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NUMERICAL AND EXPERIMENTAL STUDY OF EXTENDED EXPANSION CONCEPT APPLIED TO LEAN BURN SPARK IGNITION ENGINE

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ABSTRACT: This paper deals with numerical and experimental study of 4-stoke, Single cylinder, Spark Ignition, Extended Expansion Lean Burn Engine. Engine processes are simulated using thermodynamic and global modeling techniques. Two- zone combustion model is used for analysis of combustion process. In the simulation study the following process are considered compression, combustion, and expansion. Sub-models are used to include effect due to gas exchange process, heat transfer and friction. Also emission characteristics like unburned hydrocarbon and nitric oxide are predicted.
In the present work extended expansion concept is applied to a four-stroke lean burn engine to further improve performance and emission characteristics. The result presented compares the performance and emission characteristic of extended expansion lean burn engine with ER/CR ratio 1.5 with base lean burn engine. Compared to base lean burn engine extended expansion lean burn engine shows better performance and emission characteristics i.e. 17.95 % improvement in the thermal efficiency, 16.63 % improvement in fuel consumption, 48.27 % reduction NO_X emission and 48.4 % reduction UBHC emission for engine with a CR of 8.5.
KEYWORDS: Late Closing of Intake Valve (LIVC), Lean Burn Engine, Extended Expansion Engine (EEE)

INTRODUCTION

The Otto-Atkinson Engine is much more efficient at part loads than the present day spark-ignition (Otto cycle) engine. To achieve this efficiency, the Otto-Atkinson engine greatly reduces two of fundamental losses of the Otto cycle engine. One loss occurs because of the less than optimal expansion ratio at part loads, and the other loss occurs when drawing the air into the cylinder at less than atmospheric pressure. Increasing the expansion ratio (by means of increasing the geometric compression ratio) will extract more work from the combustion gas [1]. Many developers focus primarily on the improvement of thermal efficiency at light loads and low speeds. One way to improve efficiency at light loads is through the use of a Late Intake Valve Closing technique (LIVC) [2].

The intake valve is then kept open well into the compression stroke allowing part of the charge to be returned to intake manifold. The movement of the charge takes place at approximately atmospheric pressure and hence little pumping work should be expended. The power output can be regulated by the amount of LIVC selected [3, 4]. Figure.1 illustrates the pumping loop for an ideal late IVC strategy and the reduction in pumping loss compared to conventional throttled operation. The pumping losses of conventional engine operates at part load are substantial. Fuel Economy of the homogenous-charge spark-ignition engine can be increased by reducing the losses incurred during conversion of the indicated work to shaft output. The pumping loss at the full load consumes about 5% of the indicated power, whereas the pumping loss at light load consumes about 50% of indicated power [5].

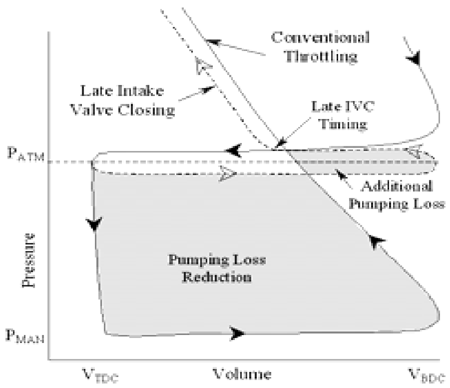

Figure 1. Late Intake Valve Closure operation

In the development of internal combustion engines, there has been a continuous effort to reduce fuel consumption and exhaust emissions. Lean combustion is a preferred concept for reducing exhaust emissions for meeting stringent emission standards. Improved fuel efficiency at constant or even further improved exhaust gas emissions is one of the major challenges that engineers and scientists in the automotive industry and its partners are currently facing.

Considering SI engine, especially the reduction of pumping and heat losses during part load operation offers the potential for performance improvement. Shifting of operating points to high loads by either downsizing or reduction of pumping and heat losses by mixture dilution give similar possibilities. However, the highest potential to reduce fuel consumption is given by concepts which combine the advantages of the lean burn engines [6].

In a conventional spark ignition (SI) engine, the compression ratio is equal to the expansion ratio. 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 [7].

