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Thermodynamic Simulation and Performance Evaluation of Diesel Engine with Blends of Biogas, Methane and Hydrogen under Dual Fuel Mode

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Abstract:
This present investigation focusses on utilizing biomass materials in a CI engine used in agricultural applications. The simulated results were generated for a maximum 70% substitution of biogas, methane and biogas 50% plus hydrogen 20% with 30% diesel as a pilot fuel to assess the engine performance. Cylinder pressure and heat release rate under neat diesel operation as well as under dual fuel mode were validated with experimental results [23] under the same engine operating conditions. The engine power drop operating under pure biogas mode is improved by adding hydrogen under dual fuel mode. Brake power under 70% substitution for biogas and methane is around 5.116 kW and 11.28 kW respectively and for 20% hydrogen with 50% biogas is 11.2 kW at 1600 rpm for four cylinder engine. The long term use of this engine fueled with biogas and hydrogen under dual fuel is feasible with improved combustion and reduced cost of biogas over hydrogen.

Keywords: Diesel Engine, Alternative Fuels, Simulation, Blending, Performance

1. INTRODUCTION

With increasing emphasis on renewability and environment friendliness, biogas is of immense importance particularly for rural areas. Biogas operated dual fuel diesel engines can be a better solution to the problems of acute power shortage. Biogas produced by the anaerobic fermentation of cellulose biomass materials is a clean fuel for automotive engines. Its composition may vary from 50-70% CH_4, 25-50% CO_2, 1-5% H_2, 0.3-3% N_2 and some traces of H_2S. Because of its high compression ratio, biogas can successfully be used in diesel engines under dual fuel mode.

Gaseous fuels cannot be used directly in conventional diesel engines even at a very high compression ratio because of their high self-ignition temperatures. For igniting them, an intense source of energy is required such as spark or a pilot diesel injection. In the fumigated dual fuel mode biogas mixes with air before the mixture enters the combustion chamber at the end of compression stroke, an amount of diesel fuel called the pilot fuel is injected to ignite the gaseous fuel called the primary fuel. The dual fuel mode of operation has advantage of providing flexibility of using biogas and diesel and may easily revert back to pure diesel mode.

This can provide replacement of diesel up to 70%-80% for safe operation. Researches were carried out for the last many more decades using gaseous fuels more efficiently under dual fuel mode with CNG [1-10], liquefied petroleum gas [11-15], producer gas [16-21], biogas [22-27] and hydrogen [28-30]. Gaseous fuels generally experience low NOx and PM, however when a dual fuel engine operates at part load conditions with high substitution levels, the thermal efficiency is lower than diesel engine and CO and UHC emissions are increased. Biogas with its high octane number finds wide use in diesel engine with high compression ratio [24]. The presence of CO_2 and N_2 in the primary fuel increases the negative effects at part load operation due to its influence in burning rate inhibition [31 and 32]. Further investigations have been reported on using biogas under dual fuel engine operation [33-35], but still available information about strategies to improve part load operation in dual fuel engine is not self-sufficed. Biogas under dual fuel mode delivers low pollution as the main burning part of biogas is methane, which has low carbon content [36]. High biogas substitution levels on dual fuel engine performance result increased delay period and poor flame propagation of the mixture which may cause lower flammability limit [37].

Computer simulation to predict engine performance can be very helpful to understand engine characteristics under a wide range of parametric variations compared to experimental work. However modeling in dual fuel engine is quite complex. Hence semi-empirical models should be much more convenient to assess various parametric combinations.

The cycle simulation models for diesel engine and spark ignition engine are well established and are reviewed in the available literature [38-43]. However, the combustion in dual fuel engines incorporates the features of both diffusive burning of diesel fuel and pre-mixed burning of gas-air mixture and their complex interactions.

1. Objectives

The gaseous and alcoholic fuels because of having low carbon are extremely suitable for power production with novel technology. The present theoretical investigations are reported on using biogas, methane and hydrogen under dual fuel operation in a direct injection diesel engine to assess the potential of using them in automotive sectors under engine limiting conditions.

