Al-Nahrain University, College of Engineering Journal (NUCEJ) Vol.19 No.1, 2016 pp.76 - 85

Effect of Cool and Hot Exhaust Gas Recirculation (EGR) on the Performance of Multi-Cylinder Compression Ignition Engine Fueled With Blends of Diesel and Methanol

Miqdam Tariq Chaichan
Machines & Equipments Eng. Dept.,
University of Technology
miqdam_tc@uotechnology.edu.iq

Qahtan Adnan Abass
Machines & Equipments Eng. Dept.,
University of Technology
qahtanalnakeen@yahoo.com

Abstract

Performance of a direct injection multi-cylinders diesel engine fueled with diesel-methanol blends using hot and cool EGR were investigated. The influence of some engine variables such as equivalence ratio, load, engine speed and injection timing were studied. The test were conducted with hot EGR and repeated with cool EGR. The results were compared with that from engine operation with neat diesel and 90% diesel+10% methanol.

The results show that different behaviors were existed in the combustion between the diesel-methanol blends than that of diesel fuel. EGR which is a useful tool in reducing NOx affects engine performance largely. Adding methanol to diesel reduced the harm effects of EGR on engine performance by increasing available oxygen inside combustion chamber. Cool EGR gives better thermal efficiency, volumetric efficiency, brake specific fuel consumption (bsfc) and brake power (bp) compared to hot EGR. Advancing injection timing improved engine performance when EGR was added.

Introduction

With the increasing concern about fuel shortage and environmental protection, the research on improving fuel economy and decreasing exhaust emissions has become the major activity in the engine combustion and development [1]. The development of alternative fuel engines has attracted increasing attention. Alternative fuels are usually clean compared with diesel and gasoline in the combustion processes [2]. Among these fuels, the simplest of alcohol and originally produced by the destructive distillation of wood which is called methanol [3]. Today it is produced in very large quantities from natural gas by the reformation of the gas into carbon monoxide and hydrogen followed by passing these gases over a suitable catalyst under appropriate conditions of pressure and temperature [4].

Methanol (CH_3OH) is a clear, colorless, high-performance liquid fuel that can be used In both spark-ignition and diesel engines. It can be burned as a neat (100% methanol) or near-neat fuel in internal combustion engines. A blend of 85% methanol and 15% gasoline, M85, is the most common form of methanol fuel used in light-duty vehicles. M100, or neat methanol, is used in some heavy-duty trucks and buses [5].

Methanol's ceta ne rating is between 0 and 10, while diesel fuel has cetane ratings in the range of 40 to 55. Diesel engines must incorporate special features to overcome the low cetane of methanol, or additives can increase the cetane rating of methanol to levels similar to diesel fuel [6 & 7]. Methanol also has less lubricity than diesel fuel, which causes problems with some fuel injection equipment. However, lubricity additives are available to mitigate this disadvantage. Diesel engines of all types could use ignition-improved methanol with modifications to the fuel injection system to enable the proper amount of fuel to be injected [8].

As the name implies, EGR involves recirculating a portion of the engine's exhaust back to the charger inlet or intake manifold, in the case of a naturally aspirated engines. In most systems, an intercooler lowers the temperature of the recirculated gases. The cooled recirculated gases, which have a higher heat capacity than air and contain less oxygen than air, lower combustion temperature in the engine, thus inhibiting NOx formation. EGR systems are capable of achieving NOx reductions of more than 40 percent [9].

EGR systems lower NOx emissions in different ways. First, the oxygen concentration in the combustion mixture is reduced by diluting the inlet air with combustion products (exhaust gas). Second, the peak combustion temperature is reduced by dilution and introduce in combustion products; such as CO_2 and H_2O as heat absorbers [10 & 110. This process cannot be carried too far. If too much EGR is used, an increase in particle emissions is observed. Some reports have stated that EGR increases emitted particulate emissions [12].

The objectives of this study were to form the oxygenated blends by adding methanol and solvent in diesel fuel and then study the characteristics of performance in a compression ignition engine fueled with the oxygenated blends under various engine variables. The performances parameters were compared with EGR (hot and cold) and without EGR for same engine output conditions.

Experimental Setup

Materials

In this study the procedure reported by references [8 & 13] was used. Three kinds of diesel/methanol blends with different methanol additions were selected. Due to the low solubility of the methanol in diesel fuel, a solvent consisting of oleic acid and iso-butanol was added into the diesel/methanol blends to develop the stabilized diesel/methanol blends.

Fuel properties and the constitutions of three blends are given in **Tables 1, 2** and **Fig. 1**, and the oxygen fraction in the fuel blends ranged from 5.87 to 11.1 as shown in **Table 2** and **Fig.2**.

