The Iraqi Journal For Mechanical And Material Engineering, Vol.13, No2, 2013

PRACTICAL INVESTIGATION OF PERFORMANCE OF SINGLE CYLINDER COMPRESSION IGNITION ENGINE FUELED WITH DUEL FUEL

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ABSTRACT

This paper studies the performance of a single cylinder compression ignition engine Ricardo E6, fueled with different blends of ethanol added to diesel.

The brake specific fuel consumption for dual fuel, in general, is higher than that for the diesel engine for low and medium loads; however, it becomes less than the diesel engine for high loads. The increase in the proportion of the ethanol in the mixture improves the specific fuel consumption at any specific load.

The brake thermal efficiency for diesel engine is higher than that for dual fuel engine for low and medium loads, while it becomes lower for high loads. In general, the addition of ethanol in the mixture improves the efficiency.

Retarding the injection timing of diesel fuel caused the specific fuel consumption to increase steadily to about 40% when the timing is too late. The thermal efficiency of dual fuel is lower, in general, than that of the diesel engine when retarding the injection timing.

Key words: duel fuel, ethanol, diesel, optimum injection timing, brake power, specific fuel consumption, brake thermal efficiency.

دراسة عملية لأداء محرك اشتعال بالانضغاط احادي الاسطوانة يعمل بوقود ثنائي

الخلاصة:

تضمن هذا البحث دراسة أداء محرك اشتعال بالانضغاط أحادى الاسطوانة نوع Ricardo E6، يعمل بوقود ثنائي بإضافة الأيثانول إلى وقود الديزل بنسب حجمية مختلفة، ومقارنة عمل الوقود الثنائي مع عمل محرك الديزل التقليدي في ظروف تشغيلية مختلفة.

إن استهلاك الوقود النوعي المكبحي للوقود الثنائي عموما أعلى من استهلاك وقود الديزل للأحمال المتوسطة والخفيفة، ولكنه يصبح أقل من وقود الديزل عند الأحمال العالية، وزيادة نسبة الأيثانول بالخليط الثنائي تحسن من الاستهلاك عند حمل معين. تكون الكفاءة الحرارية المكبحية للمحرك عند عمله بوقود الديزل أفضل منها لحالة عمله بالوقود الثنائي للأحمال المتوسطة والخفيفة، ولكن يتفوق محرك الوقود الثنائي على محرك الديزل عند العمل بأحمال عالية، إذ تتحسن كفاءته بشكل ملحوظ، وتحسن زيادة نسبة الأيثانول بالخليط من الكفاءة بشكل

عام. يزداد الاستهلاك الوقود النوعي المكبحي مع تأخير توقيت حقن الديزل عن التوقيت الأمثل للحقن تدريجيا، ليصل لحدود 40% أعلى من الاستهلاك عند التوقيت الأمثل، وتكون الكفاءة الحرارية للوقود الثنائي أقل عموما من محرك الديزل عند تأخير توقيت الحقن.

NOMENCLATURE

[air]	molar concentration of air	$m_{a,act}$	actual air flow rate
Bmep	brake mean effective pressure	$V_{s.n}$	displacement volume
Bp	brake power	Q_t	engine total heat
Bsfc	brake specific fuel consumption	Ø	Equivalence ratio
BTE	brake thermal efficiency	ρ_f	fuel density (kg/m^3)
CA	crank angle	v_f	fuel volume
[D]	molar concentration of diesel fuel		
EVF	Ethanol volume fraction		
[E]	molar concentration of ethanol		
HUCR	higher useful compression ratio		
LHV	lower heating value		
N	engine speed (rps)		
OIT	Optimum injection timing		
T	engine torque (Nm)		
TDC	top dead centre		

INTRODUCTION

The alcohols are fuels of the family of oxygenates. As is known to all, the alcohol molecule has one or more oxygen atoms, which contributes to the combustion. The alcohols are named accordingly to the basic molecules of hydrocarbon which derive from them: methanol (CH_3 OH); Ethanol (C_2H_5 OH); Propanol (C_3H_7 OH); Butanol (C_4H_9 OH). Theoretically, any of the oxygenic molecules of the alcohols family can be used as fuel. The list is somehow more extensive; however, only two of the alcohols are technically and economically suitable as fuels for internal combustion engines. These alcohols are those of the simplest molecular structure, i.e., methanol and ethanol. (Zappoli, 1991)

Ethanol, which is our interest in this paper, produced mainly from biomass transformation or bioconversion. It can be produced by synthesis from petroleum or mineral coal. Ethanol has been the alternative fuel chosen to replace petrol, due to the fact that alcohol is a renewable source of energy. Currently, ethanol is produced from sugar beets and from molasses (Dodge, 1998).