This paper mainly deals with the numerical and experimental studies on single cylinder, four stroke, spark ignition, Extended Expansion Lean Burn Engine. Extended Expansion Engine with intake valve closure delayed to produce an expansion ratio that is larger than the compression ratio. The Engine processes are simulated on a computer using thermodynamic and global modeling techniques.

❖ MODELING OF EXTENDED EXPANSION LEAN BURN SPARK IGNITION ENGINE

In this work, theoretical models for the engine processes have been developed by adopting thermodynamic and global modeling techniques. The processes include compression, combustion, and expansion. The effect of gas exchange process, heat transfer and friction were also included in the model [8,9]. The following assumptions were made in the development of models 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
- There are no deposits on the walls of the combustion chamber
- Temperature of the burnt gases at the beginning of combustion is equal to the adiabatic flame temperature.
- Metal surface temperatures of all the boundary surfaces are constant throughout the cycle

Compression Process

The computations were started from the inlet valve closing and the compression process was assumed to start from this point. During the compression period, the charge in the cylinder was assumed to consist of the air fuel mixture and the residual gas. During the first iteration, pressure and temperature at the start compression was assumed and the residual gas fraction was assumed to be zero. At the end first iteration, the parameters are calculated and the iteration is repeated until the intake temperature converges.

Combustion Process

The Wiebe's function has been used for the mass fraction burned

$$x(\theta) = 1 - \exp\left[-a\left(\frac{\theta-\theta_i}{\Delta\theta_c}\right)^{m+1}\right]$$

where, θ-Crank angle (deg), θ_i-Crank angle at the start of combustion (deg), $\Delta\theta_c$ - Total combustion duration (deg)

The a and m are adjustable parameters that are selected to provide a match with experimental information. For this work a = 5 and m = 2.2 as recommended by Heywood, et al. [11].

Expansion Process

During the expansion process the model was solved as described in the compression process. The actual expansion process starts at the end of combustion as was assumed to be when the exhaust valve opens. Dissociation is taken into account. The chemical reactions continue to take place during the expansion stroke, this is because the chemical species are in thermodynamic equilibrium. The composition of the gas mixture will there varies with pressure and temperature. Twelve species are considered to be present in the product in the cylinder during combustion and the early part of expansion. Twelve equations are required to calculate the twelve-mole number in the equilibrium state. These are supplied by the conservation and by the mass action equations. This equation is solved by the Newton Raphson method to determine the product mole number. Also the gas exchange process, friction calculation and equilibrium calculation of species were included in the modeling process [8,13].

Emission Characteristics

Rate Kinetics of Formation of Nitric Oxides

The calculations were based on the equilibrium assumption except for NO_x formation where the extended Zeldovich Mechanism was used.

$$N+NO \leftrightarrow N_2 + O$$
$$N+O_2 \leftrightarrow NO+O$$
$$N+OH \leftrightarrow NO+H$$

The final rate equation for [NO]

$$\frac{1}{Ven}\cdot\frac{d([NO].V_{en})}{dt} = 2(1-\alpha^2 NO)\left[\frac{R1}{1+\alpha NO(R1/R2+R3)}\right]$$

where $R1=K_{1f}[NO]_e[N]_e$; $R2=K_{2f}[N]_e[O2]_e$; $R3=K_{3f}[OH]_e[N]_e$

Unburned Hydrocarbon Formation

A reasonable fit to experimental data on Unburned Hydrocarbon burn up is the rate expression [14] is used for prediction of Unburned Hydrocarbon

$$\frac{d[HC]}{dt} = -6.7X10^{15}\exp\left[\frac{-18735}{T}\right][HC][O_2]\left(\frac{P}{RT}\right)$$

Carbon Mon-Oxide (CO) Formation

The carbon monoxide emissions lie between those for equilibrium at peak pressure and the equilibrium at exhaust valve opening [14]. In this study the CO emission was predicted based on the equilibrium at peak pressure and temperature.

❖ DESCRIPTION OF COMPUTER PROGRAM

The computer program was written in 'c' language. The program consists of following parts main program, function and subroutines. The main program reads all the necessary input data such as engine geometry, type of fuel, engine speed, valve opening and closing timing and equivalence ratio. The constants and variables required for this program was defined as a separate function and which is included in the main program. Functions like C_v, U_p calculate the values of specific heat and heat of reaction with respect to variations in temperature. There are fourteen subroutines used to simulate the performance and emission characteristics namely pre-calculations, volume calculations, compression, combustion, equilibrium calculations expansion, gas exchange, overlap, heat transfer, inlet and exhaust valve area, FMEP, NO_X, UBHC. The simulated value for different conditions stored as date file.

