EGR effect on performance of a spark ignition engine Fueled with blend of methanol-gasoline

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Abstract

This paper examines the results of performance of a single cylinder spark- ignition engine fuelled with 20% methanol +80% gasoline (M20), compared to gasoline. The experiments were conducted at stoichiometric air–fuel ratio at wide open throttle and variable speed conditions, over the range of 1000 to 2600 rpm. The tests were conducted at higher useful compression ratio using optimum spark timings and adding recirculated exhaust gas with 20% to suction manifold.

The test results show that the higher compression ratio for the tested gasoline was 7:1, 9.5:1 for M20 and 9:1 for M20 with added EGR. M20 at higher useful compression ratio (HUCR) and optimum spark timing (OST) characteristics are significantly different from gasoline. Within the tested speed range, M20 consistently produces higher brake thermal efficiency by about 6%. Also it resulted in approximately 3.06% lower brake specific fuel consumption compared with gasoline. Adding EGR to M20 caused reduction in HUCR and advancing the OST. This addition increased brake specific fuel consumption (BSFC), reduced brake thermal energy, volumetric efficiency and exhaust gas temperatures.

Keywords: Methanol; Compression ratio; Specific fuel consumption; Exhaust gas temperature, optimum spark timing.

تأثير تدوير الغاز العادم EGR على أداء محرك اشتعال بالشرارة أحادي الأسطوانة يعمل بخليط من الميثانول-الكازولين

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الخلاصة

تختبر هذة الدراسة نتائج اداء محرك اشتعال بالشرارة احادي الأسطوانة جهز بميثانول بنسبة 20% + كازولين 80% (M20) ، ومقارنتها بالكازولين. تمت التجارب عند نسبة مكافئة مثالية للهواء-الوقود وبفتحة خنق مفتوحة تماما وبظروف سرع محرك متغيرة بمجال من 1000 rpm لغاية 2600 rpm . كما تمت التجارب عند نسبة الانضغاط النافعة العليا وباستخدام توقيت أمثل للشرر وباضافة غاز عادم مدور بنسبة 20% لمشعب الدخول تبين النتائج أن نسبة الأنضغاط النافعة العليا للجازولين المستخدم كانت 7:1، اما نسبة M20 فكانت 9.5:1، أما بالنسبة لخليط M20 مع تدوير الغاز العادم فكانت نسبة الأنضغاط النافعة العليا له 9:1. يمتلك M20مواصفات نسبة انضغاط نافعة عليا وتوقيت أمثل للشرر تختلف عن الكازولين. فخلال مجال السرع المدروسة، انتج M20 كفاءة حرارية مكبحية أعلى بحدود 6% ، كما نتج عنه نقصان بحدود 3.06% للأستهلاك النوعي المكبحي للوقود مقارنة بالكازولين. تسببت اضافة EGR الى M20 انخفاضا بنسبة الأنضغاط النافعة العليا وتقديما لتوقيت الشرر المثل. كما زادت هذة الأضافة من استهلاك الوقود النوعي المكبحي، وقللت الكفاءة الحرارية المكبحية و الكفاءة الحجمية ودرجات حرارة غازات العادم .

Introduction

In recent decades, greater emphasis has been made to improve the fuel economy and reduce the tailpipe emissions from vehicles due to the concerns of energy supply and global warming **[1]**. Alcohol fuels, in particular methanol, resulted in advanced auto-ignition and faster combustion than that of gasoline. In addition, their use could lead to substantially lower HC, NOx and CO exhaust emissions **[2]**. In transportation, methanol is used as a vehicle fuel by itself, blended with gasoline, or as a gasoline octane enhancer and oxygenate. There is little doubt that methanol can improve the overall energy efficiency of the vehicle fleet **[3]**. Many experimental studies have confirmed that methanol, especially high-percentage-methanol fuels or neat methanol, in gasoline engines increases engine efficiency, torque, and power compared to baseline gasoline tests, mainly because of a superior fuel octane rating **[4]**.

Some methanol properties are attractive as an engine fuel. The initial boiling point of methanol (63 ^oC) is much closer to gasoline (32.8 ^oC). Its density (913.2 kg/m^3 at 20 ^oC) and its flash point (-22^oC), which would also overcome the cold engine start problems usually associated with bio-ethanol **[5]**. A little is known about the combustion of methanol. Some researchers found that methanol is more robust to cold engine starts than ethanol due to higher rates of vaporization and higher combustion stabilities. The knock suppression ability of methanol was shown to be superior to gasoline, which would support the use of higher compression ratio SI engines in the drive for greater efficiencies **[6 & 7]**.

