Journal Pre-proof

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PII: S2468-2276(20)30028-4
DOI: https://doi.org/10.1016/j.sciaf.2020.e00290
Reference: SCIAF 290

To appear in: *Scientific African*

Received date: 4 November 2019
Revised date: 19 December 2019
Accepted date: 22 January 2020

Please cite this article as: Sayon SIDIBE , Joel BLIN , Tizane DAHO , Gilles VAITILINGOM , Jean KOULIDIATI , Comparative study of three ways of using Jatropha curcas vegetable oil in a direct injection diesel engine, *Scientific African* (2020), doi: https://doi.org/10.1016/j.sciaf.2020.e00290

Comparative study of three ways of using *Jatropha curcas* vegetable oil in a direct injection diesel engine

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Abstract

Although vegetable oils are apparently an advantageous alternative fuel for direct use in traditional diesel engines with no modification necessary, in practice many problems are regularly discussed in the literature including filter clogging, breakage of certain types of injection pumps, and deposits of carbon on the cold parts of engines.

Several technological solutions have been proposed to overcome these problems but the majority of papers discuss them individually and have not actually compared them in similar conditions. The purpose of the present study was to use the same experimental device to compare the three most widely recognised technological options for the use of *Jatropha curcas* vegetable oil as a fuel in a direct injection diesel engine: preheating, blending with diesel, and recirculating exhaust gases. Power output, specific consumption, thermal efficiency and exhaust gas emissions were compared to those of diesel used as the reference. The results obtained were similar for preheated and non-preheated Jatropha oil, but differed from the results obtained with diesel. Similar combustion performance and similar emissions were obtained with a blend of 20% Jatropha oil and diesel to those obtained with diesel alone. Exhaust gas recirculation (EGR) with Jatropha oil could lead to fouling in the combustion chamber. In contrast to widely accepted theory, this study also clearly demonstrates that the viscosity of vegetable oil is not the main cause of poorer combustion quality and, consequently, of deposits in the combustion chamber.

Keywords: Jatropha *curcas*, vegetable oil, biofuel, direct injection diesel engine

1. Introduction

Increasing consideration is being given to renewable energy to gain multiple co-benefits including cost savings, access to modern energy systems, treatment of organic wastes, improved human health, local employment opportunities, social cohesion of communities, improved livelihoods, sustainable development, as well as reduced GHG emissions [1–4] .

The economies of most developing countries rely on agriculture whose development involves mechanization. Upstream and downstream efforts towards mechanization of the agricultural sector are making very little headway in developing countries today due to the high cost of liquid fossil fuels[5,6], for engines. This is even more apparent in land-locked countries with no liquid fossil fuel resources like Mali and Burkina Faso in West Africa. Petroleum products are very expensive in the rural areas of both countries, where purchasing power is very low [7]. On the other hand, such countries generally have a potential for the production of biofuel, which, if successful, could substantially advance agricultural mechanization. One possible solution for agricultural mechanization in such countries is the use of pure vegetable oil, produced and consumed on the farm. This option would make it possible to produce the fuel needed to develop, diversify and intensify agricultural production itself, with a very worthwhile ratio: agriculture could produce five to ten times the amount of energy it consumes [8].

Vegetable oils can be used in both indirect injection and direct injection diesel engines, although the latter are less tolerant of vegetable oils [9–11]. Direct injection diesel engines are mostly used for transport, the production of electricity, and for motive power because they are more efficient than other engines [12]. This type of engine is found in isolated rural zones, where fuel supplies are difficult.

While vegetable oils are an advantageous alternative for specific applications, including agricultural mechanization or the production of electricity, their direct use in traditional diesel engines without any modification, can lead to operating problems. Vegetable oils are 10 to 20 times more viscous than pure diesel, are less volatile, and their chemical nature differs from that of diesel[5,9,13–15] .

The problems resulting from the use of vegetable oil are regularly described in the literature; they include filter clogging, breakage of certain types of injection pumps, and deposits of carbon on the cold parts of engines (the combustion chamber wall, cylinder head, injector nozzles, etc.)[9,16]. Several technological solutions have been proposed for the use of pure vegetable oil with no chemical modification such as transesterification or catalytic cracking of the oil. However, the cause of the problems and the recommended solutions are still the subject of debate. The most frequently proposed solutions are:

- preheating the vegetable oils to reduce their viscosity to the equivalent of diesel [16–22];

- blending vegetable oil and diesel oil in different proportions to reduce viscosity, but also to benefit from what is assumed to be better combustion of the oil thanks to the diesel [9,18–26];
- recirculating exhaust gas (EGR) in the engine to benefit from a supply of heat on admission, which enhances ignition and combustion, and reduces NO_x [22,27–31].

However, most publications have discussed such options without actually comparing them [25,27,32–39]. Several studies have identified viscosity as the main cause of problems in the use of HVP in diesel engines. The experimental conditions described in these studies are different [18,20,42–45,23–25,33,36,39–41].

The purpose of these works was thus to use the same experimental device to compare the three solutions proposed in the literature, and to test them under the same trial conditions to assess how each solution could contribute, if at all, to the use of Jatropha oil as a diesel engine fuel. We explain how the viscosity of vegetable oil actually affects engine performance, and how exhaust gas recirculation affects the preparation of the air/fuel mix and the emission of exhaust gases.

2. Materials and methods

The tests were carried out using pure Jatropha oil (PJO), the same oil was preheated to 100°C (HJO), and some preheated blends of 20, 40, 50 and 75% Jatropha oil in diesel, called J20, J40, J50 and J75, respectively. Engine fuel conversion efficiency and gas emissions were recorded for each fuel then compared with reference tests using non-preheated Jatropha oil, PJO, and pure diesel. The EGR work was carried out in the same way using PJO.

2.1 Engine test bench

The tests were carried out on an engine test bench mainly comprising a direct injection diesel engine, an electric alternator providing a three-phase voltage of 400 V, and two VIKINS type resistive load banks of 4 kW each to load the engines.

The engine was a LOMBARDINI 9LD561/2L, French-made, twin-cylinder, with natural suction, air-cooled, running at a constant speed of 1,500 rpm. The technical characteristics of the engine are listed in Table 1. Fuel was fed to the engine by a dual-fuel system (Fig. 1). Dual-fuelling enables an engine to be fed with two fuels successively. An electrically controlled valve, commonly called a “solenoid valve”, is used to switch between the two tanks .

The combustion products were analysed with a ***TESTO*** type ***350XL*** gas analyser. The analyser probe was placed inside the silencer, at a distance of 5 cm from the cylinder outlet. The gases measured by the analyser were carbon monoxide (CO), nitrogen oxides (NO and

NO_2), unspent hydrocarbons (HC), dioxygen (O_2) and sulphur dioxide (SO_2). CO_2 emissions were calculated based on the fuel characteristics and the residual oxygen measured in the gases. The characteristics of the analyser are listed in Table 2.

The exhaust gas temperature was measured using a type K NiCr-Ni thermocouple connected to an ALMEMO type 2690-8 data logger.

2.2 Preheating of Jatropha/diesel blends

A heating magnetic stirrer was used to preheat the *Jatropha curcas* oil and the *Jatropha* diesel blends to the chosen test temperatures. A thermocouple was used to regulate the temperature. The test temperature was obtained with an error of ±5°C. The engine feed circuit was heat-insulated to prevent temperature variation.