2. **Theoretical findings and methods**

2.1 Test engine

Table .1. Engine specifications [23]

Engine Type	Two cylinders, air - cooled, 4-stroke, DI diesel engine.
Displacement	1550 CC
Bore x Stroke	98 x 101 mm
Compression Ratio	15.5 : 1
Rated Power	20 kW at 3000 rpm
Maximum Torque	76 Nm at 1800 rpm
Inlet Valve Open	36° BTDC
Exhaust Valve Close	32° ATDC

2. Procedure

The energy substitution was considered for each fuel. The performance analysis was fixed up for a maximum diesel substitution up to 70% as given in Table 4. Firstly the engine was run for baseline conditions for pure diesel and later a blend of fixed proportion was taken into account for each fuel. While running the test program the pressure and heat release histories were generated for pure diesel and various chosen blends of diesel and methane, diesel and biogas and diesel and biogas and hydrogen to assess the performance behavior of the dual fuel diesel engine and well depicted in the figures. Comparative analysis are reported to assess the validity of the computer program under given operating conditions for a range of engine speed, energy substitution and fuel injection timing.

3. Formulation of the model

The model is classified for the purpose of analysis, namely the closed period corresponding to the power cycle.

3.1 Power Cycle Simulation

The period during which both the inlet and exhaust valves are closed, represent the significant part of the engine cycle. In this period, the power is developed by the engine. Hence the power cycle has been assumed to start from the point of inlet valve closing and extend until the exhaust valve opens. It consists of compression, combustion and expansion processes. The model detail is given in reference [44].

In the present study an empirical correlation [45] for delay period in milli seconds has been used.

$$Delay = \frac{3.52\exp(2100/T_{av})}{P_{av}^{\;1.022}} \qquad (1)$$

For the integration of eq (2), heat release rate during combustion process is to be computed, which is obtained from the combustion model. The heat release rate dQ/dθ analysis was performed by choosing double Wiebe function [46] for premixed and diffusive combustion periods observed in diesel engine.

$$\frac{dQ}{d\theta} = 6.9\frac{Q_p}{\theta_P}(MP+1)(\frac{\theta}{\theta_P})^{MP}\exp\left[-6.9(\frac{\theta}{\theta_P})^{MP+1}\right] + 6.9\frac{Q_d}{\theta_d}(MD+1)(\frac{\theta}{\theta d})^{MD}\exp\left[-6.9(\frac{\theta}{\theta_d})^{MD+1}\right] \qquad (2)$$

Where MP=3.0 and MD = 0.5, θ_P =7 degree and θ_d = 108 degree.

Mean piston speed (CM) can be calculated as

$$CM = \frac{SxN}{30} \qquad (3)$$

Cylinder volume as a function of crank angle can be known by using the equation of crank slider mechanism. Cylinder volume at any crank angle (θ) is given by

$$V = Vcl + \frac{\Pi}{4}.D^2[L + \frac{S}{2}(1-\cos\theta) - \sqrt{L^2 - \frac{S^2}{4}\sin^2\theta}] \qquad (4)$$

Further the new value of cylinder volume at an increment of θ may be calculated as

$$\dot{V} = (V_{\theta+\Delta\theta} - V_\theta) \qquad (5)$$

The brake power (BPW) is given by

BPW = IP – FP (6)

The indicated power (IP) is calculated by

IP = P x DV x N x 100 / (2 x 60) (7)

The friction power (FP) of the engine parts is calculated by determining the friction mean effective pressure (FMEP) [47].

FMEP = 75 + 48 x (N/1000) + 0.4 x (2 x S x 10^{-2} x N/ 60)^2 (8)

FP = FMEP x 1000 x S x 10^{-2} x (PI/4) x D^2 x 10^{-4} x N/ (60 x 1000) (9)

Brake torque (BT) can be calculated as

BT = BPW x 60 x 1000 / (2 x PI x N) (10)

Brake mean effective pressure (BMEP) is

BMEP = WD x 4/ (PI x D^2 x S) (11)

Work done is given by the relation

WD = BPW x 60 / N (12)

4. Results and discussion

4.1 Combustion analysis

In this work, four different fuels with various energy substitutions are given in Table 2 and their fuel properties are summarized in Table 3.

Table. 2. Energy substitution in engine.

DSL %	CH4 %	DSL %	Bio %	DSL %	Bio + H2 %
100	0	100	0	100	0
70	30	70	30	70	25+5
60	40	60	40	60	30+10
50	50	50	50	50	35+15
40	60	40	60	40	40+20
30	70	30	70	30	50+20

Biogas mainly contains methane and CO_2 in its chemical composition. Increase of CO_2 makes the combustion affected badly inside the engine cylinder, which introduces a less engine performance. For this reason, methane gas was also tested to experience the engine performance without CO_2.