There are some important differences in the combustion characteristics of ethanol and hydrocarbons. Ethanol has higher flame speed and extended flammability limits (Ajav, 1997). Ethanol mixes in all proportions with water due to the polar nature of OH group. Low volatility is indicated by high boiling point and high flash point. Ethanol burns with no luminous flame and produces almost no soot (Mendez, 2006).

Combustion of ethanol in presence of air can be initiated by an intensive source of localized energy, such as a flame of a spark, and also, the mixture can be ignited by application of energy by means of heat and pressure, such as happens in the compression stroke of a piston engine. The energy of the mixture reaches a sufficient level for ignition to take place after a brief period of delay called ignition delay, or induction time, between the sudden heating of the mixture and the onset of ignition (formation of a flame front which propagates at high speed through out the whole mixture) (Hiroyasu, 1998). The high latent heat of vaporization of ethanol cools the air interring the combustion chamber of the engine, thereby increasing the air density and mass flow. This lead to increased volumetric efficiency and reduced compression temperatures. Together with the low level of combustion temperature, these effects also improve the thermal efficiency by 10% (Nagaraju, 2008).

The higher flame speed, gives earlier energy release in the power stroke, which results in a power increase of 11% at normal conditions and up to 20% at the higher levels of compression ratios (Hansen, 2005).

For a fuel to burn in a diesel engine, it must have a high cetane number or ability to self ignite at high temperatures and pressures. There exist a signification difference among gasoline, diesel and ethanol in term of cetane number and auto ignition. A high cetane number leads to a short ignition delay period. While a low cetane number results in a long ignition delay period (Myo, 2008). Ethanol has lower cetane number than that of diesel engines, which is not desired when diesel engine are converted to ethanol. Fortunately, some additives, an example of which is nitrogen glycol, can increase the cetane number of ethanol. This mean that the ignition delay period will become short which will reduce the tendency to cause a diesel knock. However, too short ignition delay period will cause a lower rate of heat release which is not wanted (Schaus, 2000).

There are many techniques by which ethanol can be used as a fuel in compression ignition engines as reported by (Brusster, 2002). The methods of using alcohol in internal combustion engines were developed from traditional figures to recent technologies. These methods can be abstracted as: solutions, emulsions, fugmations, dual injection, spark assisted and ignition improvers. These methods are the most interesting and acceptable, most of it do not achieve total change of diesel fuel, except in spark ignition and using of ignition improvers, the solution and emulsion can replace 25% of diesel fuel, the fumigation reached to about 50%, and for dual injection it become close to 90% of diesel fuel requirements (Williams, 2006).

For each method from the former there are advantages and disadvantages. This is related with the output power and exhaust gases and the required modification on engine parts. Also its applications gave a changeable degree of success, this depend on the engine type, and the alcohol used.

From the above it is clear that alcohol fuels are not suitable for Otto engines only, but its properties made it a good fuel to diesel engine also (Agrawal, 2007).

The main objective of the present research is to investigate practically the effect of ethanol addition in variable proportions to Iraqi conventional diesel fuel on the engine performance.

EXPERMINTAL SETUP

Apparatus

A single cylinder, naturally aspirated, four stroke, variable compression ratio, injection timing and speed Ricardo E6 was used in this study. Further details regarding this engine are given in **table 1**.

Diesel fuel (cetane No.= 48.5) was supplied to the injection system while ethanol was supplied to Alcohol carburetor type Zenith WIP supplied with a choke, 26 mm, consists of a main variable spray, supplied with needle valve to control the alcohol flow rate through spray opening.

Fig. 1 represents schematic layout of the test set-up, while **Fig. 2** shows a photographic picture of the engine and its accessories.

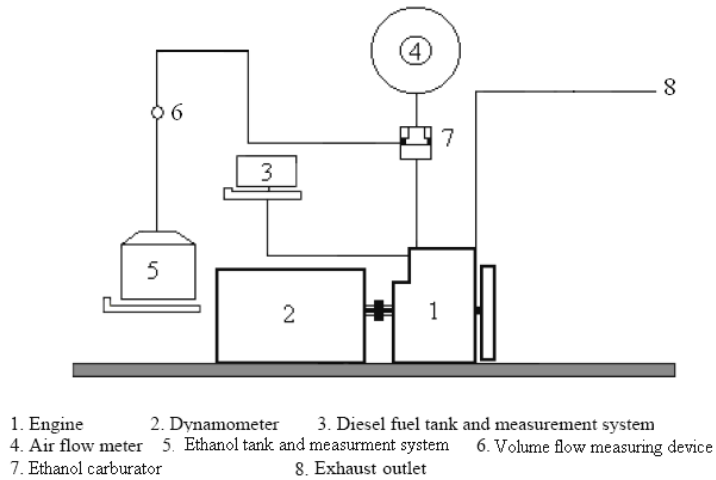

Fig. 1, Schematic of experiment setup

Fig. 2, photographic picture of the used engine and its accessories

Methods

Different types of test were conducted during this investigation. In the first set the engine were run at speed (25 rps) with diesel fuel alone, and the higher useful compression ratio (HUCR) for this fuel was defined using optimum injection timing (OIT) and full load for each tested point. At this HUCR engine performance was studied in details, using supplementary ethanol to diesel fuel, to find the effect of engine speed, injection timing and equivalence ratio on engine performance.

