

EFFECT OF NAPHTHA ON BLENDING ALCOHOL DIESEL FUEL ON C.I. ENGINE PERFORMANCE

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ABSTRACT

A comparison of diesel engine performance landmarks when blending ethanol alcohol and Naphtha as assistant factor with diesel fuel had been experimentally made. The used blending ratio is 5% Naphtha, 90% pure Gas oil and 5% ethanol alcohol. The exhaust temperature has been decreased, and the fuel consumption seen to be less accompanied with increasing in energy due to ability of blend to burn completely, which yields lower economic point, in addition to the brake thermal efficiency and Air/Fuel ratio increased in mixed fuel with increasing the engine speed, which means an enhancement in operating performance.

KEYWORD: Naphtha, Ethanol Alcohol, C.I. Engine performance, Blended fuel.

1. INTRODUCTION

The internal combustion features of Ethanol & Methanol Alcohol compared with gasoline combustion alone are: the thermal value of Alcohol is less than that of gasoline which is half the thermal value of gasoline, the air amount required for Alcohol burning is half that required for gasoline burning. The vaporizing heat for Alcohol is higher relatively and reach third time that of gasoline, this lead to best cooling of Air/fuel charge and decreasing the high temperature of the engine, Alcohol has high-octane No. about (98-99), so it tends to lift the octane No. of gasoline when blend with it, as an example; when blend 20% of Methanol with gasoline, the octane no. will increase to approximately (90-98). The exhaust gases of Alcohol make to decrease the pollution due to reducing of CO gases causes by poor mixture of Alcohol comes from containing of Oxygen in its composition, the amount of NOx in exhaust gas is also less. Alcohols are best solvents for gums, in addition to its higher ability to burn [1].

It has been revealed that combustion of the compression-ignition diesel engine takes place in three main phases such as ignition delay, premixed combustion phase and diffusion combustion phase. The ignition delay period is the time between the start of the injection and the start of combustion, it has an influence on all combustion processes. Many researchers have made the number of experimental research on the use of vegetable oils as fuel in diesel engines, and reported that the high viscosity and low volatility of planting oils resulting in low atomization, slow burning, more smoke emissions and uncontrolled combustion. It is increasing the exhaust gas temperature and NOx reduction with a slight increase in CO emissions by mixing with vegetable oils such as diesel fuel consumption observed [1]. The use of palm oil as fuel in diesel engines reported that short term use of palm oil significantly increases yield and output levels and prolonged use causes carbon deposits and piston rings sticking [2]. The higher viscosity of biodiesel tends to reduce engine power and engine torque, even the lowest calorific value of biodiesel-fueled, high specific fuel

consumption results and decreasing temperature of combustion [3]. The use of biodiesel derived from rice bran oil concluded that there is an increase in NO_x emissions due to the presence of molecular oxygen in biodiesel [4]. The n-propanol mixture with diesel shows almost the same performance of brake thermal efficiency at low and medium loads, and a higher percentage of n-propanol addition increases the brake thermal efficiency, reducing the brake specific energy consumption at medium and high loads [5]. Experimental results indicate that the addition of 4% ethanol to diesel fuel increases the engine's output power and efficiency, while reducing the specific fuel consumption for different compression ratios. Greater efficiency was achieved by compression ratio of 21 with an increase ratio of more than 3.5% [6]. The brake thermal efficiency with fumigation run was lower than diesel [7]. Using 20% of ethanol as a fumigant can

produce an increase about 7.5% in brake thermal performance [8].

2. METHODOLOGY

It is impossible for high density diesel fuel to mix with low density Alcohol unless that the Alcohol should being pure (100%) from grays such as water which help to separate the blend, or by using catalytic which help for blending. The process of purifying Alcohol include adding Calcium Chloride to Alcohol then by distillation in order to get pure Alcohol (99- 100)%. Whereas the catalytic used is Naphtha which must be added to Alcohol-diesel fuel to assist for blending due to consisting of several chains of paraffin and aromatics. The experimental investigation was carried out in two stages: the first stage was to test the blends of diesel and Ethanol with and without the additive of Naphtha. Then, the fuels are used to run the diesel engine to test its characteristics of performance. The specification of diesel fuel, Naphtha and Alcohol used is indicated in tables (1).

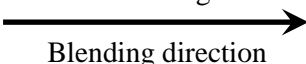
Table (1): specification of Diesel fuel, Naphtha and Alcohol used:

Pure Gas Oil	Naphtha	Alcohol
Chemical formula: C ₁₂ H ₂₆	Chemical formula: C ₁₆ H ₃₄	Chemical formula: C ₂ H ₅ OH
Specific density: 0.84 kg/m ³	Specific density : 0.79 kg/m ³	Specific density: 0.794 kg/m ³
L.C.V.: 44109 kJ/kg	L.C.V.: 46046 kJ/kg	L.C.V.: 29800 kJ/kg
Cetane No.: 42	Cetane No.: 50	Octane No.: 99
Boiling temp.: 190-280 °C	Paraffin (max): 70%	Boiling temp.: 77.8 °C
	Aromatics (max): 9%	
	Initial boiling Temp.: 30 °C	

3. EXPERIMENTAL OUTFIT

An amount of Naphtha has been added (50 cm³) to pure gas oil with the same amount (50 cm³) and blend by mixing. The mixture left for 48 hours to see its possibility of occurring separation of its compositions. Many experiments ratios of both elements has been processed according to the following blending direction, and the best blending values was 5% Naphtha, 90% pure gas oil and 5% Ethanol.

5% Naphtha + 90% Pure gas oil + 5% Ethanol



Blending direction

V6 - 3.7L FIAT water-cooled engine with compression ratio 17.5, bore 87.5 mm, and a stroke 110 mm, has been used as shown in figure (1). The engine had operated with above blending without any problem. Several results had been concluded when using pure gas oil and mixed fuel.

Four rotational speeds have been taken under consideration in making our calculations (1200, 1400, 1600 & 1800) r.p.m for both cases: pure gas oil and mixed fuel. The torque reading values also specified to be (40, 60, 70 & 80) N.m. in case of pure gas oil, and (20, 40, 60 & 80) N.m in case of mixed fuel. The consumed fuel volume specified to be 100 cm³, then the fuel consumption

determined after calculated the required time in seconds to consume above fuel volume by using time gauge. Atmospheric pressure and

temperature have been measured by laboratory devices to be 98.1 kPa and 13.5 °C respectively.

