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EXPERIMENTAL INVESTIGATION OF A BUTYLCYCLOHEXANE / PROPYLCYCLOHEXANE BASED NAPHTHENIC BIO-BLENDSTOCK SURROGATE FUEL FOR USE IN A COMPRESSION IGNITION ENGINE

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ABSTRACT

Finding fuels derived from renewable sources that can displace fossil fuels is crucial to fight climate change as drop-in replacement fuels can immediately leverage ubiquitous, existing energy conversion hardware such as the reciprocating-piston internal combustion engine. However, any replacement fuel must necessarily maintain or ideally exceed the performance of fossil fuels. There are many possible pathways to obtain renewable fuels from biomass, but one pathway that has demonstrated the potential to meet the necessary thermophysical and performance requirements is the Catalytic Fast Pyrolysis pathway followed by hydrotreating to produce a bio-blendstock oil - rich in naphthenes. Due to the variability of bio-blendstock oil composition and variations in process parameters at such a production facility, developing a representative surrogate fuel is necessary for use in understanding compositional desirability and comparative experimental investigations are required to determine performance and emissions characteristics. This work builds upon past surrogate fuel formulation efforts to represent the naphthenic-rich bio-blendstock oils produced from a one ton per day (1TPD) catalytic biomass pyrolisis facility. Specifically, in this work a new formulation of surrogate fuel (SF), SF1.12, containing butylcyclohexane and propylcyclohexane, to represent the naphthenic hydrocarbon content, was created and produced and further blended (up to 30% by vol.) with research-grade No.2 diesel (DF2). These blends were then tested in a single-cylinder Ricardo Hydra compression ignition research engine. A range of conditions were evaluated by first holding the start of injection constant and varying the injection duration from max load (knock limit) to 40% of the max load value. In addition, experiments were performed by holding the fuel-air equivalence ratio constant while sweeping start of fuel injection, and the results were compared to previous generations of SF. It was demonstrated that increasing the proposed SF concentration in DF2 lead to an increase in load output, as well as to a decrease in sooting propensity without impacting NOx formation. It was also shown that this formulation of SF performs best in increasing load and reducing emissions compared to previously reported generations. This SF formulation in the concentrations evaluated (up to 30% by vol.) was shown to be a feasible candidate for displacing heavy distillate fossil fuels in engines.

Keywords: Catalytic Fast Pyrolysis Oil, Upgrading with Hydrotreating, Biofuel, Compression Ignition Engine, Performance, Emissions

NOMENCLATURE

Abbreviations

aTDC   After TDC
CAD   Crank Angle Degrees
CO   Carbon Monoxide
COV   Coefficient of Variation
DF2   No. 2 Diesel Fuel
IMEP   Indicated Mean Effective Pressure
MFB   Mass Fraction Burned
NOx   Nitrogen Oxides
RPM   Rotations Per Minute
SF   Surrogate Fuel
SOI   Start of Injection
THC   Total Hydrocarbons

Greek letters

ϕ Equivalence Ratio

1. INTRODUCTION

Engines operating on fossil fuels are under scrutiny due to the high quantity of carbon-based and other co-pollutants commonly referred to as greenhouse gas emissions they produce. However, it is important to understand that the problem in this case arises mostly from the fuel used and not necessarily the engine hardware itself. Fossil fuels are particularly harmful to the environment

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because they are carbon sources that would otherwise be stored underground, but are burned at the earth's surface and released into the atmosphere. Biofuels offer a solution to the problems encountered with fossil fuels. The carbon present in biofuels is extracted from the atmosphere by the plants from which the fuel is made, so although the combustion of such fuels releases a similar amount of carbon into the atmosphere as fossil fuels, the carbon comes from renewable sources and can enable a sustainable cycle, leading to overall lower life-cycle carbon emissions [1].

Biofuels can be produced from many different sources [2–7]. Of particular interest are second generation biofuels given that, unlike first generation biofuels, they are manufactured from non-food sources and do not interfere with resources that could go into food production. One common feedstock to produce such biofuels is woody biomass due to its abundance and low cost [8]. As for production, there are many processes to convert biomass into bio-oils including fractionation, liquefaction, pyrolysis, hydrolysis, gasification and fermentation [9]. In particular, fast pyrolysis is one of the most promising pathways for biofuel production for being the most economically viable option [10].

Due to the many years of effort in optimizing engine architectures to operate with fossil fuels, it is important that alternative fuels have similar characteristics that would allow it to be used in current engines without modifications. However, bio-oils, which are the direct product obtained from biomass, tend to have undesirable characteristics that can damage engine components and decrease performance [11–14]. The bio-oil that is the product of fast pyrolysis presents high oxygen content, which is undesirable and is the most critical difference between bio-oils and hydrocarbon fuels [15, 16]. Further, the bio-oils can also contain large amounts of water, which are responsible for the lower flame temperature and rate of combustion as well as longer ignition delay [14, 15]. It is thus necessary to further upgrade bio-oil to remove the oxygen content, which can be done catalytically during the fast pyrolysis process, known as Catalytic Fast Pyrolysis. Calalytic deoxygenation is also achieved during hydrotreating, to yield a higher grade of finished product [8].

A pilot-scale catalytic pyrolysis unit [17] was used to convert woody biomass into a bio-blendstock using the Catalytic Fast Pyrolisis with hydrotreating pathway. The process parameters were adjusted to produce a fuel with a low aromatic content, since aromatics have been demonstrated to increase soot formation [18, 19]. The aromatic content was reduced by increasing the naphthenic content. Since the production of the bio-blendstock is expensive and time-consuming, and to be able to make large enough quantities for engine testing, the use of a surrogate fuel (SF) is needed and different variations have been devised and tested by [20, 21]. These SFs were made from pure chemical compounds with the goal of mimicking the thermophysical properties of different versions of the bio-blendstock. Engine experiments were conducted to evaluate the performance of blends of different percentages of the SF and No.2 Diesel fuel (DF2).