The engine was run with diesel alone and the load was changed to the required load while keeping engine speed constant at (25 rps). Diesel fuel consumption was reduced in several volumetric quantities (10, 20, 25, and 30%). Naturally, engine speed and load will reduce. Alcohol (ethanol 99.9% pure) was allowed to enter the combustion chamber with air until the engine reached the required load and speed. Then the term 90% diesel fuel means 90% from the required (100%) diesel mass that ensure specific load and speed. At the same time, 10% ethanol doesn’t mean the same meanings as diesel. It means a mass of ethanol required to compensate the desired 10% diesel fuel energy that makes the engine runs at required load and speed. So, for every specific load and speed there will be $m_{fuel}= m_{diesel}+m_{ethanol}$. By measuring air mass flow rate (m_{air}), then the actual fuel/air ratio is defined as:

$$\text{Actual fuel/air ratio} = m_{fuel} / m_{air} = (m_{diesel} + m_{ethanol}) / m_{air} \quad (1)$$

The stoichiometric fuel/ air ratio was calculated, and then equivalence ratio can be defined as (Abdul Haleem, 2007):

$$\emptyset = \frac{\frac{[D]}{[air] - \frac{[E]}{([E]/[air])_{st}}}}{\left(\frac{[D]}{[air]}\right)_{st}} \quad (2)$$

The following equations were used in calculating engine performance parameters (Keating, 2007):

1- Brake power

$$bp = \frac{2\pi * N * T}{60 * 1000} \quad kW \quad (3)$$

2- Brake mean effective pressure

$$bmep = bp \times \frac{2*60}{V_{sn} * N} \quad kN/m^2 \quad (4)$$

3- Fuel mass flow rate

$$\dot{m}_f = \frac{v_f \times 10^{-6}}{1000} \times \frac{\rho_f}{time} \quad {}^{kg}/_{sec} \quad (5)$$

4- Air mass flow rate

$$\dot{m}_{a,act.} = \frac{12\sqrt{h_o * 0.85}}{3600} \times \rho_{air} \quad \frac{kg}{sec} \quad (6)$$

$$\dot{m}_{a_{theo.}} = V_{s.n} \times \frac{N}{60*2} \times \rho_{air} \quad \frac{kg}{sec} \quad (7)$$

5- Brake specific fuel consumption

$$bsfc = \frac{\dot{m}_f}{bp} \times 3600 \quad \frac{kg}{kW.hr} \quad (8)$$

6- Total fuel heat

$$Q_t = \dot{m}_f \times LCV \quad kW \quad (9)$$

7- Brake thermal efficiency

$$\eta_{bth} = \frac{bp}{Q_t} \times 100 \quad \% \tag{10}$$

Error analysis

All measurements have some degree of uncertainty that may come from a variety of sources. The process of evaluating this uncertainty associated with a measurement result is often called uncertainty analysis or error analysis. The complete statement of a measured value should include an estimate of the level of confidence associated with the value. The experimental accuracies of the measuring devices that were used in present study are shown in table 2.
The uncertainty in the results is calculated by the equation (ASHREA, 1986):

$$e_R = \left[\left(\frac{\partial R}{\partial V_1} e_1\right)^2 + \left(\frac{\partial R}{\partial V_2} e_2\right)^2 + \cdots + \left(\frac{\partial R}{\partial V_n} e_n\right)^2\right]^{0.5}$$

Where:
e_R : Uncertainty in the results
R : a given function of the independent variables $V_1, V_2, \ldots, V_n$ or $R=R(V_1, V_2, \ldots, V_n)$.
e_i : uncertainty interval in the nth variable.
The partial derivative $\frac{\partial R}{\partial V_1}$ is a measure of the sensitvity of the result to a single variable.
The uncertainty for present tests was:

$$e_R = [(1.2)^2 + (1.7)^2 + (0.46)^2 + (1.1)^2 + (0.76)^2 + (0.9)^2]^{0.5} = \pm 2.67\%$$

RESULTS and DISCUSSION

To evaluate Ricardo engine performance with divided chamber, when it fueled with dual fuel consisted of traditional diesel fuel, and ethanol alcohol mixture with different volumetric proportion. The results were compared to engine fueled with diesel fuel only, to conclude the effect of different factors on the engine performance by studying the operating variables.