No	Item
1	Engine
2	Dynamometer
3	R.P.M. meter
4	Lube oil temp. meter
5	Cooling water inlet pipe
6	Cooling water flow rate measuring tank
7	BOSCH smoke pump
8	Exhaust gas temp. meter
9	Exhaust pressure manometer
10	Water outlet temp. meter
11	Air flow meter
12	Inlet pressure manometer
13	Diesel flow line and burette
14	Fuel pump

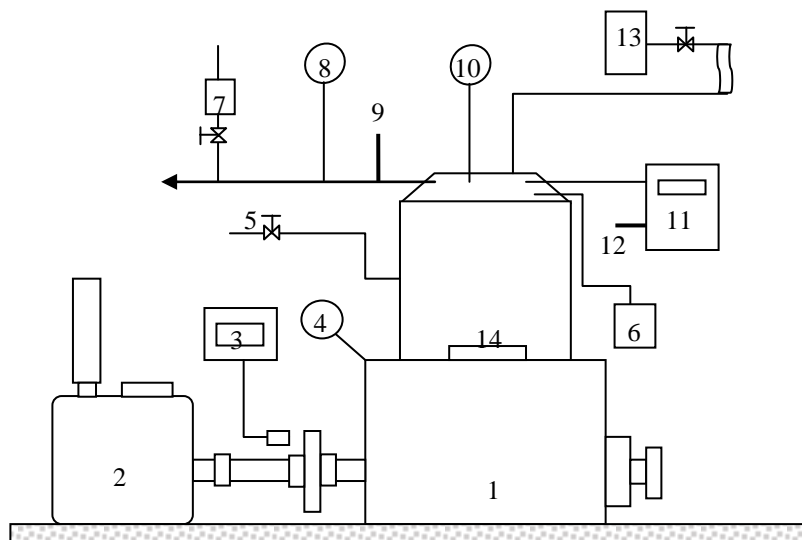


Fig. (1): schematics diagram of the test rig.

Computations

Brake power has been calculated using the formula [9]:

$$B.P. = \frac{2.\pi.N.T}{60000} \quad \dots (1)$$

Where;

N: rotational speed in revolution per minute

T: exerted torque (N.m)

B.P.: brake power in kW.

The amount of consumed fuel (\dot{m}_f) has been determined by the formula:

$$\dot{m}_f = \frac{(V_f . S)}{(1000.t)} \quad \dots (2)$$

Where; \dot{m}_f : consumed fuel in kg/sec.

V_f : consumed fuel volume in cm^3

S : Pure Gas Oil density in kg/m^3

t : calculated time of fuel consumption

(sec.)

In mixed fuel, S_{mix} . Substituted instead of S which has been evaluated

$$\text{as: } S_{mix} = \frac{S_{gasoil} + S_{alcohol} + S_{naphtha}}{3} \quad \dots (3)$$

Brake Specific Fuel Consumption (b.s.f.c.) in (kg/kw.hr) determined by:

$$b.s.f.c = \frac{(3600 \cdot \dot{m}_f)}{B.P.} \quad \dots (4)$$

In order to evaluate volumetric efficiency, actual & theoretical air consumption has been determined using the formulas:

$$\dot{m}_{air})_{act} = \frac{(12 \cdot \sqrt{h_o}) * \beta_{air}}{3600} \quad \dots (5)$$

$$\dot{m}_{air})_{theo} = \frac{(V_{sn} * N * \beta_{air})}{2 * 60} \quad \dots (6)$$

Where; \dot{m}_{air} : air consumption (kg/s)

$$L.C.V)_{mixture} = \frac{(0.9 * L.C.V_{gasoil} + 0.05 * L.C.V_{alcohol} + 0.05 * L.C.V_{naphtha})}{3} \quad \dots (9)$$

Brake & Indicated thermal efficiencies ($\eta_{b.th}$) & ($\eta_{i.th}$) have been calculated by:

$$\eta_{b.th} = \left[\frac{B.P}{Q_t} \right] * 100\% \quad \dots (10)$$

$$\eta_{i.th} = \left[\frac{I.P}{Q_t} \right] * 100\% \quad \dots (11)$$

Where;

$I.P.$: is indicated power which is equal to total sum of brake power $B.P.$ and friction power $F.P.$

Friction power can be determined by Willan's line, where the intersecting point with negative brake power axis represent friction power [9].

Mechanical efficiency can be found as:

$$\eta_m = \left[\frac{B.P}{I.P} \right] * 100\% \quad \dots (12)$$

The Air to Fuel ratio (A/F) can be determined by:

$$(A/F) = \frac{\dot{m}_{air})_{act}}{\dot{m}_f} \quad \dots (13)$$

β_{air} : air density in (kg/m^3), which is calculated by:

$$\beta_{air} = \frac{P_{atm}}{R * T_{atm}}$$

V_{sn} : Engine Capacity in cm^3

h_o : water altitude (mm H_2O), used to evaluate the air specific volume.

P_{atm} : atmospheric pressure (kPa)

T_{atm} : Air temperature ($^{\circ}\text{C}$)

R : air constant = 287 (J/kg.K)

The Volumetric Efficiency has been evaluated then after by:

$$\eta_v = \left[\frac{\dot{m}_{air})_{act}}{\dot{m}_{air})_{theo}} \right] * 100\% \quad \dots (7)$$

Total Power (Q_t) has been calculated by the formula:

$$Q_t = \dot{m}_f * L.C.V \quad \dots (8)$$

Where $L.C.V.$ is low calorific value (kJ/kg).

Lower calorific value in mixture case has been received according to ratio added to blending fuel as:

The temperatures of water entering and out from the engine also has been measured, using temperature gauges, in addition to exhaust temperature (T_{ex}).

4. RESULTS AND DISCUSSION

The relation between brake power and fuel consumption has been revealed in figure (2), which represent Willan's line. From this line, we can found friction power which are the friction losses between motive parts, oil pumps loss and cooling water in addition to suction and inject losses of air charge entered.

