Numerical Investigation of Diesel Engine Performance Under Different Fuel Mixed Modes

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Abstract - A thermodynamic numerical simulation model is developed to predict the performance of a direct injection four-stroke single cylinder diesel engine under different dual fuel modes. Diesel engine performance using neat diesel and blends of diesel with biogas, methane, and (biogas + hydrogen) in a dual fuel mode was investigated. The results were generated for a maximum energy substitution of biogas up to 70% and methane 70% with diesel 30% each. The simulated results of present study are found in good agreement with the published experimental results. The results obtained indicate that the combustion and performance characteristics are poor for (biogas + diesel) in dual fuel mode, however a small percentage of hydrogen addition in blend (biogas + diesel) significantly enhances their combustion and performance characteristics. Brake power and torque of blend DSL(20)BIO(60)H2(20) are comparable to neat diesel. As methane possesses better burning quality, it improves combustion characteristics and therefore methane blend DSL(40)CH4(60) produces higher brake power and torque than neat diesel. For maximum energy substitution, blends DSL(30)CH4(70) and DSL(30)BIO(50)H2(20) show highest brake thermal efficiency of nearly 39% and lowest brake specific energy consumption of about 15 MJ/kWh at an injection timing of 20^{o} BTDC. Injection timing of 20° BTDC gives maximum brake thermal efficiency as well as minimum brake specific energy consumption at different engine speeds for all fuel blends.

Keywords: Engine simulation, diesel engine, performance, alternative fuels

NOMENCLATURE

BIO	: Biogas
BPW	: Brake Power
BSEC Consumption	: Brake Specific Energy
BMEP	: Brake Mean Effective Pressure
BTE	: Brake Thermal Efficiency
BSEC Consumption	: Brake Specific Energy
BPW	: Brake Power
CA	: Crank Angle
CR	: Compression Ratio
D	: Diameter of Cylinder
DSL	: Diesel

HRR	: Heat Release Rate
ID	: Ignition Delay
LHV	: Lower Heating Value
MT	: Total mass
XMF	: Mass of the fuel
L	: Connecting Rod Length
N	: Engine RPM
P	: Cylinder Pressure
R	: Gas Constant
S	: Stroke Length
T	: Temperature
V_{CL}	: Clearance Volume

Θ	: Crank Angle
Φ	: Equivalence Ratio
CV_f	: Heating value of fuel
A_H, A_P, A_S	: Areas for head, piston and sleeve
T_H, T_P, T_S	: Wall Temperatures in K
Tg	: Gas temperature in K
XMFD	: Fraction of diesel burnt
XMFBIO	: Fraction of biogas burnt
XMFH2	: Fraction of Hydrogen burnt
XMFCH4	: Fraction of Methane burnt
CVD	: Heating value of diesel
CVBIO	: Heating value of biogas
CVCH4	: Heating value of methane
CVH2	: Heating value of hydrogen
ENSUB	: Energy substitution

SFC	: Specific fuel consumption
$\dot{U}$	: Rate of change of internal energy of the system of mass (m)
$P\dot{V}$	: Rate of mechanical work done at the boundary
$\sum_{i=1}^{n} \dot{Q}_i$	: Rate of heat transfer through the boundary at location i
$\sum_{i=1}^{n} \dot{m}_i h_i$	: Energy conversion in or out of the system at location i

1. INTRODUCTION

With increasing demand of clean energy and in view of excess usage of conventional fuels, it is now of utmost importance to think over other possible fuels which are renewable and less expensive.

The importance of using biomass based renewable fuels such as biogas has increased in rural areas. The increased use of biomass will also help to reduce pollution. Biogas is basically a methane gas produced through the anaerobic fermentation of animal waste and other organic waste. It is also possible to remove carbon dioxide and traces of hydrogen sulfide from biogas and then the fuel enriched gas can be compressed and filled in the cylinders [1].

Biogas with approximately 60% methane and 40% carbon dioxide has been hailed as the oldest technology as it satisfies several criteria for the basic need of cooking. Scientifically, biogas is an excellent technology that provides both fuel and manure.

Utilization of biogas is a feasible solution where organic waste is available in abundance. This technology besides solving energy crisis, will also reduce problems of waste disposal and ultimately will help improve environmental cleanliness.

Gaseous fuels are burnt in diesel engines under dual fuel mode because of their high self-ignition temperature by considering diesel as a pilot fuel and gas as a secondary fuel. First, combustion is initiated with diesel possessing an intense source of energy and later gas is gradually allowed to enter the cylinder and combustion starts taking place under dual fuel mode. Hydrogen possessing a very high heating value accelerates combustion and improves the burning quality.