❖ EXPERIMENTAL SETUP

A single cylinder, four strokes, water-cooled, diesel engine is used for the experiment. This diesel engine was converted to operate on Otto cycle by replacing the fuel injection pump and the injector with a Venturi type carburetor, a spark plug and an ignition system. Compression ratio of the engine was reduced to the required values, by changing the clearance volume. To operate engine as extended expansion lean burn engine the following modification were done.

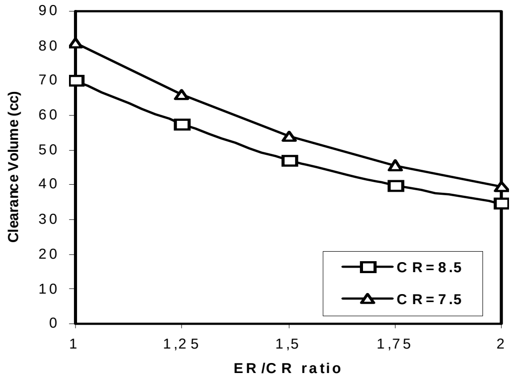

Figure 2. Variation of clearance volume with ER/CR ratio

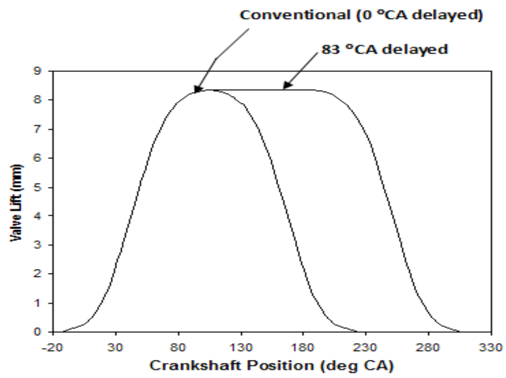

Figure 3. Cam dwell and valve lift profile

The sparkplug is fixed in the injector hole itself since valve layout did not permit the central location of the plug though the central location of the plug is considered to be ideal. Reduction in compression ratio is achieved by reducing the clearance volume. The clearance volume of the original engine was reduced by placing shims of standard thickness between the cylinder block and crankcase.

Figure 2 and 3 shows the required clearance volume and the corresponding cam dwell to maintain different ER/CR ratio. Clearance volume has been reduced to vary the ER and to keep CR (or ECR) constant depending of late IVCT. The present work compares the performance and emission characteristic of base lean burn engine with extended expansion (i.e. ER/CR ratio 1.5) lean burn engine for a CR of 8.5.

Table.1 Valve Timing

Intake-Valve Dwell (°CA)	Intake Valve		Exhaust Valve	
	Open	Close	Open	Close
0 (Conventional)	13°bTDC	30°aBDC	20°bBDC	14°aTDC
83 (Modified)	13°bTDC	113°aBDC	20°bBDC	14°aTDC

The top dwell of the intake cam were increased by building-up the material in the intake cam. The materials were build-up in the intake cam top portion and it is machined to get required top dwell. The material build-up is done to only vary top dwell. The other part profiles of the cam were not disturbed. Similarly in the other cams no changes were made they are kept as it is since our aim is vary only intake valve closing timing. However, the disadvantage of this method is that for each valve-timing camshaft has to be dismantled and material has to build-up and machined to required shape. This way of testing is only for the research application. Table.1 shows the valve timing of conventional and modified engine.

Squish and Swirl levels were enhanced during compression process by having a hemispherical bowl and swirl grooves on the crown. This type of piston configuration was test earlier by other researcher [15] found to improve performance with lean mixture. Hence the same configuration is used to enhance the swirl level. Copper as a catalyst was coated on the cylinder head and piston crown by using the plasma spray technique. This coating is capable of improving combustion of lean mixture, which has proved by other researcher [15, 16, 17]. 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. In order to prevent back flow of charge through the carburetor in to the surge tank, a suitable reed valve was used [18, 19, 20]. A 12 volt high energy TCI (Transistorized Coil Ignition) system was used in this study.

