PERFORMANCE AND ENERGY BALANCE OF A LOW HEAT REJECTION DIESEL ENGINE OPERATED WITH DIESEL FUEL AND ETHANOL BLEND

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ABSTRACT

In this study, it was aimed to investigate the effect of ceramic coating on a turbocharged diesel engine performance and energy balance. For this purpose, cylinder head, valves and pistons of the engine were coated with yttria stabilized zirconia layer with a thickness of 0.35 mm nickel-chromium-aluminium bond coat, as well as the atmospheric plasma spray coating method with a thickness of 0.15 mm. Then, the engines were tested for full load. The heating values of the diesel fuel and ethanol were 46.2 and 25.182 MJ/kg, respectively. Because of the lower heating values of the ethanol, compared with the diesel fuel, it appears to have lower following to engine power, torque and SFC. Compare to engine power of SDE, LHRe has increased about 2 %, LHReth has decreased about 22 % at all engine speed. Compare to engine torque of SDE, LHRe has increased about 2.5 %, LHReth has decreased about 23 % at all engine speeds. Compare to SFC of SDE, LHRe has decreased about 1.1 %, LHReth has increased about 54 % at all engine speeds. Compare to exhaust turbine inlet temperature of SDE, LHRe has increased about 15 %, LHReth has decreased about 17 % at all engine speeds.

PERFORMANCE ET BILAN ÉNERGÉTIQUE D'UN MOTEUR DIESEL À FAIBLE REJET DE CHALEUR ALIMENTÉ AU CARBURANT DIESEL ET AU BIOÉTHANOL

RÉSUMÉ

Dans cette recherche, nous étudions l'effet d'un revêtement de céramique sur la performance et le bilan énergétique d'un moteur diesel turbocompressé. À cette fin, la culasse, les soupapes et les pistons du moteur ont reçu une couche d'un revêtement à l'oxyde d'yttrium stabilisé au zirconium avec un alliage de scellement au nickel-chrome-aluminium d'une épaisseur de 0.35mm, ainsi que d'un revêtement par pulvérisation atmosphérique d'une épaisseur de 0.15mm. Ensuite les moteurs ont été testés en condition de pleine charge. Les taux de chaleur de combustion du carburant au diesel et à l'éthanol ont été évalués à 46.2 et 25.182 MJ/kg, respectivement. À cause du taux plus faible de chaleur de la combustion de l'éthanol, comparé au carburant diesel, il semble y avoir une puissance d'entraînement réduite du moteur, du couple, et de la consommation spécifique de carburant (CSC). Si on compare la puissance des petits moteurs diesel, le flux de chaleur rejetée a augmenté d'environ 2%, le taux de chaleur rejetée a diminué de 22% à tous les régimes du moteur. Si on compare au couple des petits moteurs diesel, le rejet de chaleur a diminué de d'environ 2.5%, le taux de chaleur rejeté a diminué d'environ 23%, à tous les régimes. Comparé à la consommation spécifique de carburant des petits moteurs diesel, la diminution du rejet de chaleur a diminué d'environ 1.1%, le taux de chaleur rejetée a augmenté de 54% à tous les régimes. Comparé à la température à l'entrée de la turbine d'échappement d'un petit moteur diesel, le flux de chaleur rejetée a augmenté d'environ 15%, le taux de chaleur rejetée a diminué d'environ 17% à tous les régimes.

Nomenclature

BSFC, Brake Specific Fuel Consumption
CA, Crank Angle
$C_{p,}$ Specific Heat of Air at Constant Pressure, [kj/kgK]
$C_{w,}$ Specific Heat of Water, [kj/kgK]
DI, Direct Injection
DME, Dimethyl Ether
LHR, Low Heat Rejection
LHRe, Low Heat Rejection Engine
LHReth, Low Heat Rejection Engine with Ethanol
$\overset{a}{m}$, Fuel Consumption Rate [kg/s]
m_f, Consumption Rate [kg/s]
m_w, Cooling Water Flow [kg/s]
Q_c, Energy Transferred to Coolant [kW]
Q_{ex}, Energy Lost to Exhaust [kW]
rpm, Revaluation per Minute
SDE, Standard Diesel Engine
T_a, Inlet Air Temperature [K]
T_c, Exhaust Temperature [K]
T_1, Cooling Water Inlet Temperature [K]
T_2, Cooling Water Outlet Temperature [K]
TS, Turkish Standard

1. INTRODUCTION

The increase in prices of petroleum based fuels, strict governmental regulations on exhaust emissions and future depletion of worldwide petroleum reserves have encouraged studies to search for alternative fuels [1]. Considerable attention has been focused on the development of alternative fuel sources, with particular reference to the alcohols [2].