Many researches have been done in last few decades on gaseous fuels utilizing more efficiently gaseous fuels in dual fuel mode such as CNG [2-10], LPG [11- 15], producer gas [16-21], biogas [22-27] and hydrogen [28-30]. Generally low NOx and PM emitted through the combustion of gaseous fuels. However, at part load operating conditions dual fuel engine with high substitution levels shows lower thermal efficiency than diesel engine and high CO and UHC emissions. Biogas contains substantial amount of methane and CO_2. Therefore, using biogas in engine increases amount of CO_2 concentration that deteriorates combustion inside the engine cylinder and results in reduced engine performance. For this reason, pure methane gas (without CO_2) was also used to analyze the engine performance parameters.

2. OBJECTIVE

The objective of present study is to predict the engine performance under different mixed fuel modes by developing a computer simulation model. Specifications of the diesel engine whose performance has been evaluated are given in table 1.

Table 1: Engine Specifications

Engine Type	Single Cylinder Diesel Engine - Kirloskar
Bore x Stroke	87.5 mm x 110 mm
Compression ratio (CR)	17.5:1
Rated Power	4.4 kW @ 2000 rpm
Maximum Torque	39 Nm at 1500 rpm
Inlet Valve Open	4.5^o bTDC
Exhaust Valve Close	4.5^o aTDC

3. Engine Cycle Simulation

The computer simulation model is classified for assessing the engine performance based upon power cycle analysis.

The entire simulation has been divided into two periods. One is closed period, that corresponds to power cycle simulation and another one is open period that represents gas exchange process. Further, the power cycle consists of compression, combustion, and expansion process, while the open period includes the exhaust and intake process. The significant part of the engine cycle is represented by period in which both inlet and exhaust valves are closed, power is developed by the engine in this period. It has been assumed that engine power cycle starts from the point of inlet valve closing and end when exhaust valve opens.

The general form of energy equation for an open system [31] is given as:

$$\dot{U} = -P\dot{V} + \sum_{i=1}^{n} \dot{m}_i h_i + \sum_{i=1}^{n} \dot{Q}_i \quad (1)$$

Energy substitutions in terms of fraction of fuel burnt are expressed as;

$$ENSUBT = ENSUBBIO + ENSUBCH4 + ENSUBH2 \quad (2)$$

$$MT = (P * 1.0E + 05 * V)/(R * T) \quad (3)$$

$$XMF = (MT - MRG)/(AFR + 1.) \quad (4)$$

$$XMFD = XMF * (1. - ENSUBT) \quad (5)$$

$$XMFBIO = XMF * ENSUBBIO \quad (6)$$

$$XMFCH4 = XMF * ENSUBCH4 \quad (7)$$

$$XMFH2 = XMF * ENSUBH2 \tag{8}$$

$$XMFT = XMFD + XMFBIO + XMFCH4 + XMFH2 \tag{9}$$

Total heat liberated during combustion

$$TQ = (XMFD * CVD + XMFBIO * CVBIO + XMFH2 * CVH2 + XMFCH4 * CVCH4) * 1000000. \tag{10}$$

For premixed and diffusive combustion stages in diesel engine, (dQ/dθ) analysis was done by double Wiebe function [32]

$$\frac{dQ}{d\theta} = 6.9\frac{Q_p}{\theta_P}(MP+1)(\frac{\theta}{\theta_P})^{MP}\exp\left[-6.9(\frac{\theta}{\theta_P})^{MP+1}\right] + 6.9\frac{Q_d}{\theta_d}(MD+1)(\frac{\theta}{\theta d})^{MD}\exp\left[-6.9(\frac{\theta}{\theta_d})^{MD+1}\right] \tag{11}$$

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

Mean piston speed can be evaluated as

$$CM = \frac{SxN}{30} \quad \text{(m/s)} \tag{12}$$

Cylinder volume as a function of crank angle (CA) (θ) can be calculated through equation of slider crank mechanism and is given as:

$$V = Vcl + \frac{\Pi}{4}.D^2[L + \frac{S}{2}(1-\cos\theta) - \sqrt{L^2 - \frac{S^2}{4}\sin^2\theta}] \tag{13}$$

From the ideal gas equation, pressure can be calculated as

$$P = \frac{mRT}{V} \tag{14}$$

Summation of heat transfer rate is given as

$$\sum \dot{Q}_i = \dot{Q}_h + \dot{Q}_p + \dot{Q}_s \tag{15}$$

The heat transfer rate is calculated for all the three surfaces as

$$\dot{Q}_h = h\ A_h(T_g - T_h)$$
$$\dot{Q}_p = h\ A_p(T_g - T_p) \tag{16}$$
$$\dot{Q}_s = h\ A_s(T_g - T_s)$$

$\dot{Q}_h$, $\dot{Q}_p$ and $\dot{Q}_s$ are the heat transfer rate through head, piston and sleeve respectively.

The gas side heat transfer coefficient (h) is determined from Hohenberg's correlation [33] as

$$h = 130.V^{-0.06}\ P^{0.8}T^{-0.4}(CM + 1.4\ )^{0.8} \tag{17}$$

The heat transfer coefficient $(Watt/m^2K)$ is assumed to be same for all the three surfaces of the cylinder.