To obtain equal viscosity values of 5.9 mm²/s limit at 37.8°C according to NF T60 100, the J20, J40, J50 and J75 blends were preheated to respective temperatures of 50, 60, 70 and 100°C (Table 3).

2.3 Description of the exhaust gas recirculation (EGR) valve

To study the thermal impact of exhaust gas recirculation on exhaust gas emissions, we chose hot EGR, i.e. after filtering, the exhaust gases were returned to the engine without cooling.

Figure 2 is a diagram of the EGR system. It was installed between the exhaust pipe and the engine's new air intake. The walls of the tube were 50 mm thick and the internal diameter of the tube was 50 mm. The system was made from steel and could withstand high temperatures.

2.4 Characteristics of the fuels used

The characteristics of the fuels used are listed in tables 4 and 5. Viscosity was determined in accordance with the NF EN ISO 3104 standard, the low heating value (LHV) in accordance with the ASTM D 240 standard and the density in accordance with the NF EN ISO 3675 standard.

2.5 Test procedure

Tests were carried out at different engine loads and the efficiency and gas emission tests were carried out with the different fuels (pure diesel, pure Jatropha oil, diesel and Jatropha oil blends). For each test, the engine was started with diesel and run at zero load for 15 minutes. Two resistive load banks were used to load the engine up to a rated power of 6 kW at 1,500 rpm. That load rate was maintained for 5 minutes to stabilise the engine. This enabled the engine combustion chamber to reach a temperature of 500°C, the optimum condition for complete combustion of pure vegetable oils. The tests for the three options (preheating, blend and EGR) were carried out at an average ambient temperature of 36°C and in four stages:

i. First, tests were carried out with diesel to generate reference data. For each engine load ranging from 0% to 100% of rated power (with a step of 20), the exhaust gas temperature, exhaust gas emission and fuel consumption were measured after running at constant load for five minutes.
ii. Second, the same test was run as in i. above, with pure Jatropha oil as fuel.
iii. Third, the same test was run as in i. above with different concentrations (20%, 40%, 50% and 75%) of Jatropha oil blended with diesel as fuel.
iv. Fourth, the exhaust gas recirculation tests on the engine were carried out with a manual valve fitted between the exhaust pipe and the fresh air intake. Once the EGR system was installed on the engine, the tests were carried out as in i. above with pure Jatropha oil and diesel as fuel.

3 Results and discussion

3.1 Viscosity of the fuels used

In order to compare the effect of the different types of fuel under the same conditions, all the fuels had to have the same viscosity. Figure 3 presents the effect of temperature on the viscosity of each of the fuels used. For PJO, by preheating, the viscosities fell significantly, reaching values close to that fixed by standard NF T60 100 equivalent to ASTM D 97-93, at temperatures approaching 100°C. Standard NF T60 100 gives a maximum kinematic viscosity limit of 5.9 mm²/s at 37.8°C for the use of diesel in diesel engines.

3.2 Results and analysis of the Jatropha oil preheating tests

3.2.1 Comparison of the performance of the engine with Jatropha oil, preheated Jatropha oil and diesel

Specific fuel consumption, thermal efficiency, and the engine exhaust temperature for Jatropha oil (preheated or not) and diesel, depending on the increase in load rate, are shown in Figures 4, 5 and 6.

The specific fuel consumption of the engine decreased with an increase in load with all the fuels (Figure 4). It increased more than 11% on average with preheated Jatropha oil and 13% with non-preheated Jatropha oil compared to that with pure diesel. This result tallies with the results of earlier studies [22,44,46,47]. The difference in the low heating value (LHV) between diesel and Jatropha oil may explain the extra consumption. Indeed, the LHV of diesel is 14% higher than that of Jatropha oil. To achieve the same power output from the engine, a higher mass of Jatropha oil was injected. The specific consumption of non-preheated Jatropha oil was, on average, 2% higher than that of preheated Jatropha oil. This difference is too small to be able to enable any conclusions to be drawn. However, the high viscosity and density of non-preheated Jatropha oil compared to preheated oil resulted in poorer oil atomization. It can thus be imagined that some large droplets underwent poor combustion inside the chamber, which may have led to greater consumption of non-preheated oil than of preheated oil.

The thermal efficiency of the engine increased with an increase in load, whatever the fuel used (Figure 5). With low loads of up to 40%, the efficiency of diesel, preheated Jatropha oil and non-preheated Jatropha oil was comparable. With engine load between 40% and 100%, efficiency was better with the preheated Jatropha oil, followed by non-preheated Jatropha oil and pure diesel. However, these differences were not significant and a tendency emerged for Jatropha oil to be more efficient than diesel. This result differs from those obtained in studies by other authors [47,48], in which the efficiency achieved with diesel was better than that of Jatropha oil. However, it should be noted that the equipment and operating procedures in our tests were not the same as those used by the other authors.

The engine exhaust gas temperature values (Figure 6) were lower with diesel with all the engine loads. Nevertheless, the temperatures were comparable to those for the two Jatropha oils, with slightly higher values for the non-preheated Jatropha oil between 40 and 100% of maximum engine load. In the literature, the highest exhaust temperature was reported with preheated Jatropha oil . This can be explained by numerous larger droplets formed with Jatropha oil compared to diesel, leading to more diffusive combustion [49].

3.2.2 Comparison of emissions of engine exhaust with Jatropha oil, preheated Jatropha oil and diesel

Changes in carbon monoxide (CO) and nitrogen oxide (NOx) emissions as a function of the engine load with Jatropha oil (preheated and non-preheated) and with diesel are presented in Figures 7 and 8, respectively.

At an engine load of 0% to 50%, CO emissions were comparable for diesel fuel and HJO, whereas PJO displayed a higher CO rate (Figure 7). Above a load of 50%, CO emissions with Jatropha oils were higher than with diesel. This result tallies with those obtained in other studies [47–53]. Another finding with our engine, was that, when the load was higher than 80%, the CO emission levels increased significantly with all the fuels (Figure 7). At a constant rotational speed, load is directly proportional to the quantity of fuel injected into the combustion chamber. In our engine, at more than 80% of the maximum injection rate, the fuel: air ratio increased such that the lack of oxygen caused incomplete combustion (thermal degradation) of the fuels and the rate at which hydrocarbon radicals was converted into CO may have been greater than that of CO conversion into CO_2.

The fact that with the vegetable oils CO emissions were higher can be explained by the formation of CO during combustion. CO forms as an intermediate stage in the fuel oxidation process, leading to CO_2 as the end product. In our tests, the time available for the reactions to take place was the same for all the tests (constant engine speed), but the larger size of the Jatropha oil droplets (due to high viscosity and surface tension) led to slow evaporation and poorer preparation of the air/fuel mixture compared to diesel fuel [25].

For an engine load of 0% to 80%, NOx emissions were comparable for diesel fuel and HJO, as PJO presented a lower rate (figure 8). Preheating up to 100°C appeared to make Jatropha oil behave in the same way as diesel fuel. At high engine loads, temperatures in the combustion chamber were high with fuel-rich mixtures. The high temperature in the combustion chamber increased the thermal formation of NO (Figure 8). At very high loads (100%), the high richness led to a slight increase in NOx due to incomplete combustion. Some zones of gas cooling in the combustion chamber appeared. These results clearly show that the decisive factors in NOx formation are temperature and local fuel/air richness. However, the higher NOx emissions with HJO than with diesel and PJO are more difficult to explain, as there were no significant differences in exhaust temperature between HJO and PJO. A similar result was also described by Chauhan, B.S., et al.[20,54] .