Various techniques involving ethanol–diesel dual fuel operation have been developed, to make diesel engine technology compatible with the properties of ethanol-based fuels. They can be divided into the following three categories:

(1) Ethanol fumigation to the intake air charge, by using carburetion or manifold injection.
(2) Dual injection system that is not considered very practical, as requiring an extra high pressure injection system for the ethanol and, thus, a related major design change of the cylinder head.
(3) Blends (emulsions) of ethanol and diesel fuel by using an emulsifier to mix the two fuels in order to prevent separation; these require no technical modifications on the engine side [3].

Generally, ethanol can be blended with diesel with no engine modifications required. However, a major drawback in ethanol–diesel fuel blends is that ethanol is immiscible in diesel fuel over a wide range of temperatures and water content because of their difference in chemical structure and characteristics. These can result in fuel instability due to phase separation. Prevention of this separation can be accomplished in two ways: by adding an emulsifier acting to suspend small droplets of ethanol within the diesel fuel, or by adding a co-solvent acting as a bridging agent through molecular compatibility and bonding to produce a homogenous blend [4]. A new trend in the field of internal combustion engines is to insulate the heat transfer surfaces as a combustion chamber, cylinder wall, cylinder head, piston and valves by ceramic insulating materials for the improvement of engine performance and elimination of the cooling systems [5].

The quest for increasing the efficiency of an internal combustion engine has been going on since the invention of this reliable workhorse of the automotive world. In recent times, much attention has focused on achieving this goal by reducing energy lost to the coolant during the power stroke of the cycle. A cursory look at the internal combustion engine heat balance indicates that the input energy is divided into roughly three equal parts: energy converted to useful work, energy transferred to coolant and energy lost to exhaust. This phenomenon, of course, should conform to some physical laws. The first law of thermodynamics is satisfied as long as energy is converted, regardless of how that energy is apportioned between various categories. The second law stipulates that all the input energy cannot be converted into work; in other words, it is impossible to obtain 100 % of the efficiency, so some heat has to be rejected, preferably at the lowest possible temperature to achieve highest possible efficiency. The reduction in the in-cylinder heat transfer to either the coolant and / or the environment does not violate the second law of thermodynamics and also, according to the first law, has the potential of producing more work. Added to this, another important advantage of the concept is the great reduction in parasitic losses due to reduction of cooling system, thus increasing the brake horsepower of the engine. These prospects of improving the design and performance have generated impetus to active research on adiabatic or more appropriately, low heat rejection (LHR) or insulated engines [6].

Al-Baghdadi [7] declared that the effect of the amount of hydrogen / ethyl alcohol addition on the performance and pollutant emission of a four-stroke spark ignition engine has been studied. The addition of 8 mass % of hydrogen, with 30 vol % of ethyl alcohol into a gasoline engine operating at 9 compression ratio and 1500 rpm causes 58.5 % of a reduction in specific fuel consumption. Moreover, the engine thermal efficiency and output power increase by 10.1 and 4.72 %, respectively.

Li et al. [8] show that the effects of different ethanol–diesel blended fuels on the performance and emissions of diesel engines were evaluated experimentally and compared. The experiments were conducted on a water-cooled single-cylinder direct injection (DI) diesel engine using 0 % (neat diesel fuel), 5 % (E5–D), 10 % (E10–D), 15 % (E15–D), and 20 % (E20–D) ethanol–diesel blended fuels. In light of literature, the results provided from a different study indicated that the brake specific fuel consumption and brake thermal efficiency results in an increase of ethanol contents in the blended fuel at overall operating conditions; smoke emissions decreases with ethanol–diesel blended fuel, especially with E10–D and E15–D.

In another study it was suggested that an investigation of the effect of DME (Dimethyl ether) or ethanol on fuel consumption is conducted in a four-stroke, one-cylinder, direct-injection diesel engine. The results show that BSFC can be decreased by about 10 % and diesel fuel consumption can be decreased by 18 %. High saving rate of BSFC up to 10 % is also acquired using ethanol instead of DME. To achieve high saving rate of BSFC, the heating temperature of about 1000 K is needed for DME operation, while the diesel engine exhaust temperature of about 750 K is enough for pyrolyzing ethanol [9].

Abu-Qudais [10] explained that within the accumulated data, the effects of ethanol fumigation (i.e. the addition of ethanol to the intake air manifold) and ethanol diesel fuel blends on the performance and emissions of a single cylinder diesel engine have experimentally been investigated and compared. The results show that both the fumigation and blends methods have the same behavior in affecting performance and emissions, but the improvement in using the fumigation method was better than when using blends. The optimum percentage for ethanol fumigation is 20 %. This percentage produces an increase of 7.5 % in brake thermal efficiency.

