THERMODYNAMIC ANALYSIS OF DIESEL ENGINE USING PRODUCER GAS AS SECONDARY FUEL

Vivek Kumar Gaba*, AnimeshDewangan, ShubhankarBhowmick
(*Corresponding author)
Department of Mechanical Engineering,
National Institute of Technology Raipur
Raipur(CG),492010,
India,
E-mail: vgaba.mech@nitrr.ac.in

ABSTRACT

The proposed work reports a combustion model for a dual fuel CI engine for Diesel cycle and Dual cycle. The work then analytically examines the performance of a CI engine with the minimum use of diesel as primary fuel, with producer gas as secondary fuel. The performance has been analysed for work output and thermal efficiency. The combustion model has been developed for ideal diesel engine, diesel-producer gas engine with diesel as primary fuel. The combustion models have been simulated using MATLAB® codes. The specifications of a standard CI engine have been used for numerical calculations. The equation of state in each process in the cycles has been critically analysed based on laws of thermodynamics. The performance analysis shows that, in a diesel-producer gas engine, thermal efficiency increases and work done decreases as the percentage of diesel supplied during combustion process decreases. Also, the addition of producer gas in a dual-fuel (Diesel-Producer gas) engine allows the engine to run over a lower range of maximum temperature of the cycle which reduces the chances of Nox formation.

NOMENCLATURE

D	[m]	Bore diameter of cylinder
L	[m]	Stroke length
m_{air}	[mole]	Mass of air
m_f	[mole]	Mass of fuel
P	[N/m^2]	Pressure
R	[J/g-mol-K]	Universal gas constant
T	[K]	Temperature
U	[J]	Absolute Internal Energy
V	[m^3]	Volume
x	[-]	Fraction of diesel converted in each division
y	[-]	Fraction of diesel burned in constant volume cycle
δQ	[J]	Heat transfer
dU	[J]	Change in Internal Energy
δW	[J]	Work transfer

Special characters

β_{st}	[-]	Stoichiometric air-fuel ratio
β_{act}	[-]	Actual air-fuel ratio
ϕ	[-]	Equivalence ratio

Subscripts

n	After nth division
$n-1$	Before nth division
fd	For Diesel fuel
fp	For Producer gas

INTRODUCTION

Compression ignition (CI) engines are important components of the transportation and energy sectors of the world. On daily basis millions of units are used due to which the emission of CI engines is an important concernunder pollution control. In case of CI engines, it is mandatoryto characterize and model each of the processes relevant to the fuel injection and combustion inside the combustion chamber. Although CI engines are designed to operate on less refined distillate than gasoline, yet these engines are more efficient at converting fuel energy content into work output compared to spark ignition engine. CI engines also have fewer overall emissions than spark ignition engines. Depleting fuel reserves and increasing emission levels have forced researchers to look for alternative/renewable energy sources. The present work proposes the combustion characteristics of one such combination of alternative fuel in form of diesel and producer gas. Modelling of mixture formation and combustion processes of the proposed fuel mixture is a very useful tool to analyse and predict the performance of reciprocating internal combustion engines.

In [1], constant eddy diffusivity and a one-step chemical reaction has been developed to study the flame propagation in a Spark Ignition (SI) engine at 1600 and 4200 rpm under rich fuel

