

Theoretical Investigations on Utilization of Alternative Fuels in an IDI Diesel Engine under Dual Fuel Mode

Nafis Ahmad*, Nawaf Al Khamis, Abbas Lal Khan

Mechanical Engineering, King Abdulaziz University, Jeddah, Saudi Arabia

ABSTRACT

The study explores the impact of ammonia and DME on the combustion / performance characteristics and exhaust emissions of a dual-mode IDI diesel engine without making any major alterations to the engine. Fortran-77 programming is used to develop a computer simulation program. The findings show that blending in dual mode raises pre and main chamber temperatures and pressures, as well as IMEP and BP, and lowers BSEC as ammonia substitution increases from 5% to 25%. Blending (diesel + ammonia) increases BP by 15.93 percent, 37.25 percent, and 72.42 percent in dual mode at 20%, 40%, and 60% NG substitutions respectively, and blending (diesel + DME) increases BP by 17.79 percent, 40.17 percent, and 76.99 percent in dual mode at 20%, 40%, and 60% NG substitutions respectively.

Keywords: IDI diesel engine, dual mode, blending, performance parameters, combustion characteristics.

Nomenclature

A	Ammonia	DI	Direct injection
ATDC	After top dead centre	DME	Dimethyl ether
BP	Brake power	EVO	Exhaust valve open
BSEC	Brake specific energy consumption	HC	Hydrocarbon
Blending	(Diesel + Ammonia / DME)	IMEP	Indicated mean effective pressure
BTDC	Before top dead centre	IDI	Indirect injection
CA	Crank angle	IC	Internal combustion
CO	Carbon monoxide	NG	Compressed natural gas
CO ₂	Carbon dioxide	NO	Nitric oxide
D	Diesel	TDC	Top dead centre
Dual mode	(Diesel + NG) mode		

Numbers represent the percentages e.g., D40NG60A0 means 40% diesel, 60% NG, 0% A

1. INTRODUCTION

As the researchers switching over to green energy in the future, alternative fuels are becoming increasingly crucial. The availability of the alternative fuel in a country is the most important factor to consider when choosing an alternative fuel [1]. Typical automotive fuels such as gasoline and diesel, emit harmful pollutants such as CO₂, nitrogen oxides (NO_x), particulate matter (PM) and soot, all of which contribute to the environmental pollution [2]. HC's have been the principal fuel in diesel engines since their inception, and have undergone various design and development changes. The researchers first goal was on increasing engine power and efficiency but in recent decades, the importance of diesel engine emissions in climate change has alarmed scientists working on IC engines optimization [3]. Nitrogen mixes with oxygen when the temperature inside an IC engine's combustion chamber hits 1600 °C, it results in the formation of NO_x and soot and these emissions are one of the major factors in the gradual decline of air quality [4].

Although conventional fossil fuels such as diesel and gasoline have better thermal efficiency and are easier to use as fuels in IC engines but their impact on the atmosphere, environment and pollution have led to the exploration of alternative/renewable fuels such as compressed natural gas, liquefied petroleum gas, liquefied natural gas, bio diesels and so on. In terms of operational cost and pollutants, NG in dual mode outperforms conventional fuels [5]. Other advantages of employing NG in dual mode include the fact that current engines just require modest modifications and knocking impact is reduced at high compression ratios [6-7]. Experimental work performed by [4] found that by using 40% NG in a dual mode reduces soot and NO_x by 74 percent and 54 percent respectively. Because of these benefits, the automotive industry is considering NG as a viable alternative to conventional fuels and the automotive vehicles switching from diesel to NG have increased over the period of time [8]. Ammonia contains no carbon, it is a CO₂ free fuel as well as a hydrogen (H₂) [9]. H₂ is one of the cleanest

energy sources used as a fuel around the world but it has obstacles in terms of storage, transportation and extraction. Ammonia, on the other hand, is easy to make due to the abundance of nitrogen in the atmosphere and liquid ammonia is easy to store. It also evaporates under atmospheric conditions due to its 10.2 bar saturated vapor pressure at 25°C [10]. Furthermore, in the 1960s, ammonia was employed for the first time as a fuel in an IC engine and effective combustion of ammonia in a diesel engine resulted in a 10% increase in BP as compared to HC fuel under the same conditions [11] and [12]. Due to its low flame speed and resistance to self-ignition, the use of ammonia as a single fuel in IC engines is not suggested because in that case, very high compression ratios are required for successful combustion of ammonia in IC engines [13]. Ammonia is also a cost-effective chemical as its price is comparable to that of diesel and gasoline [14]. CI engines can use DME as a replacement for diesel because of its cetane number of 55-60. DME's physical qualities are similar to those of liquefied petroleum gas (LPG), making it one of the world's cleanest fuels. Moreover, DME helps reduce PM, Sulphur oxides (SO_x), HC and CO in diesel

engines as compared to emissions generated by conventional fuels in IC engines [15].

The primary purpose of the research is to identify the best alternative fuels such as NG, ammonia and DME and to use them in an IDI diesel engine without any major modification as well as to determine engine performance. Since IDI diesel engines are less polluting than DI diesel engines.

2. METHODOLOGY

A computer simulation model is developed using Fortran-77 code. The program simulates the entire IDI diesel engine cycle including intake, compression, fuel injection, combustion and exhaust processes. The influence of various operational factors, such as fuel injection timings, varying percentages of NG, ammonia and DME, on the combustion characteristics and engine performance output, such as temperatures and pressures history, BP, IMEP and BSEC, are investigated. The zero-dimensional model is used to construct the program. The engine specifications and fuel properties are given in Table 1 and Table 2 respectively.

Table 1: Diesel Engine Specifications

Diameter of cylinder (D _c) x Stroke (S)	8.42 cm X 10.22 cm
Compression ratio (CR)	21:1
Connecting rod length (L)	17.165 cm
Injection Timing	14° BTDC
Rated power under naturally aspirated	16.3 kW @ 1700 RPM
Injector Opening Pressure	120 bar
Engine type	Water cooled, naturally aspirated IDI diesel engine
No. of cylinders (n _c)	4

Table 2: Fuel properties

Properties	Diesel [11], [16]	NG [16]	Liquefied ammonia [10]	Liquefied DME [15], [17], [18]
Lower heating value Q_{LHV} (MJ/kg)	43	47.1	18.6	28.8
Stoichiometric air-fuel ratio	15.0	17.2	6.1	8.9
Ignition temperature (K)	505	813	935	508
Density (kg/m ³)	817	0.75	602.8	667
Flame speed (cm/s)	82	38	7	50

2.1 MATHEMATICAL EQUATIONS

In order to simulate the performance of an indirect injection diesel engine under different fuels blending modes, a Triangular Heat

Release Method is used to determine the mass of fuel burnt [19].

