EFFECT OF EGR INJECTION RATE AND ENGINE LOAD ON CYCLIC VARIATIONS OF AN LPG DIESEL DUAL FUEL ENGINE

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ABSTRACT

Operation of dual fuel engine is limited due to higher cyclic variations in comparison to diesel engine. Cycle by cycle variability is an important constraint on engine operation. It is known, that the cycle-by-cycle variability in cylinder peak pressure, Indicated Mean Effective Pressure (IMEP) and engine speed is the result of variations in the combustion process. The cyclic variations in dual fuel engine tend to be greater than that in the corresponding diesel operation yet much smaller than that of the spark ignition engine. During the present study, efforts have been made to measure the cyclic variations in an LPG diesel dual fuel engine. Effect of Exhaust Gas Recirculation (EGR), the rate of injection and load has been studied in depth. It has been concluded that liquid pilot fuel controlled pre-mixed combustion in first phase remained practically unchanged with change in engine operating conditions and thus do not affect the cyclic variability. Diffusion burning of remained diesel fuel and flame propagation of gaseous fuel in the second stage of combustion is mainly responsible for cyclic variations. Even though, cyclic variations in injection timing were observed but these did not affect the start of ignition point. The Coefficient of Variation (COV) of peak pressure is higher than diesel mode at all temperatures and outputs but the values within 4% up to 60% of full load. COV of peak pressure is mainly affected by the irregularities in the combustion of the diesel + gas air mixture in the second stage of combustion and not by the combustion of the pilot diesel.

Key Words: Cyclic variation, Dual fuel engine, Exhaust Gas Recirculation (EGR), LPG, Coefficient of Variation (COV)

INTRODUCTION

In a dual fuel engine, two fuels are used simultaneously. Primary fuel that is usually gaseous forms the major content of the total energy supplied to the engine. After the compression of the primary fuel air mixture, small quantity of pilot diesel initiates the combustion near top dead centre. The pilot diesel auto ignites first and works as an intense ignition source after which combustion of the inducted fuel occurs. 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. It is estimated that at full load the dual fuel engine requires a retarded injection timing to assure a safe level of operation. This would result in loss of thermal efficiency by 1 to 2%. Increased admission of gaseous fuel results uncontrolled reaction rates near the pilot fuel spray leading very high combustion rates and hence very high rates of pressure rise leading to knock. The resort of Exhaust Gas Recirculation (EGR) 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. Hannu4 used EGR in natural gas fueled SI engine and concluded that upto 8% EGR, NOx emission reduces. Further addition of EGR resulted in higher hydrocarbon emissions and
Cycle by cycle variability is an important constraint on engine operation. It is known that the cycle-by-cycle variability in cylinder peak pressures and indicated mean effective pressure is the result of variations in combustion process itself. Cyclic variations in the cylinder pressure development have been observed involving both single and multi-cylinder engine. The cyclic variations in dual fuel engine tend to be greater than that in the corresponding diesel operation yet much smaller than the spark ignition engine.\textsuperscript{5,6} Zarling et al.\textsuperscript{7} studied the cyclic variations of pressure development in the combustion process in diesel engines as indicated by Indicated Mean Effective Pressure (IMEP). It was found very low, less than two percent of the mean. Karim et al.\textsuperscript{8} studied the cyclic variations in a methane fuelled dual fuel engine. The extent of cyclic variations in peak cylinder pressure is much greater than that in the ignition delay. Lower pilot fuel, higher substitution of gas and unmatched injection system increase the cyclic variations. Karim\textsuperscript{9} investigated the cyclic variations in SI engines fueled with gaseous fuel and concluded that the peak values of maximum rate of pressure rise and peak pressures are more variable in comparison to cyclic variations in indicated mean effective pressure. Indicators of cycle by cycle variations may be grouped into four main categories that are pressure related, combustion related, flame fronts related and exhaust gas related parameters.\textsuperscript{10} Cyclic variations were found to be severe if the mixture is lean or exhaust gas circulation is high.\textsuperscript{11,12} In spark ignition engines, the cyclic variations of combustion originate primarily in the conditions of the internal mixture that varies from one cycle to another.\textsuperscript{13,14} Peters\textsuperscript{15} observed that in the case of a spark ignition engine, difference in mixture motion near the spark plug at the time of ignition may be the reason of cycle to cycle variation. Mohamed Y.E. Salim\textsuperscript{16} studied the cyclic variations in a dual fuel engine by using LPG and Methane. The cyclic variations have been compared with diesel also and concluded that cyclic variation in peak pressure and brake mean effective pressure was found to be maximum while LPG was used as secondary fuel and diesel as pilot fuel. In diesel mode, cyclic variations observed to be the lowest. The reduction in combustion noise was postulated to the reduction in the maximum rate of heat release and COV in pressure rise rate may be caused by the reduction in heat release fluctuations.\textsuperscript{17-20} With the presence of gaseous fuel in the mixture, any injection advance in pilot fuel timing would result in a longer ignition delay period and hence the pressure rise rate is expected to increase.\textsuperscript{21,22} Cyclic variability in combustion in a dual fuel engine using LPG has not been studied in depth so far particularly when special measures are used to improve the performance of the engine. There is a thumb rule among engine designers world over that any engine with its COV of IMEP being over 10\% indicates that it is extremely high from a thermodynamic point of view.\textsuperscript{23} Diesel engine's combustion is assumed to be very stable where COV of IMEP is less than 3\%.\textsuperscript{24}