conditions using one and two zone thermodynamic models. The thermodynamic models have been compared with 1-D model for average mixture temperature, the temperatures of the burned and unburned gases and the flame surface area and indicate that the one-dimensional model predictions are very sensitive to the eddy diffusivity and reaction rate data whereas the two-zone thermodynamic model predicts, first, a monotonically increasing flame surface area with time and, then, a monotonically decreasing surface area.Fuel evolution is modelled by studying the spray atomization, penetration, air entrainment, and evaporation; the air motion; and the ignition delay according to [2]. The model consists of analytical and semi-empirical sub models for all the above mentioned phenomena.Typical emphasis is laid on spray behaviour during the atomization process as well as after the impingement against the walls. In [3], the problems associated with the conversion of a direct-injection diesel engine to dual-fuel operation have been addressed. The effects of changes in different operating parameters such as inlet temperature and pilot liquid fuel quantity and quality on knock-free performance limit are studied, using propane, methane and hydrogen and their blends. The importance of volumetric efficiency in form of a measure of the effectiveness of an air pumping system is reported in [4]. The results from multidimensional numerical simulations are reported in [5] in an effort to optimize the performance of a fuel-lean-burn, homogeneous charge, natural gas spark-ignition internal combustion (IC) engine. The design and operational variables of inlet - exhaust system are observed in [6] to be decisive in determining overall engine performance. In [7], development of a complete cycle model has been attempted by preparing a source code for determining complete cycle performance characteristics, and exhaust emissions of diesel engines. An emission model based on the chemical equilibrium and kinetics of NO has also been proposed to calculate the pollutant concentrations within each zone and the whole of the cylinder. In [8] in-cylinder pressure, being an important parameter in study of the ICE combustion, has been investigated by introducing heat release function and an empirical conversion of efficiency factor into the model. For the oxidation of representative surrogate components of a typical multi-component automotive fuel,use of reduced chemical kinetic mechanisms have been reported in [9] where in the results show that the multi-component combustion model yields reliable performance for combustion predictions, as well as computational efficiency improvements. The performance analysis carried out in [10] observes that in a diesel-producer gas CI engine, thermal efficiency increases with increase in the percentage of diesel supplied during constant volume process with insignificant effect on work output. The insignificance of the quantity of producer gas burnt during constant volume process on thermal efficiency for fixed quantity of diesel is also reported.In another work [11], the combustion of Rapeseed Methyl Ester (RME), has been reported using a chemical model prepared by supplementing the existing detailed mechanism of methyl butanoate (MB) ester oxidation by sub-mechanisms for C_7H_{16} and C_7H_8O to represent combustion of RME, $C_{19}H_{34}O_2$ or $C_{19}H_{36}O_2$.

ANALYSIS METHODOLOGY

The present work deals with simulation of diesel engine working on dual fuel using MATLAB® codes. Simulation has been carried out in the following steps:

Step-1 Development of relationship between temperature and absolute internal energy.
Step-2 Modeling of ideal dual diesel cycle.
Step-3 Modeling of Producer gas fuel and diesel fuel dual cycle

A real single cylinder dual fuel engine specification (Table 1) has been used for all calculations.

TABLE –1: Specification of Engine

Engine Type	4 Stroke diesel fuel engine
Stroke	125mm
Bore	105mm
Number of cylinders	1
Combustion system	direct injection
Compression ratio	17.5
RPM	2500

Assumptions in Combustion Modeling:
1. An ideal diesel cycle is considered for the analysis.
2. Thermodynamic properties are assumed uniform throughout the combustion chamber volume. The gaseous mixture behaves as an ideal gas.
3. The combustion process results in complete combustion.
4. To simplify the combustion process, the intermediate steps of chemical kinetics are not considered.
5. The combustion products are limited to water vapor, carbon dioxide and inert nitrogen gas.
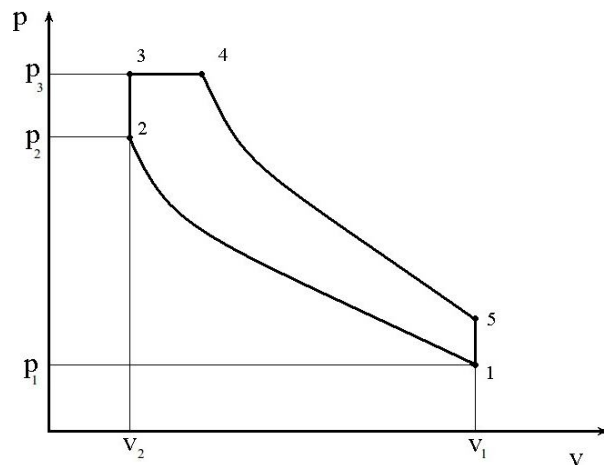
Figure 1 Ideal Diesel Engine

Governing Equations
The governing equations for the cycle 1-2-3-4-5 (fig1) are analyzed separately one by one as given below.

