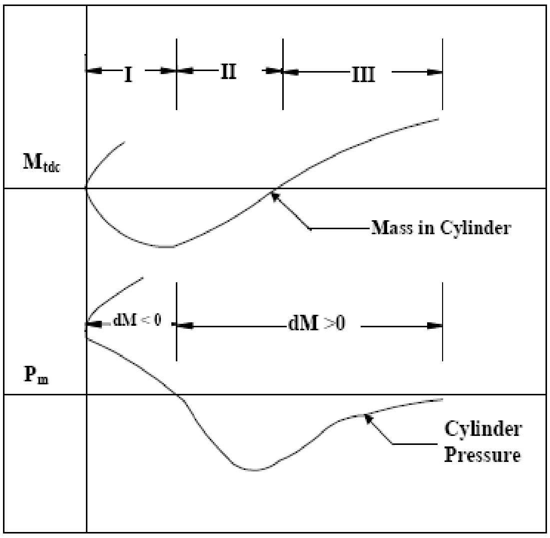

Fig 5 Particulars of Events in the Early Intake Stroke (Ganesan 1999).

To describe the NOX formation process the extended Zeldovich mechanism is adopted (Heywood 1988).

$$N + NO \leftrightarrow N_2 + O \qquad K_{1f} = 3.1 \times 10^{10} \exp\left[\frac{-160}{T}\right]$$

$$N + O_2 \leftrightarrow NO + O \qquad K_{2f} = 6.4 \times 10^{6} . T . \exp\left[\frac{-3125}{T}\right]$$

$$N + OH \leftrightarrow NO + H \qquad K_{3f} = 4.2 \times 10^{10}$$

IV. EXPERIMENTAL IVESTIGATIONS

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. Raising ER. Will raise geometrical compression ratio .also. However, by suitably, modifying the Inlet Valve Closing Timing (IVCT) it practically possible to increase the ER alone without increasing effective compression ratio (i.e., Vat IVC/VTDC).

Reduction in CR by late IVCT calls for a reduction in the clearance volume to obtain original CR. This in turn increases the geometric ER. Thus by suitably altering the IVCT and clearance volume, ER alone can be
increased without increasing the CR. The engine working on this modified Atkinson cycle hereafter will be referred to as Extended Expansion Engine (EEE).

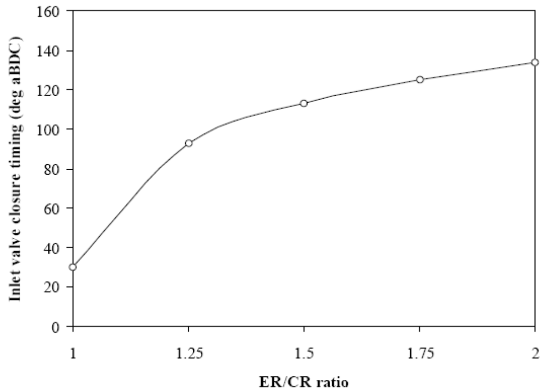

Fig 6 Variation of Intake Valve Closing Timing with ER/CR Ratio to Maintain Constant CR.

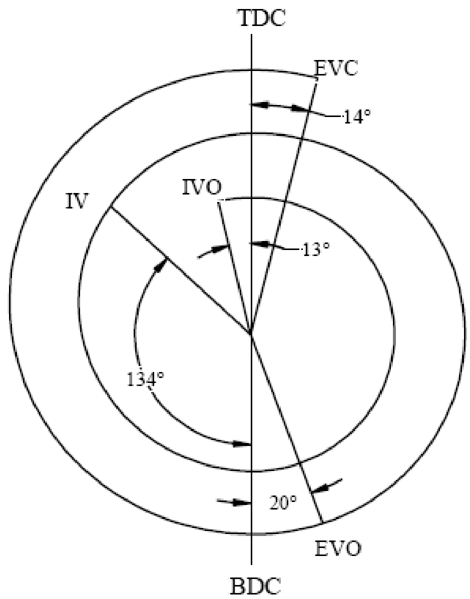

Fig 7 Valve Timing Diagram for ER/CR Ratio 2.

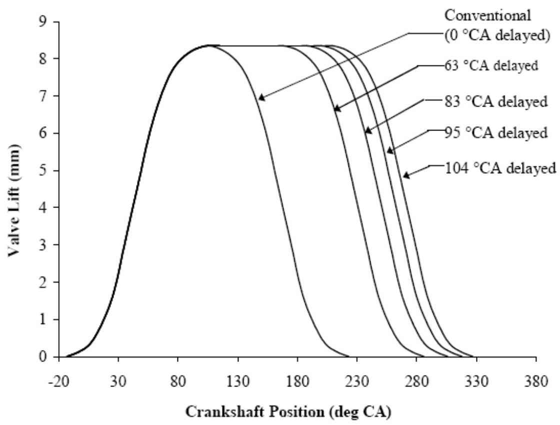

Fig . 8 Valve Lift Diagram.

