Effect of Exhaust Gas Recirculation on the Combustion of an LPG Diesel Dual Fuel Engine

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Abstract: **In this study, the effects of EGR on the combustion characteristics of a dual fuel engine are investigated. Tests were conducted on a single cylinder diesel engine with a compression ratio of 15. LPG is supplied during the suction stroke while diesel pilot fuel is injected for the purpose of initiation of combustion near the end of compression stroke. The power output of the engine is normally controlled by changing the amount of primary gaseous fuel added to the intake manifold. Diesel is injected directly into the cylinder at an optimum injector needle pressure of 150 bars. Pilot fuel quantity, intake air temperature and Exhaust Gas Recirculation (EGR) are the important operating parameters controlling the combustion process in a dual fuel engine.**

The main objective of the present study was to maintain high fuel efficiency at part load conditions. Experiments were conducted on an LPG diesel dual fuel engine. Intake air temperature as well as pilot fuel quantity was maintained at their optimum values. Whereas the flow of exhaust gases with an intake air gas mixture was varied. Ignition delay period, rate of pressure rise, combustion duration, heat release patterns and relative cycle efficiency have been presented at light and high load. Increased heat release rates have been observed in pre-mixed combustion phase with EGR at higher loads. Due to rapid combustion near top dead centre, the brake thermal efficiency was found to be higher with hot EGR. When the EGR percentage in suction becomes very high, the delay period and combustion duration increase substantially, resulting in poorer brake thermal efficiency. Higher rate of pressure rise is observed with hot EGR. It has been concluded that at higher loads EGR cannot be used to control the knock. Total duration of combustion was found longer with both hot and cold EGR.

Keywords: ***Exhaust Gas Recirculation (EGR), LPG, Dual Fuel Engine, Combustion***

I. INTRODUCTION

In a dual fuel engine, two fuels are used simultaneously. The primary fuel that is usually gaseous forms the major content of the total energy supplied to the engine. Gaseous fuels such as Natural gas, Producer gas, Liquefied Petroleum Gas (LPG), Hydrogen (H_2), etc. are promising alternative fuels. Liquid fuels like Methanol, Ethanol, and Gasoline etc.; are also being used in the fumigated form as the primary fuel and secondary fuel or pilot fuel (generally diesel) is used to initiate combustion process. Near the end of the compression stroke, pilot fuel is injected to initiate the combustion. Most of the advantages of Compression Ignition (C.I.) engines can be achieved when they work on the dual fuel mode. The main problem associated with dual fuel engines are poor utilization of the inducted fuel at light loads and loss of combustion control at high loads that is commonly termed as the onset of knock.

Karim [1, 2] has reported that in dual fuel engines at low loads when gaseous fuel concentration is low, ignition delay period of the pilot fuel increases and most of the homogeneously dispersed gaseous fuel remains unburned and results in poor performance. The pilot fuel quantity and intake temperature is the most important variables controlling the performance of dual fuel engines at light loads [3]. In dual fuel engines most probably a gaseous fuel air mixture nearby the pilot fuel spray auto ignites resulting into knock. At higher loads and higher intake temperatures, increased admission of gaseous fuel results in uncontrolled combustion leading to knock [4].

The resort of exhaust gas recirculation in a dual fuel engine without much cooling can help in improving light load performance through increased initial charge temperature and seeding with active products [5, 6]. Hannu [7] used EGR in natural gas fueled S. I. engine and concluded that up to 8% EGR, NOx emission reduces. The addition of the higher percentage of EGR resulted in higher hydrocarbon emissions and decreased exhaust temperatures.

A.S. Ramdas et al. [8] describe experimental investigation of performance and emission characteristics of single cylinder, indirect injection diesel engine fueled with biodiesel-diesel blend and LPG in dual fuel mode operation with exhaust gas recirculation (EGR). Recirculation of exhaust gases into the engine decreases the combustion chamber temperatures whereas marginal improvement in thermal efficiency was observed with EGR.

Bose and Banerjee [9] studied the suitability of different EGR techniques in a single-cylinder four-stroke diesel engine running on hydrogen and diesel in dual fuel mode under different EGR conditions both qualitatively and quantitatively, were examined in reducing NOx concentrations. The NOx level decreased from 1211 ppm to 710 for hydrogen enrichment (0.15 kg/hr) at 80% of the rated load under 20% cooled EGR.

Eiji Tomita et al. [10] studied the effect of EGR on combustion and exhaust emissions in supercharged dual fuel natural gas engine ignited with diesel fuel. They have observed two-stage combustion under the condition of low EGR rate and advanced injection timing. The indicated mean effective pressure and thermal efficiency increased, during two-stage combustion; however, NO_x emissions also increased by a large amount compared to normal combustion. High indicated mean effective pressure, high thermal efficiency, and low NO_x emissions were achieved just before the occurrence of two-stage combustion by changing the EGR rate and injection timing of diesel pilot fuel.