$\frac{\partial u}{\partial T}$ is calculated after differentiating the internal energy equation as given below:

$$u = C_{11}T_g + C_{12}T_g{}^2 + C_{13}T_g{}^3 + C_{14}T_g{}^4 + C_{15}T_g{}^5 \tag{18}$$

$$\frac{\partial u}{\partial T} = C_{11} + 2C_{12}T_g + 3C_{13}T_g{}^2 + 4C_{14}T_g{}^3 + 5C_{15}T_g{}^4 \tag{19}$$

Values of constants used in above energy equation are given in table 2.

Table 2: Values of Constants

Constants	Values in kJ
C11	0.6919943
C12	$-0.3917296\text{x}10^{-4}$
C13	$0.5292534\text{x}10^{-7}$
C14	$-0.2286286\text{x}10^{-10}$
C15	$0.277589\text{x}10^{-14}$

Further, new value of cylinder volume at an increment of θ may be given as

$$\dot{V} = (V_{\theta+\Delta\theta} - V_{\theta}) \quad (20)$$

Brake power (BPW) is given by

BPW = IPower – TFPower (21)

The indicated power (IPower) is calculated by

IPower = P x DV x N x 100 / (2 x 60) (22)

Friction mean effective pressure (FMEP) and later total friction power (TF Power) can be used to determine friction of the engine parts [34] as

FMEP = 75 + 48 x (N/1000) + 0.4 x (2 x S x 10^{-2} x N/ $60)^2$ (23)

TF Power = FMEP x 1000. x S x 10^{-2} x (PI/4) x D^2 x 10^{-4} x N/ (60 x 1000) (24)

Brake thermal efficiency (BTE) for diesel fuel is given by

BTE = (BPW x 60) / (MFD x CVD x 1000 x N) (25)

BTE for biogas blend is given by

BTE = (BPW x 60) / ((MFD x CVD + MFBIO x CVBIO) x 1000 x N) (26)

Brake specific energy consumption (BSEC) can be determined as

BSEC = {(MF x CV_f x N x 1000)} / BPW (27)

Brake torque (BT) can be calculated as

BT = BPW x 60 x 1000 / (2 x PI x N) (28)

Brake mean effective pressure (BMEP) and specific fuel consumption (SFC) are expressed as

BMEP = WD x 4/ (PI x D^2 x S) (29)

Work done is calculated as

WD = BPW x 60 / N (30)

$$SFC = (XMFTxNx3600.)/(BPWx60.) \quad (31)$$

The delay period is determined from the relation given by Watson et. al [35].

$$\text{Delay Period} = \frac{3.52e^{\left[\frac{2100}{T}\right]}}{P^{1.022}} \quad (32)$$

4. RESULTS AND DISCUSSION

Four different fuel compositions (diesel, biogas, methane and hydrogen) are considered in this research work to evaluate the engine performance under blending in a dual fuel mode.

Energy substitutions considered to assess the overall performance of a diesel engine are provided in Table 3 and their fuel properties are given in the study [36].

Table 3: Energy Substitution

(Diesel + Methane)		(Diesel + Biogas)		Diesel + (Biogas + H2)	
Diesel	Methane	Diesel	Biogas	Diesel	(Biogas + 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

Pressure CA diagrams under pure diesel and 70 % biogas substitution, obtained from the simulation as well as from experimental results are shown in figures 1 and 2 at engine rpm of 1500 and CR of 17.5 for injection timing 28^o BTDC under fuel/air equivalence ratio ϕ = 0.67. It may be observed that computed results of cylinder pressures are in good agreement with the experimental results [25].

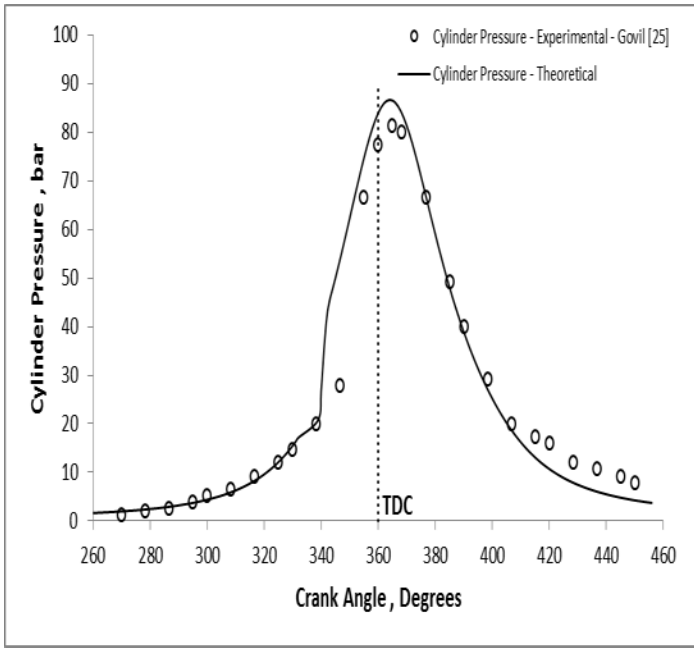

Figure 1: Cylinder pressure versus CA diagram under neat diesel operation.

