ANALYSIS OF COMBUSTION CHARACTERISTICS, ENGINE PERFORMANCES AND EXHAUST EMISSIONS OF LONG-CHAIN ALCOHOL-DIESEL FUEL BLENDS.

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ABSTRAK

Di zaman kini, bahan api berasaskan petroleum telah menjadi salah satu sumber tenaga yang mempunyai paling banyak permintaan bagi pelbagai tujuan dan aplikasi. Disebabkan ini, masalah seperti pelepasan gas yang tinggi daripada bahan api berasaskan petroleum telah memaksa banyak kerajaan untuk memperkenalkan peraturan yang ketat dan kebimbangan terhadap keselamatan tenaga. Langkah ini juga meningkatkan minat para penyelidik di seluruh dunia untuk menghasilkan bahan bakar alternatif. Sebelum ini, alkohol rangkaian pendek seperti metanol, etanol dan propanol telah digunakan sebagai elemen oksigen untuk meningkatkan kandungan oksigen dalam bahan api diesel. Walau bagaimanapun, campuran alkohol-diesel rantaian pendek mempunyai kelemahan seperti nombor cetane yang rendah, nilai kalori yang rendah serta kenaikan NOx dan kesesuaian campuran yang rendah dengan bahan api diesel (DF). Disebabkan masalah dengan penggunaan alkohol rantaian pendek, ramai penyelidik telah menunjukkan minat terhadap alkohol rantai panjang seperti pentanol dan heksanol. Ini disebabkan oleh sifat termofisika alkohol rantai panjang yang jauh lebih baik daripada alkohol rantaian pendek dari segi ketumpatan, kelikatan kinematik, nombor cetane, nilai kalori yang rendah dan kesesuaian campuran. Sehubungan dengan ini, alkohol rantaian panjang, 1-pentanol (1-PN) dan 2-etil 1-hexanol (2-EH) dengan nilai kalori yang lebih tinggi, ketumpatan tenaga, nombor cetane dan kekenyalan yang lebih baik dicampur dengan DF untuk menghasilkan campuran bahan api alkohol-diesel rantaian panjang. Oleh kerana itu, objektif pertama bagi kajian ini adalah untuk menganalisis formulasi baru untuk campuran 1-pentanol-diesel dan campuran 2-etil 1-heksanol-diesel dengan mengkaji sifat termofisika-nya. Selain itu, ianya perlu untuk menyiasat kesan campuran 1-pentanol diesel dan campuran 2-etil 1-heksanol-diesel terhadap ciri-ciri pembakaran, prestasi engine dan pelepasan gas dalam enjin diesel. Objektif terakhir adalah untuk mendapatkan korelasi antara nisbah campuran dan prestasi engine. Dalam eksperimen ini, 5%, 10% dan 20% daripada 1-PN dan 2-EH dimasukkan ke dalam DF untuk menghasilkan campuran bahan api alkohol-diesel rantai panjang. Campuran bahan api telah disediakan dengan menggunakan mesin pengemulsi ultrasonik Hielscher UP400S pada kelajuan kacau 40% Hz dan amplitud 0.5%. Perbincangan akan memberi tumpuan kepada ciri-ciri pembakaran, prestasi enjin dan pembebasan ekzos enjin diesel silinder tunggal YANMAR TF120M pada kelajuan enjin tetap 1800rpm di bawah pelbagai beban enjin (0%, 25%, 50%, 75%, dan 100%). Keputusan lengkap untuk ujian sifat termofisika dan ujian enjin diperolehi untuk DF, PE5, PE10, PE20, HE5, HE10 dan HE20. Ketumpatan, nombor cetane, nilai kalori dan kelikatan campuran bahan api menurun berbanding dengan DF. Analisis prestasi menunjukkan bahawa BTE telah meningkat sebanyak 7.03%, 12.09%, 17.55%, 12.25%, 12.95%, dan 19.67% untuk PE5, PE10, PE20, HE5, HE10 dan HE20 daripada DF. BSFC untuk campuran bahan api juga berkurangan sebanyak 8.51% untuk PE5, 10.16% untuk PE10, 12.08% untuk PE20, 8.93% untuk HE5, 10.87% untuk HE10 dan 10.99% untuk HE20. EGT untuk semua campuran bahan api adalah lebih rendah kecuali PE5 yang telah meningkatkan EGT disebabkan peningkatan O_2 mengandungi. Selain itu, keputusan telah melaporkan bahawa campuran bahan api mempunyai CO dan EGO yang lebih tinggi. Tetapi, pelepasan CO_2 dan NO_x yang lebih rendah didapati dalam campuran bahan api berbanding dengan DF. Selain itu, HC lebih tinggi untuk semua campuran bahan api daripada DF kecuali untuk HE5 dengan pengurangan pelepasan HC. Berdasar keputusan, kajian mendapati 2-EH adalah aditif yang lebih baik untuk diesel berbanding 1-PN.

ABSTRACT

In the present life, petroleum-based fuels have been one of the most in demand energy source for various purposes and application. Due to this, problems such as higher gas emissions from petroleum-based fuels have forced many governments to introduce stringent regulations and concerns over energy security. Previously, the short-chain alcohol such as methanol, ethanol and propanol had been used as an oxygenated element to increase oxygen content in diesel fuels. However, short-chain alcohol-diesel blends have disadvantages such as low cetane number, low calorific value as well as increase of NO_x and low miscibility with diesel fuel (DF). Due to the problems with the used of short-chain alcohol, many researchers have shown interest on long-chain alcohol such as pentanol and hexanol. This is due to the thermophysical properties of long-chain alcohol which are much better than short-chain alcohol in terms of density, kinematic viscosity, cetane number, low calorific value and miscibility. In relation to this, long-chain alcohols, 1-pentanol (1-PN) and 2-ethyl 1-hexanol (2-EH) are blended with DF to produce long-chain alcohol-diesel fuel blends. Thus, the first objective of this study was to analyze new formulation of 1-pentanol-diesel fuel blends and 2-ethyl 1-hexanol-diesel fuel blends by study their thermophysical properties. Besides that, it was necessary to investigate the effect of 1-pentanol-diesel fuel blends and 2-ethyl 1-hexanol-diesel fuels on combustion characteristic, performance and emissions in diesel engine. Last objective was to determine the correlation between the blend ratio and engine performance within research scope. In order to obtain objectives, 5%, 10% and 20% of 1-PN and 2-EH are added into DF to produce long-chain alcohol-diesel fuel blends. The fuel blends were prepared by using Hielscher UP400S ultrasonic emulsifier machine at stirring speed 40% Hz and amplitude 0.5%. The discussion will focus on combustion characteristics, engine performance and exhaust emissions of single cylinder diesel engine YANMAR TF120M at constant engine speed of 1800rpm under various engine loads (0%, 25%, 50%, 75%, and 100%). The complete results for thermophysical properties test and engine test obtained for DF, PE5, PE10, PE20, HE5, HE10 and HE20. The density, cetane number, calorific value and viscosity of fuel blends decrease compared to DF. Performance analysis showed that BTE had increased by 7.03%, 12.09%, 17.55%, 12.25%, 12.95%, and 19.67% for PE5, PE10, PE20, HE5, HE10 and HE20 than that DF. The BSFC for fuel blends also decreased 8.51% for PE5, 10.16% for PE10, 12.08% for PE20, 8.93% for HE5, 10.87% for HE10 and 10.99% for HE20. The EGT for all fuels blends are lower except for PE5 which had increase of EGT due to increase of O_2 contains. Furthermore, results have reported that fuel blends had higher CO and EGO. But, lower CO_2 and NO_x emissions found in fuel blends compared to DF. Additionally, the HC was higher for all fuel blends than DF except for HE5 with reduced of HC emission. Based on the results, the study found that 2-EH was better additive to diesel than 1-PN.

TABLE OF CONTENT

LIST OF TABLES

LIST OF FIGURES

LIST OF SYMBOLS

%	Percentage
g	Gram
m3	Cubic Meter
wt	Molar weight
r/min	Revolution per minute
kW	kiloWatt
Nm	Newton meter
V	Cylinder geometry
$dV/d\theta$	Pressure
π	Pie
λ	Lambda
θ	Theta
$dQn/d\theta$	Heat release rate
γ	Ratio of specific heat
°C	Degree of Celcius
$\dot{m}$	Mass flow rate
Pe	Engine power
mm	Milimeter
mPa.s	Mili pascal per second
ml	Mili liter
s	second
L	Liter
°	Degree
p-value	Probability value
R	Realibility

LIST OF ABBREVIATIONS

1-PN	1-pentanol
2-EH	2-ethyl 1-hexanol
BP	Brake power
BT	Brake torque
BSFC	Brake specific fuel consumption
BTDC	Before top dead center
BTE	Brake thermal efficiency
C	Carbon
CAD	Crank angle degree
CCRD	Central composite rotatable design
CO	Carbon monoxide
CO2	Carbon dioxide
DAQ	Data acquisitions
DF	Diesel fuel
DI	Direct injection
DoE	Design of Experiments (DoE)
EGO	Exhaust gas oxygen
EGT	Exhaust gas temperature
H	Hydrogen
HC	Hydrocarbon
HRR	Heat release rate
HE5	5% 2-ethyl 1-hexanol+95% diesel fuel
HE10	10% 2-ethyl 1-hexanol+90% diesel fuel
HE20	20% 2-ethyl 1-hexanol+80% diesel fuel
NOx	Nitrogen oxide
OH	Hydroxyl radical
O2	Oxygen
PE5	5% 1-pentanol+95% diesel fuel
PE10	10% 1-pentanol +90% diesel fuel
PE20	20% 1-pentanol+80% diesel fuel
RSM	Response surface method

CHAPTER 1

INTRODUCTION

1.1 Introduction

In recent decades, researchers and manufacturers have been interested in producing alternative fuels to reduce usage of diesel and decrease exhaust emissions emitted by the diesel engines due to increasing global concern about the air pollution. Moreover, an increasing number of usages of diesel vehicles will not solve the air pollution problems and save the diesel. In order to solve these problems, various studies are conducted in many countries in order to begin reducing the dependency on petroleum fuels. New alternative fuels, including alcohol-based fuels, such as propanol, ethanol, methanol, butanol, pentanol, hexanol and octanol from renewable sources, are attractive alternative solutions to meet the energy demand and to regulate the emissions (Çelik, Örs et al., 2017; De Poures, Sathiyagnanam et al., 2017; Gnanamoorthi & Devaradjane, 2015; Khalife, Tabatabaei et al., 2017; Kumar, Cho et al., 2013). Owing to this renewability, biodegradability, and superior fuel properties of gasoline, biodiesel and diesel, the alcohol-based fuels are currently considered as one of the future alternative to the diesel fuel which shows positive improvement in performance and emission of the engine (Çelik, Örs et al., 2017; Sathiyagnanam, Saravanan et al., 2010).

Previous studies show that alcohols can be used in compression ignition (CI) engines when blended with conventional diesel fuel or biodiesel. One of the advantages of alcohols as fuel is lower viscosity compared to diesel fuel which makes it easily injected, atomized and mixed with air. Besides that, its high stoichiometric fuel–air ratio, high oxygen content, high H/C ratio and low sulphur content help in reducing the emission. In addition, the high evaporative cooling, which results in a cooler intake process and compression stroke increase the volumetric efficiency of the engine and thereby lessen the required work input needed in the compression stroke. Combustion process can also finish early due to high laminar flame propagation speed, thus improving engine thermal efficiency (Sayin, 2010).

The short-chain alcohols, namely; methanol, ethanol and propanol are widely known to be an oxygenated fuel that increases the availability of oxygen during combustion, and reduce the smoke emission. Although the utilization of alcohols enrichment with oxygen content improves both the premixed and diffusive burning stages, their low calorific value and low cetane number have issues on miscibility and stability, weak auto ignition quality, and improper lubrication behaviour limit their use in diesel. The advantage of using short-chain alcohol is the reduction of CO and HC emission with the addition of short-chain alcohol. However, most works also revealed that, NOx failed to reduce short-chain alcohol-diesel fuel blend. In addition, short-chain alcohol-diesel fuel blends are reported to increase brake specific fuel consumption and lower brake thermal efficiency compared to DF. In relation to this, the present research focus is to study on how alternative fuels such as long-chained alcohol-diesel fuels, improve combustion characteristics, engine performance and exhaust emissions of the diesel engine. The reason is because of the interests are growing to improve fuel properties and save the usage of pure diesel.

Nowadays, long-chain alcohols such as butanol (C4 alcohol) have been studied to overcome the weakness of short-chain alcohol-diesel fuel blends. Past literatures showed that butanol was capable to mix well with 100% neat DF and various biodiesel. The reason was due to higher energy density, cetane number, viscosity, flashpoint and boiling point compared with ethanol. In 2014, (Balamurugan & Nalini, 2014) found CO and NOx emissions to be reduced with presence of 4% and 8% butanol. The possible explanation was due to the high latent heat of vaporization and calorific value of butanol that reduced the operating temperature, CO and NOx. A similar trend of falling formation of NOx was reported by (Ileri, 2016) who found that longer chain alcohol had higher cooling effect. Additionally, (Ileri, Atmanli et al., 2016) found higher CO_2 and reduced HC emission which indicated a more complete combustion. Thus, many researchers started to replace ethanol with butanol as additives in DF or biodiesel, which later investigated the engine combustion. The used of butanols as one of oxygenated alcohols from long-chain alcohol type had been proven effective by many researchers in their studies.

A recent development of alcohol based fuel showed another type of long-chain alcohol, such as 1-pentanol (1-PN) and 2-ethyl 1-hexanol (2-EH) as new alcohols that are introduced to mix together with pure diesel fuels, and DF as new alternative of long-chain alcohol-diesel fuels. Both 1-PN and 2-EH are among the new generations of long-chain alcohol, however many studies have not used this type of alcohols so far. Alternatively, long-chain alcohols, such as 1-PN and 2-EH provide advantages as they have higher calorific value, energy density, cetane number and better miscibility compared to short-chain alcohol (Imdadul, Masjuki et al., 2016; Nanthagopal, Patel et al., 2018), thus allowing it to blend with diesel fuel in a higher fraction. Both are long chain alcohols produced from renewable sources and considered as a promising blending component with diesel or biodiesel blends. Moreover, many researchers reported using long-chain alcohols, such as 1-pentanol or 1-octanol, together with biodiesel such as Calophyllum inophyllum methyl ester blends (Nanthagopal, Korah et al., 2018) and Mahua oil (Mahalingam, Munuswamy et al., 2018).

Based on findings from previous studies, it is observed that many studies were carried out using short-chain alcohols such as methanol, ethanol or propanol. Besides that, there are long chain alcohol, butanol. However, the present study introduces 1-PN and 2-EH, blended at 5%, 10% and 20% instead of at ratio range of 10%, 20% and 30% which was used by many researchers before. The new formulation of 5%, 10% and 20% of long-chains alcohol fuel blends with pure diesel was used for properties test following ASTM standard. It was observed that, there are lack of investigation on application of 1-PN and 2-EH as alcohol-diesel fuels on combustion characteristics, engine performance and emissions, particularly for CO_2 and EGO. As a result, a complete analysis of fuel blends and DF, including combustion characteristics, engine performance and emissions (CO, CO_2, HC, EGO and NOx) was carried out. In addition, the Response Surface Modelling (RSM) modelling is also developed based on actual factors and quadratic equations to validate the output from the carried out experiment. In this direction, this thesis shows the potential of using 1-PN and 2-EH, blended with pure diesel as one of the alternatives of fuels in diesel engines.

1.2 Problem statement

In modern contemporary, diesel fuels are an important energy source that is used in over 80% of our daily life (Atabani, Mahlia et al., 2013). They are being used in transportation, railway, aircraft, heavy equipment and other ramification. The excessive demand and usage of diesel fuels has led to emissions problems. The main problem is the processing and the usage of diesel fuel energy gives negative effects, such as pollutant emissions to the environment. The four main pollutant emissions reported by (Reşitoğlu, Altinişik et al., 2015) are carbon monoxide (CO), hydrocarbon (HC), particulate matter (PM), and nitrogen oxides (NOx) that affected human respiratory system, which has led to various health problems, such as asthma, asphyxiation and cancer. Moreover, the HC and NOx emission caused by diesel fuels may also cause depletion of the ozone layer and greenhouse effect.

The result of these stated problems makes researchers to focus more on producing alternative fuels. Also, it is well known that researchers used to produce alternative fuel by adding short-chain alcohol to diesel fuel, namely; ethanol and methanol which are well known as being an oxygenated fuel to increase the availability of oxygen during combustion which reduce CO and HC emissions. However, the short-chain alcohol-diesel fuel blends showed decrease in cetane number, calorific value and viscosity which poorly effected the performance of engine combustion (Gomasta & Mahla, 2012; Masimalai, 2014). Therefore, it was reported increase of NOx formation with used of short-chain alcohol-diesel fuel blends. Moreover, short-chain alcohol also had low miscibility with diesel fuels. In recent year, researchers have been interested to replace short-chain alcohol with long-chain alcohol. In the present study, long-chain alcohol; 1-PN and 2-EH with higher calorific value, cetane number, viscosity, and better miscibility, was used and thus, allowing it to be blended with diesel fuel in a higher fraction to reduce emission produce. Therefore, it increases the stability of the long-chain alcohol-diesel fuels blends which give better performance in engine combustion. The lack of study on 1-PN and 2-EH had been a research gap for this study.

1.3 Objectives of the study

The objectives of the project are as follows:

1. To analyze new formulation of 1-pentanol-diesel fuel blends and 2-ethyl 1-hexanol-diesel fuel blends by study their thermophysical properties.

2. To investigate the effect of 1-pentanol-diesel fuel blends and 2-ethyl 1-hexanol-diesel fuels on combustion characteristic, performance and emissions in diesel engine.

3. To determine the correlation between the blend ratio and engine performance within research scope.

1.4 Scope of research

The scope of this research involves fuels blends preparation and experimental work on fuels samples. Detailed system design will be described in the research methodology. In summary, the scopes of this project are as follows:

1. Identification of parameters / variables of 5%, 10% and 20% of long-chain alcohol diesel fuel.

2. Build up database for pure diesel and 5%, 10% and 20% of long-chain alcohol-diesel fuel blends on combustion characteristic, performance and exhaust emissions of diesel engine.

3. Single cylinder diesel engine experiment with engine loads (0%, 25%, 50%, 75% and 100%) and engine speed (1800 rpm).

4. Analysis of combustion, performance, and emissions.

5. Build up modelling and correlation between blend ratio and engine performance using response surface methodology (RSM).

1.5 Thesis outline

This transfer thesis consists of five chapters including this chapter. The content of each chapter can outlined as follows:

Chapter 2 provides literature review, background, previous research done by other researchers in the same area and relevant issues related to combustion engine. This included an overview of producing new alternative from various type of alcohol based diesel fuel blends.

Chapter 3 present brief methodology for producing and study new alternative long-chain alcohol-diesel fuel blends. Details of the methodologies are explained in this chapter accordingly. The experimental set up and flowchart had been included in the chapter to illustrate how experiment worked.

Chapter 4 presents experimental results that covered the properties and the effect of long-chain alcohol-diesel fuel blends on performance, combustion characteristics and emissions on the direct injected diesel engine.

Chapter 5 provided general conclusion of the research work. Other research recommendation for future work are presented.

CHAPTER 2

LITERATURE REVIEW

2.1 Background study

Diesel engine is a compression-ignition (CI) internal combustion engine in which ignition of the fuel is caused by the high temperature of the gas (air) and compressed completely (Ferguson & Kirkpatrick, 2015). CI engine is manufactured in two-stroke engine and four-stroke engine versions. The engine work by compressing the air inside the combustion chamber. The compression work done by the piston increases the air temperature inside the cylinder in such extremely high degree that it ignites the atomized diesel fuel injected into the combustion chamber. This contrasts with the spark-ignition (SI) engine, such as a gasoline engine, which has a spark plug to ignite an air-fuel mixture in the combustion chamber. In some CI engine, a glow plug (combustion chamber pre-warmers) may be used to aid starting the engine in cold weather, or when the engine uses a lower compression-ratio. The CI engine also operates in the constant pressure cycle of gradual combustion and produces no audible knock. The CI engine has the highest thermal efficiency (engine efficiency) of any practical internal or external combustion engine, due to its very high expansion ratio and inherent lean burn which enables heat dissipation by the excess air (Ferguson & Kirkpatrick, 2015). In addition, low-speed compression-ignition engines has been used in ships and other applications where overall engine weight is relatively unimportant and can have a thermal efficiency that exceeds 50%.

Rigorous emission regulations and concerns over energy security have recently increased researchers interest on alternative renewable fuels. Alternative fuel in transportation, industry, power and agriculture is dependably in expanding pattern, because of the consumption of petroleum reserves throughout the world. Numerous countries have taken action, while several others are remarkably contributing to the climate change that must be under better control. Environmental Protection Agency (EPA), the directive in the USA and Euro VI standards in Europe have set the new emission regulations, which was acknowledged in 2017 where the NO, CO and PM exhausts ought to be lessened by 25%, 24% and 10% individually by 2030 (Ogunkoya, Li et al., 2015).

Diesel engine is one of the major contributions to excessive emissions causing pollution and natural disaster. Due to these problems, several countries have expressed concerns over climate issues and energy security, thus strictly stating their emission regulations to overcome the issue. This move has urged many researchers to explore and focus more on producing alternative fuels to replace the dependence on petroleum fuels. Some examples of alternative fuels are synthetic fuels, methane, dimethyl ether, biodiesel, hydrogen, alcohols, emulsion fuels and much more (Ashraful, Masjuki et al., 2014; Atabani, Mahlia et al., 2013; Fahd, Wenming et al., 2013; Fattah, Kalam et al., 2014; Hasannuddin, Yahya et al., 2018; Wan Ghazali, Rizalman et al., 2015). Each alternative fuels have their own advantages and disadvantages in their performance engine. Recently, alcohol-based fuels have been one of the major interesting diesel alternative fuels.

Numerous studies have explained that alcohols such as ethanol, methanol, butanol, pentanol, and hexanol have the qualities to meet the energy demand and control emissions (Atmanlı, Yüksel et al., 2013; Shahir, Masjuki et al., 2015; Yusri, Mamat, Najafi et al., 2017). This is because alcohols are being used as fuel blending components to improve unleaded cetane quality, which will increase oxygen content in diesel fuel blends, thus improving the blends knock resistance. Firstly, the thermophysical properties of alcohols such as density, cetane number, kinematic viscosity, and calorific value are suitable to act as additive for diesel fuels. The comparable properties of conventional diesel and alcohol-based fuels had offer an excellent solution to the above stated problems. Diesel fuels added with alcohols are attractive alternative solutions to meet the energy demand and regulate emissions. Alcohol based fuels also capable to reduce emissions due to high fuel–air ratio, high H/C ratio and low sulphur content. In addition, the high evaporative cooling of alcohols caused cooling effect process and decrease work input needed in the compression stroke which increases the efficiency of the engine.

Based on this, interests are growing on how to improve fuel properties and to overcome the problems. Although, the utilization of the short –chain alcohol that is enriched with oxygen content, improves both the premixed and diffusive burning stages, their low calorific value, low cetane number, miscibility and stability issues, weak auto ignition quality, and improper lubricating behavior thereafter limit their use as a pure diesel engine fuel. Long-chain alcohols have advantages with higher cetane number, viscosity, calorific value and heat of vaporization. Therefore, long-chain alcohol with longer carbon chain seems to be a possible solution to replace short-chain alcohol.

2.2 Thermophysical properties of alcohol-based fuel:

Table 2.1 Thermophysical properties of diesel and alcohols.