One of the possible ways of using methanol is to mix it in certain proportions with gasoline to improve its qualities. This has been the subject of extensive research for many years **[8]**. **Pearson et al [9]**, **Ozsezen et al [10]** & **Bromberg et al [11]** have investigated the suitability of methanol as a fuel, especially for heavy duty commercial vehicles using both spark and high compression ratio as means for initiating combustion. They came to a basic conclusion that it is suitable for both the systems and for certain heavy-duty applications. They observed that by using 15% methanol-gasoline blend, the engine performance curves (brake power, mechanical efficiency, and thermal efficiency) showed reduction. **Koenig et al [12]** studied the technical and economical aspects of methanol as an automotive fuel. From their results conducted on a single-cylinder engine and using methanol-gasoline blend as alternative fuel for motor vehicles, they found that the utilization of antiknock effect of methanol could lead to competitive gasoline-methanol blend vehicle operation at the present cost of gasoline and methanol.

Exhaust gas recirculation (EGR) is a strategy employed in many modern gasoline engines to reduce NOx emissions; it involves recirculating a fraction (5–30%) of the exhaust gas to the intake manifold **[13]**. The dilution effect, combined with replacement of air with the exhaust gases CO_2 and H_2O which have higher heat capacities, leads to lower combustion temperatures and hence reduced NO formation. The mechanics of EGR involve appropriate piping between engine exhaust and inlet systems and a control valve to regulate the amount of exhaust that is recirculated. When implemented properly, EGR can increase fuel economy under cruise conditions **[14]**. Unfortunately, there is no free lunch: EGR increases soot production, decreases thermal efficiency, and can cause misfire at excessive levels. As usual, fine control is required to balance effects **[15]**.

The aim of this study is to evaluate the best conditions for using methanol-gasoline blend. The effect of EGR on single cylinder SI engine fueled with methanol-gasoline blend was tested. For this purpose, the study was conducted using several engine variables like engine speed, load, compression ratio and spark timing.

1. Experimental Setup

1.1 Experimental apparatuses

Experiments were performed using petrol engine, type (PRODIT GR306/0001). The engine is designed to be a spark ignition of a single cylinder, water cooled, four strokes and variable compression ratio engine. The general arrangement of the experimental rig is shown in **figures (1 & 2)**, while **table (1)** illustrates engine specifications.

The engine rig is coupled to the following equipments:

● The suction side of the engine cylinder was connected to an air tank. Air tank damped out the pressure variations in air that was entering into carburetor. The atmospheric air was drawn into the engine cylinder through air tank. A manometer provided to measure the pressure drop across the orifice was used to calculate the volume of air drawn into the cylinder. This set was calibrated in the laboratory, by using a calibrated set and compared the readings of the two sets.

● Fuel was supplied to the engine from the main fuel tank through a graduated measuring fuel gauge (burette).

● A hydraulic dynameters was used to measure the torque of the output engine. This dynamometer was calibrated in the laboratory using calibrated weights.

● Exhaust gas temperature were measured by using thermocouples type K (Ni-Cr/Ni-AL) at the beginning of the exhaust tube. These thermocouples were calibrated in the laboratory by comparing its readings with that of a set of calibrated thermocouples.

EGR System: In order to furnish the tested engine with EGR, a supply system was fitted to the engine, as shown in **Fig.3**. The exhaust gas was extracted immediately above an intermediate flange connecting between the exhaust gas manifold and exhaust pipe, which is 35cm downstream from

● confluence point. By this arrangement, the driving force for the EGR was the pressure difference between the exhaust and the intake manifold

● pressure. In case of hot EGR without cooling, the desired amount of EGR was controlled by a flow control valve, which was placed after a 50cm copper tube from the extraction point. The feedback point of the EGR was located at the end of a plenum chamber, that is 3 cm downstream of the mixer in order to avoid the interaction between recycled exhaust gas and residual gases at valve overlap as effectively as possible. EGR measurement was evaluated by:

$$EGR\ (\%) = \frac{\dot{m}_{EGR}}{\dot{m}_{EGR} + \dot{m}_a} \times 100$$

Where $\dot{m}_{EGR}$ - the mass flow rate of EGR, and $\dot{m}_a$ is the mass flow rate of fresh air. In order to determine how far the EGR valve should be opened to achieve a desirable EGR mass ratio, different EGR rates were extracted from a simple computer code based on the equation of gas state and the method of trial and error.

