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ENGINE PERFORMANCE PARAMETRES AND EMISSIONS REDUCTION METHODS FOR SPARK IGNITION ENGINE

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ABSTRACT

An experimental study is carried out to investigate engine performance parameters and methods of reducing emissions from spark ignition engine. The used engine is four stroke four cylinder naturally aspirated spark ignition engine with compression ratio of 9, bore diameter of 80 mm and stroke of 90 mm. The engine performance parameters are presented with and without exhaust gases recirculation (EGR). Engine cylinder pressure measurements and engine geometry are used for calculating indicated engine performance parameters. UHC and CO concentrations are measured with EGR, catalyst converter and air injection in the exhaust manifold. UHC and CO concentrations for different methods are compared with the original engine emissions. The investigated parameters are indicated and brake engine performance parameters, air/fuel ratio (AFR) and exhaust gases temperature ($T_{exhaust}$). Also, engine cycle to cycle variation (CCV) and sound pressure level (SPL) generated from engine are calculated using cylinder pressure, inlet and exhaust manifold pressure measurements. EGR rate of 5%, 7%, 8%, 10% and air injection rate of 3%, 4%, 5%, 6% are used in the present work.. Catalyst converter and air injection in exhaust manifold are useful methods for reducing UHC and CO concentrations on contrast UHC and CO concentrations increase with the increase of EGR. Air injection in the exhaust manifold represents method for reducing UHC and CO exhausted from spark ignition engine. The present work is useful for improving engine performance parameters, reducing engine emissions and further development of spark ignition engines.

Key words (Performance, emissions, EGR, noise, catalyst, air injection)

1-INTRODUCTION

The methods and techniques used to reduce emissions from spark ignition engines have some impact effects on engine performance. So, many researches directed their researches to increase spark ignition engine efficiency. For spark ignition engine a reasonable solution for reducing emissions is by controlling some combustion parameters, in such way engine performance is kept unaltered. Choongsik and Junemo (2001) carried out an experimental study to investigate the effect of EGR on combustion characteristics of spark-ignition engine. Their results show that decreasing EGR temperature by 180 ^oC enabled the reduction of exhaust gases temperature by 15 ^oC in cooled EGR test at 1600 rpm, 370 kPa BMEP and consequently the reduction of thermal load at exhaust. EGR is widely used to reduce NO_X, pumping loss, and to increase thermal efficiency in interna combustion engines. However, it does have some disadvantages such as its detrimental effects on combustion stability. The coefficient of variation of indicated mean effective pressure (COV_{imep}) was shown in order to depict the change of cycle-by-cycle variation. Engine speed of 1200 rpm and light load operation resulted in significant rises in COV_{imep} up to 17% at 30% of EGR rate, while under medium and high loads conditions combustion stability was assured within 3% of COV_{imep}, although EGR rate was increased up to 30%.

Modern catalyst converters are highly effective at eliminating the majority of the three regulated pollutants at well defined operating conditions. A three-way catalyst is generally used to reduce NO_X and oxidize CO and UHC. An intelligent catalyst is one of the modern methods for reducing UHC emissions, whereby the catalyst agents and catalyst substrate are designed such that the catalyst reacts to the exhaust environment in order to achieve a higher efficiency Hirohisa et al. (2001). An alternative approach to oxidizing UHC and CO emissions during cold start is the store UHC and CO on-board and then release them into the catalyst once light off has been achieved, at which point the catalyst is capable of converting UHC and CO to CO_2 and H_2O. This can be accomplished by using UHC adsorbed in line with the catalyst converter Kanazawa (2001). Catalyst operating temperature range must be carefully controlled. The catalyst is generally not highly active below 300 ºC and begins to age rapidly above 800-900 ºC. Above 1000 ºC severe aging of the catalyst can occur and at temperatures in the range of 1400 ºC the substrate itself can melt.

Synchronized secondary air injection (SAI) method has been proposed by Sim et al. (2001) to reduce UHC and CO emissions by injecting secondary air intermittently into the exhaust port. Results show that UHC and CO reduction rates and exhaust gases temperature are sensitive to the timing of SAI. Two optimum SAI timings are observed which are at 100 CA ATDC and 230 CA ATDC depending on spark timing. At the cold steady conditions, SAI is found to be more effective on UHC and CO reduction and also has much higher exhaust gases temperature than either the continuous SAI or the baseline condition. Onorati et al (2003), and Errico and Onorati (2006) have been extended GASDYN code to model the injection of air in the exhaust manifold. The code has been extended to simulate the injection of air in the exhaust manifold and predict the consequent post oxidation of pollutants in the ducts. The heat released in the gases due to the exothermal reactions has been taken into account to evaluate the exhaust gases temperature along the ducts with injection of air. The results show that UHC and CO decrease with the increase of air injection rate. An experimental study was performed by Mendillo and Heywood (1981) to determine the fraction of UHC and CO emissions which oxidizes in the exhaust port of a spark ignition engine. The technique used was injection of a CO_2 quench gas into the exhaust port at various planes along the port centerline to cool the exhaust gases and "freeze" the hydrocarbon oxidation reactions. By quenching the reactions of hydrocarbons at the cylinder exit plane, cylinder exit hydrocarbon emissions levels were determined. By differencing the concentration of hydrocarbons observed during quenching and non quenching operations, fraction of hydrocarbons reacting in each section of the exhaust was determined and UHC and CO exit from exhaust manifold found to be decreased. Injection of compressed air into engine has been used to improve the mixing of fuel and air during the intake stroke of the engine. This technique was used to inject the air directly into the cylinder of spark-ignition engine Gruenefeld et al (1997). The combustion performance of the engine during cold start was shown to be equivalent to that of a fully warmed-up engine after only a few cycles.

Aaron Hein et al. (2000) faced several challenges in dealing with the noise generated by the snowmobile. Noise coming from the exhaust, intake, and from the engine through the hood vents had to be reduced to an acceptable level based on the criteria stated in the rules for the competition. Russell et al. (2000) studied torque, emissions and noise generated from internal combustion engine. They found that the control of the pressure rise and rate of fuel burning are essential to control noise generates from combustion especially in heavy traffic. A minor contribution to the noise exciting propensity of the cylinder pressure development comes from the compression ratio. The engines of high compression ratio have slightly more energy in their high frequency harmonics than the low compression ratio. Hans Bodén and Albertson (1998) studied sound pressure level (SPL) generated from inlet and exhaust systems of internal combustion engine. They found that, SPL generated from inlet and exhaust systems are high typically between 130 and 170 dB. The comparison between measured source data for different speeds and load shows that both the source strength and the source impedance depend more on the speed than on the engine load. The source impedance is determined more by exhaust or inlet manifold geometry than by engine operating conditions.

Cycle to cycle variation (CCV) in combustion occurs during the early stages of the combustion process. A study done by Soltau (1961) show cyclic variations occur from the time of spark to the establishment of a fully developed flame. This is the period in which the developing flame kernel is susceptible to factors that cause CCV. Cyclic variations of the turbulence level are a major cause for cycle-to-cycle variations in combustion. A study done by Hamai et al. (1986) showed that incomplete mixing of fuel/air and residual contribute to CCV in combustion. Also, the increased of residuals resulted in increased cyclic pressure variability. Engine speed also contributes to CCV in combustion, where increasing the engine speed resulted in an increase in flame speeds and cyclic flame speed variations. Increase in turbulence has also been attributed to engine speed and the higher turbulence is the main reason for the increase in flame speed variations. Brehob and Newman (1992) obtained experimental results from 500 consecutive individual cycles. Their results included several burn durations, peak cylinder pressures, crank angles of peak pressure and indicated mean effective pressures. They determined the mean and standard deviation for each of the above quantities. For the most part, the results were consistent with a normal distribution.

Scherer et al. (1997) carried out simulation model for analysis of manifold pressure pulsations, its influence on mean value throttle and manifold models. The simulation results of the throttle and manifold model are discussed and compared with experimental results from a steady state flow test bench and engine experiments. By calculating the correct geometrical area of flow, at a constant discharge coefficient, the air mass error is less than 10% compared with steady flow experiments. But with manifold pressure pulsations, there is a large difference between simulated and measured air mass flow which has to be compensated when using mean value models. The influence of the pressure pulsations is compensated by modeling the effective area of flow as a noise process and combining its estimated state with the estimated mean value of the map and the air mass flow at the throttle. The periodic motions of the intake valves and the suck cycle of the engine produce strong changes in the air mass flow and pressure waves in the manifold. By optimizing the geometry of the manifold the pressure resonance points can be tuned to increase the volumetric efficiency of the engine at low speed. The aim of the present work is to study the effect of using EGR on engine performance, different methods for reducing emissions, engine noise and cycle to cycle variations for spark ignition engine.