To conclude, the aim of these tests was to identify how the viscosities of preheated and non-preheated Jatropha oil affect combustion. Our results show that HJO and diesel produced similar CO and NOx emissions, except for load above 80% impacted by a high fuel:air ratio.

3.3 Results and analysis of the tests of the Jatropha oil/diesel blends

The results of engine efficiency and emissions when running on blends with different proportions of Jatropha oil blended with the diesel, were analysed and compared with those for diesel, PJO and HJO.

3.3.1 Comparison of engine performances with Jatropha oil, diesel and Jatropha oil/ diesel blends

Figures 9 to 11 show specific fuel consumption, thermal efficiency and engine exhaust temperatures depending on the engine load for PJO, HJO, the blends and diesel, respectively.

The specific fuel consumption curves all show the same trend, i.e. a decrease with an increase of the load (Figure 9). Specific fuel consumption was lower with diesel and increased with an increase in the proportion of Jatropha oil in the blends for all the engine loads. With 20% Jatropha oil in blend J20, specific consumption was comparable to that of diesel. This result tallies with the results of some earlier studies [35,47]. The increase in specific fuel consumption resulted from the low LHV of the Jatropha oil and its blends with diesel.

Overall engine efficiency increased with the load for all the fuels (Figure 10). It was lowest with pure diesel and increased with an increase in the proportion of Jatropha oil in the blend. The highest efficiency was obtained with J75 followed by HJO. The efficiency with PJO was comparable to that with J20 and J40.

The engine exhaust gas temperature curves for all the fuels used showed the same trends, increasing in line with an increase in the load (Figure 11). The temperature was lowest with J40 and J50. Other researchers have obtained the same results [47].

3.3.2 Comparison of engine exhaust emissions with Jatropha oil, diesel and Jatropha oil/ diesel blends

Changes in carbon monoxide (CO) and nitrogen oxide (NOx) emissions in the engine exhaust gases with the different fuels as a function of the load are shown in Figures 12 and 13, respectively.

CO emissions (Figure 12) were stable with all fuels and increased beyond a load of 80% of Jatropha oil. These emissions were lowest with diesel. No relationship was found between the increase in CO emissions increase and the proportion of Jatropha oil in the blends, for any engine load. NO_x emissions (Figure 13) were very similar with all the fuels, increasing in line with an increase in the engine load. This result tallies with those in some other studies [55,56]. However, it should be noted that blending reduces the viscosity of Jatropha oil to the point at which it is comparable to diesel. However, during the combustion of these blends problems in performance and gas emissions persist. Consequently, differences in combustion emissions and performances can be attributed more to the intrinsic characteristics of the oil than to its viscosity.

3.4 Results and analysis of the exhaust gas recirculation tests

Based on the literature, EGR tests with diesel engine were carried out with exhaust gas recirculation rates ranging from 10% to 30% [57,58]. Some preliminary engine efficiency and emission tests carried out with 10, 15, 20, 25 and 30% of exhaust gas recirculation established the optimum EGR rate for the engine used at 20%. At this stage of the tests, recirculation was conducted with that EGR value. The fuels used were PJO and diesel. It must be noted that the air inlet configuration was modified, as shown in Figure 2, compared to the previous tests. Therefore, differences in CO and NOx rates are reported for all fuels increasing CO by 40 % and reducing NOx by 10% for diesel, and increasing CO by 100% and reducing NOx by 40% in average for PJO.

3.4.1 Comparison of energy efficiency of the exhaust gas recirculation mode with Jatropha oil and diesel

Figures 14 to 16 show respectively specific consumption, overall efficiency and engine exhaust temperatures as a function of the load for PJO and diesel, with an EGR rate of 20%.
Specific fuel consumption decreased with an increase in the load, be it for diesel or PJO with or without EGR (Figure 14). The specific consumption obtained with PJO with or without EGR was higher than that obtained with pure diesel. Maintaining power was responsible for this extra consumption [41,59]. We found that EGR did not significantly affect the specific consumption of the two fuels.
Thermal engine efficiency increased with an increase in the engine load with all the fuels, with or without EGR (Figure 15). Efficiency was higher with PJO with or without EGR.
The engine exhaust gas temperature increased with recirculation at all the engine loads (Figure 16). PJO with EGR resulted in the highest temperatures followed by diesel without EGR, then by PJO without EGR. This was partly due to recirculation, which increased the average temperature of the gases in the chamber, and to the extra consumption of PJO. This result tallies with the results obtained by Shehata [41]. In that study, the author showed how increasing the exhaust temperature affected combustion, and revealed a substantial systematic change in the heat release rate. The surface of the pre-mixture phase decreased in line with the increase in recirculation. This phenomenon was also linked to the increase in the temperature of the intake gas. In fact, the increase in temperature made it easier to achieve self-ignition conditions for the fuel blend. This reduced the length of the pre-mixture combustion phase to the benefit of the diffusion phase, with a slight reduction in the ignition lag.

3.4.2 Comparison of engine exhaust emissions of the exhaust gas recirculation mode with Jatropha oil and diesel

Figures 17 and 18 show respectively, CO emissions and NOx emissions as a function of the load with PJO and diesel, with an EGR rate of 20%.

EGR greatly influenced CO emissions, with both diesel and PJO (Figure 17). The CO emitted was low with diesel without EGR, and slightly below that with pure vegetable oil without EGR. Emissions were substantial with diesel with EGR and even with for PJO with EGR. The differences were greater between the EGR and non-EGR tests at high loads. This situation can be explained by the richness of the fuel blend. Combustion in a diesel engine requires a fair amount of excess air [60]. With EGR, as the load increases excess air decreases. The blend then becomes fuel-rich and combustion quality begins to decline. The products of incomplete combustion, such as carbon monoxide (CO), are then greater, which was the case with diesel and PJO.

NO_x emissions varied greatly depending on the exhaust gas recirculation conditions (Figure 18). A marked drop in such emissions was found with recirculation, be it with diesel or PJO. The reduction was estimated to be:

- 25% on average with PJO at low loads and 66.5% at the maximum load;
- 34.5% with diesel at low loads and 47.5% at the maximum load.

It turned out that the EGR system enabled a substantial reduction in nitrogen oxide emissions. This phenomenon was mainly observed at high loads when the energy contribution of the fuel was very low [41].

It is clear that EGR affects NO_x formation mechanisms [41,57,58,61]. This could be explained by the existence of low-reactive burned gases in the intake, which reduces the instantaneous temperature of combustion, thereby limiting the Zeldovich mechanism. However, our analysis of the exhaust temperature showed that it increased with recirculation, resulting in an increase in temperature in the combustion chamber. The reduction in NO may have been due to the influence of recirculation on local fuel richness, which can be high in some zones during combustion, thereby reducing NO formation by the “prompt NO” mechanism.

3.4.3 Critical comparison of the three main solutions regularly proposed in the literature for the sustainable use of vegetable oils as fuel in diesel engines

This work compared the three main solutions regularly proposed in the literature for efficient and sustainable use of vegetable oils as fuel in diesel engines.