The fundamental equations describing the performance of spark ignition engine, with EGR and without EGR are **[16]**:

● The brake power:

$BP=w_b*N/348.067$ ……………………………………..(1)

Where: W_b= the load in (N)
N= speed engine (r.p.m.)

● The gasoline brake specific fuel consumption:

$BSFC= m^o_f*3600/BP$ ……………………………….(2)

Where, m^o_f= fuel consumption mean (kg/kw.hr)

● The M20 brake specific fuel consumption (Benjamin, 2010):

$$BSFC = \frac{\left(m^o_{methanol}\times LHV_{methanol}+m^o_{gasoline}\times LHV_{gasoline}\right)*3600}{BP} \qquad \text{……….(3)}$$

Where: $m^o_{methanol}$ and $m^o_{gasoline}$ are the mass flow rates (g/s) of the methanol and gasoline fuels. $LHV_{methanol}$ and $LHV_{gasoline}$ are the lower heating values of the methanol and gasoline fuels. BP is the engine brake power.

● The volumetric efficiency:

$\eta_{vol}= (m_a)_{act}/(m_a)_{theo}$ ………………………………..(4)

● The brake thermal efficiency:

$\eta_{bth}= BP/ m^o_f*(L.C.V)$ ……………………………(5)

The stoichiometric fuel/ air ratio was calculated, and then equivalence ratio can be defined as **[17]**:

$$\emptyset = \frac{\dfrac{[G]}{[air]-\dfrac{[M]}{([M]/[air])_{st}}}}{\left(\dfrac{[G]}{[air]}\right)_{st}} \qquad \text{……………………………….(6)}$$

Where rate of heating energy = fuel mass flow rate * LHV. The denominator in Eq. (5) is the rate of the total heating energy of the two fuels. The MGR was varied by changing the mass flow rates of both methanol and gasoline fuels.

1.2 Materials

Combustion tests were carried out using, as baseline fuel, the Iraqi gasoline with ON=82 produced by Al Doura refinery; moreover, a blend with the volume of 20% methanol with gasoline were tested. Methanol is also known as methyl alcohol and its chemical formula is CH_3 OH. Methanol possesses high octane number, and is often used as octane improver in reformulated gasoline blends. Commercially, methanol is most commonly produced by steam reforming of natural gas **[18 & 19]**. Iraq is considered as one of the largest ambushes of natural gas. The used quantities of NG in Iraq are very low compared with its high stores. In this study, the blends were prepared on volume basis. Methanol was blended with gasoline in concentration of 20% and gasoline, this blend is known as M20. Fuel properties of the gasoline were determined in the Fuel Laboratory of the Department of Chemical Engineering, UOT. Methanol of 99% purity was purchased from local markets. **Table 2** represents the typical properties of gasoline, methanol and ethanol.

1.3 Test procedure

The tests were carried out under steady-state conditions. The engine was allowed to run until it reached steady-state conditions, and then, the data were collected subsequently. The engine was firstly warmed up with the coolant and lubricating temperatures stabilized. All the tests were carried out at stoichiometric air–fuel ratio (AFR).

The experimental tests started with pure gasoline, to set a database performance level on the basis of which the comparison will be carried out. The experiments were conducted for gasoline starting from CR=6:1, and on, to find used gasoline higher useful compression ratio. The same tests were conducted for M20 and (M20 + 20% EGR) starting from HUCR for gasoline, because methanol has higher octane number. All fuel tests were conducted at wide-open throttle conditions. The tests were repeated three times and average values were presented to reduce the experimental uncertainties.

All the tests for each fuel carried out in this work were done under the fuel-specific optimum spark timings, known as the maximum brake torque (MBT) timings. Spark sweeps were performed for each fuel at various engine torques (starting by 10 Nm till 25 Nm). The definition used

for the MBT timing was the spark timing which provides the maximum IMEP for a fixed throttle position.
The basic objective of this experiment was to conduct performance tests on the variable compression ratio engine using methanol blended with petrol. The experiments were conducted at different compression ratios and at different speeds starting by 1000 till 2600 rpm. Various parameters were calculated, such as, brake
thermal efficiency, brake power, brake specific fuel consumption and volumetric efficiency. Engine performance at two cases was evaluated: At HUCR and OST for gasoline and at HUCR and OST for M20 and M20+20% EGR. The performance tests were consisted of:

1- When the torque is constant at (20 N.m) and engine speed was varied (1100, 1500, 1900, 2300 and 2700 r.p.m.).

2- When engine speed was fixed at (1500 r.p.m.) and engine torque was changed (10, 15, 20 and 25 N.m.).