N. K. Miller et al. [11] conducted experiments on a LPG fueled diesel engine using diethyl ether with exhaust gas recirculation. Experimental results showed that by EGR technique, at part loads the brake thermal efficiency increases by about 2.5% and at full load, NO concentration could be considerably reduced to about 68% as compared to LPG operation without EGR. However, the higher EGR percentage affects the combustion rate and significant reduction in peak pressure at maximum load.

Saleh [12] studied the effect of variation in LPG composition on emissions and performance in a dual fuel diesel engine. From the results, it is observed that the exhaust emissions and fuel conversion efficiency of the dual fuel engine are found to be affected when different LPG composition is used as a higher butane content lead to lower NOx levels while higher propane content reduces CO levels. Also, tests were conducted at part loads to improve the engine performances and exhaust emissions by using the Exhaust Gas Recirculation (EGR) method.

Mahala S.K. [13] studied the effect of EGR on the performance and emission characteristics of natural gas fuelled diesel engine and concluded that the application of EGR reduces substantially NOx, CO and smoke. Results indicate that the performance parameters are comparable to the baseline diesel operation.

V. Pirouzpanah [14] made an attempt to investigate the combustion phenomenon at part load and using exhaust gas recirculation (EGR) to improve the part load performance of dual fuel engines. The results of this work show that each of the different cases of EGR (thermal, chemical and radical cases) has an important role in the combustion process in dual fuel engines at part loads.

In the present work, combustion characteristics of an LPG diesel dual fuel engine with different percentage of EGR are analyzed. The effect of EGR on different combustion parameters like delay period, peak cylinder pressure, rate of pressure rise, combustion duration, heat release rate, mass fraction burnt and relative cycle efficiency was measured and analyzed. With the optimum injection timing of 27.4º CA BTDC, the change in ignition delay with the addition of different quantities of EGR at optimum intake temperature, injector opening pressure and pilot fuel quantity was investigated at light loads as well as at high loads.

II. APPARATUS AND EXPERIMENTAL PROCEDURE

The schematic layout of the test setup used is indicated in **Fig. 1**. A single cylinder, direct injection, water cooled diesel engine was used for this experimental work. The specifications of the engine are given in **Appendix [A]**

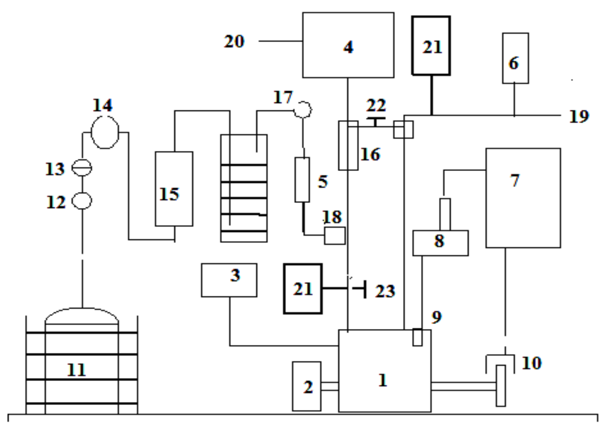

Figure 1. Experimental Setup

1-Engine, 2-Dynamometer, 3-Diesel tank and Measurement system, 4-Air flow Surge tank and Meter, 5-Air Pre Heater, 6-HC/CO Analyzer, 7-PC Based Data Acquisition system, 8-Charge Amplifier, 9-Pressure Pickup, 10-Shaft Position

Encoder, 11-LPG cylinder in a Constant Temperature water Bath, 12-Control valve, 13-Pressure Regulator, 14-Positive Displacement Gas Flow Meter, 15-Rotometer, 16-Flame Trap, 17-Control Valve, 18-Gas Mixer, 19-Exhaust Outlet, 20-Air Inlet, 21-Temperature and Pressure Measurement Points, 22-EGR Valve, 23-Throttle Valve

An LPG connection was made on the intake manifold. Governor varied pilot diesel flow while the LPG flow rate was varied manually. Cylinder pressure signals, obtained from a flush mounted quartz pressure pickup were recorded on a personal computer. The heat release rate and other combustion parameters were computed using software developed for the purpose after obtaining the pressure signals 100 consecutive engine cycles. Based on a previous experimental program injection timing of 27.4º BTDC and 24º BTDC were selected for dual fuel and diesel operation respectively. Based on the same work, injection pressure of 150 bars and 200 bars were used up to 80% load and beyond that respectively in the dual fuel mode. The optimum pilot fuel quantities and intake temperatures set for the experimentation at different loads are given in **Table 1**.