The first generation SF, SF1, documented by Ran et al. [20] demonstrated good performance and overall potential for use as a diesel replacement. However, it still contained larger than desired aromatic content and the only representative naphthenic compound was decalin. The work by Ran et al. was further investigated in a second generation SF, SF1.01, by Hadlich et al. [21] which reduced the aromatic content by replacing it with naphthenic content. This increased naphthenic content was divided into equal parts of decalin and butylcyclohexane. This yielded an improvement in performance, nearing pure diesel levels.

In the bio-oil produced by the pilot-scale facility, there are many more naphthenic hydrocarbons that are produced than the two present in SF1.01, so there was a desire to understand how changing the representative naphthenic compounds would affect the performance of the SF during engine testing. From chemical analysis of the pilot-scale facility bio-oil, notable quantities of propylcyclohexane were found in the bio-blendstock. Pure compounds to act as the representative of naphthenic hydrocarbon content were selected based on availability and cost. Decalin was more readily available at lower cost so it was selected first in [20]. Wanting to understand the effects that a specific representative pure compound to all naphthenic hydrocarbons may have on performance, a more expensive and lesser available compound of butylcyclohexane then trialed and evaluated in [21]. As a performance difference was noticed between decalin and butylcyclohexane in prior work, the logical progression was to use the most expensive, least commercially available, most abundant naphthenic hydrocarbon present in the produced bio-blendstock oil, propylcyclohexane. An equal composition of butylcyclohexane and propylcyclohexane is now considered in this third generation SF study, thus entirely replacing the decalin content from the second generation SF1.01 studied by Hadlich et al. [21].

Therefore, in this work an improved formulation of a high-naphthenic content surrogate fuel containing butylcyclohexane and propylcyclohexane, in equal parts, was designed and produced. Experiments were conducted to test the overall performance and emissions profile of different blends of this SF with No.2 Diesel in a single-cylinder compression ignition research engine. The results were then compared to the pure diesel baseline case as well as with previous generations of the SF reported in [20, 21].

2. METHODOLOGY
2.1 Surrogate Fuel Composition and Mixing

The surrogate fuel detailed in this work, labeled SF1.12 is the third generation of surrogate fuels based on the same bio-blendstock previously studied [20, 21]. All three surrogate fuels target to mimic a bio-oil containing a 1 wt% oxygen content. The main differences between the three generations are the ratio of naphthenes to aromatics and the quality of the naphthenic content. In the first generation [20], labeled SF1, the aromatic content was 28.5% by volume and the only compound utilized to represent the naphthenes was decalin. In the second generation [21], labeled SF1.01, the aromatic content was reduced to 14.3% and replaced with additional naphthenes. The two representative naphthenic compounds used were decalin and butylcyclohexane, in equal quantities. In SF1.12 there remained the reduced 14.3% of aromatics present and the naphthenic blend now contained equal amounts of butylcyclohexane and propylcyclohexane. The detailed composition of all SF generations is listed in Table 1, while details regarding the bio-blendstock can be found in [20]. SF1.12 was mixed in-house using pure chemical compounds with

purity listed in Table 1.

2.2 Engine Setup

The engine experiments were performed at Stony Brook University for all SF generations. The engine used for the experiments was a single-cylinder compression ignition Ricardo Hydra research engine. Details about the engine build can be found in [20, 22]. The specifications of the engine are listed in Table 2. The laboratory research engine facility is fully instrumented so as to be able to measure performance parameters as well as exhaust gas emissions. The intake, in-cylinder, and exhaust pressures time-histories were measured by Kistler 4011A, 6045B, and 4049B pressure transducers, respectively, at a resolution of 0.1 crank angle degrees (CAD). The engine speed was measured and modulated by a 30hp DyneSystems controlled Eddy Current dynamometer, which also measures the engine's torque output. A Horiba MEXA 7100 D-EGR was used to measure the emissions species in the exhaust gas, and thus measures the equivalence ratio, ϕ, in addition to the ϕ measurements from an ECM LambdaCAN. A TSI 3795-HC Nanoparticle Emission Tester (NPET) was used to measure the concentration of soot particles in the exhaust gas stream. The fuel injection pressure is controlled using a control system that makes use of a Bosch CP3 pump as the actuator, a Kistler 4067E as the sensor, and a PID controller built in an in-house LABVIEW code. This LABVIEW code was used to receive, process, and send signals to two NI CompactRIO chassis and an array of NI and Drivven modules that comprises the data acquisition and engine control system.

2.3 Uncertainty Analysis

It is important to quantify the uncertainties associated with findings to be able to discern the quality of the data. That is true even more so for a system as complex as an engine, and where the error in measurements will propagate to many parameters that are calculated from them. In this work, uncertainty analysis needs to not only be considered for the experimental setup, but also the fuel composition itself, given that the surrogate fuel was prepared in-house. Details regarding the errors associated with the instruments of the experimental setup and fuel composition can be found at [20] and [21], respectively, and they apply to this work given that the experimental setup as well as the mixing and blending processes were identical. The amount of SF1.12 produced was also the same as the amount of SF1.01 produced, a total volume of 2.5 liters each. The total mixing error associated with the mixing of SF1.12 can be found in Table 3.

The uncertainty shown in the error bars in this work represent the standard error, which is given by the standard deviation divided by the squared root of the number of points used to obtain the average of a given measurement. For the high speed measurements such as the $IMEP_g$ the number of data points used to calculate the mean was of 300, or one value per cycle of a data point. As for the data collected by low-speed sensors, as well as the values derived from them, the number of data points used for averaging was of 15, or one measurement for every 20 cycles at an engine speed of 1200RPM. The cycle chosen as the representative for a given data point of 300 cycles was selected as follows: first the average of the pressure values was calculated;

then the mean absolute difference was calculated between average cycle and each of the 300 individual cycles and the cycle with the smallest difference was chosen as the representative cycle. This way it captures the average behavior observed, but the values are still calculated based on a real cycle.