In case of pure gas oil fuel, this power increasing gradually with increasing the engine rotational speed, while in blending fuel figure (3) the increasing is not constant or not happened gradually. This was due to

fluctuating of engine speed especially at high speeds and low loads, so this lead to unstable increasing in friction power in case of blending.

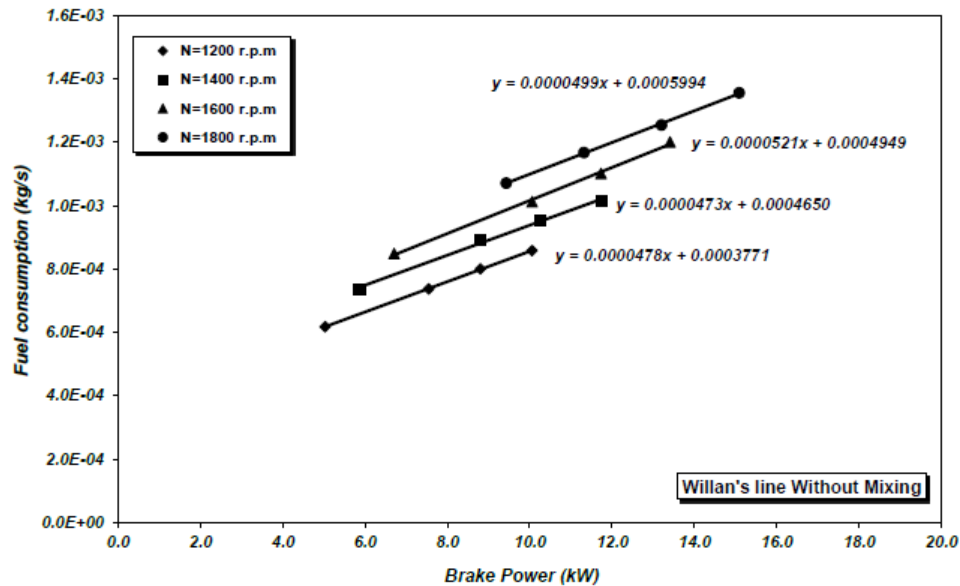


Fig. (2): relation between fuel consumption and brake power for pure gas oil fuel

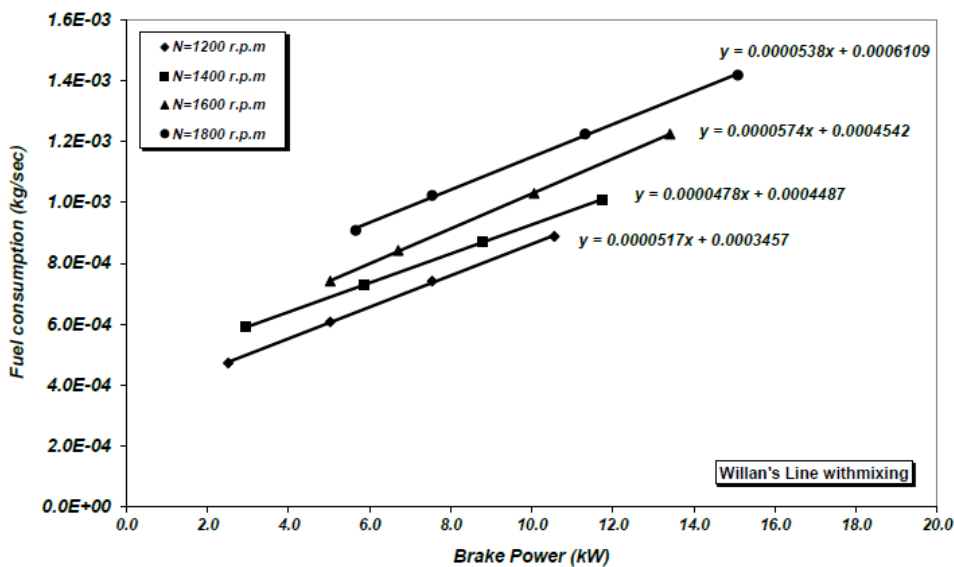


Fig. (3): relation between fuel consumption and brake power for mixing fuel

Performance curves at several speeds are drawn in one graph as shown in figures (4 and 5). The graph shows how specific fuel consumption

will change as the brake power changes at a specific speed which called fish hook curve [9].

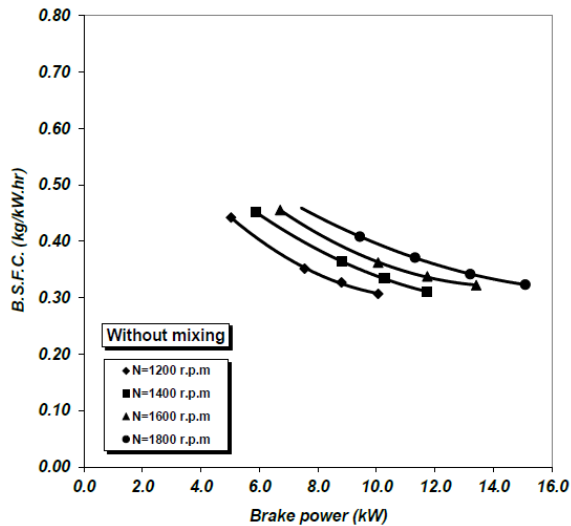


Fig. (4): Fish Hook relation between B.s.f.c. & B.P. for unmixed fuel and at all speeds

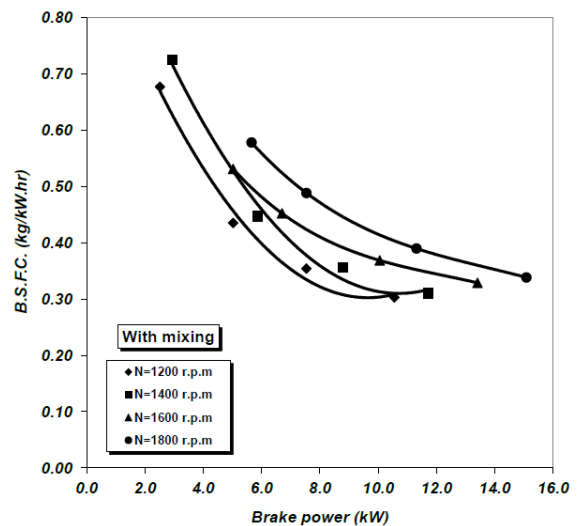


Fig. (5): Fish Hook relation between B.s.f.c. & B.P. for mixed fuel and at all speeds