4.2 Validation results

The pressure crank angle diagram obtained from the model and that of the experimental results [23] corresponding to dual fuel system for an engine speed of 1800 rev/min at an injection timing of 5° BTDC and compression ratio of 15.5 under stoichiometric conditions ϕ = 1.0, is shown in Fig. 1(a). It may be observed that there is no appreciable difference in cylinder gas pressure during most part of the power cycle, except near the peak pressure region. This may be due to the reason that the experimental diagram is based on a single cycle whereas in compression ignition engine there is likely to be considerable cycle by cycle variation in cylinder pressure pattern. It may further be observed from the Fig. 1(b) that the heat release rate (dQ/dθ) is more in pure diesel mode in comparison to dual fuel mode with biogas. The main reason seems that the biogas has a lower heating value as compared to diesel fuel which results in poor combustion. It may also be noticed that experimental results [23] have slightly lesser heat release rate than theoretical one. Fig. 2(a) shows the effect of blending of fuels on brake power and brake mean effective pressure at 1500 rpm at fuel injection Timing of 332^0 CA. The brake power and brake mean effective pressure increase with an increase in methane substitution and in case of (biogas + hydrogen) blending up to 60% as shown in Table 3, the brake power and brake mean effective pressure increase and after it decreases due to more increase in biogas percentage. Although under biogas mode, both decrease throughout with an increase in biogas substitution. However, due to better combustion characteristics and also lower temperature in the intake manifold, the engine power increases when the methane or (biogas + hydrogen) are added in diesel. Generally methane or (biogas + hydrogen) will ignite faster than diesel, therefore engine burning methane or (biogas+ hydrogen) would produce slightly more power. It is also possible to take the advantage of the higher octane rating for methane or biogas as shown in Table 4. This would increase the efficiency of converting the potential combustion energy into mechanical power. The addition of methane or (biogas + hydrogen) blend with diesel show higher brake mean effective pressure compared to pure diesel because of the unavailability of CO2 in the chemical composition of methane and high concentration of hydrogen which improves the combustion efficiency. However in Fig. 2(b), the brake torque and friction power increase with an increase in methane substitution as well as with (biogas + hydrogen) till 60 %, then decrease due to increase in biogas substitution.

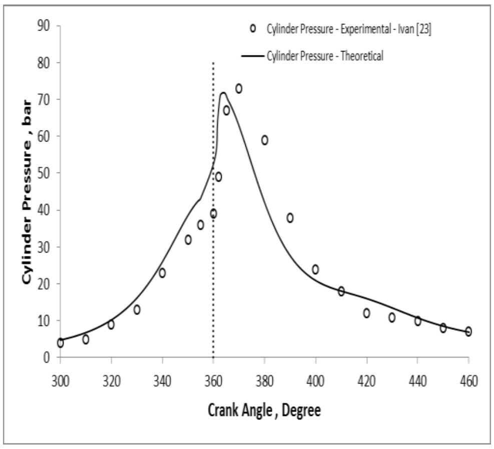
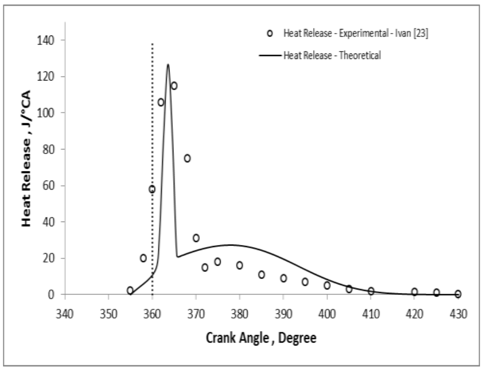


Figure.1.(a) Computed and Experimental cylinder pressure-crank angle diagram for dual fuel diesel engine.



Figute.2. (b) Computed and experimental heat release and crank angle diagram under dual fuel mode.

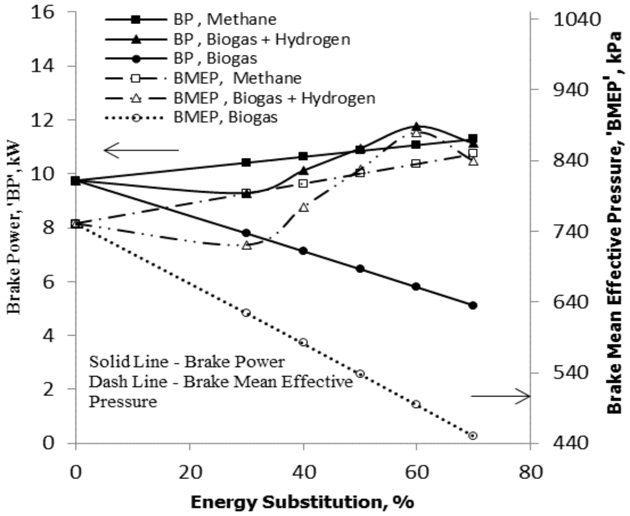


Figure.3. (a) Effect of blending of fuels on brake power and brake mean effective pressure.