A charge of pure air and producer gas is compressed isentropically from state 1 to state 2 and after compression the pressure and temperature increases and which are calculated by applying first law of thermodynamics.

The whole compression process is assumed to be divided into 'N' number of small steps where,

$$dV = V_n - V_{n-1} \qquad (1)$$

(n varies from 1 to N)

From first law of thermodynamics, $dU = \delta Q - \delta W$, where dU is the change in internal energy after very small change in volume during the process, δQ is the heat transfer which is zero during isentropic compression process and $dW = p.dV$. In this case it is the work done by the system during n^{th} interval

$$\frac{P_n + P_{n-1}}{2}\left(V_n - V_{n-1}\right) \qquad (2)$$

Hence equation (1) yields,

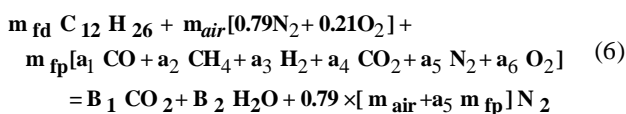$$\left[U(T_n) - U(0)\right] - \left[U(T_{n-1}) - U(0)\right] = -\left(\frac{P_n + P_{n-1}}{2}\right)(V_n - V_{n-1}) \qquad (3)$$

For iterative solution technique, the above equation assumes the form as given below

$$f(U) = \left[U(T_n) - U(0)\right] - \left[U(T_{n-1}) - U(0)\right] + \frac{P_n + P_{n-1}}{2}\left(V_n - V_{n-1}\right) \qquad (4)$$

where, P_{n-1}, V_{n-1}, T_{n-1} are the pressure, volume, and temperature before n^{th} interval and are known apriori; P_n, V_n and T_n are the pressure, volume and temperature after the nth interval during the compression stroke. From equation of state

$$P_n = \frac{V_{n-1}}{V_n} \cdot \frac{T_n}{T_{n-1}} \cdot P_{n-1} \qquad (5)$$

The solution of equation (4) is obtained numerically assuming the value of T_1 so that $f(U) = 0$. By repeating the same process the pressure and temperature at the end of compression can also be calculated. For the combustion process (i.e. process 2-3 and process 3-4), the internal energy values at different temperatures for these gaseous components present in air and alternative fuel were obtained from standard data [12]. An empirical relationship between temperature and absolute internal energy for different gaseous components that are present in air and producer gas is obtained by plotting absolute internal energy values at different temperatures and fitting 5th degree polynomial curve using curve fitting algorithm in MATLAB® and the correlations obtained are shown in Table-III. General combustion equation for blend of Producer gas and Diesel is given in equation (6), in which m_{fd} and m_{fp} are the total number of moles of diesel & Producer gas respectively.

$$m_{fd}\, C_{12}H_{26} + m_{air}[0.79N_2 + 0.21O_2] + $$
$$m_{fp}[a_1\, CO + a_2\, CH_4 + a_3\, H_2 + a_4\, CO_2 + a_5\, N_2 + a_6\, O_2] \qquad (6)$$
$$= B_1\, CO_2 + B_2\, H_2O + 0.79 \times [\, m_{air} + a_5\, m_{fp}]\, N_2$$

Here a's are the mole fractions of different components presents in the producer gas.

Assume that 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_{fd} g-moles of fuel are supplied in each interval. Therefore,

$$m_{fd} = n_T \times dm_{fd} \qquad (7)$$

It is also assumed that in each interval x mole fraction of total moles of diesel present is burned. Before combustion in 1st interval the moles of diesel present are dm_{fd}.