Stationary, single cylinder water cooled developing 3.7 kW @ 1500 revolutions per minute diesel engine was selected for present investigations. These engines are very common in India for operating pumps to supply water for drinking and irrigation purposes. The effect of EGR, the rate of injection and engine load on cyclic variations in diesel and dual fuel mode was studied during the present investigations. The engine cylinder pressures, and dynamic injection timing data for 100 cycles were stored on a personal computer. A program was developed to calculate the mean effective pressure, the maximum cycle pressure, the crank angle at which maximum cycle pressure occurs, the ignition delay period for every cycle and hence the angle at which combustion starts. The same program also computed the standard deviation and COV for mean effective pressure, peak pressure, angle at which this pressure occurs, starting point of injection and engine speed, computer program was also developed to view the pressure crank angle traces on the computer screen its first and second derivatives and average injector needle lift diagrams.

**Statistical evaluation of cyclic variations**

It is expected that the cyclic variations in the combustion process under constant operating conditions, would follow essentially a normal distribution. However, due to the piston motion during the combustion process the variations in peak pressure, rate of pressure rise and indicated mean effective pressure would not necessarily follow a normal distribution. The extent of statistical variations in any engine performance
parameter 'X' can be established with Coefficient of Variations (COV).

\[
(COV)_x = \frac{1}{X_m} \left\{ \sum_{i=1}^{N} (X_i - X_m)^2 / N \right\}^{0.5}
\]

Where, \(X_m\) and \(X_i\) refer to the mean and typical observation values respectively. \(N\) is the number of observations made for the value of \(X_i\) under the same engine operating conditions.

**AIMS AND OBJECTIVES**

The present work was to gain an insight into the combustion of dual engines using LPG at part loads as well as full load to identify the optimal performance domains. In addition, it was intended to study the effect of special measures such as EGR, the rate of injection, intake air heating etc. to improve the dual fuel engine performance. Such a study, it is hoped will facilitate the design of the kits for conversion of existing engines and will also be useful in the design of new dedicated engines. The optimization of the engine is intended for maximum thermal efficiency as well as maximum substitution of alternative gaseous fuel so that conventional fuels like diesel can be saved which are being depleted at a very rapid rate.

**MATERIAL AND METHODS**

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 Table 1. An LPG connection was made on the intake manifold. Governor varied pilot diesel flow while the LPG flow rate was varied manually. 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.