It can be seen that the oxygen in the fuel blends come mainly from methanol addition although the mass fraction of methanol and solvent has the same level, so it is reasonable to consider the influence of oxygen in the fuel blends to be the influence of oxygen from the addition of methanol. The solvent was added to methanol in the exact wanted rate first. Then the mixture was added to diesel fuel performing the combustible mixture. The fuel properties show that methanol has high oxygen content, while the heat value is low, and cetane number is low compared to diesel fuel.

Table 1: Fuel properties of diesel, methanol and blended fuel constitutions

property	diesel	methanol	Solvent	
			Oleic acid	Iso-butanol
Chemical formula	$C_{10.8}H_{18.7}$	CH_3OH	$C_{18}H_{34}O_2$	$C_4H_{10}O$
Mole weight (g)	148.3	32	282	74
Density (g/cm^3)	0.86	0.796	0.8905	0.802
Lower heating value (MJ/kg)	44.40	19.68	38.65	33.14
Heat of evaporation (kJ/kg)	260	1110	200	580
Self-ignition temperature (^oC)	200-220	470	335	385
Cetane number	45	5	40	10
C wt%	86	37.5	76.6	64.8
H wt%	14	12.5	12	13.5
O wt%	0	50	11.4	21.7
Blended fuel 1 wt%	79.86	8.96	10.1	1.08
Blended fuel 2 wt%	71.28	13.33	14.47	0.92
Blended fuel 3 wt%	63.94	17.66	16.6	1.8

Table 2: fuel properties of diesel/methanol blended fuel constitutions

Property	Blended fuel 1	Blended fuel 2	Blended fuel 3
Lower heating value (MJ/kg)	41.73	39.89	38.64
Heat of evaporation (kJ/kg)	333.53	367.57	405.94
Cetane number	40.41	38.4	36.21
C wt%	80.47	77.98	75.5
H wt%	13.66	13.5	13.4
O wt%	5.87	8.52	11.1
O wt% contributed from methanol	4.48	6.67	8.83
O wt% contributed from solvent	1.39	1.85	2.27

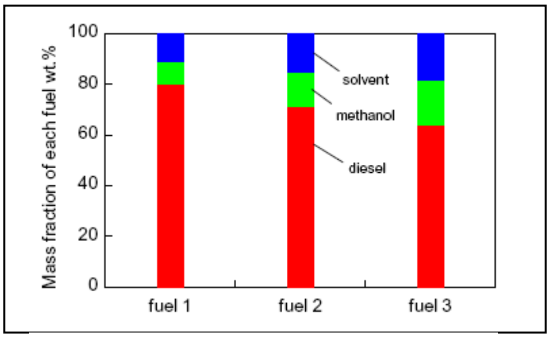

Figure 1: Constitution of the fuel

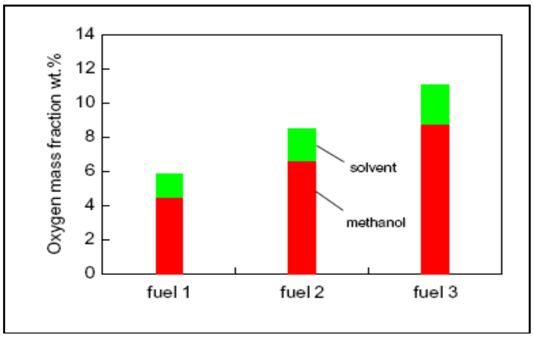

Figure 2: Oxygen mass fraction in the fuel blends

Engine and accessories

A four stroke, direct injection, naturally aspirated four-cylinder four-stroke diesel engine is employed for the present study shown in **Fig. 3**. The engine specifications are listed in **table 3**. The engine is coupled to a hydraulic dynamometer, and the engine speed was measured with a tacho-generator connected to the dynamometer.

The load and speed of the engine were controlled by adjusting the dynamometer resistance. The fuel consumption of the engine was determined by measuring the fuel level decrease in a measurement container in a given period of time. The volumetric flow rate of the intake air was measured using orifice plate.

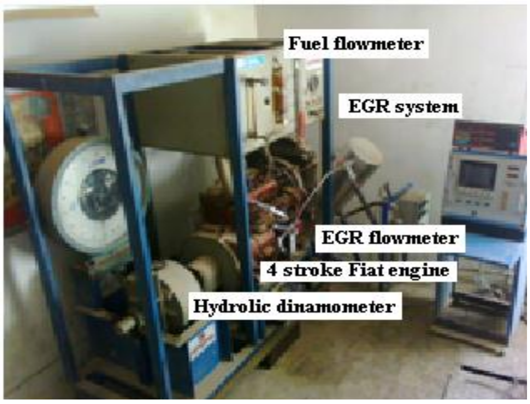

Figure 3: photographic picture of the experimental rig.