Table 6 shows the advantages and disadvantages of the three solutions for the use of vegetable oils in diesel engines. It clearly shows that EGR is not an appropriate solution for the use of PJO in diesel engines. Indeed, the use of EGR in an engine depletes combustion due to the admission of non-combustible gases into the engine (CO_2, H_2O, etc.). Hence the reduction in the combustion temperature, which limits combustion reactions and clogs the engine with unburned residues of vegetable oil.

Once preheated, the vegetable oil can be used directly in the engine, being more fluid, it is more easily injected into the engine, but as discussed above, when the engine is running at low load, this option does not solve the problems of poor performance and emissions of pollutants in exhaust gases.

When the engine is subjected to high load variations or no vegetable oil is available to replace the 100% diesel, mixtures of vegetable oil and diesel may be a good alternative. Indeed, PJO / gas oil mixtures decrease the viscosity of vegetable oil while increasing its cetane number [58], which improves fuel injection and combustion. However, although this solution does not require modifications to the diesel engine, it requires the use of a mixer to ensure that the mixture remains homogeneous while in use. The limit of using PJO/diesel blends remains the homogeneity of the mixture and the low oil content in the mixture (30% PJO).

For the best use of 100% oil as engine fuel, bi-fueling is preferable [62]. The bi-fuel system consists of starting and stopping the engine with diesel and using PJO only when the engine is under load (i.e. with a hot combustion chamber). This way of using the PJO not only makes it possible to ensure complete combustion of the oil, but also to clean the engine supply circuit and to remove the deposits of unburned matter on the combustion components [62].

4 Conclusion

The purpose of this study was to investigate and compare the adaptations most frequently mentioned in the scientific literature (preheater, oil/diesel blends, exhaust gas recirculation) to enable vegetable oils used as fuel to behave similarly to diesel in a direct injection diesel engine. The tests were carried out with pure Jatropha oil (PJO), pure Jatropha oil preheated to 100°C (HJO), and some preheated blends of 20, 40, 50 and 75% Jatropha oil in diesel. Engine fuel conversion efficiency and gas emissions were recorded using each type of fuel and compared with reference tests using non-preheated Jatropha oil, PJO, and pure diesel. The EGR work was carried out in the same way using PJO.

Based on the analysis of the results of these tests, using an EGR system to re-inject hot gases into the combustion chamber for fuelling with pure vegetable oils did not appear to be worthwhile. The recommendation is therefore to inject vegetable oils directly into the engine, or to blend them with diesel oil or PJO (depending on the available quantities of vegetable oil), while ensuring that combustion conditions are optimum in the combustion chamber (high engine load). Under these conditions, the low viscosity of vegetable oils does not affect the quality of their combustion in the diesel engine. However, the high viscosity of certain vegetable oils may be problematic for their flow and for their injection into the engine, and preheating may be required to make them more fluid.

We can thus conclude this work as follows: whatever the solution (preheating, mixing, EGR) of the oil, bi-fuel guarantees the durability of the engine.

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Table 1. Technical characteristics of the diesel engine.

	Characteristics	Specification
Engine	Type	Lombardini 9LD 561-2/L, four strokes, air cooled, direct injection, two cylinders, compression ignition engine, constant speed
	Stroke/Bore	90mm/88mm
	Displacement volume	1120 cm^3
	Rated power	12 kW at 2200 rpm
	Compression ratio	17.5:1
Generator	Type	Genelec
	Power	8 kVA
	Cos φ	0.8
	RPM	1500

Table 2. Technical characteristics of the Testo 350 XL combustion gas analyser.

Gas	Detector	Resolution	Scale	Accuracy
CO_2	Evaluated from O_2	0.01%	0 to CO_2 max	-
CO	Electrochimical cell	1 ppm CO (0 to +10000 ppm CO)	0... +10000 ppm CO	±5% of mv (+100 to +2000 ppm CO) ±10% of mv (+2001 to +10000 ppm CO) ±10 ppm CO (0 to +99 ppm CO)
O_2	Electrochimical cell	0.1%	0 to 25%	± 8%
NO	Electrochimical cell	1 ppm NO (0 to +3000 ppm NO)	0 to +3000 ppm NO	±5% of mv (+100 to +1999.9 ppm NO) ±10% of mv (+2000 to +3000 ppm NO) ±5 ppm NO (0 to +99 ppm NO)
NO_2	Electrochimical cell	0.1 ppm NO_2 (0 to +500 ppm NO_2) 0.1 ppm NO_2 (0... +500 ppm NO_2)	0 to 500 ppm NO_2	±5% of mv (+100 to +500 ppm NO_2) ±5 ppm NO2 (0 to +99.9 ppm NO_2)

Table 3: Kinematic viscosity trend depending on the temperature

Fuel	Viscosity (mm²/s)					
Temperature (°C)	**37.8**	**50**	**60**	**70**	**80**	**100**
Diesel fuel	3.44	2.67	2.22	-	-	-
J20	6.37	4.73	4.26	3.07	2.81	2.45
J40	8.48	6.86	5.9	4.9	3.78	2.83
J50	12.1	8.67	6.77	5.57	4.58	3.33
J75	21.7	15.1	11.41	9.27	7.4	5.41
PJO	36.8	25.5	18.4	14.36	11.44	7.1

Table 4: Fuel physical characteristics

Fuel	Density at 15°C (kg/dm^3)	Viscosity at 37,8°C (mm²/s)	PCI (kJ/kg)
Diesel fuel	0.855	3.44	45749
J20	0.869	6.37	43580
J40	0.882	8.48	42934
J50	0.888	12.12	42250
J75	0.904	21.74	39200
PJO	0.919	36.79	39104

Table 5: Physico-chemical characteristics of the fuels

Fuel	Diesel fuel	Jatropha oil
Masse volumique à 15°C (kg/dm^3)	**0,855**	**0,919**
Viscosity at 37.8°C (mm^2/s)	3.37	36.8
Low Heating value (kJ/kg)	44 868	39 104
Cetane number (%)	49.1	45
Pour point (°C)	<-5	-3
Cloud point (°C)		-3
Flash point (°C)	64 ,2	
Acidity (mg KOH /g)	-	0.66

Table 6: advantages and disadvantages of the three solutions for using vegetable oils in the diesel engine

	Advantages	**Disadvantages**
HJO	• The viscosity of HVP decreases thus less mechanical fatigue of the power supply components of the engine. • Engine performance comparable to that of diesel • The difference in engine emissions is more due to the characteristics of HVP and diesel and also to the combustion process.	• Adaptation of the components of the supply circuit to oil preheated to 100 ° C • The problem of depositing in the combustion chamber remains
Blend PJO/fuel	• The viscosity of PVO thus decreases the mechanical tired of the motor feeding members. • Engine performance comparable to that of diesel • The difference in engine emissions is more due to the characteristics of HVP and diesel and also to the combustion process. • Less fouling in the combustion chamber.	• Phase separation problem in the tank • Installation of a mixer device

EGR	• Reduction of NOx emissions from the engine	• Increase in engine consumption • Decreased thermal efficiency engine • Increased CO emissions • Incomplete combustion of fuel or deposition of unburned in the combustion chamber.