Fuel properties	**DF**	**Ethanol**	**Pentanol (1-PN)**	**Hexanol (2-EH)**
Molecular formula	$C_{12}H_{23}$	C_2H_5OH	$C_5H_{11}OH$	$C_8H_{17}OH$
Density at 40 °C (g/m^3)	0.837	0.785	0.815	0.818
Viscosity	5.8	1.13	2.89	5.1
Cetane number	52.0	5-8	20.0	23.2
Caloric value (MJ/kg)	42.8	26.9	32.16	34.7
Heat of vaporization	270.00	904.00	308.05	358.00
C content (wt%)	86.14	52.14	68.13	71.12
H content (wt%)	13.86	13.13	13.74	15.35
O content (wt%)	0	34.73	18.15	13.53

There are two different types of alcohol, which are; short-chain alcohol and long chain alcohol. These are an important class of alcohols, in which methanol, ethanol, and propanol are the simplest members in the saturated straight chain alcohols. The general formula for methanol is CH_4O, ethanol is C_2H_6O and propanol is C_3H_8O (Wallner, Ickes et al., 2013). In terms of the general formula, it can be seen that short-chain alcohols, such as methanol, ethanol and propanol have alkyl chains of 1–3 carbons. While long-chain alcohols, which is also known as fatty alcohols have alkyl chains of 4–21 carbons, and very long-chain alcohols have alkyl chains of 22 carbons or longer. An example of long-alcohol are pentanol and hexanol with a general formula of $C_5H_{12}O$ and $C_6H_{14}O$ (Wallner, Ickes et al., 2013) respectively. From Table 2.1, long chain alcohol, 1-PN and 2-EH has higher carbon, hydrogen and oxygen content by molecular weight than DF.

As shown in Table 2.1, the thermophysical properties of 1-PN and 2-EH (long-chain alcohols), are better than ethanol (short-chain alcohol), which is closer to thermophysical properties of DF. The percentage of O_2 for 1-PN and 2-EH are higher compared to ethanol and DF. Also, it is observed that ethanol has lower density than DF by 6.21%. Meanwhile, the density of 1-PN and 2-EH is only 2.63% and 2.27% less than DF. Moreover, the cetane numbers show for ethanol is 8, 1-PN is 20.0 and 2-EH is 23.2. The cetane numbers for long-chain alcohol are higher than short-chain alcohol. With higher cetane number and density, the performance and combustion of the DF can be improved during engine testing. Even though calorific value of 1-PN and 2-EH are slightly lower compared to DF, but still higher compared to ethanol by 16.36% and 22.48%. Thus, better combustion is expected, and emissions can be improved at the end of this experiment. Also, the thermophysical properties of long-chain alcohol are better than the short-chain alcohol, which is closer to thermophysical properties of DF. In previous studies, long-chain alcohol seems to be capable of replacing short-chain alcohol for alcohol-diesel blend (Kanase-Patil, Tekadeb et al., 2014).

2.3 Complete combustion reaction

The complete combustion below showed the reaction involved in diesel and alcohol with atmospheric O_2. From the combustion reaction, DF has the highest number of O_2 needed in order to obtain complete combustion. The oxygen molecule in alcohol structure had decrease the number of O_2 needed in fuel blends to gain complete combustion. The oxygen molecules in the alcohols structure definitely improve combustion of diesel engine.

1. Pure diesel (DF) combustion: $\mathbf{4C_{12}H_{23}+71O_2 \rightarrow 48CO_2+46H_2O}$.

2. 1-Pentanol (1-PN) combustion: $\mathbf{2C_5H_{11}OH+15O_2 \rightarrow 10CO_2+12H_2O}$.

3. 2-Ethyl 1-hexanol (2-EH) combustion: $\mathbf{C_8H_{17}OH+12O_2 \rightarrow 8CO_2+9H_2O}$.

2.4 Combustion characteristics of alcohol-diesel fuel blends

2.4.1 Peak pressure curves

Previous research by (Balki, Sayin et al., 2014) has found that in-cylinder pressure increased with ethanol and methanol. It was observed that in-cylinder pressure of the fuel blends, occurs earlier than DF. Several authors have suggested that higher flame speed and cetane number of ethanol and methanol have caused shorter combustion duration and increase in-cylinder peak pressure. Furthermore, higher latent heat of evaporation of alcohol fuels and gasoline also increased the volumetric efficiency and BTE, thereby considered for the increased of in-cylinder pressure. However, this study also found in-cylinder pressure for the methanol to be higher than the other fuels. This increasing value may be explained with the oxygen ratio within the chemical structure of the methanol which is higher than the gasoline and ethanol.

This result is consistent with the finding of (Zhu, Cheung et al., 2011). Other authors have also reported the same finding to discover higher maximum pressure with the addition of ethanol into biodiesel. The lower cetane number is higher in-cylinder pressure. Moreover, due to the lower density and viscosity of ethanol, the ethanol-biodiesel blends could improve the spray characteristics and enhance the mixing of fuel and air, and hence increase the premixed heat release rate and the maximum pressure.

Another investigation made by (Wei, Cheung et al., 2014) employed different approach and used n-pentanol as the oxygenated fuel. These authors found that, the ignition is retarded and the ignition will delay longer with the addition of n-pentanol in the blended fuel. The reason for the long ignition delay is due to the decrease of a cetane number of the blended fuels than that of DF. Besides that, these authors also reported that the in-cylinder pressure decreases with addition of n-pentanol at low engine load which is not significant, whereas in-cylinder pressure increases with n-pentanol at medium and high engine load. Based on these, it can be explained that at the low engine load, the addition of n-pentanol increases the latent heat of evaporation of the fuel and delays the combustion further into the expansion stroke, thus decrease peak in-cylinder pressure. While the increase in ignition delay leads to more fuel being burned in premixed mode, which may lead to increase in peak in-cylinder pressure. In that study, the authors concluded that the conflicting factors lead to an insignificant decrease in the peak in-cylinder pressure to increase of n-pentanol and the higher in-cylinder temperature in the medium and high engine loads which may weaken the cooling effect of n-pentanol, and leads to increase in ignition delay with the increases of n-pentanol in fuel blends.

In general, increase in maximum in-cylinder pressure was found as engine load increase (Wei, Cheung et al., 2014; Zhu, Cheung et al., 2011). While the maximum pressure occurs further away from the top dead centre (TDC) with an increase in engine load. This is due to increase of in-cylinder temperature significant to increase of engine loads caused higher pressure in cylinder.

2.4.2 Heat release rate

According to an investigation by (Zhu, Cheung et al., 2011), the heat release rates (HRR) of all the fuels have similar shape with having a premixed combustion phase that is followed by a diffusion combustion phase. The result presented that, the premixed combustion phase for all the fuels is shortened, while the diffusion combustion phase is lengthened with the increase of engine load. Also, it is observed that with increase of engine load, the maximum HRR occurs slightly closer to the TDC. Comparing this to biodiesel, it is reported that with the increase of ethanol fraction, HRR increases and occurs further away from the TDC, as there is a delay in the start of the combustion. A possible explanation is that, the start of the combustion was delayed due to the lower cetane number, density and viscosity of ethanol, which increases the mixing region of fuel and air, improved spray characteristic leading to more fuel combusted in the premixed phase, and finally resulting to the higher maximum pressure and higher premixed HRR.

Moreover, higher latent of vaporization of ethanol lowers the in-cylinder temperature and hence, increase the ignition delay. At the diffusion combustion phase, the blends give higher heat release rate than that of biodiesel and Euro V diesel fuel, indicating that the diffusive combustion phase is improved due to the higher oxygen content of the blends, which also leads to a reduction of the combustion duration.

An experimental investigation from (Siwale, Kristóf et al., 2013) was conducted to explore about the engine pressure after the initiation of burning, and a small simplification in piston chamber insistence was experienced irrespective of all engine loads condition of 5%, 10% and 20% n-butanol-diesel fuels. This study has also identified a little difference in the pressure curves with increased premixed heat that was released due to the blends. The increased HRR in the crank angle region indicated the event of premixed combustion. The possible explanation of this is due to decrease of a cetane number of the blends with the addition of n-butanol, as well as the lower boiling point of n-butanol. Again, this study found that the higher oxygen content of n-butanol has enhanced the mixing ability of the blend. Therefore, HRR increased with addition of n-butanol ratio in the premixed phase. In the mixing controlled combustion phase, it was also observed that the HRR for the blends, slightly shortened in comparison to DF as a result of a better combustion efficiency of the blends.

In recent year, (Zhu, Xiao et al., 2016) have also reported increase of HRR with an increase of pentanol fraction. However, this study was in contrast to the previous literature, because the increase of pentanol fraction in the biodiesel–pentanol blends have increased the HRR and makes it occur further away from the top dead center (TDC) compared to biodiesel, which indicated a longer ignition delay due to the lower cetane number of pentanol. It is expected for a better atomization of pentanol-biodiesel as pentanol had a lower viscosity compared to biodiesel, which enhanced the fuel/air mixing, resulting in increases in the premixed heat release rate and the maximum pressure.

2.5 Performance of alcohol-diesel fuel blends

2.5.1 Brake specific fuel consumption

Over the past century, the increase and fluctuation in prices of diesel fuels and petrol as well as shortage of petroleum have been one of the reasons to produce alternative fuels. Thus, one of important trait for an effective alternative fuel is to have minimum brake specific fuel consumption. Brake specific fuel consumption (BSFC) is a measure combustion efficiency to measure how efficient a given amount of fuel is being converted into a specific amount of horsepower. An improved combustion allows the same amount of fuel to produce an increase in power to improve combustion efficiency, which thereby results in decrease of BSFC.

A previous study by (Yasin, Mamat et al., 2017) discovered higher BSFC for biodiesel-methanol-diesel. The authors reported an increase of BSFC as 5%, 10% and 20% concentration of methanol increase. In 2015, (Oliveira, Morais et al., 2015) found similar finding of an increase of BSFC, with 5%, 10% and 15% biodiesel-ethanol-diesel fuels due to lower calorific value and density of ethanol. A research by (Atmanli, 2016a) had investigated the BSFC for short-chain, propanol and long-chain alcohol, n-butanol and 1-pentanol together with biodiesel and diesel at 20% concentration. His research further presented higher BSFC for n-propanol-biodiesel-diesel blend by 5.28% and lower BSFC for n-butanol-biodiesel-diesel blend and 1-pentanol-biodiesel-diesel blend by 0.89% and 0.95% respectively. The author also reported that propanol, n-butanol and 1-pentanol have 26.7%, 21.59% and 18.15% oxygen (by weight), respectively, in their atomic structures, which is one of the factors that influence measured BSFC.

In addition, a study by (Campos-Fernández, Arnal et al., 2012) found lower BSFC, with the addition of butanol and pentanol. The calorific value of oxygenated fuel decrease as the oxygen content increases that makes the BSFC higher which lead for need of more fuel to keep the same engine output (Kumar, Cho et al., 2013; Yilmaz & Vigil, 2014). The decrease of density is also another factor that contributes to higher BSFC as lower density indicted to a lighter fuel which made it easier to inject more fuel into the combustion chamber. Overall, available literatures showed that long-chain alcohols, such as butanol and pentanol have advantage than minimum BSFC compared to short-chain alcohol, ethanol and methanol.

Besides that, previous researchers have also shown similar trend of decrease of BSFC with an increase in engine load (Alptekin, Canakci et al., 2015; Qi, Chen et al., 2011). The BSFC curve plotted in decreasing trend is due to higher fuel combustion efficiency in higher engine load related to higher in-cylinder temperature.

2.5.2 Brake thermal efficiency

Definition of brake specific thermal efficiency (BTE) is the measured ratio of the work performed by the engine to the heat supplied with engine, which is stated as the efficiency of the heat engine (Atmanlı, Yüksel et al., 2013). There are several reports describing BTE in their experimental work which indicated thermal combustion efficiency.

In an experimental study of (Subbaiah, Gopal et al., 2010), it was found that higher BTE for all the diesel–biodiesel–ethanol blends compared to DF at all engine loads conditions. The comparable result showed that BTE increase by 1.5%, 2.2% and 2.91% in addition of 5%, 10% and 15% ethanol in the ethanol-biodiesel-diesel blends. 15%

ethanol-biodiesel-diesel was recorded as the best fuel with maximum BTE 28.2% higher than biodiesel and 3.67% higher than DF. The reason for this increase of BTE for biodiesel and biodiesel blends might be due to the extended ignition delay and their leaner combustion. These authors also found that the BTE increased because the ethanol in the blend increased due to a reduction of density and viscosity. There are also some researchers who found BTE of ternary blends to be lower than diesel fuel (Barabás & Todoruţ, 2011). This decrease is between 1.3% and 21.7% when only 10% diesel and 10% ethanol is used in ternary blends. In this case, the decrease of BTE is due to the higher content of biodiesel in blends.

Another experiment made by (Atmanlı, Yüksel et al., 2013) stated that the presence of n-butanol reduce BTE of n-butanol-croton oil-diesel blend. This reduction may be because of both cotton oil and n-butanol had a lower calorific value which lower the heat of combustion and hence providing less BTE due to poorer atomization and poorer combustion efficiency. Moreover, the lower density and a cetane number of n-butanol of DF caused a prolong injection duration and ignition delay, therefore more fuel will burn during the expansion of stroke with relatively lower combustion efficiency, and thus decreases BTE. However, blending long-chain alcohol with DF may only result in the increase of BTE as reported in several studies. As such, in an example of (Campos-Fernández, Arnal et al., 2012) study who proved that the addition of 1-butanol and 1-pentanol has led to higher BTE. These authors further stated that the presence of oxygen in fuel blends will involve a higher combustion efficiency and a reduction of heat losses due to the lower boiling point of butanol and pentanol compared to DF. Similar finding was reported by (Sathiyagnanam, Saravanan et al., 2010) which used long-chain alcohol, hexanol as a co-solvent to ethanol-diesel fuel blends. The study found hexanol to give higher BTE to ethanol-diesel fuel blends.

In general, both alcohols, short-chain or long-chain alcohols may increase the BTE of engine performance. Furthermore, the ignition delay caused a rapid rate of releasing energy which reduces the heat loss because there is a shorter time for this heat to leave the cylinder through heat transfer to the coolant. The higher premixed combustion part exhibited by the alcohol blends was due to their lower cetane number, leading to a higher percentage of 'constant volume' combustion and lower cylinder gas temperature and better combustion efficiency.

2.5.3 Brake power and torque

In regard to this, there are few authors who have reported the effect of alcohols on engine power and torque. Based on their findings, they reported that the engine power and torque dropped with the use of alcohols due to its lower calorific value. In research by (Guo, Li et al., 2011), the engine power slightly decreased with the increased proportion of methanol (MTD). These authors reported that the maximum engine power and torque for DF was 11.9 (kW) and MTD10, MTD20 and MTD30 were 11.69 (kW), 11.67 (kW) and 11.66 (kW). In comparing to DF, the engine power decreased to about 1.7%, 1.9% and 2.0% with MTD10, MTD20 and MTD30 at 2000 r/min full engine load operating conditions. This can be explained by the fact that the deceased of calorific values of the blends with MTD and the engine operating parameters were not adjusted which reduced the engine power correspondingly when blend fuels were used. While the maximum torque point of the engine was found at 1200 r/min and DF was 63.3Nm, MTD30, 62.5 Nm respectively, with a maximum decreasing ratio of 1.3%. Another reason for reducing torque was due to low cetane number of blend fuels, which affected combustion characteristics. A review by (Shahir, Masjuki et al., 2015) also reported the reducing effect of ethanol on engine power and torque output. Their findings stated that engine power and torque, reduced as the portion of oxygenated compounds (biodiesel and ethanol/bioethanol) in the blends increases. The possible reason was due to the low cetane number and calorific value and higher ignition delay of the blends, compared to DF.

Moreover, (Ileri, Atmanli et al., 2016) also reported the same findings when they reported lower brake torque and brake power for fuel blends containing n-butanol than DF, which is due to lower calorific value and cetane number. In addition, higher density and viscosity is also another cause leading to the lower brake torque and brake power due to poor spray characteristics in the injector. These authors also stated that high latent heat of evaporation of n-butanol creates cooling effect in the combustion chamber, decreases combustion temperature and influences combustion efficiency negatively, as a result, brake torque and brake power decrease. In the same year, (Ileri, 2016) also discovered the same thing, as lower brake power was found with addition of n-butanol and pentanol. The findings also reported 10% butanol and pentanol production by 3.94% and 2.46% with lower maximum brake power compared to DF, respectively. The reasons were due to the higher densities and kinematic viscosities of fuel blends that caused poorer atomization and lower combustion efficiency despite having higher oxygen content in their chemical structure. This was because the oxygen content of the fuel created fuel-lean regions in the combustion chamber which provided some advantages in terms of exhaust gas emissions, also causing a reduction of the brake power as a result of lower calorific value.

2.5.4 Exhaust gas temperature

Previous research by (Yasin, Mamat et al., 2015) found higher exhaust gas temperature for the biodiesel-methanol-diesel blends in comparison to biodiesel, B20 and mineral diesel. In view of this, it can be seen that biodiesel-methanol-diesel blends produce increase by 4.96% for B20 M5 and 12.21% for B20 M10 in exhaust temperatures compared to DF. The reason may be due to the higher oxygen content of the biodiesel-ethanol-diesel blends that could increase the exhaust gas temperature.

In 2017, (Gangwar, Saraswati et al., 2017) found the opposite result of the variation of exhaust gas temperature versus speed, at constant engine load. They reported using 1-butanol and 1-propan-ol fuel blends to give a lower effect on the exhaust temperature than that for DF. Furthermore, having increase in alcohol percentage of 1%, 2% and 3%, may reduce the exhaust temperature. This can be explained as 1-butanol and 1-propanol diesel fuel blends with slightly higher thermal efficiency due to lower calorific value and the higher latent heat of evaporation that finally leads to lower exhaust temperatures.

This can be supported by an investigation made by (Rajesh Kumar & Saravanan, 2015) who found that exhaust temperature decreases with increasing pentanol content in the blend. According to them, this can assume the lower energy content in pentanol, causing less combustion temperatures. Their findings also reported that 45% pentanol has the lowest energy content among all the blends and produces lowest exhaust temperatures at all engine loads. Moreover, the higher latent heat of vaporization of pentanol-diesel blend also reduces the exhaust temperature.

In addition, the decrement or increment of exhaust gas temperature depends on what type of alcohols used. The longer chain alcohol was expected to give lower effect on exhaust gas temperature due to the higher latent heat of vaporization that gives cooling effect in the combustion chamber. It is also expected from all cases that the exhaust gas temperature increases regardless of the engine loads.

2.6 Exhaust emission of alcohol-diesel fuel blends

2.6.1 Carbon monoxide emission

Generally, fuel blends containing short-chain alcohol such as ethanol and methanol produced higher CO emission compared to DF. This was due to fact that higher oxygen atom in ethanol and methanol have stronger bond with carbon atom, which difficult to break especially because of decrease of EGT. This may lead to incomplete combustion and increase of CO emission formed. In addition, (Khandal, Banapurmath et al., 2015) stated in their study, lower cetane number and high latent heat of evaporation of ethanol caused for less vaporization and hence very less time to burn fuel completely that results in considerable increase in CO emissions.

Previously, (Mahalingam, Munuswamy et al., 2018) found positive effect of pentanol on CO emission as CO formed reduced for 3.1% to 4.2% with the introduction of pentanol to biodiesel. This is due to the increase of oxygen atom in fuel blends that increase available oxygen which enhanced combustion process. With the same reason, (Mahalingam, Devarajan et al., 2017) also find lower CO emission with increase proportion of n-octanol. Authors also stated that CO emissions from mahua oil biodiesel with n-octanol are reduced due to enhanced burning pace. The oxygen atoms of n-octanol enhance the availability of oxygen during combustion.

2.6.2 Carbon dioxide emission

According to (Gomasta & Mahla, 2012), higher CO_2 emission with reduction of HC emission and CO emission indicates a successful combustion. CO_2 is a normal product of combustion. However, most literature found lower CO_2 with the addition of alcohol, due to the increase of H and O_2 molecule in the fuel blends. An example of lower CO_2 was also reported by (Alptekin, 2017) in his study, where he stated addition of ethanol to biodiesel contribute to lower CO_2 formation. The result was consistent with another study by (Alptekin, Canakci et al., 2015) as lower CO_2 was reported in ethanol biodiesel. In the same study, the authors said the reason for the decrease CO_2 with the addition of ethanol was due to low C/H ratio in the fuel blends.

Long-chain alcohol, such as pentanol and n-butanol also gave the same effect on CO_2 emission. (Imdadul, Masjuki et al., 2016) found that the addition of pentanol at 10%, 15% and 20% for biodiesel have decreased the CO_2 formed. (Ileri, Atmanli et al., 2016) also supported the finding of lower CO_2. In his study, they found lower CO_2 in n-butanol-biodiesel fuel blends. Both studies stated that the increasing O_2 and H molecules in fuel structures have decreased CO_2. This was because O_2 needed by alcohol-fuel blends were less than DF due to their higher content of O_2. However, the usage of long-chain alcohol was better than short-chain alcohol as long-chain alcohol contain less O_2 for higher CO_2. Moreover, the higher carbon chain of long-chain alcohols also can be added as advantage for higher CO_2 formed.

2.6.3 Hydrogen carbon emission

According to previous studies, the short-chain alcohol-diesel fuel blend or biodiesel-short-chain alcohol-diesel fuel short-chain have increased HC emission formed. HC emission is a product of incomplete combustion. In 2012, (Yilmaz, 2012) showed that 10% and 20% ethanol-blended fuels have a significant increase of HC emissions as compared to DF. He also found that 10% and 20% methanol-blended fuels have constant result with DF. The results from (Yilmaz, 2012) indicated that, methanol proportion up to 20% may increase oxygen content of the mixture, which leads to better combustion and lower HC emissions where with more methanol addition, it is expected to show a cooling effect, which causes incomplete combustion and higher HC. Besides that, ethanol concentration should be much lower than 10% in order to have the positive effect of higher oxygen content, rather than the cooling effect of ethanol.

The increase of HC emissions was also found in a study using long-chain alcohol diesel fuel blends. A study by (Atmanli, 2016b) explained that, the used of n-butanol and n-pentanol have increased HC emission formed, due to high latent heat of evaporation. Higher latent heat of evaporation caused sudden temperature dropped, quenching effect and expand, cooling region, resulting in incomplete combustion and increased HC formed. His study used cetane improver, 2-ethylhexyl nitrate (EHN) to reduce HC emission. A similar finding was reported by (Yilmaz & Atmanli, 2017b) where pentanol-diesel blends at different ratio emitted higher HC emission. Further explanation show that HC emission increased as decreased in cetane number and EGT affected the slower combustion.

2.6.4 Nitrogen oxide emission

Generally, the usage of alcohols may increase the NOx emission. Several reasons were reported to increase NO_x emission. The first was the increase of oxygen in the fuel, which might increase the formation of NO_x. Secondly, the lower cetane number can also lead to an increase in the combustion temperature, increasing NO_x emission. However, higher latent heat of evaporation of alcohol may result in a cooling effect that might lower the combustion temperature, and hence reduce NO_x formation. Thirdly, alcohol can lead to an increase of burn in the premixed mode, because of its lower cetane number and thus an increase in the combustion temperature.

Previously, (Sayin, 2010) reported that there was an increase in NO_x emissions of about 23.6%, 17.4%, 13.1% and 11.3% for 10% methanol, 5% methanol, 10% ethanol and 5% ethanol. This author stated that, fuel blends have higher NO_x emission than DF despite having the higher latent heat of evaporation. The reason was due to the lower cetane number and higher oxygen content that is more effective to increase peak temperature in the cylinder. Therefore, the concentration of NO_x increased as the alcohol content increased in the fuel blend.

