Open Access Journal

Journal of Sustainable Research in Engineering Vol. 2 (4) 2015, 139-147

Journal homepage: www.jkuat-sri.com/ojs/index.php/sri/index

An Experimental Investigation on Engine Performance and Emission of a Varied Injection Pressures Compression Ignition Croton-Diesel Blends Engine

Obadiah Maswai[1*], Robert Kiplimo[2] and Hiram Ndiritu[1]

[1] *Department of Mechanical Engineering, Jomo Kenyatta University of Agriculture and Technology, P.O. Box 62000 - 00200 Nairobi, Kenya.*

[2] *Department of Marine Engineering, Jomo Kenyatta University of Agriculture and Technology, P.O. Box 62000 - 00200 Nairobi, Kenya.*

[*] *Corresponding Author - Email: maswaiobadiah@jkuat.ac.ke*

Abstract In this study, a direct injection compression ignition (DICI) engine was modified to utilize croton esters-diesel blends in equal ratio and at varied fuel injection pressures. The main aim of this research is to make use of alternative fuels for diesel engines and to reduce the emission of exhaust gases while minimizing the reduction of brake thermal efficiency associated with the low calorific value of bio-fuel. The engine was tested at varied injection pressures of 220, 240, 260 and 280 bar and at 0%, 10%, 20% and 30% exhaust gas recirculation (EGR) and the performance and emission characteristics of the modified engine were analyzed. The effect of higher injection pressures than the engine's standard value of 220 bar and the varied EGR percentages on performance parameters such as brake thermal efficiency (BTE), brake specific fuel consumption (BSFC) and exhaust gas temperature (EGT) were studied. A thorough analysis on the effect of croton oil ester and diesel blends on smoke, CO, HC and CO_2 emissions is also presented. The performance and emission characteristics were compared among the different injection pressures and exhaust recirculated gas percentages and the results show that brake thermal efficiency increases up to 240 bar before decreasing while the brake specific fuel consumption decreases and then increases at this injection pressure. It was observed that the use of EGR reduces the brake thermal efficiency except for 280 bar where 20% EGR enhanced the engine's efficiency. The EGT and emission parameters were found to decrease at higher injection pressures.

Keywords Croton esters, Injection pressure, Bio-fuel, DICI engine, Performance parameters, Emission characteristics

1. Introduction

The fossil fuel depletion and pollutant emissions from diesel engines have become two major problems of the world today. This is due to the increasing demand of petroleum in automotive, thermal power generation and manufacturing sectors of economy worldwide [1], [2]. In Kenya, for example, there is an increase of automobiles imported every year as it can be witnessed in our roads and also thermal power investment to meet the demand of the growing population. Consequently, fossil fuel reservoirs nears depletion in oil rich countries like United Arab Emirates (UAE) (estimated to be depleted by 2016) and also the risk of harmful global warming due to air pollution associated with these fuels. Stationary engines use heavier fuels thereby producing higher emissions but at the same time they can be operated on alternative fuels more comfortably compared to automobiles. Researchers have sought viable alternative fuels either as straight form fuel or blends in engine operation. Other approaches have included vegetable oils converted to bio-fuel in straight form or as fuel blends which have also shown interesting results [3]. The use of straight diesel in running diesel engines is associated with high particulate emissions. On the other hand, using biodiesel solves the problem of particulate emissions but emissions of nitrogen oxides rises significantly due to oxygen content present in the fuel. Biodiesel causes an increase in fuel consumption proportional to its lower net heating value (NHV), but the engine efficiency does not significantly decrease [4].

Clean energy alone cannot meet the power demand for domestic and manufacturing industry. Thermal energy has been a key source in supplying additional megaWatts to the national grid. Fossil fuels therefore, are mainly consumed in Kenya by the stationary diesel engines generating this thermal power [5]. The Kenyan Government in drafting vision 2030 identified energy as a key foundation and one of the enablers upon which the economic, social and political pillars of the long term strategy is built [6]. The fuel cost as reflected in electricity bills in Kenya stands at 40% and this can be reduced by blending diesel with biodiesel for running power generators. Fossil based thermal power contributes 32.5% to the national grid and the peak load is projected to grow to about 2,511 MW by 2015 and 15,026 MW by 2030 [7].

Bio-fuels are viable alternatives for running these power generating engines despite the accumulation of carbon deposits inside the engine cylinder after long use [8]. However, esterification process converts vegetable oil to esters which has properties comparable to that of fossil fuels hence minimizing the problems associated with straight vegetable oil. Most of the automobile engines have been designed to run on light fuel (petrol) but bigger vehicles like lorries and tractors uses heavy fuel (diesel) and thus they can be operated on biodiesel-diesel blends. This will reduce emissions associated with straight diesel (100% diesel) but higher blend levels such as B50 (50% biodiesel-50% diesel) require special handling and may require equipment modification [9]. The challenge of biodiesel having higher viscosity compared to straight diesel can be partly solved by designing a heat exchanger that uses the heat generated at the exhaust pipe. This will raise the temperature of the fuel and consequently reducing viscosity before its injection into the combustion chamber [10]. Another approach is by increasing the injection pressures so as to atomize the fuel into fine form during injection into the combustion chamber and consequently improving engine performance and reducing emissions [11]–[14].