2. EXPERIMENTAL SETUP

The present study is conducted on the engine at the research lab of Higher Institute of Technology, Benha University. The experimental setup is shown in Fig.(1). The engine is four stroke, four cylinder naturally aspirated spark ignition engine with compression ratio of 9, bore diameter of 80 mm and stroke of 90 mm. The present system provides a facility to conduct engine performance tests at different engine loads and speeds. The engine fuel system is modified by adding a custom tank and a flow metering system which used for fuel consumption measurement. A two glass burette of known volume and one 3 ways hand operated control valve which allowed rapid switching between the base fuel and the test fuel with stop watch to measure the time for complete evacuation of the glass burette from the fuel sample feeding the engine. A large air box fitted with an orifice plate is used for measuring air consumption rate using differential pressure transducer model Setra 239 having differential pressure range of 0-12.7 cm water column with accuracy of 1%. The exhaust gases temperature is measured using calibrated K type thermocouple with accuracy of ± 0.5%. Ambient air temperature, intake air temperature inside the intake air manifold is measured using calibrated K type thermocouple probes. The cylinder pressure is measured using piezoelectric pressure transducer model Kistler 6123, 0-200 bar as pressure rang with sensitivity of 16.5 pc/bar and accuracy of 1.118 %. A slotted disk is fitted to the end of the crankshaft and an optical sensor for measuring engine speed and crankshaft angle position. The signals from the pressure transducers, optical sensors, and thermocouples are digitised and recorded in IBM compatible PC with the help of Lab View software for later analysis using a data acquisition card model CIO-DAS1602/12, 12 bit, 32 channel single ended 16 differentials. The signal of cylinder pressure is acquired for every 0.1^0 CA and the acquisition process covered 50 completed cycles, the average value of these 50 cycles being outputted the pressure data used for calculation of combustion parameters. The brake mean effect pressure (BMEP) is calculated from engine power, speed and engine geometry specification. Sample of combustion gases are analyzed by gas analyzer model HORIBA to measure UHC, CO, CO_2, and O_2. A stainless steel water cooled sample probe of 10-mm o.d is used to draw the gas sample. The concentrations are measured on dry base and UHC is measured as propane (C_3H_8). The outgoing signals of the cells are manipulated and digitized by a built in A/D converter. Digital readouts of UHC, CO, CO_2 and O_2 are available through the analyzer screen.

Part of the exhaust gas is recirculated (EGR) to engine inlet manifold. EGR is accomplished by direct link between the exhaust and the intake surge tank. Therefore, the exhaust back pressure is maintained slightly above the intake manifold pressure such that EGR could occur. The quantity of this EGR is to be measured and controlled accurately; hence a by-pass for the exhaust gases is provided along with the manually controlled EGR valve in the circulation line between the exhaust gases manifold and the intake surge tank. The exhaust gases come out of the engine during the exhaust stroke at high pressure. It is pulsating in nature. It is desirable to remove these pulses in order to make the volumetric flow rate measurements of the recirculating gases possible. For this purpose, EGR is directed to air box line before entering engine cylinder to improve mixing process with fresh air. An orifice plate is used for measuring EGR flow rate using differential pressure transducer model Setra 239 having differential pressure range of 0-12.7 cm water column with accuracy of 1%. The detailed schematic line drawing of the experimental EGR system is shown in Fig.(1). Thermocouples are provided at the intake manifold, exhaust manifold and along the EGR route. This work studies the interaction resulting from the application and control of EGR rate and its effectiveness on emissions as well as engine performance parameters.

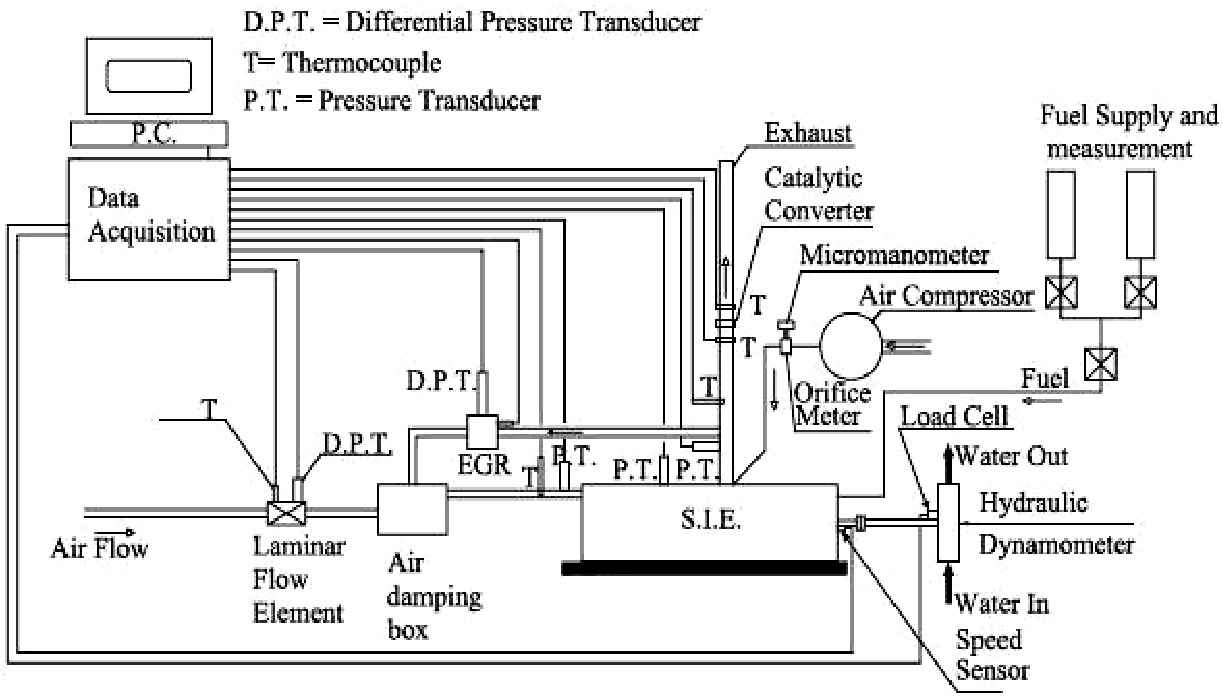

Fig. (1) Experimental Setup

Three ways catalyst converter installed in the exhaust manifold to study its effects on UHC and CO concentrations. Two thermocouples K type are installed before and after the catalyst for measuring exhaust gases temperatures. UHC and CO concentrations are measured to compare with the base case. Also, air injected adjusting to exhaust valves to study its effects on UHC and CO concentrations. In this case, pipe of the exhaust manifold is insulated to prevent heat losses to increase oxidation rate of UHC and CO species to CO_2 and H_2O.

Exhaust gases temperature is measured after air injection at the same position for compression with the base case. Air compressor is used as a source for air injection. Air injection flow rate is measured using orifice plate. Air temperature is measured using K type thermocouple and the pressure difference across the orifice plate is measured using digital micromanometers model MP5KD.

3. RESULTS AND DISCUSSIONS

3.1 Engine Performance

Brake power, brake specific fuel consumption (BSFC) and brake thermal efficiency versus engine speed for different engine loads are shown in Fig.(2). For different engine loads, brake power increases with the increase of engine speed up to 3000 rpm. After that, brake power decreases with the increase of engine speed due to increase of engine friction. For difference engine load, the difference in brake power increases with the increase of engine speed. Maximum brake power is 24 kW at 3000 rpm at engine full load and equivalence ratio (Φ) =1.2 [from Fig.(7), at AFR= 12.3 where the theoretical AFR is equal 14.8). For different engine speeds, BSFC increases with the increase of engine load due to increase mass of fuel entering engine. At engine speed of 2400 rpm, BSFC has minimum values and increase before and after this speed for different engine loads. Where maximum combustion rate and the shortest combustion duration occur at a mixture slightly richer than the stoichiometric (Φ=1.15) and the maximum heat input is at a mixture 20% richer than stoichiometric mixture i.e. at Φ =1.2. So, at 20 % load the minimum BSFC is occurred at Φ =1.16 while at 100% load minimum BSFC is occurred at Φ =1.2. The mixture with Φ =1.16 to 1.2 for all engine loads range is the mixture at which the mean effective pressure and maximum effective power are obtained, and these values decrease as the mixture becomes leaner.

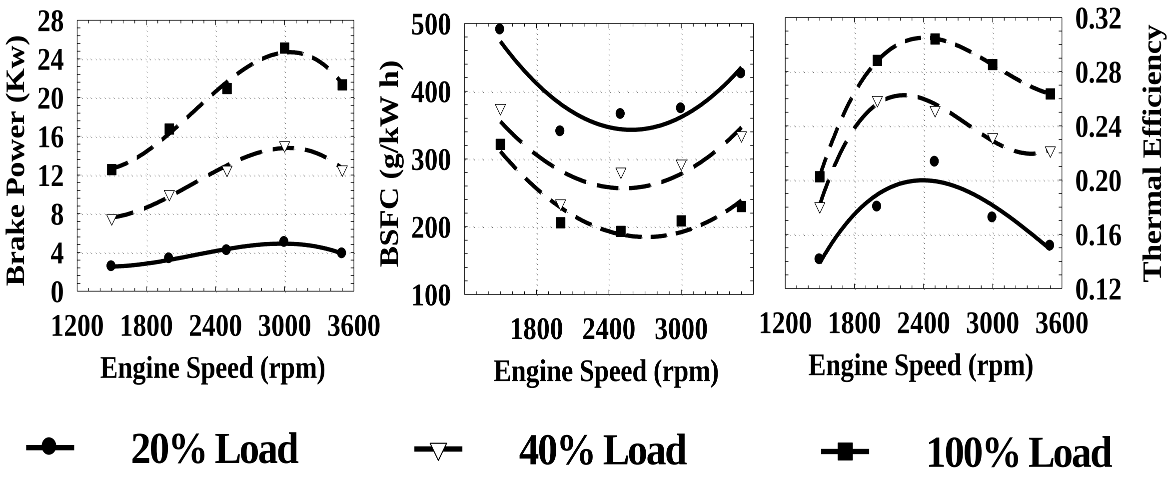

Fig.(2) Brake Power, BSFC and Brake Thermal Efficiency for Different Engine Loads