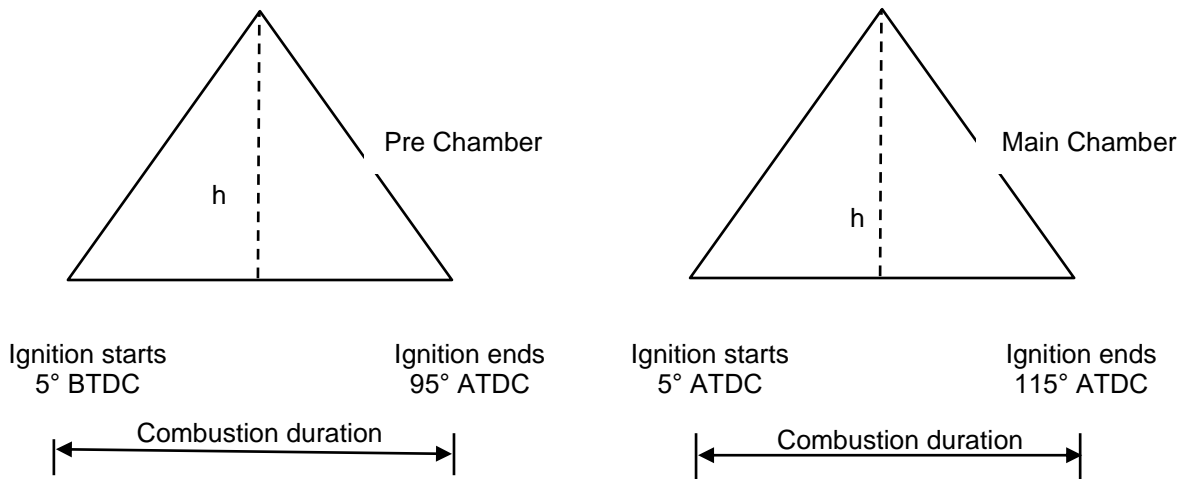


Figure 1: Triangular Heat Release Method for pre and main chambers.

$$\text{Total mass of fuel} = m_f = \text{Area of Triangle}$$

$$\text{Total mass of fuel} = m_f = \frac{1}{2} \times \text{Combustion duration} \times h \quad (1)$$

Figure 1 depicts the Triangular Heat Release Method used in the pre-chamber and main chamber to determine the mass of fuel burned

and maximum heat release. The beginning and ending locations of the triangles represent the start of ignition during the combustion stroke

and the end of ignition during the expansion stroke respectively. The whole combustion period is represented by the bottom line of the triangle and 'h' is the maximum height at the specified crank angle that can be calculated as:

$$h = \frac{2 \times \text{Area of Triangle}}{\text{Combustion Duration } (^{\circ}\text{CA})} \quad (1)$$

Similarly, maximum heat release 'h' can be determined for each $^{\circ}\text{CA}$.

$$\begin{aligned} S_{CA} \\ = L + (S/2)(1 - \cos(CA)) \\ - \sqrt{L^2 - (S^2/4)(\sin^2(CA))} \end{aligned} \quad (4)$$

$$\begin{aligned} \frac{dS}{dCA} \\ = (S/2)(\sin(CA)) \\ + \left[\frac{(S^2 * \cos(CA) * \sin(CA))}{4} \right. \\ \left. * \sqrt{L^2 - (S^2 * \sin^2(CA)/4)} \right] \end{aligned} \quad (5)$$

The cylinder volume (V_c) at any CA is given by:

$$\begin{aligned} V_c \\ = V_{cl} \\ + (\pi/4)(D_c^2) \left[L \right. \\ \left. + (S/2)(1 - \cos(CA)) \right. \\ \left. - \sqrt{L^2 - (S^2/4)(\sin^2(CA))} \right] \end{aligned} \quad (6)$$

$$\begin{aligned} \frac{dV_c}{dCA} \\ = (\pi/4)(D_c^2) \left[(S/2)(\sin(CA)) \right. \\ \left. + \left\{ \frac{(S^2 * \cos(CA) * \sin(CA))}{4} \right\} \right. \\ \left. * \sqrt{L^2 - (S^2 * \sin^2(CA)/4)} \right] \end{aligned} \quad (7)$$

The change in internal energy w.r.t T1 and T2 are given by:

The swept volume (V_{sw}) and the clearance volume (V_{cl}) of the engine can be calculated by:

$$V_{sw} = \pi \times D_c^2 \times S/4 \quad (2)$$

$$V_{cl} = V_{sw}/(\text{CR} - 1) \quad (3)$$

The stroke at any CA (S_{CA}) is calculated as [20]:

$$\begin{aligned} dU_m/dT_1 = & 0.16528 + 2 \\ & * 0.93563 * 10^{-5} \\ & * T_1 + 3 \\ & * 0.12641 * 10^{-7} \\ & * T_1^2 - 4 \\ & * 1.054607 \\ & * 10^{-10} * T_1^3 + 5 \\ & * 0.66301 \\ & * 10^{-15} * T_1^4 \end{aligned} \quad (8)$$

$$\begin{aligned} dU_p/dT_2 = & -0.013623 * \phi - 2 \\ & * 0.22689 * 10^{-4} \\ & * T_2 * \phi + 3 \\ & * 1.051419 \\ & * 10^{-7} * T_2^2 * \phi \\ & - 4 * 0.47836 \\ & * 10^{-12} * T_2^3 * \phi \end{aligned}$$

Where

T_1 and T_2 are temperatures of main and pre

chambers respectively. ϕ

= Equivalence ratio

Heat transfer coefficient [21] for main chamber (h_m) and pre chamber (h_p) are determined as:

$$\begin{aligned} h_m (W/m^2.K) = & 130 * P_m^{0.8} * \\ & (C_M + 1.4)^{0.8} * T_m^{-0.4} * \\ & V_m^{-0.06} \end{aligned} \quad (9)$$

$$h_p (W/m^2.K) = 130 * P_p^{0.8} * (C_M + 1.4)^{0.8} * T_p^{-0.4} * V_p^{-0.06} \quad (10)$$

C_M = Mean piston speed (m/s). V_m and V_p represent total Main chamber volume and Pre chamber volume (m^3) respectively.

Heat input (HI) is obtained by:

P_m and P_p represent Main and Pre-chamber pressure (bar) respectively,

T_m and T_p represent Main and Pre-chamber temperature (K) respectively,

$$HI (KW) = m_f (kg/s) \times Q_{LHV} (MJ/kg) \times 10^{-3} \quad (11)$$

The indicated power (IP) is given by the following equation:

$$IP (KW) = \frac{P_m \times dV \times N \times 10^2}{2 \times 60} \quad (12)$$

Frictional mean effective pressure (FMEP) [22]:

$$FMEP (bar) = \left(75 + \left(\frac{48 \times N}{1000} \right) \right) + \left(0.4 + \left(\frac{2 \times S \times N}{100 \times 60} \right)^2 \right) \quad (13)$$

Frictional power (FP) in 'KW' is obtained by:

$$FP = \frac{FMEP \times 1000 \times S \times \pi \times D_c^2 \times N}{100 \times 10000 \times 60 \times 1000} \quad (14)$$

Brake power is represented by:

$$BP (KW) = IP - FP \quad (15)$$

Brake specific energy consumption (BSEC) and Brake thermal efficiency are related to each other and are obtained by:

Brake Thermal Efficiency (%) =

$$3.6 \times 100/BSEC \quad (17)$$

$$BSEC (MJ/KWh) = \frac{HI}{BP} \times 3.6 \quad (18)$$

Indicated mean effective pressure is calculated by:

$$IMEP (bar) = \frac{IP \times 2 \times 60 \times 4}{\pi \times D_c^2 \times S \times N \times n \times 100} \quad (19)$$

ENGINE CYCLE SIMULATION

The engine cycle modeling has been divided into two parts. One for the power cycle simulation and another for the gas exchange process. Furthermore, the compression, combustion and expansion processes are all part of the power cycle whereas the exhaust and intake processes are part of the open phase. The portion of the engine cycle when both the inlet and exhaust valves are closed is the most important because the engine generates power during this time period. As a result, the power cycle has been assumed to

begin with the inlet valve closing and end with the exhaust valve opening.