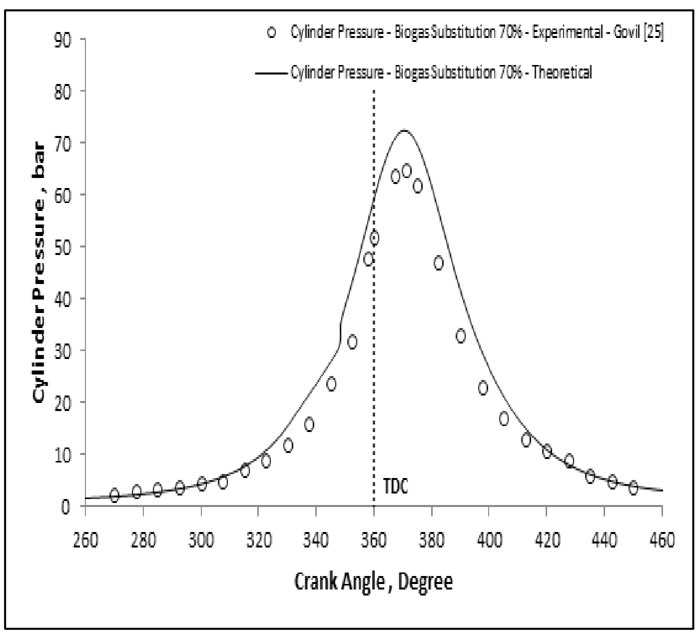

Figure 2: Cylinder pressure versus CA diagram under (diesel + biogas).

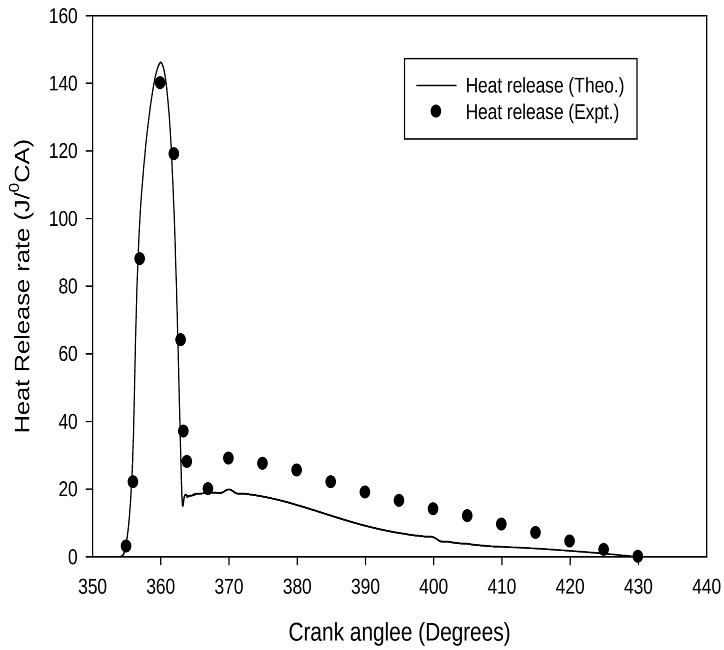

Figure 3: Computed and experimental [25] HRR versus CA diagram under neat diesel operation.

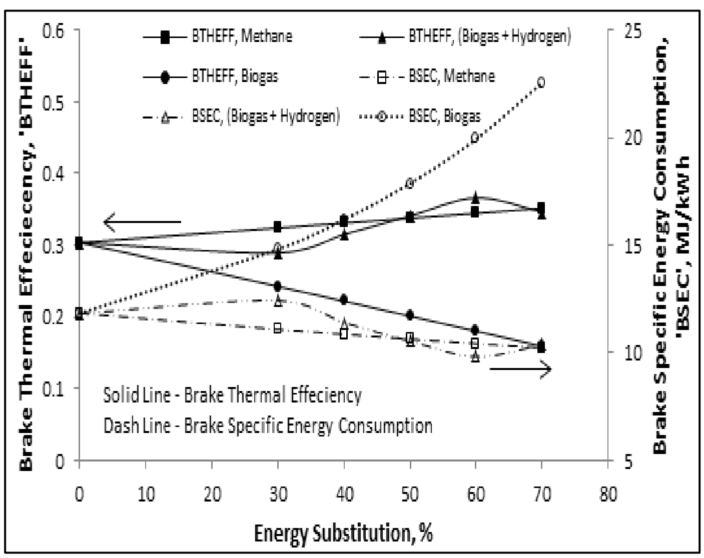

Figure 4: Blending effects of fuels on BTE and BSEC.