Engine produces its maximum power at engine speed where the power increase provided by the frequency of cycles is completely balanced by the decrease in torque. In this situation the cylinders take in a maximum amount of mixture per second. However, the engine power decreases at higher speeds because the increase in engine speed can not compensate the decrease in torque. The maximum power decreases with a reduction of 7% when Φ decreased from 1.2 to 0.9 at 3300 rpm. Thus, engine speed affects the effective power positively whereas the increase of Φ affects negatively. Mean effective pressure has a maximum value at a specific engine speed corresponding to maximum brake power and it decreases at higher speeds due to a reduction in volumetric efficiency. According to this result, it was observed that BMEP decrease from 8.8 bar to 7 bar with a reduction of 12.5% when Φ is decreased from 1.2 to 0.9 (2400 rpm). For different engine loads, brake thermal efficiency increases with the increase of engine speed up to 2400 rpm which cross bonding to minimum BSFC. After that it is decreased due to increase engine friction power losses. Thermal efficiency increases as the mixture becomes richer because the mixture properties approach to those of ideal gases. At engine speed of 2400 rpm, maximum thermal efficiencies are 30% for 100% load and Φ =1.2 and 20% for 20% load and Φ =1.17 (minimum specific fuel consumption speed) respectively. At 100% load, as engine speed increases from 2400 rpm to 3500 rpm, the maximum brake thermal efficiency decreases from 30% to 25% (as 17% decreases). At 20 load, as engine speed increases from 2400 rpm to 3500 rpm, the maximum brake thermal efficiency decreases from 20% to 15% (as 25% decreases). Besides, specific fuel consumption represents the opposite of the variation in volumetric efficiency by engine speed. BSFC decreases by decreasing Φ for all engine loads. According to this result, the minimum SFC decreases from 360 g/kW-h to 190 g/kW-h with a reduction of 16.4% when engine load decreased from 100% to 20% and Φ decreased from 1.2 to 1.17.

Brake thermal efficiency and BSFC versus engine speed for different EGR at 40 % of engine load are shown in Fig.(3). BSFC increases with the increase of EGR for different engine speeds due to need more fuel to over come dilution effects of incoming charge, increase specific heat of exhaust gases, decreasing rate of heat release and rate of reactions of different species. AFR for optimum fuel consumption at a given load depends on mixture preparation quality. It also varies for a given chamber design with the part of throttle load and speed range (Heywood, 1988). It is clearly from Fig. (3) and Fig.(7) (AFR variation) that the higher fuel is consumed at higher speeds and AFR less than stoichiometric (AFR less than 14) due to the greater friction losses that can occur at high speeds. It is easy to perceive from the figures that the decrease of BSFC with the decrease of engine speed and the increase of AFR. However, the required minimum BSFC occurs within a range of AFR from 12 to 14 for the selected range of engine speeds with different EGR rates. At very lean conditions, higher BSFC can be noticed. After AFR of 14.5, BSFC goes up rapidly, especially for high speed. At very lean conditions with AFR greater than 14.5, BSFC of 390 g/kW-h is observed for the speed of 1500 rpm while 380 g/kW-h for speed of 3500 rpm. BSFC at speed of 1500 rpm is observed to be equal at speed of 3500 rpm. Because of very lean operation conditions can lead to unstable combustion and more lost power due to a reduction in the volumetric heating value of the air/gasoline mixture. This behavior can be more clarified by referring to Fig.(3) and Fig.(7)(AFR variation), where the brake efficiency reduced considerably at very lean operation conditions.

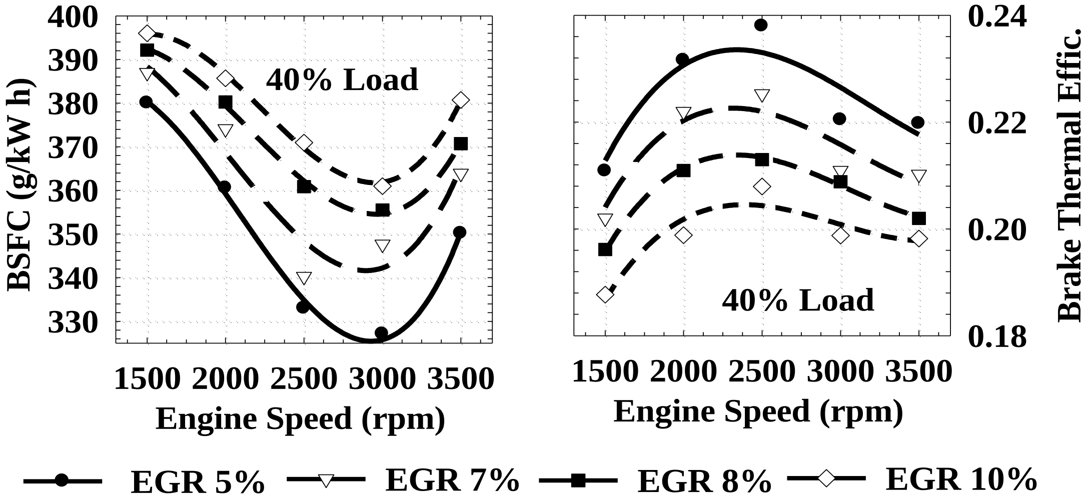

Fig. (3) BSFC and Brake Thermal Efficiency for Different EGR with 40% Load

Brake thermal efficiency decreases with the increase of EGR for different engine speeds due to decrease brake power and increase BSFC. Thermal efficiency tends to decrease slightly; this may be due to the fact that the amount of fresh oxygen available for combustion decreases due to replacement by exhaust gases. The presence of inert molecules reduces the temperature and increases the detonation tolerance and may leads to considerable losses in brake thermal efficiency. Engine thermal efficiency tends to decrease with EGR as a result of decreasing indicated work and increasing pumping work. Combustion deterioration is predominant at low speed conditions due to a low AFR with the increase of EGR. Decreasing AFR, combustion deterioration and efficiency losses are largely attributed to increase pumping work with the increase of EGR. The brake power (useful part) is a percentage from the intake fuel energy. The fuel energy are also covered to the friction losses and heat losses (heat loss to surroundings, exhaust enthalpy and coolant load). The brake thermal efficiency increases nearby the slightly lean condition (AFR = 15.5) and then decreases with the decrease of AFR and the increase of engine speed as shown in Fig.(7) (AFR variation). The operation within a range of AFR from 15.5 to 12 (Φ = 0.95 to 1.2) gives the values of brake thermal efficiency for all rang of engine speeds. At minimum EGR, maximum brake thermal efficiency is 23 % at speed 2500 rpm comparing to 22% at speed 3500 rpm. Clearly, engine speed and AFR have major effects on engine performance with different EGR.

Engine performance indicated parameters are calculated using the measured cylinder pressure and engine geometry at different engine speeds. Indicated mean effect pressure (IMEP), pumping mean effect pressure (PMEP), indicated power (IP), indicated specific fuel consumption (ISFC), indicated thermal efficiency (η_{ith}), volumetric efficiency (η_V), mechanical efficiency (η_m)} are calculated using the measured cylinder pressure versus engine crank angle at different operating conditions. The model equations are found in Heywood (1988). The calculated engine performance parameters are shown in Figs.(4& 5).