Figure 4: EGR assembly used in the present study

Table 3: Tested engine specifications

Engine type	4cyl., 4-stroke
Engine model	TD 313 Diesel engine rig
Combustion type	DI, water cooled, natural aspirated
Displacement	3.666 L
Valve per cylinder	two
Bore	100 mm
Stroke	110 mm
Compression ratio	17
Fuel injection pump	Unit pump 26 mm diameter plunger
Fuel injection nozzle	Hole nozzle 10 nozzle holes Nozzle hole dia. (0.48mm) Spray angle= 160^o Nozzle opening pressure=40 MPa

Known quantity of exhaust gas is re-circulated into the combustion chamber with air. EGR quantity was achieved with manually controlled EGR valve. The exhaust gas comes out exhaust valves at very high temperature and pressure (more than 1000°C and 30 bars). An EGR cooler (type Prodit) was used to reduce the temperature of exhaust gas. Gate valve is used to control the flow of exhaust gas. Valves are used to control the inlet and outlet water supply to the cooler. The exhaust gas temperature was measured using a thermocouple connected to the exhaust pipe just downstream of the exhaust manifold. The cooling water temperatures at the inlet and outlet of the engine were measured using calibrated thermocouples. **Fig. 4** illustrates a photographic picture of the used EGR system.

External EGR system use piping to route exhaust gas to the intake system was the preferred approach at present work. The EGR ratio is the ratio of the amount of EGR to the charge aspired into an engine cylinder. In this study, the EGR ratio was calculated with the following equation:

$$EGR = \frac{\dot{m_{EGR}}}{\dot{m}_{air} + \dot{m}_{EGR}} \qquad \ldots\ldots (1)$$

Where: $\dot{m_{EGR}}$ - is the flow mass rate of EGR air, and $\dot{m}_{air}$ - is the flow mass rate of fresh air. To calculate correct EGR ratios, the flow EGR rate has to be defined exactly. However, this is very difficult because of the high temperature and contamination by ash, soot and unburned hydrocarbon.

The following equations were used in calculating engine performance parameters:

1- Brake power

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

2- Brake mean effective pressure

$$bmep = bp \times \frac{2*60}{V_{sn} * N} \quad kN/m^2 \qquad \ldots\ldots (3)$$

3- Fuel mass flow rate

$$\dot{m}_f = \frac{v_f \times 10^{-6}}{1000} \times \frac{\rho_f}{time} \quad {kg}/_{sec} \qquad \ldots\ldots (4)$$

4- Air mass flow rate

$$\dot{m}_{a,act.} = \frac{12\sqrt{h_o * 0.85}}{3600} \times \rho_{air} \quad \frac{kg}{sec} \qquad \ldots\ldots (5)$$

$$\dot{m}_{a_{theo.}} = V_{s.n} \times \frac{N}{60*2} \times \rho_{air} \quad \frac{kg}{sec} \qquad \ldots\ldots (6)$$

5- Brake specific fuel consumption

$$bsfc = \frac{\dot{m}_f}{bp} \times 3600 \quad \frac{kg}{kW.hr} \qquad \ldots\ldots (7)$$

6- Total fuel heat

$$Q_t = \dot{m}_f \times LCV \quad kW \qquad \ldots\ldots (8)$$

7- Brake thermal efficiency

$$\eta_{bth.} = \frac{bp}{Q_t} \times 100 \quad \% \qquad \ldots\ldots (9)$$

Tests procedure

To evaluate engine performance and the effect of different engine variables as well as the fuel and EGR mode effects, tests were carried out on the engine using diesel fuel alone in order to provide base line data. The engine was warmed up until engine cooling water temperature reached 70°C and it was maintained at this temperature for about half an hour. Engine performance for each diesel-methanol blends were tested and recorded. Methanol percentage which achieved maximum brake power was chosen to continue other tests with EGR.

Then, 15% hot EGR was used to evaluate the impact of EGR on engine performance. This EGR percentage was chosen because it is not low percentage so the dilution effect will not appear; also it is not too high so dilution effect dominated. After complete performance tests with hot EGR, cooling water was allowed to flow in EGR cooler, and leaving the engine operating until recirculated exhaust gas temperature fixed at 50°C. Then performance tests repeated, and the results were compared for each mode.

In the second set of experiments engine speed was changed between 1250 and 2500 rpm with intervals of 250 rpm, while the engine was operated at medium load and full load. The fuel injection timing was kept constant at 38° BTDC with variable speed tests. Before each fuel test, the fuel tank and fuel lines were drained, and the engine was operated at least 15 minutes to stabilize on the new fuel. At each speed, the engine was operated 5 minutes to achieve steady-state conditions, and the data were collected at sixth minute. Each test was repeated 3 times and the results of the 3 repetitions were averaged.

Results & Discussion

An experimental study was conducted to investigate the practical application of EGR mode (hot or cool) on a 4-cylinders direct injection diesel engine fueled with methanol-diesel blends, and its effect on performance. The study addressed EGR effects on brake power, combustion efficiency, exhaust gas temperatures and volumetric efficiency. In particular, the study examined in detail operation with constant level of EGR (15%) had two gas temperatures. To limit the number of experiments and concentrate on

EGR effect, three methanol- diesel blends were examined to choose the best of them.