Figures (6, 7, 8 & 9) revealed the relationship between brake power and efficiencies without mixing and figures (10, 11, 12 & 13) for mixing fuel. It seems clearly that the volumetric efficiency was stable in both cases and at the same speed because it doesn't depend on the fuel type entering the engine, but on the amount of air sucked by the engine. The curved relation of indicated thermal efficiency and brake power has

seen to increase with increasing brake power and it's larger than the brake thermal efficiency, and this is normally happened due to friction power losses that always cannot be exceeded. The mechanical efficiency seen to be increased with brake power and engine speed increasing, the mechanical efficiency is lower in case of blending fuel than that in case of pure gas oil fuel due to fluctuating in engine speed.

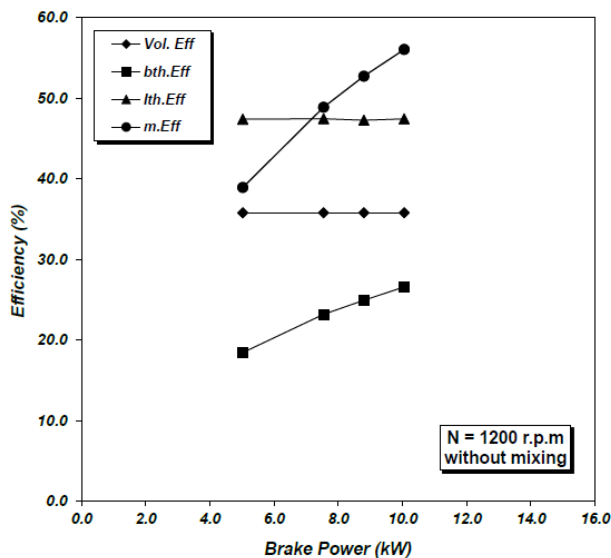


Fig. (6): relation between efficiency and brake power at N=1200 rpm without mixing

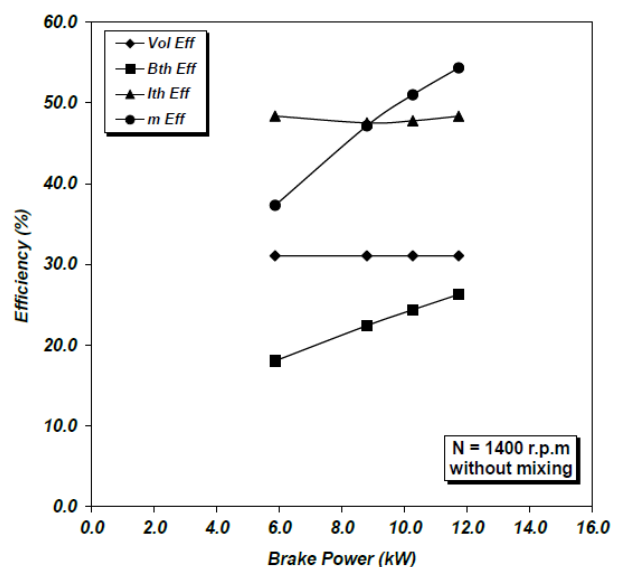


Fig. (7): relation between efficiency and brake power at N=1400 rpm without mixing

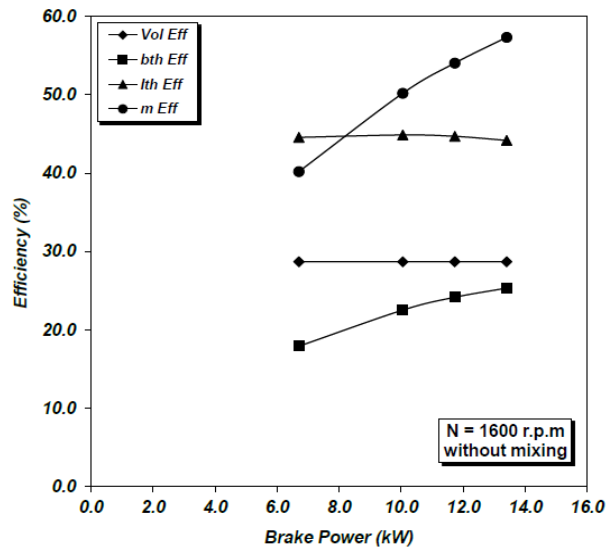


Fig. (8): relation between efficiency and brake power at N=1600 rpm without mixing

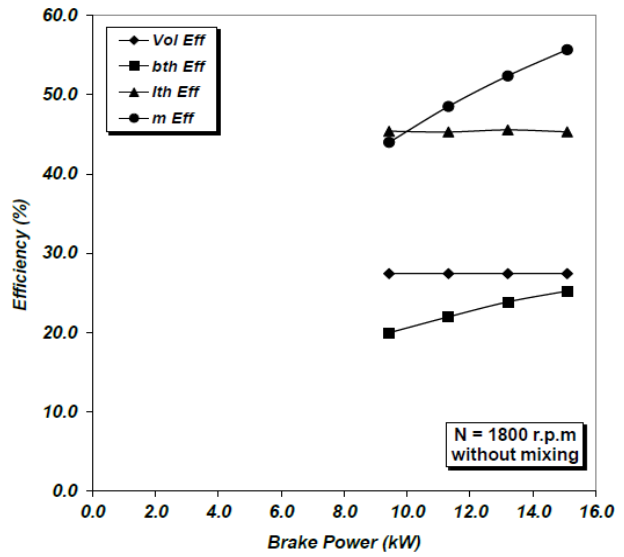


Fig. (9): relation between efficiency and brake power at N=1800 rpm without mixing