For an open system, the general form of energy equation [23] is expressed as follows:

$$U = -PV + \sum_{a=1}^n m_a h_a + \sum_{a=1}^n Q_a \quad (20)$$

Where,

U = rate of change in internal energy of the system mass

PV = rate of mechanical work done at the boundary

$\sum_{a=1}^n m_a h_a$ = energy conversion at location "a" from system in/out

$\sum_{a=1}^n Q_a$ = rate at which heat transfer occurs through the boundary at location "a"

Figures 2 and 3 show the schematic diagrams of all fuels and their blends. In Fig. 2, diesel is replaced by NG at first, resulting in dual mode and then ammonia (5% to 25%) is added to replace diesel at fixed 20%, 40% and 60% NG substitutions in dual mode. Similarly, DME is substituted instead of ammonia as depicted in Fig. 3.

3. RESULTS AND DISCUSSION

Figure 4 shows the experimental data used to validate theoretical cylinder pressures. It can be observed that theoretical values almost reflect the similar pattern as that of experimental results. The IDI diesel engine used for validation [24] is naturally aspirated with 22:1 compression ratio, 24° BTDC injection timing and 2400 RPM engine speed.

Figure 5 represents the pressures in the pre and main chambers with crank angle for pure diesel case. Under turbocharged conditions, the intake pressure is 1.3 bar. The pre and main chamber pressures are nearly equal at the start of the compression stroke, but as the ignition begins, the pre chamber pressure rises above the main chamber pressure because the ignition occurs first in the pre chamber and then the mixture (air + fuel) moves through the passageway to the main chamber. The difference in pressures is recorded near TDC (where the compression stroke terminates). The expansion stroke begins as the piston advances from TDC to BDC, and the pressures begin to fall until the exhaust valve opens.

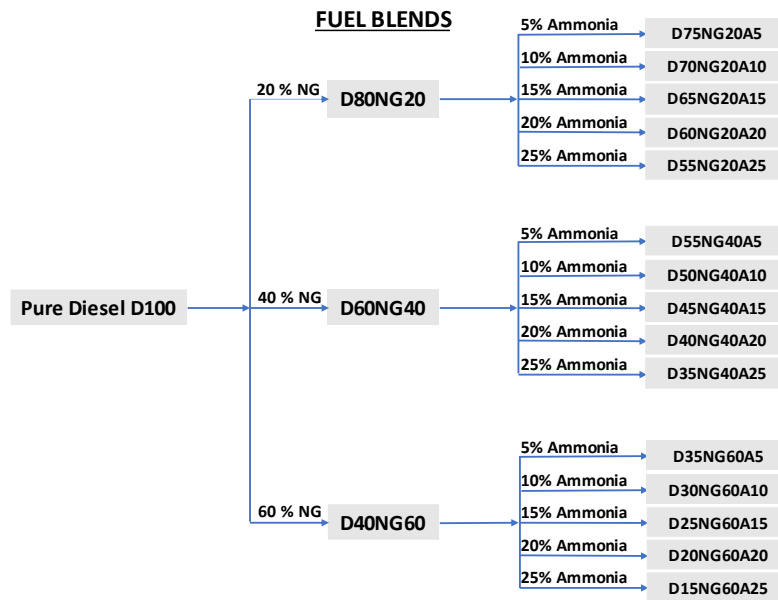


Figure 2: Schematic diagram of fuels (D+NG+A)

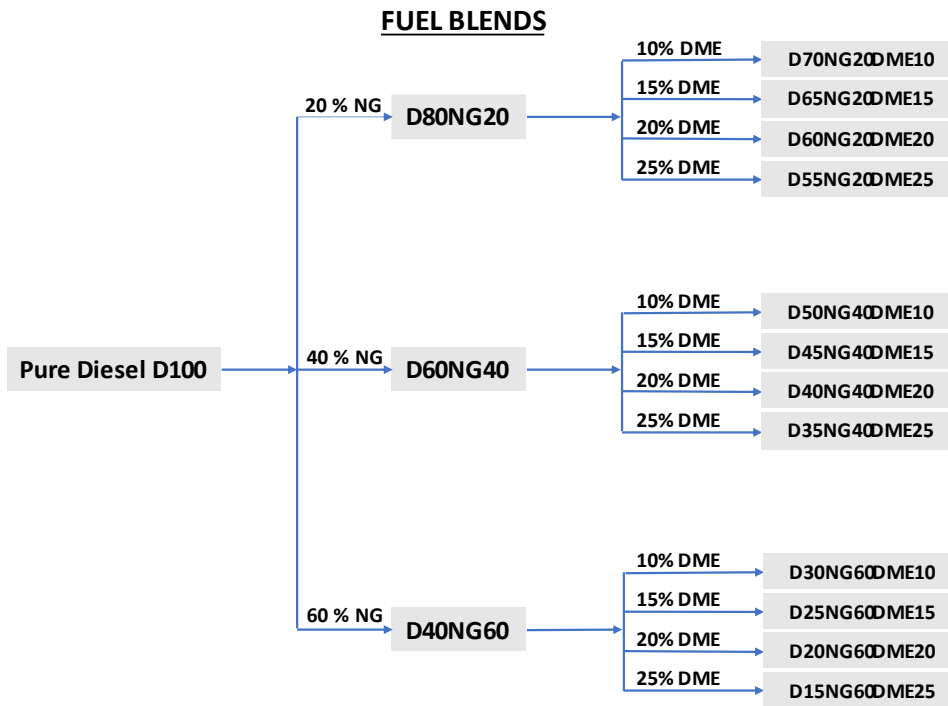


Figure 3: Schematic diagram of fuels (D+NG+DME)

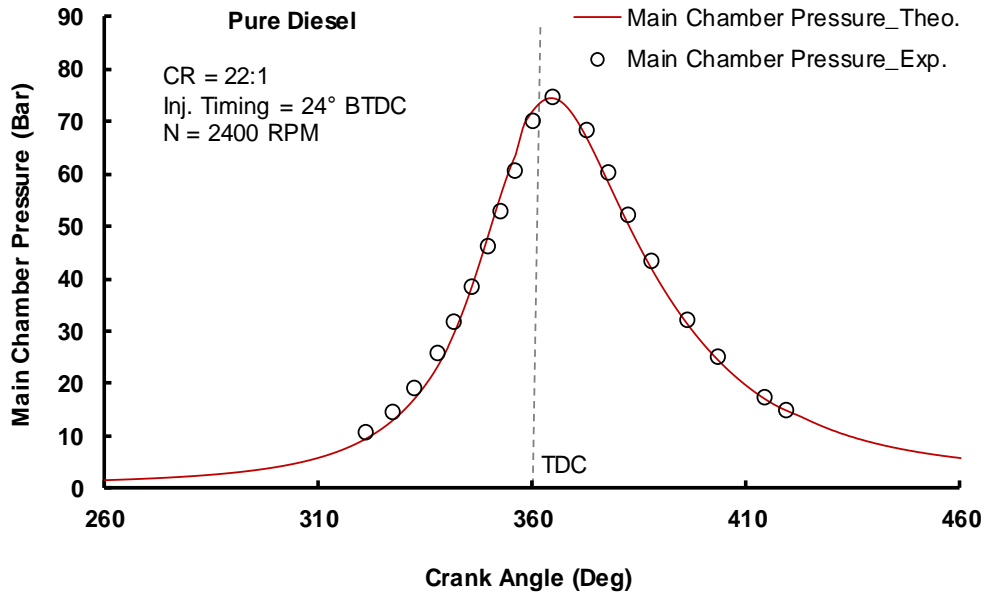


Figure 4: Experimental and theoretical comparison of cylinder pressures with crank angle.