The second set of tests: the experiments were conducted on the engine with recirculating exhaust gas by (20% EGR volumetric percentage), higher useful compression ratio and optimum spark timing for M20 and M20+20% EGR. Engine performance was evaluated and compared with the first case at the same variable speeds and loads

2. Results and Discussion

Figures 4, 5 &6 represent the first part from tests, in this part the HUCR was found for each fuel. For gasoline the HUCR was found 7:1. This low CR is due to low octane number for Iraqi conventional fuel. **Fig. 4** shows that BP curve at CR=7.5:1 starts to decline at medium and high speeds. It can be seen the same decline in curves of CR= 10:1 for M20 (**Fig. 5**) and CR= 9.5:1 for M20+20%EGR (**Fig. 6**). The reason for this decline in BP is the occurrence of engine knock which requires reducing engine torque to get rid of knock. The methanol portion in M20 caused higher octane number and high knocks resistance. This effect was exposed for the effect of EGR that reduced its activity, resulting in CR=9:1 which is less than that for M20.
All tests presented in **Figures 7, 8 & 9** were carried out at a variable engine speed, constant torque 20 Nm, wide-open throttle and at HUCR for each fuel. The spark timing was changed in the range 12 to 20 crank angle degree before top dead centre (ºBTDC) in order to identify the maximum brake torque and the knocking limit.

Fig. 7 shows that OST for gasoline fuel is 19ºBTDC, which is not the designed OST for the engine (17 ºBTDC). Conventional Iraqi gasoline has low octane number (ON=82) caused this advanced timing.
When methanol is introduced as in M20 that means the flame propagation speed is faster. So, the M20 has a higher flame speed. Also, methanol has higher octane number. These two factors together made optimum spark timing of M20 to be at 17 ºBTDC, as **Fig. 8** represents.
EGR addition to M20 slow down the flame propagation, causing lower BP, as **Fig. 9** illustrates. Advancing spark timing create higher pressure and temperature that improved generated BP. Severe spark advance causes high pressure rates that may generate knock. From the figure OST for M20+20% EGR was 19ºBTDC. Any other advancing was a cause for knock that influence to reduce engine load which resulted in BP reduction.
Fig. 10 shows that adding EGR to M20 at CR=7:1 caused high decrement in BP, contrary to running the engine at HUCR=9:1 where the resultant BP was higher than that for gasoline. Two reasons for this result they are adding methanol with higher ON which increased the methanol-gasoline blends overall ON. The second reason is the EGR effect inside the combustion chamber. EGR absorb part of the generated heat and reducing the overall heat that maybe cause fuel self ignition. Another effect for EGR addition is its taking over apart of air-fuel mixture reducing the reactants and causing lower HUCR compared with M20. Adding methanol to gasoline increases its resistance to knock by increasing the blend ON. The reduction in BP for M20 at CR=7:1 is due to lower heating value of methanol compared to gasoline. The increment in BP at M20 HUCR=9.5:1 is due to higher energy resulted from better combustion at this CR. The oxygen fraction in methanol improved combustion resulting in higher BP.
Bsfc depends on resulted BP as equations 2 and 3 represents. BSFC increased as the speed increased and decreased as the compression ratio and brake power increased. Increasing BP reduced BSFC and verse versa, as **Fig. 11** declares. If gasoline is taken as baseline then operating the engine with M20 and M20+20% EGR at CR=7 and HUCR increased bsfc with about 12.26, 30.9 & 7.27% respectively. In the same time, engine operation at HUCR and OST for M20, reduced engine bsfc with about 3.06%. This was due to the fact that density of charge increased with the compression ratio, which leads to more efficient combustion improving the fuel combustion characteristics. BSFC decreased with adding

methanol content, because methanol content increased combustion rate as more oxygen was available which allowed complete combustion.
Fig. 12 represents the effect of engine speed on volumetric efficiency for the five studied cases. Adding EGR reduced volumetric efficiency in both cases compared with gasoline, because EGR took a portion of the interring air-fuel charge. While adding methanol increased volumetric efficiency in both cases. This was due to presence of oxygen in methanol, which required less air for combustion. When the speed of the engine was increased, volumetric efficiency decreased, due to less time available for suction. Methanol is considered as oxygenate due to oxygen molecular in its structure. Adding EGR at CR=7:1 and HUCR reduced volumetric efficiency with about 14 & 3.5% respectively compared with gasoline. In the same time, operating the engine with M20 at CR=7:1 and HUCR increased volumetric efficiency with about 0.6 & 3.22% respectively.
Brake thermal efficiency increases by increasing BP or reducing $m^o{}_f$ * *(LHV)*, as equation 5 clarifies. **Fig 13** manifests that due to this reason M20 and M20+20% EGR at HUCR and OST for each surpassed gasoline brake thermal efficiency, with about 5.87 and 1.12% respectively. In contrast, M20 and M20+20% EGR retracted below gasoline brake thermal efficiency with about 7.9 & 15.1% respectively.
Operating engine with low compression ratio and speed resulted in low exhaust gas temperatures, as **Fig. 14** illustrates. In contrast operating the engine with HUCR and high speed resulted in high exhaust gas temperatures. Increasing engine speed.