Table 1. Optimum Engine Operating Conditions

Load %	Intake Temp. (º C)	Pilot Fuel Quantity (mg/cycle)	Injector Needle Lift Pressure (bar)	% LPG Substitution	% Optimum EGR by Volume
20	70	8.4	150	25.0	18
40	70	10.7	150	36.0	10
60	60	10.7	150	46.0	8
80	60	5.9	150	71.0	6
100	40	7.1	150	70.0	4
100*	40	7.1	200	71.023	4

*200 bar Injector Needle Lift Pressure

III. EXHAUST GAS RECIRCULATION (EGR)

Exhaust gas was extracted from the exhaust manifold very near to the exhaust valve. A single pass parallel flow heat exchanger using cooling water circulation was used to reduce the temperature of the exhaust gas. A regulating valve controlled the flow of exhaust gas. The temperature of the air gas and EGR mixture was controlled by using an electrical heater. Gas flow meter was used to measure the flow of re-circulated gas. Pressure and temperature values of EGR were measured before the gas flow meter. Blower was used to supply the EGR to engine during suction stroke.

A simple one-dimensional heat release scheme similar to one suggested by Hayes and Savage [15] was developed to process experimentally obtained pressure crank angle data. The estimate of the initial mass in the cylinder was made with the help of the complete cycle simulation program. In the present study both mass fractions burnt and heat release rate as a function of crank angle were calculated for diesel and dual fuel engines for average pressure crank angle diagram (100 cycles). The simple method assumes that the mass fraction burnt is in proportion to the fraction of the total pressure rise due to combustion. The same program was also used for calculating the heat release rate of an individual cycle.

Woschni's relation was used to calculate the gas side heat transfer coefficient. The combustion wall chamber was assumed to be 450º C for calculating the heat transfer rate. The heat transfer area was calculated by considering the bowl in the piston.

Relative cycle efficiency is defined as the ratio of brake thermal efficiency with the heat added at any particular crank angle in comparison to this if it would have been added ideally at TDC. The relative cycle efficiency can be used to compare the effects of heat release patterns on the engine performance. The relative cycle efficiency can help us to correlate the heat release rate diagram with the expected thermal efficiency of the engine.

The relative cycle efficiency is expressed as;

$$RCE_i = \frac{\{CR^{k-1}\}}{CR^{k-1} - 1} \qquad (1)$$

Where,

K = Polytrophic index of compression & expansion process

V = Cylinder volume in any particular crank angle, cm^3

V_0 = Cylinder volume at TDC, cm^3

CR = Compression ratio

The overall relative efficiency of all the heat added is the weighted average of the relative efficiency of each amount of heat added at its corresponding positions. Practically different amounts of heat are added at different crank angles in the engine and in that case the overall relative cycle efficiency is [16];

$$RCE = \sum \frac{\{h_i (RCE_i)\}}{\sum h_i} \quad (2)$$

h_i = heat added J/^{o}CA

During present investigation, equation (2) has been used to calculate the relative cycle efficiency of diesel and LPG + diesel dual fuel engine at different engine operating conditions.

IV. RESULTS AND DISCUSSION

The analysis of engine pressure crank angle diagrams is very important in the study of dual fuel engines because of the complex nature of combustion processes in these engines. A convenient way of depicting the combustion process is by the heat release rate and mass fraction burnt diagram of fuel as a function of crank angle. During present study heat release diagrams have been obtained from experimental crank angle diagrams with optimum EGR at the crank angle step of 1^0 CA.

It has been established during the present study shown in **Fig. 2** at different intake temperatures and optimum pilot, that the combustion in the dual fuel mode is composed of three stages namely;

I stage: Pre-mixed burning of part or whole of the pilot quantity plus small part of gas entrained in the spray.

II stage: Auto ignition of gas air mixture in the close vicinity of pilot spray and diffusive burning of the remaining pilot.

III stage: Burning of gas air mixture by flame propagation initiated from spray zone.

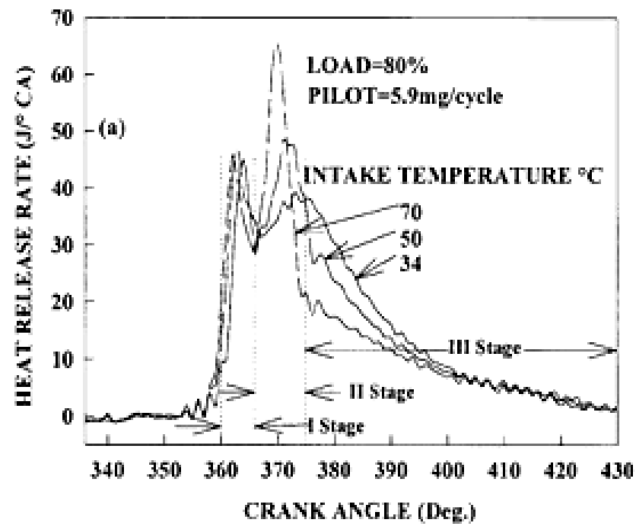

Figure 1. Combustion Stages in LPG + Diesel Dual Fuel Engine

A. *Effect of EGR (Cold and Hot)*

When the temperature of the intake air, LPG and EGR mixture were maintained at optimum conditions as given in **Table 1**, the EGR was termed as **hot EGR** whereas when the mixture was supplied at ambient intake temperature, it was called **cold EGR**. The main objective of studying the **hot EGR** was to improve its effect on engine fuel economy particularly at low outputs whereas **cold EGR** was expected to suppress the rate of pressure rise and hence knock at higher outputs.