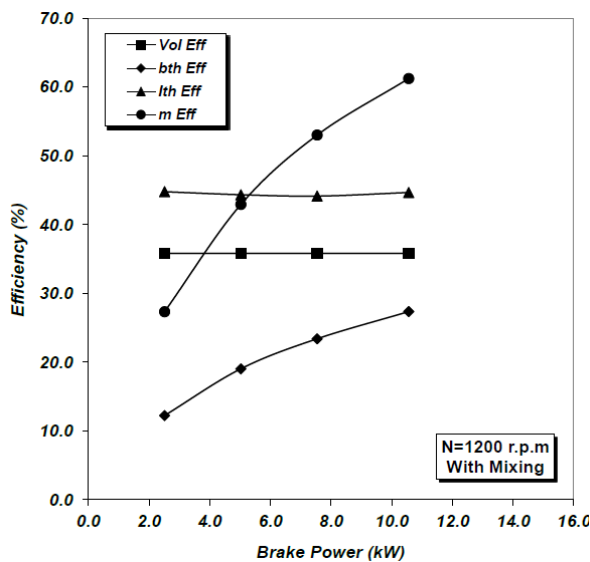


Fig. (10): relation between efficiency and brake power at N=1200 rpm with mixing

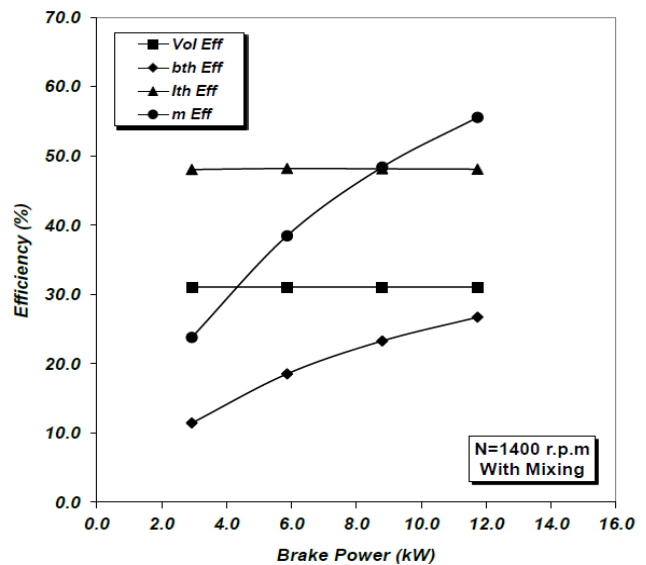


Fig. (11): relation between efficiency and brake power at N=1400 rpm with mixing

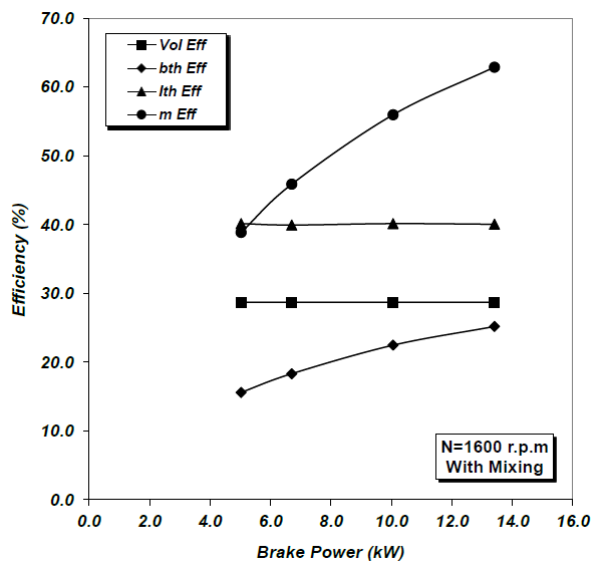


Fig. (12): relation between efficiency and brake power at N=1600 rpm with mixing

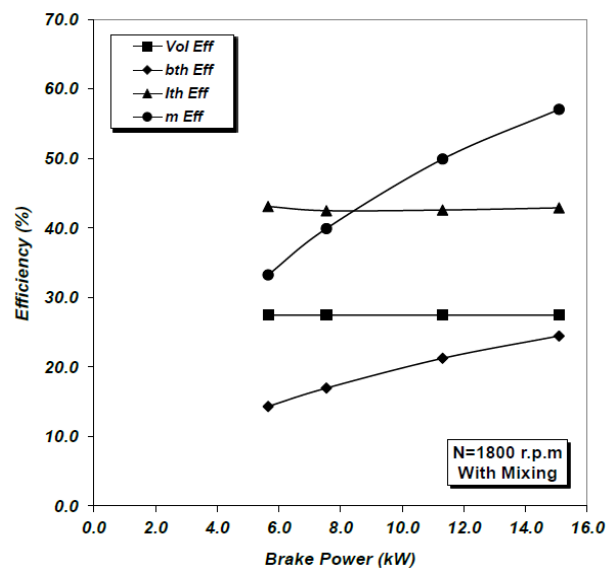


Fig. (13): relation between efficiency and brake power at N=1800 rpm with mixing

Figure (14) represents the relationship between brake thermal efficiency and engine speed which can tell an increasing of it in mixed fuel comparing with unmixed fuel. It can be seen that a slight improvement in the efficiency could be noticed when blending diesel fuel with ethanol. The improvement in the thermal efficiency is attributed to the physical and chemical changes

occurring in the combustion process, this yield to a slight gains in the thermal efficiency due to the increase in the ignition delay. This will produce a rapid rate of energy released which reduces the heat loss from the engine due to lack in time for heat to get out of the cylinder by heat transfer coolant [8].