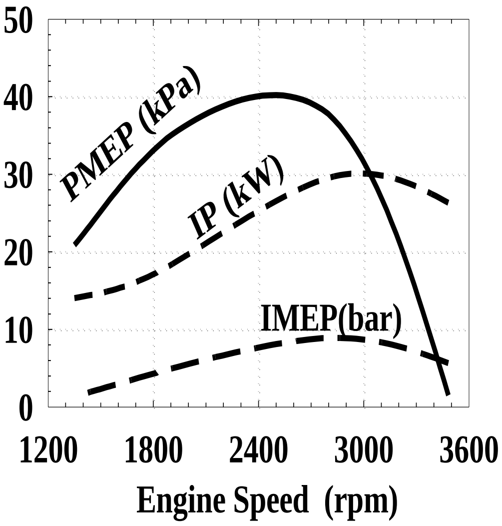

Fig.(4) IMEP, IP and PMEP at 100% Load

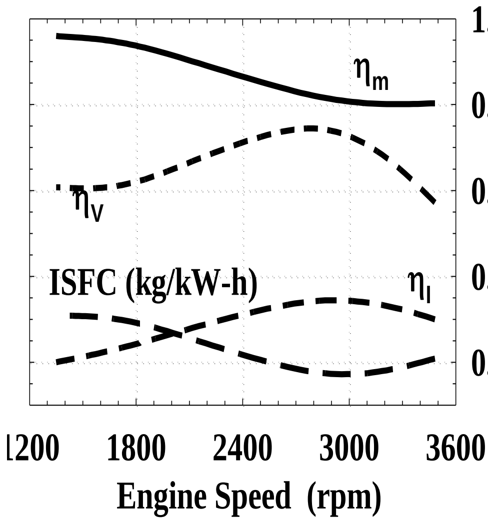

Fig.(5) η_m, η_V η_{Ith} and ISFC at 100% Load

All engine performance parameters are compatible with each other. The maximum volumetric efficiency is equal to 75% at engine speed of 2500 rpm. The highest volumetric efficiency is obtained with the valve timing gives a relatively small back flow into the intake manifold. High speed leads to high volumetric efficiency because of the high speed gives high vacuum at the intake port and consequent large air flow rate that goes inside the cylinder. Further increase in engine speed leads toward the maximum value of η_V. At engine speed higher than 2500 rpm, the flow into the engine during at least part of the intake process becomes chocked. Once this occurs, further increase in speed do not increase the flow rate significantly so volumetric efficiency decreases sharply. This sharp decrease happens because of high speed is accompanied by some phenomenon that have negative influence on η_V. These phenomenons include the charge heating in the manifold and high friction flow losses which increases as the square of engine speed. In general, the functional dependence on the volumetric efficiency with respect to engine speed will be similar to the corresponding dependencies of IP, η_{ith} and IMEP. The maximum mechanical efficiency is equal 95 % at engine speed less than 1800 rpm and steadily decreases to 80% at engine speed higher than 2500 rpm. This result of the mechanical efficiency reflects the increasing importance of the friction as square of engine speed increases. The indicated thermal efficiency varies between 20 and 35%. The brake thermal efficiency decreases rapidly for increasing engine speed due to increase mechanical friction. The indicated specific fuel consumption and indicated thermal efficiency curves are inverses for each other, and show maximum fuel efficiency for engine speed range of 2500 to 3000 rpm. The maximum value of IMEP, is 9 bar at engine speed of 2600 rpm. IMEP is a good parameter for comparing engines with regard to design due to its independent on the engine size and speed. If torque used for engine comparison, a large engine is always seem to be better when considering the torque, however, speeds become very important when considered the power (Pulkrabek, 2003). As engine speed increases, IMEP increases with the decrease of AFR. IMEP falls with a non-linear behavior from slightly rich condition where AFR is 14 to the richest condition where the AFR is 11.5. The difference in IMEP increases with the increase of engine speed and AFR. IMEP decreases from 9 bar at engine speed of 2600 rpm to 2 bar at speed of 1500 rpm for the same range of AFR. This implied that, the engine gives the maximum power (IMEP = 9 bar) at speed of 2600 rpm comparing to the minimum power (IMEP =2 bar) at speed of 1500 rpm. Due to dissociation at high temperatures following combustion, molecular oxygen is present in the burned gases under stoichiometric conditions. Thus some additional fuel can be added and partially burned. This increases the temperature and the number of moles of the burned gases in the cylinder and consequently increases IP and IMEP. The maximum value of IP is 30 kW at engine speed of 3000 rpm. The minimum value of ISFC is 190 g/kW-h. Pumping mean effect pressure (PMEP) at different engine speeds are also calculated. PMEP increases with the increase of engine speed to maximum value at 2300 to 2600 rpm then decreases with the increase of engine speed. All these results are consistent with reported results for similar automotive engines and are within the acceptable range. This means that, the using measuring system is accurate and suitable for obtaining many other data.

Air/fuel ratio (AFR) versus engine speeds for different engine loads is shown in Fig.(6). AFR decreases with the increase of engine speed for different engine loads due to increase amount of fuel conducting to engine cylinder resulting in high pressure different around fuel jet at throttle position. The increase in the area of inlet manifold at throttle position is not liner with throttle angle. So, the increase of throttle angle contributes in increasing vacuum beyond the throttle gate which increases fuel mass flow rate than the increase of air mass flow rate which decreases AFR with the increase of engine speed. Also, AFR decreases with the increase of engine load due to increase fuel mass flow rate required to increase heat release rate to over coming engine load.

AFR versus engine speeds for different EGR is shown in Fig.(7). AFR decreases with the increase of EGR due to increase the dilution effects. When EGR displaced a unit of fresh air with an equal unit of burned exhaust products, it is not only altering AFR, but also causes a dilution effects. By reducing the oxygen concentration, it expects to increase the ignition delay and reduces the burning rate. Other effects, such as delayed maximum cylinder pressure and impact ignition delay. Decreased temperature and reduced AFR seem to offset the dilution effect on ignition delay, hence the ignition delay increases with the increase of EGR. As EGR increases, the boost pressure and the peak pressure increase as well resulting in high pressure. Consequently, negative side effects of the increase EGR on indicated efficiency appears closely related to the overall AFR. In other words when the engine operated with overall low AFR, relative changes of mixture composition due to EGR have impact effect. Careful optimization of system parameters in order to minimize fuel economy penalties associated with the application of EGR. Combustion deterioration is the predominant reason for efficiency losses under low speed mid load conditions where relatively low boost pressure levels might lead to critically low AFR. For conditions characterized by higher overall AFR, most of the fuel economy deterioration can be attributed to the increase in pumping work.

AFR versus engine speeds for different air injection in exhaust manifold is shown in Fig.(8). For air injection, AFR decreases with the increase of engine speed as in the two cases of engine loads and EGR. For the two cases of engine loads and engine EGR, AFR traces are concave on contrast in the case of air injection AFR traces is convex. AFR is varied from lean (AFR= 16) based on mass where the equivalence ratio = 0.925 to rich (AFR=14.3) where the equivalence ratio = 1.034 and engine speed varied from 1500 rpm to 3500 rpm.

The percentage of change in AFR for different engine loads, EGR and air injection for engine speed range of 1500 to 3500 rpm is shown in Fig.(9). The percentage of reduction in AFR for engine speed range of 1500 to 3500 rpm are

17.5%, 18.8%, 19%, 15.5 % for different percentages of engine loads of 20%, 40%, 60%, 100%. The percentage of reduction in AFR for engine speed range of 1500 to 3500 rpm are 18%, 19%, 22% and 25% for different percentages of EGR of 5%, 7%, 8%, 10 %. The percentage of reduction in AFR for engine speed range of 1500 to 3500 rpm are 13%, 12%, 11% and 10% for different percentages of air injection of 3%, 4%, 5%, 6 %. From comparison the change in AFR for different three cases it can be concluded that the effect of EGR on ARF is higher than engine load and air injection cases.

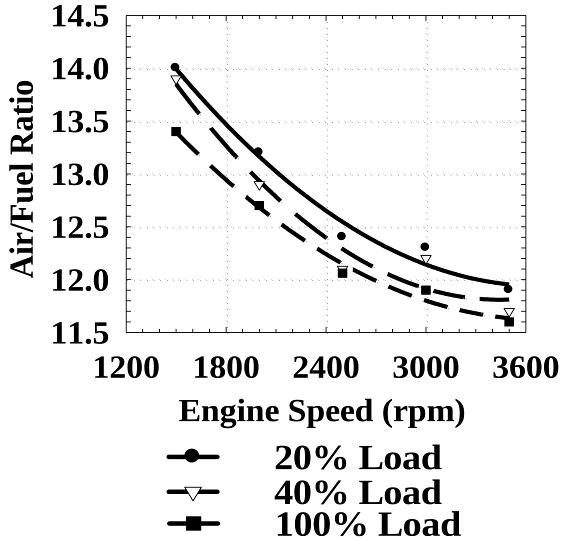

Fig. (6) Air/Fuel Ratio for Different Engine Loads

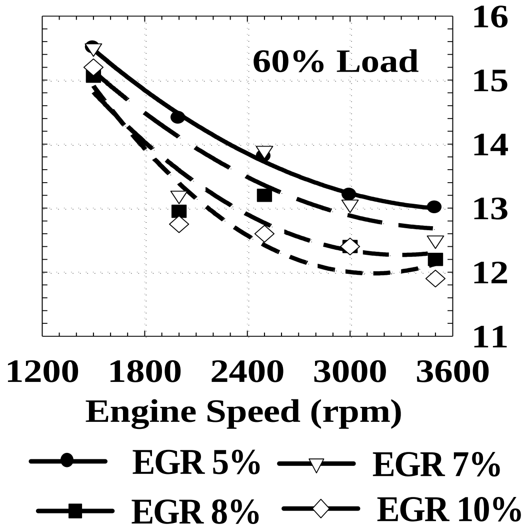

Fig. (7) Air/Fuel Ratio for Different EGR

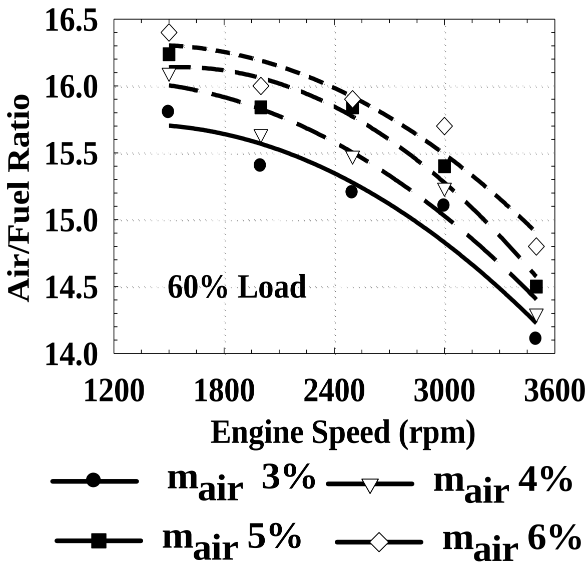

Fig. (8) Air/Fuel Ratio for Different Air Injection at 60% Load

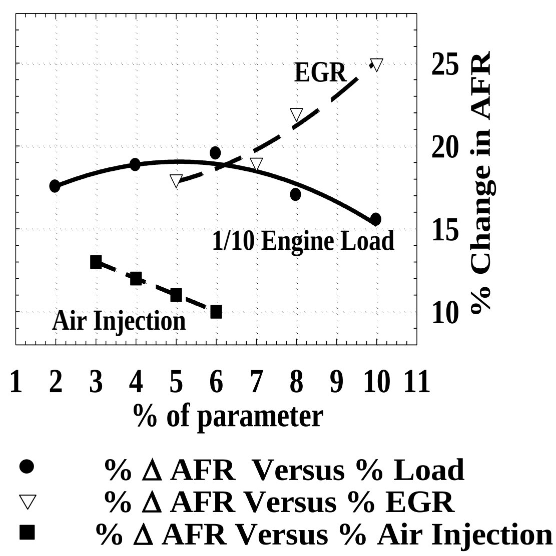

Fig. (9) Effect of Load, EGR and Air Injection on Percentage Change of AFR

$T_{exhaust}$ versus engine speed at different engine loads is shown in Fig.(10). $T_{exhaust}$ increases with the increase of engine speed and engine load due to increase mass of fuel conducting to engine cylinders, increase turbulence intensity, increase heat release rate, and increase maximum flame temperature. $T_{exhaust}$ affects by engine load than engine speed due to engine load increases by the increase amount of fuel burning and heat release rate but engine speed can be increased by decreasing engine load.