Heat Release Analysis (Hot EGR)

As shown in the **Fig. 3**, the heat release rate was not significantly influenced by hot EGR (70 ˚C) in magnitude and phase at 20% load. However, the second phase of heat release slightly increases. This is probably due to the presence of some active species from previous cycle supplied through EGR. Presence of EGR at 20% load does not affect the total combustion duration.

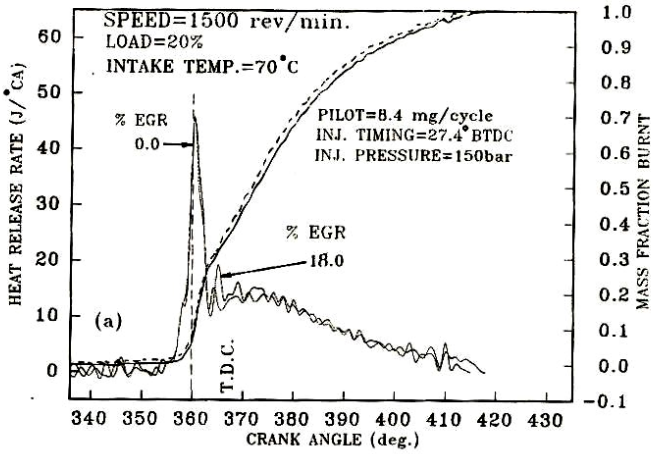

Figure 2. Effect of Hot EGR on Heat Release Rate and Mass Fraction Burnt

The effect of EGR on combustion at 80% load was found to be different as compared to 20% load as indicated in **Fig. 4**. The rate of heat release reveals three stages of combustion at 80% load and using the pilot fuel quantity of 5.9 mg/cycle. In the first stage (pre mixed stage), mainly the diesel burning takes place with a very small amount of gas available in the pilot fuel spray zone. The heat release rate in the first phase may reach up to 45 J/˚CA. The first stage lasts until 5-6˚ CA after TDC. Total heat release during the first phase of combustion is approximately 15-20% of the total energy supplied which is around 50% of the energy supplied by pilot fuel. The end of the first stage of combustion is followed by a dip in the rate of heat release rate. With EGR, heat release rate increases in premixed combustion phase. EGR also increases the peak pressure of the cycle. The remaining 50% pilot combustion burns in diffusion controlled mode in the second stage of combustion.

In the second stage combustion was different as compared to that without EGR. The rate of heat release was higher in this stage and after 50% burning of the total fresh charge, combustion shifts to the third phase. During the second stage, the homogeneous gas air mixture near the pilot fuel spray is consumed very rapidly and approximately 50% heat release in the next 6-7 ˚CA is noticed.

During the third phase when heat is mainly released by flame propagation, the heat release rate becomes lower due to slower combustion and total combustion duration is found to be longer. Due to rapid combustion near top dead centre, the brake thermal efficiency was found to be higher with hot EGR. When the EGR percentage becomes very high, the delay period and combustion duration increase substantially, resulting in poorer brake thermal efficiency. The highest rate of pressure rise is observed with hot EGR.

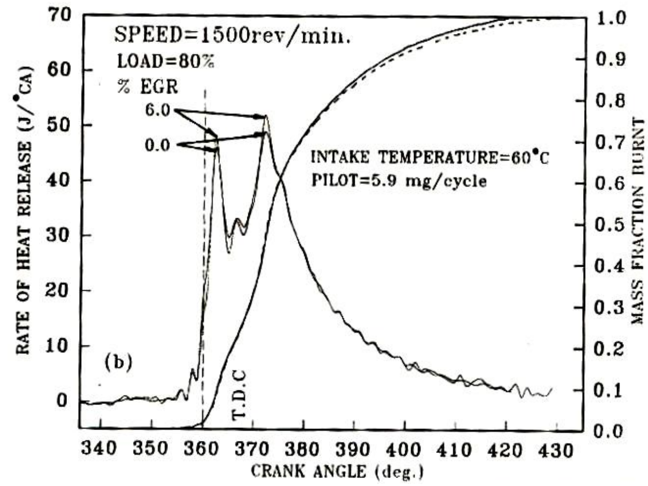

Figure 3. Effect of Hot EGR on Heat Release Rate and Mass Fraction Burnt

Fig. 5 indicates the mass fraction curve of three typical consecutive cycles at 80% load with 6% EGR. The curves indicate that significant change exists in the second and third phases of combustion in dual fuel mode.