A study by (T. Zhang, Munch et al., 2015) also reported the increase of NO_x emission due to higher peak pressure and a peak temperature of fuel blends containing butanol and octanol. In addition, the lower calorific value of butanol and octanol were another reason for higher NO_x emission. However, opposite results of decrease NO_x were found by (Doğan, 2011). The author explained that despite having lower cetane number and increasing O_2 content, n-butanol-diesel fuel blends generally have lower flame temperature due to their lower energy content and higher heat of evaporation, thereby spotted the falling trend of NO_x in their study. Their result was consistent with the study of (Li, Wang et al., 2015) as the reduce in NO_x was reported with increasing content of pentanol. The reason for decreased NO_x was due to the strong mixing and homogeneity increase in the fuel blends, because the ignition delay increases. Thus, the premixed combustion became weaker and it relatively lowers the combustion temperature and the combustion that reduced formation of NO_x.

2.7 Theories

2.7.1 Peak pressure curves

The cylinder geometry, V and pressure data, $\boldsymbol{dV/d\theta}$ terms are shown in Equation 2.1 and Equation 2.2 below.

$$V = VC + Ar\left[1 - \cos\left(\frac{\pi\theta}{180}\right) + \frac{1}{\lambda}\left\{1 - \sqrt{\lambda^2 \sin^2\frac{\pi\theta}{180}}\right\}\right] \quad 2.1$$

and

$$\frac{dV}{d\theta} = \left(\frac{\pi\theta}{180}\right) \times r\left\{\sin\left(\frac{\pi\theta}{180}\right) + \frac{\lambda^2 \ \sin^2\left(\frac{\pi\theta}{180}\right)}{2\times\sqrt{1-\lambda^2 \ \sin^2\left(\frac{\pi\theta}{180}\right)}}\right\} \quad 2.2$$

2.7.2 Heat release rate

The formula in Equation 2.1 is heat release rate (HRR) arranging and simplified from the first law of thermodynamic, an open quasi static system. In the experiment, heat release rate obtained from cylinder pressure data and crank angle data.

$$\frac{dQn}{d\theta} = \frac{1}{\gamma-1}\left(\gamma p\frac{dV}{d\theta} + V\ \frac{dp}{d\theta}\right) \quad 2.3$$

where $\frac{dQn}{d\theta}$ = Heat release rate, γ = the ratio of specific heats, c_p/c_v, V= cylinder geometry.

2.7.3 Brake specific fuel consumption

The brake specific fuel consumption (BSFC) was calculated using Equation 2.4.

$$\text{BSFC} = \frac{mass\ flow\ rate\ (g\ s^{-1})}{engine\ power} \times 3600 = \frac{\dot{\text{m}}}{\text{Pe}} \quad 2.4$$

where BSFC is brake specific fuel consumption, ṁ is the consumed fuel amount (gs^{-1}) and P_e is the engine power

2.7.4 Brake thermal efficiency

In Equation 2.6, the brake thermal efficiency (BTE) is expressed in formula. The fuel power is calculated using Equation 2.5.

$$Fuel\ power = mass\ flow\ rate \times calorific\ value \qquad 2.5$$

$$\text{BTE} = \frac{Engine\ power}{fuel\ power} \qquad 2.6$$

2.7.5 Brake power and torque

The brake power (B.P.) of an internal compression (IC) diesel engine is the power measured at the crankshaft. The brake power of an IC engine is, usually, measured by means of a brake mechanism of prony brake or rope brake. The brake power can be calculated using Equation 2.7.

$$\text{Brake power (W)} = \frac{2\pi NT}{60} \qquad 2.7$$

where the N is the rpm and T is the torque.

Next, torque is a term use to describe turning or twisting force. It is about tendency of a force to rotate an object about an axis. In engine, torque is the measured rotational effort applied on engine crankshaft by the piston. The formula to calculate torque as in Equation 2.8.

$$\text{Torque (T)} = \text{WR} \qquad 2.8$$

where W = net load, R = radius

2.8 Optimization: Response Surface Methodology (RSM)

Optimization was developed to improve performance of a system, a process, or a product that an applied procedure produces the best possible response. According to (Abuhabaya, Fieldhouse et al., 2013) and (Atmanlı, Yüksel et al., 2015), the most relevant multivariate techniques used in analytical optimization is response surface methodology (RSM). In complex variables processes, conducting many experiments would be time consuming and expensive. It is essential to have a well-designed experimental plan in order to capture more information from fewer experiments compared to the conventional methods (one factor at a time). RSM is a statistical and mathematical tool useful for analyzing, modelling, optimizing and determining the interactions between the variables and responses (Adam, Aziz et al., 2016). The aim is to build models, evaluate the effects of variables and establish the optimum performance conditions by means of experimental design and regression analysis. In the RSM the relationship between the responses and variables is presented by Equation (2.5).

$$\gamma = f(x_1+x_2+x_3+\ldots\ldots\ldots x_n) \pm \varepsilon \qquad 2.9$$

where y is the dependent variable, f is the response function, xi are the independent variables and ε is the fitting error.

2.8.1 Response Surface Methodology using alcohol based fuels.

The Response Surface Methodology (RSM) was a powerful tool that designed with an objective to obtain the best performance of the engine by using alcohol based fuels. There are several type of method that used to design RSM which are Central Composite Design, Box-Behnken, One Factor, Miscellaneous, D-Optimal, Distance Based, Used Defined and Historical Data (Shirneshan, Almassi et al., 2014; Yusri, Mamat, Azmi et al., 2017) (D. Patel, Lakdawala et al., 2015). Method to design RSM models was developed according to suitable variable and responses.

A study by (Khoobbakht, Najafi et al., 2016) aimed to investigating the effect of operating factors of engine load and speed as well as blended levels of biodiesel and ethanol in diesel fuel on the emission characteristics of DI diesel engine. The experiments were designed using a statistical tool known as Design of Experiments (DoE) based on central composite rotatable design (CCRD) of response surface methodology (RSM). The resultant quadratic models of the response surface methodology were helpful to predict the response parameters such as oxides of nitrogen (NOx), carbon monoxide (CO), carbon dioxide (CO2) and total hydrocarbon (THC) and smoke opacity and further to identify the significant interactions between the input factors on the responses (Fang, Kittelson et al., 2015; Khoobbakht, Najafi et al., 2016). The authors reported that biodiesel and ethanol could reduce CO and HC emissions as well as smoke opacity and enhance CO2 which indicated a more quality in fuel combustion. However, over adding these biofuels in diesel led their detrimental impacts to be dominant over advantages of biodiesel and ethanol and as result emissions tended to increase. Optimization of independent variables was performed using the desirability approach of the response surface methodology with the goal of minimizing CO, THC, NOx and smoke opacity and maximizing CO2. An engine load of 80% of full load bar, speed of 2800 rpm and a blend of 26% biodiesel, 11% ethanol and 63% diesel were found to be optimal values with a high desirability of

74% for the test engine having 0. 013% of CO, 41 ppm of HC, 643 ppm of NOx, 12% of smoke opacity and 7.3% of CO2. Similar study provided by (Fang, Kittelson et al., 2015) was successful to find the correlation of variables such as engine load, speed and blends ratio on response which are emissions. In addition, (Najafi, Ghobadian et al., 2015) also find the correlation between blend ratio and loads to engine performance such BSFC and BTE. The optimization of minimum BSFC and maximum BTE to find the best set of speed, load and blend ratio.

Moreover, (Atmanlı, Yüksel et al., 2015) used in-depth mathematical optimization to analyses diesel butanol vegetable oil (cotton oil), based on engine operating parameters using RSM (response surface methodology). The objective of authors was to achieve the maximum power and torque for customers while keeping the emissions low enough due to government regulations and certifications. Thus, in the RSM three optimization studies were conducted at 2200 rpm, which corresponds to the maximum brake torque, and engine emissions were fixed at a maximum possible value based on emission standards, for all three studies. In order to understand the impact of other engine parameters on the blend ratio, as well, various combinations of BTE (brake thermal efficiency), maximum brake power, maximum brake torque, BSFC (brake specific fuel consumption) and BMEP (brake mean effective pressure) were fixed. Optimization studies used experimentally determined emissions and performance data of a diesel engine based on 7 different concentrations of diesel-butanol-cotton oil blends. Optimum values of the blends corresponding to the optimization studies were mathematically determined as Opt-1 (optimization 1) (61.7 vol.% diesel, 34.75 vol.% butanol, 3.55 vol.% cotton oil), Opt-2 (optimization 2) (64.5 vol.% diesel, 28.7 vol.% butanol, 6.8 vol.% cotton oil), and Opt-3 (optimization 3) (65.5 vol.% diesel, 23.1 vol.% butanol, 11.4 vol.% cotton oil). When compared to diesel, BSFCs of Opt-1, Opt-2 and Opt-3 blends at 2200 rpm increased 41.57, 33.87 and 24.53%, respectively. In terms of basic exhaust gas emissions, optimum fuel blends decreased NOx (oxides of nitrogen), CO (carbon monoxide) and HC (hydrocarbon) emissions as compared to diesel found at the end of the experiment.

In another study by (Saravanan, B et al., 2017), authors carried out an experimental and statistical investigation is carried out to analyze the effects of injection-pressure, timing and exhaust gas recirculation (EGR) on performance and emissions of a

DI diesel engine fueled with 40% by vol. of iso-butanol/diesel blend. Response surface methodology was used to model all measured responses like nitrogen oxides (NOx), smoke opacity, brake thermal efficiency (BTE) and brake specific fuel consumption (BSFC). Analysis of variance (ANOVA) revealed that all developed models were statistically significant. Interactive effects between injection pressure, injection timing and EGR for all blends were analyzed using response surface plots that were plotted using developed regression models. Optimization was performed using desirability approach of the RSM with an objective to minimize NOx and smoke emissions simultaneously with maximum BTE and minimum BSFC. Iso-butanol/diesel blend injected at 240bar pressure, 23°CA bTDC under 30% EGR was predicted to be optimum for this particular engine. The predicted combination was validated by confirmatory tests and the error in prediction was found to be within 4%.

Besides that, (Rajesh B., Muthukkumar T. et al., 2016) also utilizes three high carbon bio-alcohol/diesel blends prepared by mixing 40% by vol. of n-propanol, n-butanol and n-pentanol individually with fossil diesel in a DI diesel engine in their study. The RSM was developed using historical data which designed using previous experimental data. Engine performance and emission characteristics were measured under high-load conditions based on a 33 full-factorial experimental design matrix using exhaust gas recirculation (EGR) rate, injection-timing and alcohol type used in the blends as factors for controlling charge-dilution and combustion-phasing. A statistical investigation was then carried out to compare and analyzed the effects of these factors on all measured responses like nitrogen oxides (NOx), smoke, hydrocarbons (HC), carbon monoxide (CO), brake thermal efficiency (BTE) and brake specific fuel consumption (BSFC). Multiple regression models were developed for all responses using a response surface methodology (RSM) and were found to be statistically significant at 99% confidence levels. Interactive effects between injection timing and EGR for all blends were compared and analyzed through response surface plots fitted using developed models with high R2 values. Optimization was performed using a desirability approach with an objective to minimize NOx, smoke and BSFC with maximum BTE. n-Propanol/diesel blend injected at 25° CA bTDC under 30% EGR with a desirability of 0.965 was predicted to be optimum for this engine. Similarly, n-butanol/diesel and n-pentanol/diesel blends injected at 24° CA bTDC under 10% EGR were found to be optimum in their respective category. Confirmatory tests validated that the developed

RSM models were adequate to describe the effects of injection timing and EGR on the engine characteristics as the predicted error is within 5%.

Next, (Campos-Fernandez, M. Arnal et al., 2013) also developed an RSM modelling to evaluate the performance of a direct-injection diesel engine, without any modifications, fueled with 1-pentanol/diesel fuel blends. Blends with 10% pentanol/90% diesel fuel, 15% pentanol/85% diesel fuel, 20% pentanol/80% diesel fuel and 25% pentanol/75% diesel fuel (v/v) were tested and engine performance results were compared with those provided by neat diesel fuel. Experimental results showed insignificant engine power, brake thermal efficiency and brake-specific fuel consumption variations when the engine was fueled with the majority of the blends instead of straight diesel fuel. Moreover, statistical analysis showed no significant differences between the blends and diesel fuel (EN 590) tests. During engine starting, no difficulties were experienced and the engine performed satisfactorily on the blends throughout the entire test. On the basis of this study, pentanol/diesel fuel blends can be considered acceptable diesel fuel alternatives if exhaust emissions and long-term engine tests show acceptable results.

From the previous studies, it was revealed RSM was proven to be a powerful tool to study the correlation between variables or input factors and responses. The example of input factors such as loads, blend ratio and speed to determine responses such BTE, BSFC, EGT and emissions. In addition, the selection of model in RSM was influenced by higher desirability and p-value less than 0.05. While, the optimization was determined by study the set of experimental run that obtain the best performance as the objectives. The lack of study of alcohol based fuels in optimization made finding was limited. However through study in previous literature, the most common methodology was Central Composite Design and Historical data.

2.9 Summary

In order to reduce the emissions, the researchers and manufacturers have been triggered to take different approaches, including producing new alternative fuels. New alternative fuels, such alcohol-based fuels have been shown to reduce the particulate emissions. Thus, alcohol have been a potential oxygenated additive to blend with pure diesel, DF. Short-chain alcohol such as propanol, methanol, and ethanol had been widely used in studies. Propanol, methanol, and ethanol had lower miscibility with DF as it possesses a lower energy density compared DF. Based on Table 2.1, lower cetane number of ethanol, thus cetane improver is required in ethanol-diesel fuel blends as in the study by (Ciniviz, Örs et al., 2017). Furthermore, lower flashpoint, boiling point and viscosity also effect power safe supply lubricity and storage. According to (Balki, Sayin et al., 2014), it had been revealed that the addition of short-chain alcohol; methanol, ethanol and propanol had higher in-cylinder peak pressure and HRR. However, short-chain alcohols degrade the performance of diesel engine as higher BSFC and lower BTE reported (Oliveira, Morais et al., 2015; Yasin, Mamat et al., 2015). From the previous studies, propanol, methanol and ethanol had reduced some particulate emissions such as CO_2 and HC, but reported higher CO, NO_x and NO_2 (Alptekin, 2017; Shahir, Masjuki et al., 2015; Zhu, Cheung et al., 2011).

Thus, longer-chain alcohols, such as butanols (C_4 alcohols) have been studied to overcome the weakness of ethanol-diesel fuel blends. From literature, butanols capable to mix well with 100% neat DF and various biodiesel (Atmanlı, Yüksel et al., 2013; Campos-Fernández, Arnal et al., 2012; Emiroğlu & Şen, 2018a). The reason was due to higher energy density, cetane number, viscosity, flashpoint and boiling point compared with ethanol (Campos-Fernández, Arnal et al., 2012). In 2016, (Balamurugan & Nalini, 2014b) found CO and NO_x emissions reduced with presence of 4% and 8% butanol. The possible explanation is due high latent heat of vaporization and calorific value of butanol reduced the operating temperature, reduced CO and NO_x. Similar trend of falling formation of NO_x reported by (Atmanli, 2016a) as authors found that longer chain alcohol had higher cooling effect. Additionally, (Ileri, Atmanli et al., 2016) found higher CO_2 and reduced HC emission which indicated a more complete combustion. Thus many researchers start replaced ethanol with butanol as additives in DF or biodiesel which later investigated the engine combustion.

Recently, the development of alternative fuels had introduced long-chain alcohols, pentanol (1PN) and 2-ethyl 1-hexanol (2-EH) as oxygenated additives in alcohol-diesel, alcohol-biodiesel and diesel-alcohol-biodiesel. As shown in Table 2.1, the physical-chemical properties of 1-PN and 2-EH (long-chain alcohols) are better than ethanol (short-chain alcohol) and even than butanol. The thermo-physical properties of 1-PN and 2-EH were closer to physical-chemical properties of DF in terms of density, calorific value, cetane number and viscosity. The percentage of O_2 content in 1-PN and

2-EH is higher compare to ethanol and DF. With higher cetane number and density, the performance and combustion of the DF can be improved during engine testing. Thus, better combustion expected and emissions can be improved at the end of experiment. The thermo-physical properties of long-chain alcohol are better than the short-chain alcohol which is closer to physical-chemical properties of DF. Through previous studies, long-chain alcohol seems to be capable of replacing short-chain alcohol for alcohol-diesel blend.

Pentanol is a five carbons alcohol in its atomic structure which shown potential as oxidation liquid as it has better overall physical-chemical properties than ethanol and even butanol. Low polarity and being hydrophobic leads pentanol to form homogenous mixing with DF and even with biodiesel, without obvious phase separation shown when blended with DF due low polar interaction parameter which make pentanol more promising and reliable. According to a study by (Campos-Fernández, Arnal et al., 2012), 1-pentanol can be added up to 25% by volume without any engine performance problems. In the study, the 10%, 20% and 30% pentanol-diesel fuels blends exhibit similar heat release rate curves with pure diesel fuel with slight increase at the peak. Meanwhile, the brake specific fuel consumption (BSFC) for pentanol-diesel fuels blends is the lower with DF and higher in brake thermal efficiency (BTE). Similar investigation reported by (Wei, Cheung et al., 2014) as the authors also investigated pentanol at the same ratio (10%, 20% and 30%). In the paper, authors stated that n-pentanol and DF can be blend up to 30% by volume without any additional solvents at room temperature. The addition pentanol enriched oxygen content, as well as improved both the premixed and diffusive combustion stage. However, due to the blends low CN, the ignition delay was longer with the addition pentanol. For gaseous emissions, HC and CO emissions were increased with increased volume of pentanol in the blends especially at low and medium engine loads due to the low number of cetane for the pentanol-diesel fuel blends (Atmanli, 2016a; Wei, Cheung et al., 2014; Zhu, Xiao et al., 2016). Opposite to the results, (Agrawal, Sharma et al., 2015) and (Imdadul, Masjuki et al., 2016) found reduced of CO and HC was achieved in their works. (Imdadul, Masjuki et al., 2016) stated that the decreasing of CO and HC was due to the increase of O_2 in fuel blends led to complete combustion. Through previous literature studies, NO_x emissions had increased with addition of pentanol especially at higher load (Imdadul, Masjuki et al., 2016; Yilmaz & Atmanli, 2017b). NOx formation at higher engine load was attributed with higher in-cylinder temperature due to the long ignition delay. Moreover, increase in n-pentanol will increase the formation NO_2 emissions affected by the e-OH functional group. However, (Yilmaz & Atmanli, 2017b) found less NO_X for 5% pentanol with 95% DF even though higher percentage content of pentanol has higher NO_X than DF.

Another potential long-chain alcohol is 2-ethyl 1-hexanol which is also known as an organic alcohol with eight-carbon chain. This type of hexanol also known as octanol due similar chemical structure. According to my best knowledge, there only a few researchers that use 2-ethyl 1-hexanol their studies (De Poures, Sathiyagnanam et al., 2017; Suhaimi, Adam et al., 2018). In a journal, (Duncan, Adebayo et al., 2007) indicates that 5% hexanol with DF shows properties similar to DF and higher aniline point, hence the fuel blends can be used well in diesel engine with less emission. It was also reported that the fuel blend density, flash point and viscosity were above the requirement than DF according to ASTM standard. In order to find an alternative fuel instead of fuel diesel with less emission, authors studied the effects of hexanol-diesel blends at 5% to 45% by volume. The results showed improvement in performance with hexanol-diesel blends compared to DF with less smoke formed but increase of NOx emissions. Increase in hexanol ratio influence combustion analysis by increasing the maximum peak pressure as well as rate of pressure. The heat value of the hexanol was lower than heat value of the blend and hence increases the BSFC. The presence of O_2 due to the addition of hexanol in the DF improves the combustion, especially diffusion combustion and hence increases the BTE. This can be associated due to the higher premixed combustion of the blends because of the lower cetane number of hexanol, thus resulted increase in percentage of "constant volume" combustion, and to the lower heat losses and "leaner" combustion as explained by (T. Zhang, Munch et al., 2015). Long chain alcohols contain O_2 molecule for better combustion process of the engine. (T. Zhang, Nilsson et al., 2016) found the addition of 2ethylhexanol had increased formation of NO due higher O_2 content. In another experiment, hexanol was used as the one of the co-solvents which used ethanol-diesel blend fuels and biodiesel, neat shell oil (A. K. Pandian, Munuswamy et al., 2018) The additional of hexanol was to improve blend tolerance for ethanol-diesel and stabilize blend fuel. By adding hexanol, ethanol-diesel blend fuels can be stored for longer time. In addition, the smoke emission was reported to have decreased significantly with the rise of oxygen content in the fuels (Sathiyagnanam, Saravanan et al., 2010). This concludes that, the usage of hexanol as a co-solvent or a direct blend with DF, the hexanol is a high potential long-alcohol that could improve engine performance, fuel combustion characteristic and exhaust gas emission due to its better physical-chemical properties of fuel blends compared to short-chain alcohol-diesels fuel blends. Another investigation by (Devarajan, Munuswamy et al., 2018) found reducing effect of n-octanol to biodiesel on formation of CO, HC and NO_x. The explanation of decrease of HC and CO was due to the n-octanol enhance combustion rate and O_2 supplied. And, the lower heating value of n-octanol gave cooling effect which lower exhaust temperature, led to decrease NO_x.

The above reviews show that there is lack of investigation on the application of 1-pentanol and 2-ethyl 1-hexanol to diesel engines, especially on its influence on particulate emissions. The objective of the present work is to examine the potential of 1-pentanol and 2-ethyl 1-hexanol as an additive to diesel fuel. Specifically, for 5%, 10% and 20% by volume of 1-pentanol and 2-ethyl 1-hexanol in the blended fuel, the overall gaseous emissions. It is observed that previously most researchers do not include carbon dioxide (CO_2) and exhaust gas oxygen (EGO) in their studies. CO_2 and EGO are important to analyses the combustion. Thus, overall exhaust gas emissions such as the CO, CO_2, HC, EGO and NO_x emissions are investigated. Moreover, interestingly that 1-PN and 2-EH are new generation alcohols to replaced short-chain alcohols.

CHAPTER 3

METHODOLOGY

3.1 Introduction

In this chapter, the details about the methods used in the project were discussed. The process and streamline of the project was utilized to carry out all the steps from the beginning of the projects until the project ends. The summary of methodologies used was illustrated in Flowchart 1. All experimental procedures were conducted in order to achieve all the objectives of the project. The experiment was designed carefully to study the combustion characteristic, performance and emissions of diesel engine fueled with long-chain alcohol-diesel fuel blends.

3.2 Thermophysical properties test

The thermophysical properties test was conducted to study the stability and properties of the fuel blends. The measured properties were density, cetane number, kinematic viscosity and calorific value. The equipment and ASTM standard used were listed in Table 3.1. The details procedures conducted will be explained further.

Table 3.1 Table of equipment for properties tested

Properties	Equipments	ASTM
Density	Microbalance, A&D GH-252	ASTM D1298
Kinematics Viscosity	Viscometer, G.D 265-D	ASTM D445
Calorific Value	Bomb calorimeter, Parr 6772	ASTM D976

3.3 Preparation of long-chain alcohol-diesel fuel blends

The long-alcohols such pentanol (1-PN) and 2-ethyl 1-hexanol (2-EH) were mixed with pure diesel (DF) at 5%, 10% and 20% in volume (v/v). Details of fuels were recorded in Table 3.2. The blending process was done by using Hielscher UP400S ultrasonic processor referring to the figure. The apparatus and equipment used for blending and store the fuel blends were washed and wiped with laboratory acetone. The precaution was done to prevent impurity of the mixed the fuel.