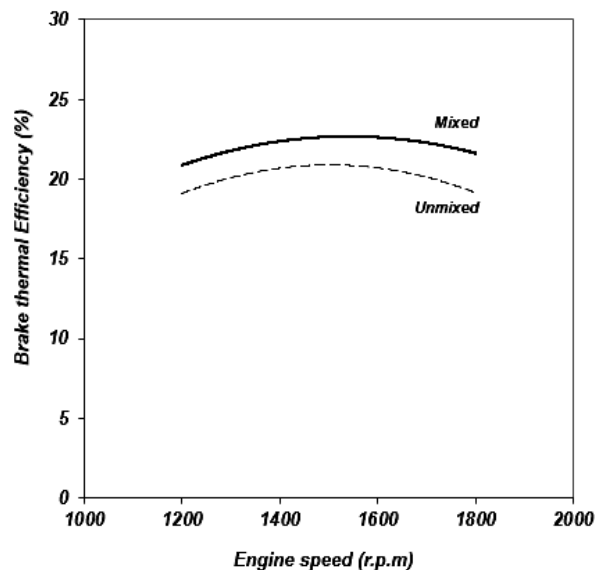


Fig. (14): relation between brake thermal efficiency and engine speed for both mixed and unmixed fuel

The relation between brake power and exhaust temperatures figure (15) has seen to be proportional, this is due to increasing the loads at a certain speed, because the amount of fuel required has increased to maintain the same engine speed. So this consequently lead to increase peak pressure and temperature inside engine combustion chamber during power stroke. Figures (17, 18, 19 & 20) indicate this relation for each speed separately and for both cases. In blending fuel figure

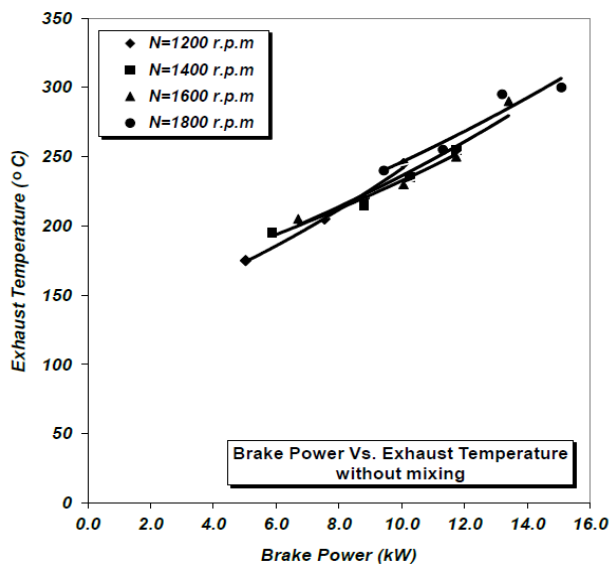


Fig. (15): relation between exhaust temperature and brake power for pure gas oil fuel.

Relations have been obtained of economic point through which we can reach less fuel consumption and larger power, this point is necessary to specify the engine performance. In mixing fuel, we found that the fuel consumption is lower than its value without mixing because of using larger amount of air to burn the mixture. This reflect the ability of mixing fuel to burn completely without fume. Figures (21, 22, 23 & 24) indicates the

(16), the exhaust gas temperatures observed to be higher at lower loads, but at medium and high loads, the exhaust gas temperatures is lower than in pure gas oil fuel. The reduction is due to higher latent heat of vaporization of blending elements and the quenching effect of alcohol in the combustion chamber. So the peak temperature and pressure inside combustion chamber in power stroke will be less than without mixing [5].

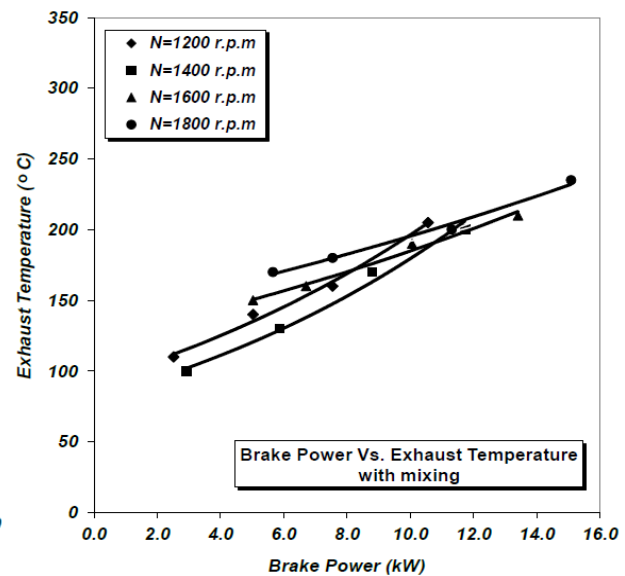


Fig.(16): relation between exhaust temperature and brake power for all speeds and with mixing fuels

relations between fuel consumption and brake specific fuel consumption (B.s.f.c).

Regarding A/F ratio in figure (25), it can be observed an increase of the ratio for mixing fuel and this ratio will decrease with increasing brake power resulted from increasing rotational engine speed as shown in figure (26). The reason of this decrease is due to decrease in volumetric efficiency of the engine.

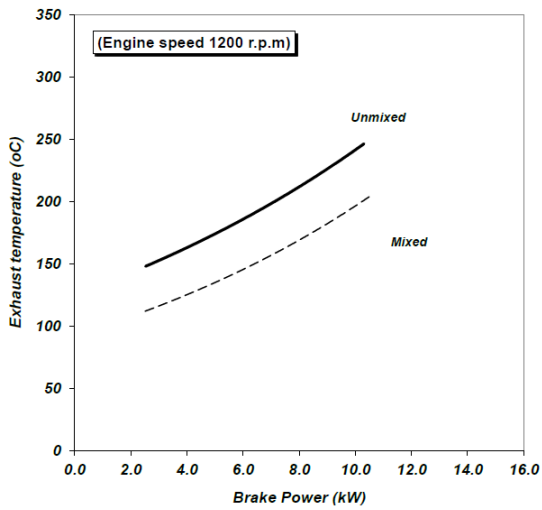


Fig.(17): relation between exhaust temperature and brake power for both mixed and unmixed fuel at N=1200 rpm