❖ RESULTS AND DISCUSSION

The results presented here compares the lean operation performance characteristics of base lean burn engine (ER/CR ratio 1) and extended expansion lean burn engine (ER/CR ratio 1.5) for a compression ratio of 8.5 and speed 1500 rpm for different air fuel ratios. In the graphical plots continuous line shows the simulated results and symbols shows experimental values. The experiments were conducted for MBT timing. Trends of simulation results are in good agreement with experimental trend. There is about 4 to 12 % percentage variation between the simulation and experimental values. This deviation may be due to theoretical assumptions made in the simulation procedure.

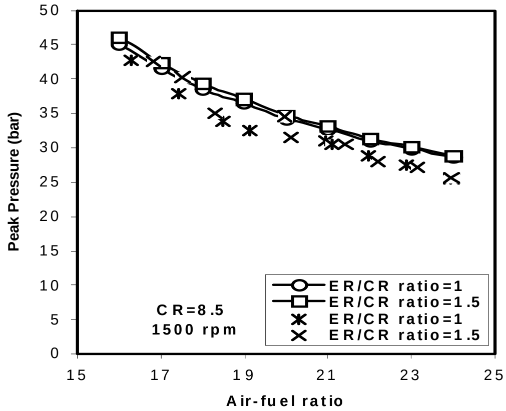

Figure 4. Variation of cylinder peak pressure with air-fuel ratio

Peak Pressure

Figure.4 shows the peak pressure variation with air-fuel ratio for the conventional and extended expansion lean burn engine. As the air-fuel ratio increases the peak pressure decreases. This trend can be attributed to the dissociation of triatomic molecules CO_2 and H_2O. Thus the fraction of chemical energy of fuel, which is released as sensible energy near TDC is greater; hence greater fraction of fuel energy is transferred as work to piston during expansion [14]. The percentage decrease in peak pressure is about 17.89 %, when air-fuel ratio increased from 17 to 20 for EEE with ER/CR ratio 1.5 and the same for base engine.

17.4 %. Because of maintaining constant compression ratio (CR=8.5) in EEE the percentage reduction of peak pressure in the same order. This percentage reduction in peak pressure intern reduces the output of the engine.

Brake Power

Figure.5 shows the variation of brake power with air-fuel ratio for ER/CR ratio of 1 and 1.5. The brake power output decreases as the mixture becomes lean. With very lean mixtures power output falls suddenly may be due to reduced peak pressure when mixture becomes lean. The percentage reducing of brake power when the air-fuel ratio increased from 17 to 20 is about 17.32 % in EEE with an ER/CR ratio of 1.5 and for the base engine 19.72 %. Because of the extended expansion concept even though brake power decreases, the percentage reduction in brake power due to increase in air-fuel ratio is lower i.e. there is about 2.4 % less in EEE engine.

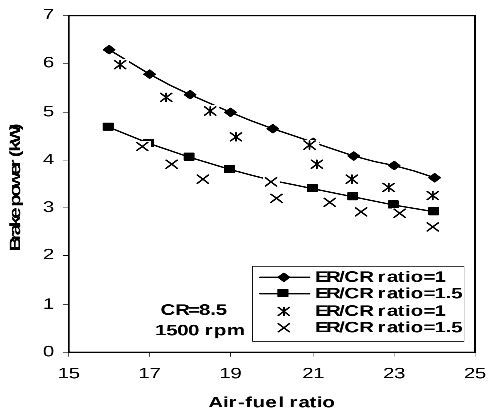

Figure 5. Variation of brake power with air-fuel ratio

Reduction in brake power due to increase in air-fuel is not affecting the sustainable operation of engine till an air-fuel ratio of 24, when the air-fuel ratio further increased sustainable operation of the engine is not possible. This is may be due misfiring of charge.

Brake Thermal Efficiency

Thermal efficiency of internal combustion engine increases as the air-fuel ratio increases and reaches a theoretically maximum value when air-fuel ratio becomes infinite (i.e., only air is used as working medium). The increase in efficiency is mainly due to three basic effects, which results from lower combustion temperature associated with lean mixtures; increased ratio of specific heats, reduced dissociation losses and reduced heat transfer through combustion chamber walls.