Figures 1 and 2 show the three processes inside the engine cylinder such as compression, combustion, and expansion. Biogas 70% shows the lowest value of pressure. It is due to the reason that pure biogas as such contains CO_2 in its chemical composition, which causes deterioration in combustion and ultimately reflect on poor engine performance.

A comparison of the experimental and simulated results of HRR with respect to CA under pure diesel mode is shown in figure 3 for the same operating conditions. Due to long ignition delay (ID) period, peak value of HRR during premixed combustion under dual fuel mode have been observed. In long ID period, a large portion of pilot fuel was mixed with air producing a rapid heat release. Although under diffusion combustion, quite appreciable change in the results can be observed.

Figure 4 shows the effect of blending on BTE and BSEC. The BTE increases with an increase in substitution of methane and for (biogas + hydrogen) up to 60% substitution and after that it decreases slightly as further biogas is added in the blend. Also, it may be noticed that under (biogas + diesel) mode, the BTE shows a decreasing trend throughout as biogas has a much lower heating value. Although, BSEC for methane and (biogas + hydrogen) up to 60% energy substitution show a decreasing trend. BSEC for blend (biogas + diesel) shows an increasing trend throughout the energy substitution as biogas has a lower flame velocity and exhibits slower combustion. The (biogas + hydrogen) blend improves the burning quality and experiences better combustion.

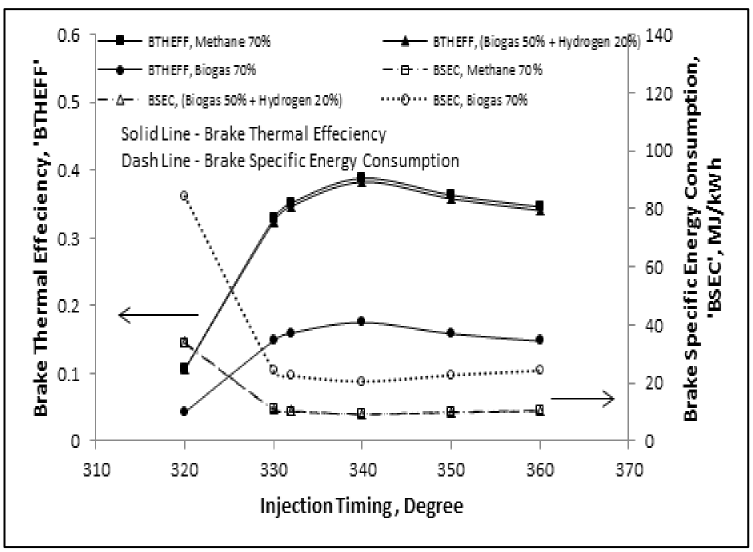

Figure 5: Effect of injection timings on BTE and BSEC.

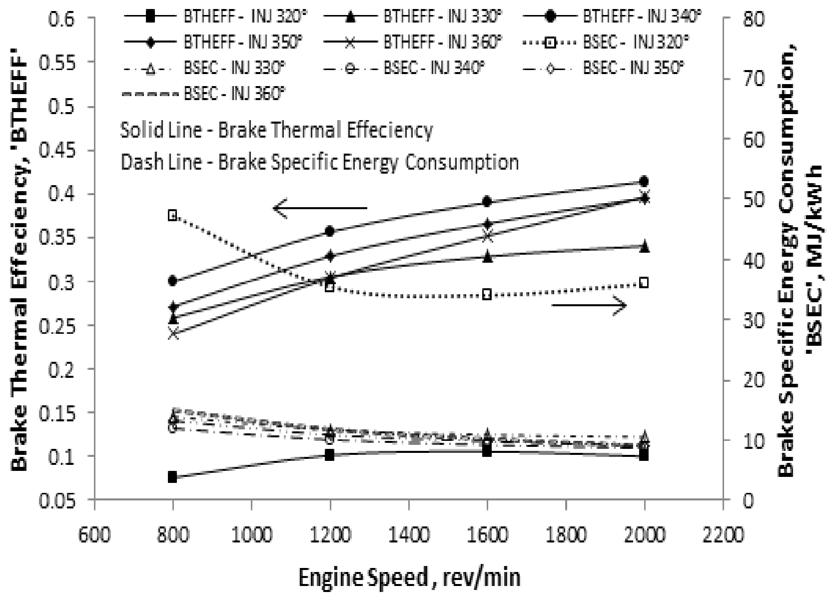

Figure 6: BTE and BSEC for (biogas 50% + hydrogen 20%).