2.4 Experimental Procedure

In order to ensure the collection of useful and reliable data, certain procedures were followed throughout the experiments. Once the desired operating conditions were reached by varying the combustion parameters, the engine was allowed to operate continuously for 20 seconds to reach steady state operation. A threshold on the coefficient or variation (COV) of 5% was set to ensure the combustion process was stable and the data did not contain an excessive amount of cycle-to-cycle variation. Once both steady state operation and a COV below 5% were reached, 300 consecutive cycles were recorded for each set of operating conditions and the results were averaged.

The experiments were performed using two different methods. First a load sweep was performed, where the fuel injection timing and pressure were held constant at -1.5 degrees aTDC and 550 bar, respectively, for all blends. The maximum Indicated Mean Effective Pressure ($IMEP$) was then achieved by varying the injector opening duration and change the fuel-air charge mixture equivalence ratio. The limiting factor for the $IMEP$ was knock which is quantified by the Maximum Pressure Rise Rate, and has a maximum value of 10 bar/CAD so that the engine components are not damaged. The data point at maximum $IMEP$ was evaluated and recorded. The load was then reduced to approximately 80, 60, and 40% of the maximum $IMEP$, and data was collected at each of these operating conditions. The second sweep was of injection timing. For this, the equivalence ratio ϕ was held constant at 0.25, as well as the fuel injection pressure which was constant at 550 bar. The fuel injection timing was then swept from knock limit to misfire limit in steps of 1 CAD. The knock threshold used was the same as for the load sweep of a Maximum Pressure Rise Rate of 10 bar/CAD or less.

Both processes of load and injection timing sweep were repeated for pure diesel fuel, 10%SF1.12/90%DF2, and 30%SF1.12/70%DF2. During all the experiments the intake and exhaust gas pressures were held constant at ambient conditions, and the intake air temperature was also at ambient conditions. The engine speed was fixed at 1200 rotations per minute (RPM). The temperatures of the coolant and oil were controlled to be at 358K and 348K, respectively.

3. RESULTS AND DISCUSSION

3.1 Comparison of SF1.12 Blends and Diesel

In this subsection the load performances of 10 and 30% SF1.12 blends with diesel are compared to that of pure diesel, as well as the emissions associated to each.

3.1.1 Effects of Fuel in Fuel System Components. It is known that an increased naphthenic content is generally accompanied by a decrease in fuel lubricity, which has been shown to damage the components of fuel systems designed for operation using diesel fuel [23]. However, it was found that in blends of up to 30% of SF1.12 with 70% No.2 diesel by volume there was no

TABLE 1: COMPOSITION OF SURROGATE FUELS

Chemical Compound	SF1 [Wt.%]	SF1.01 [Wt.%]	SF1.12 [Wt.%]	Purity/Description
Phenanthrene	3.5	1.8	1.8	98%
Decane	4.5	4.5	4.5	For Synthesis
4-Methylphenol	5.0	5.0	5.0	≥ 99%, FG
Octadecane	7.0	7.0	7.0	99%
Tetralin	25.0	12.5	12.5	Anhydrous, 99%
Decalin	55.0	0.0	0.0	Anhydrous, ≥ 99%
Butylcyclohexane	0.0	34.6	34.6	≥ 99%
Propylcyclohexane	0.0	0.0	34.6	98.0%+

TABLE 2: EXPERIMENTAL SETUP DETAILS

Exhaust Valve Opening (EVO) [°aTDC]	122
Exhaust Valve Closing (EVC) [°aTDC]	366
Intake Valve Opening (IVO) [°aTDC]	-354
Intake Valve Closing (IVC) [°aTDC]	-146
Connecting Length Rod [mm]	160
Number of Valves per Cylinder	4
Piston Pin Offset [mm]	0.6
Stroke [mm]	86
Bore [mm]	79
Compression Ratio	15.5:1

TABLE 3: SURROGATE FUEL BLENDING ERROR

Blend Composition	Total Blending Error [%]
0% SF 100% DF2	0
10% SF1.12 90% DF2	2.94
30% SF1.12 70% DF2	5.54

FIGURE 1: IGNITION DELAY AND BURN DURATION OF SF1.12 BLENDS AND DIESEL

observable degradation in fuel system components. During the course of the engine experiments no part failures occurred and no visible damage was observed or needs to be reported.

3.1.2 Ignition Delay and Burn Duration. Ignition delay and burn duration are two important parameters in describing the behavior of fuels. It is desirable that both of these parameters are as low as possible. A low ignition delay allows for better control of when the fuel will ignite, making it easier to define fuel injection timings that will yield a heat release at a timing that is mechanically advantageous. As for the burn duration, the lower it is, the higher the energy produced per unit time will be. When more heat is released it instantaneously increases the temperature of the surrounding gases and makes the fresh air-fuel mixture more prone to autoignite, leading to an overall higher percentage of fuel being consumed. The burn duration, also referred to as CA10-90, represents the time (measured in CAD) between combusting 10% and 90% of the total fuel burnt in the cycle. Figure 1 shows a plot with two curves, one for the burn duration (CA10-90) and the other for the ignition delay. As can be observed, although the ignition delay for all blends remains approximately constant, the CA10-90 is lower for the SF1.12/DF2 blends compared to pure diesel by as much as 4 CAD, or around 30%. This ultimately means that even though it does not reach
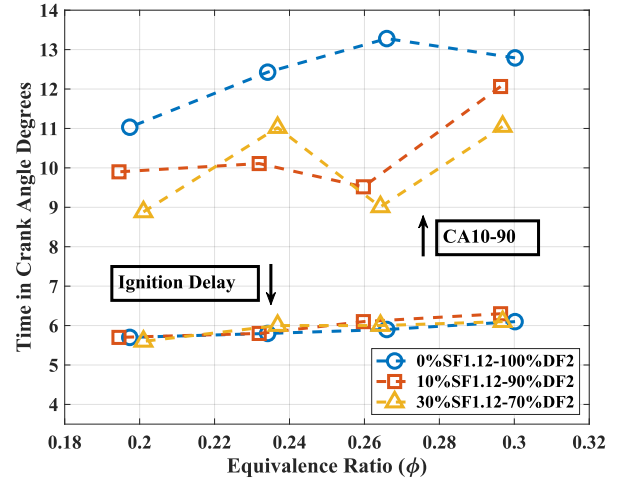
autoignition conditions faster, the fuel is consumed faster once it starts to autoignite and thus releases heat quicker. This significant difference explains many of the trends observed in upcoming subsections.