After combustion in 1st interval the moles of diesel present are $[(1-x)\, dm_{fd}]$. So, amount of diesel fuel burned in n^{th} interval is

$$\left\{\frac{1-(1-x)^n}{x}\right\}dm_{fd}\, x = \left\{1-(1-x)^n\right\}dm_{fd} \qquad (8)$$

So, total number of moles of air required for complete combustion is given by

$$m_{air} = \frac{\left[170\, m_{fd} + 24.744\left(\frac{PV}{RT}\right)b_{act}\right]}{\left\{29 + 24.744 b_{act}\right\}} \qquad (9)$$

Similarly, equivalence ratio, f is given by

$$f = \frac{b_{st}}{b_{act}} \qquad (10)$$

b_{st} can be calculated by taking the value of molecular weight of air and producer gas. Using Eq. (6-10), the moles for different composition of gaseous mixture present before and after each small step can be found out for both constant volume combustion (process 2-3) and constant pressure combustion (process 3-4) can be evaluated from the TABLE-II and TABLE-III respectively, for n^{th} step. The process 4-5 of isentropic expansion is simulated in a similar manner as process 1-2.

PROCESSING OF RESULTS

The proposed numerical model using Eq. (1-10), is simulated using MATLAB® codes. For solving the model, each co-ordinate of the cycle has been calculated along with the work done and thermal efficiency for different equivalence ratio on dual cycle. The result obtained are plotted in Fig. 2, 3 and 4. For each process the number of division taken is 100 and for simplicity the percentage of diesel burned at constant volume is assumed to be 50% (i.e. y = 0.5). A single cylinder 4 stroke direct injection dual fuel engine is used for all calculations. Table-I gives the complete specification of the engine under consideration. The values of pressure, volume, and temperature at the beginning of compression process are taken as 101325N/m² (P_1), 1147.57×10^{-06} m³ (V_1), 300K (T_1) respectively.

TRENDS AND RESULTS

It is observed from the variation of efficiency with equivalence ratio plotted in Fig. 2 that the efficiency decreases as ϕ

increases and it is also clear from the Fig. 2, that efficiency is higher when the percentage of diesel is less (or in other words percentageproducer gas is high) due to homogenous mixture of producer gas and air prepared before the combustion.
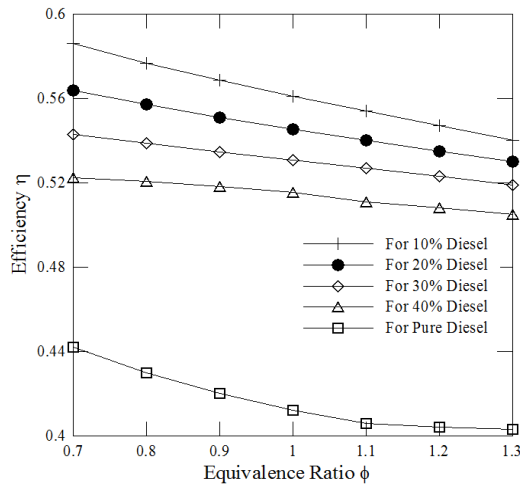
Figure 2 Variation of efficiency for different percentage of diesel fuel used in combustion with ϕ

The variation in work output with ϕ is plotted in Fig. 3 and it is observed that work done is maximum for pure diesel in all range of ϕ, reason being the higher calorific value of diesel compared to producer gas.
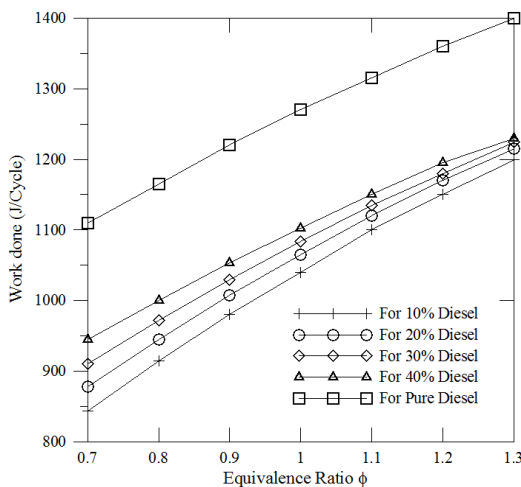
Figure 3 Plot of work done at different percentage of diesel fuel used in combustion with ϕ