$T_{exhaust}$ versus engine speed for different percentage of EGR is shown in Fig.(11). $T_{exhaust}$ increases with the increase of EGR due to increase inlet mixture temperature. As EGR increases, oxygen concentration decreases, start of combustion delays, duration of combustion increases, fuel burning velocity decreases, heat release rate decreases. So for the same mass of fuel entering engine cylinders, complete of combustion occurs in exhaust process and in exhaust manifold which increases exhaust gases temperature. For certain engine speed with and without EGR at the same engine load AFR decreases with the increase of EGR which means increasing mass of fuel burning and consequently $T_{exhaust}$ increases with the increase of EGR. EGR has high effect on $T_{exhaust}$ especially around maximum engine power due to maximum flame temperature.

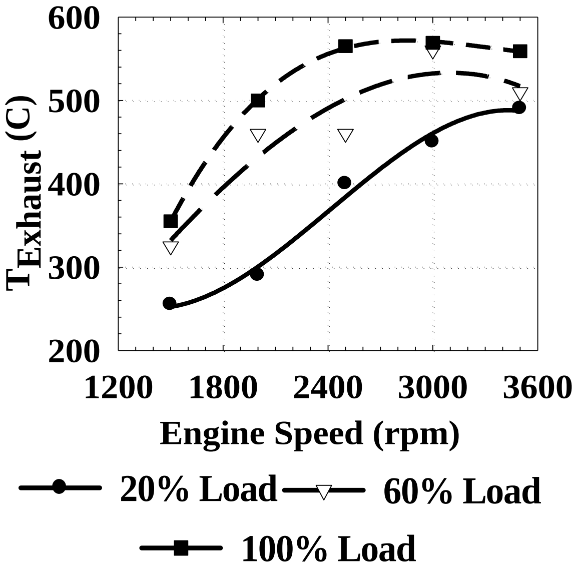

Fig.(10) $T_{Exhaust}$ for Different Loads

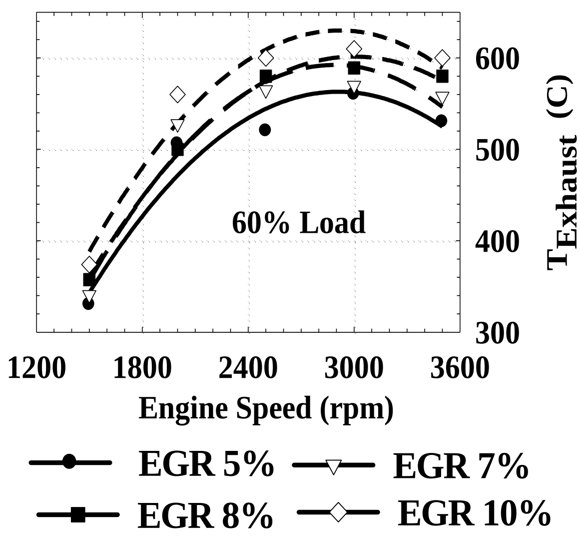

Fig.(11) $T_{Exhaust}$ for Different EGR with 60% Engine Load

$T_{exhaust}$ before and after catalyst converter versus engine torque at different engine speeds is shown in Fig.(12). There are some of heat generated in the exhaust manifold due to some of reactions occur on the surface of catalyst converter to oxidize CO, UHC to CO_2 and H_2O. Consequently, at different engine speeds $T_{exhaust}$ after catalyst is higher than before catalyst. The converter has high surface area and so high heat losses which contributes in decreasing $T_{exhaust}$ but the heat generated due to reaction dissipates this effect. Also, $T_{exhaust}$ before and after catalyst increased with the increase of engine speed.

$T_{exhaust}$ versus engine speed at different percentage of air injection is shown in Fig.(13). $T_{exhaust}$ decreases with the increase of the percentage of air injection due to exhaust gases is lean mixture and the degree of lean increases with the increase of air injection. Also, there is some reactions occur in the exhaust manifold which generates heat contributing in increasing gases temperature and decreasing the effect of increase AFR in the lean mixture. Form comparison $T_{exhaust}$ for the case of air injection with the case without air injection in the exhaust manifold (first case), $T_{exhaust}$ increases in the case of air injection than in case without air injection due to insulation of exhaust pipe system.

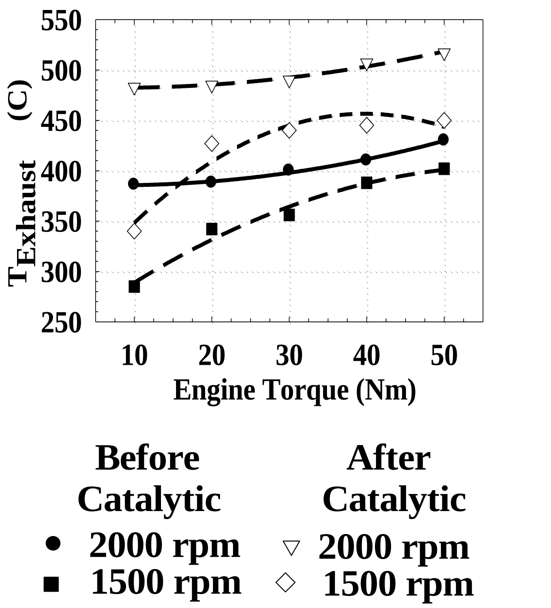

Fig.(12) $T_{Exhaust}$ Before and After Catalytic Converter for Different Engine Speeds

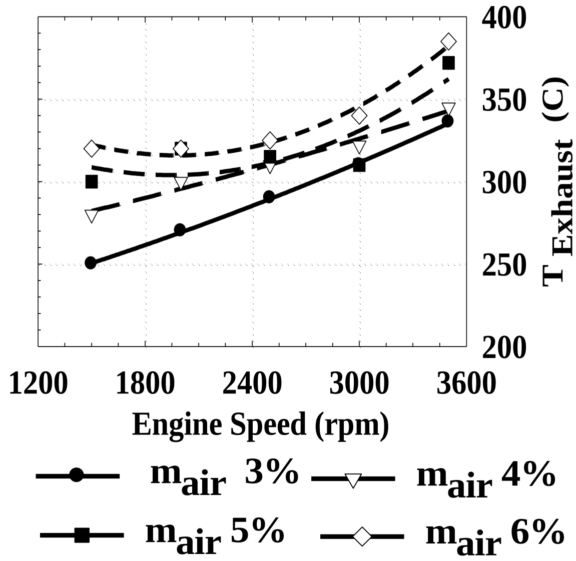

Fig.(13)$T_{Exhaust}$ for Different Percentages of Air Injection at 60 % Load

3.2 Engine Emissions

CO and UHC concentrations are measured for four different cases. The first case is the measuring of CO and UHC concentrations versus BMEP at different engine speeds as shown in Fig.(14). From the figure it can be seen that, as BMEP and engine speed increase, UHC and CO concentrations decrease due to increase turbulence intensity, mixing process of burnt and unburnt gases, combustion efficiency, gases temperature which increases oxidation rate of UHC and CO to CO_2.and H_2O. At low engine speed CO and UHC concentrations are affected by increasing BMEP and engine speed than at high engine speed. UHC concentrations are affected by BMEP and engine speed than CO.

The second case is the measuring CO and UHC concentrations versus engine speed at different EGR at equal engine torque of 30 Nm as shown in Fig.(15). Using EGR increases CO and UHC concentrations due to decrease O_2 concentration in the fresh charge which decreases rate of different reactions. Rate of increasing CO and UHC concentrations at high EGR is greater than that at low EGR due to decrease flame temperature and mixture became richer. Also, EGR is more effect at high engine speeds than at low engine speeds where the traces of CO and UHC are diversion with the increase of engine speed.