The use of straight diesel in direct injection compression ignition (DICI) engines emits harmful exhaust gases such as smoke, carbon monoxide, hydrocarbon and carbon dioxide. Studies conducted over the last decade have well established a direct relationship between deteriorating human health and diesel engine exhaust. Biodiesel has shown a lot of promise in terms of both its relatively higher combustion efficiency and lower harmful emissions [15]. Releasing these harmful emissions to the environment causes respiratory diseases and discomfort to the asthmatic leading to loss of income in households due to medical expenditures and reduced productivity. Carbon dioxide is the main greenhouse gas that leads to global warming and subsequently unfavourable climate changes. It therefore necessitates the use of croton oil which releases lesser amount of these emissions compared to fossil diesel.

The anticipated depletion of fossil fuels demands for the use of alternative fuels such as croton oil which is readily available in Kenya. This will lead to the combined benefit of utilization of lower price alternative fuels and farmers benefiting from their produce, therefore modification of a DICI engine to utilize this biodiesel is essential. This comes in handy with an ongoing study that assesses the business case for croton in Kenya and examine potential livelihood benefits for growers as it has been in jatropha growing countries [16], [17].

Croton oil cannot be utilized as straight diesel oil in engines. This is because of high viscosity and low energy density that affect the performance of engines. Moreover, the high viscosity causes emission of particulate in form of thick smoke. This is what discourages the use of unblended croton oil and other vegetable oils in internal combustion engines. It would be necessary to not only blend the oil with straight diesel but also vary combustion parameters such as injection pressure and timing in order to minimize emissions from their use. This demands a modification of the engine to accommodate the use of the blend.

2. Equipment and Experimental Procedure

2.1. The Experimental Setup

The engine used for this study is a single cylinder, four stroke direct injection compression ignition (DICI) engine. It is a water cooled, naturally aspirated engine with its specification shown in Table1. Loading was applied by regulating the water supply to the rotor blades of the hydraulic dynamometer that was coupled to the engine through adjustment of inlet and outlet water valves. The loading conditions considered for the engine testing were at 25% (quarter load), 50% (half load), 75% (three-quarter load) and 100% (full load). The engine was modified to run on a proportion of C50 blend (50% croton oil esters-50% diesel) and at increased injection pressures by replacing the injection system integral to the engine with an electronic injection system via a pressure gauge. Fuel was supplied to the independent electronically controlled high pressure fuel pump from a blending stand adjacent to the engine with 2 fuel tanks, one for diesel and another for the blend so as to allow the switching of fuel from diesel to C50 without stopping the engine. A variable-frequency drive was used for controlling the high pressure fuel pump and an arduino microcontroller controlled the electronic injector and the pump. The experimental apparatus consisted of the test engine, variable-frequency drive, electronically controlled high pressure fuel pump, microcontroller, dynamometer, emission analyzers, exhaust gas temperature measurement system, analogue pressure gauge and the fuel tanks stand as shown in Fig. 1.

2.2. Physio-Chemical Properties of Diesel and C50

The crude croton oil was converted into esters (biodiesel) through esterification process before blending in equal ratio with fossil diesel. The C50 (50% croton oil esters-50% diesel) was taken to Kenya Industrial Research and

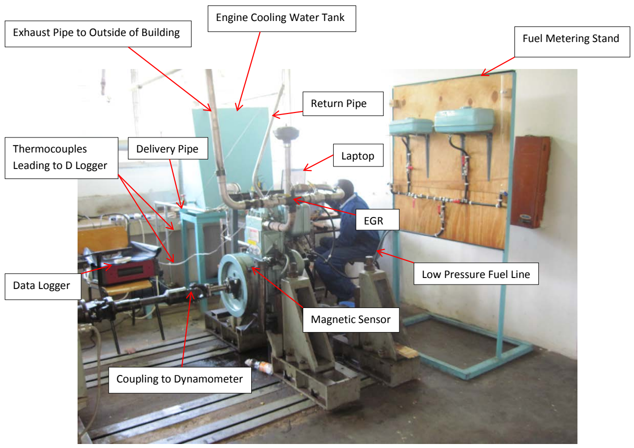

Fig. 1. Pictorial view of the experimental setup

Table 1. Engine specifications

	Specification	Indicated
1	Number of cylinders	1
2	Number of strokes	4
3	Type	Vertical
4	Model	SI-100H
5	Speed (rpm)	1500
6	Rated Power (kW)	7.5
7	Compression Ratio	16.5:1
8	Specific Fuel Consumption (g/kWh)	250
9	Injection Timing (^{0}bTDC)	27
10	Injection Pressure (bar)	220
11	Mode of Injection	Direct Injection
12	Cylinder Capacity (cc)	950

Development Institute (KIRDI) to ascertain its physical and chemical properties as shown in Table2.

Table 2. Physio-chemical properties of diesel and C50

	Diesel	C50
Kinematic Viscosity at 40^{0}C (mm^2/s)	2.98	5.82
Flash Point (^{0}C)	62	85
Specific Gravity at 20^{0}C	0.8403	0.8717
(LHV) Calorific Value (kJ/kg)	43940	40743

2.3. Experimental Procedure

In testing the influence of higher injection pressures on the performance and emissions characteristics of the engine, the C50 proportion of blend was tested at injection pressures of 220, 240, 260 and 280 bar. The engine was first run on straight diesel and then the blending proportion of C50 was introduced with each test procedure done for the different injection pressures at a time.

During the experiment, a stop watch was used to measure the time in seconds it took the engine to consume 10 milliliters (ml) of fuel for every set of readings on the burette. The stop watch was a CASIO model capable of measuring up to milliseconds (ms). In avoiding parallax errors when using the burette, readings were viewed at the bottom of the fuel's meniscus. The relationship between the time and the amount of fuel consumed yields specific fuel consumption (SFC) for the engine, whereby the SFC of the engine was determined in kilograms per hour.