In fig 4, it is further observed that pure diesel, when runs on leaner mixture i.e. $\phi < 1$, themaximum temperatureattained in the cycle is less compared with other cases when it runs on dual fuel but it increases when the engine runs on richer mixture i.e. $\phi > 1$ which causes the formation of NO_x.
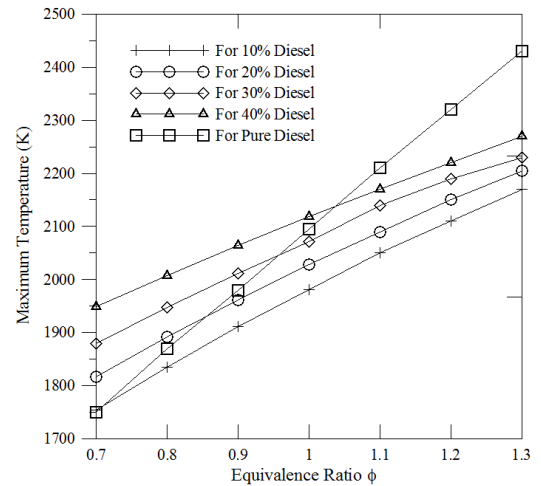
Figure 4 Plot of maximum temperature after combustion at various equivalence ratio, ϕ

CONCLUSION

From the results and discussions presented in the preceding section following important conclusions are drawn.

I. The thermal efficiency of a pure diesel engine as well as of dual fuel engine (producer gas and diesel) decreases with the equivalence ratio. The decreasing trend is obvious also since increase in equivalence ratio implies rich mixture.

II. The maximum cycle temperature is higher for pure diesel amongst all the combination of dual fuel when engine runs on richer mixture. So it can be concluded that by using producer gas, good efficiency can be achieved alongwith lower value of maximum temperature which constrainsNO_x formation temperature.

III. A sharp increase in work output is observed as the equivalence ratio increases due to more fuel injection.

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TABLE II: Number of moles for different components (for constant volume process)

Components	nth interval before combustion	nth interval after combustion
DIESEL	$\left[\left\{\dfrac{1-(1-x)^n}{x}\right\}(1-x)dm_f\right]$	$\left[\left\{\dfrac{1-(1-x)^n}{x}\right\}dm_f\right]$
CO	$\left[1-\left(\dfrac{y\times(n-1)}{n1}\right)\right]m_{pg}\times a_1$	$\left[1-\left(\dfrac{y\times n}{n1}\right)\right]m_{pg}\times a_1$
CH$_4$	$\left[1-\left(\dfrac{y\times(n-1)}{n1}\right)\right]m_{pg}\times a_2$	$\left[1-\left(\dfrac{y\times n}{n1}\right)\right]m_{pg}\times a_2$
H$_2$	$\left[1-\left(\dfrac{y\times(n-1)}{n1}\right)\right]m_{pg}\times a_3$	$\left[1-\left(\dfrac{y\times n}{n1}\right)\right]m_{pg}\times a_3$
CO$_2$	$12\left[(n-1)-\dfrac{1-x}{x}\left\{\dfrac{1-(1-x)^{(n-1)}}{x}\right\}\right]dm_f$ $+\left[\left(\dfrac{y\times(n-1)}{n1}\right)\right]m_{pg}$ $\times(a_1+a_2)+a_4\times m_{pg}$	$12\left[n-\dfrac{1-x}{x}\left\{\dfrac{1-(1-x)^n}{x}\right\}\right]dm_f$ $+\left[\left(\dfrac{y\times n}{n1}\right)\right]m_{pg}$ $\times(a_1+a_2)+a_4$ $\times m_{pg}$
O$_2$	$0.21m_{air}+a_6m_{pg}$ $-\left[\left(\dfrac{y\times(n-1)}{n1}\right)\right]m_{pg}(0.5a_1+2a_2+0.5a_3)$ $-18.5\Big[(n-1)$ $-\dfrac{1-x}{x}\left\{\dfrac{1-(1-x)^{(n-1)}}{x}\right\}\Big]dm_f$	$0.21m_{air}+a_6m_{pg}$ $-\left[\left(\dfrac{y\times n}{n1}\right)\right]m_{pg}(0.5a_1+2a_2$ $+0.5a_3)$ $-18.5\Big[n$ $-\dfrac{1-x}{x}\left\{\dfrac{1-(1-x)^n}{x}\right\}\Big]dm_f$
N$_2$	$0.79m_{air}+a_5m_{pg}$	$0.79m_{air}+a_5m_{pg}$