Figure

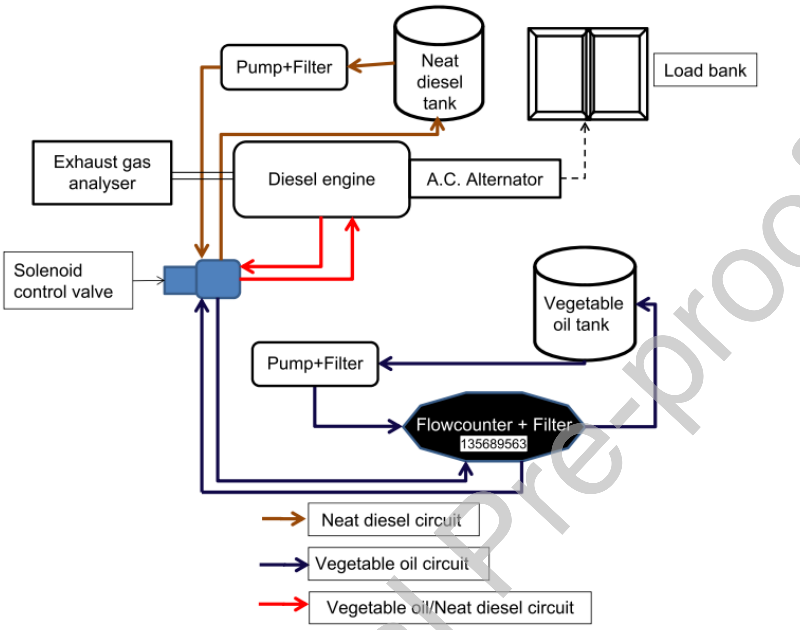

Fig. 1. Schematic diagram of experimental setup.

Fig. 1: Schematic diagram of the experimental setup

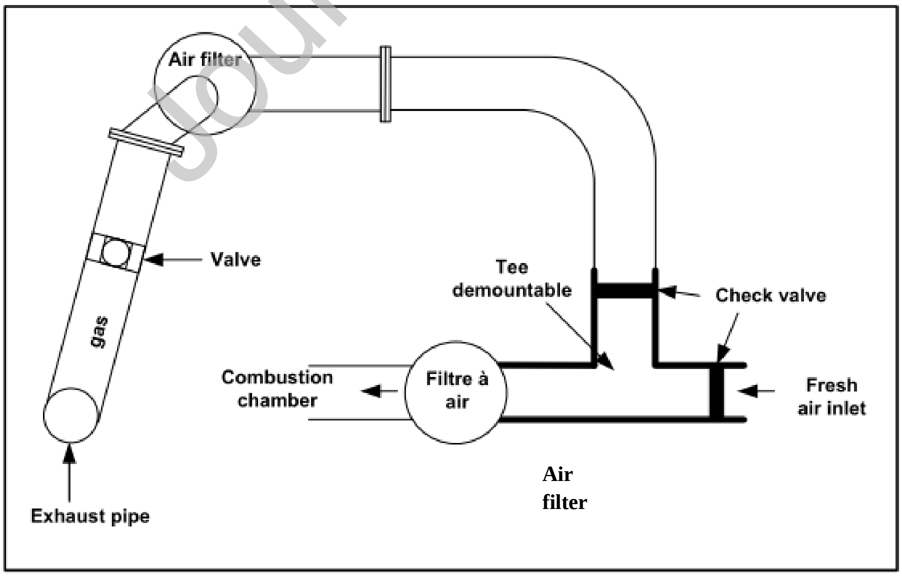

Fig. 2 : Diagram of the EGR valve

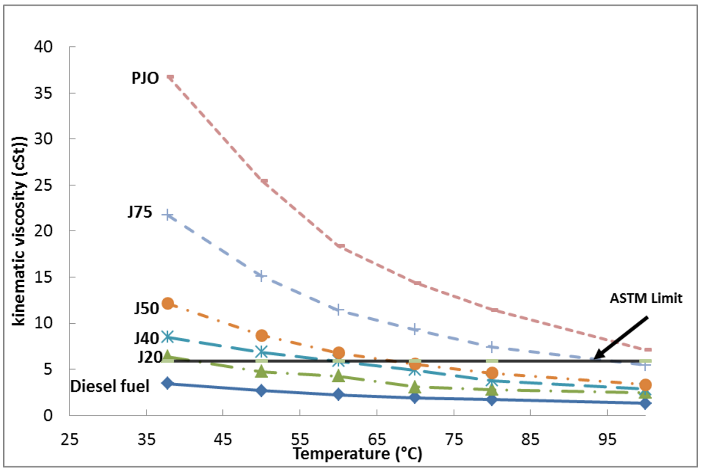

Fig. 3: evolution of the kinematic viscometer according to the temperature

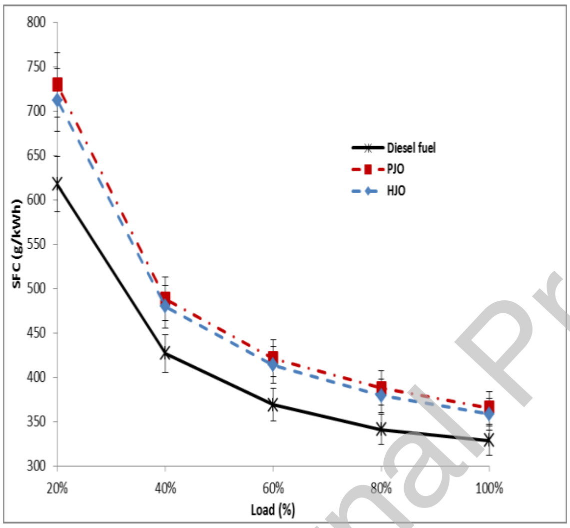

Fig. 4: Specific consumption of the engine depending on the load for different fuels (PJO: pure Jatropha oil - PJO Jatropha oil preheated to 100°C)

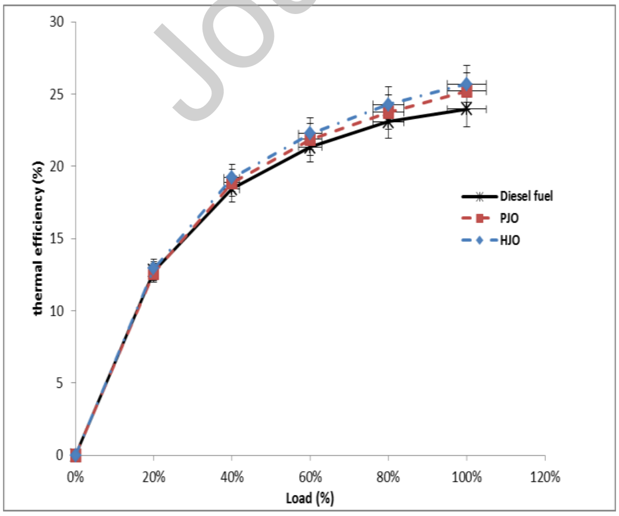

Fig. 5: thermal efficiency of the engine depending on the load for different fuels (PJO: pure Jatropha oil - PJO Jatropha oil preheated to 100°C)