NOMENCLATURE

Degree before top dead centre	ºBTDC
Brake specific fuel consumption	BSFC
Brake power	BP
Compression ratio	CR
Crank angle	CA
Exhaust gas recirculation	EGR
Higher useful compression ratio	HUCR
Optimum spark timing	OST
Spark ignition engine	SIE
Fuel flow rate	$m^o{}_f$
Brake thermal efficiency	η_{bth}
Volumetric efficiency	η_{vol}
Equivalence ratio	Ø

Table (1) Engines pacifications

Manufacturer	PRODIT	No load speed range	500-3600 rpm (Otto cycle)
Cycle	OTTO or DIESEL, four strokes	Load speed range	1200-3600 rpm (Otto cycle)
Number of cylinder	1 vertical	Intake star	54^o before T.D.C
Diameter	90mm	Intake end	22^o after T.D.C
Stroke	85mm	Exhaust start	22^o before T.D.C
Compression ratio	4-17.5	Exhaust end	54^o after T.D.C
Max .power	4 kWat 2800 rpm	Fixed spark advance	10^o (spark ignition)
Max .torque	28 Nm at 1600 rpm	Swept volume	$541cm^3$

Table (2) Properties of typical gasoline, methanol and ethanol [20]

Property	Gasoline	Methanol	Ethanol
Chemical formula	Various	CH_3OH	C_2H_5OH
Oxygen content by mass [%]	0	50	34.8
Density at NTP [kg/l]	0.74	0.79	0.79
Lower heating value [MJ/kg]	42.9	20.09	26.95
Volumetric energy content [MJ/l]	31.7	15.9	21.3
Stoichiometric AFR [kg/kg]	14.7	6.5	9
Energy per unit mass of air [MJ/kg]	2.95	3.12	3.01
Research octane number	89-95	109	109
Motor octane number	85	88.6	89.7
Boiling point at I bar [ºC]	25-215	65	79
Heat of vaporization [kJ/kg]	180-350	1100	838

Reid vapor pressure [psi]	7	4.6	2.3
Flammability limits in air [λ]	0.26-1.6	0.23-1.81	0.28-1.99
Laminar flame speed at NTP, Ø=1 [cm/s]	28	42	40
Adiabatic flame temperature [ºC]	2002	1870	1920
Specific CO_2 emissions [g/MJ]	73.95	68.44	70.99