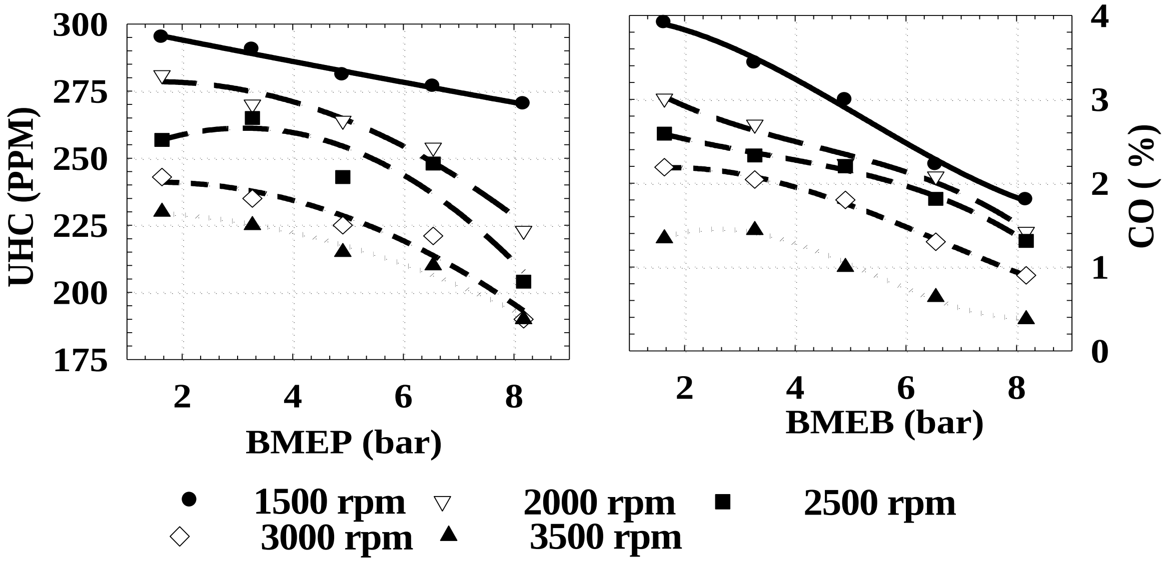

Fig.(14) UHC and CO Concentrations for Different Engine speeds

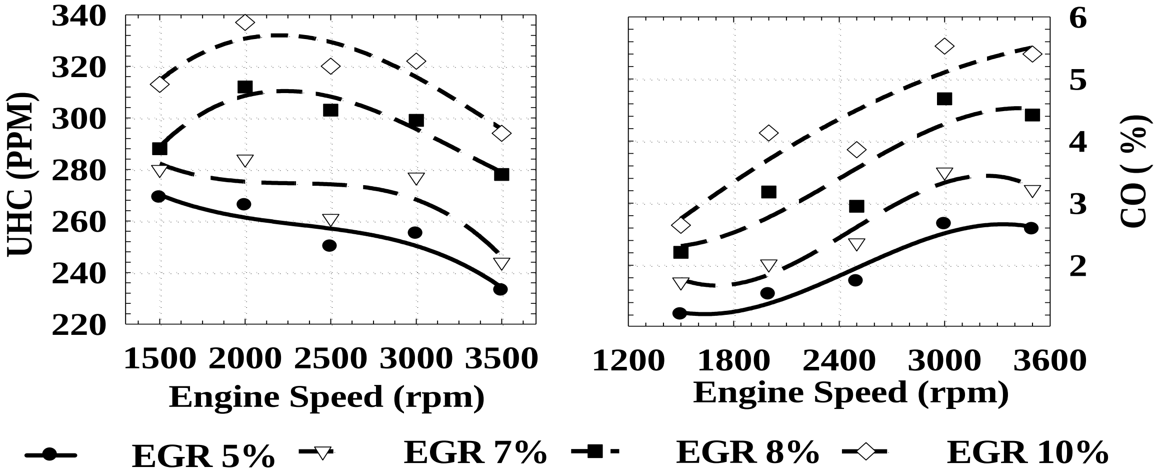

Fig.(15) UHC and CO Concentrations for Different EGR at Torque 30 Nm

According to the dilution theory, the effect of EGR is caused by increasing amount of inert gases in the mixture, which reduces the adiabatic flame temperature. At high loads, it is difficult to employ EGR due to deterioration in combustion. At low loads, UHC contained in the EGR would possibly re-burn in the mixture, leading to lower unburnt fuel in the exhaust and thus improved brake thermal efficiency. Apart from this, hot EGR would raise the intake charge temperature, thereby influencing combustion and exhaust emissions. At mid load and low engine speed, as EGR rate increases the peak pressure, as well as the integrated area of the high pressure loop decreases significantly. In contrast, at high engine speed and mid load, as EGR rate increases the boost pressure and the peak pressure increase. Consequently, negative side effects of increased EGR on indicated efficiency appear closely related to the overall AFR. In other words when the engine operates with overall low AFR, relative changes of mixture composition due to EGR have impact. Combustion deterioration is the predominant reason for efficiency losses under low speed mid load conditions where relatively low boost pressure levels might lead to low AFR.

The third case is the measuring UHC and CO concentrations versus engine torque for different engine speeds using three ways catalyst converter as shown in the Fig.(16). The concentrations measured at the same time after and before catalyst converter for 1500 and 2000 rpm. CO and UHC concentrations before the catalyst are higher than after catalyst due to CO and UHC oxidized by converter to CO_2 and H_2O. The effectiveness of the catalyst agents used for these conflicting tasks requires operation of the engine very close to stoichiometric conditions.

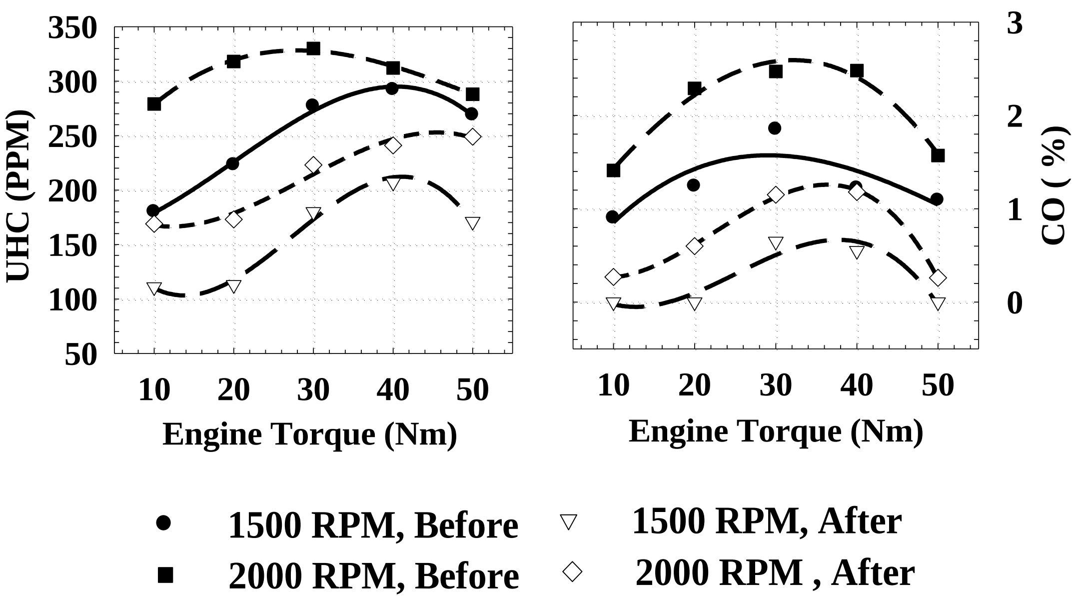

Fig.(16) UHC and CO Concentrations Versus Engine Torque Before and After Catalyst for Different Speeds

Any deviations away from stoichiometric result in rapid decreases in the conversion efficiency of at least one pollutant. Additionally, the catalyst reaction of the precious metal agents in the catalyst converter is highly temperature dependent. The catalyst is considered ineffective below approximately 250 ºC. At stable operating temperature and stoichiometric conditions, the catalyst converter generally achieves conversion efficiencies in excess of 90%.

The fourth case is the measuring UHC and CO concentration versus engine brake power at different air injection in the exhaust manifold adjusting to exhaust valves at equal torque of 30 Nm is shown in Fig.(17). The use of secondary air injection is currently under research as a means to decrease the time for the catalyst converters to light off which is the temperature at which 50% of exhaust emissions can be effectively converted.