3.1.3 Cylinder Pressure and Heat Release. Figure 2 shows a double y-axis plot with cylinder pressure on the left y-axis and mass fraction burned (MFB) on the right y-axis. The values are reported for an equivalence ratio of 0.26 for all blends. A clear trend in the cylinder pressure can be seen that, as the surrogate fuel content increases, the cylinder pressure after start of combustion increases. Given that the fuel injection timing is held constant for all blends and the ignition delay was demonstrated to not vary significantly, the explanation for this increase can be found in the final stages of the heat release process. The inset plot shows an exploded view of the last stages burn process in the MFB curve for all blends. It is noticeable that, although all blends combust at similar rates initially, at approximately 70% of the total heat release the rate at which the fuels are consumed increases with increasing SF concentration. This means the higher the content of SF, the quicker it burns after approximately 70% of the total fuel combusted in the cycle. This explains the behavior observed in Fig 1 which shows burn duration decreasing with increasing SF content in the blend.
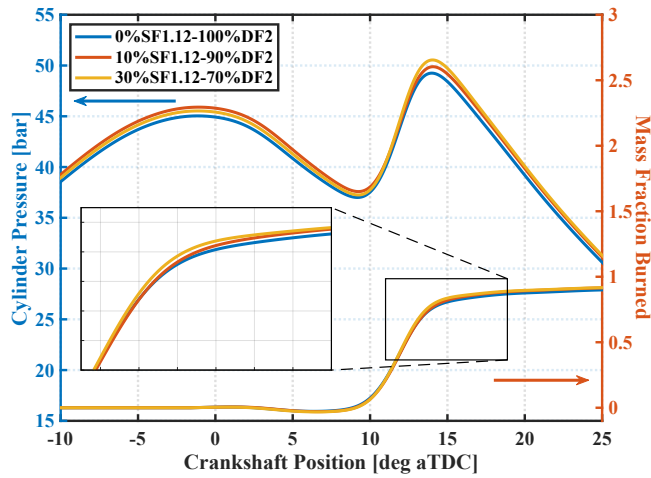
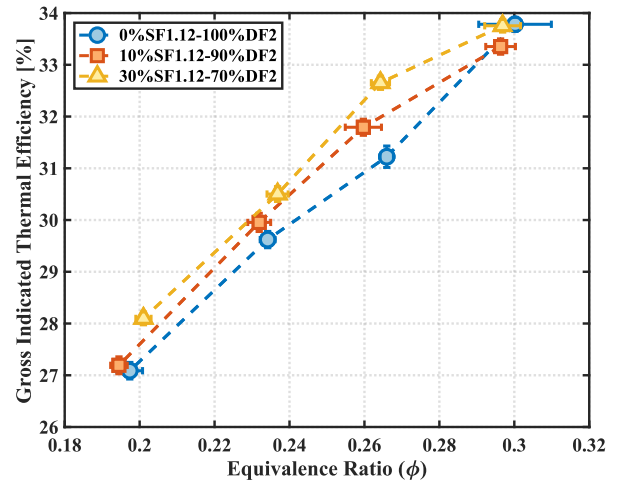**FIGURE 2: CYLINDER PRESSURE AND HEAT RELEASE RATE OF SF1.12 BLENDS AND DIESEL**