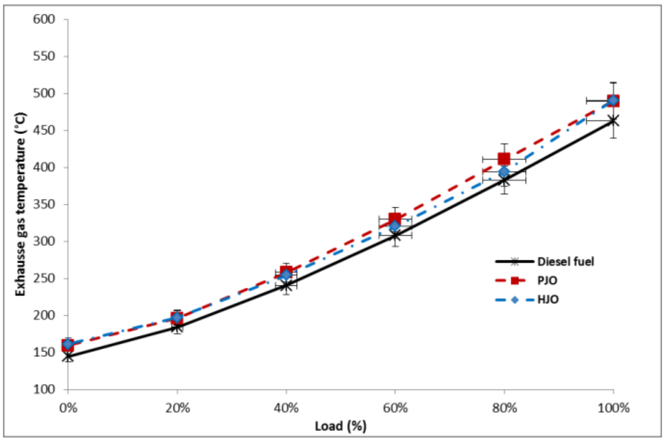

Fig. 6: Engine exhaust gas temperatures depending on the load for different fuels (PJO: pure Jatropha oil - PJO Jatropha oil preheated to 100°C)

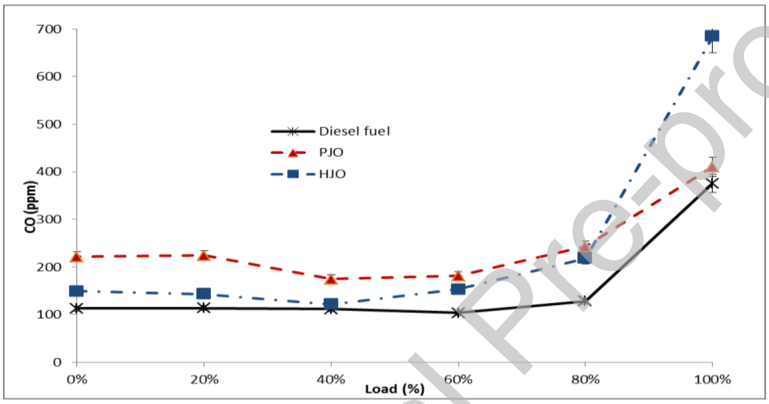

Fig. 7: Engine CO emission depending on the load and for different fuels (PJO: pure Jatropha oil - PJO Jatropha oil preheated to 100°C)

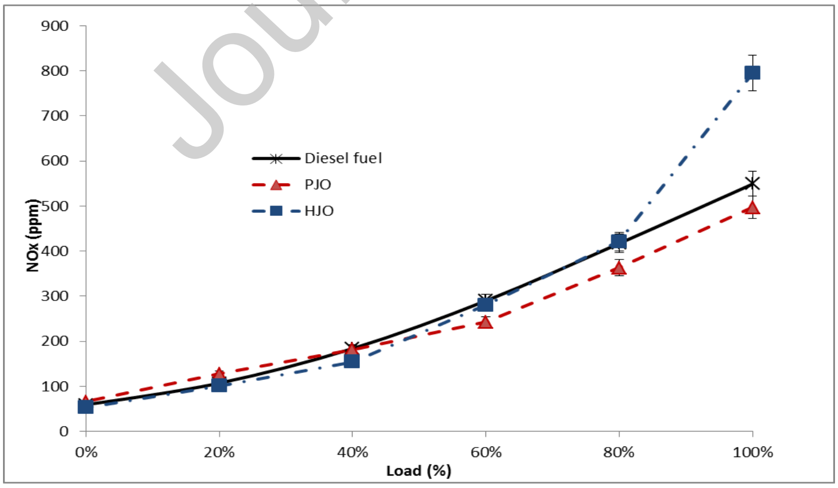

Fig. 8 : NO_x emission of the engine based on the load and for different fuels (PJO: pure Jatropha oil - PJO Jatropha oil preheated to 100°C)

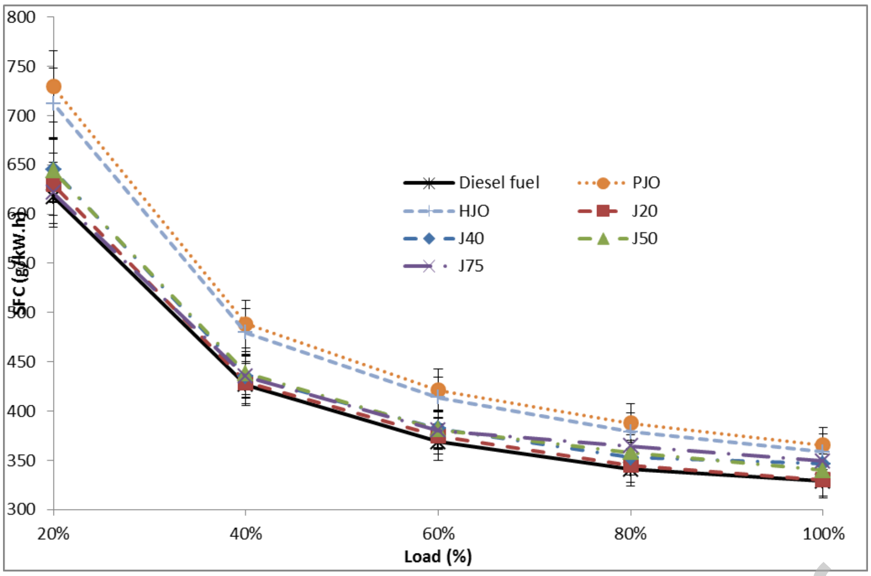

Fig. 9: Specific fuel consumption of the engine depending on the load and for different fuels (J20: 80% diesel fuel and 20% Jatropha - J40: 60 diesel fuel and 60% Jatropha - J50 : 50% diesel fuel and 50% Jatropha - J75 : 25 diesel fuel and 75% Jatropha)

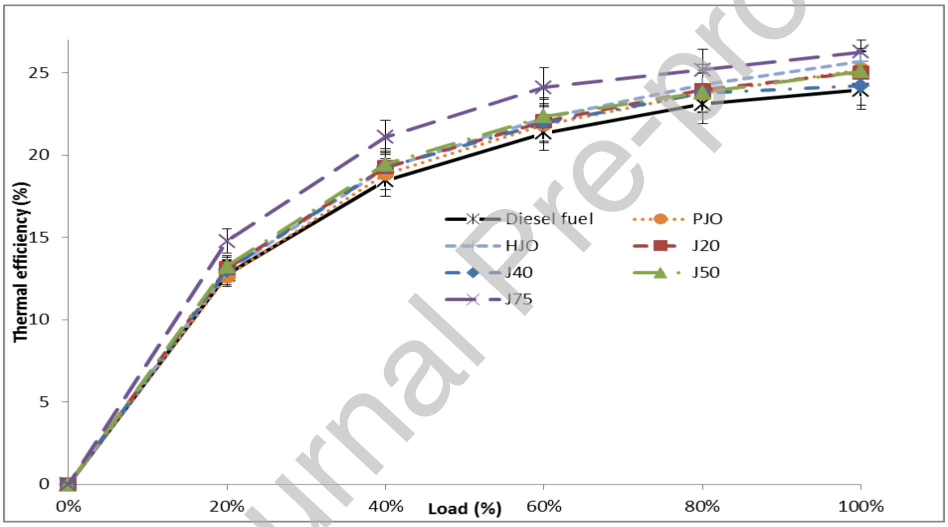

Fig. 10: thermal efficiency of the engine depending on the load and for different fuels (J20: 80% diesel fuel and 20% Jatropha - J40: 60 diesel fuel and 60% Jatropha - J50 : 50% diesel fuel and 50% Jatropha - J75 : 25 diesel fuel and 75% Jatropha)