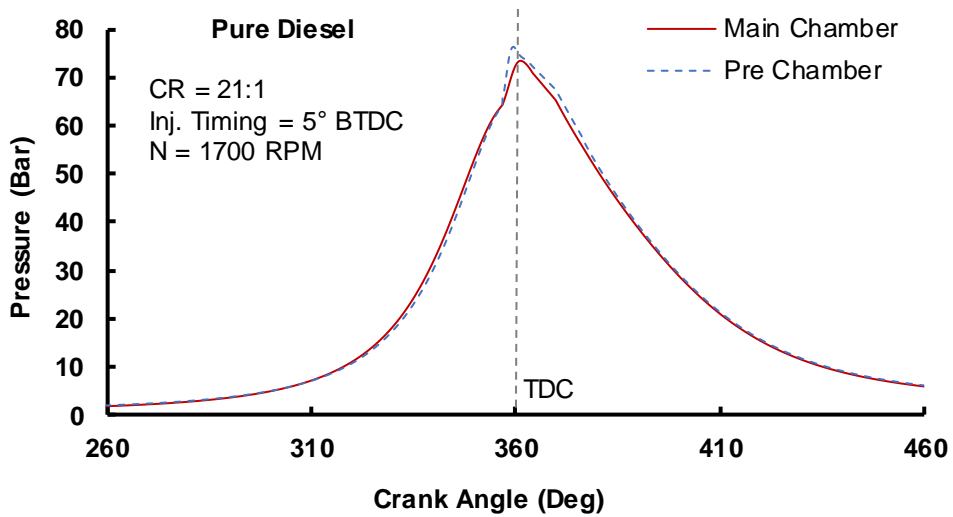


Figure 5: Cylinder pressure in pre and main chambers with crank angle.

The temperature profiles of the pre and main chambers for the pure diesel are shown in figure 6. In the case of temperature profiles, the maximum temperatures of the pre and main chambers occur during the expansion stroke and the temperature of the pre chamber reaches to maximum first, followed by the main chamber temperature for the same reason as mentioned in figure 5. The heat losses caused by hot gases moving from the pre-chamber to the main chamber through the passageway are also a factor in lowering of pre-chamber temperature during expansion stroke.

Figure 7 demonstrates how the crank angle affects the mass flow rate in the pre and main chambers. During the intake stroke, air enters into the main chamber as the piston gets closer to TDC during the compression stroke. Similarly, during the expansion stroke, hot

burn gases stream through the passageway from the pre chamber to the main chamber delivering engine power to the engine's output shaft.

The change in mass flow for D100, D15NG60DME25 and D15NG60A25 as a function of crank angle is shown in Figure 8. The negative values indicate that pre chamber pressure is higher than the main chamber pressure. Since the ignition begins in the pre chamber in an IDI diesel engine that resulted in sudden pre chamber pressure rise and as the piston moves toward BDC during the expansion stroke, the pre chamber and main chamber pressures become closer to each other.

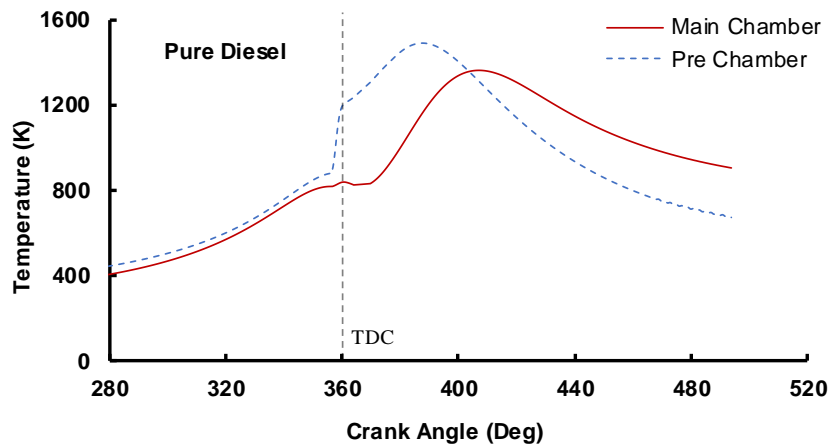


Figure 6: Cylinder temperatures in pre and main chambers with crank angle.

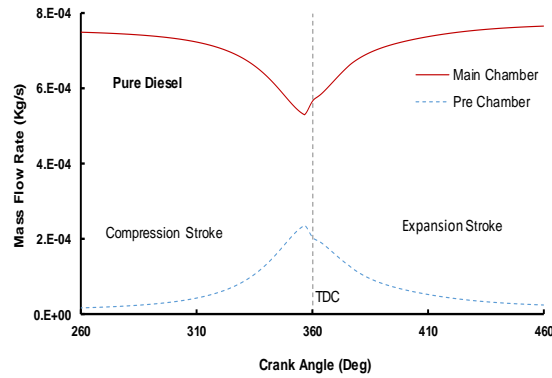


Figure 7: Mass flow rate in pre and main chambers with crank angle.

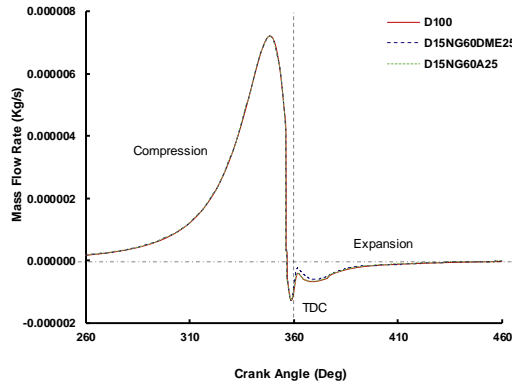


Figure 8: Variation of mass flow rate with crank angle for different fuel blends.

Figure 9 shows the trend of IMEP and BP change with injection timing. The power output of an engine with volumetric efficiency is best described by IMEP [12]. It is also a critical performance output parameter since it provides the average pressure produced in the combustion chamber throughout the course of the whole thermodynamic cycle [25]. Both IMEP and BP show their maximum values during the compression stroke which is at injection timing of 20° BTDC. The BP and IMEP decrease before and after that angle,

when the piston crosses 340° CA and moves towards TDC as well as during the expansion stroke, because the fuels are assumed to burn completely and generate their full heat energy before the piston touches TDC in order to achieve maximum power during the expansion stroke. As the ignition timing approaches TDC (retarded), the time required for complete burning of fuel decreases resulting in lower cylinder temperatures and pressures which reduces engine output. The engine's optimal injection timing is around 20° BTDC.