It can be seen in figure 5 that the BTE increases with an increase in injection timing up to 340° CA and then it decreases for all fuel blends. An opposite trend is observed for BSEC. Further, blends DSL(30)CH4(70) and DSL(20)BIO(50)H2(20) have the higher values of BTE and lower value of BSEC compared to pure diesel and (diesel + biogas) substitution. It can also be noticed from the figure 5 that BSEC for methane and (biogas + hydrogen) overlap approximately. This means that blending of hydrogen in a certain proportion with biogas is a good solution to improve the engine performance (heating value of the fuel blend is increased to give relatively better combustion characteristics).

A similar trend can also be observed in figure 6 for BTE and BSEC for different injection timings

and engine speeds. An injection timing of 20° BTDC gives maximum BTE as well as minimum BSEC for different engine speeds. The optimum value of injection timing 20° BTDC gives the satisfactory results of the engine performance.

Figures 7, 8, 9, and 10 depict the variation in engine BPW and torque for different fuel blends under dual fuel mode. Figure 7 shows that the BPW is slightly increased under diesel and methane mixed running mode. It is due to improved combustion characteristics of methane. On the other hand, biogas is a slow burning gas, thereby experiences slow combustion and causes power loss. A 20 %

hydrogen addition in the diesel and biogas fuel blend shows a noticeable improvement in brake power. In figure 8, at low engine speed (i.e., 1500 rpm) similar characteristic of brake power can be observed with lower values. Blending diesel with methane i.e., DSL(40)CH4(60) gives higher torque compared to blend DSL(20)BIO(60)H2(20), but torque of neat diesel (DSL100) and blend DSL(20)BIO(60)H2(20) are comparable to each other as shown in figure 9. In figure 10, similar torque behavior is observed with lower values at lower engine speeds.

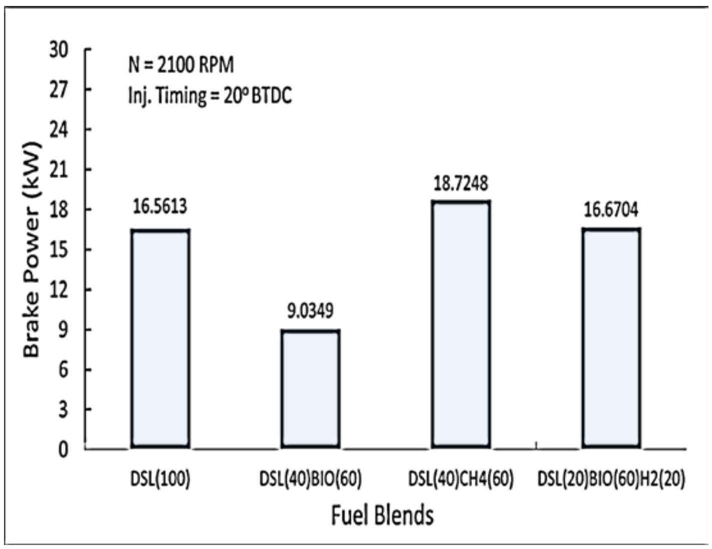

Figure 7: BPW with various fuel blends at 2100 rpm.

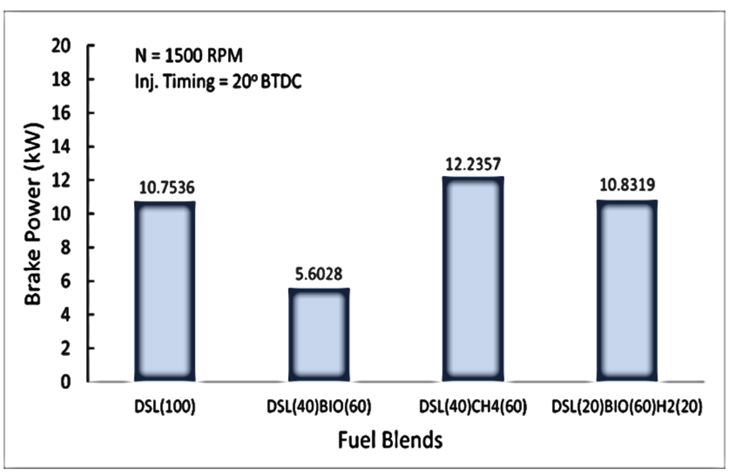

Figure 8: BPW with various fuel blends at 1500 rpm.

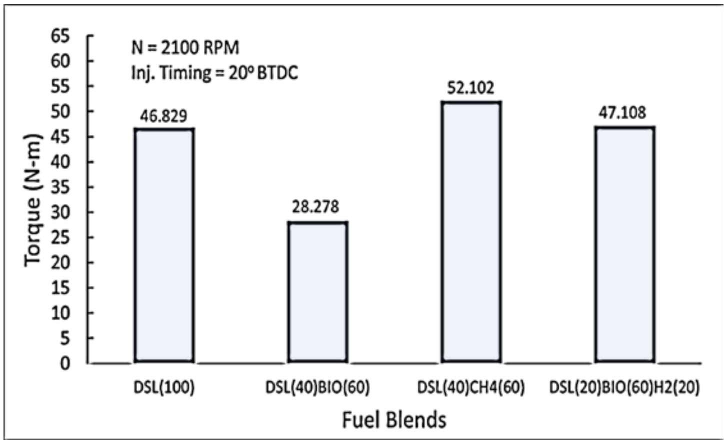

Figure 9: Engine torque with various fuel blends at 2100 rpm.