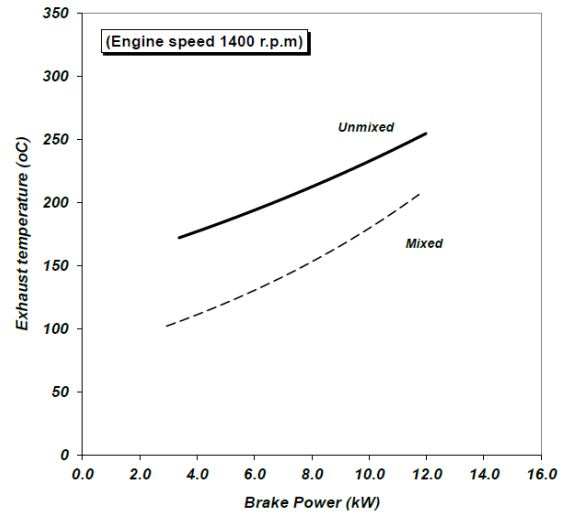


Fig.(18): relation between exhaust temperature and brake power for both mixed and unmixed fuel at N=1400 rpm

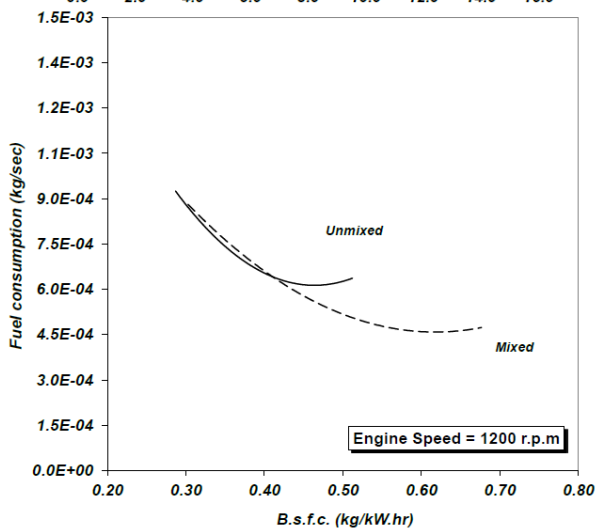
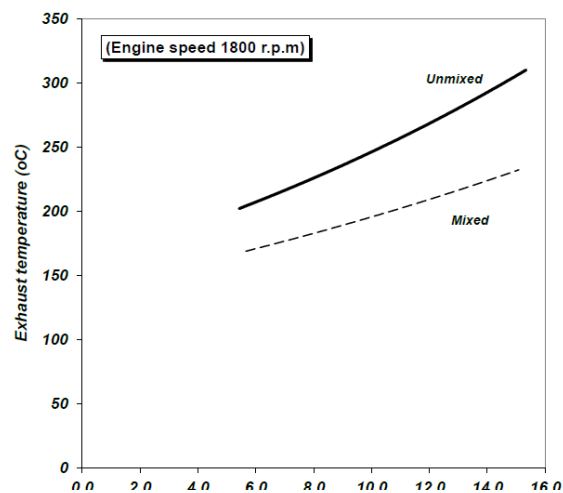
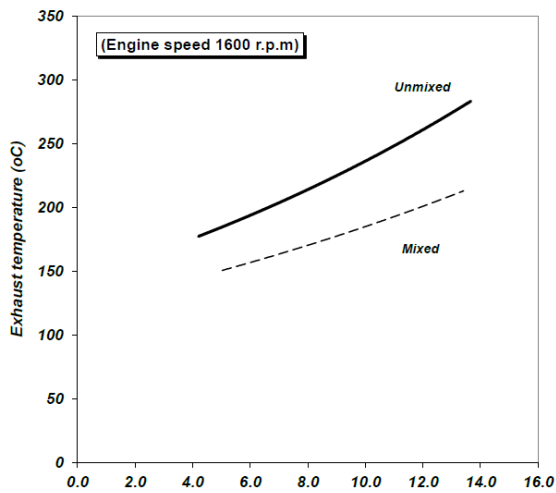


Fig. (21): relation between fuel consumption and brake specific fuel consumption for both mixed and unmixed fuel at N=1200 rpm

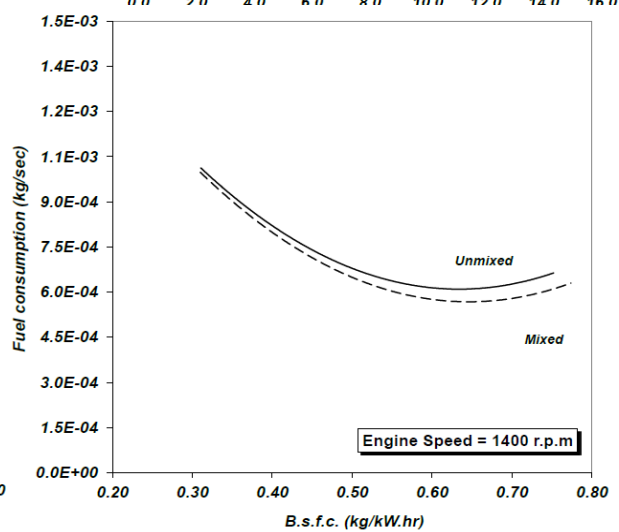


Fig. (22): relation between fuel consumption and brake specific fuel consumption for both mixed and unmixed fuel at N=1400 rpm

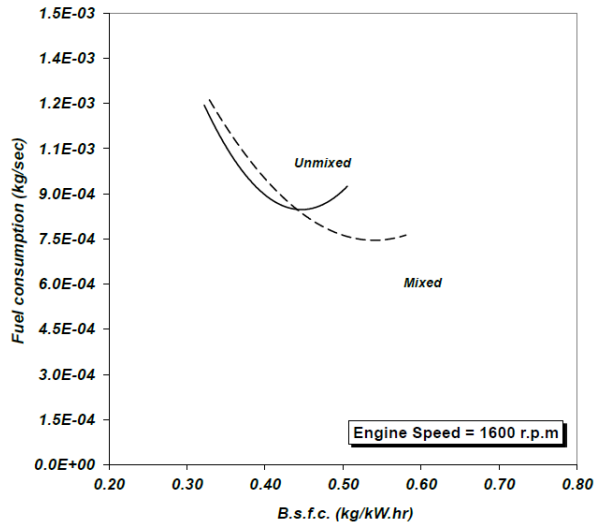


Fig. (23): relation between fuel consumption and brake specific fuel consumption for both mixed and unmixed fuel at N=1600 rpm