3.3.1 Preparation method

The fuel blends were prepared for thermo-physical test and engine performance test. The blending procedure was conducted by using Hielscher UP 4000S as in Figure 3.1. Amplitude at 40% and the cycle at 0.5 were set as the blending parameter. The lid was closed and stirring process was set for 2 minutes. The blending mixture was named accordingly to PE5, PE10, PE20, HE5, HE10, and HE20. Name and details of test fuels refered in Table 3.2. The details procedure was listed in APPENDIX A.

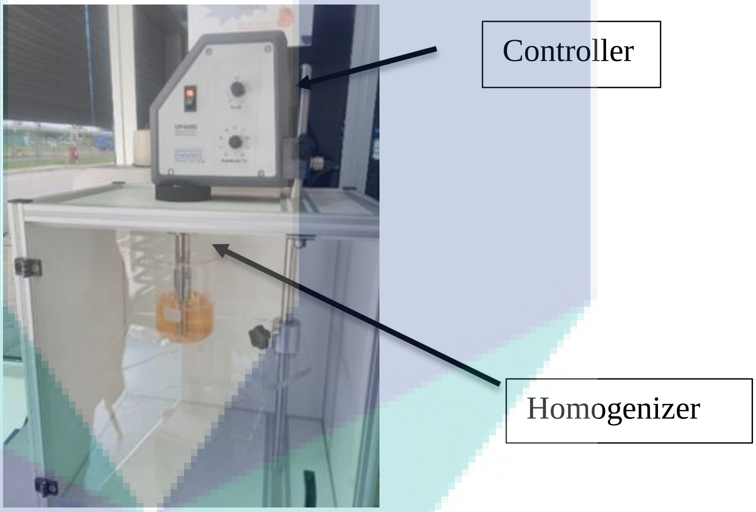

Figure 3.1 Hielscher UP400S.

Table 3.2 Details of test fuels

Test fuels	Percentage of fuels (v/v)
DF	100% DF
PE5	5% PN, 95% DF
PE10	10% PN, 90% DF
PE20	20% PN, 80% DF
HE5	5% 2-EH, 95% DF
HE10	10% 2-EH, 90% DF
HE20	20% 2-EH, 80% DF

3.3.2 Density measurement

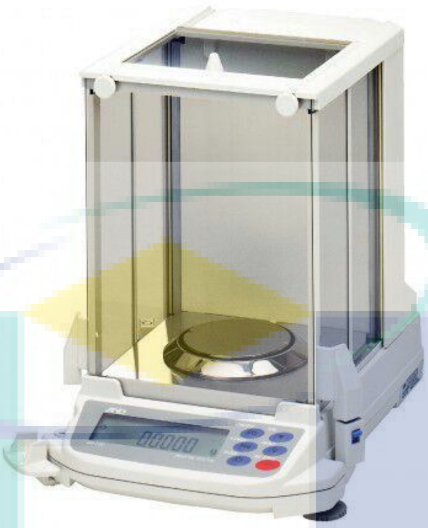

Figure 3.2 Microbalance GH-252

The density of the fuel blends measured using A&D GH-252 precision microbalance. The microbalance was capable to weight precise measurements of relatively small mass of the order of a million parts of a gram. Microbalances are generally used in a laboratory as standalone instruments but are also incorporated into other instruments, such as thermos-gravimeter, sorption/desorption systems, and surface property instruments. The readability of this microbalance model was 0.00001g. The precaution to take was to make sure there is no fuel that spill out from the beaker and close the microbalance cover to prevent misreading. The procedure undertaken strictly following ASTM D1298 standards. Procedure was detailed in APPENDIX B.

3.3.3 Kinematic viscosity measurement

Kinematic viscosity testing was carried out at laboratory in Faculty of Chemical and Natural Sources (FKKSA), Universiti Malaysia Pahang. The kinematic viscosity was measured using an individually tank viscometer model G.D 265D. The precision of the viscometer was ±0.2% as stated in the specification sheet. For measurement of kinematic viscosity of transparent Newtonian liquids, particularly petroleum products or lubricants, the ASTM D445 and ISO 3104 requirements were followed. From Equation 3.1 and 3.2, kinematic viscosity and viscosity were measured by measuring the effux time of test fuels for given tank. The procedure of kinematic viscosity was listed in APPENDIX C.

$$Kinematic\ Viscocity\ \left(\frac{mm^2}{s}\right) = effux\ time \times constant\ viscometer \qquad 3.1$$

$$Viscocity\ (mPa.s) = kinematic\ viscosity\ \left(\frac{mm^2}{s}\right) \times Density\ \left(\frac{g}{ml}\right) \qquad 3.2$$

where constant viscometer for 100 ml = 0.015120 (mm^2/s^2)

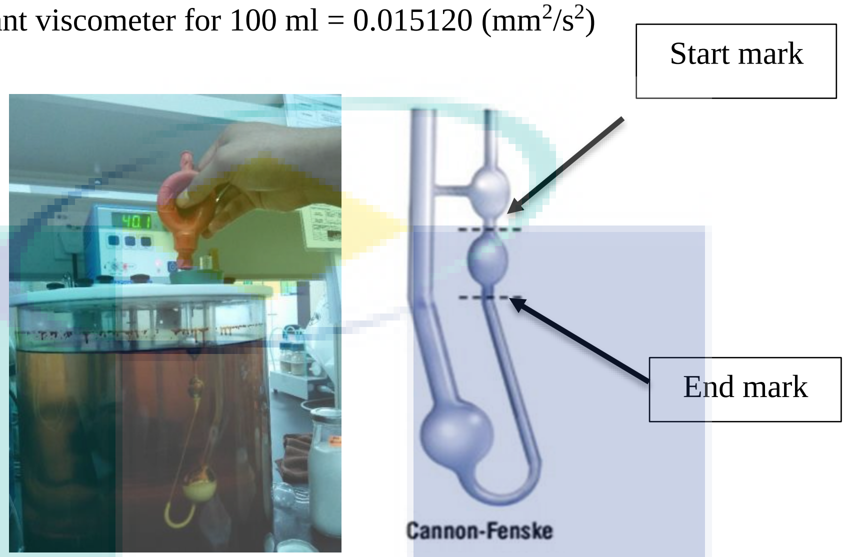

Figure 3.3 Procedure of sucked test fuels into start mark.

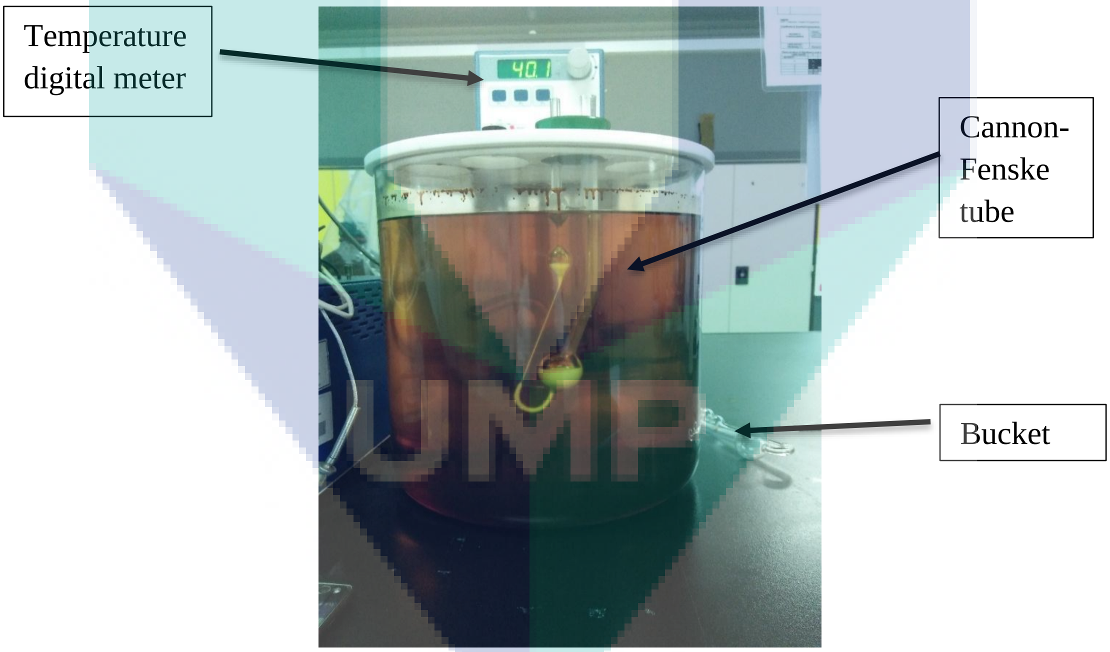

Figure 3.4 Time consuming for test fuels move from start to end mark observed.

The capillary viscometer style Cannon-Fenske tube used inside the kinematic viscosity tester conforms to ASTM D445. The time consuming observed for test fuels was the effux time in seconds as in Figure 3.3 and 3.4. Precaution for this testing, all the apparatus must be rinse with ethanol. Finally, the kinematic viscosity and viscosity obtained using the equation 3.1 and 3.2.

3.3.4 Calorific value measurement

Calorific value measurement was determined by using an oxygen bomb calorimeter of model Parr 6772 in Figure 3.5. Details procedure was listed in APPENDIX D. The procedure of finding calorific value was found the heat capacity or energy content of a material. The procedure for calorific value was conducted by following steps and meet standard ASTM D967.

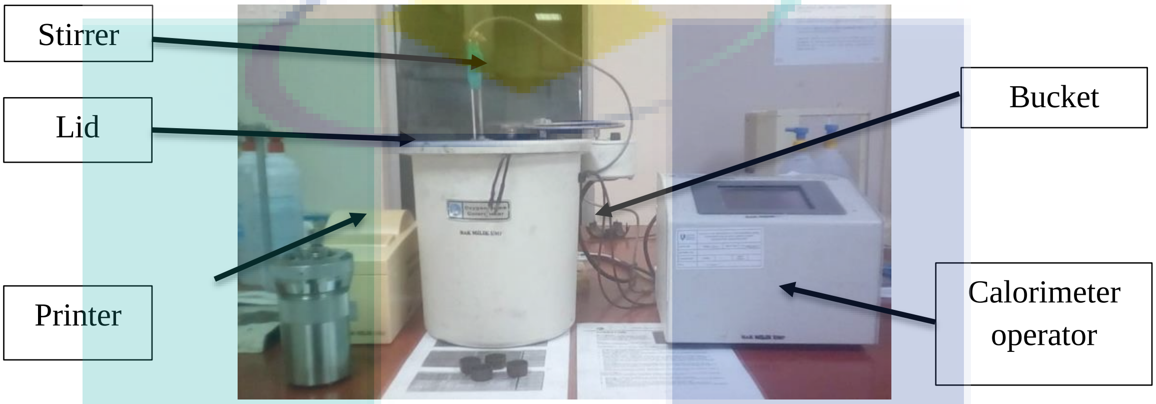

Figure 3.5 Bomb Calorimeter Parr 6772

3.4 Engine test bed and apparatus

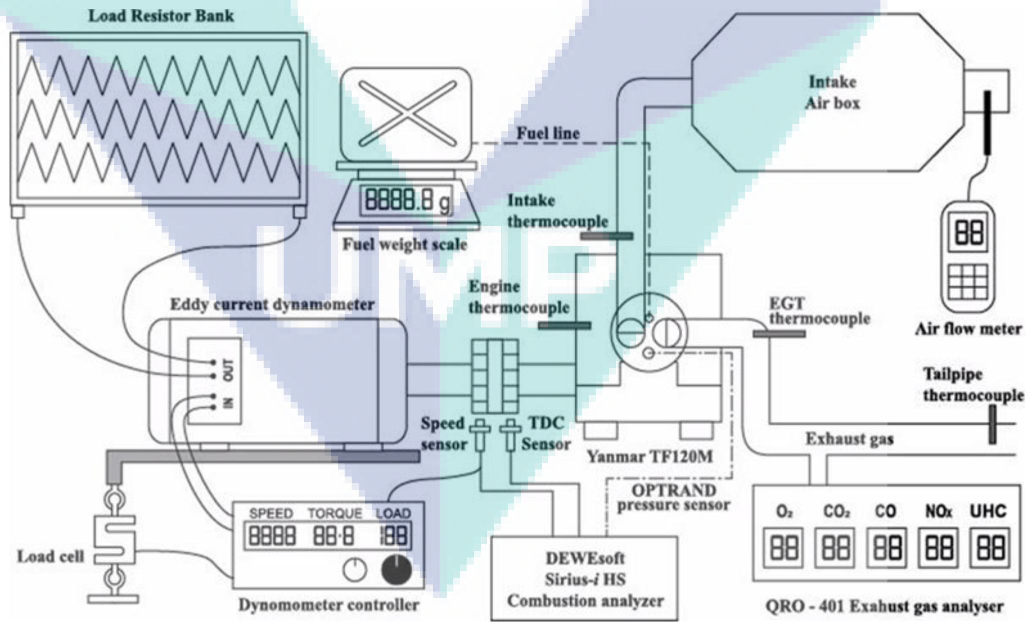

Figure 3.6 Schematic diagram of experimental setup for diesel engine.

In this work, a diesel engine model YANMAR TF120M research was setup in the Power Engine Laboratory of Universiti Malaysia Pahang (UMP). This research diesel engine consists of a single cylinder, four-stroke, water cooled system and natural aspirated direct injection (DI) in diesel engine. The engine fuel injection is at 17°CA before top dead center (BTDC). The test engine does not undergoes any alterations, no facilities of units in the test engine that could enable to enhance the fuel spray/ atomization characteristics or any modification on the engine to reduce the emissions (such as common rail injection unit, exhaust gas recirculation or diesel oxidation catalyst). The specification details of the engine are listed in Table 3.3.

The engine fuel system consists of 1 tank which has only 1 valve for DF and test fuel blends to flow into the engine during experiment. An eddy current dynamometer was connected to the engine to control the engine speeds and engine loads. In addition, the engine was employed with a data acquisition unit (DAQ), DEWESOFT SIRIUS-i. The combustion analyser, SIRIUS-i was installed at the combustion chamber to provide the engine combustion characteristics data. A OPTRAND pressure sensor with specification of AutoPSi-TC ±1% combustion and frequency response of 1 kHz-20 kHz, was connected to a charge amplifier used to obtain in-cylinder pressure. A magnetic-type crank angle sensor was mounted parallel to the trigger wheel on the engine. The timing in crank angle degree (CAD) during the combustion will be recorded by a crank angle decoder. The clearance between the crank angle sensor tip and trigger wheel was calibrated (maximum 3mm or less) and was adapted in order to increase its effectiveness at the piston top dead center (TDC).

The combustion characteristics were collected using DEWESOFT-X2 software. The performance data were collected and monitored from a Dynomax 2000 data-acquisition system. All of the parameter was calculated by using the equation that specific in Chapter 2. A QRO-401 exhaust gas analyzer was used to determine the exhaust gas parameters (CO, CO_2, O_2, HC, and NO_x emissions). The schematic of the test engine with the necessary accessory is drawn in Figure 3.6. The parameters measured and analyzed in this experiment were combustion characteristics (In-cylinder pressure and HRR), performance (BSFC, BTE and EGT) and exhaust emissions (CO, CO_2, HC, EGO, and NO_X) of the diesel engine which run at different engine loads (0%, 25%, 50%, 75%, and 100%) at the constant speed of 1800rpm. Each experiment for PE5, PE10, PE20, HE5, HE10 and HE20 was run according to load for 3 times. Complete data of experiment was collected in order to obtain the best results to achieve the objectives of study.

Table 3.3 Details of engine specifications

	Specification
Engine type	YANMAR TF120M
Number of cylinder	1
Bore x stroke	92 x 96 mm
Displacement	0.638 L
Compression ratio	17.7
Injection timing	17^o BTDC
Continuous output	10.5 HP at 2400 rpm
Rated output	12 HP at 2400 rpm
Cooling system	Water cooled

3.5 Test method

The test engine was operated in full throttle opening under constant engine speeds 1800rpm. The engine works on four-stroke cycle and operates with different engine loads (0%, 25%, 50%, 75%, and 100%). Before the start of the experiment, the engine was operated with DF for a couple of minutes at 1200 rpm to warm up the engine. Then, tested fuels (DF, PE5, PE10, PE20, HE5, HE10 and HE20) were run to flow in setup valve for 5 minute before collecting the test data to attain engine stability. Data was recorded by the installed DEWESoft X2 and Dynomax 2000 data-acquisition system (DAQ) after achieving the steady-state condition. All data was recorded after the engine stabilized at the setup operating condition for each of the tested fuels. After a completed the test for each test fuel, the engine was run for a few minutes to ensure that the remaining fuel in the fuel system was completed empty.

All the data was obtained and processed using the data acquisition unit (DAQ) by DEWESoft X2 and Dynomax 2000 data-engineering software. The QRO-401 exhaust gas analyzer was used for analyzing the exhaust emissions (CO, CO_2, EGO, NO_x and HC). The parameters measured and analyzed in this experiment were recorded and discussed.

3.6 Uncertainty analysis

An uncertainty analysis is use to achieved the accurate data of the experiments as error and uncertainties in the experiment from the instruments can affect the results. The list of instrument used for measuring various parameters and the percentage of uncertainties are presented in Table 3.4. The uncertainty for BSFC, BTE and brake power were calculated by using dynamometer and gas emissions collected using QRO-401.

Table 3.4 List of instrument and the percentage of uncertainty

`	Measurement	± Uncertainties	Unit
Gas Analyser	HC	1.9	ppm
QRO-401 (5 gases)	CO	0.01	vol %
	CO_2	0.2	vol %
	O_2	0.15	vol %
	NO_X	19.7	ppm
Dynamometer	Brake Power	0.18	kW
Calculated	BSFC	53.17	g/kW.h
Calculated	BTE	7.53	%

3.7 Experimental design and statistical analysis.

In experimental design, a standard response surface methodology (RSM) modeling was applied to study the modelling and analysis of the response variables at varies of engine loads in order to obtain the characteristic of engine working on fuel blends. By using Design Expert 7.0, the design of experiments was provided not only the individual effect of BSFC and BTE with engine loads but also their interactions with the minimum number of experiments for achieving the optimum conditions. In addition, RSM included both the mathematical and statistical techniques to describe the influence of interactions of parameters on the response when they are varied simultaneously. The RSM was designed for two factors-four levels historical data. The RSM for 1PN-diesel fuel blends and 2-EH-diesel fuel blends were designed separately. The independent variables, engine load (A) and percentage of alcohol in fuel mixture (B) were taken as the input parameters. The engine loads (denoted as load) were varied at 5 levels from 0% to 100% in steps of 25%. The percentage of alcohol in fuel mixture (denoted as fuel) was varied at 4 levels, (DF, PE5, PE10, PE20) and (DF, HE5, HE10, HE20).

The response (Y) for BSFC and BTE was evaluated. The design matrix contained 20 experimental runs. The experimental readings were fitted to second order polynomial equation by the design expert software. A multiple regression analysis was carried out to obtain coefficients and equations which use to predict the responses. Using the statically significant model, the correlation between parameters and responses were obtained. The optimum values of the input parameters were obtained by using desirability approach of the designed RSM.

3.8 Flowchart

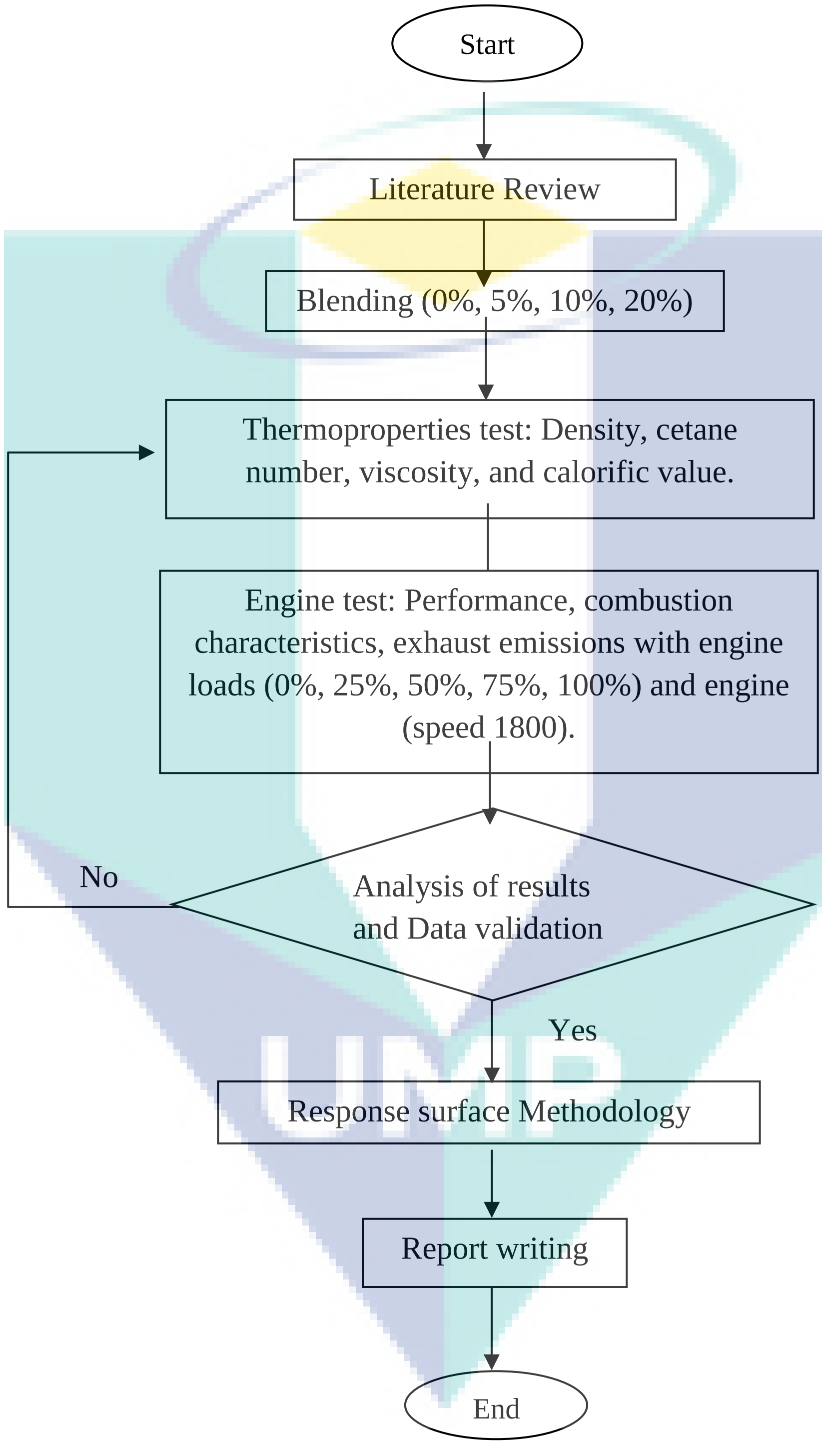

Figure 3.7 Flowchart 1.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Introduction

The purpose of this chapter is to verify that the experimental process of the methodology or the flow work come out with the results, analysis and discussion. The result included thermophysical properties test of fuel blends at different ratio, and engine test for combustion characteristic, engine performance and exhaust emission. All data of the experiment were recorded and analyzed accordingly.