The hydraulic dynamometer manual indicated the procedure for loading the engine using the water inlet and outlet valves. For *quarter* load condition, the water outlet valve was opened upto 50% while the inlet valve was opened upto 25%. During *half* loading, the water outlet valve was opened upto 50% while the inlet valve was opened upto 50% and at *full* load condition, the water outlet valve was opened upto 25% while the inlet valve was 100% opened. The *three-quarter* (75%) load of the engine was calculated as the ratio between brake power and maximum brake power at the same engine speed.

For every set of fuel consumption readings, the speed of the engine in revolutions per minute (rpm) was taken at the flywheel using a digital tachometer. At the same time the dynamometer readings in kilograms (kg), data logger readings in degrees Celsius (^{0}C) and emissions analyzer readings in percentage volume (% Vol.) and parts per million (ppm) were also recorded. This was done repeatedly 3 times for *quarter*, *half*, *three-quarter* and *full* load with an allowance of 5 minutes before taking the readings after change of load. The reason for taking 3 sets of data in every engine loading condition was to improve the accuracy of results. The 5 minutes allowance was for ensuring that the engine attained a steady state condition after variation of the load.

The recirculated exhaust gas (EGR) percentages were achieved by turning the ball valve that returned the exhaust gas to the inlet air manifold of the engine proportionately according to the total number of turns of the valve handle. Exhaust gas recirculation (EGR) was then introduced starting with 10% EGR for *quarter*, *half*, *three-quarter* and *full* load before increasing to 20% and finally to 30% EGR for all loads and data recorded.

2.4. Performance Evaluation of the Engine

2.4.1. Brake Thermal Efficiency

Brake thermal efficiency (BTE) is defined as brake power of a heat engine as a function of the thermal input from the fuel. It is used to evaluate how well an engine converts the heat from a fuel to mechanical energy. It is the ratio of the fuel energy changed into mechanical power by the engine to the energy of the fuel supplied to the engine.

$$BTE = \frac{brake\ power}{fuel\ power} = \frac{3600\ \times\ P}{SFC\ \times\ LHV}, \qquad (1)$$

where P is the brake power in kW, SFC is the specific fuel consumption in kg/h and LHV is the lower heating value of the fuel in kJ/kg. The specific fuel consumption in kg/h was calculated by multiplying the fuel consumption expressed in L/h with the fuel density expressed in kg/L. Brake power P was determined as follows: P = M $\times$ ω (W), where M is the torque in Nm and ω is the angular velocity in rad/s. Using:

$$\omega = \frac{2 \times \pi \times n}{60}, \tag{2}$$

where n is the engine speed in revs/min (rpm). Upon dividing by 1000 to obtain brake power in kW,

$$P = \frac{M \times n}{9549}(kW). \tag{3}$$

Torque (M) was calculated using the formula: M = S × g × L, where S is the weight indicated by the scale, g = 9.81 m/s^2 and L is the length of the arm of the scale.

2.4.2. *Brake Specific Fuel Consumption*

Brake specific fuel consumption (BSFC) is a measure of the fuel efficiency of an internal combustion engine. It is the ratio of fuel consumption rate in kilograms per hour (kg/h) to the brake power produced in kilo-Watts (kW). The brake specific fuel consumption (BSFC) of the engine was obtained using the expression below:

$$BSFC = \frac{SFC}{P}, \tag{4}$$

where SFC is the specific fuel consumption in kg/h and P is the brake power in kW. From capacity and quantity of liquids, density of diesel is reported as 0.832 kg/l [18]. Therefore, 0.01 l of diesel which was the predetermined amount of fuel for every run during the experiment was equivalent to 0.00832 kg. The value of SFC for every reading was determined through dividing this value by the stop watch reading.

3. Results and Discussion

3.1. Brake Thermal Efficiency

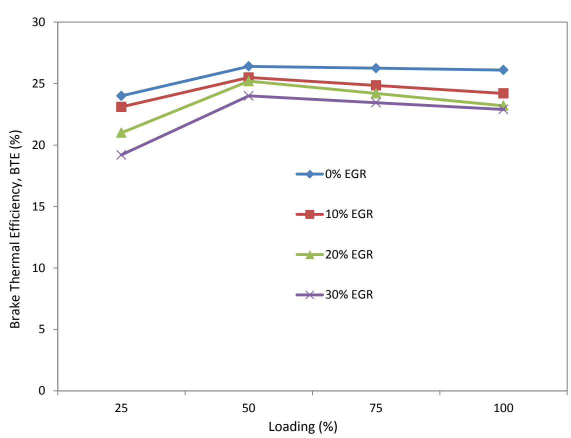

Fig. 2. Variation of brake thermal efficiency with engine load at 220 bar

From Fig. 2, there was an increase in brake thermal efficiency with engine loading up to half load before decreasing again towards full load for 0% exhaust gas recirculation (EGR). As the EGR amount was increased from 0 to 30%, there was a decrease in brake thermal efficiency at all loads by an average of 7%. The results show that the recirculated exhaust gases had the greatest negative impact during low loading condition. It was attributed to the insufficient oxygen due to the displacement of incoming air by exhaust gases resulting to incomplete combustion of fuel. Moreover, at lower injection pressures, there is poor atomization and mixture formation of bio-fuel during injection [19].