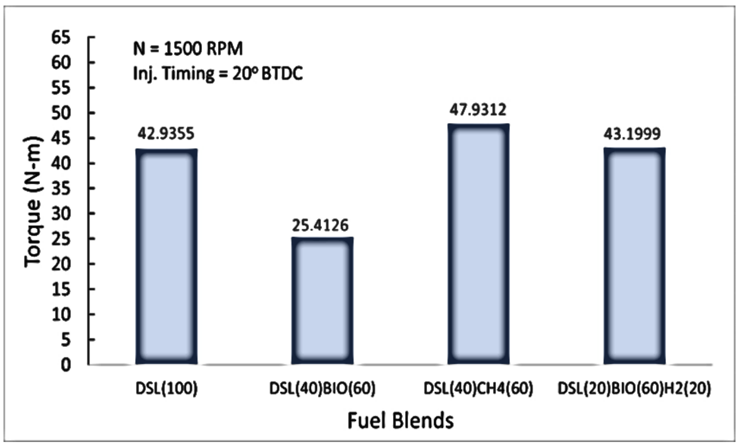

Figure 10: Engine torque with various fuel blends at 1500 rpm.

Figure 11 shows the effect of different engine speeds on torque and brake mean effective pressure (BMEP) under pure diesel mode for an injection timing of 30^o BTDC and CR of 21 and validation with the study [24]. The brake torque and BMEP increase with an increase in engine speed and reaches a maximum at a certain engine speed and the it decreases due to mechanical loss which has been more significant. Figure 12 also depicts the effect of engine speed on BTE and SFC under the same conditions. It can be seen that BTE increases gradually up to 1800 rpm and beyond that it nearly shows same trend as similar to SFC.

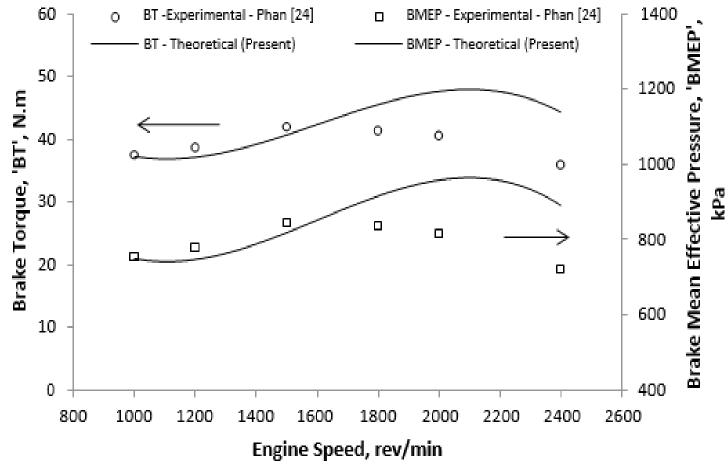

Figure 11: Computed and Experimental [24] Brake Torque and BMEP with engine speed for pure diesel.

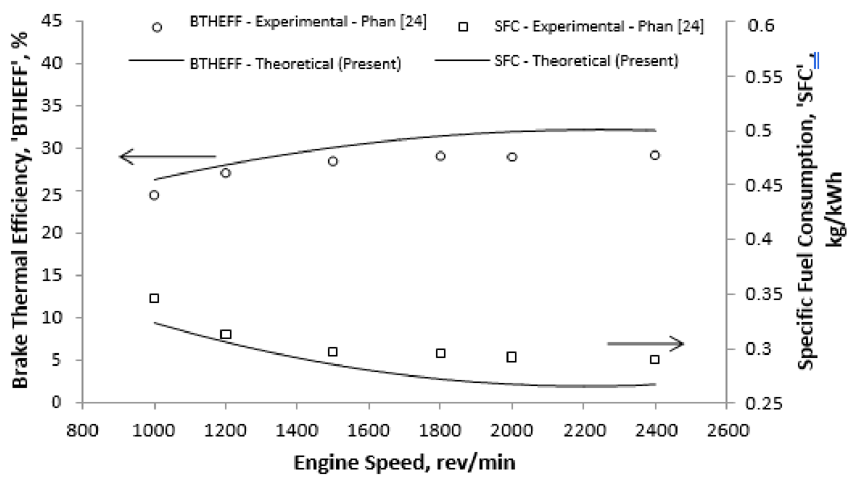

Figure 12: Computed and Experimental [24] BTE and SFC with engine speed for pure diesel.