4.2 Materials

The long-chain alcohols, 1-pentanol (1-PN) and 2-ethylhexanol (2-EH) are chosen due to their long carbon chain compared to short-chain alcohol. Due to that, 1-PN and 2-EH have more stability and miscibility with diesel fuels. Besides that, the short-chain alcohol-diesel fuel blends with diesel fuel needed additional additive, such as cetane improver make fuel blends became complicated and their stability worsens (Ciniviz, Örs et al., 2017; Venu & Madhavan, 2017). However, the long-chain alcohols blend with diesel fuel has a high cetane number, high kinematic viscosity, high calorific value and does not need additional additive to improve the combustion process (De Poures, Sathiyagnanam et al., 2017). Another reason to choose 1-PN and 2-EH were because of higher cetane number and calorific value compared with short-chain alcohol, such as ethanol or long-chain alcohol, butanol. The higher calorific value of the fuel blends also highly influenced the ignition delay of the fuel blend during the combustion process by shorter ignition delay (Imdadul, Masjuki et al., 2016). In addition, it is because1-PN and 2-EH are easily purchased and safe to use. As stated, in this experiment the long-chain alcohols which are 1-PN and 2-EH was the chosen alcohol to mix with DF and the application of 1-PN and 2-EH was studied at the end of experiment.

4.3 Thermophysical properties test results

The long-chain alcohols, 1-PN and 2-EH blended with DF for a total 2L at 5%, 10%, 20% using ultrasonic processor at constant 2 minutes. The fuel blends were prepared using amplitude of 40% and cycle at 0.5. Observation of long chain alcohol-diesel blend PE5, PE10, PE20, HE5, HE10, and HE20 show no separation even after 5 days as occurred in Figure 4.1 and 4.2.

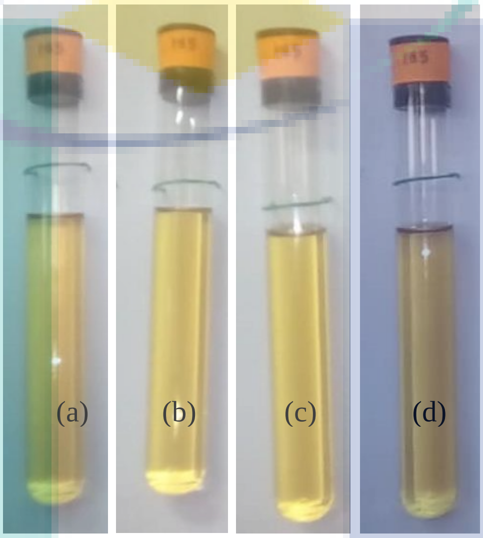

Figure 4.1 PE5 condition after blends (a), 1 hour (b), 1 day (c) and 5day (d).

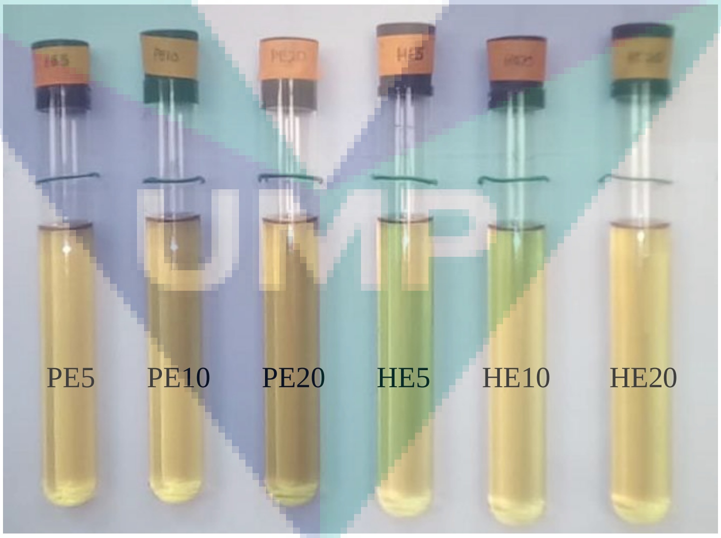

Figure 4.2 PE5, PE10, PE20, HE5, HE10 and HE20 condition at 5 days.

Thermophysical properties test for DF, PE5, PE10, PE20, HE5, HE10 and HE20 were recorded in Table 4.1. Cetane number calculated by using formula. The fuel density was measured using analytical balancer and the calorific value was measured using the bomb calorimeter. The thermal viscometer was used to measure viscosity, while the equipment models that was used for properties test is shown in Table 3.1.

Table 4.1 Thermophysical properties of the fuel blends.

	DF	PE5	PE10	PE20	HE5	HE10	HE20
Density (kg/m^3)	837.1	831.0	825.0	822.0	834.8	829.8	824.1
Cetane number	52.0	50.4	48.8	45.6	50.56	49.12	46.24
Kinematics viscosity (mm^2/s)	4.300	3.739	3.683	3.193	3.837	3.801	3.747
Viscosity (mPa·s)	3.60	3.10	3.04	2.62	3.20	3.15	3.09
Calorific value (MJ/Kg)	48.29	46.76	46.49	44.37	47.90	46.78	46.09

Based on Table 4.1, the overall thermophysical properties values of fuel blends were lower than DF. From Figure 4.3, PE5, PE10, PE20, HE5, HE10 and HE20 all have lower density compared to DF. In Figure 4.4, the cetane number for fuel blends lower than DF respectively. The kinematic viscosity and viscosity also decrease with the addition of 1-PN and 2-EH. Figure 4.5 showed viscosity for fuel blends decreased than DF. The results in Figure 4.6 showed that calorific values for fuel blends were lower than DF.

It was also noted that 1-PN has a lower value of density and calorific value compared to 2-EH, because 2-EH has a long carbon-chain in its molecule structures. Thermophysical properties of fuel blends helped the combustion process to complete efficiently due to higher viscosity, calorific value and cetane number compared to short-chain alcohol. Overall results indicated that the long-chain alcohols provide more oxygen molecules and stability than DF as lower density and viscosity improved atomization in fuel blends (Campos-Fernández, Arnal et al., 2012). This property can present an advantage in terms of engine performance.

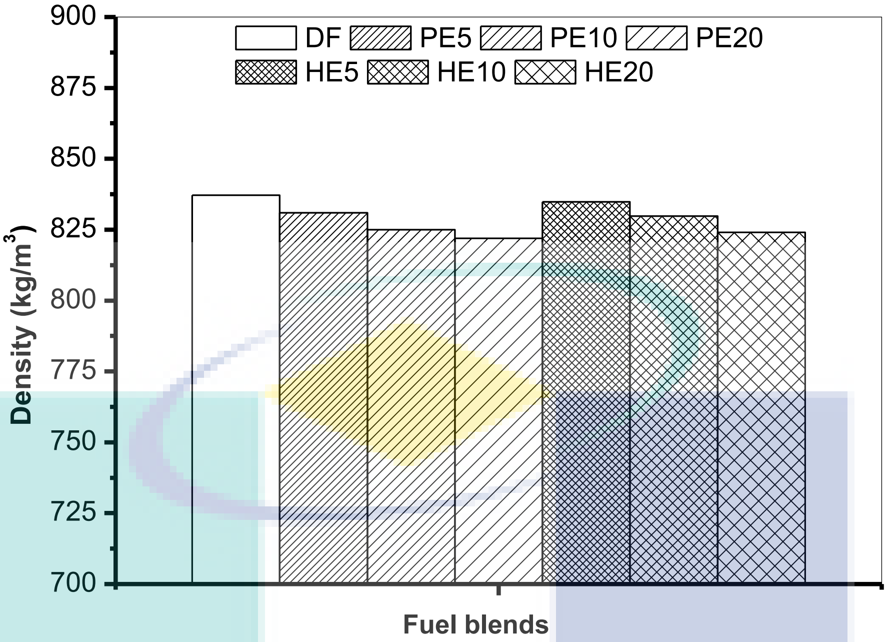

Figure 4.3 Density of DF and long-chain alcohol fuels blends.

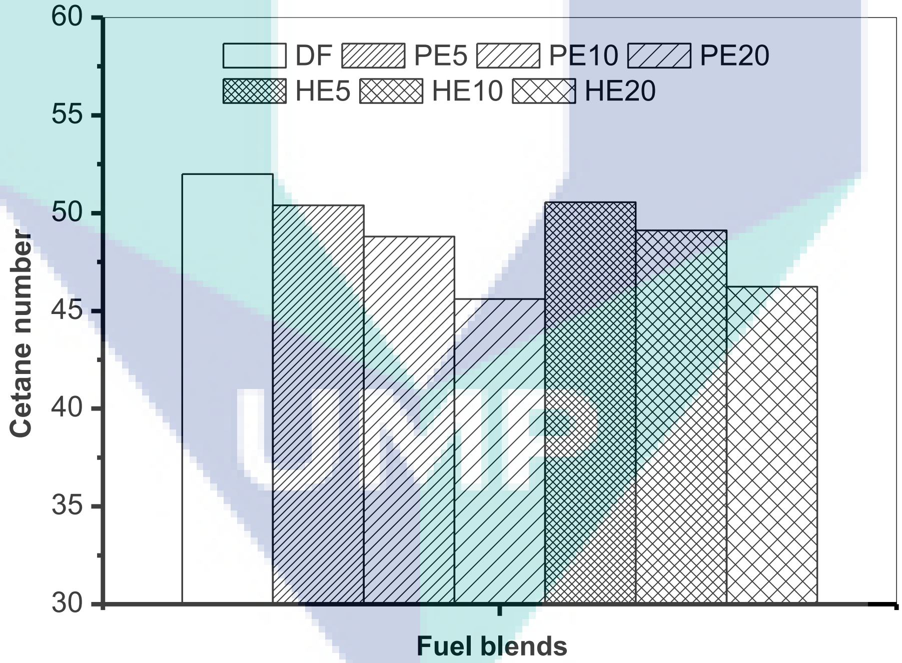

Figure 4.4 Cetane number of DF and long-chain alcohol fuels blends.

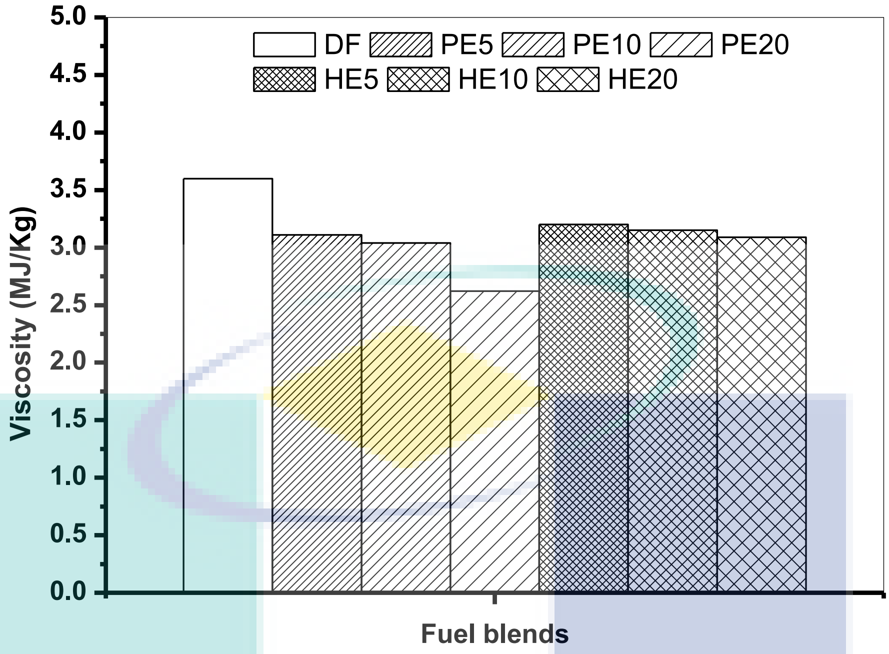

Figure 4.5 Viscosity of DF and long-chain alcohol fuels blends.

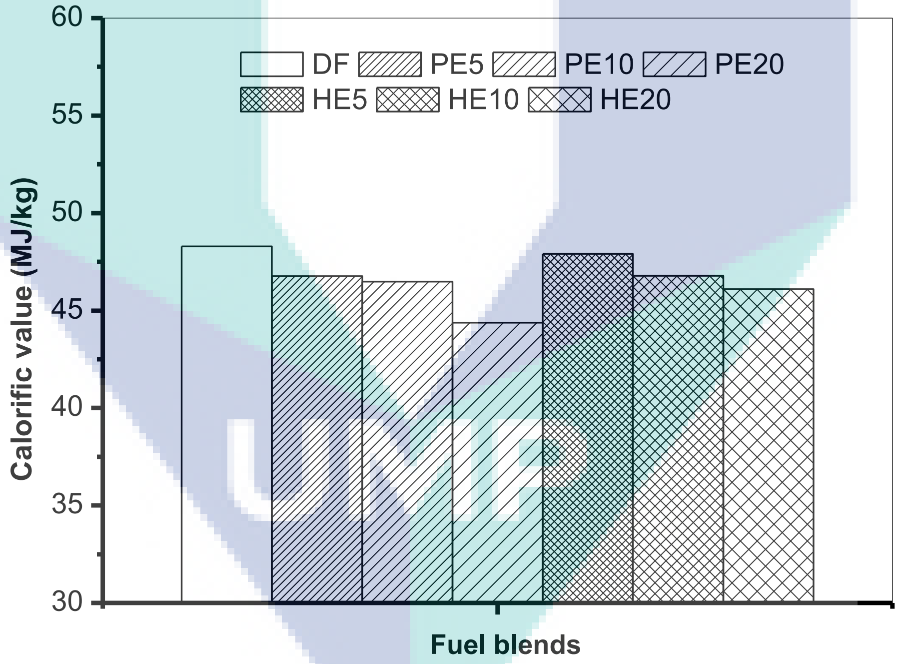

Figure 4.6 Calorific value of DF and long-chain alcohol fuels blends.

4.4 Combustion characteristics

4.4.1 In-cylinder pressure

The variation of in-cylinder pressure from engine loads 0% to 100% were demonstrated in Figure 4.7, 4.8, 4.9, 4.10 and 4.11. As shown in the Figure 4.12, the peak in-cylinder pressure increased as engine load increased, due to more fuel injected at higher engine load. At low engine load 0% to 50%, lower in-cylinder pressure of fuel blends due to lower in-cylinder temperature in low engine load as well as low density and viscosity of fuel blends. At full load, higher peak pressure observed in fuel blends than DF due to rapid combustion of the accumulated fuel as the temperature of the combustion increases at higher loads. At 100% engine load, it was seen that the peak pressure for PE5, PE10, PE20, HE5, HE10 and HE20 were higher than DF by 0.91%, 0.98%, 2.24%, 1.19%, 2.43%, and 4.77%, respectively. Higher peak pressure leads to longer premixed combustion.

Based on Table 2.1, the presence of 1-PN or 2-EH in the long-chain alcohol-diesel blends increased the essential oxygen that resulted for a stronger premixed combustion phase and higher peak pressure. Moreover, the lower viscosity and higher volatility in long-chain alcohol has increased fuel-air mixture regions during the ignition delay period which is important for complete combustion. Among fuel blends, HE20 has the highest peak pressure due to higher cetane number of 2-EH and lower heat of vaporization at 20% content of 2-EH (Prbakaran & Viswanathan, 2018). The trend found in Figure 4.12, for in-cylinder peak pressure can be supported by the study of (Imdadul, Masjuki et al., 2016) who reported rise of pressure for the increase of pentanol proportion in the blends to 15% and 20% than neat calophyllum inophyllum biodiesel. Another research by (Lujaji, Kristóf et al., 2011) also found and supported higher peak pressure with addition of butanol. These authors reported higher pressure in mixed of 15% croton oil, 5% butanol and 80% DF, followed by the 10% croton oil, 10% butanol and 80% DF and DF fuel blend samples.

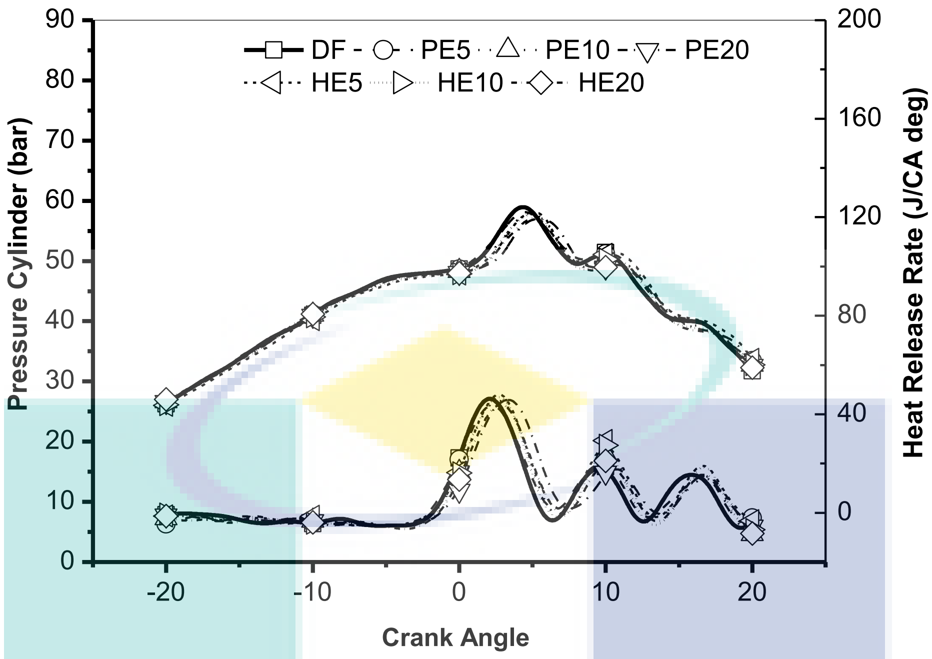

Figure 4.7 Variation of pressure and HRR for engine load 0%.

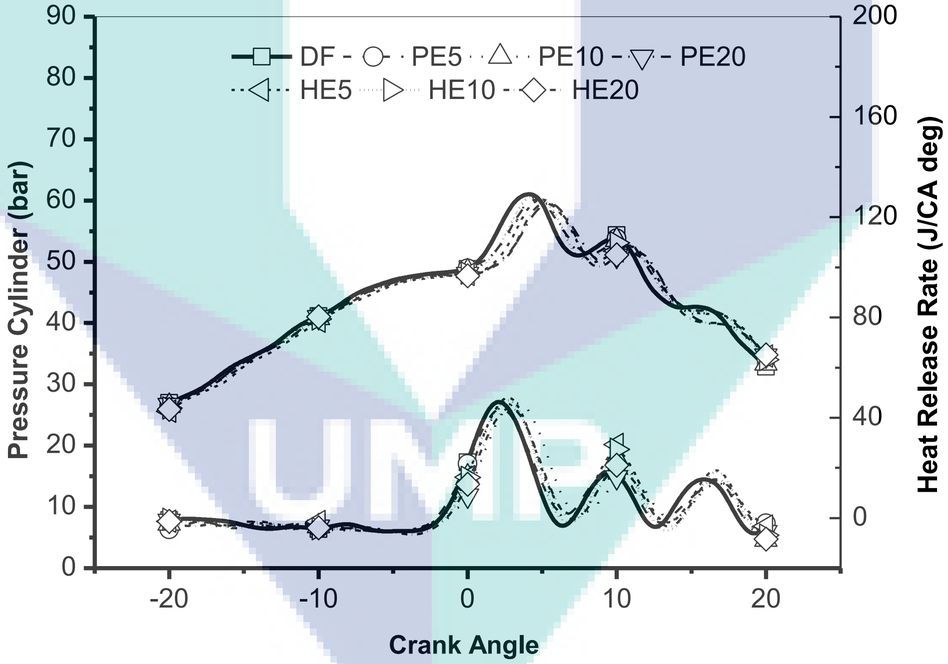

Figure 4.8 Variation of pressure and HRR for engine load 25%.

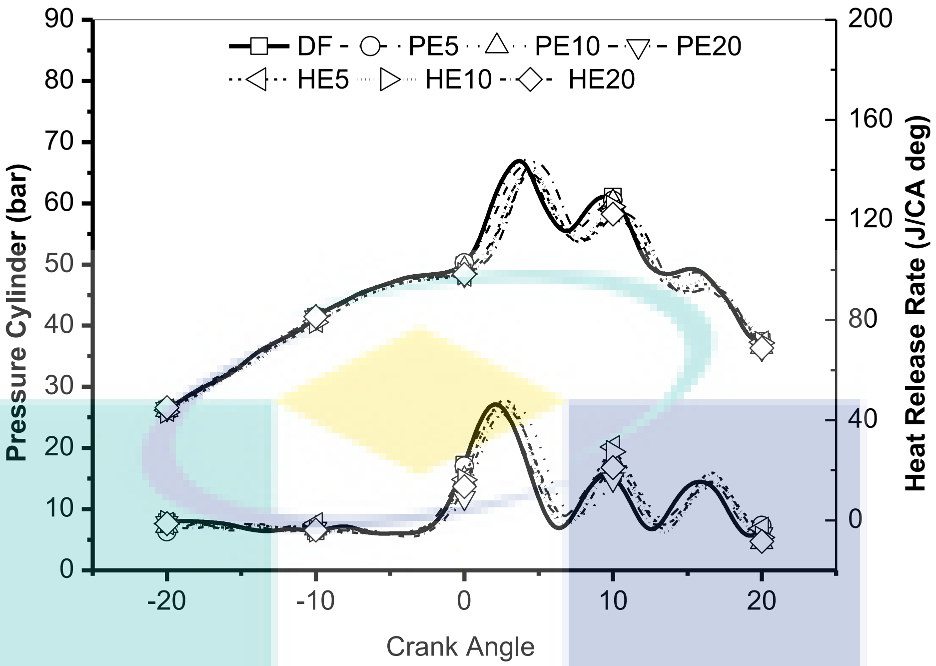

Figure 4.9 Variation of pressure and HRR for engine load 50%.

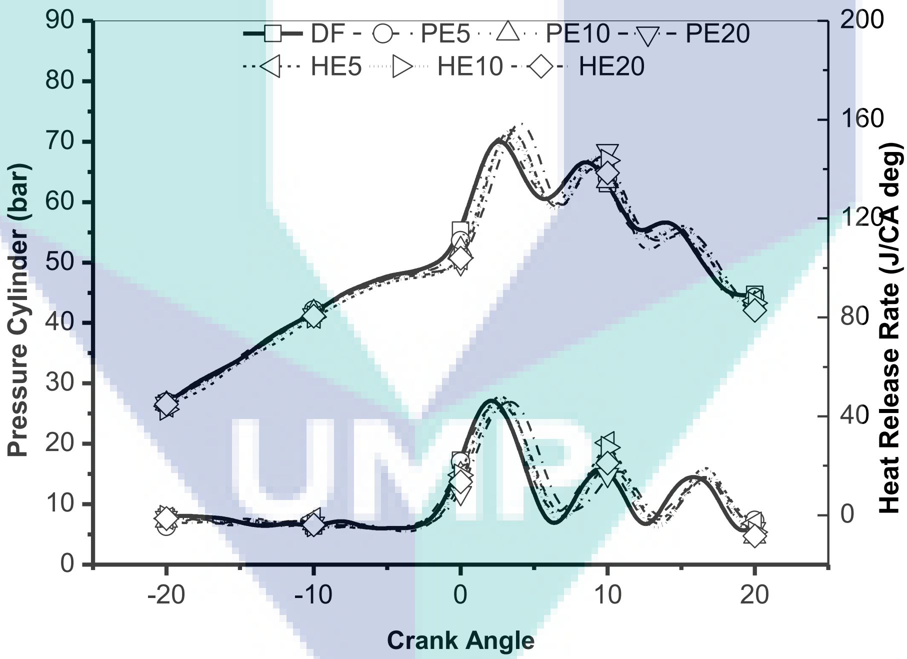

Figure 4.10 Variation of pressure and HRR for engine load 75%.