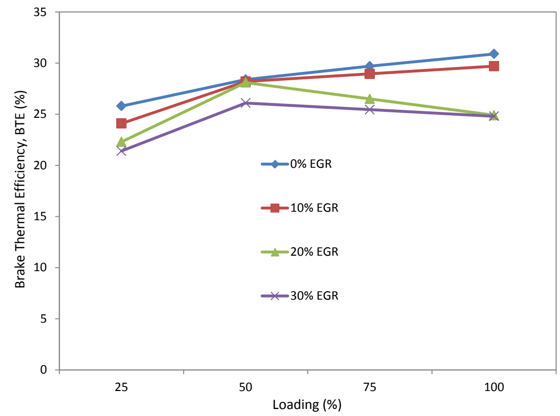

Fig. 3. Variation of brake thermal efficiency with engine load at 240 bar

In Fig. 3, brake thermal efficiency increased linearly with load when no exhaust gases were recirculated. An increase of exhaust recirculated gases up to 30% resulted in a reduction of BTE by an average of 7% for all loads. This was with the exception of when EGR was raised from 10 to 20% at full load in which 16% reduction was recorded. At this injection pressure of 240 bar, it was observed that brake thermal efficiency improved at increased EGR rates. This improvement was due to better fuel spray that resulted in improved combustion of the fuel even at oxygen-deprived in-cylinder conditions.

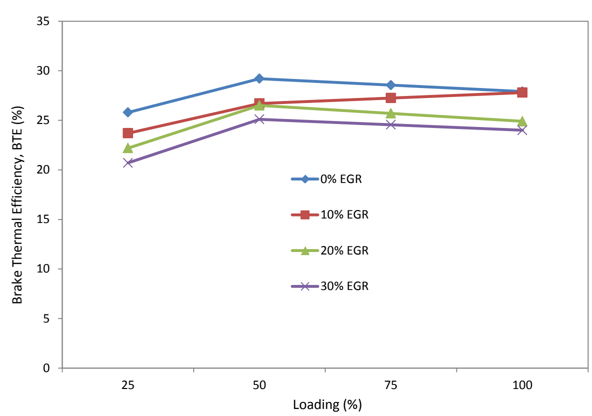

Fig. 4. Variation of brake thermal efficiency with engine load at 260 bar

In Fig. 4, brake thermal efficiency increased with loading up to half load before decreasing at no exhaust gas recirculation. Raising the exhaust recirculated gases to 10% reduced the brake thermal efficiency by 8% at lower loads and 2% at higher loading. Further increase in recirculated exhaust gases reduced the brake thermal efficiency with an average of 3%. The impact of 30% EGR on brake thermal efficiency was on a reducing percentage average of 15% across all loads. The minimal reduction in BTE even with the recirculation of exhaust gases is attributed to reduced viscosity, improved atomization and better combustion at the enhanced injection pressure of 260 bar. At higher injection pressures the size of fuel droplets decreases resulting to very high fine spray hence penetration and momentum will be reduced during injection [20].

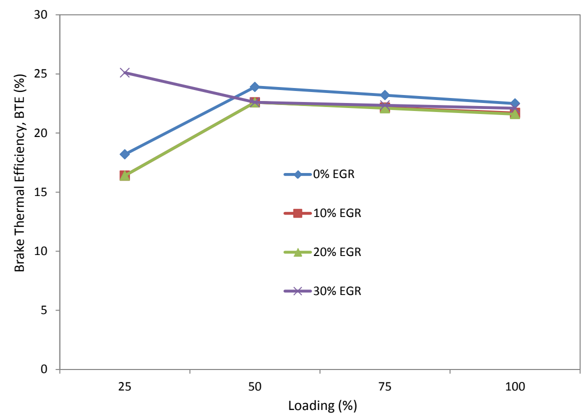

Fig. 5. Variation of brake thermal efficiency with engine load at 280 bar

Increased load raised the brake thermal efficiency by almost 30% at 0% EGR as shown in Fig. 5. This is attributed to improved burning of the fuel due to rising in-cylinder temperatures. Increased recirculated exhaust gases up to 10% reduced the brake thermal efficiency by 5%. An increase in exhaust gas recirculation from 10 to 30% raised the thermal efficiency by the same margin. Further increase in exhaust recirculated gases had little impact on brake thermal efficiency except for 30% EGR which resulted in 35% remarkable rise in BTE at quarter load. This is attributed to enhanced injection pressure that improved the fuel spray. Also the recirculation of already heated air aided combustion and consequently the brake power.

3.2. Brake Specific Fuel Consumption (BSFC)

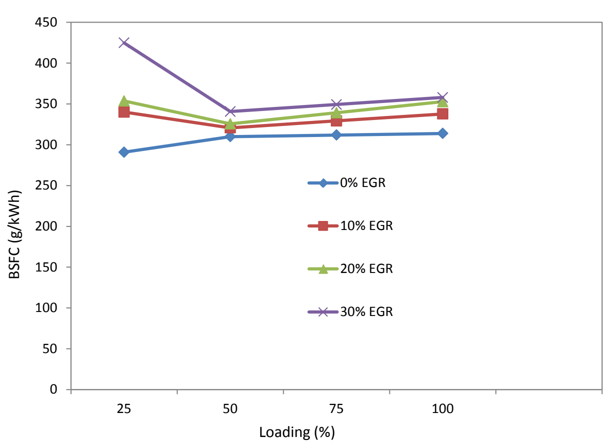

Fig. 6. Variation of brake specific fuel consumption with engine load at 220 bar

Increased exhaust recirculated gases raised the brake specific fuel consumption by almost 5% at medium and high loads as shown in Fig. 6. At low load however, the increase was about 20% when exhaust recirculated gases were raised from 0 to 10% and from 20 to 30%. The significantly high increase in brake specific fuel consumption at low load is attributed to the low injection pressure of 220 bar coupled with choked combustion due to the recirculated exhaust gases.