![Fig. 1: Experimental setup](image)

**Table 1: Test engine specifications**

<table>
<thead>
<tr>
<th>Engine type</th>
<th>Single cylinder, four stroke, direct injection diesel engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated Power</td>
<td>3.7 kW @ 1500 RPM</td>
</tr>
<tr>
<td>Bore × Stroke</td>
<td>80 mm × 110 mm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>15:1</td>
</tr>
<tr>
<td>Injection timing, injector opening pressure</td>
<td>27.4°CA BTDC (static), 170 bar</td>
</tr>
</tbody>
</table>
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.

On a personal computer, the cylinder pressure signal obtained from a flush mounted quartz pressure pickup recorded. Rate of heat release and other combustion parameters were computed using a software for the purpose after obtaining the average pressure signals from 100 consecutive engine cycles. Based on a previous experimental program injection timing of 27.4ºCA BTDC and 24ºCA BTDC were selected for dual fuel and diesel operation respectively. Based on the same work, injection pressure of 150 bars up to 80% load and 200 bars beyond that were used in the dual fuel mode. Very low (125 bar) and very high (225 bar) injector opening pressure and retarded injection timing of 24ºCA BTDC were also tried in the dual fuel engine but the engine performance was found very poor.

Based on the brake thermal efficiency of the engine at different loads optimum injector needle lift pressures were obtained. An improvement in thermal efficiency up to 3% can be obtained at loads less than 80% by reducing the injector needle lift pressure from 170 bars to 150 bars. It has been observed that the engine knocking is severe at 150 bar injectors openings pressures, and 100% load. In dual fuel engine, at 100% load, peak cycle pressure go down significantly using 200 bars injector needle lift pressure in comparison to 150 bars. In diesel fuel mode, injector needle lifts pressure of 170 bars, was found to be optimum. It has been observed that at 200 bars injector opening pressure, knocking found to be less probably due to more spread of the pilot fuel in the combustion chamber in comparison to 150 bars. The optimum pilot fuel quantities and intake temperatures set for the experimentation at different loads are given in Table 2.

Table 2: Optimum engine operating conditions

<table>
<thead>
<tr>
<th>Load (%)</th>
<th>Intake temp. (ºC)</th>
<th>Pilot fuel quantity (mg/cycle)</th>
<th>Injector needle lift pressure (bar)</th>
<th>% optimum EGR by volume</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>70</td>
<td>8.4</td>
<td>150</td>
<td>18</td>
</tr>
<tr>
<td>40</td>
<td>70</td>
<td>10.7</td>
<td>150</td>
<td>10</td>
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<td>60</td>
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<td>100</td>
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<td>7.1</td>
<td>150</td>
<td>4</td>
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<tr>
<td>100</td>
<td>40</td>
<td>7.1</td>
<td>200</td>
<td>4</td>
</tr>
</tbody>
</table>

RESULTS AND DISCUSSION

The analysis of engine pressure crank angle diagrams is important in the study of dual fuel engines because of the complex combustion processes in these engines. A convenient way of depicting the combustion process and cyclic variations are by the heat release rate and mass fraction burnt diagram of fuel as a function of crank angle. During the present study, heat release diagrams have been obtained from experimental crank angle diagrams at a crank angle step of 1ºCA. Three stages of combustion, have been observed at 80% load and using the pilot fuel quantity of 5.9 mg/cycle as shown in Fig. 2. In the first stage (pre mixed stage), mainly the diesel burning takes place with a very small amount of gas available in the pilot fuels spray zone. In the first phase of combustion, heat release rate may reach up to 45 J/ºCA. Heat release continues 5-6ºCA after TDC in the first phase of combustion. During first phase of combustion, approximately 15-20% of the total energy supplied release in the form of heat 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. During the second phase of combustion, remaining 50% pilots burns in diffusion controlled mode. The second stage releases approximately 50% of the total heat in next 6-7\(^{\circ}\)CA. In this stage, homogeneous gas air mixture near the pilot fuel spray is consumed very rapidly. The remaining gas air mixture is consumed through flame propagation in the third stage of combustion just like after burning in diesel engines. The amount of heat released in the second stage of combustion is mainly responsible for the cyclic variations. Experiments for obtaining cyclic variations were conducted in the whole load range but the results are presented mainly for 20% and 80% loads. In order to represent performance at low loads, 20% of full load have been selected, whereas, 80% load represents the higher load side.