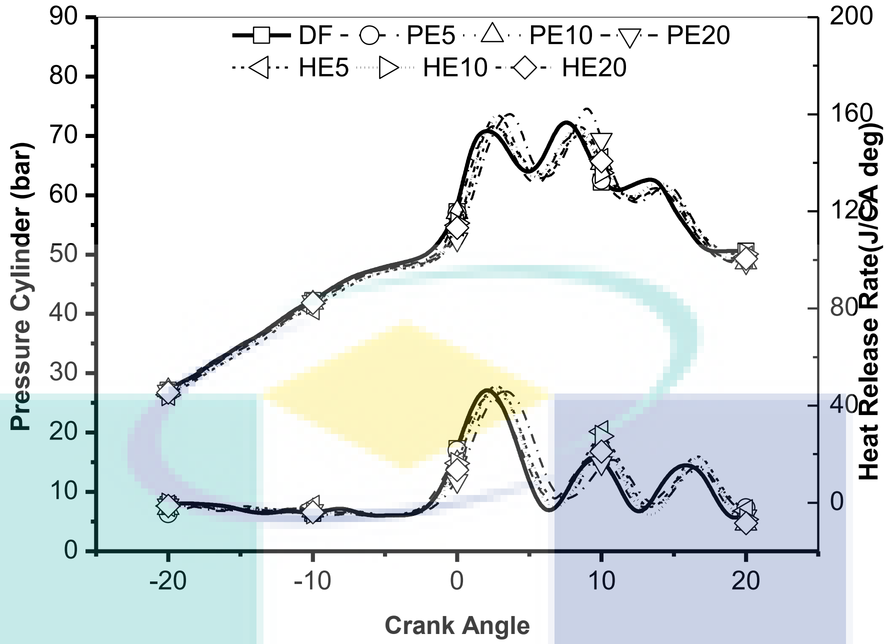

Figure 4.11 Variation of pressure and HRR for engine load 100%.

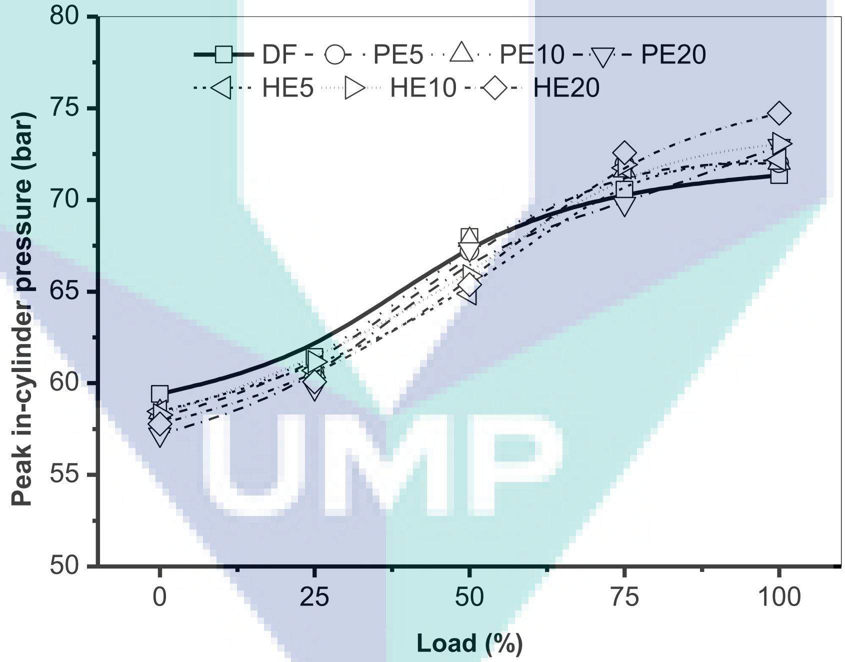

Figure 4.12 Variation of peak pressure for different engine load.

4.4.2 Heat release rate

In Figure 4.7 to 4.11, the heat release rate (HRR) for all test fuels at 0% to 100% engine loads conditions were shown. It is observed that HRR increased as engine loads increased. In addition, the plotted data in Figure 4.13 showed that fuel blends with 2-EH has higher HRR than fuel blends with 1-PN at the same ratio. At 100% engine load, the HRR was increased by 4.21% for PE5, 6.33% for PE10, 9.97% for PE20, 5.64% for HE5, 11.91% for HE10 and 15.6% for HE20 compared to DF. The higher HRR found in fuel blends were found in fuels with higher cetane number and oxygen in the structures. From Table 2.1, the presence of oxygen molecules in the long-chain alcohol-diesel fuel blends higher than DF which increased HRR for the preparation of a larger fuel of rapid burning during longer ignition delay. (Sathiyamoorthi & Sankaranarayanan, 2017) supported this findings by stating the addition of ethanol into 25% neat lemongrass and 75% DF increased the HRR. In 2014, (Balamurugan & Nalini, 2014a) also reported increase of HRR with addition of n-propanol due to the increase of in-spray characteristic. The higher latent heat of vaporization of 1-PN and 2-EH caused for quenching effect and lower in-cylinder temperature, delayed maximum heat release rate.

The above figures show that the maximum HRR increased as the engine load increased for all fuel blends. The higher engine loads influence the rise for high temperature and high cylinder pressure for better fuel–air mixing, and higher flame velocity that had caused combustion to start slightly early for HRR at the premixed combustion period (Z.-H. Zhang & Balasubramanian, 2014).

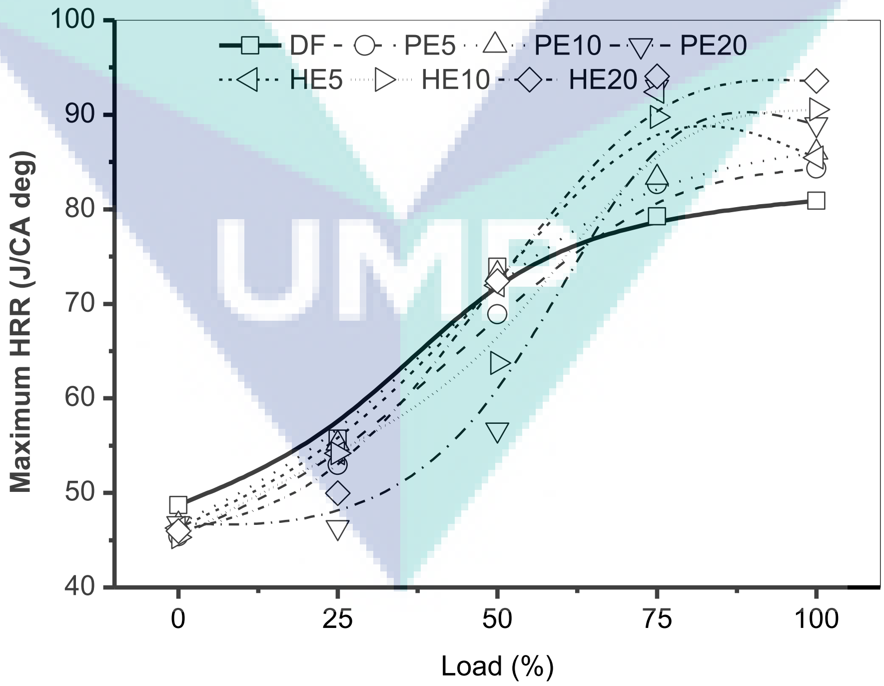

Figure 4.13 Variation of maximum HRR for different engine load.

4.5 Engine performance

4.5.1 Brake specific energy consumption

The brake specific energy consumption (BSFC) and engine loads are plotted in Figure 4.14. Referring to the figure, the BSFC decreased for all test fuels at 0% and up to 75% and slightly increased at 100%. At low load 0% to 50%, the lower in-cylinder gas temperature leading to incomplete and low efficiency of the combustion, resulting in higher BSFC (Agrawal, Sharma et al., 2015). In Figure 4.14, the decrease of BSFC found were 8.51%, 10.16%, 12.08%, 8.38%, 10.87%, and 10.99% for PE5, PE10, PE20, HE5, HE10 and HE20. According to (Z.-H. Zhang & Balasubramanian, 2014), lower density and decrease of calorific value in the fuel blends would injected more fuels into combustion chamber and made it easier for injection of fuel blends as the energy content decrease which increase BSFC. However, in this case higher O_2 in the fuel blends resulted for a more complete combustion and increase efficiency, which decrease BSFC. Besides that, lower density and viscosity of fuel blends also improved the atomization between DF and alcohols which also improved the combustion (Campos-Fernández, Arnal et al., 2012). Besides that, higher latent heat of vaporization of 1-PN and 2-EH have resulted less BSFC. Higher latent heat of vaporization caused cooling effect which reduce fuel intake in combustion chamber. Similar to study (Rajesh Kumar & Saravanan, 2016) where decrease of BSFC found in 40% iso-butanol and n-pentanol diesel fuel blends.

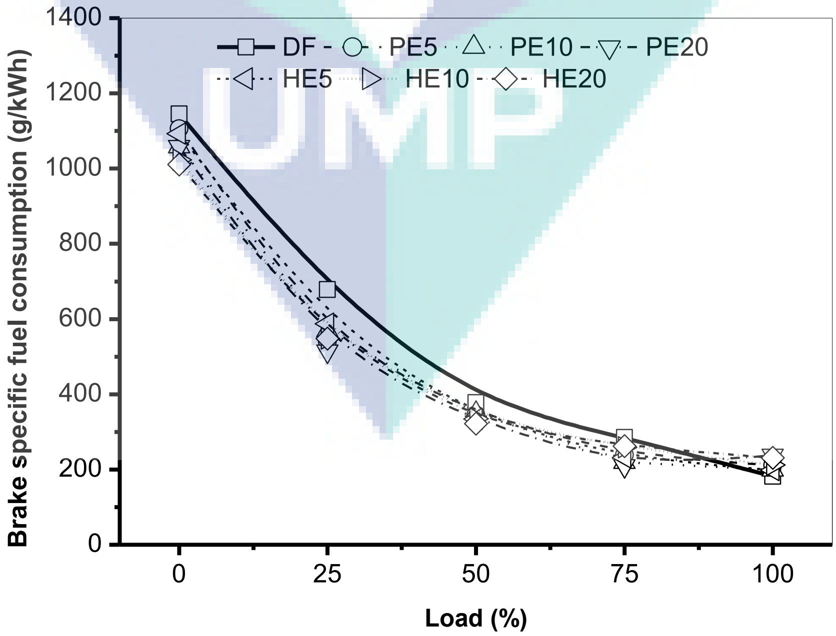

Figure 4.14 Variation of BSFC for different engine loads.

4.5.2 Brake thermal efficiency

Figure 4.15 depicts the brake thermal efficiency (BTE) versus engine loads for all DF and fuel blends: PE5, PE10, PE20, HE5, HE10, and HE20. Based on the graphs shown, the BTE for blended fuels increased gradually as engine loads increased. At engine loads 0% to 75%, BTE of 2-EH fuel blends at the highest, followed by 1-PN fuel blends and DF, before gradually decreases at 100% engine load condition. Clear increase of BTE can be seen at engine load 50%, where HE20 have the highest BTE. The BTE increased by 7.03%, 12.09%, 17.55%, 12.25%, 12.95%, and 19.67% for PE5, PE10, PE20, HE5, HE10, and HE20 respectively when compared with DF. Relevant to lower BSFC, higher BTE was also found in fuel blends compared to DF, due to increase of oxygen content in fuel blends as stated in Table 2.1, which contributed to higher BTE. The presence of oxygen molecules by addition of alcohols, improved combustion especially diffusion combustion and increased efficiency. This can be supported by (Z.-H. Zhang & Balasubramanian, 2014) who stated that increase BTE of 3.7%, 3% and 2.7% from low to high engine load were found in 15% n-butanol with 85% DF. The authors further found that addition of n-butanol increase oxygen in fuel blends which prolonger ignition delay. In another study (Doğan, 2011) found the significant increase of BTE as n-butanol added increased at 5% to 20% of DF. Authors suggested that another factor that increase BTE was lower cetane number of 1-PN and 2-EH causes of a longer ignition delayed, which led to having more fuel burnt during premixed mode that increase BTE. Despite to that, HE20 has highest BTE due to highest maximum HRR which reduce heat losses with lower in-cylinder temperature (Imdadul, Masjuki et al., 2016). Moreover, both references of (Doğan, 2011; Z.-H. Zhang & Balasubramanian, 2014) reported the same findings with increase of BTE.

However, at maximum engine loads, the efficiency of fuel blends decreasing indicating deterioration in engine performance at this engine load. As the amount of fresh air in the fuel blends is lower with the increase of temperature, which is significant to high engine load. At maximum engine load, fuel blends experienced slight reduced BTE, as the amount of fresh O_2 available for combustion gets decreased due to replacement of exhaust gas (Sundar & Saravanan, 2011). The same trend curve of BTE have been reported by (Zhu, Cheung et al., 2011) as decrease of BTE illustrated at engine load 70%.

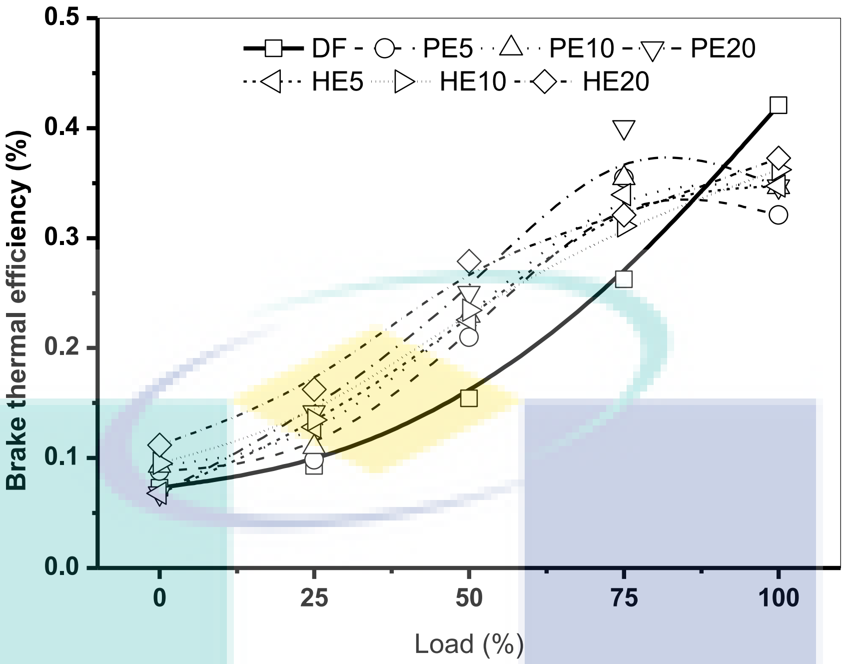

Figure 4.15 Variation of BTE for different engine loads.

4.5.3 Exhaust gas temperature

Figure 4.16 presents the exhaust gas temperature (EGT), respected in engine loads at constant 1800rpm. From the graph, EGT increases, as the engine load increases in all the tested fuels. Although calorific value of PE5, PE10, PE20, HE5, HE10 and HE20 was lower than DF, the higher content of oxygen had expanded oxygen-rich regions, which lead to higher EGT values as engine loads increases. According to (Yilmaz & Atmanli, 2017a), more fuels enters the combustion chamber as the engine load increases from 0% to 100%, due to presence of higher atmospheric O_2 content at higher loads, causing increase of in-cylinder temperature. Higher temperature and pressure in combustion chamber also increased EGT at full engine load condition.

At full engine load condition, the values of EGT found in DF, PE5, PE10, PE20, HE5, HE10 and HE20 were 390.35 °C, 394.20°C, 384.83°C, 381.20°C, 388.12°C, 386.57°C and 381.66°C. Referring to the figure, it was noted that, the EGT for all blended fuels was slightly lower than DF, except for PE5. The reason for lower EGT found in PE10 PE20, HE5, HE10 and HE20 was due to lower density and calorific value of fuel blends. This can be supported by (Anand, Sharma et al., 2011) who also finds biodiesel-methanol lowers EGT compared to neat biodiesel. Among the fuel blends the highest EGT found in PE10 and the lowest found in HE10.

Lower EGT found in PE10, PE20, HE5, HE10 and HE20 was opposite to the finding by (Atmanli, 2016a). The author claimed that higher exhaust gas temperatures were achieved with the additional of higher alcohols as the EGT of D40B40Pro20, D40B40nB20 and D40B40Pn20 increased 30.54%, 31.58% and 27.23%, respectively, as compared to D50B50. However, a slight increase of EGT was found in PE5 which was opposite to results found in other fuel blends. Despite having low calorific value and energy density, another reason that influence EGT were the increase of oxygen content in PE5. 1-PN expanded the regions of oxygen-rich in the combustion chamber, which leads to regional temperature peaks and higher exhaust gas temperatures. (Yilmaz & Atmanli, 2017a) supported this report by also reporting increase of EGT of 5% 1-PN added into diesel-biodiesel-1-pentanol blends.

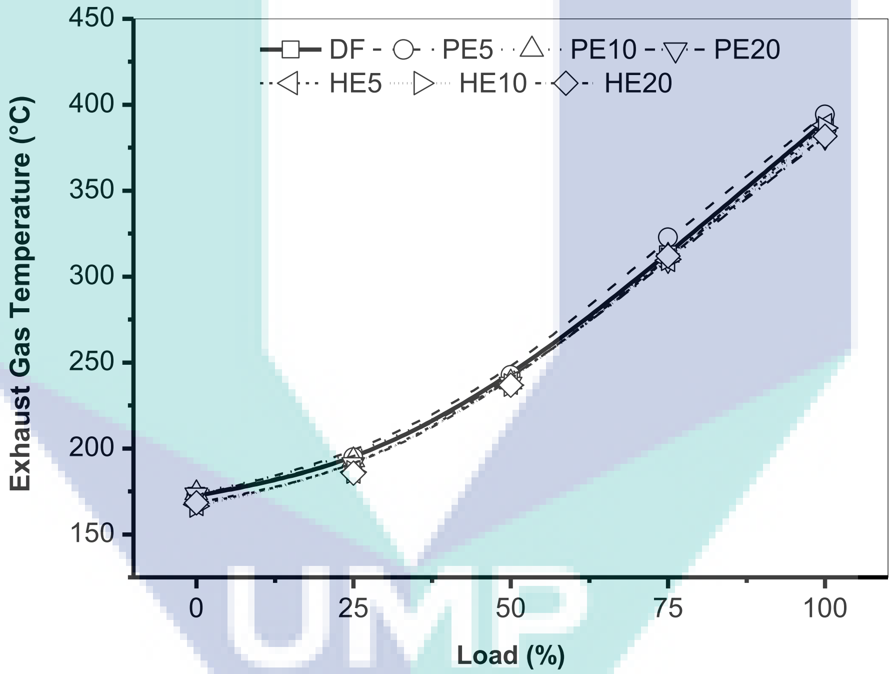

Figure 4.16 Variation of EGT for different engine loads.

4.6 Exhaust gas emissions measurements

4.6.1 Carbon monoxide emission

Figure 4.17 shows relation of carbon monoxide (CO) emission which varies to different engine loads at constant 1800 rpm. Generally, CO emission increases as engine loads increase. At maximum engine load, CO emission rapidly increases because engine experiences high volume of fuel blends, supplying fuels to the higher in-cylinder pressure and temperature which also significant to the increase of the engine loads. Refer to the figure below, at engine load 100% PE5 and HE5 there is a similar CO emission of DF, because of the fuel blends had sufficient O_2 to react during combustion. Even with increase of O_2 with rise of alcohols, the formation of carbon oxide (CO) happens because of incomplete combustion occurrence and controlled primarily by the fuel/air equivalence ratio which happens in high concentration of additive.

Compared to DF, the total average CO emission for PE5, PE10, PE20, HE5, HE10 and HE20 were higher by 12.64%, 23.51%, 35.28%, 2.94%, 14.70%, and 32.35%. From the results, the highest CO emission among fuel blends found in PE20. Meanwhile, the lowest CO emission among fuel blends found in PE5. The reason for higher CO emission found in fuel blends compared to DF was due to lower cetane number of fuels blends increased the duration of the premixed combustion stage which caused timing problems in terms of the combustion and expansion stages. This results in less oxidation of carbon and oxygen that rise formation of CO emission. The same finding was been reported by (Tutak, Lukács et al., 2015) where the highest CO emission was observed at 20% of methanol with DF. These authors stated that CO was a product of incomplete combustion, and usually occurred at high O_2 deficiency in cylinder. Moreover, the whole results can be explained by the cooling effect of long-chain alcohols due higher latent heat of vaporization caused incomplete oxidation of the CO to CO_2 that can happens during the expansion stroke. Furthermore, with increasing engine load, temperature of charge increases as well, leading to formation of CO emission (Tutak, Lukács et al., 2015). Also, at partial engine loads, the emission of CO increases linearly for DF and fuel blends.

The findings was different to the report by (Mahalingam, Devarajan et al., 2017) and (Ramakrishnan, Kasimani et al., 2018) as both authors claimed decrease of CO emissions in alcohols-diesel fuels blends which indicated more complete combustion. According to the authors, the rise of n-pentanol and n-octanol increase the availability of O_2 in the modified fuel blends. Moreover, (Ramakrishnan, Kasimani et al., 2018) found that the lower density of n-pentanol caused less formation of CO emission. Besides, the biodiesel has low carbon to hydrogen ratio which leads to reduced CO emission than diesel fuel (Mahalingam, Devarajan et al., 2017; Ramakrishnan, Kasimani et al., 2018). Besides that, (Mahalingam, Devarajan et al., 2017) highlighted that presence of n-octanol increased the burning pace of n-octanol with mahua oil biodiesel.

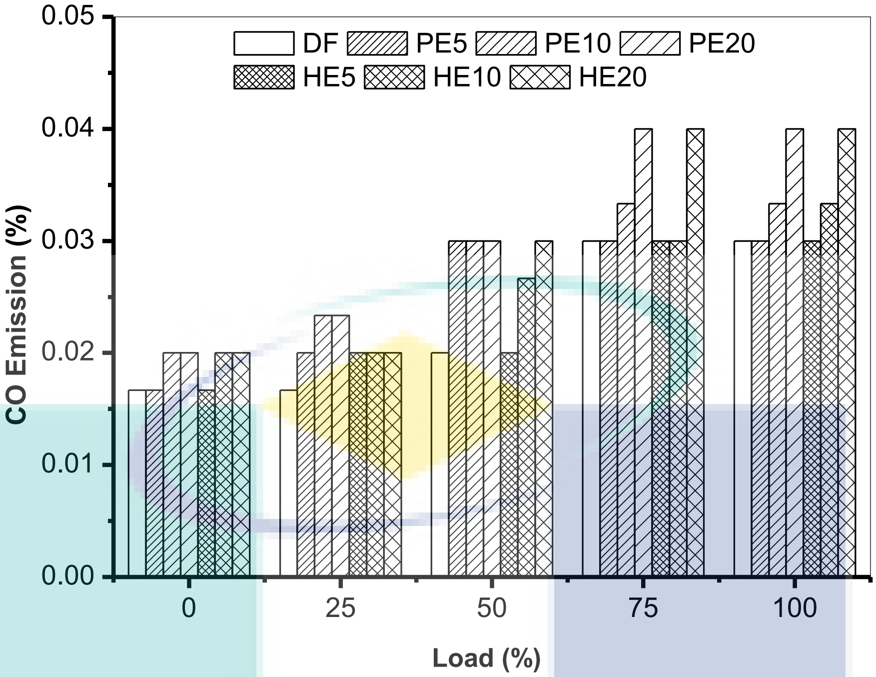

Figure 4.17 Variation of CO emissions for different engine loads.

