

Numerical Simulation of Combustion Mechanism of Dual Fuel Diesel-Producer Gas Compression Ignition Engine

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Abstract—*The answer to increasing energy requirements and decreasing oil reserves are probably alternative fuels and renewable energy sources. In this work combustion modelling of diesel-producer gas dual fuel compression engine is carried out. The relation between temperature and absolute internal energy is developed and used in first law of thermodynamics to solve unknown temperature. The relationship between equivalence ratio and thermal efficiency of diesel cycle is obtained which is consistent with the available data. Then the model of dual fuel dual cycle is solved for various combinations of diesel and producer gas burned at constant volume combustion process and their effect is shown. It is inferred that the amount of diesel burned at constant volume combustion process is 20% of total diesel supplied. It can also be inferred that the fraction of producer gas burned in constant volume combustion process does not affect the thermal efficiency significantly.*

Keywords— *Compression ignition, Dual fuel, producer gas and simulation.*

I. INTRODUCTION

Diminishing oil reserves and increasing power requirements are posing a steep challenge for researchers to search for alternative fuels and renewable energy sources. Studies suggest that oil reserves will probably last only for next 50 years, natural gas for 60 years, and coal for another 200 years. So there is a tremendous need for alternative fuels or renewable energy sources or to develop technology to reduce consumption of these fuels especially liquid fuels. Nowadays natural gas, LPG, methane, ethanol, hydrogen, producer gas, vegetable oil, and burned edible oil etc. are under research to use them as alternative fuels. In 1938 Europe operated about 9000 producer gas buses and trucks, and there were almost none on any other continent. By 1941, however, about 450,000 vehicles were in operation in all parts of the world, and by 1942 the number had grown to approximately 920,000 [1]. Producer gas were then in use not only in land vehicles, but also in boats, barges, and stationary engines. Compression ignition (CI) engines, commonly known as diesel engines, are designed to operate on a less refined petroleum distillate than gasoline which contains less energy content. Yet CI engines are more efficient at translating fuel energy content into output than their spark ignition counterparts. CI engines also have fewer overall emissions than spark ignition engines and are preferred for emissions reductions [2]. The earliest operations of the dual fuel system were in those experiments performed by Cave in 1929 and Helmore and Sokes in 1930 [3], in which hydrogen was introduced as a secondary fuel in diesel engines. It was stated that a 20% saving in diesel fuel was possible if hydrogen was burned instead of being discharged, as the liquid fuel load decreased. By 1939, the first commercial dual fuel engine, fuelled by town gas or other types of gaseous fuels, was produced by the National Gas and Oil Engine Co. in Great Britain. Different kinds of gaseous fuels, such as coal gas, sewage gas or methane, were employed in conventional diesel engines [4]. One of the main problems with dual fuel operation is that, at light load, dual fuel engines usually exhibited a drop in efficiency and power output with respect to diesel operation. The emissions of unburned hydrocarbons and carbon monoxide under dual fuel operation were higher in comparison to diesel operation [5,6]. Nielsen et al. [7] in 1987 indicated the amount of pilot fuel and the intake temperature necessary to avoid ignition failure at part load operation. Soliman and Isaïd [8] in 1993 investigated the effect of using LPG or gasoline vapor as a secondary fuel for a single cylinder diesel engine, on its performance and exhaust smoke level. They found that the addition of LPG or gasoline vapor, while reducing the liquid fuel at constant speed, causes a sharp reduction in the smoke number. D.H. Qi et al. [9] in 2006 experimentally investigated to examine the effect of blended fuel combustion on the performance and pollutant emissions of a DI Diesel engine using LPG–Diesel blended fuel. It revealed that blended fuel operation results in lower peak cylinder pressure with increase of the LPG mass fraction in blended fuels. It also revealed that, NO_x concentration is lower compared to diesel operation.