Fig.1, Single cylinder prudent spark ignition engine

Fig.2, Single cylinder Prodet spark ignition engine

Fig. 3 EGR assembly used in present

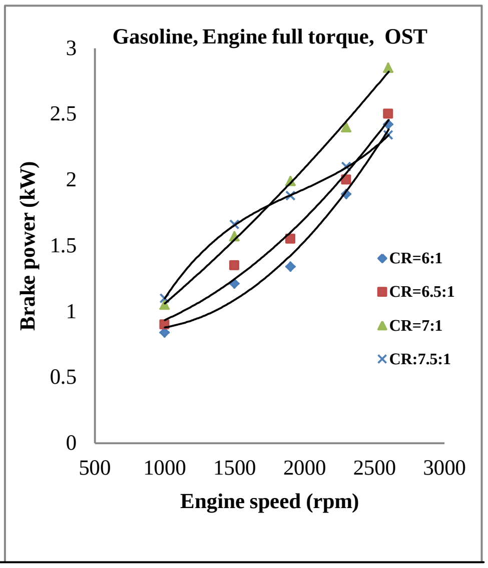
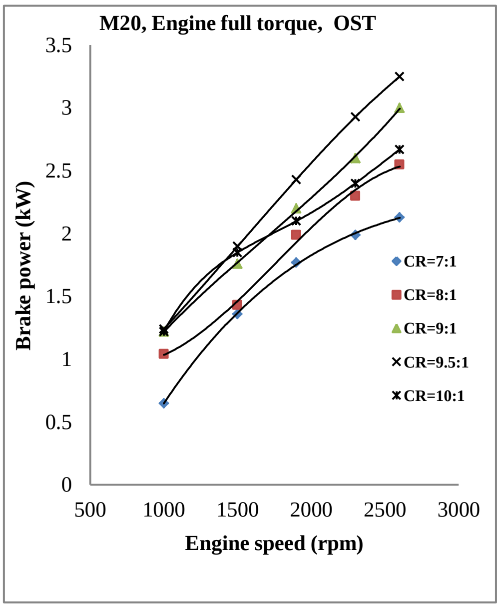
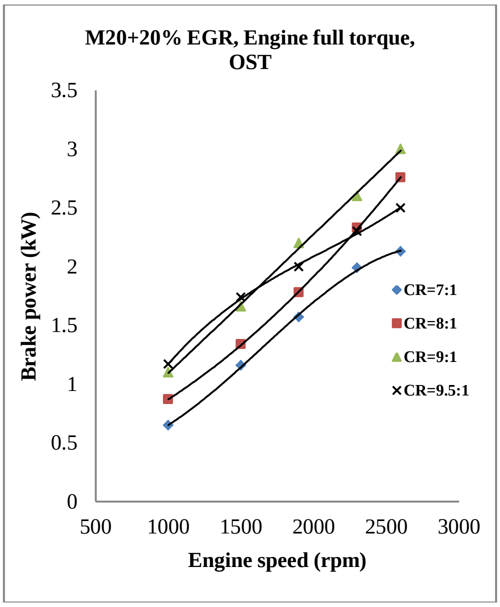


Fig. 4, Compression ratio effect on BP for gasoline at variable speed and OST and constant torque





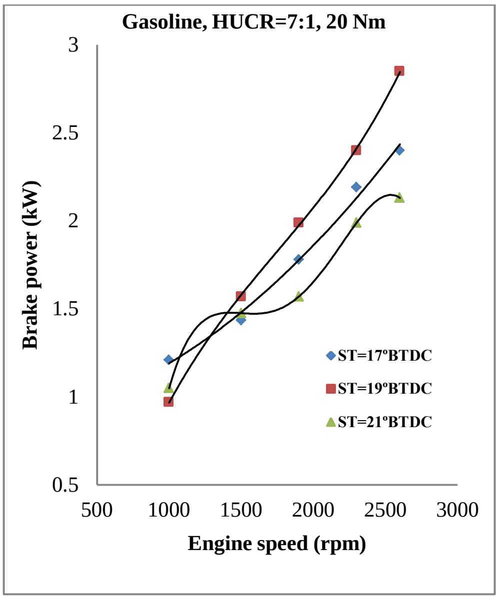


Fig. 7, Spark timing effect on BP for gasoline at variable speed, HUCR and constant torque

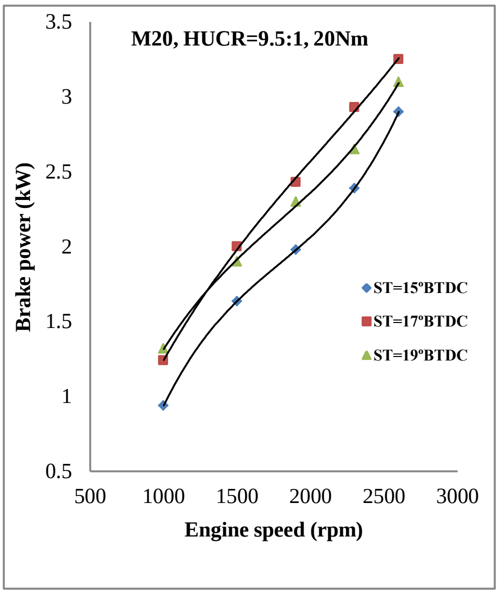
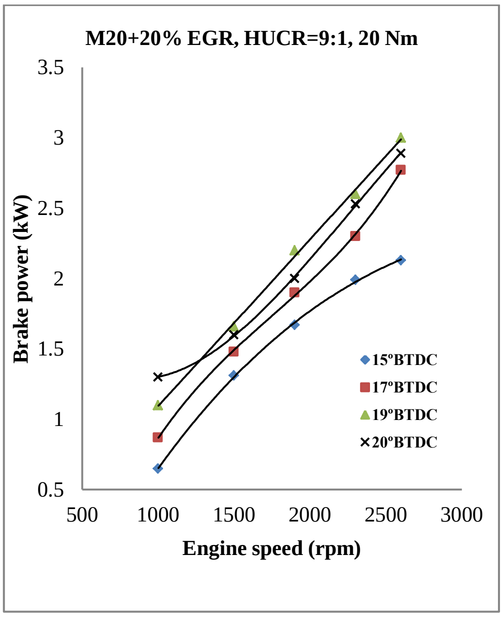