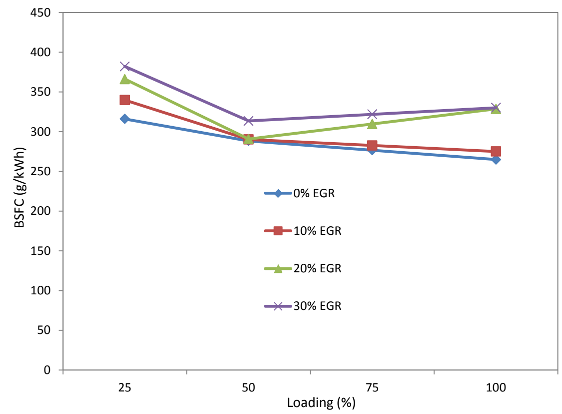

Fig. 7. Variation of brake specific fuel consumption with engine load at 240 bar

From Fig. 7, brake specific fuel consumption increased by equal rate of about 5% at low and medium loads with the rise in recirculated exhaust gases. This low percentage increment is attributed to the improved fuel consumption efficiency as a result of enhanced spray characteristics of the fuel. During high loading condition, increased exhaust recirculated gases from 10 to 20% raised the brake specific fuel consumption by almost 20%. Further increase in the exhaust gases recirculation however had insignificant impact on the brake specific fuel consumption. The stability in the rate of fuel consumed by the engine even with further exhaust gas recirculation is due to complete combustion of the fuel. As injection pressure increases the penetration length and spray cone also increases, so that at optimum pressure, air-fuel mixing and spray optimization will be improved [21], [22].

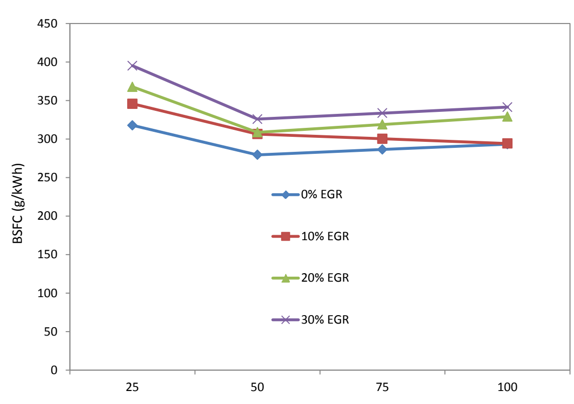

Fig. 8. Variation of brake specific fuel consumption with engine load at 260 bar

Brake specific fuel consumption increased by about 8% with raised exhaust recirculated gases as shown in Fig. 8. Increased loading however decreased the BSFC up to medium load before gradually rising again towards full load. A finer spray pattern of the fuel at the injector results in a better in-cylinder combustion and hence less fuel consumption per unit brake power output. At the injection pressure of 260 bar, 30% exhaust recirculated gases increased the brake specific fuel consumption by an average of 15% across all loads. Too fine fuel spray pattern becomes weaker as it vaporizes away from the intended engine in-cylinder point of effective fuel combustion. This enhanced injection pressure coupled with oxygen-deficient recirculated gases may have deteriorated the efficacy of fuel consumption.

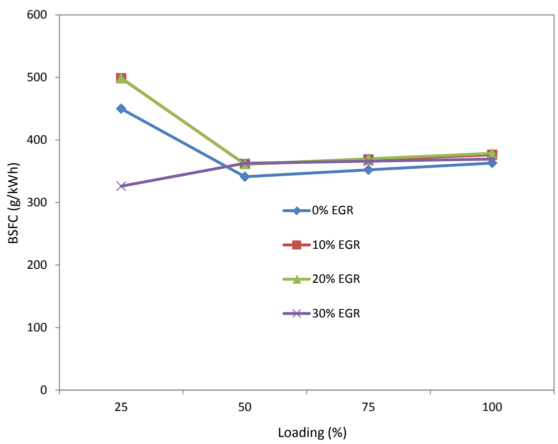

Fig. 9. Variation of brake specific fuel consumption with engine load at 280 bar

In Fig. 9, the highest percentage increase in the brake specific fuel consumption was 10% at quarter load as the exhaust recirculated gases were raised from 0 to 10%. Medium and high loads were marginally affected by the recirculated exhaust gases. However, raised EGR from 20 to 30% resulted in a tremendous decrease in the brake specific fuel consumption by almost 35%. Similar findings have been encountered in the previous studies by Ankit A. et al. [23] who observed that increased injection pressures gives better results of brake specific fuel consumption compared to the original injection pressure.

3.3. Exhaust Gas Temperature (EGT)

From Fig. 10, exhaust gas temperature increased by 2% with increase in percentage EGR up to 20% before reducing by 1%, at quarter load. Beyond this exhaust gas recirculation percentage, there was an insignificant EGT change. As for medium loads, the reduction in exhaust gas temperature up to 20% EGR was 1%. At full load, the percentage increase in EGT beyond 20% exhaust gas recirculation was 2%. The drop in EGT at part load could be attributed to reduced energy input as a result of less fuel consumption by the engine at that condition.