TABLE III: Number of moles for different components (for constant pressure process)

Components	nth interval before combustion	nth interval after combustion
DIESEL	$\left[\left\{\dfrac{1-(1-x)^{n1+n}}{x}\right\}(1-x)\right]dm_f$	$\left[\dfrac{1-(1-x)^{n1+n}}{x}\right]dm_f$
CO	$\left[1-\left(\dfrac{(n-1)}{n_T-n1}\right)\right]m_{pg}\times(1-y)\times a_1$	$\left[1-\left(\dfrac{n}{n_T-n1}\right)\right]m_{pg}\times(1-y)\times a_1$
CH_4	$\left[1-\left(\dfrac{(n-1)}{n_T-n1}\right)\right]m_{pg}\times(1-y)\times a_2$	$\left[1-\left(\dfrac{n}{n_T-n1}\right)\right]m_{pg}\times(1-y)\times a_2$
CO_2	$12\left[(n1+n-1)\right.$ $\left.-\dfrac{1-x}{x}\left\{\dfrac{1-(1-x)^{(n1+n-1)}}{x}\right\}\right]dm_f$ $+\left[y+\dfrac{(1-y)\times(n-1)}{n_T-n1}\right]\times(a_1+a_2)+a_4$ $\times m_{pg}$	$12\left[(n1+n)\right.$ $\left.-\dfrac{1-x}{x}\left\{\dfrac{1-(1-x)^{(n1+n)}}{x}\right\}\right]dm_f$ $+\left[y+\dfrac{(1-y)\times n}{n_T-n1}\right]\times(a_1+a_2)+a_4$ $\times m_{pg}$
O_2	$0.21m_{air}+a_6m_{pg}$ $-\left[\left(\dfrac{(1-y)\times(n-1)}{n_T-n1}\right)\right]m_{pg}(0.5a_1+2a_2)$ $-18.5\left[(n1+n-1)\right.$ $\left.-\dfrac{1-x}{x}\left\{\dfrac{1-(1-x)^{(n1+n-1)}}{x}\right\}\right]dm_f$ $-0.5a_3\times m_{pg}$	$0.21m_{air}+a_6m_{pg}$ $-\left[\left(\dfrac{(1-y)\times n}{n_T-n1}\right)\right]m_{pg}(0.5a_1+2a_2)$ $-18.5\left[(n1+n)\right.$ $\left.-\dfrac{1-x}{x}\left\{\dfrac{1-(1-x)^{(n1+n)}}{x}\right\}\right]dm_f$ $-0.5a_3\times m_{pg}$
N_2	$0.79m_{air}+a_5m_{pg}$	$0.79m_{air}+a_5m_{pg}$