R.G. Papagiannakis and D.T. Hountalas [10] in 2004 have made conclusive studies on the potential benefits of using natural gas in diesel engines are both economical and environmental. The high auto ignition temperature of natural gas is a serious advantage since the compression ratio of conventional diesel engines can be maintained. Dual fuel combustion using natural gas is a technique that results in the reduction of both pollutants NO and soot, which is extremely difficult to achieve in direct injection Diesel engines, and contributes to the reduction of carbon dioxide emissions due to the low Carbon-Hydrogen ratio. Producer gas is generated from wood, charcoal or coal which is gasified with air and consists of some 40 per cent combustible gases, mainly carbon monoxide, hydrogen and some methane. The knock tendency with producer gas can be expected to be better on account of large fraction of inert gas it contains as compared to natural gas [11]. In India, work in the area of producer gas engine has been reported by Indian Institute of Technology, Mumbai. Shashikantha et al [12, 13] and Parikh et al [14] have reported work on a gas engine converted from a naturally aspirated diesel engine at CR of 11.5.

The reason for limiting the CR is cited to be the knocking tendency; however, no experimental evidence is provided in support of it. The only earlier experimental work in the higher CR range is reported by Ramachandra [15] on a single cylinder diesel engine (CR=16.5) coupled to a water pump. A power de-rating of 20% was reported at an overall efficiency of 19% without any signs of detonation. In order to predict the engine performance working with such a fuel, an Engine Fuel Quality (EFQ) parameter was developed by Francisco V. Tinaut et al. in 2006 [16]. This parameter considers the combined effect of stoichiometric air–fuel ratio and stoichiometric mixture heating value, both depending on the producer gas composition. The estimation of engine power made by using the EFQ parameter indicates that power at full load is reduced to about two-thirds of the maximum obtained with a conventional liquid fuel.

In this work a dual cycle is considered. For this analysis a gaseous fuel-air mixture is considered to be admitted to the cylinder. A single cylinder 4 stroke direct injection dual fuel engine was used for all calculations. Table I gives the complete specification of the engine under consideration.

Table I: Specification of Engine

| | |
|---------------------|---|
| Engine Type | 4 Stroke Direct Injection Dual fuel Engine |
| Stroke | 125mm |
| Bore | 105 mm |
| Number of Cylinders | 1 |
| Combustion System | Direct Injection |
| Compression Ratio | 17.5 |
| RPM | 2500 |

Table II: Composition of Producer gas [17]

| Components | Composition | mole fraction |
|-----------------|----------------|---------------|
| CO | a ₁ | 0.27 |
| CH ₄ | a ₂ | 0.03 |
| H ₂ | a ₃ | 0.14 |
| CO ₂ | a ₄ | 0.045 |
| N ₂ | a ₅ | 0.509 |
| O ₂ | a ₆ | 0.006 |

II. MODELING OF DUAL CYCLE

Initially a charge of producer gas and air is compressed is entropically from a top to bottom dead centre and after compression the pressure and temperature will increase and that will be calculated by applying first law of thermodynamics. The second step is combustion process, where m_f moles of diesel fuel are supplied from time t_1 to t_2 seconds after compression and this interval is divided into a large number of intervals i.e. n_T , so that in dt time dm_f g moles of fuel is supplied in each interval.

therefore $m_f = n_T \times dm_f$

It is also assumed that in each interval x fraction of total moles of diesel present in each interval is burned.

$$\text{Amount of diesel fuel burned in } n^{\text{th}} \text{ interval is } \frac{1-(1-x)^n}{x} dm_f \times x = [1 - (1-x)^n] dm_f \quad (1)$$

Combustion process initiates with adiabatic combustion at constant volume. Energy balance equation by applying 1st law of thermodynamics

For combustion in n^{th} interval the above equation become

$$u_n(T_n) - u_{n-1}(T_{n-1}) + \left[\left\{ \frac{1-(1-x)^n}{x} \right\} dm_f \right] \times CV_{\text{diesel}} = 0 \quad (2)$$

$CV_{\text{diesel}} = \text{lower calorific value of diesel} = 43.1 \text{ MJ/kg}$

Composition of the gaseous mixture present before and after each small step of combustion at constant volume for dual fuel dual cycle are calculated from combustion reaction. The equation (2) will be solved by assuming the value of $n=n_1$ and y where the above compositions are calculated from mole balance during reaction and internal energy equation are substituted in $u_n(T_n), u_{n-1}(T_{n-1})$. After adiabatic constant volume combustion adiabatic Combustion at Constant pressure takes place. The composition of producer gas is given in table II which is used to calculate the total change in internal energy during the cycle.