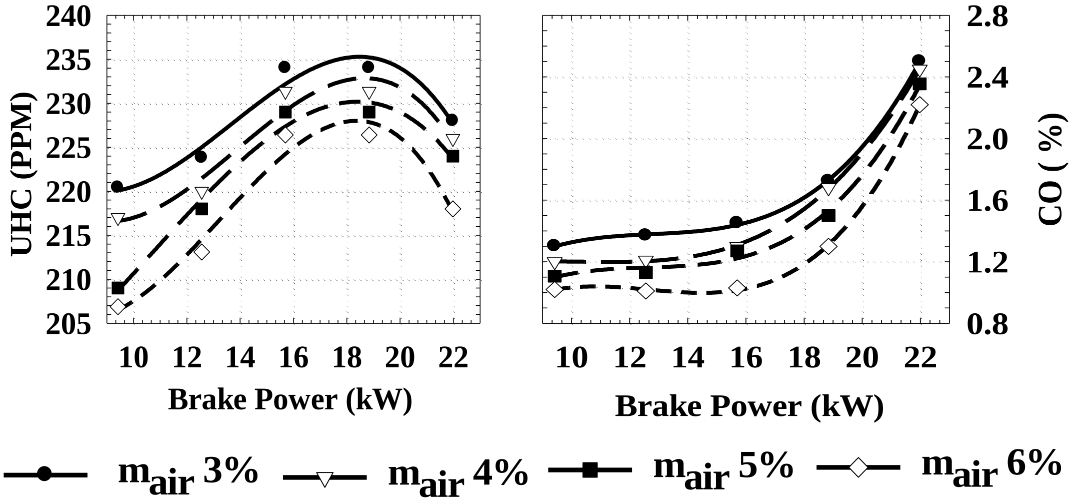

Fig.(17) UHC and CO Concentrations for Different Air Injection at equal Torque of 30 Nm

If air is introduced into a fuel rich environment such as the one present in the exhaust manifold during a cold start, the high concentrations of UHC and CO can be oxidized and converted into harmless derivatives, such as CO_2 and H_2O. The values of UHC and CO concentrations are less than in the case of without air injection due to concentrations is diluted by the fresh air. CO concentration increases with the increase of engine speed due to the exhaust leaner. CO trace in the case of air injection is contracting to CO trace in the case without air injection with the increase of engine speed because the exhaust in the case of air injection is very leaner but in the case without air injection near stoichiometric conditions. UHC concentration is more affected by air injection than CO concentrations. The results show that the reduction in UHC concentrations is 7-10% due to oxidation in the exhaust port. These variables that cause substantial changes in gases temperature, port residence time, or oxygen concentration are most significant. A set of experiments with air injection into the exhaust, at the cylinder exit plane, showed that substantial additional oxidation of UHC can be obtained with injection into the exhaust gases before significant cooling occur. When air injected into exhaust manifold any UHC will ignite. This combustion will not produce any power, but it will reduce excessive UHC emissions. Unlike in the combustion chamber, this combustion is uncontrolled, so if the UHC concentrations in exhaust are excessive, explosions sound waves like popping will occur. Since all of this is done after the combustion process is completed, this is one emission control that has no effect on engine performance. From discussion the three cases for measuring emissions, UHC and CO concentrations decrease with using catalyst converter and air injection on contrast UHC and CO concentrations increase with the increase of EGR. Using air injection is a simple method for decreasing UHC and CO concentrations.

Sound pressure level (SPL) generated from cylinder pressure, inlet and exhaust pressure measurements for different engine speeds are shown in Fig. (18 &19). SPL generated from cylinder pressure, inlet and exhaust manifolds pressure increase with the increase of engine speed due to increase pressure, heat release rates and turbulence intensity in engine cylinder. SPL generated from inlet and exhaust manifolds also increase with the increase of engine speed due to increase flow velocity and pressure difference in inlet and exhaust manifolds. SPL generated in inlet and exhaust manifolds is mainly due to pressure waves generated by opening and closing valves.

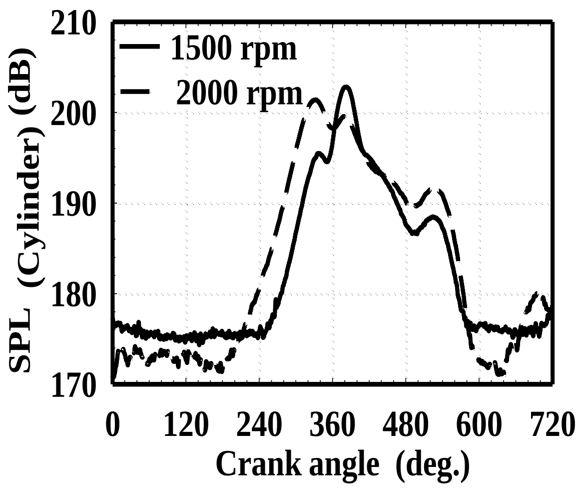

Fig. (18) SPL Generated from Cylinder Pressure at Different Speeds

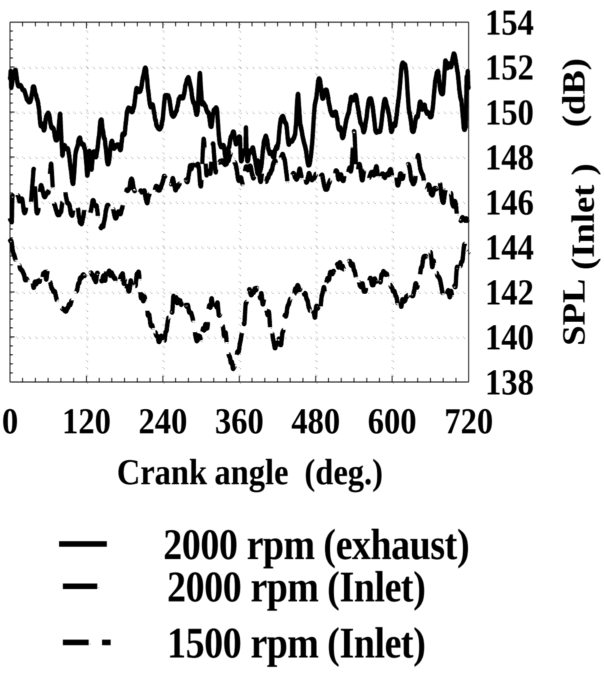

Fig. (19) SPL Generated from Inlet and Exhaust Manifolds at Different Speeds and 50% Load

SPL generated from exhaust manifold is higher than that generated from inlet manifold due to higher back pressure. SPL generated from cylinder pressure is higher than that from exhaust and inlet manifold due to higher fluctuation resulting in different rates of inter mediate reactions, different rates of fuel burning, pressure waves generated with different process inside engine cylinder due to piston motion, fluctuations in heat release rate, mixing process of burnt and unburnt gases, compression and expansion of gases inside the cylinder due to piston motion. Position of maximum SPL is advanced BTDC as engine speed increases due to start of combustion occurs earlier with the increase of engine speed. Traces of SPL generated from cylinder pressure have two noses; the first is related to position of maximum heat release and maximum in cylinder pressure; the second is related to exhaust valve opening due to high exhaust flow velocity resulting in more expansion of exhaust gases in the exhaust manifold. Trace of SPL generated from exhaust manifold is more fluctuated than from inlet system and in cylinder pressure due to many reactions occur in exhaust manifold.

3.3 Engine Cycle to Cycle Variation

Cylinder pressures cycle to cycle variations (CCV) for two different speeds are shown in Figs.(20 & 21). Cylinder pressures CCV are mainly occurred during early stage of combustion and around peak pressure. Cyclic variations occur from the time of spark to the establishment of a fully developed flame. This is the period in which the developing flame kernel is susceptible to factors that cause CCV. The arrival time of the flame front at the first ionization gap from the spark gap varied greatly from cycle-to-cycle.

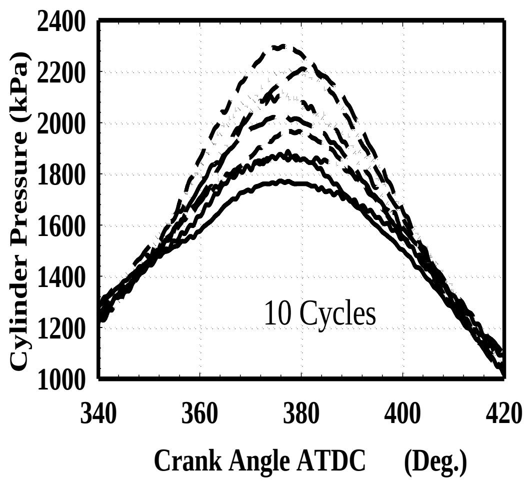

Fig.(20) Cycle to Cycle Variations of Cylinder Pressure at Speed of 1500 rpm and 60% Load

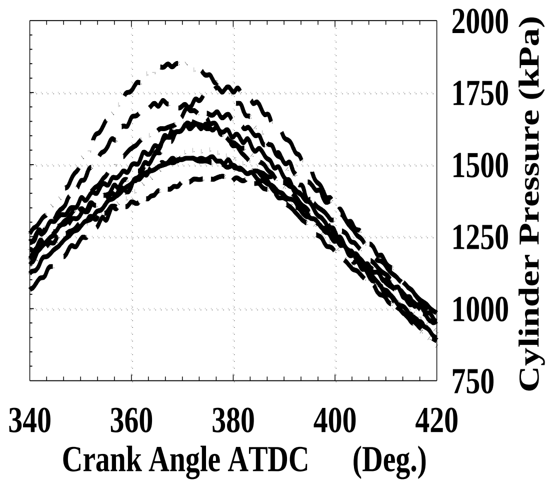

Fig.(21) Cycle to Cycle Variations of Cylinder Pressure at Speed of 2000 rpm and 60% Load

Cyclic variations of the turbulence level are a major cause for cycle-to-cycle variations in combustion where movies of turbulent flow in the combustion chamber do not follow any pattern. The randomness of the flow causes velocity gradients within the flow field. The cyclic variation of the velocity near the spark plug at the time of ignition contributes to the cyclic variations in combustion. The velocity gradient during the early flame development smears the kernel and increases the surface area. This causes the combustion rate to increase because of the increase of flame area. Variation in duration of combustion, peak cylinder pressure, crank angle position of peak pressure and indicated mean effective pressure all these parameters are contributed in cylinder pressure CCV. Incomplete mixing of fuel, air and residual contribute also to CCV in combustion. The increase of residuals resulted in increase cyclic pressure variability. Engine speed also contributes to CCV in combustion, where increasing the engine speed resulted in an increase in flame speed and cyclic flame speed variations. Cyclic turbulence variation attributes to engine speed variation and the high turbulence variation cause high variation in flame speed. The variation of CCV of cylinder pressure increases with the increase of engine speed due to increase of turbulence intensity and decrease of flame stability. The period of combustion process (as crank angle) increases with the increase of engine speed where period of combustion for 2000 rpm is longer than that for 1500 rpm due to decrease duration of combustion process and increase turbulence intensity. Cylinder pressure CCV is varied from 13% to 15% for different engine speeds which is very high comparing to CCV from inlet pressure.