4.6.2 Carbon dioxide emission

Figure 4.18 illustrated carbon dioxide (CO_2) emission versus engine loads at constant 1800 rpm. The formation of CO_2 happens where CO convert into CO_2 in the presence of adequate O_2 using hydroxyl radical OH as the oxidizing agents. Observation from Figure 4.6 shows that CO_2 emissions is the highest from DF, PE5, PE10, PE20, HE5, HE10 and HE20. It is also noted that emission of CO_2 is almost less for all long-chain alcohol fuel blends at all engine load ranges, compared to pure diesel and DF. Among fuels blends, the lowest total average of CO_2 emission was found in HE20 with 3.37% and highest CO_2 emission was found in PE5 at 3.91%. (Ileri, Atmanli et al., 2016) stated that increasing oxygen and hydrogen molecules in fuel structures decrease CO_2 emissions. Based on the combustion reaction in the page 10, number of atmospheric O_2 needed for DF was higher than fuel blends. In 2018, (Nanthagopal, Patel et al., 2018) also reported that n-butanol and pentanol give decreasing effect of CO_2 formation to Calophyllum Inophyllum (CIME) biodiesel. Since the CO_2 emission highly influences the greenhouse effect and global warning, it is necessary to ensure the CO_2 emission decreased from the engine emissions and achieved objective. The increase of CO emissions was significant with decrease of CO_2 emissions. However, despite 2-EH have lower oxygen, it is observed that fuel blends with 1-PN have higher CO_2 than fuel blends with 2-EH due to the insufficient of O_2 to react with unburnt carbon in the fuel blends structure, which increase CO_2 and found in PE5, PE10 and PE20.

However, (Ramakrishnan, Kasimani et al., 2018) opposed the finding as the authors found higher CO_2 emission in n-pentanol. The authors stated that the CO and CO2 emissions show trade off properties with each other as in adequate O_2, the CO oxidized into CO_2 as a result of complete combustion. The CI20 and n-pentanol fuel blends revealed an increased CO2 emission in an average of 5.3 and 7% than diesel fuel blends, respectively.

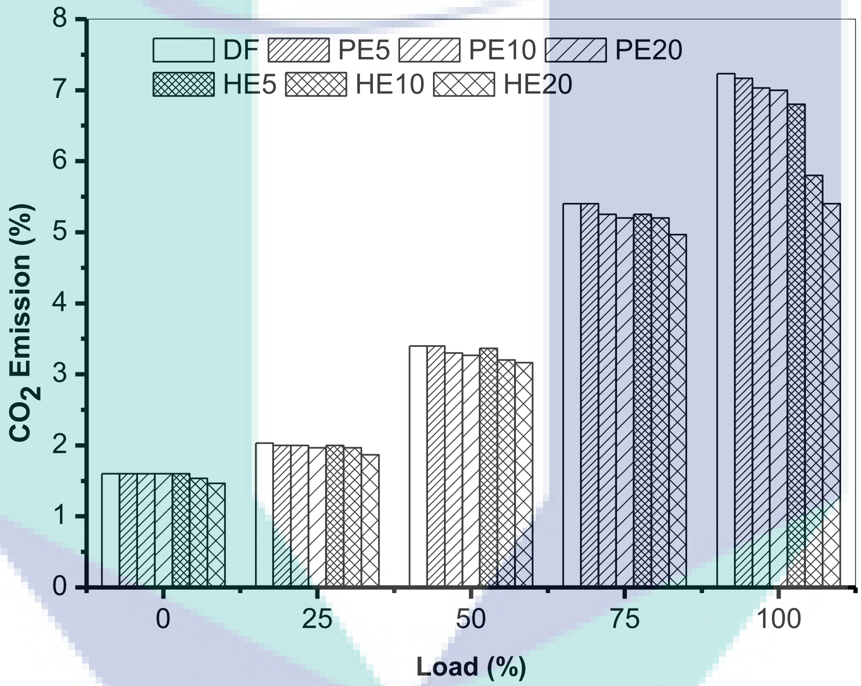

Figure 4.18 Variation of CO_2 emission for different engine loads.

4.6.3 Hydrocarbon emission

The variation of hydrocarbon (HC) emission and engine loads is demonstrated in Figure 4.19. The HC emission of DF and all three fuels blends gradually decrease with increasing engine loads. From the figure, it can be clearly seen that HC emission sharply increase as the percentage of long-chain alcohol, for 1-PN and 2-EH with diesel increase from 5%, 10% and 20% in the fuel blends. However, HE5 have decreases HC emission at engine load of 50% and above. The discussion starts with PE5, PE10, PE20, HE10 and HE20 with higher HC emission at all engine loads. This situation is expected as long-chain alcohol display lower cetane number, calorific value and higher heat of vaporization. Lower cetane number of 1-PN and 2-EH usually prolong ignition delay and allowing more time for fuel blends to evaporates. The higher heat of the evaporation causes slower evaporating which increase HC emission. The high evaporation of temperature of long chain alcohol has resulted in cooling effect in combustion, which leads to the lower temperature inside the cylinder, resulting in incomplete combustion and high HC emissions (Emiroğlu & Şen, 2018b). A research by (Doğan, 2011) reported increase HC emission from 5% n-butanol to 20% butanol. In another research, (Emiroğlu & Şen, 2018b) high formation of HC in presence of 10% butanol, 10% ethanol, or 10% methanol which supported the findings.

However, engine load 50%, 75% and 100% showed reduction of HC emissions by 46.65%, 45.45% and 71.0% which is measured than DF. The total average HC emission for HE5 was lower by 18.06% than DF. At higher engine load, lower HC emissions as HE5 possess good atomization due to fuel-air ratio in the fuel blend, thus improving combustion and reducing formation of HC emission. Higher O_2 in HE5 due to the addition of oxygenated fuel, 2-EH as the additive. A report by (Alptekin, 2017), reported usage of alcohol and ethanol reducing effect on HC emissions of 15% ethanol with 85% canola-safflower biodiesel, showing lower HC emission by 10.2%. Moreover, the reduce of HC can be at higher engine load which may be due to the higher temperature found at the engine loads for better combustion and lower HC measured.

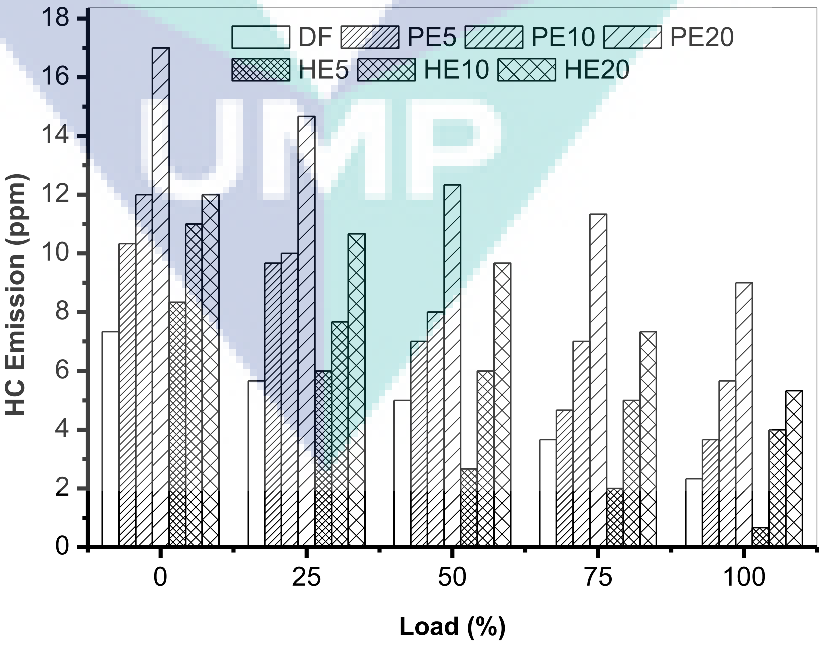

Figure 4.19 Variation of HC emission for different engine loads.

4.6.4 Exhaust Gas Oxygen emission

The exhaust gas oxygen (EGO) of the blend fuels in the experiment was plotted in Figure 4.20. EGO level indicates the amount of oxidation achieved by the test fuels during combustion. EGO is provided by the EGO sensor which gives details about the O_2 levels from exhaust and it is very useful for analyzing the transition from rich to lean mixture range. .EGO levels and was measured to study the amount of oxidation rate during combustion of DF and blended fuels, which analyze the transition from rich to lean mixtures range (Venu & Madhavan, 2017). A rich mixture of more burnt O_2 and lean mixtures of burnt less O_2 make the more O_2 escape, known as ‘‘un-combusted” for the EGO level to increase. In simple, EGO levels are inversely proportional to combustion efficiency. Figure 4.20 indicates the variation of EGO for all the test fuels with respect to engine load. As the engine load increases EGO reduces due to higher in-cylinder temperatures.

As shown in Figure 4.20, the EGO levels decrease with increase of engine loads. This trend happens as high engine loads have higher temperature of in-cylinder that decrease the O_2 in exhaust gas analyzer which indicated more O_2 burnt at higher engine loads. The same trend was found by (Venu & Madhavan, 2017) who used diethyl ether (DEE) as an addition in ethanol-biodiesel-diesel (EBD), where 0.62% was the lowest EGO found at engine load of 100%. Besides that, fuels blends with addition of 2-EH have higher O_2 emission than fuel blends with 1-PN and DF. The total average increase of EGO was found to be 0.058%, for PE5, 0.653% for PE10, 0.698% for PE20, 0.288% for HE5, 1.363% for HE10 and 1.824% for HE20. Findings showed that addition of long-chain alcohol, 1-PN and 2-EH concentration was affected by the volatile and latent heat properties of the mixture.

From Table 2.1, oxygen content in the 1-PN and 2-EH was higher than DF which explained the higher EGO levels found in fuel blends in Figure 4.20. Furthermore, the lower cetane number of alcohol has prolong ignition delay, allowing more time for fuel to evaporate and increase O_2 to escapes.(Srinivasan & Saravanan, 2010) supported the results as higher EGO was found with addition of ethanol into gasoline and increase the oxygen content in fuel blends. In summary, higher oxygen content with lower in-cylinder temperature does increase O_2 escapes during combustion.

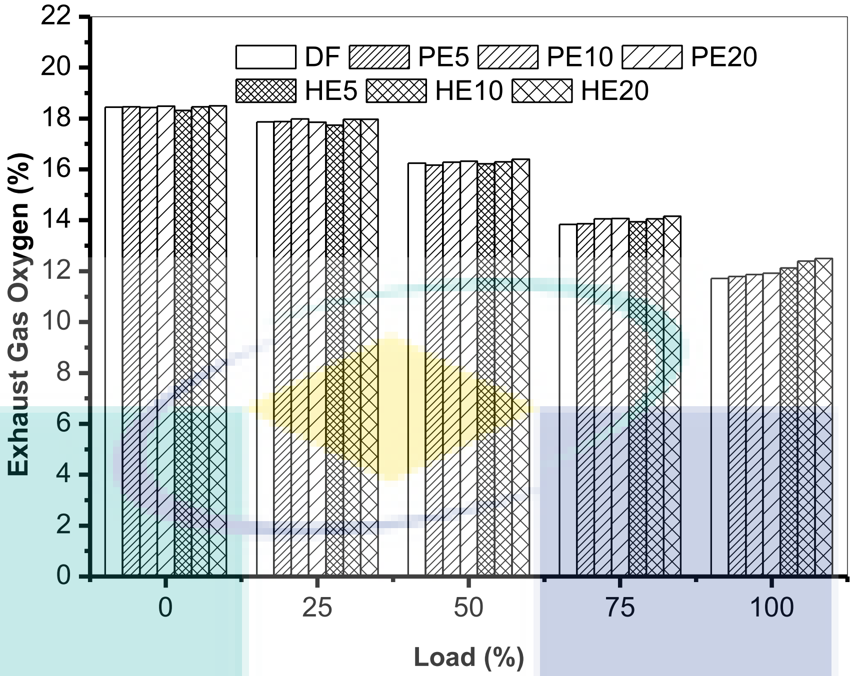

Figure 4.20 Variation of Exhaust Gas Oxygen (EGO) different engine loads.

4.6.5 Nitrogen oxide emission

In Figure 4.21, nitrogen oxide (NO_x) emissions were provided at different engine loads. The Figure 4.21 showed that as the engine loads increased, the emissions of NO_x also increases. The discussion in Figure 4.21 regarding NO_x formation happen due to the higher combustion temperature, because higher engine loads was significant to higher temperature. Generally, lower NO_x was found in fuels blends than DF. At engine load of 100%, it was clearly observed that the NO_x emission for the PE5, PE10, PE20, HE5, HE10 and HE20 fuel blends are lower compared to DF. The total average of NO_x emission was found to be at 287.27 ppm, 286.10 ppm, 278.50 ppm, 278.40 ppm, 277.13 ppm, 259.2 ppm and 245.40 ppm for DF, PE5, PE10, PE20, HE5, HE10 and HE20 respectively. The addition of long-chain alcohol of 1-PN and 2-EH have high latent of evaporation than DF and gives a lower temperature effects that helps to lower the temperature of in-cylinder and lower the formation of NO_x emissions. The lowest NO_x was found in HE20 with 14.58% lower than DF. In addition, the fuels blends of 2-EH have lower NO_x emission than fuel blends of 1-PN. This is due to lower cetane numbers, viscosity, density and high volatility of pentanol that cause longer ignition delay and more fuel accumulation, increasing the amount of fuel in premixed combustion. In statement by (Yilmaz & Atmanli, 2017a), the post-combustion temperatures increases, leading to higher NO_x. This findings is supported by (Sharon, Ram et al., 2013) who also found less formation of NO_x with addition of butanol into palm oil and DF. Also, (Joy, Devarajan et al., 2018) reported that n-octanol at 10%, 20% and 30% have lower NO_x emissions compared to DF. Both references and experimental findings, showed the effect of alcohols resulted for less formation of NO_x. The reduction of NO_x is important to be an objective to achieve, as it is the most harmful emission caused by diesel engine.

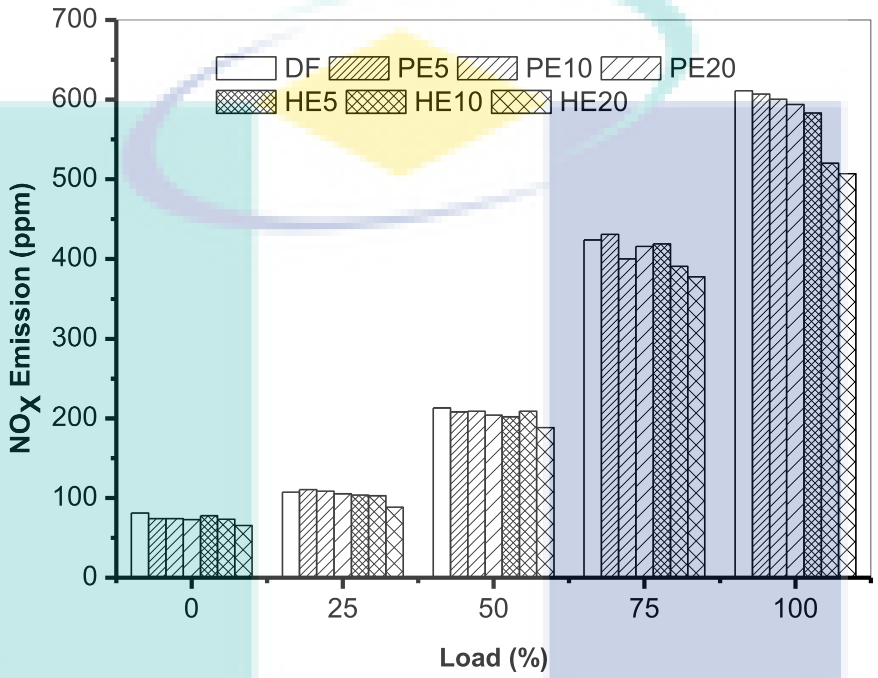

Figure 4.21 Variation of NO_x emission for different engine loads

4.7 Long-chain alcohol-diesel fuel blends optimization analysis (Response Surface Modelling)

To improve performance of a system, a process, or a product, optimizing is necessary to obtain maximum benefit from it. This is because it is commonly used in analytical chemistry to discover conditions, that an applied procedure produces the best possible response. Among the most relevant multivariate techniques used in analytical optimization is response surface methodology (RSM). This is used to collect mathematical and statistical techniques based on the fit of a polynomial equation to the experimental data, which must explain the behaviour of a data set with the objective of making statistical previsions. It can be well applied when a response or a set of responses of interest are influenced by several variables. The objective is to simultaneously correlate the levels of these variables to attain the best system performance. It is also designed to avoid failure experiment data set. In this optimization work, RSM designed by using Historical Data that developed based on imported data that already exist.

4.7.1 Analysis of the model

The design expert software was operated to fit experimental readings into the second order polynomial equation. Based on the analysis of variables (ANOVA), the principle model analysis was carried out to provide numerical information for the p-value. The selected significant models found must have p-values to be less than 0.05. The ANOVA for response variable parameters, BSFC and BTE are shown in Table 4.3. The quadratic models were developed in terms of actual factors and using quadratic equations from equation 4.1, 4.2, 4.3 and 4.4. The input parameters were engine load (A) and the percentage of alcohol in fuel mixture (B).

For pentanol,

$$BSFC = +1136.63947 - 21.42423\times A - 11.72825\times B + 0.067288\times A\times B + 0.12410\times A^2 + 0.26646\times B^2 \quad 4.1$$

$$BTE = +0.041989 + 3.14198\times10^{-3}\times A + 3.2484\times10^{-3}\times B - 2.54994\times10^{-6}\times A\times B + 3.697145\times10^{-7}\times A^2 - 5.32691\times10^{-5}\times B^2 \quad 4.2$$

For hexanol,

$$BSFC = +1129.177757 - 20.63562\times A - 13.91673\times B + 0.894987\times A\times B + 0.11582\times A^2 - 0.31794\times B^2 \quad 4.3$$

$$BTE = +0.041358 + 2.86508\times10^{-3}\times A + 5.09382\times10^{-3}\times B - 3.49998\times10^{-5}\times A\times B + 4.67371\times10^{-6}\times A^2 - 5.06782\times10\text{-}5\times B^2 \quad 4.4$$

4.7.2 Evaluation of the model

The evaluation of the models was made by using ANOVA. In Table 4.2, various responses were recorded to evaluate the suitability of the selected models. The value of p was less than 0.001, indicating a significant model terms and reference limit was set at 0.05. The regression statistics, goodness of fit (R^2) and the goodness of prediction (Adjusted R^2) were shown in Table 4.3.

The R-value was the total variability of the response after considering the significant factors. The Adjusted R^2 value accounts for the number of predictors in the model. The value of R^2 and Adjusted R^2 was studied to affirm the validation of the developed models, fitting test, data regression, significance analysis and individual model coefficients. The coefficient of R^2 was used to prove the quality of the fitted polynomial function models that represents the proportion of variability of the response as a result of the input variables. From table 4.3, it is noted that the model variable number increases with the increases of the determination coefficient (R^2). According to (Adam, Aziz et al., 2016), it is recommended to use the Adjusted R^2, that decreases if unnecessary terms are added to the model. In this case, close value of R^2 and Adjusted R^2 indicated a low probability for insignificant terms to be included in the model (M. Pandian, Sivapirakasam et al., 2011). Finally, both values indicate that the models fitted the data very well.

Table 4.2 Analysis of variables for responses (Values of p-value)

	Pentanol		**Hexanol**	
Source	BSFC (kg/kWh)	BTE (%)	BSFC (kg/kWh)	BTE (%)
Model	< 0.0001	< 0.0001	< 0.0001	< 0.0001
A	< 0.0001	< 0.0001	< 0.0001	< 0.0001
B	0.0436	0.1840	0.0297	0.0225
AB	0.1018	0.9516	0.0113	0.1919
A^2	< 0.0001	0.9719	< 0.0001	0.4760
B^2	0.2594	0.8301	0.1204	0.7414

Table 4.3 Model evaluation

	Pentanol		**Hexanol**	
Source	BSFC (kg/kWh)	BTE (%)	BSFC (kg/kWh)	BTE (%)
Mean	492.57	0.22	493.23	0.22
Std. Deviation	44.93	0.048	38.10	0.030
Model degree	Quadratic	Quadratic	Quadratic	Quadratic
R^2	0.9869	0.8859	0.9900	0.9500
Adjusted R^2	0.9822	0.8452	0.9682	0.9864
Predicted R^2	0.9704	0.7169	0.9766	0.8805

4.7.3 Interactive effect of percentage of alcohol blends and engine load

The contour and three dimensional surface response plot of the relation between the percentage of alcohol blends and engine loads are shown in Figure 4.22 to 4.29. Specifically, Figure 4.23 and 4.25 show the interaction of the percentage of alcohol blends, 1-PN and 2-EH with various engine loads in the response of BSFC. Figure 4.27 and 4.29 indicate the interaction of the percentage of alcohol blends, 1-PN and 2-EH with various engine loads in response on BTE.

From Figure 4.23 and 4.25, the RSM plot variation of BSFC varies to engine loads. From the figures, BSFC of the fuel blends decrease from 0% to 75% engine and slightly increase at 100% engine load. It is observed that total average BSFC was found lower at rated engine loads for all fuel blend compared to DF. In the Figure 4.23, the predicted BSFC reduced by 6.70%, 10.82% and 11.51% for PE5, PE10 and PE20, respectively. Also, in Figure 4.25, predicted BSFC reduced by 7.04%, 11.18%, and 10.48% for HE5, HE10 and HE20, respectively. It is observed from both figures that the BSFC decreases with the addition of 1-PN and 2-EH. The reduced of BSFC was due increase of oxygen content in fuel blends, as well lower density and viscosity have led to better atomization which caused more complete combustion. In actual data, the decrease of BSFC found were 8.51%, 10.16%, 12.08%, 8.38%, 10.87%, and 10.99% for PE5, PE10, PE20, HE5, HE10 and HE20. The predicted data and actual data had almost similar percentage of BSFC reduction than DF which validated both data. The model analysis found were relevant to BSFC in the experimental engine test, as lower BSFC was also found in fuel blends with increase engine load. The RSM models for BSFC was accepted.

Moreover, in Figure 4.27 and 4.29, the BTE was plotted against engine loads and percentage of alcohol blends. It is observed that BTE of diesel engine increases with decreasing engine loads and percentage of alcohols. The average BTE were increased by 5.97%, 12.44%, 19.90%, 7.96%, 13.93%, and 23.38% for PE5, PE10, PE20, HE5, HE10, and HE20 than DF. Maximum BTE was found to be at the high percentage of long chain alcohol-diesel fuel blends, PE20 and HE20 at the same loading conditions. Oxygen presence in the fuel improves the combustion characteristics. This was due to the increase of oxygen content in fuel blends, and burnt more fuels during combustion. From the predicted data, highest BTE found at 100% engine load which slightly different than in actual data. In the actual data, slightly deterioration of BTE spotted at engine load 100%

because of lower fresh O_2 content which been replaced by exhaust gas. From actual data, increased of BTE were 7.03%, 12.09%, 17.55%, 12.25%, 12.95%, and 19.67% for PE5, PE10, PE20, HE5, HE10, HE20than DF. The predicted data and actual data had almost similar percentage of BTE increment than DF validated both data. The same findings by experiment was proven, as higher maximum BTE found engine performance fueled with fuel blends than DF.