Fig. 4 represents the resulted brake power (bp) for diesel fuel and its blends with methanol for wide range of equivalence ratios. Brake power reduced with methanol addition. M10 (10% methanol + 90% diesel) gave the higher bp compared to the other blends. The reduction in bp compared with that for diesel were 19.37%, 16.78% and 21.5% for M5, M10 and M15 respectively.

Fig. 5 shows brake specific fuel consumption (bsfc) of tested fuels for wide range of engine loads. Diesel fuel characterized by its low bsfc, M10 followed it then M5 and M15. Adding methanol to diesel increased bsfc by 6.7%, 4.6% and 11.9% for M5, M10 and M15 respectively.

From the above figures M10 can be considered as a reasonable selection to perform the remaining tests.

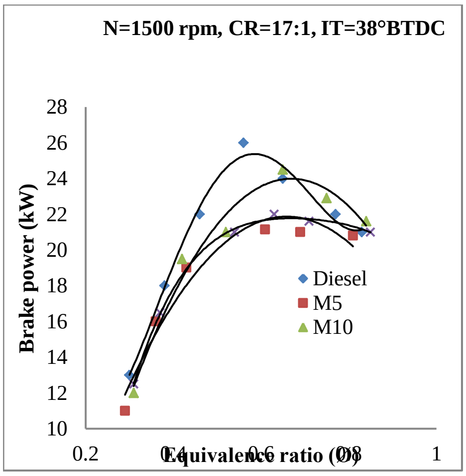

Figure 4: equivalence ratio- brake power relationship for the tested fuel

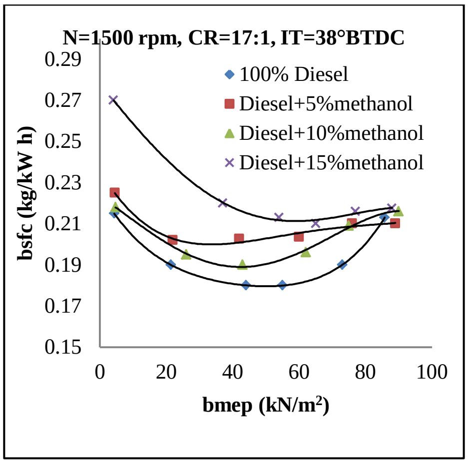

Figure 5: brake mean effective pressure- brake specific fuel consumption relationship for the tested

Fig. 6 represents the effect of hot and cool EGR on engine bp for wide range of equivalence ratios. Working with cool EGR produced higher bp than hot EGR operation, but still both less than diesel fuel. Previous studies [14] have shown that using EGR in order to reduce NOx emissions generally degrades engine performance. Flowing EGR reduces the concentration of oxygen, hence directly affecting fuel-air mixture composition in a compression ignition (CI) engine. The reductions in bp were 19.65% and 8.7% for hot EGR and cool EGR respectively.

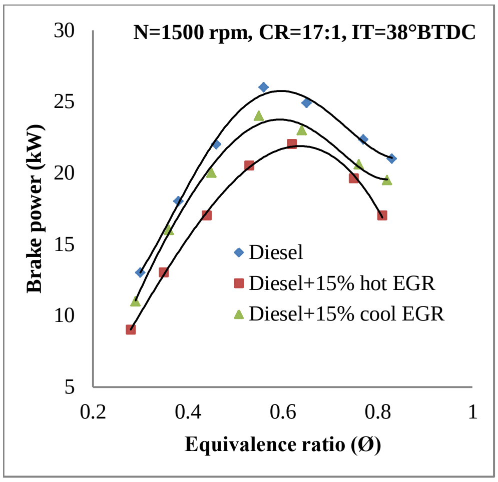

Figure 6: EGR mode effects on brake power for wide range of equivalence ratios operating with diesel fuel

The combustion process and engine performance are dependent on several important factors such as fuel properties, initial temperature and pressure, combustion chamber shape, injection pressure, etc. One important factor, denoted as Ø, indicates the equivalence ratio. It is the only factor that determines whether an engine's combustion process is complete or incomplete.

The results of using EGR modes with M10 engine operation was compared to diesel fuel results in **fig. 7**. EGR routes exhaust gas from preceding engine combustion cycles into the combustion chamber for succeeding combustion cycles. Displaces a unit of fresh air with an equal unit of burned exhaust products, will not only alter equivalence ratio, but causes a dilution effect. By reducing the oxygen concentration, the mixing time between the direct-injected fuel and the fresh oxygen increases. This is expected to increase the ignition delay and reduce the burn rate once diffusion combustion starts, in case all other parameters are kept constant. Brake power trends, shown in the figure allow better understanding of the relative contribution of two important factors (decreased combustion work versus increased pumping work accompanied with EGR utilization).