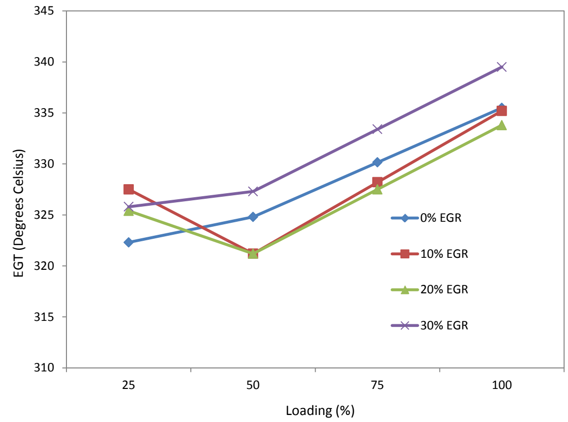

Fig. 10. Variation of exhaust gas temperature with engine load at 220 bar

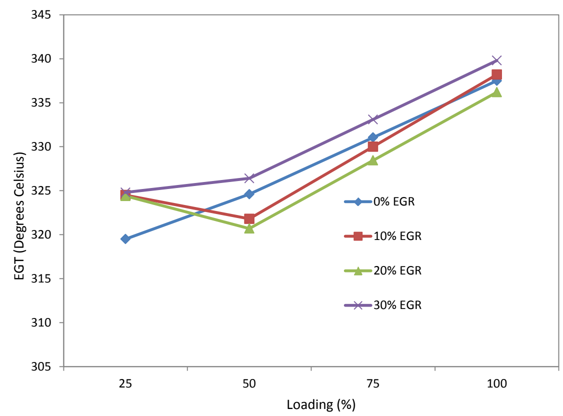

Fig. 11. Variation of exhaust gas temperature with engine load at 240 bar

Fig. 11 indicates the variation of exhaust gas temperature at various EGR engine operating mode. It was observed that EGT at quarter load increased by 2% as the EGR was increased up to 10% beyond which there was no change. As for part load, the EGT reduced by an average of 1% up to 20% EGR before increasing by 2%. The only significant change in EGT at full load was an increase by 1% as the exhaust gases were increased beyond 20% EGR. The higher trend of EGT values may be due to the inefficiency of the engine in converting the chemical energy in the fuel blend into mechanical power output.

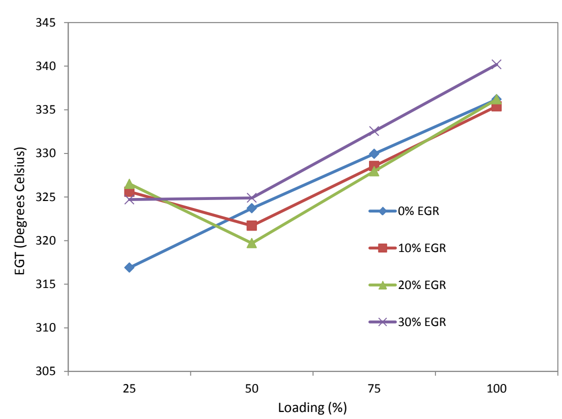

Fig. 12. Variation of exhaust gas temperature with engine load at 260 bar

From Fig. 12, the increase in exhaust gas temperature without exhaust gas recirculation was found to be linear while the rest of EGR percentages showed a drop in EGT up to half load. With further increase in load, 30% EGR was found to have recorded the highest exhaust gas temperature throughout. The maximum increase in EGT was found to be 3% at quarter load when exhaust gas recirculation percentage was increased to 10%. The increased exhaust gas temperature as the EGR ratio was raised could be attributed to the recirculated already hot gases.

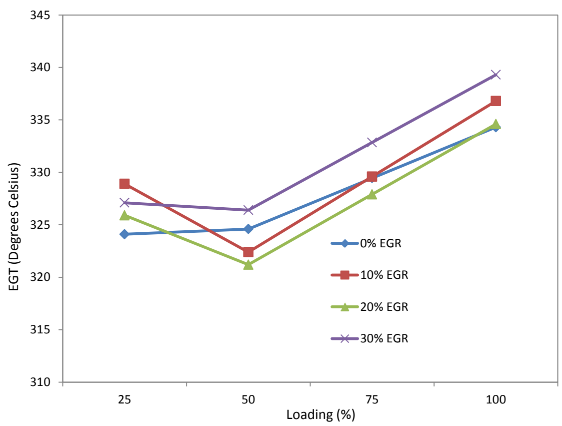

Fig. 13. Variation of exhaust gas temperature with engine load at 280 bar

The variation of EGT with load is shown in Fig. 13. It was observed that 10% EGR recorded comparatively higher EGT values than for 0% and 20% at all loads with the exception of half loading at no exhaust gas recirculation. 30% EGR also recorded the highest EGT at all loads except for quarter load for 10% exhaust gas recirculation. Moreover, the percentage difference in EGT for this EGR ratio was an average of 2%.

3.4. Exhaust Emissions

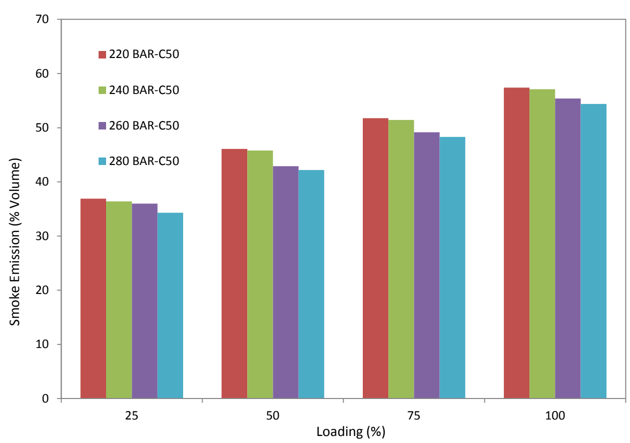

Fig. 14. Variation of Smoke with Load

Variation of smoke emission with engine load is shown in Fig. 14. It was observed that the engine emitted more smoke at the injection pressure of 220 bar and at all loading conditions. However as the injection pressure increased the percentage of emission decreased with the lowest recorded at 280 bar. The maximum reduction in smoke was at an average of 7%.