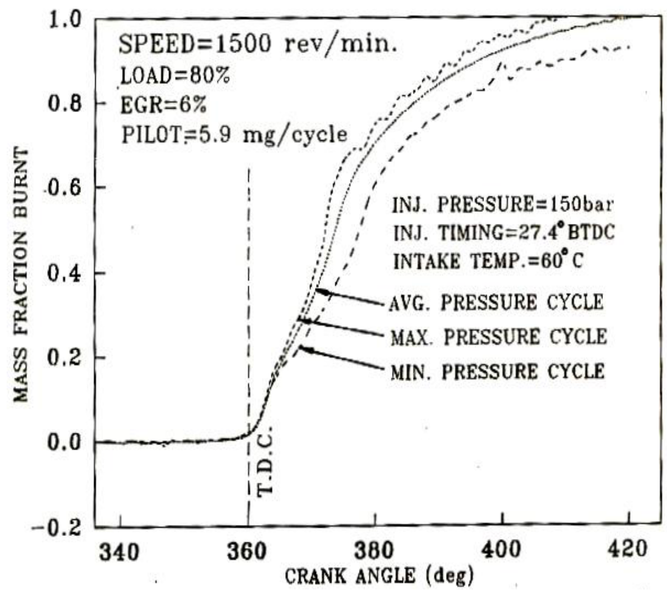

Figure 4. Crank Angle vs. Mass Fraction Burnt of Three Cycles

Heat Release Analysis (Cold EGR)

Effect of cold EGR on heat release rate and mass fraction burnt at 20 % load and 80% load is shown in **Figs. 6 & 7** respectively. At 20% load, the first phase of combustion is delayed. Mass fraction burnt in the initial stages of combustion is very low with cold EGR.

At 80% load and cold EGR using 5.9 mg/cycle pilot as shown in **Fig. 7**, combustion duration has been noticed to increase by 10º crank angle as compared to that without EGR. Rate of heat release in the first stage is higher due to increased ignition delay (approx. 2 ºCA) but in the second stage, the rate of heat release decrease. In the second stage due to the presence of relatively colder inert gases nearby pilot spray, the rate of heat release in this phase becomes very low and it merges with flame propagation phase. The decrease in heat release rate and longer combustion duration are noticed due to the slower flame propagation. However, higher peak pressures and higher rate of pressure rise are noticed with cold EGR at 20 and 80% load. With EGR (hot and cold) rate of pressure rise is higher and hence cold EGR cannot be used for the purpose of reducing the combustion knock at higher loads.

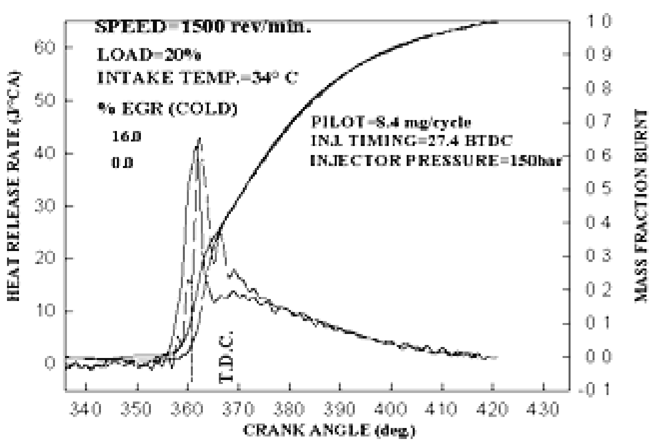

Figure 5. Effect of Cold EGR on Heat Release Rate and Mass Fraction Burnt

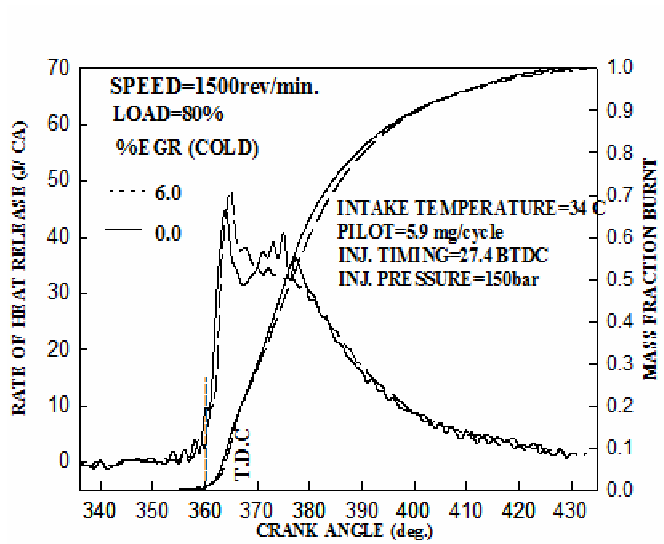

Figure 6. Effect of Cold EGR on Heat Release Rate and Mass Fraction Burnt

B. Peak Cycle Pressure

The peak cycle pressure was studied by observing the averaged cylinder pressure versus crank angle traces for hundred cycles. The ignition starting point was identified, as the point at which there is an abrupt change in the rate of pressure rise.