Equivalence Ratio Effect

The first set of experiments was to explore the HUCR for used diesel fuel, at optimum injection timing (OIT) and maximum load for each point, from **fig. 3**, HUCR for diesel fuel used was 17.7:1. Brake power (bp) increased with CR increase till 17.7:1. Then it decreased due to knock occurrence, which influences to retard injection timing (IT), reducing resulted brake power (bp). This HUCR is used as reference CR for all experiments conducted.

Fig. 4 represents the effect of ethanol addition on bp resulted from engine, at HUCR, 25 rps and OIT. Increasing ethanol proportion increases bp to a limit (25%). At ethanol rate = 30%, bp increased at very lean equivalence ratios less than Ø=0.5, and decreased at Ø>0.5. Increasing ethanol fraction (EF) extends ignition limits (from equivalence ratio Ø=0.43 for diesel fuel, to Ø= 0.38 with EF=30%, at lean side) and this appears good modification in very lean side compared with neat diesel fuel.

The optimum injection timing (OIT) retarded with mixture enrichment, but it advanced for duel fuel compared with diesel fuel alone, about 5°CA with EF=25%, as shown in **fig. 5**. Ethanol fuel needs more time to evaporate due to its high vaporization temperature. Then, using ethanol means advancing IT to get the best engine performance.

The volumetric efficiency reduced with fuel mixture enrichment, due to fuel increment without increasing inner air, but it is improved with EF increase, as **fig. 6** shows. This increase is due to OH molecule in ethanol chemical formation. Also, introducing ethanol in incoming air reduces its temperature which enhances air density.

The indicated thermal efficiency is improved with ethanol fraction increase till EF=25%, as **fig. 7** indicates. After this limit it begins to reduce due to knock appearance which influences to reduce load and retard injection timing. Which means that part of fuel will burn after TDC, during expansion stroke, reducing resulted bp from the engine? The increase of alcohol proportion in dual fuel mixture modified the efficiency, due to the homogenous mixture of alcohol and air. Ethanol evaporation reduces air temperature and made its density higher, bushing more mass into combustion chamber. This increased fuel at combustion starting, and improved combustion processes characteristics, because of excess oxygen made combustion came nearly complete with no smoke.

Figure 8 represents the effect of EF on bsfc, it is clear that ethanol addition reduces bsfc to 25% limit, at EF=30%, bsfc reduced for very lean equivalence ratios less than Ø=0.5, and increases for other equivalence ratios. Excess air quantity and its increasing density by ethanol evaporation at entering manifold makes ethanol addition improved specific fuel consumption. For low percentage of ethanol the consumption is high, because of the need to compensate the differences in heating values between fuels, and it becomes better when the ethanol percentage becomes higher in mixture. Ethanol fraction can be increased more, when using suitable modification on alcohol vaporization system.

Exhaust gas temperature increased with equivalence ratio increase, but it reduced with increasing EF till 25%, as **fig. 9** shows. Increasing equivalence ratio means more fuel enrichment while the entering air is constant, resulted in increasing exhaust gas temperatures for all blends. While the reduction in these temperatures with increasing EF to 25% compared to neat diesel fuel, is due to two reasons. First a part of diesel fuel is replaced with ethanol fuel which has lower heating value. This means the resultant heat inside combustion chamber will be reduced comparing with operating with diesel fuel alone. Second the efficient burning of fuel – air mixture gives efficient cooling to burning gases at expansion stroke. At 30% ethanol fraction exhaust gas temperatures increased, due to knock existence which influences the injection timing (IT) to be retarded, so part of the mixture continued burning in expansion stroke, and the rejected gases got out with these high temperatures.

Effect of Engine Speed

Fig. 10 represents the effect of engine speed on maximum bp at each EF. Maximum bp increases with speed increase from low to medium speeds with about 7%, but at high speeds the resultant maximum bp reduced due to reduction in volumetric efficiency.

Volumetric efficiency varied with engine speed and ethanol fraction as **Fig 11** represents. The maximum volumetric efficiency increased by increasing engine speed from low speeds (20 rps) to medium speeds (25-30 rps). At high speed (35 rps) volumetric efficiency falls down. Adding

ethanol at low proportions (5-10%) reduced volumetric efficiency, but high proportions (25-30%) improves volumetric efficiency. Increasing ethanol fraction means increasing the oxygen content inside the combustion chamber which improves the volumetric efficiency. In the same time it takes a part of air share. The resulted volumetric efficiency is the outcome of these two parameters.
OIT retarded with ethanol addition, it was retarded with about 3°BTDC for medium speeds and 6°BTDC for high speeds. Except for EF=30%, where the IT retarded apparently, as shown in **Fig. 12** due to its knock tendency.