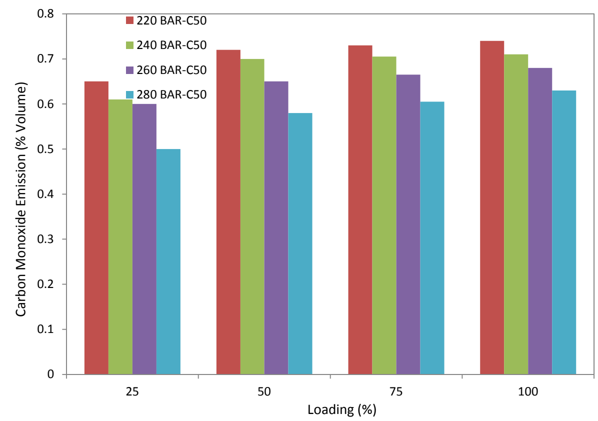

Fig. 15. Variation of CO with Load

The CO emission for different injection pressures is indicated in Fig. 15. It was observed that the CO emission decreased with the increase in injection pressure by an average of 5%, 220 bar recording the highest and 280 bar the lowest at all loads. It was attributed to proper mixing of fuel particles and effective combustion at increased injection pressures.

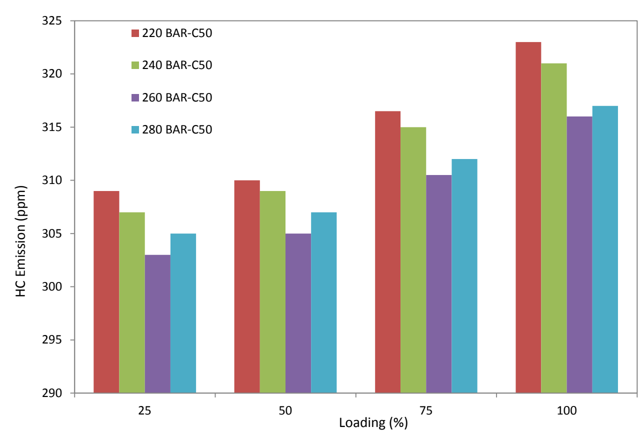

Fig. 16. Variation of HC with Load

The variation of HC emission with engine load is shown in Fig. 16. It was observed that the HC emission increased with increase in load for all the injection pressures. The values for 260 bar were comparatively lower at all loads. HC emission is a result of incomplete combustion but as the injection pressure increase, the emission is reduced. At injection pressures above 260 bar, there was an increase in HC emission and it was because of finer fuel spray which resulted in momentum and penetration of the droplets causing an incomplete combustion.

Fig. 17 shows the emission levels of CO_2 for various injection pressures. The rising trend of CO_2 emission with load is due to the higher fuel intake as the load increases. Test measurements revealed that the CO_2 emission for

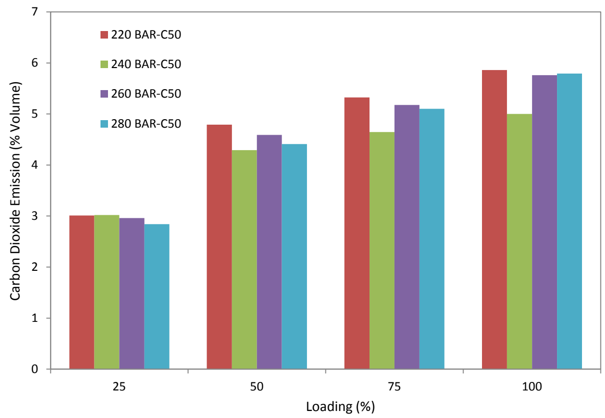

Fig. 17. Variation of CO_2 with Load

280 bar was the lowest at quarter load but 240 bar showed reduced emission as the load increased further. The maximum reduction in CO_2 emission at quarter and high loads were 6% and 10% respectively.

4. Conclusions

The study was to investigate the effects of injection pressures on performance and emission characteristics of the chosen proportion of equal ratio of croton oil esters-diesel blend. This was done using a single cylinder four-stroke DI diesel engine under various experimental conditions. The following conclusions were drawn from the analysis:

1) The brake thermal efficiency was the highest for the injection pressure of 240 bars at full load without exhaust gas recirculation. The lowest brake thermal efficiency was at the injection pressure of 280 bar and at low loading conditions with recirculated exhaust gases at 10 and 20 percent. It may be attributed to the improved fuel spray characteristics at the fuel injector with increased injection pressure though it deteriorated when it exceeded 240 bar. Increase in fuel injection pressures for higher proportion of blends is effective in improving engine performance at higher loads and at a higher EGR rates.
2) The test results showed higher brake specific fuel consumption (BSFC) of the engine at low load and comparatively much lower BSFC at higher engine loads for all injection pressures.
3) Exhaust gas temperature (EGT) increased with the increase in exhaust gas recirculation except for 20% EGR which showed relatively lower values compared to other percentages, this was the case for all injection pressures. Part load had a drop in EGT before rising again at full load.
4) The smoke emissions increased with the increase in load with the highest at 220 bar at full load and the lowest at 280 bar at low load. The highest emissions of CO, HC and CO_2 also occurred at 220 bars at full loading condition and the lowest at an injection pressure of 280 bar and at low load except for HC which occurred at 260 bar.