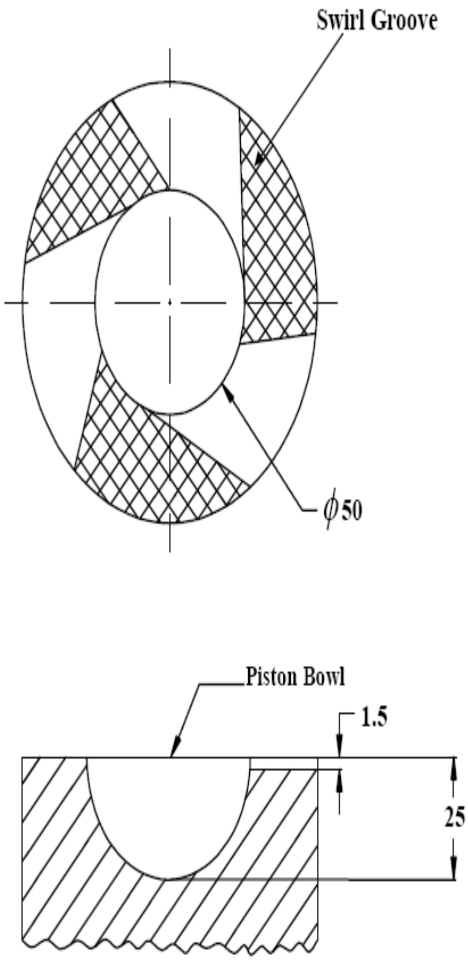

Fig 9 Geometry of Enhanced Swirl Combustion Chamber.

A 12 volt high energy Transistorized Coil Ignition (TCI) system was used in this study. Use of TCI system give high output voltage and longer spark duration (Heywood 1988, Jose Geiger et al 1999). A suitable drive hasbeen taken from the engine camshaft to operate the ignition system. The distributor points and cam assembly of the conventional system are replaced by a magnetic pulse generating system, which send an electrical pulse to trigger the ignition system.

V. RESULTS AND DISCUSSION

The variation of CO against ER/CR ratio. CO emission is mainly depends of air-fuel and other variables has less effect (Heywood 1988)). As ER/CR ratio and CR increases, there is slight increase in CO emission level. The range of CO emission values are a minimum of 0.035% Vol. and the maximum of 0.06% Vol. For both base and modified engine, because of lean operation oxygen availability is abundant, due to above reason the maximum value CO emission itself negligibly lower compared with rich operation.

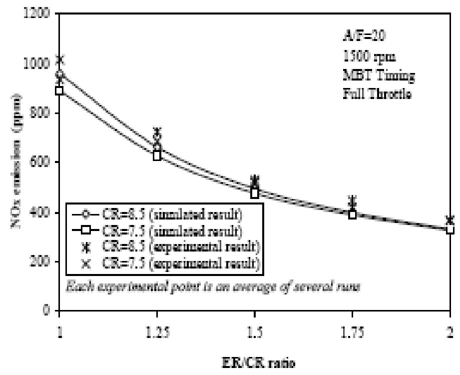

Figure 6.12 Variation of NO_x Emission with ER/CR Ratio

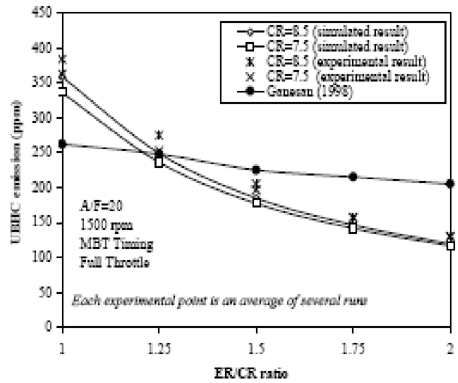

Fig 10 Variation of UBHC Emission with ER/CR Ratio.

There is a considerable improvement in the maximum brake thermal efficiency obtainable from an extended expansion lean burn engine. ER/CR ratio 1.5 is found to be the optimum ratio, giving a maximum brake thermal efficiency of 35.01% against 28.72% of the base engine. There is about 17.95% improvement in thermal efficiency. Compared to the base engine, EEE with ER/CR ratio 1.5 shows 16.63% less brake specific fuel consumption.

VI. CONCLUSIONS

- 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
- The percentage increase of brake power due to increase in engine speed from 1000 rpm to 1600 rpm is in the order of 63.9% for EEE with ER/CR ratio 1.5 and 3.73% reduction in
- break thermal efficiency and NOX emission increases by 31.9% and UBHC emission reduced by 12.94%. The

effects of speed on CO emission levels were negligibly low.

- Compare to mid-load operation, full load operation EEE with ER/CR ratio 1.5 shows 5.89% increase in brake thermal efficiency, 12.07% reduction in BSFC, 20.97% increase in NOX emission, 19.19% reduction in UBHC emission and 11.82% increase in CO emission.
- At mid-load the BSFC of EEE with ER/CR ratio 1.5 is around 5.26% less compare to base engine.

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