For combustion in n^{th} interval the above equation becomes

$$u_n(T_n) - u_{n-1}(T_{n-1}) + \left[\left\{ \frac{1 - (1-x)^n}{x} \right\} dm_f \right] \times CV_{\text{diesel}} + P(V_n - V_{n-1}) = 0 \quad (3)$$

From equation of state at constant pressure,

$$V_n = \frac{T_n}{T_{n-1}} V_{n-1} \quad (4)$$

The equation (3) will be solved by assuming the value of $n = (n_T - n_1)$ and x and composition of the different components can be calculated and internal energies are substituted in $u_n(T_n), u_{n-1}(T_{n-1})$. Next step is Isentropic expansion of combustion products. The mixture combustion products then isentropically expand from the end of combustion to bottom dead centre and after expansion the pressure and temperature will decrease and that will be calculated by applying first law of thermodynamics. The Pressure and temperature in the compression process is calculated as discussed in isentropic compression process. The effect of (n_1/n_T) and y on the performance of Diesel-Producer gas operated dual cycle engine is studied from the variation of thermal efficiency with % of diesel and producer gas burned during constant volume combustion process, and a suitable value of (n_1/n_T) and y is fixed for analysis for next stage. The proposed model as discussed is simulated using MATLAB7.0 code. The details of the results obtained are discussed in the next chapter. The number of divisions is 100 for compression, combustion and expansion processes. In dual fuel mode the sum number of divisions for constant volume and constant pressure combustion process is 100. However the number of divisions for constant volume or constant pressure combustion process is decided upon by criteria mentioned above. A single cylinder 4 stroke direct injection dual fuel engine was used for all calculations. Table I gives the complete specification of the engine under consideration.

III. RESULTS

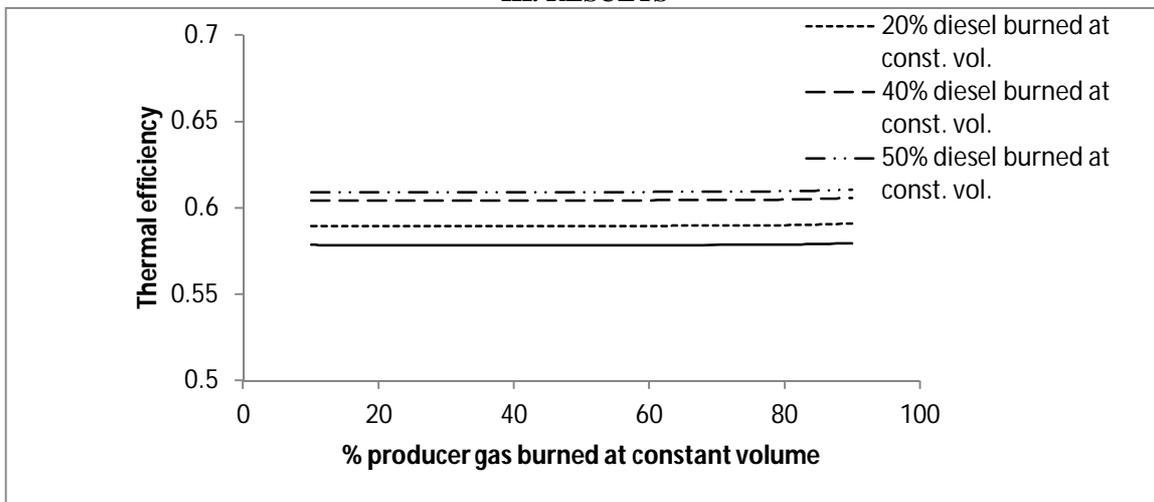


Fig 1: Effect of producer gas burned at constant volume combustion process on thermal efficiency

From the figure1 it is observed that thermal efficiency increases with the % of diesel burned in constant volume. The rate of increase in thermal efficiency is decreases after 20% of diesel burned in constant volume. So, 20% of diesel burned in constant volume is considered for the next stage of calculation.