Fig. 8, Spark timing effect on BP for M20 blend at variable speed, HUCR and constant torque



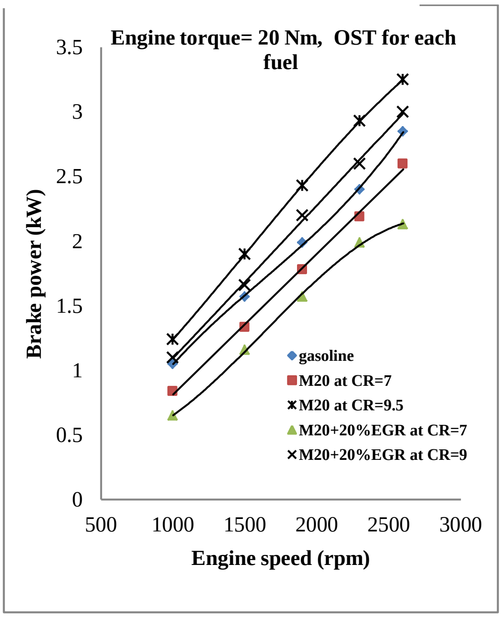
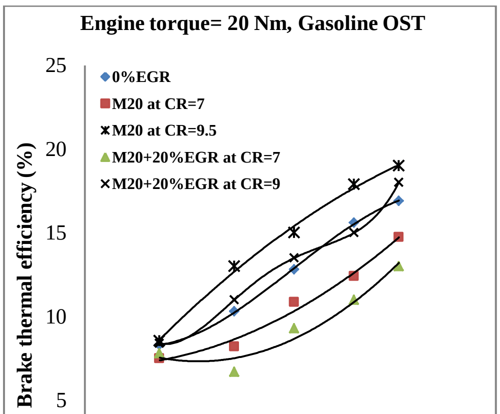
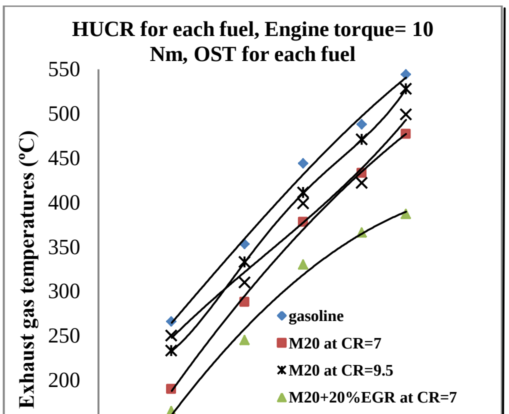


Fig. 9, Spark timing effect on BP for M20+20% EGR at variable speed

Fig. 10, Engine speed effect on BP for studied cases at HUCR and OST for


Engine torque= 20 Nm, Gasoline OST
0%EGR
M20 at CR=7
M20 at CR=9.5
M20+20%EGR at CR=7
M20+20%EGR at CR=9
Brake thermal efficiency (%)
25
20
15
10
5


HUCR for each fuel, Engine torque= 10 Nm, OST for each fuel
Exhaust gas temperatures (°C)
550
500
450
400
350
300
250
200
gasoline
M20 at CR=7
M20 at CR=9.5
M20+20%EGR at CR=7

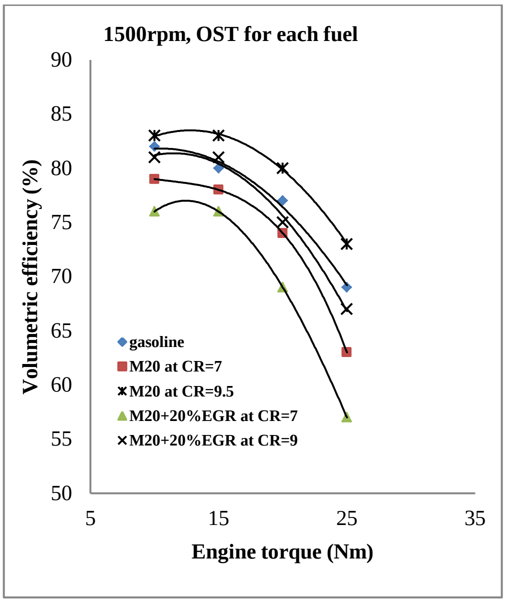


Fig. 17, Engine torque effect on volumetric efficiency for studied cases at HUCR and OST for each fuel and constant speed