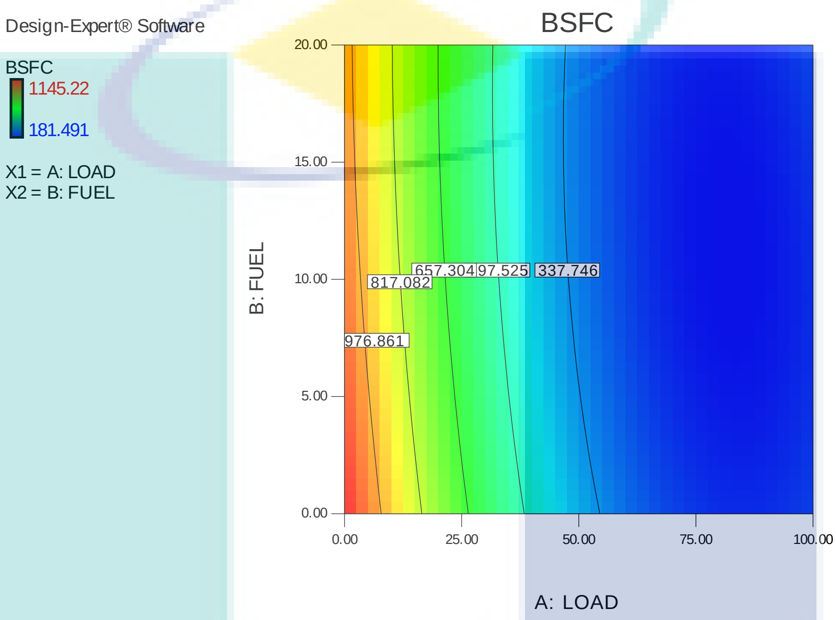

Figure 4.22 Contour plot of Effect of pentanol percentage and engine load on BSFC.

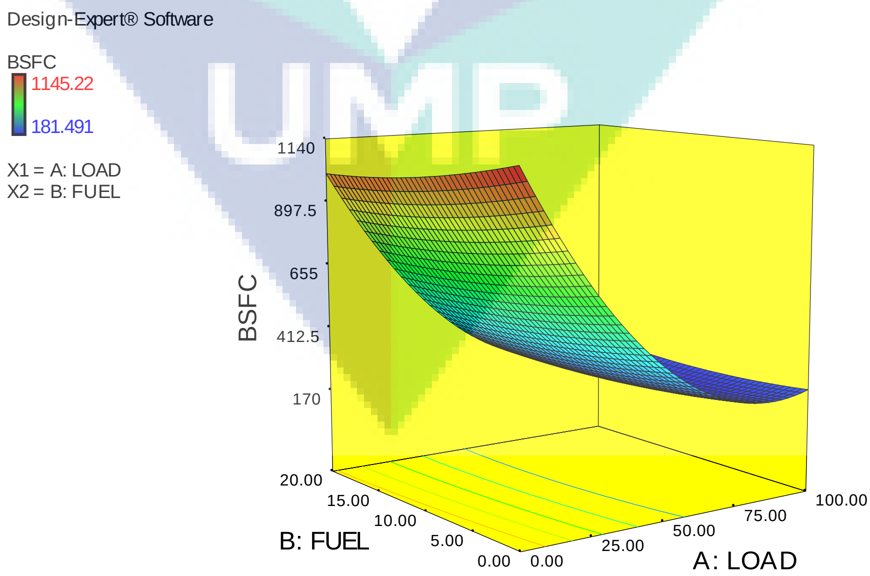

Figure 4.23 Surface plot of Effect of pentanol percentage and engine load on BSFC.

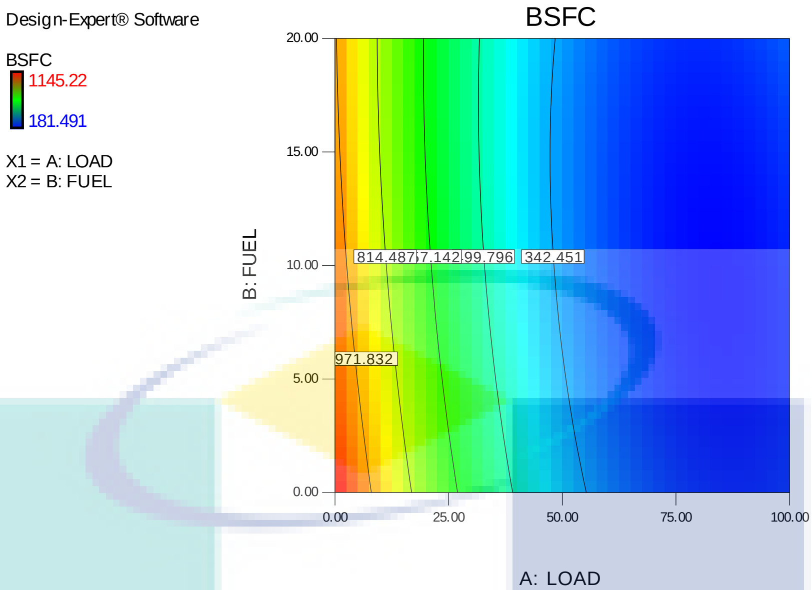

Figure 4.24 Contour plot of Effect of hexanol percentage and engine load on BSFC.

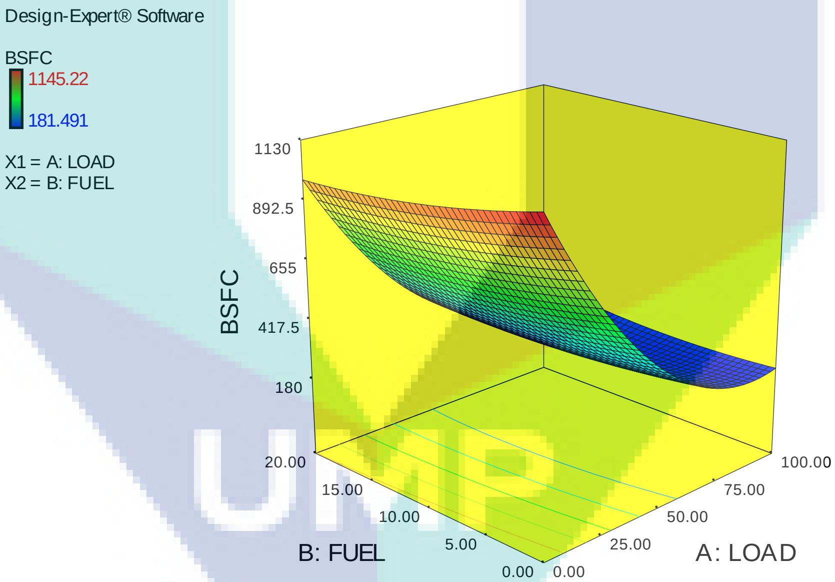

Figure 4.25 Surface plot of Effect of hexanol percentage and engine load on BSFC.

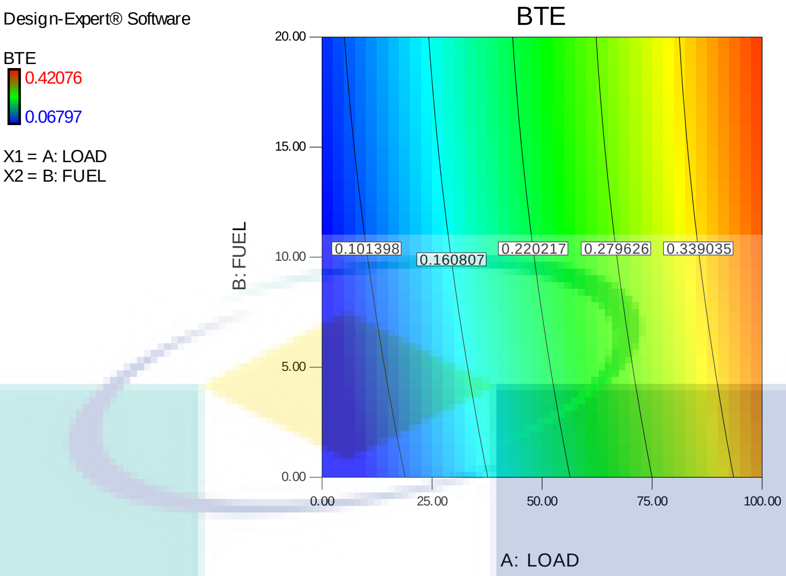

Figure 4.26 Contour plot of Effect of pentanol percentage and engine load on BTE.

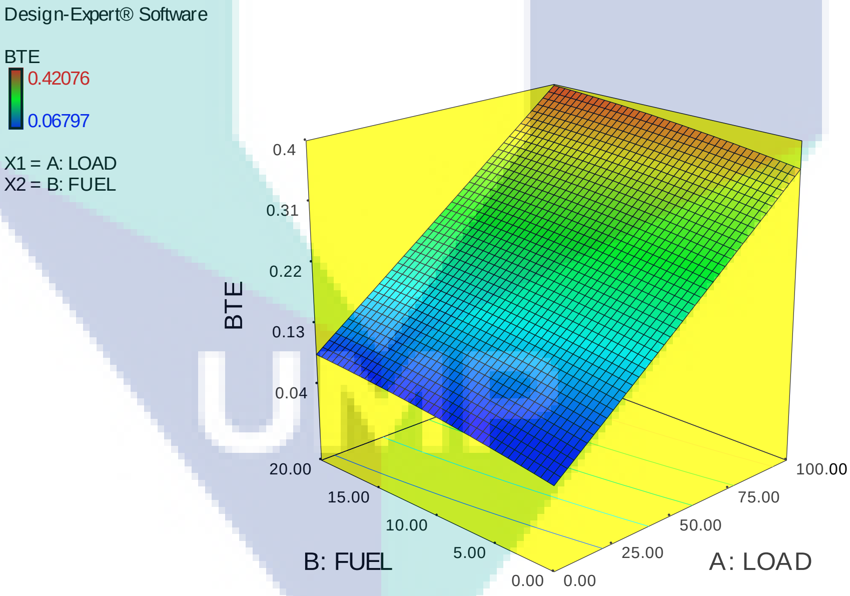

Figure 4.27 Surface plot of Effect of pentanol percentage and engine load on BTE.

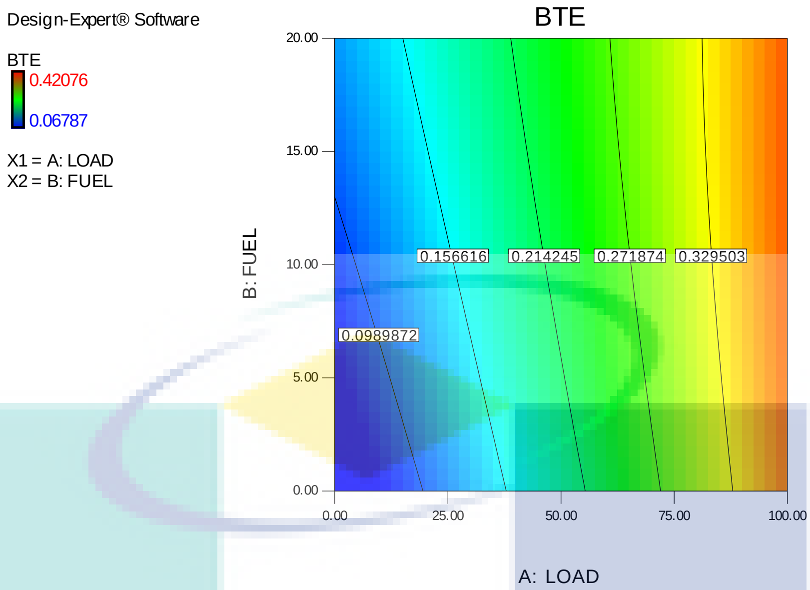

Figure 4.28 Contour plot of Effect of hexanol percentage and engine load on BTE.

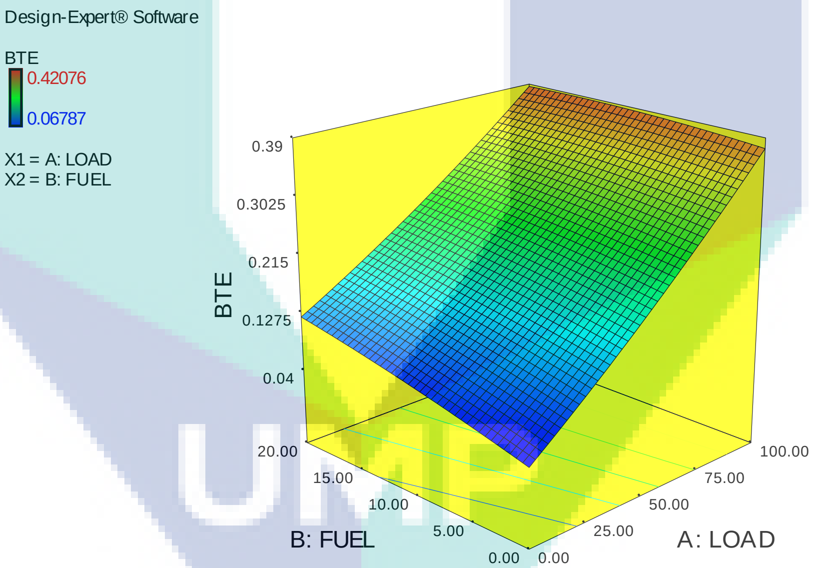

Figure 4.29 Surface plot of Effect of hexanol percentage and engine load on BTE.

4.7.4 Optimization

The Design Expert Software analyses and searches for optimum combination of factors, find optimum blends to satisfy the requirement set on each factor and responses. Optimization methodology indicated for the optimum numerical condition can be achieved with factors and responses. The criteria for optimization for engine load and blends was set in a range. While the optimal requirement for BSFC set was at a minimum and BTE was set at maximum. The numerical optimization can be achieved by pentanol-diesel fuel blends and hexanol-diesel fuel blends can be found in Table 4.4. Highest desirability based approach for different best solutions was obtained. The desirability found for pentanol-diesel fuel blends and hexanol-diesel fuel was 0.959 and 0.943 respectively. The input system parameters like 17.75% of 1-PN (PE20) at the engine load of 100% have the optimum parameters for test engine with BSFC (230.38 g/kWh) and BTE (0.39%). Similar optimum parameter was found in 18.50% of 2-EH (HE20) at the engine load of 100% and have the optimum parameters for test engine with BSFC (250.88 g/kWh) and BTE (0.39%)

4.7.5 Validation of correlation and optimization

In order to validate the optimization, the predicted and actual results for PE20 and HE20 was compared. For the actual responses, the average three measured result was calculated. The predicted values, actual values and percentage of errors are presented in Table 4.4. For pentanol, the percentage errors found was 3.08% for BSFC and -8.33% for BTE. While for hexanol, the percentage errors found was -8.40% and -3.65 %. The validation of the results shows that the percentage of errors was not higher than 10% which is in agreement as suggested by (M. Pandian, Sivapirakasam et al., 2011) and (Adam, Aziz et al., 2016). This indicates that the predicted results were acceptable.

Table 4.4 Table of correlation and optimization.

		Load	**Fuel**	**BSFC (g/kWh)**	**BTE (%)**
Pentanol	Predicted	100.00	17.75	230.38	0.39
	Actual	100.00	20.00	237.69	0.36
			Error (%)	3.08%	-8.33%
Hexanol	Predicted	100.00	18.50	250.88	0.39
	Actual	100.00	20.00	231.42	0.37
			Error (%)	-8.40%	-5.41%

CHAPTER 5

CONCLUSION

5.1 Conclusion

The study using long chain alcohol, 1-pentanol and 2-ethyl 1-hexanol as a new potential formulated alternative fuels showed potential as next generation alcohol based fuels. Based on the analysis and discussion, the following points emerged from the present investigation are as follows:

1. Objective 1 was achieved as the findings suggest that in general thermophysical properties for PE5, PE10, PE20, HE5, HE10 and HE20 has improved as the reported cetane number, viscosity and calorific value for 1-PN and 2-EH higher than compared short-chain alcohol, ethanol.

2. Objective 2 was obtained as:

    - The peak pressure for PE5, PE10, PE20, HE5, HE10 and HE20 were higher by of 0.91%, 0.98%, 2.24%, 1.19%, 2.43%, and 4.77% than DF due to the higher cetane number and lower viscosity which increase fuel-air mixing ratio with the presence of essential oxygen that resulted for a stronger premixed combustion phase and higher peak pressure.

    - Also, HRR is increased by 4.21% for PE5, 6.33% for PE10, 9.97% for PE20, 5.64% for HE5, 11.91% for HE10 and 15.6% for HE20 than DF due to the increase of O_2 content with increased ratio of 1-PN and 2-EH.

    - The BTE increased by 7.03%, 12.09%, 17.55%, 12.25%, 12.95%, and 19.67% for PE5, PE10, PE20, HE5, HE10, and HE20 respectively when compared with DF. The lower cetane number of 1-PN and 2-EH that causes for a longer ignition delayed, lead to have more fuel burnt during premixed mode that increase BTE.

- The decrease of BSFC found were 8.51%, 10.16%, 12.08%, 8.38%, 10.87%, and 10.99% for PE5, PE10, PE20, HE5, HE10 and HE20. The BSFC for all fuels blends are lower compared to DF due to their lower calorific value, lower density and higher latent heat of vaporization.

- Moreover, higher CO and EGO emissions found in fuel blends compared to DF. The blending of 1-PN and 2-EH with diesel resulted for less CO_2 and NOx emissions in entire engine load ranges. Since NOx is the most harmful, the reduction of it holds an importance in the engine research.

- In addition, the HC emissions for fuel blends also higher compared to DF except for HE5 which has lower HC at 50%, 75% and 100%.

3. Objective 3 was determined as:

- The correlation between fuel blends for 1-PN with DF was 17.75% ratio of 1-PN at engine load 100%.

- The correlation between fuel blends for 2-EH with DF was 20.00% ratio of 2-EH at engine load 100%.

The evidence from the study and analysis suggested that fuel blends, PE5, PE10, PE20, HE5, HE10 and HE20 as promising alternative alcohol based fuel-blends as the BTE was higher and BSFC was lower. Moreover, CO_2 and NO_x emission are lower for all fuels blends than DF. Meanwhile, the exhaust emission measurement of HE5 has successfully decreased the HC emission of diesel engine than that DF.

Moreover, through present study it was revealed that better combustion performed by using 2-EH as alcohol additives as lower HC, CO and NO_x found compared to fuel blends by using pentanol. In addition, the higher BTE and lower BSFC found in HE5, HE10 and HE20 compared to PE5, PE10 and PE20. The best optimum blends were HE20 with the highest BTE and lower BSFC. Additionally, HE20 also have lowest CO_2 and NO_x emission among fuel blends. Apparently, the addition of the long chain alcohol, 1-

PN and 2-EH with diesel shows positive impacts in performance and combustion as there are reduction in engine operating temperature which increases the life of the engine.

5.2 Recommendation

From the above discussions, the optimization blending of 1-PN and 2-EH with diesel from RSM are recommended and further research in this area by advancing the injection timing and introducing the exhaust gas recirculation techniques which will turn out to be highly efficacious. The correlation found from Design Expert Software should be produce for further work in order to obtain alternative long-chain alcohol with the optimum combustion condition. Besides, analysis and modelling by using ANSYS chemkin-pro also recommended for further works.

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LIST OF PUBLICATIONS

Journal of Publication

H. Suhaimi., A. Adam., et al. (2018). Analysis of Combustion Characteristics, Engine Performances and Emissions of Long-Chain Alcohol-Diesel Fuel Blends. Fuel, 220, 682-691. (ISI IF 4.601 Q1)

H. Suhaimi., Z. Abdullah., A. Adam., et al. (2018). A comparative analysis on emissions of some next generation long-chain alcohol/diesel blends in A direct-injection diesel engine. APPLIED INSTITUET OF PHYSICS, 15(3), 5435-5450. (SCOPUS indexed)

Z. Abdullah., **H. Suhaimi**., A. Adam., et al. (2018). Effect of Pentanol-Diesel Fuel Blends on Thermo-Physical Properties, Combustion Characteristics, Engine Performance and Emissions of a Diesel Engine. International Journal of Automotive and Mechanical Engineering, 15(3), 5435-5450. (SCOPUS indexed)

APPENDIX A
SAMPLE APPENDIX 1

The fuel blends were prepared for thermo-physical test and engine performance test. The blending procedure was conducted as below:

i. 5% 1-PN and 95% DF of 300 ml was prepared and mixed together in a 300 ml beaker.

ii. The beaker was placed on the plate in ultrasonic processor and a soft cloth was put between beaker and ultrasonic processor plate to avoid any vibration that could break the beaker.

iii. Amplitude at 40% and the cycle at 0.5 were set as the blending parameter. The lid was closed and stirring process was set for 2 minutes.

iv. After 2 minutes, the fuel blend was ready and stored in 8L tank. The tank was labelled as PE5.

v. Step i to iv was repeated until total of 2L of PE5 was stored.

vi. And step i to v was repeated by followed in details in Table 3.2 for PE10, PE20, HE5, HE10 and HE20.

APPENDIX B
SAMPLE APPENDIX 2

Procedure for measure the density:

i. Placed an empty 100 ml beaker on the microbalance and measured it.

ii. Filled a 12 ml syringe with 10 ml of DF.

iii. Filled the beaker on the microbalance with 2ml of DF, wait for 30 seconds for stable and accurate reading.

iv. Repeated step iii by filled the beaker for 4ml, 6ml, 8ml and 10ml.

v. And each test fuels undergoes step i to iv for 3 times.

APPENDIX C
SAMPLE APPENDIX 3

Procedure for measure kinematic viscosity:

i. Heat the transparent Newtonian liquid in the viscometer up until 40°C.

ii. Attached 100 ml capillary tube into viscometer chamber when temperature reached to 40°C for 5 minutes and took out.

iii. Suck the test fuels using vacuum until sample moving to start mark.

iv. Put the tube back into chamber and observed the time consuming for test fuels to move to end mark.

v. Repeated the operation for each test fuels for 3 times for average value.

APPENDIX D
SAMPLE APPENDIX 4

Procedure for calorific value:

i. Prepared a handling cup of calorimeter and weighted it using analytical balance.

ii. Prepared sample of test fuel on the handling cup and weighted it without the weight of handling cup. And the weight of sample test fuels must not exceed 1g.

iii. Cut a 10cm length of fuse wire and tied it at the electrodes in form of U shape.

iv. Placed previous prepared sample of test fuel in handling cup at the electrodes. (make sure the tied fuse wire touch fuel in handling cup)

v. Put the stand of electrode into the bomb cylinder and closed firmly.

vi. Closed the cap of bomb calorimeter was loosely.

vii. Filled the bomb cylinder with some oxygen to remove air in the cylinder for 20seconds.

viii. Closed the cap of bomb cylinder tightly and filled it with oxygen under allowing pressure of 25 atm.

ix. Put bomb cylinder into the tank of calorimeter that full of distilled water about 2L and placed it into bucket.

x. Connected the electrical fuse with bomb cylinder. (no polarity)

xi. Closed the lid of the calorimeter.

xii. Key in the sample name and weight of sample test fuel for calorimeter operation.

xiii. Installed the belt and on the stirrer of bomb calorimeter.

xiv. Waited the sample test fuels undergoes combustion until beep sound came from calorimeter operation.

xv. Loose the cap of bomb slowly to release the pressure inside.

xvi. Opened the bomb calorimeter and check the condition of fuse wire and fuel sample. (make sure fuse wire and fuel sample completely burnt)

xvii. Printed readings of calorific value.

xviii. Repeated the step i to xvii for 3 times to obtain average calorific value for each test fuels.