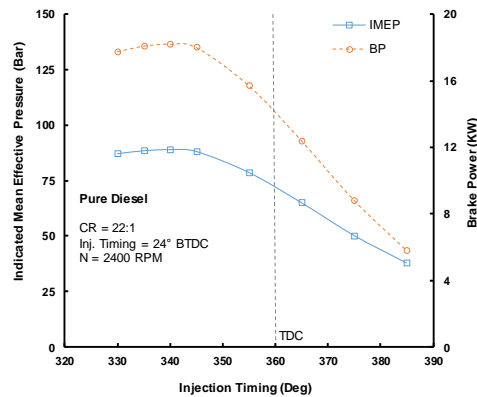


Figure 9: Variation of indicated mean effective pressure and brake power with injection timings.

Figure 10 displays the changes in engine output parameters with pure diesel and dual mode at a fixed 20% NG concentration. The temperature fluctuations in the pre and main chambers are depicted in Fig. 10 (a). Because of the rapid combustion, pure diesel has the highest temperature value among fuel blends and when the neat diesel percentage decreases, the temperature values decrease as well. Furthermore, when ammonia is added to the blending process, the temperature of the pre and main chambers rises because the blend becomes richer, reducing the mass of air and subsequently increasing the mass of fuel for the same equivalence ratio. The air to fuel ratio (AFR) increases when NG is used in dual mode, in that case, the combination becomes leaner as compared to pure diesel [7]. In Fig. 10 (b), the pressure values of the pre and main chambers nearly remain the same with the addition of 20% NG, whereas these values increase slightly with the addition of ammonia from 5% to 25% in blending. Figures 10 (c and e) exhibit the change in IMEP and BP with different fuel blends. Under dual mode and blending scenarios, pure diesel has the maximum IMEP and BP values. By mixing ammonia with diesel, these (IMEP and BP) parameters begin to rise due to the reason

presented in Fig. 10 (a). When NG is used in dual mode, the BP drops as compared to neat diesel case [7]. Fig. 10 (d) represents the variation of BSEC with fuel blends. BSEC is the ratio of HI to BP, therefore, pure diesel exhibits the lowest BSEC value because it has highest BP among fuel blends. Similarly, D80NG20A0 has maximum BSEC value because of its lowest BP.

In figure 11, NG is increased to 40% and remains fixed in all fuel blends. Figures 11 (a and b) show that by using 40% NG in dual mode, the temperature and pressure in the pre and main chambers drop because NG causes slow combustion than pure diesel case. The blending impact is identical to that shown in Fig. 10 (a) i.e., incorporating ammonia to the blending raises the temperatures of the dual mode's pre and main chambers. In Fig. 11 (b), it can be seen that for the same pure diesel case, pressure declines faster with 40 percent NG than with 20 percent NG (Fig. 10 (b)). Similarly, adding ammonia to the blending raises the pressures in the pre and main chambers. The pattern in figures 11 (c, d and e) is the same as in figures 10 (c, d and e). It can be observed that IMEP and BP values are lower and BSEC value is higher in figures

11 (c, d and e) as compared to figures 10 (c, d and e).

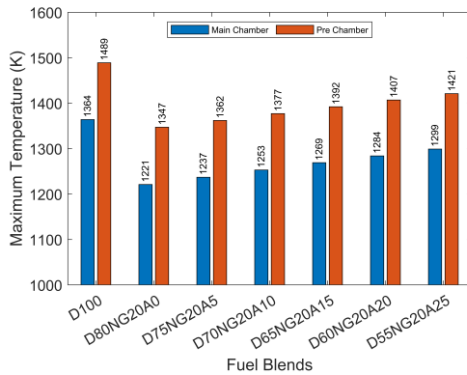
Figures 12 (a and b) show the temperature and pressure variations in the pre and main chambers in dual mode at a fixed 60% NG. It can be seen that the pre and main chambers temperature and pressure values drop as compared to figures 11 (a and b) and figures 10 (a and b) [4]. Again, by raising the proportion of ammonia in the blending increases the pressure and temperature

readings. Figures 12 (c, d, and e) and 11 (c, d, and e) have the same pattern. IMEP and BP have the lowest values in the 60% NG substitution in dual mode whereas BSEC has the highest value across all NG substitutions in dual mode.

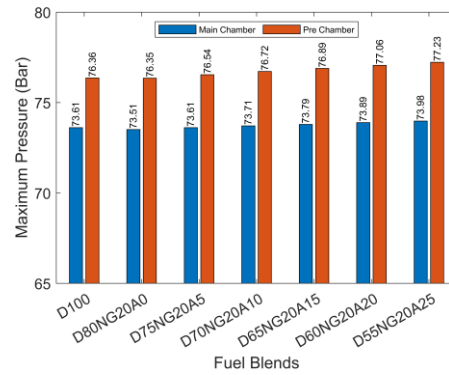
Figure 13 displays the impact of various fuel blends on BP. When compared to pure diesel, the BP drops in dual mode [7] but it increases with blending in dual mode. Effect of blending on BP is given in Table 3.

Table 3: Effect of blending on BP (KW) in dual mode

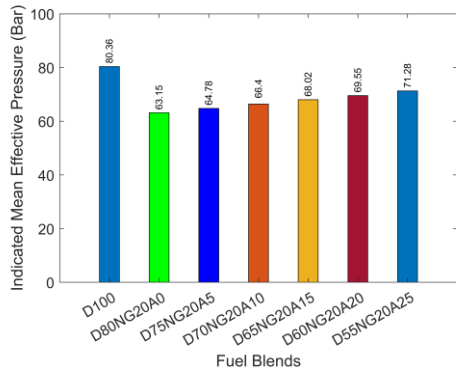
NG %	BP Dual Mode	Blending %	BP by Blending (Diesel + Ammonia)	% rise in BP by (Diesel + Ammonia) Blending	BP by Blending (Diesel + DME)	% rise in BP by (Diesel + DME) Blending
20	16.13	(55 + 25)	18.70	15.93	19.0	17.79
40	10.63	(35 + 25)	14.59	37.25	14.9	40.17
60	6.78	(15 + 25)	11.69	72.42	12.0	76.99



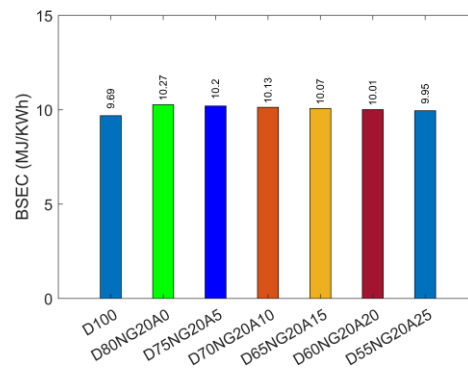
(a)



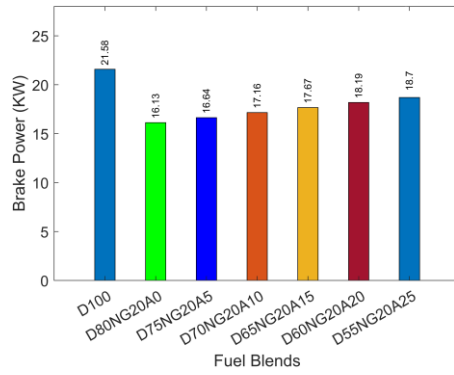
(b)