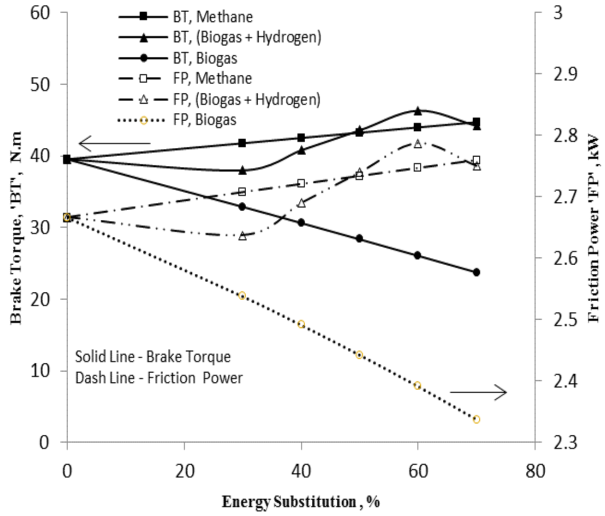


Figure.4.2.(b) Effect of blending of fuels on brake torque and friction power.

Fig. 2(c) shows the effect of injection timings on brake power and brake mean effective pressure for different fuel blends. Up to 70% maximum energy substitution, the brake power and brake mean effective pressure increase with an increase in injection timing till 340° and after that it decrease with an increase in injection timing further. Methane and (biogas + hydrogen) have the higher value of brake power and brake mean effective pressure compared to pure diesel and (diesel + biogas) blend.

In the beginning as the injection timing is advanced, the injection of fuel is early, which produces less value of brake power and brake mean effective pressure. Further near TDC, as it is retarded timing to inject the fuel as the piston is moving down from TDC to BDC for an expansion stroke, so all the heat is not utilized in converting into useful work but even after some time the exhaust valve is opened and rest of the heat is transferred to the exhaust, so experiencing less brake power and brake mean effective pressure.

In **Fig. 2(d)** the brake torque and friction power increase with an increase in injection timing till 340° and then decrease with an increase in injection timing further. Methane and (biogas + hydrogen) have the higher value of torque and friction power compared to pure diesel and (diesel + biogas) for the same reasons.

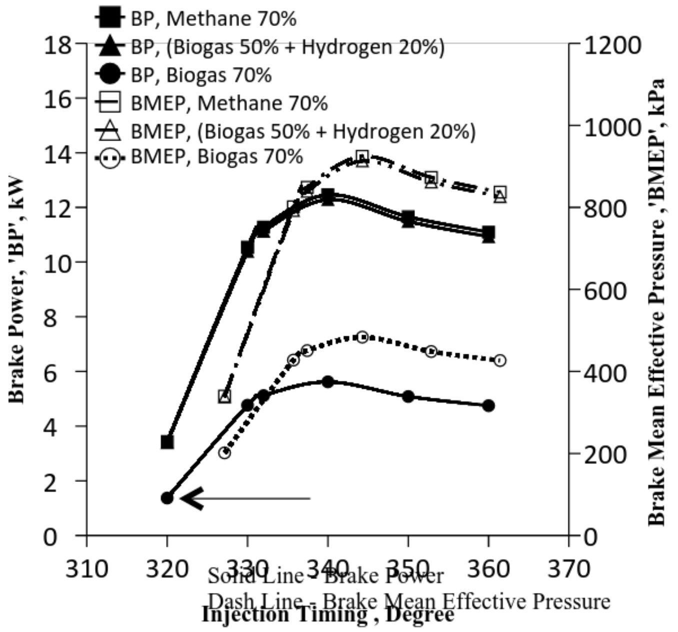


Figure.5. 2(c) Effect of injection timings on brake power and brake mean effective pressure.