**Effect of EGR on cyclic variations**

**Fig. 3** and **Fig. 4** compare the cyclic variations with and without EGR in peak pressure and IMEP in the dual fuel mode at different loads. The values for diesel operation at ambient temperature are also given for the purpose of comparison. Percent EGR, pilot and intake temperatures were set at their optimum values as given in the Table 2. The cyclic variations in peak cycle pressure with straight diesel were found to be least (less than 2\%). In the dual fuel mode, an increase in load increases the COV till 80% load. At 100% load, slight drop in COV is observed. It may be recalled that the injection pressure is significantly different at 100% load, in comparison to 80% load to insure knock free operation. Cyclic variations in IMEP are less with diesel (less than 6\%) than dual fuel operation and decreases with increase in power output. Up to an output of 60\%, it is seen that COV in peak pressure and COV in IMEP are higher without EGR as compared to those with EGR in the dual fuel mode. **Fig 3**, shows typically how the COV in peak pressure for a dual fuel engine varies with load while using different percentage of EGR. At light loads (20\%, 40\% and 60\% of full load) where the size of pilot is large and combustion is mainly controlled by pilot fuel in first and second phase, the addition of EGR tends to reduce the cyclic fluctuations. It has been observed that the addition of EGR at higher loads (80\% and 100\%) increases the combustion duration and hence the cyclic variations. It is common opinion among the investigators that the higher the burning rate and lower the combustion duration lesser are the cyclic variations.

COV in IMEP with change in percent EGR is shown in **Fig 4**. EGR seems to have a favorable effect on the first and second phases of combustion. It is seen that in the dual fuel mode, while COV in peak pressure increases with load, the COV in IMEP decreases with load generally. The COV in peak pressure is a result of the variations in the heat release rate in the first and second phase while the COV in IMEP is a result of the whole combustion process. When the load on the engine becomes higher than 60\%, the presence of EGR increases the cyclic variations both in peak pressure and IMEP. At 80% load, the cyclic variations may reach up to 6\%.
Effect of rate of injection on cyclic variations

Fig. 5 shows typically how the COV of cylinder peak pressure varies for a dual fuel engine with load and an injection pressure of 125 bars. The baseline engine was equipped with a 6 mm plunger hence in order to change the rate of injection; a 5 mm plunger was used in its place. Decrease in the diameter with no change in the cam will lead to a reduction in the rate of injection as well as injection pressure. Attempt has also been made to keep the same dynamic injection timing with 5 mm plungers as with 6 mm plunger. It has been observed that unmatched plunger increase the cyclic variations. At light load, increase in cyclic variations, using 5 mm plungers is probably a reflection of the increasingly dominating role of pilot fuel injection. It increases further by reducing the injection pressure which deteriorates the spray geometry. The energy release rate will be dominated by flame propagation at higher loads. The poor atomization of the pilot fuel at low rate of injection reduces the intensity of the ignition source, resulting in slower flame propagation and longer combustion duration. This is also reflected through the lower peak cycle pressures with lower rate of injection. This evidently emphasizes the need of optimizing the injection characteristics of the fuel pump for dual fuel operation.

The effect of load on COV of IMEP and engine speed is given in Fig. 6 to Fig. 7 respectively for varying load in dual fuel and diesel mode. As shown in Fig. 6 and Fig. 7, the COV of IMEP decrease with increase in load using 5 mm plunger and 125 bar injection pressure in dual fuel operation. At 20% load, the COV of IMEP can reach up to 13% in comparison to 6% in the case of 6 mm plungers and 150 bar injection pressures. The increase in load improves the drive-ability and engine operation both in the case of diesel and dual fuel mode. In the case of diesel operation, the cyclic variations with both the plungers are observed to be less in comparison to dual fuel operation. Same trend of cyclic variability in engine speed with load has been observed in diesel and dual fuel mode as shown in Fig. 8 and Fig. 9. At higher loads due to maximum energy substitution by gas, cyclic variations of speed were found to be more.