(c)

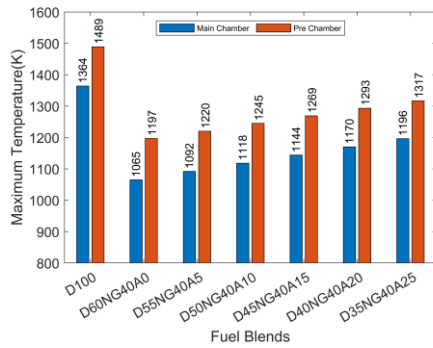


(d)

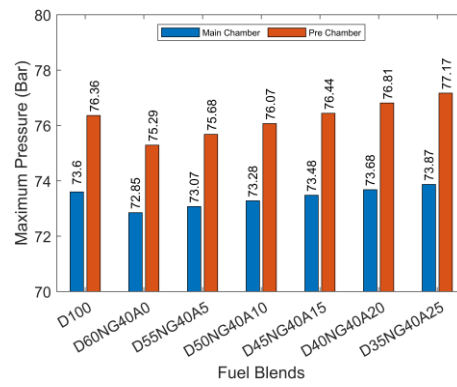


(e)

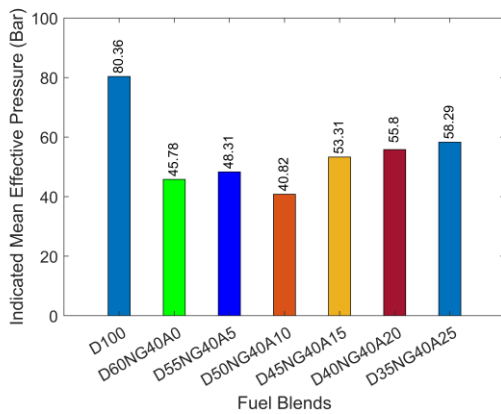
Figure 10: Variation of different parameters with fuel blends at fixed 20% NG. (a) maximum temperature in pre and main chambers (b) maximum pressure in pre and main chambers (c) indicated mean effective pressure (d) brake specific energy consumption (e) brake power



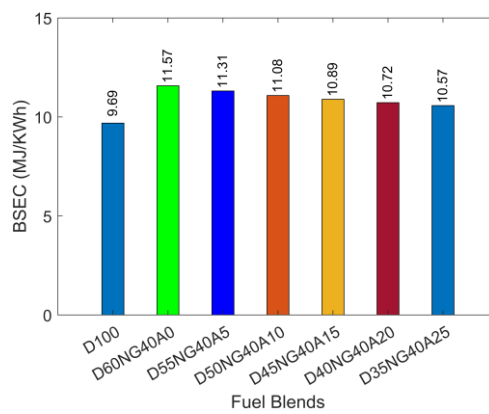
(a)



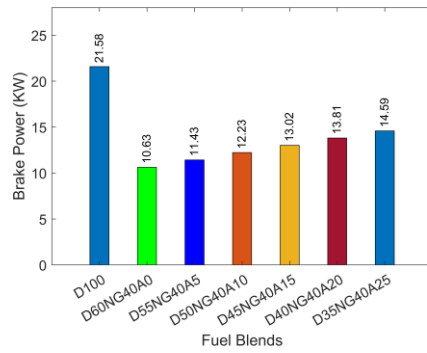
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(c)

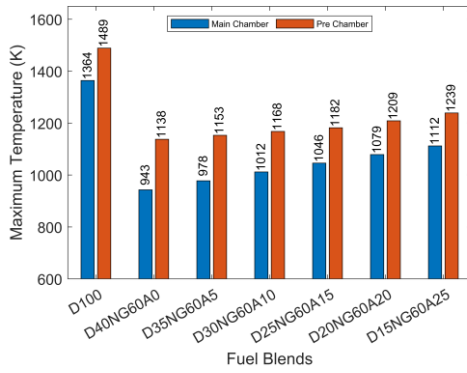


(d)

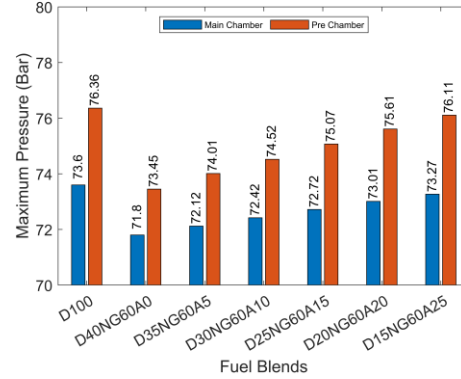


(e)

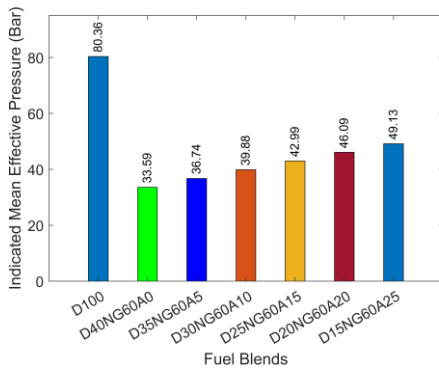
Figure 11: Variation of different parameters with fuel blends at fixed 40% NG. (a) maximum temperature in pre and main chambers (b) maximum pressure in pre and main chambers (c) indicated mean effective pressure (d) brake specific energy consumption (e) brake power



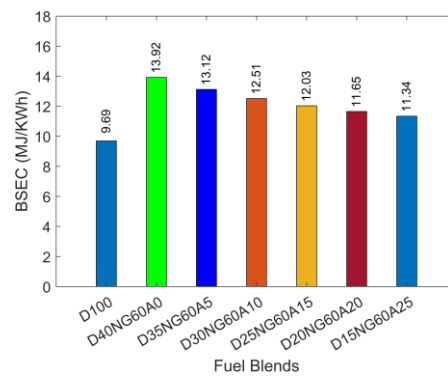
(a)



(b)



(c)



(d)

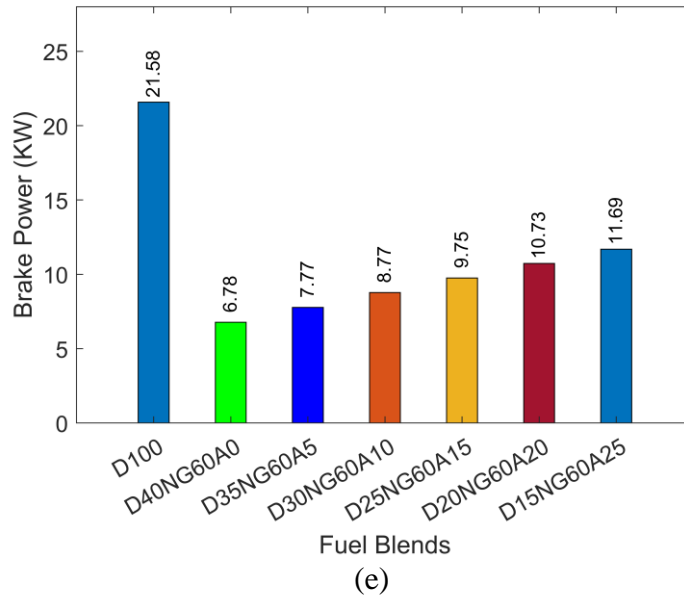


Figure 12: Variation of different parameters with fuel blends at fixed 60% NG. (a) maximum temperature in pre and main chambers (b) maximum pressure in pre and main chambers (c) indicated mean effective pressure (d) brake specific energy consumption (e) brake power