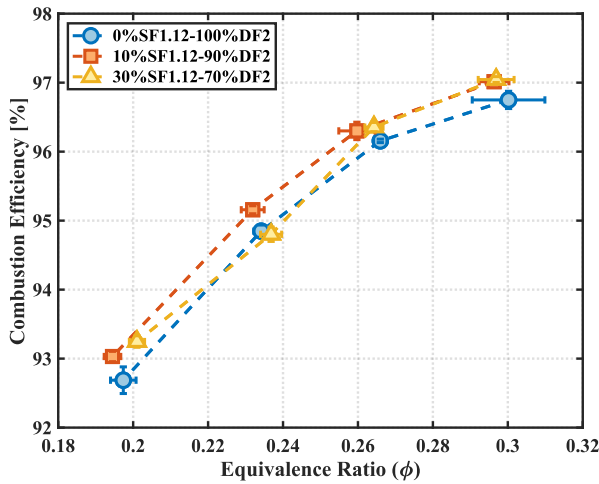**FIGURE 4: GROSS INDICATED THERMAL EFFICIENCY OF SF1.12 BLENDS AND DIESEL**

FIGURE 3: COMBUSTION EFFICIENCY OF SF1.12 BLENDS AND DIESEL
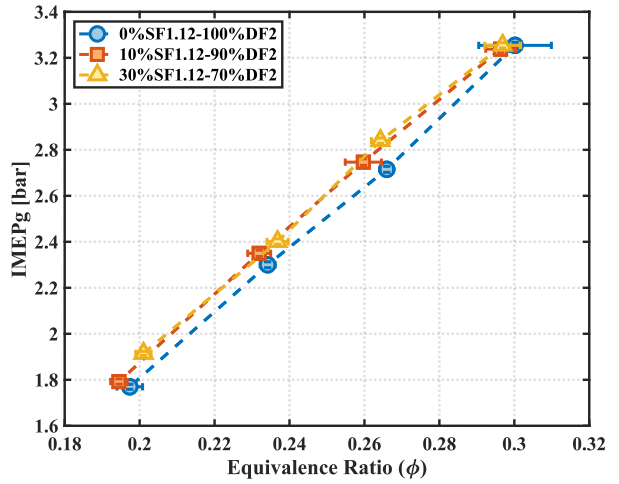

FIGURE 5: GROSS INDICATED MEAN EFFECTIVE PRESSURE OF SF1.12 BLENDS AND DIESEL

3.1.4 Combustion Efficiency. The combustion efficiency is a measure of the percentage of fuel input that was converted into heat. Figure 3 shows the combustion efficiency as a function of the equivalence ratio ϕ for all SF blends and pure DF2. It can be observed that the combustion efficiency in both SF blends is marginally higher than pure diesel, with the exception of one outlier in the 30%SF1.12-70%DF2 blends at a ϕ of 0.24. This can be explained by the mass fraction burned curve which shows the combustion rate of pure diesel becomes slower compared to the SF blends at a value of approximately 0.7. This in turn indicates that when all fuels reach a crankshaft position in which combustion cannot be sustained due to low temperature and pressure, the SF blends have combusted slightly more fuel than diesel.

3.1.5 Indicated Thermal Efficiency. In addition to the combustion efficiency, the gross indicated thermal efficiency is a ratio of how much work is produced compared to the total heat released during the cycle without considering work done during engine breathing events. The curves presented in Fig. 4 demonstrate the gross indicated thermal efficiency increases as the SF percentage increases in the blend. Because the thermal efficiency is calculated based on the work produced, it is heavily affected by the timing of the combustion process and how usefully work can be extracted from a mechanics standpoint. When heat is released early in the power stroke it produces more work since the volume inside the cylinder is small, so it will cause a greater increase in pressure and thus greater force pushing on the piston. Sections 3.1.2 and 3.1.3 demonstrate that higher SF content in the blend leads to faster burn duration and higher cylinder pressures, which explain the trend observed in Fig. 4.

3.1.6 Engine Load. When it comes to engine load indicators, the indicated mean effective pressure ($IMEP$) is widely used because it normalizes the work output by the size of the engine, and thus allows for comparison of performance between engines of different categories. It receives the subscript n, for
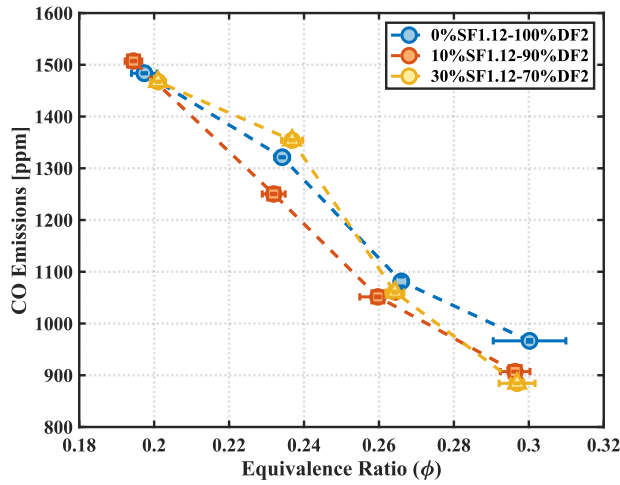
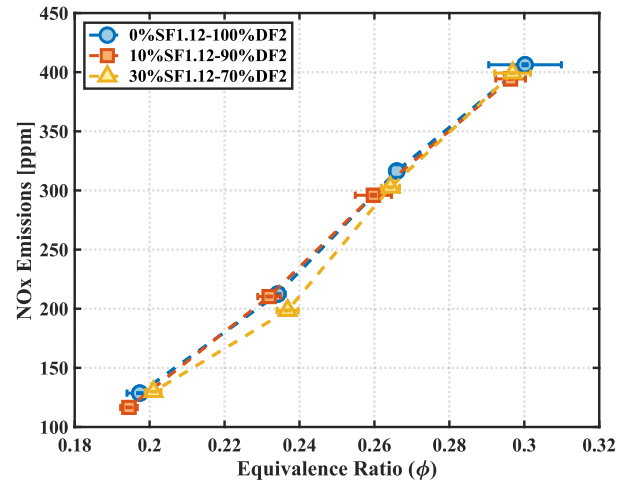**FIGURE 6: CO EMISSIONS OF SF1.12 BLENDS AND DIESEL**

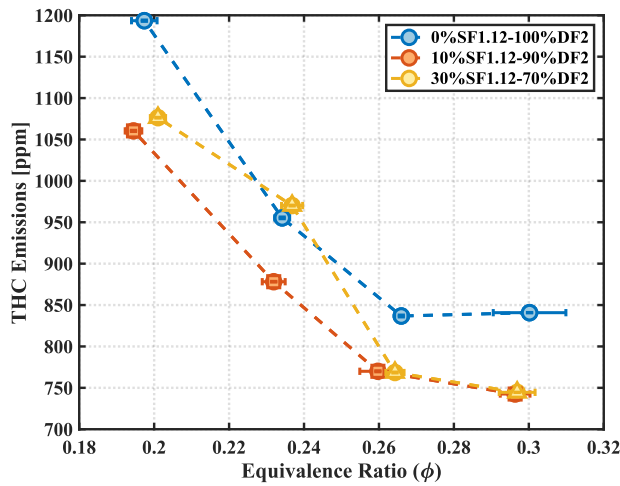**FIGURE 8: NOX EMISSIONS OF SF1.12 BLENDS AND DIESEL**