Effect of load on cyclic variations

The COV of peak pressure and Indicated Mean Effective Pressure (IMEP) with change in load depicted in Fig. 10 and Fig. 11 respectively. It is seen that COV of IMEP and peak pressure has no relationship. The COV of IMEP is a tool of the drive-ability test. The drive-ability is found to be lowest at 100% load and increases with decrease in output. The COV of IMEP is less than 10%, which indicate good drive-ability. The values for diesel operation at ambient temperature are also given for the purpose of comparison.

Cyclic variations at optimum operating conditioning

Fig. 12 compares cyclic variations in pressure traces where 10 cycles at optimum conditions are plotted. Various traces deviate substantially after 5-8°CA from the ignition initiation points. Engine operation with EGR, cyclic fluctuations are the result of the second phase of combustion only. Cyclic variations in the second phase of combustion are significantly higher. The maximum deviation can reach up to 11 bars in peak cycle pressure.
Fig. 5: Load vs COV in peak pressure at optimum conditions

Fig. 6: Effect of load on COV in IMEP in dual fuel mode at optimum conditions

Fig. 7: Load vs COV in IMEP in diesel engine at optimum conditions

Fig. 8: Load vs COV in speed in dual fuel mode

Fig. 9: Load vs COV in speed in diesel mode

Fig. 10: Load vs COV in peak pressure
The cyclic variations in peak pressure and angle at which peak pressure occurs for 100 cycles with 5.9 mg/cycle pilot and 600°C intake temperature is shown in Fig. 13. The load on the engine was 80% of full load and EGR is at its optimum value of 6%. It has been seen earlier that whenever heat release through flame propagation dominates, the COV and brake thermal efficiency both increase. The COV and Standard Deviation (SD) observed is 6.1% and 3.9 bar respectively. It can be concluded that most of the gaseous fuel at higher load is consumed by flame propagation. Due to this, COV is higher but thermal efficiency is also observed to be higher. Fig. 14 shows the cyclic variations of 10 cycles.
cycles in dual fuel mode. It is clear from the diagram that the cyclic variations are affected by both the phases of combustion in contrary to well-matched injection system (6 mm) where cyclic variations were only due to second and third stage of combustion. At 80% loads, both peak pressure and angle at which peaks pressure occurs varies as shown in Fig. 15. The maximum variation may reach of the order of 15 bars.

CONCLUSION

Salient conclusions of the present study are summarized below:

Liquid pilot fuels controlled first phase of combustion remained practically unchanged from cycle to cycle. Even though, cyclic variations in injection timing were observed but these did not affect the ignition point. The COV of peak pressure is higher than diesel mode at all temperatures and outputs but the values within 4% at up to 60% loads. COV of peak pressure is mainly affected by the irregularities in the combustion of the gas air mixture and not by the combustion of the pilot diesel. Only second and third stages of combustion are responsible for cyclic variations and not the first stage.

Up to 60% load, the cyclic variation in peak cycle pressure and IMEP are lower with optimum EGR as compared to with no EGR. Higher COV is observed at loads higher than 60% with EGR. The reason may be slower flame propagation due to the presence of EGR.

Up to 60% load, the increase in cyclic variations with 5 mm plungers is more due to larger pilots at these loads. Unmatched injection deteriorates the atomization, vaporization and mixture formation of pilot fuel. At 80 and 100% loads, the size of pilot is small and hence the effect of cyclic variations in these loads due to lower rates in injection is less.

With well-matched injection system, cyclic variations are due to the second phase of combustion but with the unmatched injection system, both the phases are responsible for variation.

For lesser cyclic variations and better engine performance, well matched injection system is needed.

REFERENCES