The effect of percent hot EGR on dual fuel engine peak cycle pressure was investigated at 20% and 80% load as shown in **Fig. 8**. The optimum pilot fuel quantity of 8.4 mg/cycle and optimum intake temperature of 70 ˚C was used at 150 bar injector opening pressure and 1500 rev/ min at 20% of full load. The peak cycle pressure, in the case of 20% load decreases continuously as percent EGR increase. Charge dilution due to the addition of EGR seems the major reason for a decrease in pressure.

At 80% load approximately 1 bar higher peak pressure is observed at optimum EGR. Better combustion near TDC seems to be the main reason for increase in peak cycle pressure at 80% load. The EGR at 80% load seems to affect the flame propagation

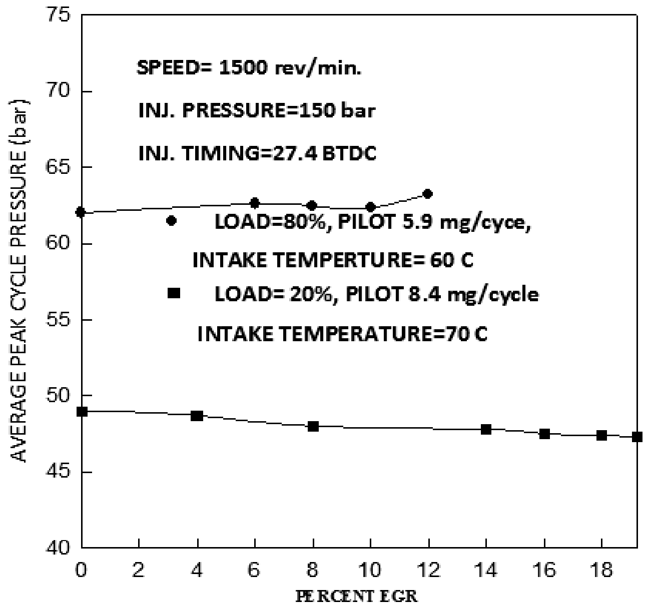

Figure 7. Effect of EGR on Peak Cycle Pressure

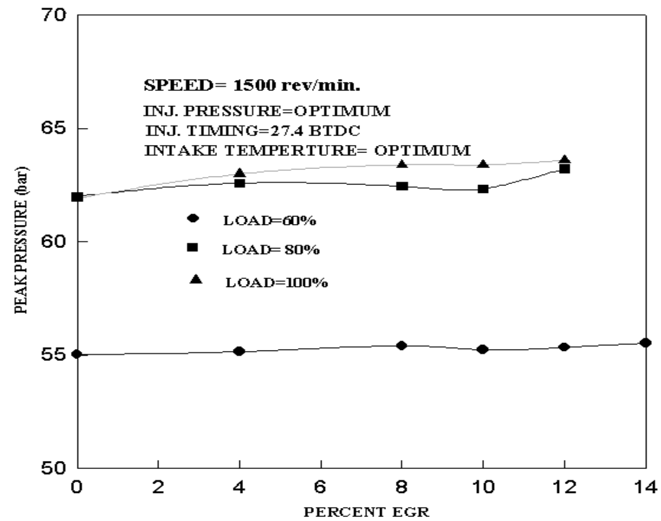

Figure 8. Effect of EGR on Peak Pressure

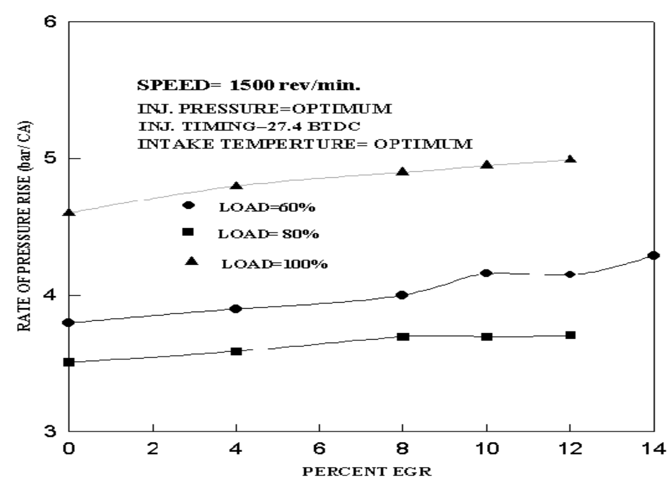

Figure 9. Effect of EGR on Rate of Pressure Rise

The effect of percentage EGR at different loads (60, 80 and 100%) on the peak pressure rise at optimum conditions is shown in **Fig. 9**. Only higher loads are considered here because knocking is likely to be significant at these conditions only. As percentage EGR increases, the peak cycle pressure increase at every load. Better combustion near TDC seems to be the main reason for increase in peak cycle pressure

Fig. 10 shows the effect of percentage EGR addition on rate of pressure rise at 60, 80, and 100% loads. The rate of pressure rise at 80% and 100% load was observed higher as compared to 60% load at every EGR percentage. Peak pressure and rate of pressure rise is higher at these loads and hence hot EGR cannot be used for the purpose of reducing the combustion knock at higher loads.