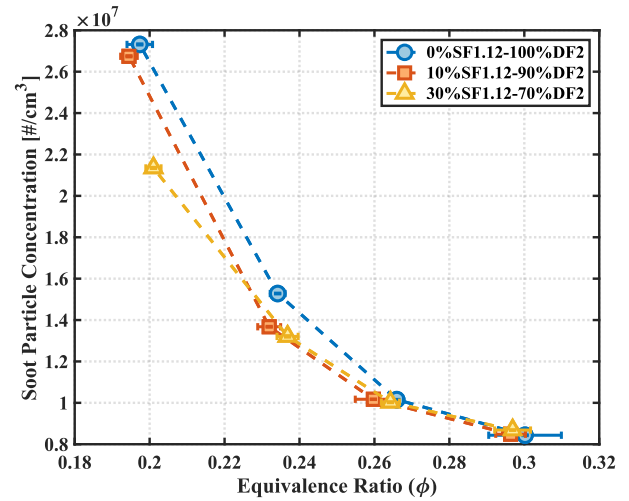**FIGURE 7: THC EMISSIONS OF SF1.12 BLENDS AND DIESEL**

FIGURE 9: SOOT EMISSIONS OF SF1.12 BLENDS AND DIESEL

net, if the breathing strokes are accounted for, and the subscript g for gross otherwise. Figure 5 shows the $IMEP_g$ of all SF blends and diesel as a function of ϕ. It can be observed that the addition of SF to diesel increased the $IMEP_g$ throughout the entire range of equivalence ratios explored. This can be explained by a combination of factors, which are manifested in the increase in the combustion and thermal efficiencies.

3.1.7 CO and THC Emissions. Carbon monoxide (CO) and unburnt hydrocarbons (THC) are two undesirable combustion products. They are not only lethal for humans if exposed to significant quantities as is the case of CO [24], but can also be strong contributors to global warming in the case of methane [25], which is one of the species defined as THC. Figures 6 and 7 show the levels of CO and THC, respectively, for all SF blends and pure diesel. For both CO and THC there is a pattern of decreasing emissions with the addition of SF, with the exception of one outlier for the 30%SF1.12-70%DF2 blend at a ϕ of 0.24. While the CO tends to decrease slightly with the addition of SF, a larger decrease is present in the THC curves. The trends

observed in both cases can be explained by the fact that CO and THC are incomplete combustion products, and are thus inversely proportional to the combustion efficiency of the cycle. This is confirmed when contrasting these two figures with Fig. 3, when the combustion efficiency is higher for any one blends compared to the others the CO and THC emissions are lower.

3.1.8 NOx and Soot Emissions. Nitrogen oxides (NOx) and particulate matter, or soot, are emissions species that must also be discussed. NOx molecules have very high global warming potential [26], whereas soot particles cause health issues due to its effect on the respiratory track of humans and wildlife [27]. The formation of NOx takes place in the presence of oxygen and nitrogen molecules at high temperatures (above 1800K) and thus is mostly formed in locally lean areas of the combustion chamber. Conversely, soot is formed in very fuel-rich regions where fuel molecules that cannot mix with air get fused together by combustion happening in its surroundings. Both soot and NOx are formed in significant quantities in heterogeneous mixtures of fuel and air found in the fuel spray of conventional diesel direct
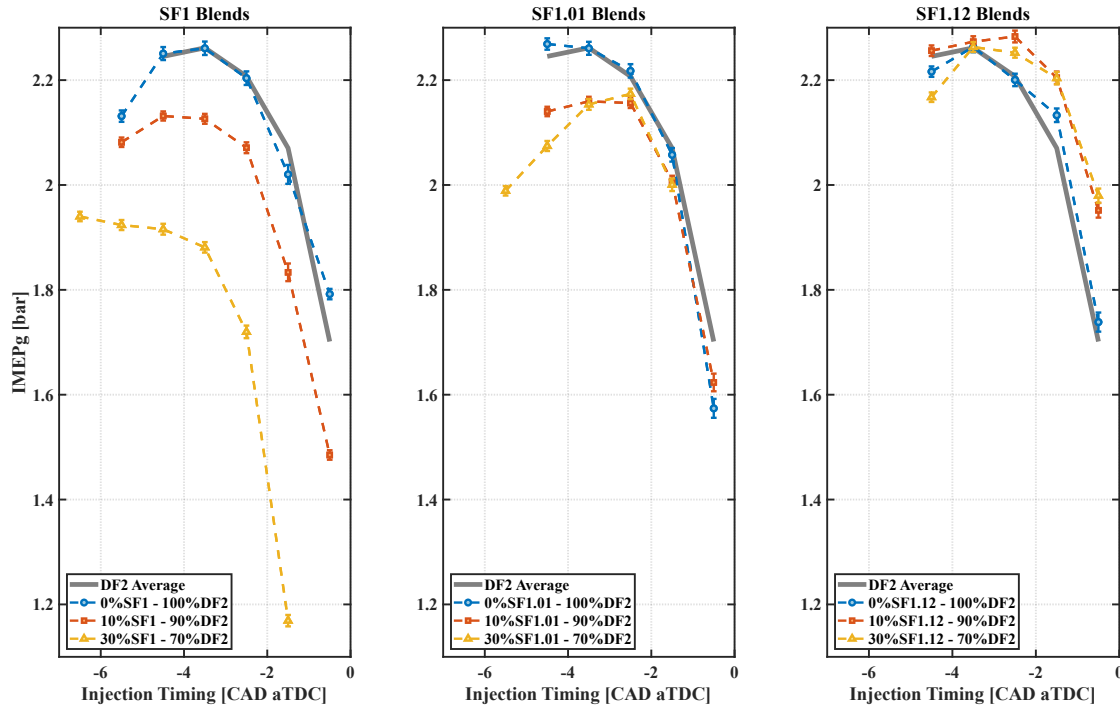
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FIGURE 10: COMPARISON OF $IMEP_g$ FOR DIFFERENT SF GENERATIONS

injection combustion.

There is a strong correlation between soot and NOx formation in diesel engines, normally referred to as the NOx-soot trade-off [28]. It dictates that in a combustion event with shorter ignition delay there is less time for fuel to mix with air which leads to a worse mixing process and overall lower temperatures in the cylinder, tending to form more soot and less NOx. Conversely, if the ignition delay is longer there is more time for the fuel spray to break apart and mix with air, having better mixing, higher cylinder temperatures and ultimately higher NOx and lower soot formation. This trend can be observed in Figs. 8 and 9 which show the NOx and soot emissions, respectively. As NOx emissions increase with increasing equivalence ratio the soot emissions decrease for all fuel blends. This also relates to the small but steady increase in the ignition delay from Fig. 1 with the increase in ϕ for all blends.