Increasing engine speed increased bsfc, as **Fig. 13** represents. Increasing engine speed means injecting more fuel causing higher bsfc. Adding ethanol reduced bsfc about 1.3% for each step, from 10 ro 25% ethanol. At 30% ethanol bsfc increased about 1.4% compared to diesel alone. Combustion improvement with ethanol addition till 25% caused bsfc to reduce with each ethanol adding step. Combustion deterioration while adding 30% ethanol caused the bsfc increments.

Effect of Injection Timing

The maximum bp is at OIT, when this IT advanced or retarded bp reduced, as appears in **fig. 14**. IT effects appears at equivalence ratios that gives maximum bp (Ø=0.575), and its effect is less clearly at other equivalence ratios. The late injection causes low pressure rate rise, because the injected fuel became near the top dead center with no available time for preparation, which made it burned after the piston came down in expansion stroke, and reduced the maximum cylinder pressure. This operation required pushing more fuel into combustion chamber to compensate the lost unburned exhausted fuel, which increases the brake specific fuel consumption as **fig. 15** represents.

Bsfc increases with retarding or advancing IT, the lower resulted bsfc was at OIT, as **fig 15** shows. There was a compound effect for dual fuel engine in elongation delay period, resulted from alcohol existence in the entering charge. So, the cooling effect of evaporating ethanol will increase the delay period, and delay pressure rise inside cylinder. The ethanol evaporation in suction air reduced its temperature and pressure, and increased the ethanol quantity drown from carburetor as well as air. Advancing IT caused the maximum pressure directed on the piston to take place near top dead center, causing rapid combustion, with high pressure rates.

CONCULSIONS

1. Brake specific fuel consumption for dual fuel engine is higher; in general, than that for traditional diesel engine, for the most of the studied loads range, except for high loads it will be less. In general, the increment was about 3% at optimum timing until 6% at very late injection timing.
2. The increase of ethanol proportion in the dual mixture for specific load improved specific fuel consumption, due to ethanol cooling effect which increases air density and incoming air mass.

3. Ethanol addition gave better brake thermal efficiency than neat diesel fuel engine, but for 30% ethanol addition brake thermal efficiency improved at very lean equivalence ratios and reduced for higher ratios.
4. Total inner energy required producing desirable output power rises for low ethanol percentage, and reduced gradually with ethanol percentage increase, the improvement becomes higher with increasing the applied load.

5. Brake specific fuel consumption for dual fuel rises gradually with retard injection timing from the engine optimum timing. It will reach very late injection timing at about 40% higher than consumption at optimum timing.
6. The thermal efficiency for dual fuel engine was less than that for diesel engine at highly retarded timing, but it became better by advancing IT toward optimum injection timing, it was the best at this timing.

Table 1: Ricardo E6 engine geometry and operating parameters.

Value	Description
Displaced Volume	504 cm^3
Bore	76.2mm
Stroke	111.1mm
Exhaust Valve Open	43° BBDC (at 5 mm lift)
Exhaust Valve Close	6° ATDC (at 5 mm lift)
Inlet Valve Open	8° BTDC (at 5 mm lift)
Inlet Valve Close	36° ABDC (at 5 mm lift)
Speed	1500 RPM (25 rps)

Table 2, Experimental Accuracies

Measurements	Accuracies in this study
Thermocouples	± 1.2 %
Engine speed tachometer	± 1.7%
Diesel fuel flow meter	± 0.46 %
Air flow meter	± 1.1 %
Ethanol fuel flow meter	± 0.76%
dynamometer	± 0.9%

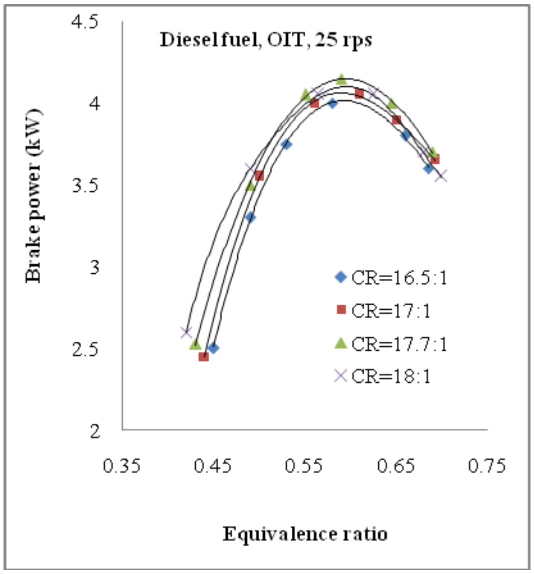

Fig3, Compression ratio effect on brake power for wide range of equivalence ratios, at 25 rps, full load and OIT