Ciniviz et al. [11] explained that Compared with a standard diesel engine, engine power was increased by 2 per cent, the engine torque was increased by 1.5–2.5 %, and brake specific fuel consumption (b.s.f.c.) was decreased by 4.5–9 %.

Haşimoğlu et al. [12] declared that the results showed that specific fuel consumption and the brake thermal efficiency were improved and exhaust gas temperature before the turbine inlet was increased for both fuels in the LHR engine.

Parlak et al. [13] showed that In comparison to a standard Diesel engine, specific fuel consumption was decreased by 6 %, and brake thermal efficiency was increased by 2 %. It was concluded that the exhaust gas process was the most important source of available energy, which must be recovered via secondary heat recovery devices. The available exhaust gas energy of the LHR engine was 3–27 % higher for the LHR engine compared to the standard (STD) Diesel engine.

In this study, the effect of ceramic coating to a turbocharged diesel engine performance and energy balance were investigated. For this purpose, cylinder head, valves and pistons of the engine were coated with plasma spray zirconium with the thickness of 0.5 mm. Then, the engine was tested for different loads and speeds at standard and low heat rejection engine (LHRe) condition.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

The present study was conducted on a turbocharged diesel engine (OM364A, Mercedes-Benz). The engine is in line four-cylinders, turbocharged, direct injection and swept volume of 3972 cm^3. The general specifications of the engine are given in Table 1. A Go-Power hydraulic dynamometer (DT 3000, Go-Power Systems) was used for engine tests. The general specifications of hydraulic dynamometer are given Table 2. The schematic view of the test equipments is shown in Fig. 1. Chemical properties of ethanol and diesel fuel are given Table 3. Experiments were performed with two different fuels at full load. These are diesel fuel (cetane number-46), 10 % ethanol+ 90 % diesel fuel. The ethanol used in the experiments was pure 99.5 %. The experiments were performed at the engine speeds of 1100–2800 rpm. Before coating process, SDE tests were carried out according to TS (Turkish Standard) 1231. To keep the same

Table 1. The general specifications of the OM364A Mercedes-Benz engine.

Item	Specification
Trademark / type	Mercedes-Benz / OM364A
Engine type	DI turbo diesel
Cylinder arrangement	In-line 4
Bore (mm)	97.5
Stroke (mm)	133
Displacement (cc)	3972
Compression ratio	17.251
Nominal speed (rpm)	2800
Max. power	66 kW at 2800 rpm
Max. torque	266 Nm at 1400 rpm
Cycle	Four stroke
Injection advance	18°CA BTDC

Table 2. The general specifications of hydraulic dynamometer.

	Measurement Range	Sensitivity
Torque (Nm)	0–999	± 0.1
Engine Speed (rpm)	0–7500	1