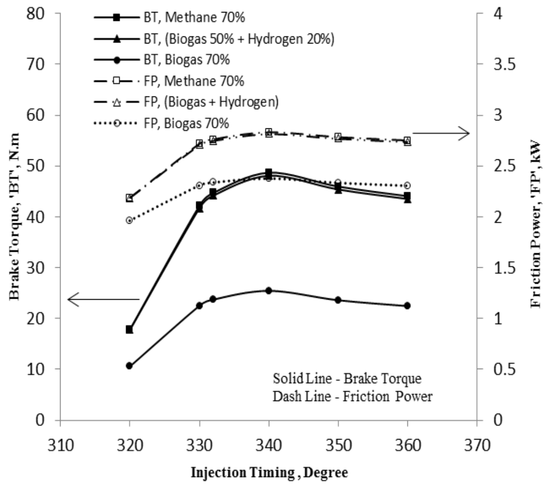


Figure.6. 2. (d). Effect of injection timings on brake torque and friction power.

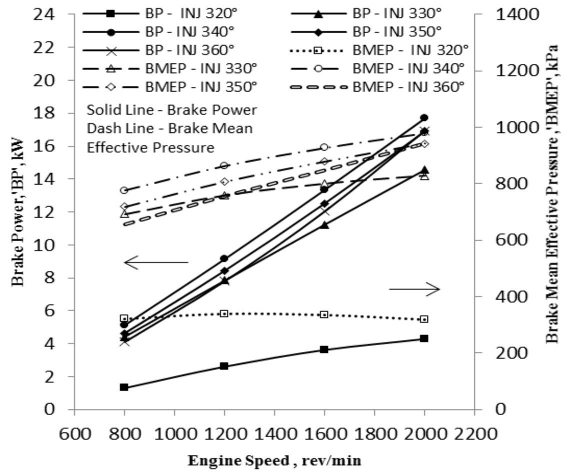


Figure.7 . 2(e). Brake power and brake mean effective pressure for (biogas 50% + H_2 20%).

Fig. 2(e) shows the effect of engine speed for a maximum of 70% energy substitution (biogas 50% + hydrogen 20%) on brake power and brake mean effective pressure for various fuel injection timings.

It is observed that for all engine speeds ranging from 800 rev/min up to 2000 rev/min with an increment of 400 rev/min, the optimum injection timing is 340^o or 20° BTDC. Similarly for brake mean effective pressure, the injection timing 20° BTDC gives the maximum value.

It is due to the fact that if injection timings are quite advanced like much before TDC, the maximum heat is released during the period when the piston is reaching toward TDC from BDC and when it reverts back on its expansion stroke from TDC to BDC, the piston is facing less pressure of the heat released of the hot burnt gas during combustion, so giving less power as well as less mean effective pressure.

If the injection timing is retarded too much like injection of fuel is taking place near TDC, then by the time the piston is returning back from TDC to BDC, the combustion is still taking place and all the fuel is not burnt completely, so again facing less pressure of the burnt gases and by the time the fuel is burnt completely, the exhaust valve starts opening and some of the heat is transferred to the atmosphere without being used through the exhaust valve.

Fig. 2(f) also shows the effect of engine speed on brake torque and friction power. It can be seen from the figure that injection timing 20° BTDC gives the maximum brake torque for all the engine speeds.

It can further be seen that friction power is also maximum at injection timing 20° BTDC. The injection timing 40° BTDC gives lowest value of brake torque as well as friction power.