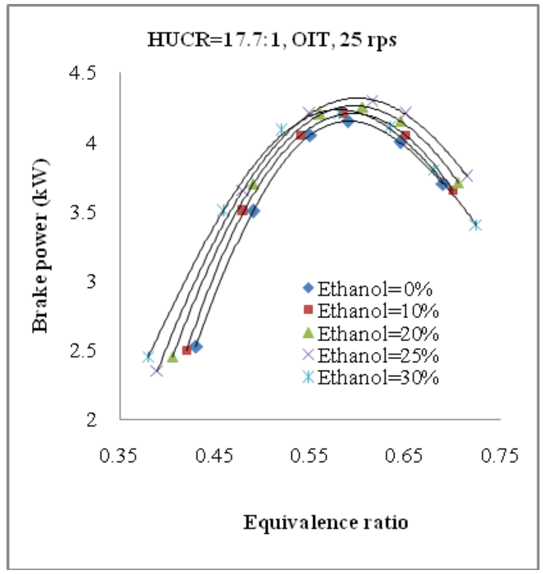

Fig4, Ethanol addition effect on brake power for wide range of equivalence ratios, at HUCR, 25 rps, full load and OIT

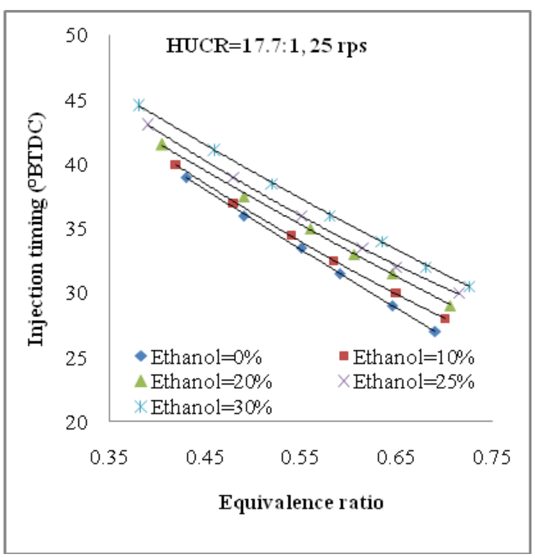

Fig. 5, Ethanol addition effect on injection timing for wide range of equivalence ratios, at HUCR, 25 rps and full load

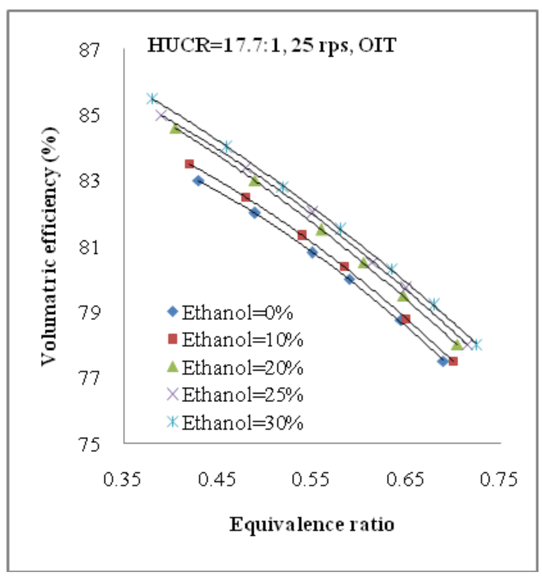

Fig. 6, Ethanol addition effect on volumetric efficiency for wide range of equivalence ratios, at HUCR, 25 rps, full load and OIT

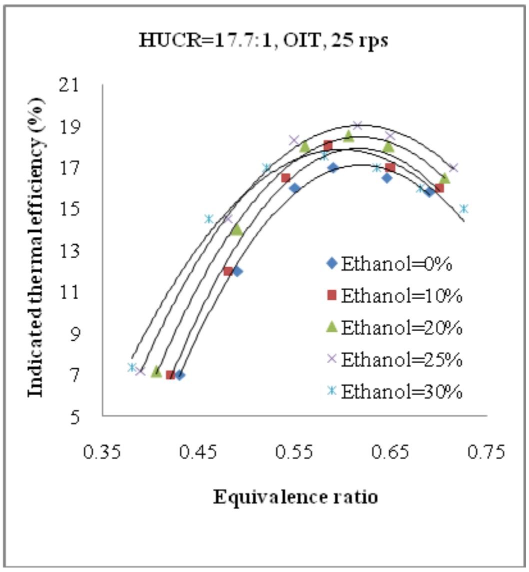

Fig7, Ethanol addition effect on indicated thermal efficiency for wide range of equivalence ratios, at HUCR, 25 rps, full load and OIT