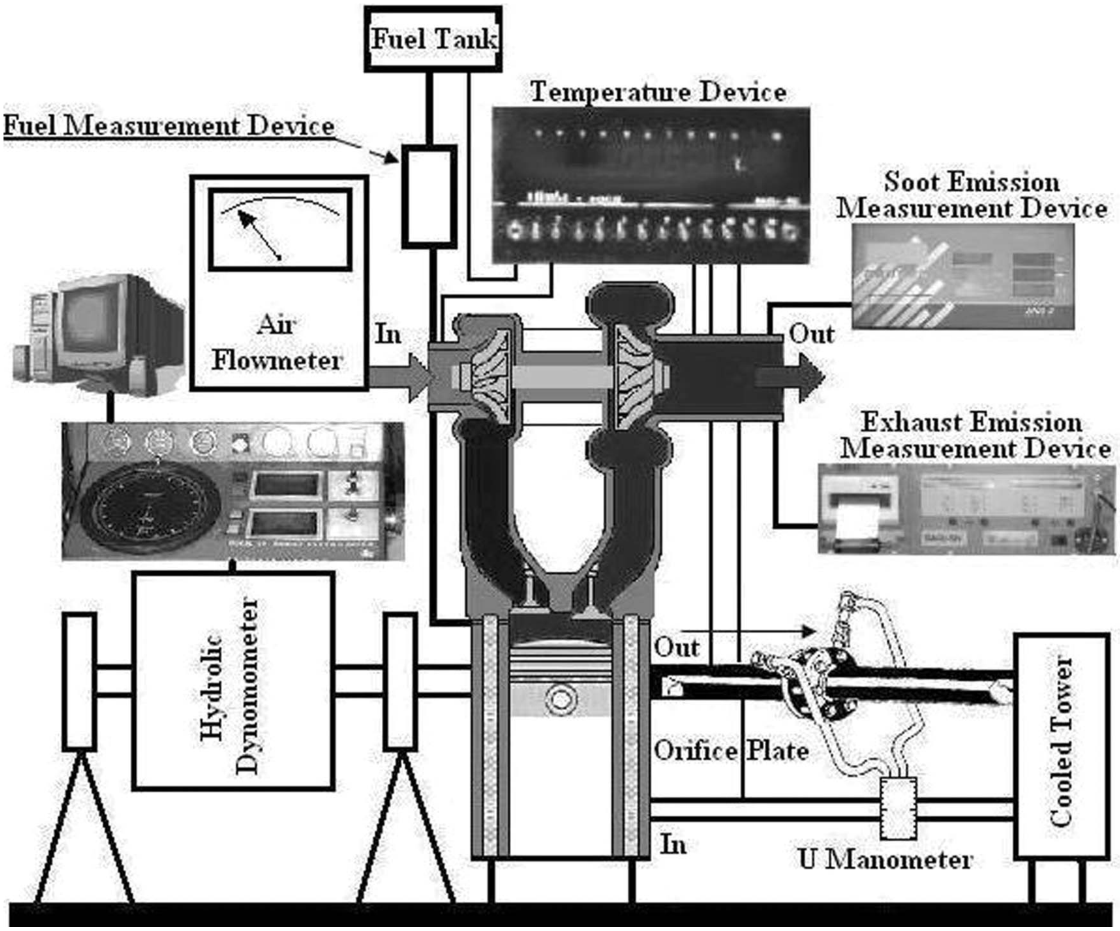

Fig. 1. Hydraulic dynamometer and cooling tower.

amount of size of SDE, 0.5mm chips were removed from the surface of cylinder heads, top of the pistons and valves.

In order to coat valves, pistons and cylinder head, following materials were used;

1. Standard Engine(SDE)
2. Main material + 0.15 mm NiCrAl + 0.35 mm + Y_2O_3-ZrO_2 (LHRe).

Engine test were conducted with full load at constant engine speeds of 1100-1200-1400-1600-1800-2000-2200-2400-2600-2800 rpm. Measurements were taken on SDE, LHRe and LHReth at 10 different revolution intervals when the engine was moved from 1100 to 2800 rpm. The values measured for each of the mentioned measurement intervals above were calculated for the

Table 3. Chemical properties of ethanol and diesel fuel.

	Formula	Mole weight (g)	Density at 20 °C (g/cm^3)	Boiling point (°C)	Flash point (°C)
Diesel	C_xH_y	190–220	0.829	180–360	65–88
Ethanol	CH_3 CH_2OH	46.07	0.789	78.5	9–11

Table 4. The uncertainties of measured and calculated parameters.

Parameter	Maximum errors ($\pm$%)
Power	0.3
Torque	0.39
Specific fuel consumption	0.45
Exhaust gas temperature	0.13

engines of SDE, LHRe and LHReth using a computer program. All measured performance values were obtained by using software. The uncertainty of the measured parameters is important for verifying the accuracies of the test results. The uncertainties of the measured parameters are shown in Table 4.

Power measurements were performed via a load cell on the moment branch of the dynamometer. The power formed while breaking the engine is transformed into tension by the load cell. Due to the decreased tension signal, it was increased by an instrumentation elevator (amplificator). The signal is administered onto a low permeable filter of 40 dB/dec in order to clear the signal out of unwanted noise. The output (opening) of the filter was transferred onto the micro controller by being turned into digital signals via the analogue digital transformer. Information of the engine speed provided from the part of revolution measurement was also passed onto the control panel. Micro controller passed the power and engine speed information into a computer via RS232 interface (circulation – connection) circuit [14]. Schematic presentation of system is in Fig. 2.

This assumes that the specific heat of the exhaust gas, the mass of which is the sum of the masses of air and fuel supplied to the engine, is equal to that of air. This is not strictly true, but permits an approximate calculation to be made if the temperature of exhaust gas is measured (exact measurement of exhaust temperature is no simple matter) [15].