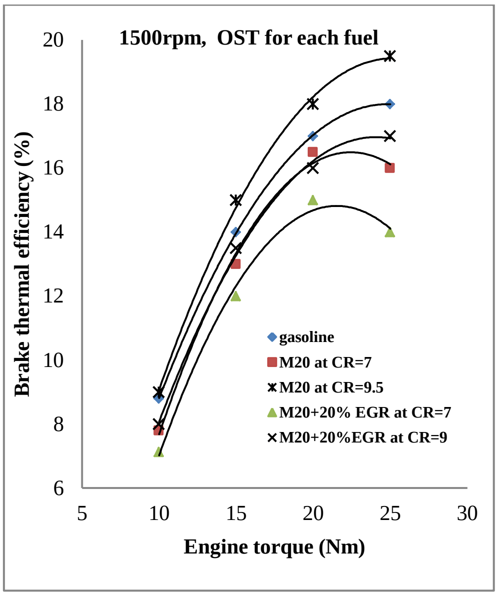


Fig. 18, Engine torque effect on brake thermal efficiency for studied cases at HUCR and OST for each fuel and constant speed

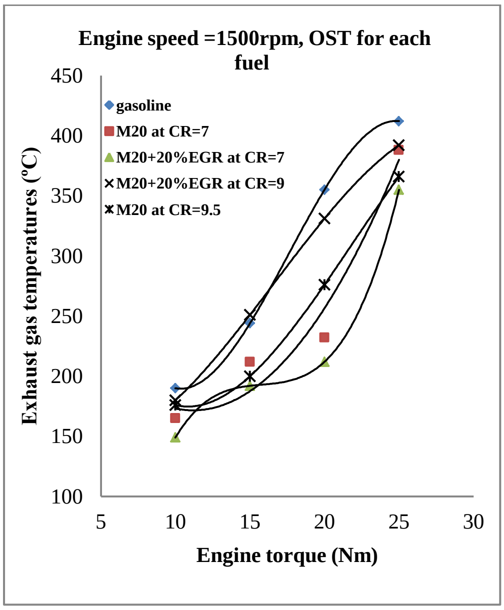


Fig. 19, Engine torque effect on exhaust gas temperatures for studied cases at HUCR and OST for each fuel and constant speed

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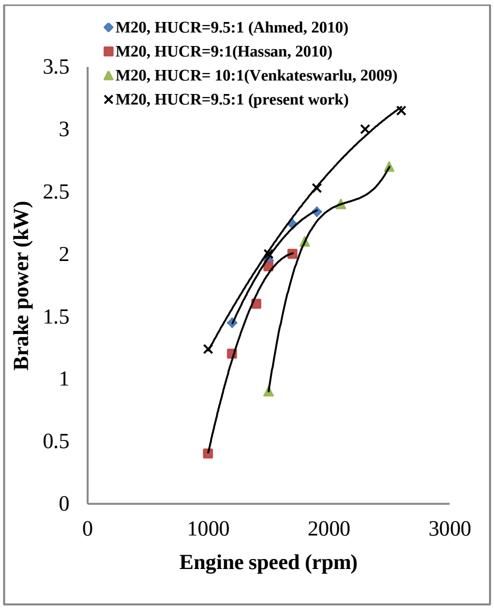

Fig. 20, Engine speed effect on BP for comparison with other researches at HUCR and OST for M20 fuel and constant speed

volume to this gasoline improved its ON that increased HUCR to 9.5:1 and caused a retard in OST to 17 ºBTDC. Adding 20% EGR to this blend reduced HUCR to 9:1 and caused an advance in OST to 19 ºBTDC again.

2. The leveraging effect of methanol fuel on improving engine performance could be attributed to factors such as the cooling effect of the methanol fuel added to the effect of methanol fuel's high combustion velocity.
3. Operating the engine with M20 increased the brake thermal efficiency compared to gasoline operation.
4. A decrease in bsfc of about 3.06% was observed when the engine was run with M20 at HUCR and OST.
5. Exhaust gas temperature decreased with methanol addition, it decreased with EGR addition, too.

Finally, from the test matrix, it was observed that M20 appeared to be a good candidate fuel in comparison to present conventional Iraqi gasoline operation. Perhaps modifying gasoline fuel will give better blend results especially with adding EGR to this blend.

4. References

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