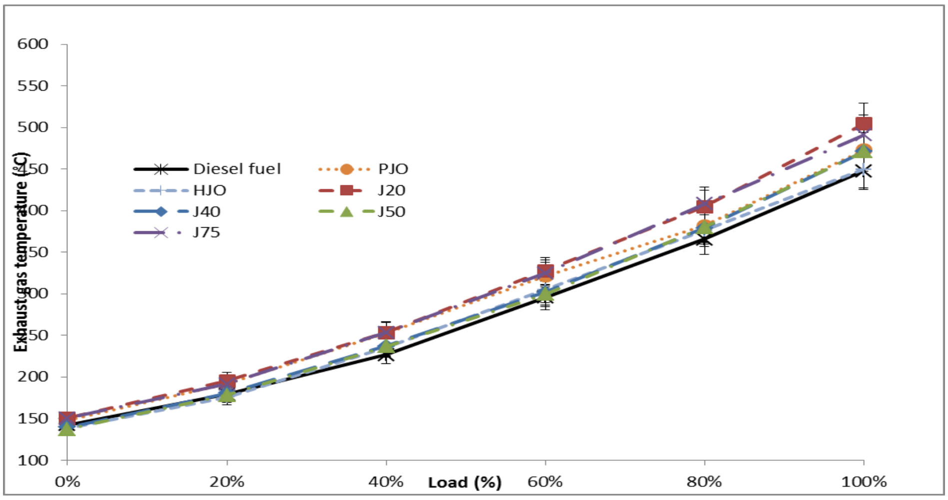

Fig. 11: Engine exhaust temperature depending on the load and for different fuels (J20: 80% diesel fuel and 20% Jatropha - J40: 60 diesel fuel and 60% Jatropha - J50 : 50% diesel fuel and 50% Jatropha - J75 : 25 diesel fuel and 75% Jatropha)

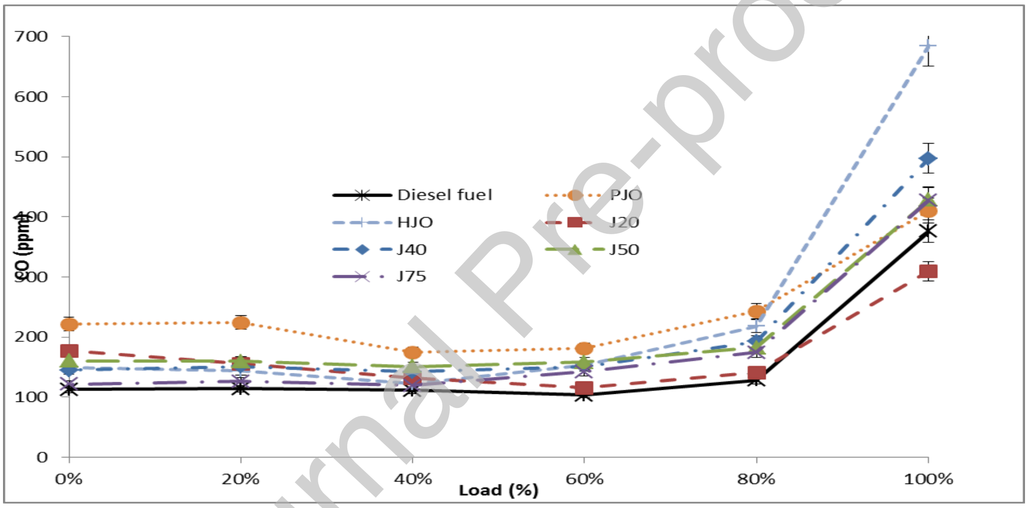

Fig. 12: Engine CO emissions depending on the load for different fuels (J20: 80% diesel fuel and 20% Jatropha - J40: 60 diesel fuel and 60% Jatropha - J50 : 50% diesel fuel and 50% Jatropha - J75 : 25 diesel fuel and 75% Jatropha)

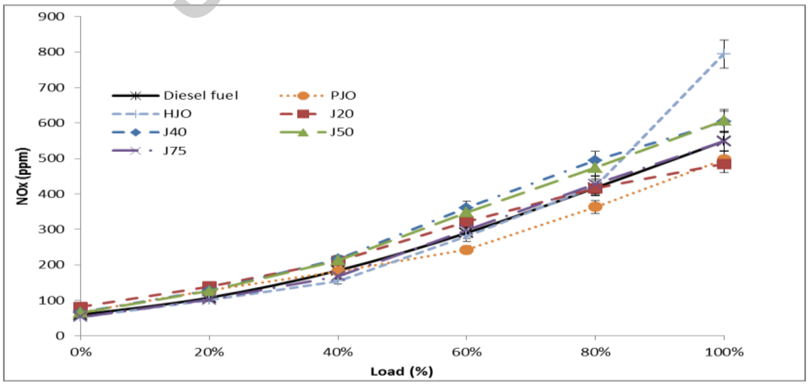

Fig. 13: Engine NOx emissions depending on the load for different fuels (J20: 80% diesel fuel and 20% Jatropha - J40: 60 diesel fuel and 60% Jatropha - J50 : 50% diesel fuel and 50% Jatropha - J75 : 25 diesel fuel and 75% Jatropha)

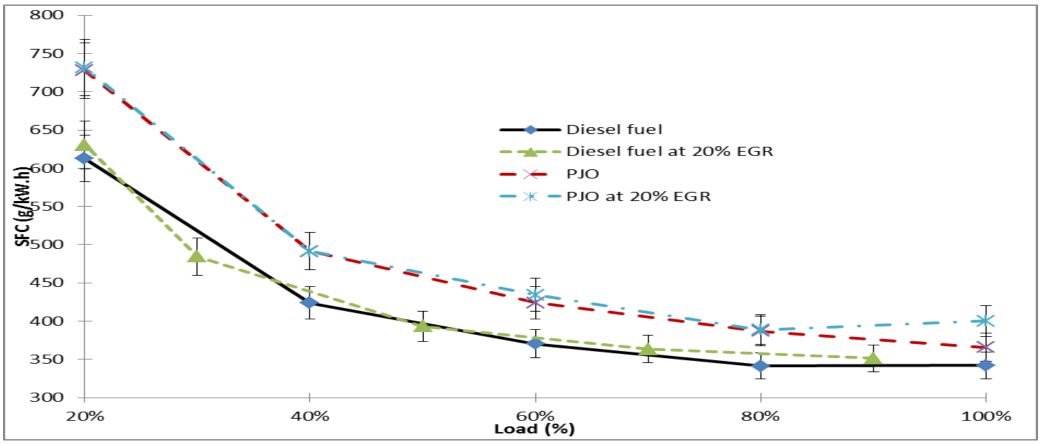

Fig. 14: Engine specific fuel consumption depending on the load for different fuels (20% EGR)