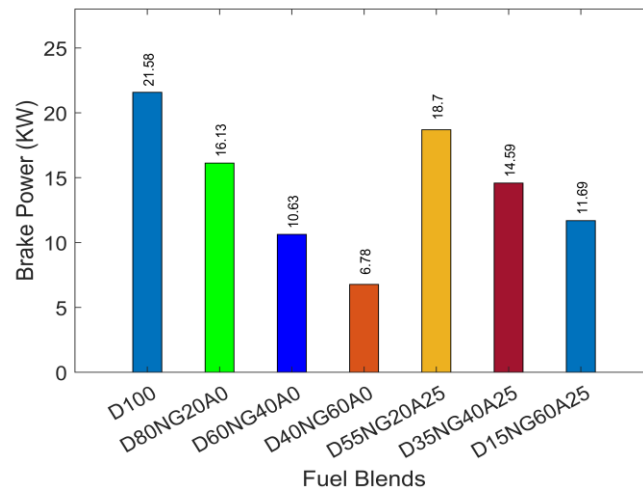


Figure 13: Brake power for different fuel blends.

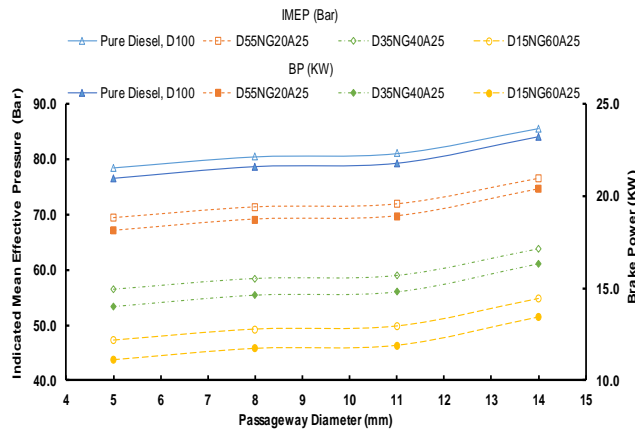


Figure 14: Variation of indicated mean effective pressure and brake power with passageway diameter for different fuel blends

The change in IMEP and BP parameters with passageway diameter is shown in figure 14. Because more fuel/mixture goes into the main chamber from the pre chamber as the passageway diameter increases, therefore, both IMEP and BP values increase with passageway diameter.

Figure 15 shows the variations in brake power and main chamber pressure for pure diesel, dual mode and diesel with (diesel + DME) blending. Figures (a, b and c) indicate the brake power of various fuel blends at fixed NG substitutions of 60%, 40% and 20%, respectively. Figures (d, e, and f), on the other hand, depict the main chamber pressure of fuel blends with 20%, 40% and 60% NG substitutions in dual mode respectively. The effects of DME and ammonia on brake power and main chamber pressure are also shown in figure 15. DME delivers slightly more BP than ammonia because when DME is mixed with diesel, the mixture becomes richer than (diesel + ammonia) blending, resulting in a higher mass flow rate within the chamber for the same equivalence ratio.

Figures 16-18 show the exhaust gas emissions at EVO for different fuel blends at injection timings of 10° and 20° BTDC. Both injection timings (10° BTDC and 20° BTDC) have

same pattern for exhaust emissions. Figure 16 shows that pure diesel has the lowest NO value whereas D40NG60 has the highest NO value among fuel blends, since the availability of oxygen is lowest in the former instance and largest in the latter case, resulting in the generation of more NO from the engine's exhaust. The amount of NO produced by the engine's exhaust increases as the mixture becomes leaner. Furthermore, blending in dual mode has little impact on NO emissions.

Figure 17 shows the change in CO levels with different fuel blends. D100 and D15NG60A25 fuels have the highest and lowest CO values respectively. Ammonia is a carbon free fuel; therefore, the CO emission decreases with the addition of ammonia in blending. Again, blending in dual mode has a negligible impact on CO emissions. In addition, with the inclusion of NG in dual mode, CO decreases significantly [26].

Figure 18 shows how CO₂ percentage change with different fuel blends. Injection timing has no influence on CO₂ emissions because both lines are overlapping. Due to high carbon content in diesel fuel, D100 has the highest CO₂ percentage, however D15NG60A25 has the lowest CO₂ percentage among the fuel blends. In comparison to pure diesel, dual

mode and blending emit less CO₂. The engine running conditions in figures (6–8) and (10–

18) are the same as in figure 5 with an intake pressure of 1.3 bar.