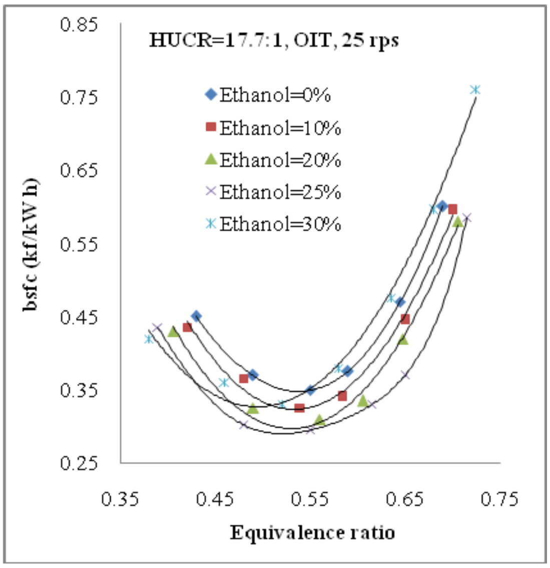

Fig 8 Ethanol addition effect on brake specific fuel consumption for wide range of equivalence ratios, at HUCR, 25 rps, full load and OIT

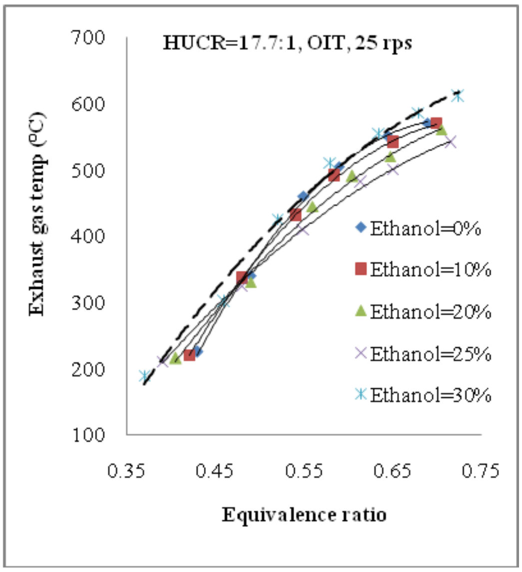

Fig. 9, Ethanol addition effect on exhaust gas temperatures for wide range of equivalence ratios, at HUCR, 25 rps, full load and OIT

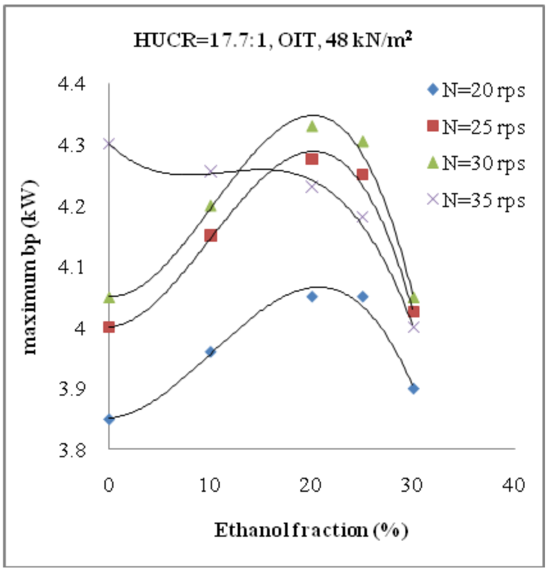

Fig. 10, Ethanol addition effect on maximum brake power for variable engine speed, at HUCR, full load and OIT

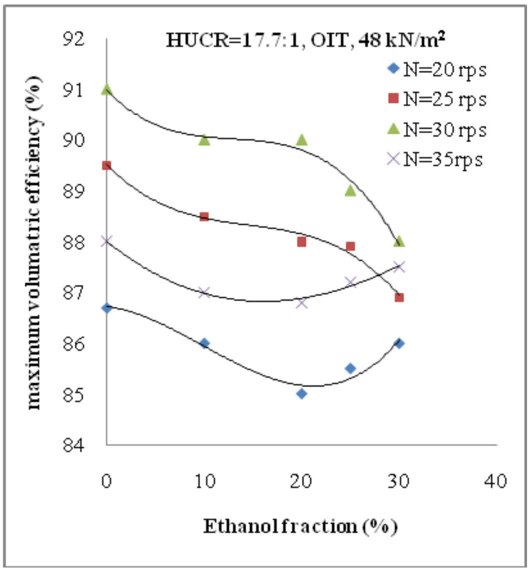

Fig. 11, Ethanol addition effect on maximum volumetric efficiency for variable engine speed, at HUCR, and full load