C. Ignition Delay

The variation of ignition delay period at 20% load and 80% load with different percentages of EGR is shown in **Fig. 11**. It is observed from the figure that delay period is not affected with EGR up to 18% at 20 % load. Further addition of EGR at this load increases the delay period. However at 80% load, the delay period has not been changed in the range of EGR investigated.

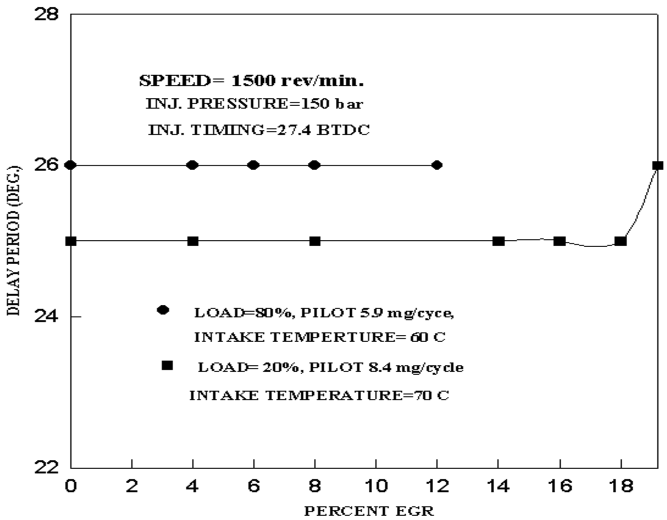

Figure 10. Effect of EGR on Delay Period

The effect of intake temperatures at low loads namely 20% and 40% load on the peak cycle pressure and delay period with and without optimum EGR are shown in **Figs. 12 & 13**. It was observed that at both 20% and 40% loads the peak cycle pressure at every intake temperature with optimum EGR were higher and ignition delay was longer as compared to operation without EGR. It has been observed that by decreasing the intake temperature, the ignition delay period increases by 1º and 2º crank angle at 20 and 40% load respectively. Due to increase in delay period, the peak cylinder pressure increases as intake temperature decreases.

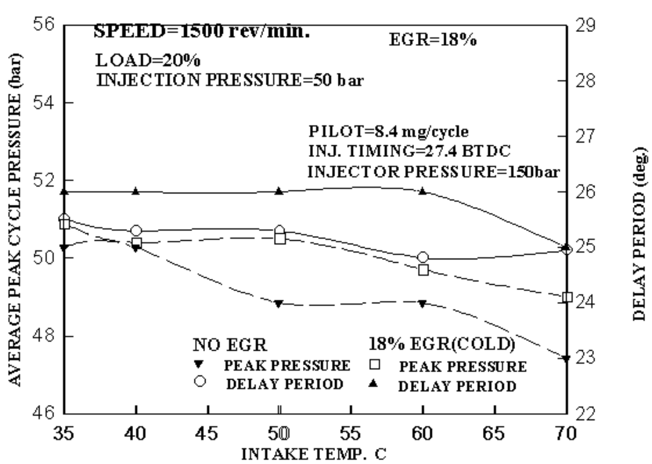

Figure 11. Effect of Intake Temperature on Peak Cycle Pressure and Delay Period

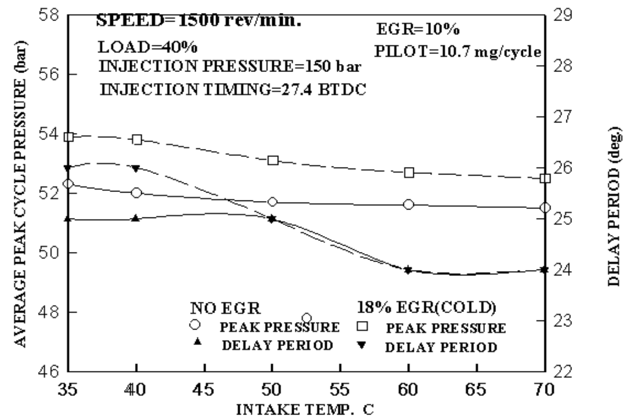

Figure 12. Effect of Intake Temperature on Peak Cycle Pressure and Delay Period

D. Relative Cycle Efficiency and Combustion Duration

The relative cycle efficiency and combustion duration are shown in **Figs 14 &15** at 20 and 80% load respectively. At 20% load, maximum substitution of diesel by LPG is only 30% and 70% heat is added by diesel only in the form of pilot fuel. Due to 70% heat input by diesel at 20% load, the maximum amount of heat is released near TDC after the completion of the delay period in premixes phase. The presence of EGR at 20% load also improves combustion in the second phase. It is observed that at 70 ˚C intake temperature, relative cycle efficiency increases up to 18% EGR. As the EGR increases, the combustion duration increases at low load. This may be due to the burning of the gas which otherwise does not take place at light loads.