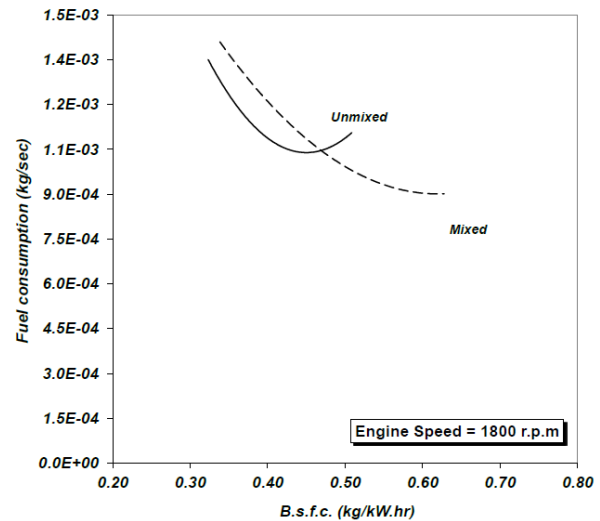


Fig. (24): relation between fuel consumption and brake specific fuel consumption for both mixed and unmixed fuel at N=1800 rpm

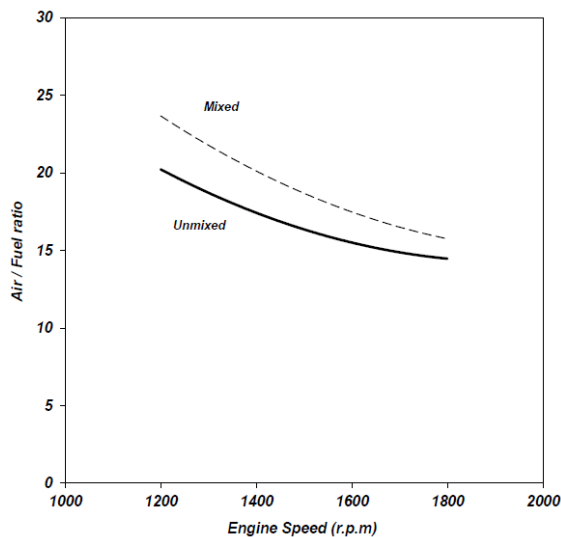


Fig. (25): relation between A/F ratio & engine speed for mixed and unmixed fuels

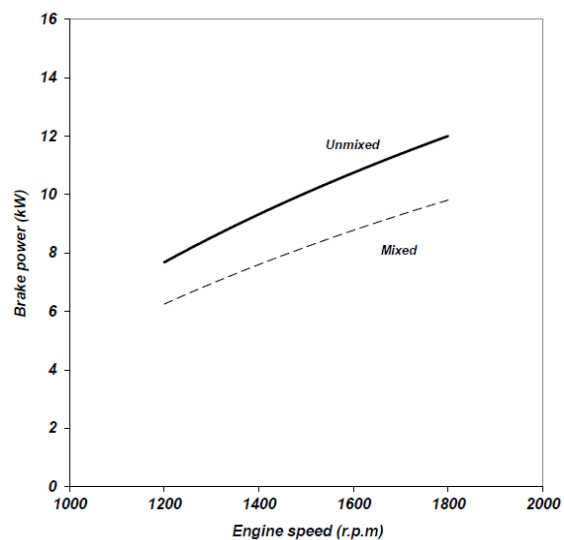


Fig.(26): relation between brake power and engine speed for both mixed & unmixed fuel

5. CONCLUSION

The general results obtained from the experiments can be summarized as follows:

- Addition of 5% Naphtha with 90% pure gas oil and 5% Ethanol are the best blending ratio for the mixture to get positive results.
- In mixing fuel by stirring and motion then leaving over for 48 hours will avoid separation of blending elements.
- Exhaust temperature is lower in mixed fuel than that of unmixed.
- The fuel consumption is lower in mixed fuel due to lower economic point resulted in which less fuel consumption against larger brake power can gain.
- The A/F ratio is higher in mixed fuel.
- Brake thermal efficiency is higher in mixed fuel and in all speeds.

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پۆختە

فەكولینا نوکە تێدا برا وەردی دەهێتە کەرن بجهئینانا بزویته ری ستونا کقاشنی کو کار دکەت ب سوتە مەنیا دیزالئ ناسایی دگەل سوتە مەنیا تیک کەل دیزل و کحولاً نیپانول ب ریکا کارکەرەکئ هاریکار کو ئەوژی (نغیا) کە کارکەرەکئ هاریکارەکو ریژا تیکەل بونئ ٥% نغیا دگەل ٩٠% سوتە مەنیا دیزلئ پاقەش و ٥% کحولاً نیپانول. تیبینی هاتە کرنا کو پلا گەرمانیا قە سنکریئ عادمی هاتە خارئ و هاتە خاراً مەزاختنا سوتە مەنیا جوری دگەل زیدەبو وزا بەرەههئەر ب ئەگر سوتنا تیکەلە ی ب رەنگەکئ تەمام، بو ئەگەرئ داکەفتنا خالا نابوو، زیدەباری زیدەبونا شیانین گەرمانیا میکە حی و ریژا هەوا / سوتە مەن دناف تیکەلەیدا دگەل زیرەبونا لە زاتیا زفرینا بزویته ری.

الخلاصة

البحث الحالي تم فيه مقارنة أداء محرك الاحتراق الانضغاطي عملياً الذي يعمل بوقود الديزل الاعتيادي، مع وقود خلط الديزل وكحول الايثانول بواسطة النفط كعامل مساعد. حيث تم خلط نسبة 5% نفثاً مع 90% وقود ديزل نقي و5% كحول ايثانول. تم ملاحظة انخفاض درجة حرارة غازات العادم وانخفاض استهلاك الوقود النوعي المصحوب بزيادة في الطاقة الناتجة بسبب احتراق الخليط بشكل كامل. أدى ذلك إلى انخفاض النقطة الاقتصادية، بالإضافة إلى ازدياد الكفاءة الحرارية المكبحة ونسبة الهواء/الوقود في الخليط مع زيادة سرعة دوران المحرك، مما يدل على تحسين أداء المحرك التشغيلي.