There is, however, another trend that can be observed when it comes to these emissions as a function of the concentration of SF in the blend. While the NOx emissions in Fig. 8 remain mostly constant as the SF concentration varies, the soot emissions become lower with increasing SF concentration. This is particularly evident in lower equivalence ratios where all fuels are more prone to form soot due to the low combustion efficiency shown in Fig. 3, and considering soot is an incomplete combustion product. This can be attributed to the extremely low aromatic content in the composition of SF1.12.

3.2 Comparison of SF Generations

It is important to understand how the different compositions of the SF generations currently reported compare to the new com-

position proposed in this work in terms of performance and emissions. This subsection presents the results of this comparison, which can be used to guide the production of the bio-blendstock with the optimal composition. All the plots in this subsection have the same format: they show a comparison of parameters for the 0%, 10%, and 30% surrogate fuel blends for all three generations of surrogate fuel, namely SF1, SF1.01, and SF1.12. The leftmost subplot displays the curves for SF1 blends, the middle subplot for SF1.01 blends, and the rightmost subplot for SF1.12 blends. All three subplots also have a curve for the average diesel baseline performance, which is shown as a thick gray line, for comparison purposes as well as the baseline diesel case collected during each particular set of experiments (labeled 0%SFX-100%DF2). The results reported here are of the injection timing sweep.

3.2.1 Engine Load. As mentioned in Sect. 3.1.6, the IMEP is a valuable parameter that can be utilized to measure an engine's capacity to produce work that is independent of engine displacement. Figure 10 presents the $IMEP_g$ as a function of start of injection (SOI). As shown in the figure, the trend for all blends is that the $IMEP_g$ increases at first and reaches a peak value close to the middle of the SOI range and then ultimately drops to the lowest value. This behavior is expected as in the earlier points the combustion experiences knock, while in the latter points the combustion is near misfire limit. Intermediate SOI values produce the highest maximum $IMEP_g$ values because the combustion is optimally phased with the crank-slider mechanism transmission efficiency, thus achieving the highest thermodynamic conversion efficiency of the engine. Figure 10 shows that for SF1 blends, the performance lowers with increasing SF1 blend content. For
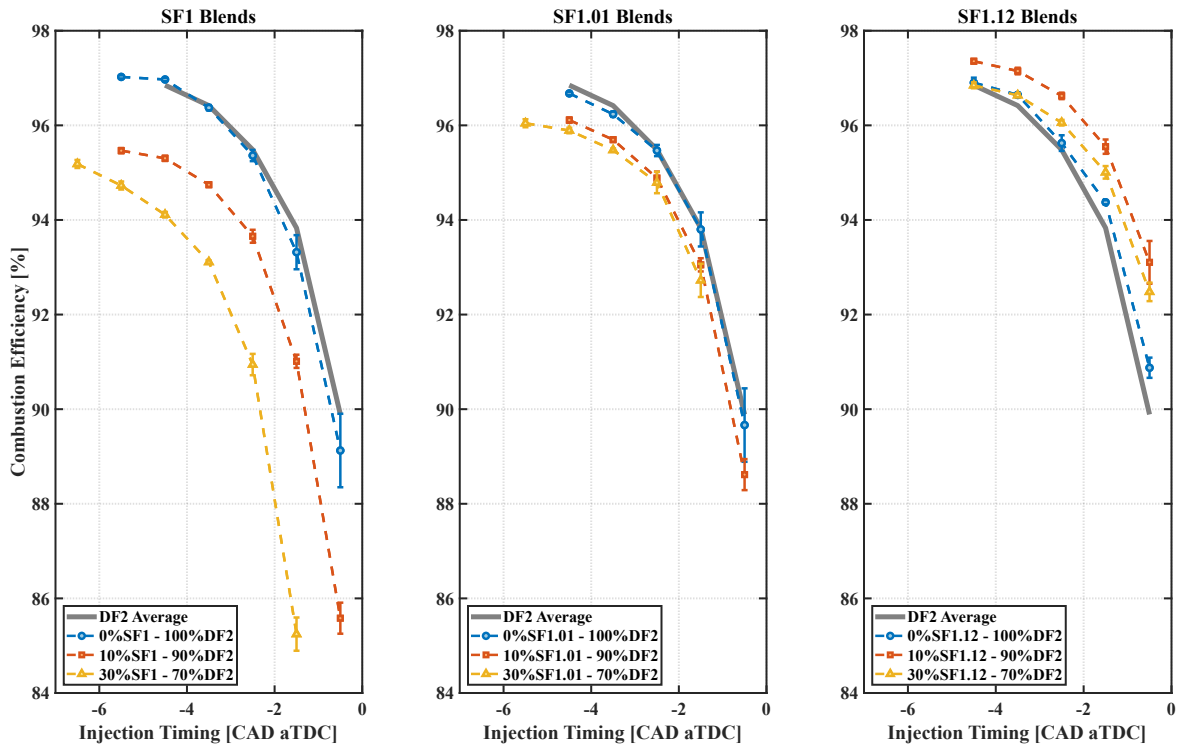
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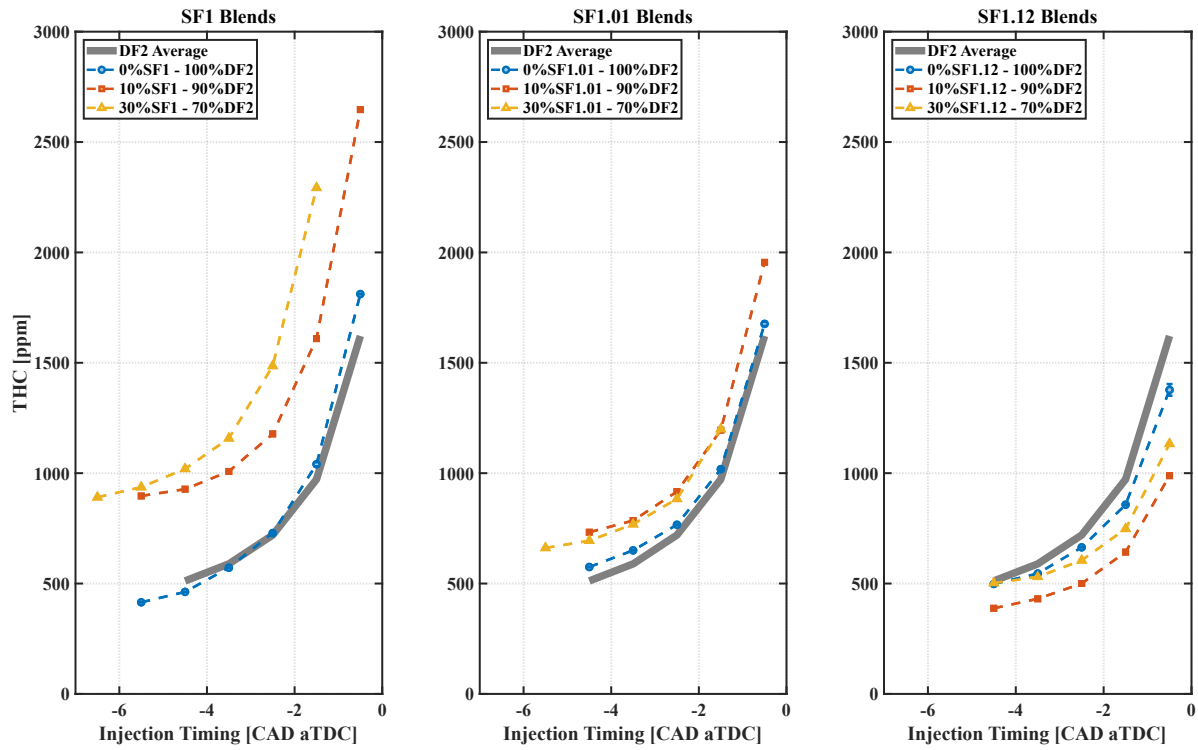**FIGURE 11: COMPARISON OF COMBUSTION EFFICIENCY FOR DIFFERENT SF GENERATIONS**