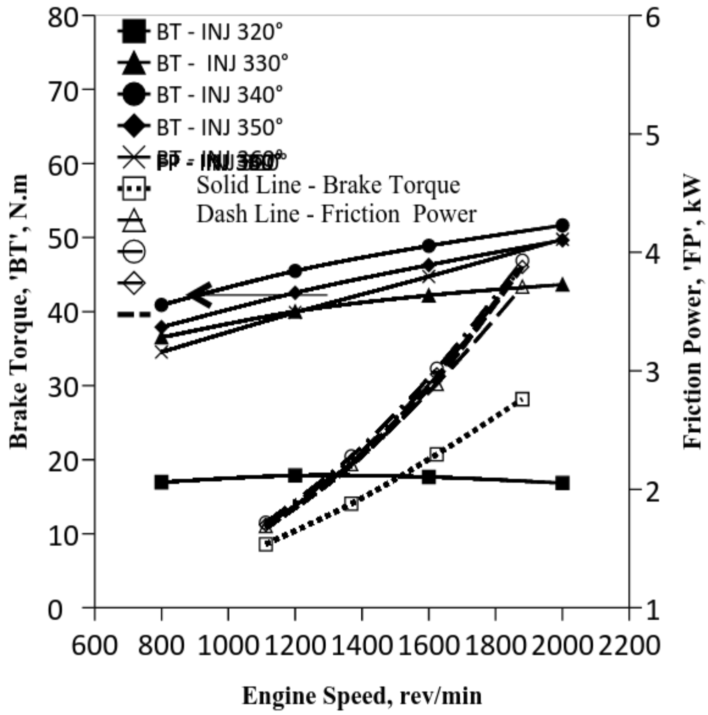


Figure.8. 2(f). Brake torque and friction power for (biogas 50% + hydrogen 20%).

Table. 3. Fuel properties [48].

Properties	Diesel Oil	Methane in Natural Gas	Biogas	Hydrogen
Composition	C = 84.8 % H2 = 15.2 % (By weight)	CH4 = 86.4 - 95.0 C2H6 = 6.0-3.0 C3H8 = 2.0-0.35	CH4 = 60 - 70 % CO2 = 30 - 40 % CO = 0.18 % H2 = 0.18 %	H2=100%
Lower Heating Value (MJ/Kg)	42	47.141	20.67	120
Density Kg/m^3	840	0.79	0.91	0.09
Flame Speed cm/sec	2.0 - 8.0	34	25	265 - 325
Octane Number	-	130	120	-
Molecular weight	144	16.043	-	-
Auto-ignition temperature °C	280	595	814	585
Stoichiometric air fuel ratio	14.92	17.233	10	34.3

4.3 Simulated results

The ignition delay in dual fuel mode is higher than under pure diesel mode. It may be due to the reason that a larger amount of biogas fuel in the intake and compression process and the reduction of charge temperature compared to pure diesel mode. Further hydrogen has a very high heating value so it helps in lowering down the delay period due to its fast combustion characteristics. The ignition delay periods calculated under different fuel blends are summarized in Table 4.

Table.4. Ignition delay period in degrees crank angle at full load at 1500 rpm.

Pure Diesel	Diesel+70% Methane	Diesel+70% Biogas	Diesel+50% Bio+20% H2
15.8	19.6	21	18.7

II. CONCLUSIONS

A computational model for evaluating the engine performance under neat diesel and blends of diesel with biogas, methane and biogas and hydrogen is developed. The combustion characteristics are poor under biogas diesel dual fuel mode because of having low heating value, although a little percentage of hydrogen addition enhances better combustion characteristics. The power drop as depicted under biogas diesel dual fuel mode can be improved with hydrogen substitution in a limited proportion as the overall heating value of mixture is increased which cause better combustion. Since biogas chemical composition is mainly methane and CO_2, so the existence of CO_2 in biogas leads to deterioration in overall engine performance in terms of low engine power. Engine performance improved with using biogas and hydrogen blend with diesel, whereas the better results of brake power, brake thermal efficiency, brake torque and brake mean effective pressure under maximum fuel substitution (biogas 50% + hydrogen 20%) were observed at injection timing of 20° BTDC and engine speed of 1600 rpm.

Nomenclature

Tav	Average temperature (k)
Pav	average pressure (bar)
S	stroke (m)
N	engine speed (rpm)
V	cylinder volume (m^3)
DV	change in volume (m^3)
VcL	clearance volume (m^3)
D	diameter of cylinder (m)
L	connecting rod length (m)
P	pressue (bar)
m	mass (kg)
R	gas constant (J/kg K)
T	temperature (k)
$dQ/d\theta$	heat release rate (J/^{o}CA)
Qp	heat release during premixed burning (j)
Qd	heat release during diffusion burning (j) MP
MP	shape factor for premixed burning
MD	shape factor for diffusion burning
θ	Crank angle (degrees)
θp	Combustion duration for premixed burning (degrees)
θd	Combustion duration for diffusion burning (degrees)
CM	mean piston speed (m/s)
ϕ	Fuel – air equivalence ratio

Acronyms

CH_4	methane
DSL	Diesel
CO_2	carbon dioxide
H_2	hydrogen
N_2	Nitrogen
NOx	oxides of nitrogen
PM	particulate matter
CO	carbon monoxide
UHC	unburnt hydro carbon
EGR	exhaust gas recirculation
CI	compression ignition engine
SI	spark ignition engine
CNG	compressed natural gas
TDC	top dead centre

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