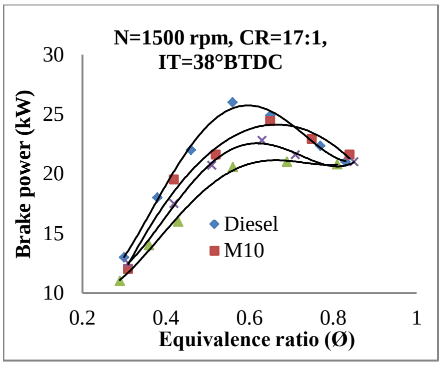

Figure 7: EGR mode effects on brake power for wide range of equivalence ratios operating with diesel and M10 fuels

Adding methanol reduced the total fuel heating value due its lower calorific value compared with diesel. But in the same time it increased oxygen content inside combustion chamber, which restricted harmful effect of EGR. Working with EGR reduced bp even more especially at equivalence ratios which the diesel engine usually produces its maximum bp. This leads to a reduction of the maximum heat release rate when EGR ratios increase. These reductions were 4.91%, 6.56% and 21% for M10, cool EGR and hot EGR respectively.

Fig. 8 illustrates volumetric efficiency variation with the tested modes. Methanol addition increases fuel blend oxygen content improving volumetric efficiency. Volumetric efficiency falls down when EGR was used, due to replace part of fresh air with gases. Hot gases occupied bigger part inside combustion chamber; this is why the volumetric efficiency reduced 9.4% compared to diesel fuel, while in cool EGR the reduction was 4%. Adding 10% methanol increased volumetric efficiency with about 2.9%.

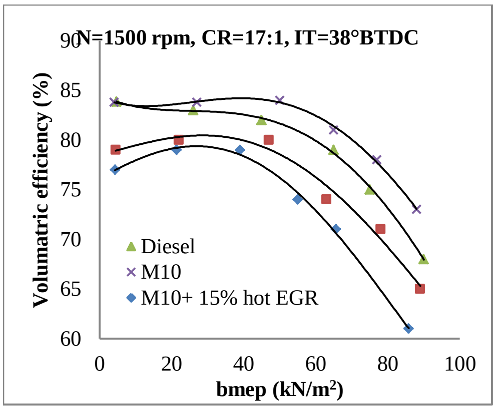

Figure 8: EGR mode effects on volumetric efficiency for wide range of loads operating with diesel and M10 fuels

Fig. 9 illustrates a decrease of engine brake thermal efficiency as a function of EGR mode for the investigated cases. Clearly, the deterioration of brake thermal efficiency with hot EGR is reasonable, and it is more pronounced at low and high loads conditions than at mid loads sets. The two main causes for decreasing brake thermal efficiency are attributed to decreased combustion work (i.e. indicated work) and increased pumping work (assuming that friction remained constant). The decreased combustion work is the consequence of combustion degradation due to lower combustion temperatures and changes in equivalence ratio. Adding 10% methanol to diesel fuel reduced brake thermal efficiency about 5.26%, while hot and cool EGR reduced it by 30% and 9.8% respectively.

Adding EGR has three effects, dilution effect, chemical effect and thermal effect. The chemical effect is associated with the dissociation of CO_2 to form free radicals and has been shown to be of minor effect [16]. However, this effect may have caused the slight increase in thermal efficiency compared with hot EGR.

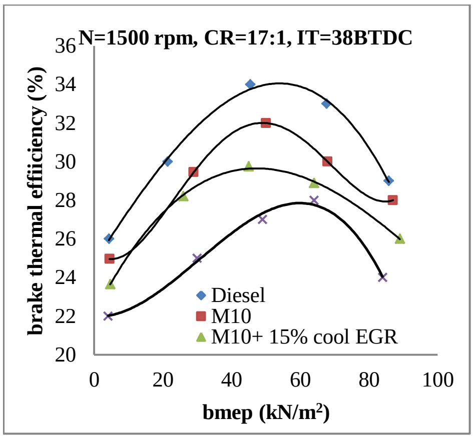

Figure 9: EGR mode effects on brake thermal efficiency for wide range of loads operating with diesel and M10 fuels

Engine brake power reduced with methanol addition for all studied engine speed range at medium loads (**Fig. 10**). Also, resulted bp with cool EGR utilization reduced, and its reduction was sharp at high speeds. Hot EGR caused high reduction in bp reached 9.7% compared to diesel, while for M10 and cool EGR it were 2.1% and 4.9% respectively.