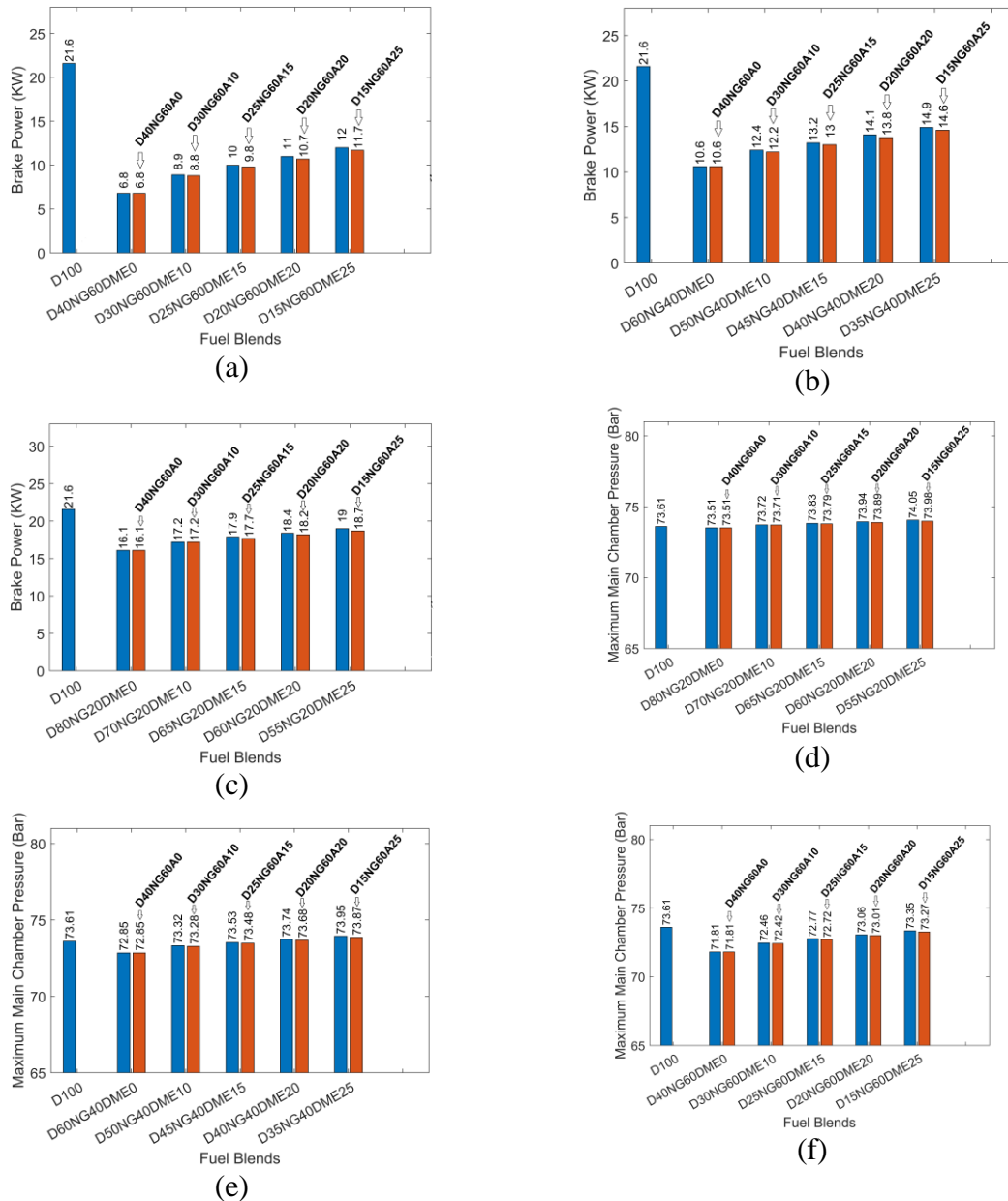


Figure 15: Variation of brake power and main chamber pressure with DME for different fuel blends. (a), (b) and (c) are brake powers at fixed 60%, 40% and 20% NG substitutions respectively. (d), (e) and (f) are main chamber pressures at fixed 20%, 40% and 60% NG substitutions respectively.

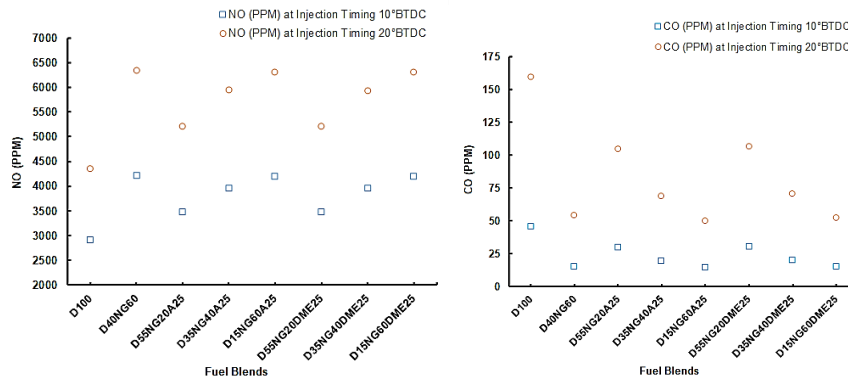


Figure 16: NO (ppm) at EVO for different fuel blends. Figure 17: CO (ppm) at EVO for different fuel blends.

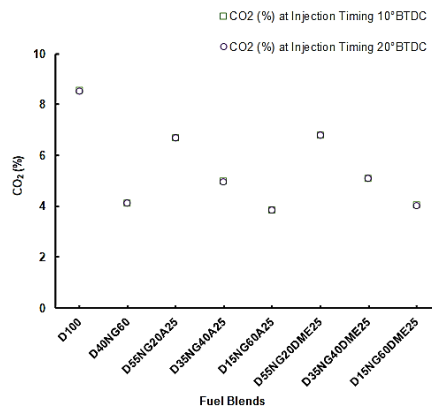


Figure 18: CO₂ (%) at EVO for different fuel blends.

4. CONCLUSIONS

Present work focused on the combustion characteristics of DME/Ammonia and key conclusions are extracted as below:

- i. In dual mode, the output parameters of the engine such as temperatures, pressures, IMEP and BP decrease and continue to decrease as the NG percentage grows from 20% to 60% in dual mode. However, with 20% NG (D80NG20A0), the pre and main chamber pressure values are nearly identical to pure diesel. Furthermore, by increasing the amount of ammonia from 5% to 25% in blending raises the above-mentioned output

parameters of the engine. Because of the rise in BP, as BSEC value falls.

- ii Pre-chamber and main chamber temperatures, IMEP and BP values are maximum and BSEC value is the lowest for the pure diesel among all dual mode and blending cases. Also, BP and IMEP for different fuel blends rise with the increase in passageway diameter.

- iii. Ammonia and DME substitutions in blending increase the BP as compared to dual fuel. In blending cases, (diesel + DME) has slightly higher BP than (diesel + ammonia). The engine's optimal injection timing is 20°

BTDC resulting in a maximum BP at around 340° CA.

iv. In both dual modes and blending cases, NO increases while CO and CO₂ decrease, when compared to pure diesel. According to study, D15NG60A25 and D15NG60DME25 are the best fuel blends in terms of BP and exhaust emissions. In comparison to ammonia, DME is more environmentally benign, non-toxic and easy to handle.

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التحقيقات النظرية حول استخدام الوقود البديل في محرك ديزل يستخدم خليط وقود مزدوج

نفيس أحمد * ، نواف الخميس ، عباس لال خان

الهندسة الميكانيكية ، جامعة الملك عبد العزيز ، جدة ، المملكة العربية السعودية

تستكشف الدراسة تأثير غاز النشادر و غاز ثنائي ميثيل الأثير في وضعية الوقود المزدوج على أداء الاحتراق و صفاته و انبعاثات العادم في محرك الديزل ذو الحقن الغير مباشر في غرفة احتراق قبل الغرفة الرئيسية، من دون إجراء تعديلات جوهرية للمحرك. تم استخدام المبرمج فورتران ٧٧ لعمل برمجة محاكاة حاسوبية. وقد أظهرت النتائج أن المزج بطريقة الوقود المزدوج أدت إلى رفع كلا من درجة الحرارة والضغط في كلا من غرفة الاحتراق الرئيسية و التي قبلها، و أدت أيضاً إلى رفع كلا من الضغط المتوسط الفعال البياني و القدرة المكبحية و تخفيض استهلاك الطاقة النوعية المكبحية . حينما زادت نسبة النشادر كبديل من ٥٪ إلى ٢٥٪. في مزيج (الديزل و النشادر) زاد القدرة المكبحية بنسب ١٥,٩٣٪ و ٣٧,٢٥٪ و ٧٢,٤٢٪ في وضعية الوقود المزدوج مقابل الاستبدال بالغاز الطبيعي بالنسب ٢٠٪ و ٤٠٪ و ٦٠٪ على التوالي. و كذلك مزيج (الديزل و غاز ثنائي ميثيل الأثير) زاد من القدرة المكبحية بنسب ١٧,٧٩٪ و ٤٠,١٧٪ و ٧٦,٩٩٪ في وضعية الوقود المزدوج مقابل الاستبدال بالغاز الطبيعي بالنسب ٢٠٪ و ٤٠٪ و ٦٠٪ على التوالي.

الكلمات المفتاحية:

محرك ديزل، معاملات الأداء، خصائص احتراق مخاليط الوقود