The variation of brake thermal efficiency with different air fuel ratios for ER/CR ratio of 1 and 1.5 is shown in Figure.6. It is observed that, brake thermal efficiency increases as the mixture becomes lean up to an air-fuel ratio of 20 and then brake thermal efficiency tends to reduce rapidly. It decreases as the mixture becomes too lean. This may be due to an incomplete combustion, which results in production of less thermal energy. The percentage increase in thermal efficiency when air-fuel ratio increased from 17 to 20 is about 12.83 % in the base engine and the same in EEE with ER/CR ratio 1.5 is about 16.55 %. Compared to base lean burn engine, percentage improvement of efficiency in EEE lean burn engine is higher and the maximum efficiency for the both engine was achieved around an air-fuel ratio of 20.

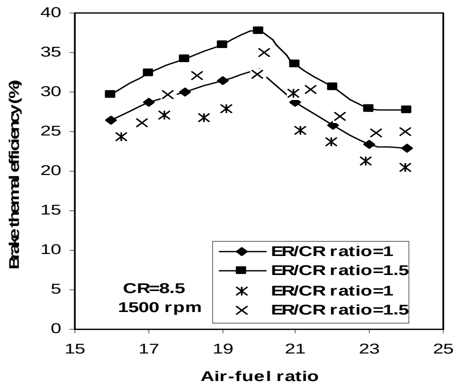

Figure 6. Variation of brake thermal efficiency with air-fuel ratio

Brake Specific Fuel Consumption (BSFC)

Figure 7 show the variation of BSFC with air-fuel ratio. As air-fuel ratio increases the BSFC decrease up to an air-fuel ratio of 20 and then start to increase there after. The engine fuel consumption strongly depends on the engine combustion chamber and mixture preparation quality. Other research also shows the same trends that the BSFC decreases till equivalence ratio is less than 0.8 [12]. The fuel consumption reduction is due to two factors, the more complete and faster oxidation and the smaller throttling losses. A reduction of 18.4 % in the fuel consumption in the case of base engine and 21.45 % reduction in the case of EEE with ER/CR ratio 1.5. Considering the improvement in the brake thermal efficiency and fuel consumption, the optimum operating is of engine is around an air-fuel ratio of 20 for both base and EEE lean burn engine. Further increase the air-fuel reduces both performance factors.

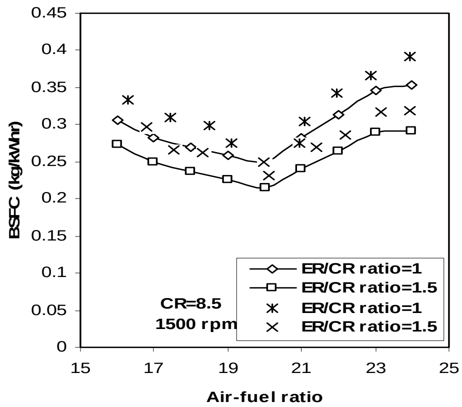

Figure 7. Variation of brake specific fuel consumption with air-fuel ratio

NO_X Emission

The formation rate of NO_X depends on the gas temperature and oxygen concentration. The maximum burned gas temperature occurs at $\phi \approx 1.1$ (A/F =13.3), at this equivalence ratio, the oxygen concentration is low. As mixture is leaned out increasing the oxygen concentration initially offsets the falling gas temperatures and NO_X emissions peak at $\phi \approx 0.9$ (A/F = 16.2). Then, decreasing temperatures dominate and NO_X emission decreases to low levels [14]. Figure 8 shows effect air-fuel ratio NO_X emission with ER/CR ratio of 1 and 1.5. The NO_X emission is found maximum when the air-fuel ratio is about 17 and then decreases drastically as the air-fuel ratio is greater than 17. The experimental values of NO_X emission were 6-12 % higher than the simulated values. The percentage decrease of NO_X emission for EEE due to increase of air-fuel ratio from 17 to 20 is about 43 %, which around 4 % higher than base lean burn engine. This observation reveals the temperature dependence of NO_X emission.