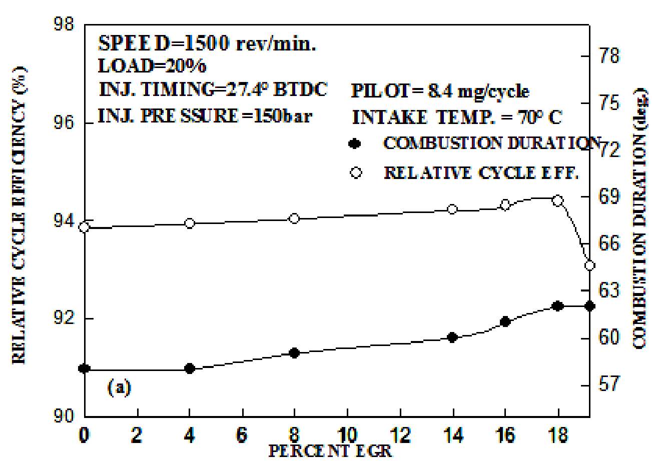

Figure 13. Effect of EGR on Relative Cycle Efficiency and Combustion Duration

Longer combustion duration with EGR addition is also observed at 80% load as shown in **Fig. 15**. The value of relative cycle efficiency also does not change much with low percentage of EGR but with higher substitution it decreases rapidly. The increase in combustion duration up to 12 ˚CA is observed with 12 % EGR that without EGR. With 6% EGR addition, the heat release near TDC in first and second phase is significantly higher than without EGR. This results in higher relative cycle efficiency. At 6% EGR, due to rapid heat release near TDC, brake thermal efficiency is also observed to be higher.

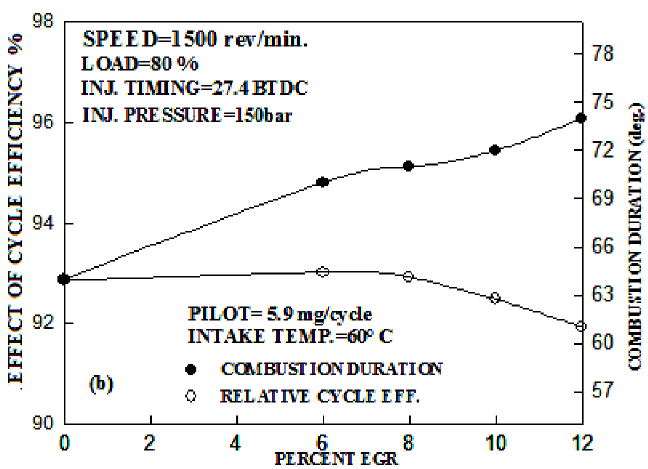

Figure 14. Effect of EGR on Relative Cycle Efficiency and Combustion Duration

V. CONCLUSIONS

Following salient conclusions have been drawn on the basis of present experimental investigations;

- It has been concluded that significant change exists in the second and third phases of combustion in the dual fuel mode due to the addition of EGR at 20% load.
- At 80% load and optimum hot EGR, the rate of heat release in first and second phases, become higher.
- At optimum hot EGR at each higher load, the peak pressure of the cycle is observed to be maximized.
- The effect of hot EGR on delay period is negligible but cold EGR increases the delay period. At optimum hot EGR at each higher load, the peak pressure of the cycle is observed to be maximized. This could be because of some active species with optimum EGR (hot and cold), increases the rate of reactions.
- With EGR (hot and cold), rate of pressure rise is higher and hence the use of EGR for reducing the combustion knock at higher loads is not recommended.
- Relative cycle efficiency is observed to be maximum at optimum hot EGR. Cold EGR does not seem to be advantageous.
- The total duration of combustion is observed to be longer with hot and cold EGR.

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ABBREVIATIONS

IMEP	**Indicated Mean Effective Pressure**
LPG	**Liquefied Petroleum Gas**
EGR	**Exhaust Gas Recirculation**
RCE	**Relative Cycle Efficiency**
TDC	**Top Dead Centre**
kW	**Kilo Watt**
BTDC	**Before Top Dead Centre**
CA	**Crank Angle**
SD	**Standard Deviation**

APPENDIX [A]: ENGINE DETAILS

Engine Type	**Single Cylinder, Four Stroke, Direct Injection Diesel Engine**
Rated Power	**3.7 kW @ 1500 RPM**
Bore × Stroke	**80 mm × 110 mm**
Compression Ratio	**15: 1**
Injection Timing, Injector Opening Pressure	**27.4 °CA BTDC (Static), 170 bar**

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