FIGURE 12: COMPARISON OF THC EMISSIONS FOR DIFFERENT SF GENERATIONS

SF1.01 there is a slight drop in performance going from 0% to 10%, but no significant effects can be observed if the SF1.01 content is further increased. Lastly, for the SF1.12 blends, the loads recorded for the 10% and 30% blends are slightly higher than that of pure diesel, which indicates that adding this surrogate fuel up to 30% by volume has the potential to improve the performance of diesel fuel.

3.2.2 Engine Combustion Efficiency. The engine combustion efficiency is a measure of how much of the fuel input into the system during a cycle was converted into energy. Figure 11 shows a comparison of the combustion efficiency as a function of SOI for all blends of all SF generations. It can be observed that, while the combustion efficiency is lower than that of pure diesel for all blends of both SF1 and SF1.01, it is higher than diesel for nearly all SOIs of all blends of SF1.12. This once again demonstrates that there are clear benefits to adding SF1.12 to diesel in blends of up to 30% by volume.

3.3 Engine Emissions

Minimizing harmful engine emissions is a crucial aspect of engine research. Figure 12 presents a comparison of the THC emissions as a function of SOI for all blends of all SF generations. From the leftmost subplot of the figure it can be stated that SF1 blends increased THC emissions as SF content increased and the lowest emissions were achieved by pure diesel. The similar statement can be made regarding SF1.01 blends shown in the middle subplot, although the behavior is less accentuated compared to the first generation SF. However, in the rightmost subplot both SF1.12 blends (10 and 30%) show a clear reduction in THC emissions compared to all other SF blends and even pure diesel. This is directly related to the combustion efficiency reported in Fig.11 given that THC is an incomplete combustion product, so higher combustion efficiency leads to lower THC emissions. This once more highlights the positive impact of blending SF1.12 in volume concentrations of up to 30%.

4. CONCLUSION

The findings reported in this work can be summarized by the following statements.

- A surrogate fuel was prepared from pure chemical compounds, designed to mimic the characteristics and composition of a high-naphthenic content bio-blendstock derived from woody biomass via the Catalytic Fast Pyrolysis with Hydrotreating pathway.

- The compounds selected to serve as the naphthenic content were Butylcyclohexane and Propylcyclohexane, which are found in significant quantities in the composition of the bio-blendstock.

- The surrogate fuel was blended in 10% and 30% concentrations by volume with No.2 diesel and successfully tested in a compression ignition engine.

- The surrogate fuel was found to have a positive effect on engine performance, having faster burn duration, higher combustion and thermal efficiencies and higher load output for the same fuel-air equivalence ratio.

- The performance improvement observed is also accompanied by a reduction in the CO, THC, and soot emissions, while NOx emissions are shown to remain constant with respect to fuel composition.

- It has been demonstrated that a bio-blendstock with composition compatible with SF1.12 can be blended in concentrations of up to 30% by volume with diesel fuel and lead to an improvement in engine performance as well as decrease in engine-out emissions without quantifiable degradation of fuel system components.

- When compared to previous generations of SF, the formulation of SF1.12 showed improvements in performance and lowered emissions, indicating Butylcyclohexane and Propylcyclohexane are the desirable naphthenic compound representatives for the bio-blendstock.

- A high-naphthenic content bio-blendstock with composition equivalent to SF1.12 was demonstrated to be a promising alternative fuel to be used as a drop in replacement for diesel fuel in up to 30% by volume.

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