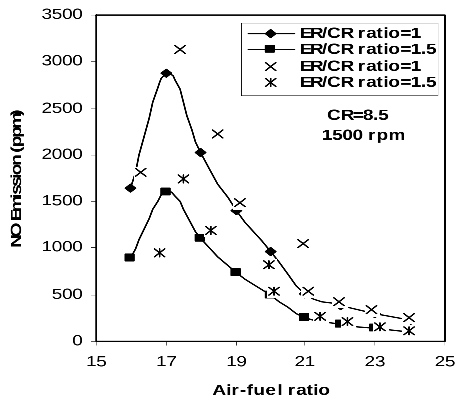

Figure 8. Variation of NO_X with air-fuel ratio

Unburned Hydrocarbon (UBHC) Emission

Figure 9 shows the variation of UBHC emission characteristics with air-fuel ratio for ER/CR ratio 1 and 1.5. As the air-fuel increases the UBHC emission initially decreases up to air-fuel 17, after that increases. When mixture is too lean i.e., after air-fuel 21 the increase in UBHC emission is steep. This increase in UBHC may be due ignition inhibition or poor combustion. The percentage increase of UBHC emission, when the air fuel ratio is increased from 17 to 20 is about 12.08 % for the base engine and 15.39 % for EEE with ER/CR ratio 1.5. This may be due of deterioration combustion quality as mixture become lean. Compare to the improvements in other operating parameter like BSFC, brake thermal efficiency, NO_X emissions increase in UBHC emissions has to be scarified.

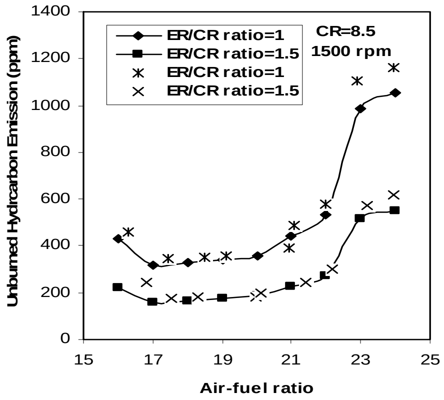

Figure 9. Variation of UBHC with air-fuel ratio

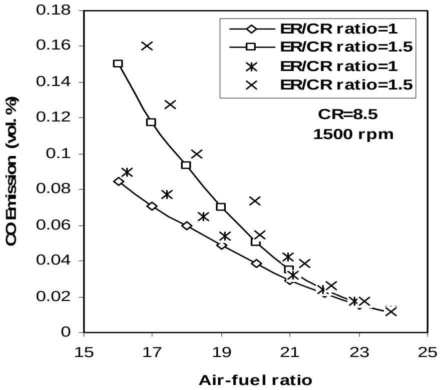

Figure 10. Variation of CO emission with air-fuel ratio

Carbon Monoxide (CO) emission

CO emissions from the internal combustion engines are controlled primarily by the air-fuel ratio. For rich mixtures, the CO concentration in the exhaust increases steadily with increasing equivalence ratio. For lean mixtures, CO concentration in the exhaust varies little with equivalence ratio (Heywood J.B (1988)). Figure 10 shows variation of CO emission with air-fuel ratio for ER/CR ratio 1 and 1.5. CO emission decrease as mixture becomes lean due to availability of abundant oxygen. The percentage reduction of CO emission when the air-fuel ratio increased from 17 to 20 for the base engine is about 56.9 % and that of EEE with ER/CR ratio 1.5 is about 45.38 %. CO emission reduction is more in the case of base lean burn engine compare to extended expansion lean burn engine.

❖ CONCLUSION

Based on the experimental and simulation results obtained the following conclusion are drawn.

- ❖ The experiments result shows the mathematical model developed is good agreement with simulation results. The mathematical model developed can be used as an effective analytical tool to calculate quickly and inexpensively the effect of design and operating parameters on engine performance and emissions.
- ❖ The optimum air-fuel ratio is found to be 20 with respect to maximum thermal efficiency and BSFC.
- ❖ There is about 1-3% improvement in fuel consumption when engine is operates in extended expansion. The optimum value of BSFC is at ER/CR ratio 1.5.
- ❖ Compare to the base lean burn engine, the NO_X, and UBHC emission level are low in the extended expansion lean burn. CO is emission relatively high in EEE engine.

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