Inlet pressure cycle to cycle variations (CCV) for two different speeds are shown in Figs.(22 & 23). Inlet pressure CCV is due to pressure wave pulsations coming from opening and closing of inlet valves. Fluctuation in flow velocities, engine speed, instantions piston speed, flow densities, are main sources for CCV for inlet pressures. Pulsations of inlet manifold pressure, air mass flow rate, and AFR are main sources for CCV of inlet manifold pressure. Also, throttle angle with the nonlinear behavior of the throttle gate and the periodic motions of the intake valves and the suck cycle of the engine which produce strong changes in the air mass flow and pressure waves in the inlet manifold are contributed in increasing CCV. Also, the amplitudes of the pulsations are not constant where they vary with engine load and speed. Moreover over lap period between inlet valve and exhaust valves are also contributed in CCV for inlet pressure and cylinder pressure. Inlet pressure CCV increases with the increase of engine speed due to increase the difference between depression and peak pressures of the pulsating pressure waves of inlet pressure. Inlet pressure CCV is varied from 0.3 to 0.6 % for different engine speeds which is very low comparing to CCV from cylinder pressure. The periodic motions of the intake valves and the suck cycle of the engine produce strong changes in the air mass flow and pressure waves in the manifold and consequently inlet pressure CCV. By optimizing the geometry of the manifold the pressure resonance points can be tuned to increase the volumetric efficiency of the engine at low speed. The amplitudes of the pulsations are not constant. They vary with load and speed of the engine and so, inlet pressure CCV change with engine load and speed.

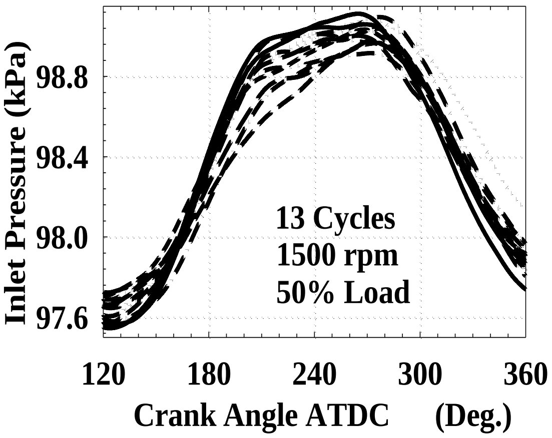

Fig. (22) Cycle to Cycle Variation for Inlet Pressure at 1500 RPM

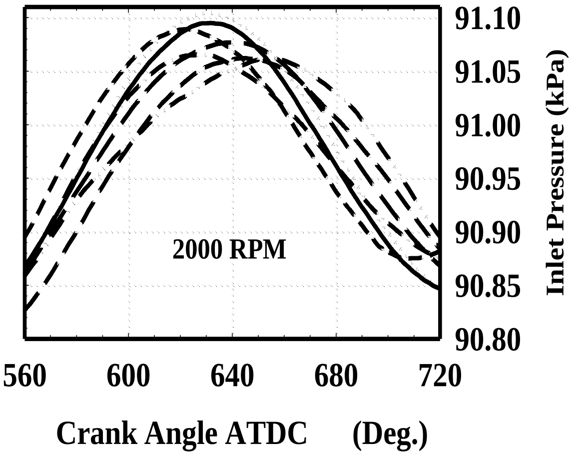

Fig. (23) Cycle to Cycle Variation for Inlet Pressure at 2000 RPM

Cycle to cycle variation (CCV) in engine speed for two mean different engine speeds of 1500 rpm and 2000 rpm are shown in the Fig.(24). The variation in engine speed is consider the main deriving force for CCV due to its effects on combustion characteristics, flow characteristics, performance parameters characteristics and engine emissions characteristics. Engine speed CCV increases with the increase of engine speed due to increase the variation of AFR, fuel burning rate, heat release rate, turbulence intensity, mean effective pressure, volumetric efficiency and engine cylinder pressure. To operate gasoline engines at part load the mixture formation has to full several requirements.

The complexity of this process requires, regarding suitable mixture transportation, vaporization of the fuel, an adjustable design of the combustion chamber and the intake ports to reliably place an ignitable mixture at ignition timing near the spark plug at any speed and load. Due to the inhomogeneous mixture distribution during different operation, the first combustion period is very sensitive to CCV. A reproducible mixture movement with high kinetic energy is necessary for stable engine operation with low fluctuations in the combustion process. Because of the high relevance of these facts, the effects of engine speed, an adjustable air guiding system in the inlet manifold on in-cylinder flow, ignition, and combustion process are main parameters affecting on engine speed CCV. Engine speed CCV is a measure of the irregularity of the angular velocity of the crank shaft which is caused by the variation in energy release from CCV as well as cylinder to cylinder variation. The variation of mean effective pressure causes torque changes and resulting angular speed changes of the crank shaft. The engine speed CCV corresponding to change of the mean angular acceleration between successive crankshaft rotations, it is approximately preoperational to the change of the mean torque or mean effective pressure during one rotation of the crank shaft. The crank shaft speed increases AFR which increases the engine speed CCV. The maximum variation of engine speed CCV is varied from 1.6 to 2.4 for engine speed varied from 1500 to 2000 rpm

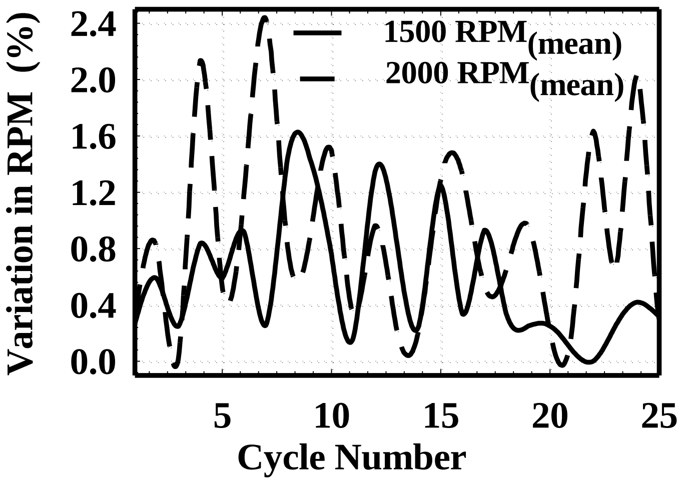

Fig. (24) Cycle to Cycle Variation of Engine Speed at Full Wide opening

4- CONCLUSTIONS

1. The use of EGR in spark ignition engine is promising method for improving part load operation conditions. BSFC, UHC, CO concentrations and $T_{exhaust}$ increase with the increase of EGR. On contrast brake power, break thermal efficiency and AFR decrease with the increase of EGR. EGR improves combustion qualities by increasing the inlet charge temperature and UHC and CO re-burned with using EGR. Decreasing AFR, combustion deterioration and efficiency losses are largely attributed to increase pumping work with the increase of EGR.
2. Catalyst converter installed in exhaust manifold provides significant reduction in UHC and CO concentrations on contrast $T_{exhaust}$ increases after catalyst than before catalyst due to some of heat release with oxidation of UHC and CO species into CO_2 and H_2O.

3. Air injection in exhaust manifold is the simplest method for reducing UHC and CO concentrations due to increase oxygen concentration after exhaust valve opening which is used to oxidize UHC and CO to CO_2 and H_2O at high exhaust temperature. $T_{exhaust}$ decreases with the increase of mass of air injection due to increase AFR to very lean conditions which over come heat release effects with air injection.
4. Sound pressures level (SPL) increases with the increase of engine speed and load. Sound pressures level generated from combustion process is higher than SPL generated from flow process where SPL calculated from cylinder pressure higher than that calculated from inlet manifold or exhaust manifold.
5. Engine cycle to cycle variation (CCV) is found to increase with the increase of engine speed due to increase the variation of AFR, fuel burning rate, heat release rate, turbulence intensity, mean effective pressure, volumetric efficiency, and engine cylinder pressure.

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NOMENCLATURES

AFR	air/fuel ratio on mass base
ATDC	after top dead center
BDC	bottom dead center
BMEP	brake mean effect pressure, bar
BSFC	brake specific fuel consumption, g/kW-hr
CA	crank angle, degree
CCV	cycle to cycle variation
COV_{imep}	coefficient of variance
EGR	exhaust gas recirculation, %
IP	indicated power, kW
IMEP	indicated thermal efficiency, kPa
SPL	sound pressure level, dB
$T_{exhaust}$	exhaust gases temperature, ºC
PMEP	pump mean effect pressure, kPa
TDC	top dead center

Greek letters

θ	crank angle, (deg.)
Φ	equivalence ratio
η_V	volumetric efficiency
η_{bth}	brake thermal efficiency
η_{ith}	indicated thermal efficiency
η_m	mechanical efficiency