Figure 13 shows the results for BPW and BSEC with different injection timings at 1700 rpm. It is predicted that around 20^o BTDC, the minimum BSEC occurs and thereby the BPW is maximum. Figure 14 shows the effect of injection timing on ID and peak cylinder pressure under dual operation. As the injection timing is advanced the delay period and peak cylinder pressure increase, as sufficient time is given for proper combustion and maximum energy is released for doing useful work during expansion stroke.

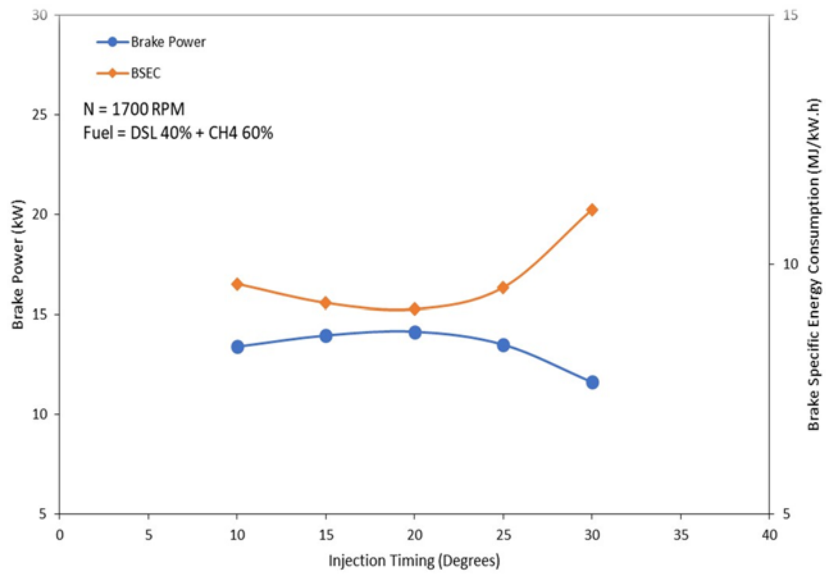

Figure 13: Effect of Injection Timing on BPW and BSEC.

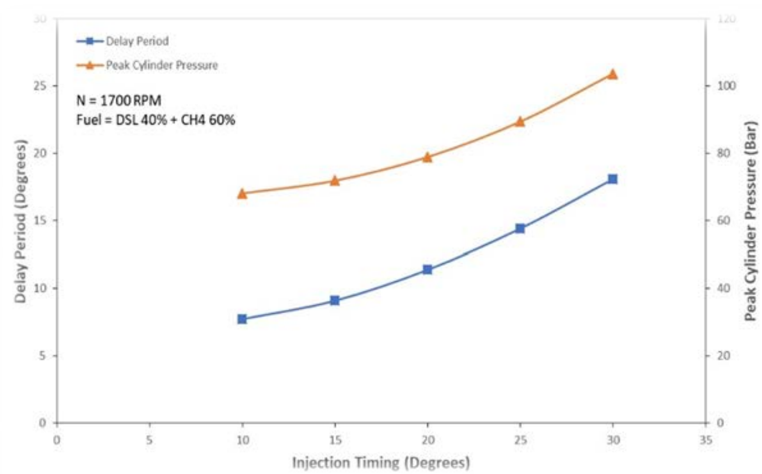

Figure 14: Effect of Injection Timing on ID and Peak CP.

5. Conclusions

In this work, diesel engine performance using neat diesel and blends of diesel with biogas, methane, and (biogas + hydrogen) in a dual fuel mode was evaluated through a computational model. The basic approach lies to satisfactorily simulate the engine performance under dual fuel mode rather than going for experiments. The following conclusions can be drawn from the present study:

- It is found that for a maximum energy substitution of 70%, combustion and performance characteristics are poor for (biogas + diesel) in dual fuel mode due to low heating value of biogas. However, a small percentage of hydrogen addition in blend (biogas + diesel) enhances their combustion and performance characteristics.

- Increase in energy substitution through methane addition in diesel fuel significantly improves engine performance.
- It is also found that an injection timing of 20° BTDC gives maximum BTE as well as minimum BSEC at different engine speeds for all fuel blends. Moreover, blends DSL(30)CH4(70) and DSL(20)BIO(50)H2(20) have the highest values of BTE i.e., nearly 39% and lowest BSEC of about 15 MJ/kWh.
- For blend DSL(30)BIO(50)H2(20), BTE increases and BSEC decreases continuously with increase in engine speed at all injection timings.
- It is also inferred that BPW and torque are highest for the methane blend i.e., DLS(40)CH4(60). BPW and torque of blend DSL(20)BIO(60)H2(20) are comparable to neat diesel.
- Advanced injection timings resulted in increase in ignition delay period and peak cylinder pressure.

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