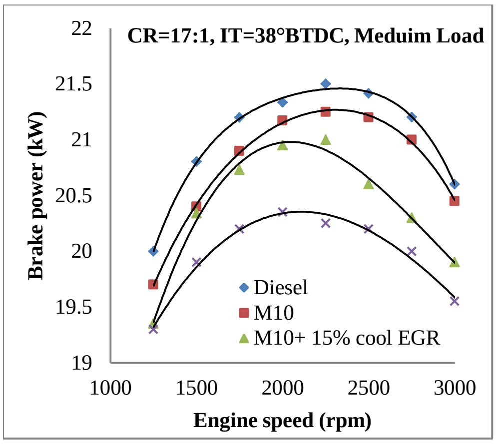

Figure 10: EGR mode effects on brake power for variable engine speed operating with diesel and M10 fuels

The same behavior resulted with high loads as **Fig. 11** represents, except the reduction percentages were changed. The droppings in bp were 1.45%, 2.95% and 5.4% for M10, cool EGR and hot EGR.

The effect of using EGR modes on bsfc is depicted in **Fig. 12** for variable engine speed. The hot EGR condition has been achieved to allow for higher gas temperature at the engine inlet. This increased the recycled gases' temperature. Since M10 has lower caloric values and densities than conventional diesel fuel, it caused specific fuel consumption to increase.

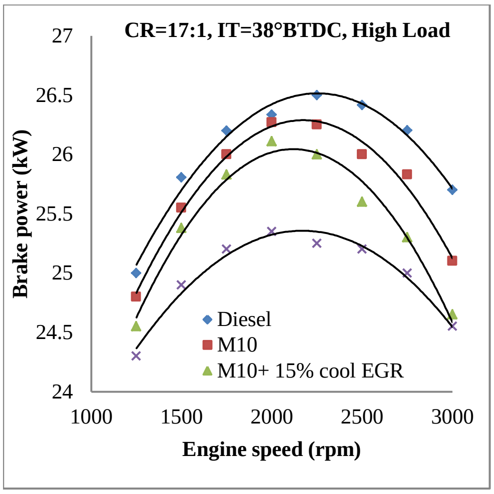

Figure 11: EGR mode effects on brake power for variable engine speed operating with diesel and M10 fuels

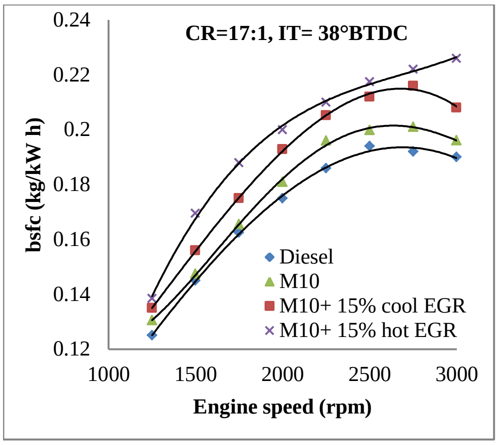

Figure 12: EGR mode effects on brake specific fuel consumption for variable engine speed operating with diesel and M10 fuels

Consequently, the bsfc has increased when using hot and cool EGR. In response to increases in incombustible components aspirated into the cylinder when EGR ratios were higher and oxygen content was lacking, specific fuel consumption was increased to keep operating condition constant when exhaust gas was recalculated and EGR was cooled more strongly. The drawbacks of specific fuel consumption were acceptable.

The gas temperature in the cylinder increases somewhat with increasing EGR. This is the consequence of increased intake temperature as well as increased internal residual fraction. Increased gas temperature has a tendency to shorten ignition delay. Secondly, the equivalence ratio decreases with increasing EGR. This is confirmed by **Fig. 13** for all operating load conditions. The increased temperature and reduced equivalence ratio due to dilution effect on combustion process. These factors caused the reduction in exhaust gas temperatures.

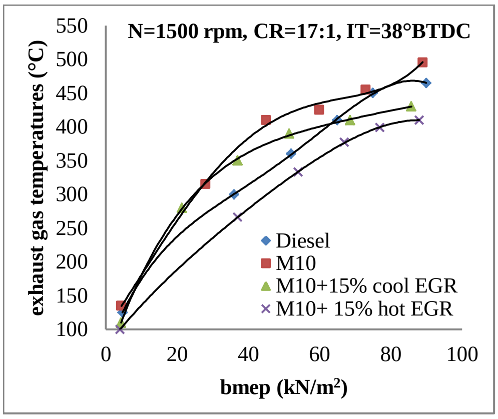

Figure 13: EGR mode effects on exhaust gas temperatures for variable brake mean effective pressure operating with diesel and M10 fuels

One of the most important thermodynamic parameters is the ignition delay period because it significantly influences an engine's combustion process, in particular the mixing process of the air charge and injected fuels, and thus also significantly influences the heat release rate. The lower heat values, higher viscosities and lower densities methanol and its blends necessitated adjusting the injection timing. **Fig. 14 & 15** represent injection timing effect on bp and bsfc for the studied modes.