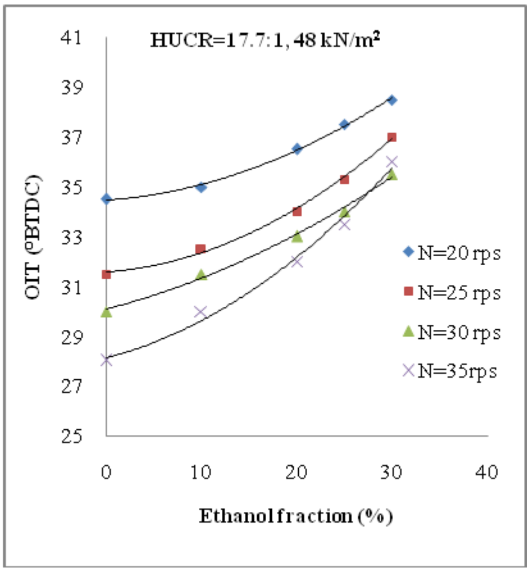

Fig. 12, Ethanol addition effect on optimum injection timing for variable engine speed, at HUCR, and full load

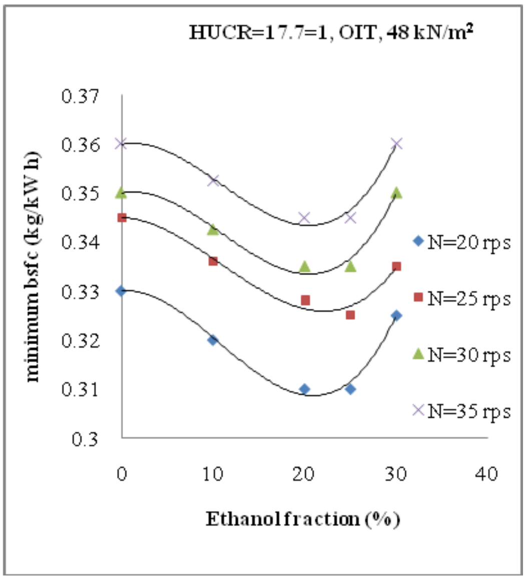

Fig. 13, Ethanol addition effect on minimum bsfc for variable engine speed, at HUCR, full load and OIT

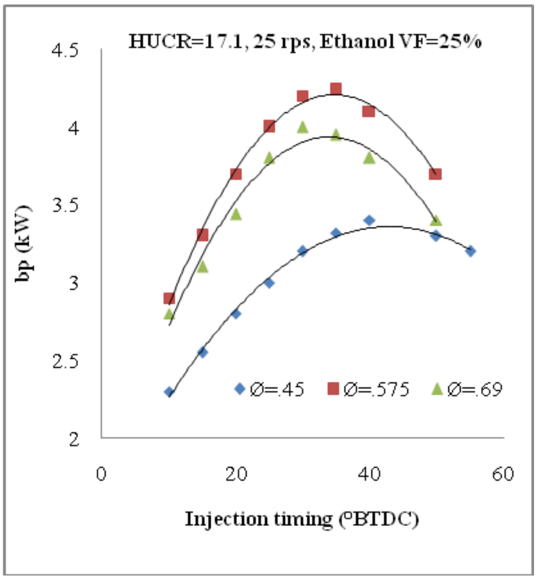

Fig. 14, Ethanol addition effect on bp for specific equivalence ratios, at HUCR, 25 rps and full load

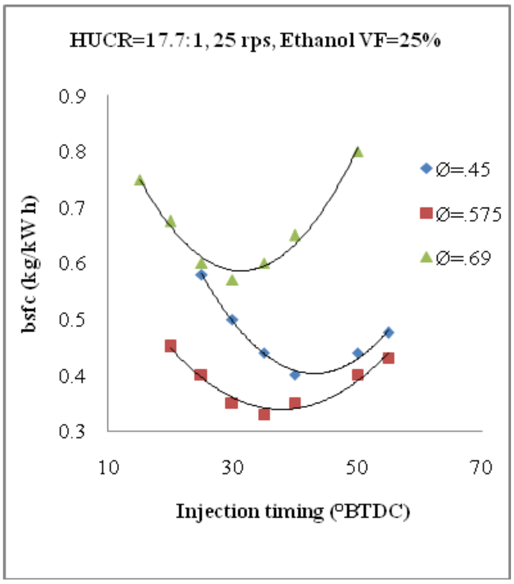

Fig. 15, Ethanol addition effect on bsfc for specific equivalence ratios, at HUCR, 25 rps and full load

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