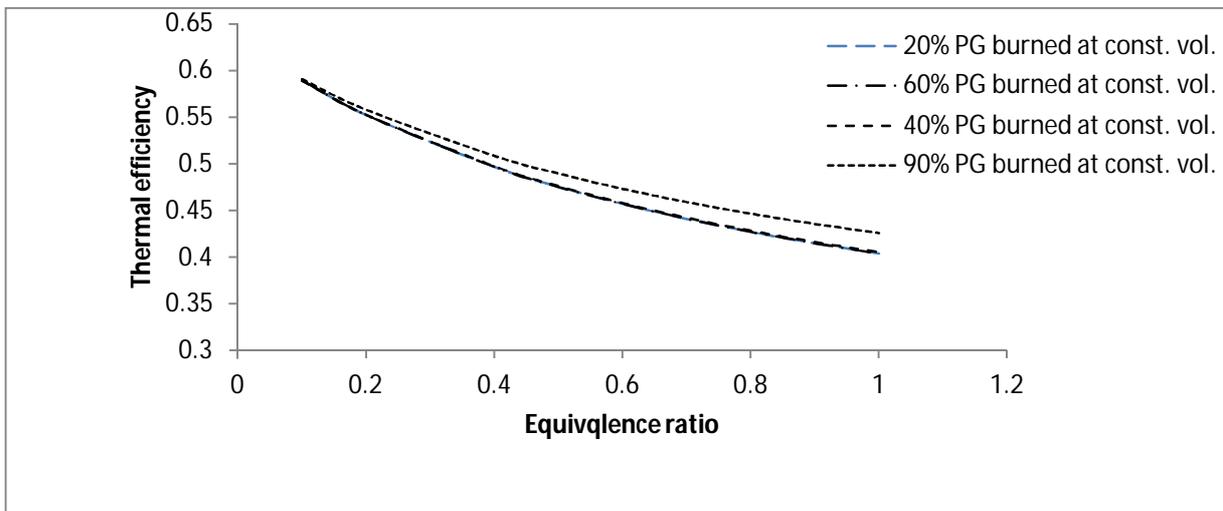


Fig 2: Effect of equivalence ratio on thermal efficiency with increasing % of producer gas burned at constant volume combustion process for different 20 % of diesel burned at constant volume combustion process.

IV. CONCLUSION

In this work the relation between temperature and absolute internal energy is developed and used in first law of thermodynamics to solve unknown temperature. The relationship between equivalence ratio and thermal efficiency of diesel cycle is obtained which is consistent with the available data [18]. Then the model of dual fuel dual cycle described is solved for various combinations of diesel and producer gas burned at constant volume combustion process and their effect is shown in figure 1 and 2. It can be concluded from figure 1, that the amount of diesel burned at constant volume combustion process is 20% of total diesel supplied. It can be inferred that the fraction of producer gas burned in constant volume combustion process does not affect the thermal efficiency significantly (fig.1 and 2). In next stage of calculation, hydrogen fuel is mixed with a mixture of producer gas and air. To study the effect of hydrogen addition a very small fraction of producer gas has been burned in constant volume combustion process.

LIST OF SYMBOLS

| Symbol | Description | Units |
|---------------|---|--------------------|
| n | nth Number of division | |
| x | Fraction of diesel converted in each division | |
| y | Fraction of producer gas burned in constant volume combustion process | |
| ϕ | Equivalence ratio | |
| dm_f | Gram moles of diesel supplied in each division during combustion | gmol/cycle |
| CV_{diesel} | lower calorific value of diesel | MJ/kg |
| CV_{PG} | lower calorific value of producer gas | KJ/Nm ³ |
| m_{air} | Gram moles of air supplied | gmol/cycle |
| m_f | Total gram moles of diesel supplied during combustion | gmol/cycle |
| m_{PG} | Gram moles of producer gas supplied | gmol/cycle |
| n_1 | Total number of division in constant volume combustion process | |
| n_T | Total number of division in combustion process | |
| P_n | Pressure after nth division | N/m ² |
| P_{n-1} | Pressure before nth division | N/m ² |
| T_n | Temperature after nth division | K |
| T_{n-1} | Temperature before nth division | K |
| $u_n(T)$ | Function of absolute internal energy with temperature of nth species | J |
| $u_n(0)$ | absolute internal energy at zero degree Kelvin of nth species | J |
| V_n | Volume after nth division | m ³ |
| V_{n-1} | Volume before nth division | m ³ |

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