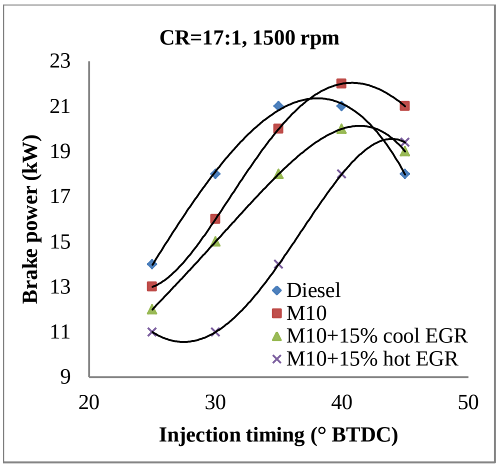

Figure 14: EGR mode effects on bp for variable engine injection timing with diesel and M10 fuels

When injection timing was retarded; combustion started later, so that, bp reduced and specific fuel consumption increased. When exhaust gas was recalculated the cylinder pressure exhibited decreases in cylinder pressure gradients and maximum cylinder pressures. The decreases were greater when the EGR was cooled. Advancing injection timing can reduce these penalties.

Jacobs et al concluded when many more inert components are in the cylinder and oxygen deficiencies increase because the EGR ratios increase, ignition points appear later. Initial conditions such as temperature and pressure greatly affect the emergence of the first ignition point.

With most of the fuels, ignition delays increased when the recirculated exhaust gas was cooled more strongly. Longer ignition delay periods accompanied with lower calorific heat blend caused the resulted bp to be less than with diesel fuel, and bsfc to be more than diesel fuel [15].

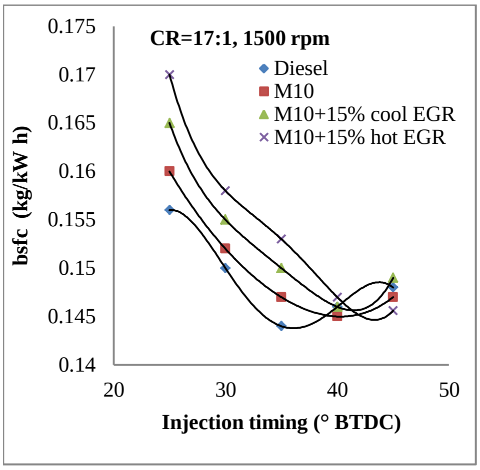

Figure 15: EGR mode effects on bsfc for variable engine injection timing with diesel and M10 fuels

Conclusions

This study was conducted to expand basic understanding of the differences in performance and behavior that result from the use of alternative fuels (methanol) and their blends in conjunction with recirculated exhaust gas in a diesel engine. The experimental results obtained provide a basis to optimize engine operating points with methanol.

1. Adding methanol with a rate of 10% to diesel fuel presented better specifications on bsfc and bp basis compared with neat diesel.
2. Adding methanol to diesel reduced the harm effects of EGR on engine performance by increasing available oxygen inside combustion chamber.
3. At constant engine speed (1500 rpm) increasing the load increased the thermal efficiency. Introducing EGR caused the thermal efficiency to decline.
4. At constant engine speed the use of cool EGR may be favorable in terms of less reduction in thermal efficiency.
5. Combining exhaust gas recirculation with lowered EGR temperatures produced better operating conditions. Combining oxygenated alternative fuels with exhaust gas recirculation substantially improves volumetric efficiency.
6. Injection timing optimizations can reduce the losses in bp and bsfc largely by advancing it to improve ignition delay.
7. Periods of ignition delay increased when EGR is cooled, decreasing initial temperatures lead to advancing injection timing. The higher initial temperature and pressure are (using hot EGR), the less injection timing crank angles advanced.
8. Ignition delay shortens when engine load increases. As the load increased the residual

gas temperature and the wall temperature raised this resulted in higher exhaust gas temperatures.

References

[1] C. Sayin, A. N. Ozsezen, M. Canakci, "The influence of operating parameters on the performance and emissions of a DI diesel engine using methanol-blended-diesel fuel," Fuel, Vol. 89, pp. 1407-1414, 2010.

[2] L. J. Wang, R. Z. Song, H. B. Zou, S. H. Liu, L. B. Zhou, "Study on combustion characteristics of a methanol–diesel fuel compression ignition engine," Proc Inst Mech Eng D – J Auto; Vol. 222, pp. 619–27, 2008.

[3] C. Sayin, K. Uslu, M. Canakci, "Influence of injection timing on the exhaust emissions of a dual-fuel CI engine", Renew Energy; Vol. 33, pp. 1314–1323, 2008.

[4] G. Najafia and T. F. Yusaf, "Experimental investigation of using methanol-diesel blended fuels in diesel engine," Proceedings of the Fourth International Conference on Thermal Engineering: Theory and Applications, January 12-14, Abu Dhabi, UAE, 2009.

[5] Y. Ren, Z. Huang, H. Miao, Y. Di, D. Jiang, K. Zeng, B. Liu, X. Wang, "Combustion and emissions of a DI diesel engine fuelled with diesel-oxygenate blends," Fuel, Vol. 87, pp. 2691–2697, 2008.