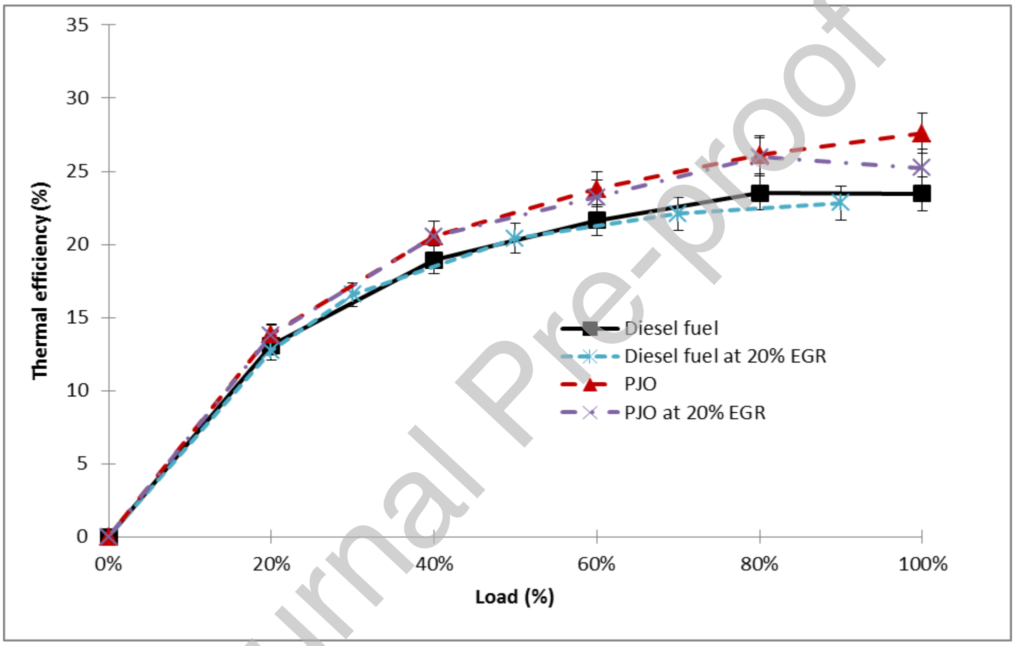

Fig. 15: thermal efficiency engine depending on the load for different fuels (20% EGR)

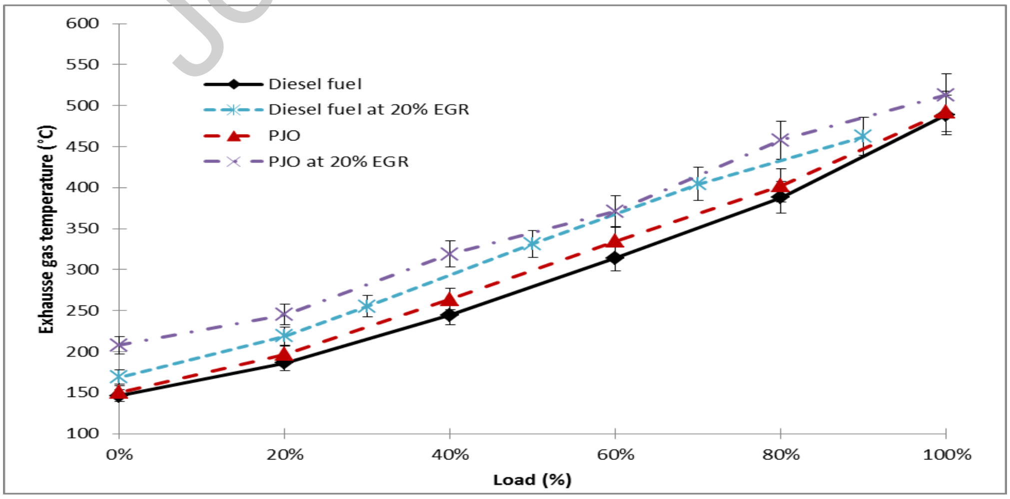

Fig. 16: Engine exhaust gas temperature depending on the load for different fuels (20% EGR)

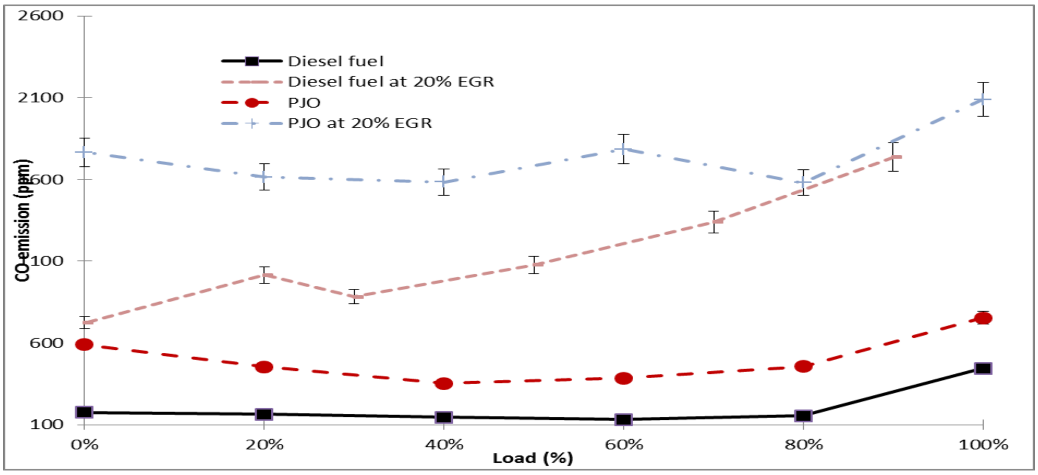

Fig. 17: Engine CO emissions depending on the load for different fuels (20% EGR)

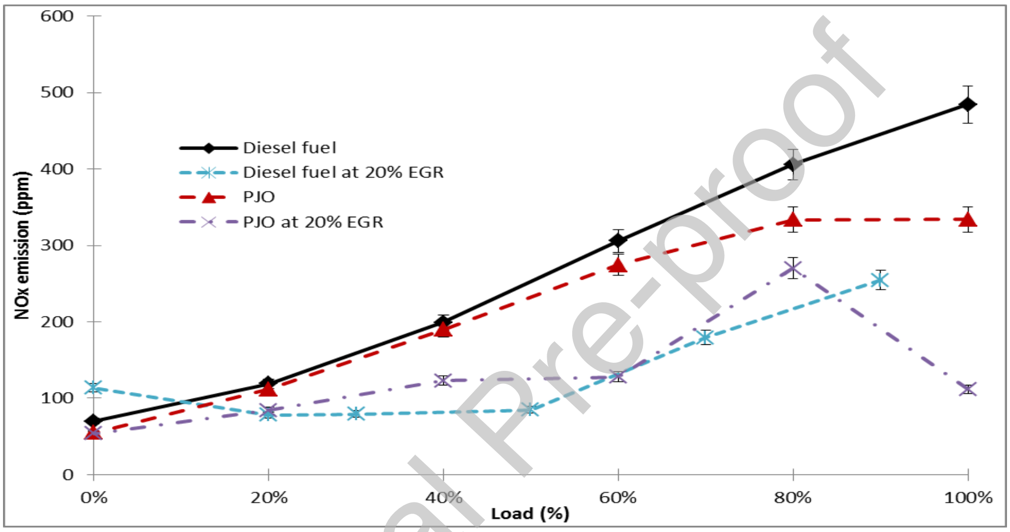

Fig. 18. Engine NOx emissions depending on the load for different fuels (20% EGR)