Acknowledgment

The authors would like to thank Jomo Kenyatta University of Agric. & Tech. (JKUAT) for funding this research and providing facilities for conducting the experiment. The authors are also grateful to Japan International Corporation Agency (JICA) for their support in providing the diesel engine, electronically controlled fuel injection pump and the electronic fuel injector used in this research.

Nomenclature

DICI - Direct Injection Compression Ignition
EGR - Exhaust Gas Recirculation
BTE - Brake Thermal Efficiency
BSFC - Brake Specific Fuel Consumption
EGT - Exhaust Gas Temperature
CO - Carbon Monoxide
HC - Hydrocarbon
CO_2 - Carbon Dioxide
DI - Direct Injection
NHV - Net Heating Value
bTDC - before Top Dead Center
LHV - Low Heating Value
SFC - Specific Fuel Consumption

References

[1] Swarup K. N., Bhabani P. P., “Experimental investigation on performance and emission characteristics of a diesel engine fuelled with mahua biodiesel using additive.,” *Energy Procedia*, vol. 3, pp. 569–579, 2014.
[2] Meshack H., Robert K., Hiram N., “Effect of exhaust gas recirculation on performance and emission characteristics of a diesel-piloted biogas engine,” *Smart Grid and Renewable Energy*, vol. 6, pp. 49–58, 2015.
[3] Barabas I., Todorut A., Baldean D., “Performance and emission characteristics of a ci engine fueled with diesel-biodiesel-bioethanol blends.,” *Fuel*, vol. 89, pp. 3827–3832, 2010.
[4] Qi D., Chen H., Geng L., Bian Y., “Experimental studies on the combustion characteristics and performance of a direct injection engine fueled with biodiesel/diesel blend.,” *Energy Conversion and Management*, vol. 51, pp. 2985–2992, 2010.
[5] Kanini M., “Energy efficiency from the kenya power perspective.,” *Africa Carbon Exchange (ACX) Energy Efficiency Workshop*, 2013.
[6] Government of Kenya, “Implementation of the first medium term plan of kenya vision 2030.,” *Annual Progress Report*, 2011.
[7] Stanley Hotel, “Energy in kenya.,” *Energy Scenarios Workshop*, 2013.
[8] Timothy P. G., “Characterization of engine performance with biodiesel fuels.,” *MSc*, 2008.
[9] National Renewable Energy Laboratory, “Biodiesel handling and use guide.,” *Innovation for Our Energy Future*, 2009.
[10] Dinesha P., Mohanan P., “Experimental investigations on the performance and emission characteristics of diesel engine using

preheated pongamia methyl ester as fuel.," *Energy Scenarios Workshop*, vol. 5, pp. 591–600, 2012.

[11] Keerthi K., Kiran C. K., Sravya S., "Performance characteristics of four stroke single cylinder diesel engine with 10% iso butanol at different injection pressures.," *International Journal of Modern Engineering Research*, vol. 3, pp. 311–316, 2013.

[12] Deva K. M. L. S., Drakshayani S., Vijaya K. R. K., "Effect of fuel injection pressure on performance of single cylinder diesel engine at different intake manifold inclinations.," *International Journal of Engineering and Innovative Technology*, vol. 2, 2012.

[13] Subba R. C. V., Eswara R. C., Hemachandra R. K., "Effect of fuel injection pressures on the performance and emission characteristics of di diesel engine with biodiesel blends cotton seed oil methyl ester," *International Journal of Research and Reviews in Applied Sciences*, vol. 2, 2012.

[14] Mahalingam S., Suresh M. K. P., Pranesh R. V., "Experimental study of performance and emission characteristics of a bio dual fuel blends in diesel engine for variation of injection pressures.," *Proceedings of the World Congress on Engineering*, vol. 1, 2013.

[15] Avinash K. A., Tarun G., "Particulate emissions from biodiesel vs diesel fuelled compression ignition engine.," *Renewable and Sustainable Energy*, 2011.

[16] Michael J., "An intriguing bioenergy possibility for east africa.," *Renewable and Alternative Energy*, 2015.

[17] Yuka T., Brent S., "Jatropha curcas biodiesel production in kenya.," *Economics and Potential Value Chain Development for Smallholder Farmers*, vol. 54, 2007.

[18] Fraunhofer, "Dokumentation transport processes," *Transport*, vol. 04, 2012.

[19] Purushotham N. D. S., Sreekantha A., "Effect injection parameters on ci engines performance and emissions using sesame and pongamia pinnata methyl ester as fuels.," *Jordan Journal of Mechanical and Industrial Engineering*, vol. 2, 2014.

[20] Mohd Y. A., Mehd S. N., Reddy P. R., "Performance and emission characteristics of a diesel engine with cottonseed oil plus diesel and blend.," *International Journal of Engineering*, vol. 18, 2005.

[21] Naga S. S., Shailaja M., Sita Rama Raju A. V., Kalyani R. K., "Optimization of injection pressure for a compression ignition engine with cotton seed oil as an alternative oil.," *International Journal of Engineering, Science and Technology*, 2012.

[22] Nagaraja A. M., Prabhukumar G. P., "Effect of injection pressure on the engine performance with ricebran oil as biodiesel.," *XVIIINCICEC*, pp. 581–587, 2003.

[23] Ankit A., Soni S. L., Dinesh K. S., Sandeep K. B., "Effect of variation of compression ratio and injection pressure on performance and emission characteristics of ci engine using various alternative fuels: a review.," *International Journal of Research in Engineering and Technology*, vol. 04, 2015.