[6] T. Kulakoglu, "Effect of injection pressure on the performance and emissions of a diesel engine fueled with methanol–diesel blends," M. Sc. Thesis, Marmara University, Turkey, 2009.

[7] J. A. Araujo, B. Barajas, M. Kleinman, X. Wang, B. J. Bennett, K. W. Gong, M. Navab, J. Harkema, C. Sioutas, A. J. Lusis, A. E. Nel, "Ambient Particulate Pollutants in the Ultrafine Range Promote Early Anther sclerosis and Systemic Oxidative Stress," Circulation Research, Vol. 102, pp. 589- 596, 2008.

[8] Z. Huang, H. Lu, D. Jiang, K. Zeng, B. Liu, J. Zhang and X. Wang, "Performance and emissions of a compression ignition engine fueled with diesel/oxygenate blends for various fuel delivery advance angles,' Energy & Fuels, Vol. 19, pp. 403-410, 2005.

[9] Manufacturers of Emission Controls Association (MECA), "Retrofitting emission controls for diesel-powered vehicles," October-2009.

[10] K Nord, "Particles and unregulated emissions from CI engines subjected to emission control," PhD thesis, Luleå University of Technology Department of Chemical Engineering and Geosciences, Division of Chemistry, 2005.

[11] S. Chatterjee, R. Conway, S. Viswanathan, M. Blomquist, B. Klusener, S. Andersson, "NOx and PM Control from Heavy Duty Diesel Engines Using a Combination of Low Pressure EGR and a Continuously Regenerating Diesel Particulate Filter," SAE Technical paper No. 2003-01-0048.

[12] S. A. Niemi, T. P. J. Paanu, M. J. Laurén, "Effect of injection timing, EGR and EGR cooling on the exhaust particle number and size distribution of an off-road diesel engine," SAE Technical paper No. 2004-01-1988.

[13] Kh. I. Abaas, "Performance Characteristics of Methanol-Diesel Blends in CI Engines," Journal of Engineering, Vol. 17, No. 6, pp.1492-1501, 2011.

[14] K. Rajan and K. R. Senthilkumar, "Effect of exhaust gas recirculation (EGR) on the performance and emission characteristics of diesel engine with sunflower oil methyl ester," Jordan Journal of Mechanical and Industrial Engineering, Vol. 3, No. 4, pp. 306-311, 2009.

[15] T. Jacobs, D. Assanis and Z. Filipi, "The impact of exhaust gas recirculation on performance and emissions of a heavy-duty diesel engine," SAE paper No. 2003-01-1068, 2003.

[16] N. Ladommatos, S. M. Abdelhalim, H. Zhao, Z. Hu, "Effects of EGR on heat release in diesel combustion," SAE Paper No. 980184, 1998.

Notation

IT	injection timing
CN	cetane number
DI	direct injection
N	engine speed (rpm)
T	engine torque
V_{sn}	swept volume
°BTDC	degree before top dead center
bmep	brake mean effective pressure
BTE	brake thermal efficiency
CA	crank angle
CR	compression ratio
LCV	Lower calorific value
EGR	Exhaust gas recirculation

تأثير تدوير الغاز العادم البارد والساخن على أداء محرك اشتعال بالأنضغاط متعدد الأسطوانات يعمل بخلائط من وقود الديزل والميثانول

مقدام طارق جيجان قحطان عدنان عباس
قسم هندسة المكائن والمعدات، الجامعة التكنولوجية، بغداد، العراق

الخلاصة:

تم فحص أداء محرك ديزل متعدد الأسطوانات ذي حقن مباشر مزود بخليط من ديزل-ميثانول ويستخدم تدوير الغاز العادم المبرد والساخن، ودرس تأثير بعض متغيرات المحرك مثل النسبة المكافئة والحمل وسرعة المحرك وتوقيت الحقن له، وتمت الدراسة بتدوير غاز عادم ساخن وأعيدت بتدوير غاز عادم مبرد، وقورنت النتائج بمثيلاتها عند عمل المحرك بالديزل وبخليط ديزل+10%ميثانول.

تبين النتائج وجود تصرف مختلف للأحتراق لخلائط ديزل-ميثانول مقارنة مع ديزل صافي. ويؤثر تدوير الغاز العادم الذي يعتبر أحد أفضل الوسائل في تقليل ملوث NOx على أداء المحرك بشكل كبير. إن اضافة الميثانول الى الديزل تقلل من التأثيرات المؤذية لتدوير الغاز العادم على أداء المحرك بزيادة الأوكسجين المتوفر داخل غرفة الأحتراق. وينتج عن تدوير الغاز العادم المبرد كفاءة حرارية وكفاءة حجمية واستهلاك نوعي مكبحي للوقود وقدرة مكبحية أفضل مقارنة بالعمل بتدوير غاز عادم ساخن. ويحسن تقديم توقيت الحقن من أداء المحرك عند العمل بتدوير الغاز العادم.