Energy of exhaust transferred is defined as follows.

$$Q_{ex} = \left(\dot{m}_f + \dot{m}_a\right) + C_p T_e - \dot{m}_a C_p T_a \tag{1}$$

The heat transfer to cooling is found from the temperature rise in coolant as it passes through the engine and mass flow rate of coolant. Energy of cooling transferred is defined as follows.

$$Q_c = \dot{m}_w C_w (T_2 - T_1) \tag{2}$$

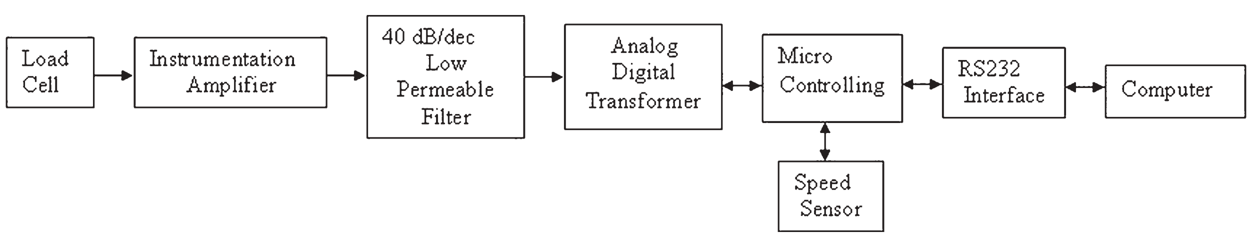

Fig. 2. The schematic view of the system of strength measurement.

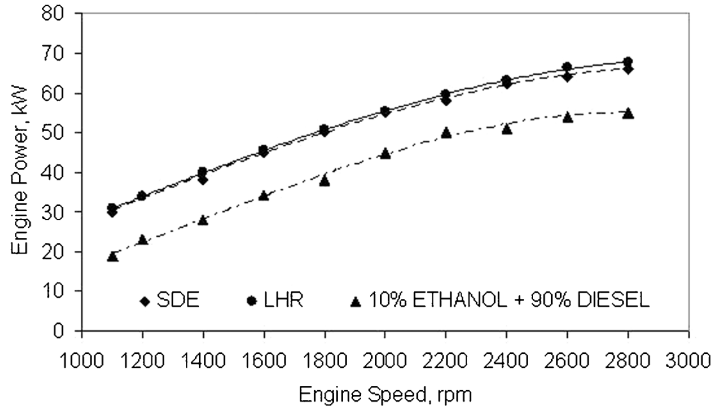

Fig. 3. The change of engine power depending on engine speed at full load.

3. RESULTS AND DISCUSSION

The results of the engine tests are presented in Figs. 3–8 for each condition. These conditions are SDE diesel (uncoated engine, diesel fuel), LHR diesel (coated engine, diesel fuel) and LHReth (coated engine, ethanol fuel). All comparisons were made according to the SDE diesel condition. The graphics include engine power, torque, specific fuel consumption, exhaust gas temperature, and energy balance changes according to engine speed (1400 and 2800 rpm).

The engine power versus engine speed is seen in Fig. 3 for each condition. The engine power increases by 2 % at all speeds in LHR diesel engine condition. In LHReth condition, the engine power decreases by 22.5 % at all speeds. For each condition, the variations of engine torque depending on engine speed are shown in Fig. 4. The engine torque increases by 2.5 % at all engine speeds in LHR diesel engine condition. In LHReth condition, the engine torque decreases by 23 % at all engine speeds. The engine brake power output increased with an increase in engine speed. Both the amount of injected liquid fuel and the amount of inlet air increased with engine speed, accordingly. In addition, a higher engine speed also caused an increase in friction frequency between the engine piston and the cylinder wall and thus reduced the heat transfer of the burning gas from the combustion chamber to the engine surround. As a consequence, the burning gas temperature in the cylinder was raised, thus resulting in an increase of the exhaust gas temperature with an increase in engine speed, as shown in Fig. 7. The

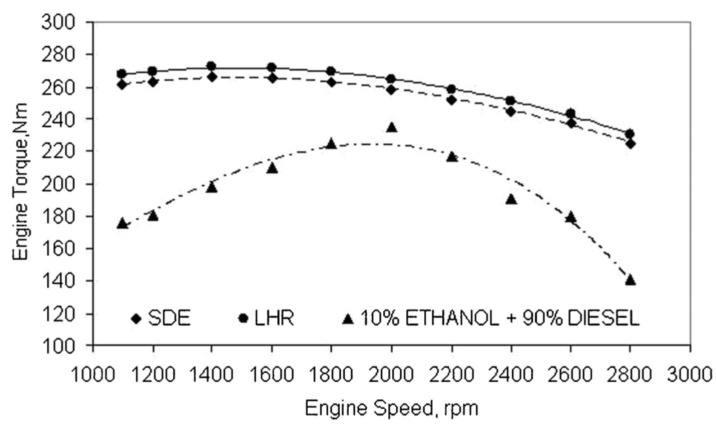

Fig. 4. The change of engine torque depending on engine speed at full load.

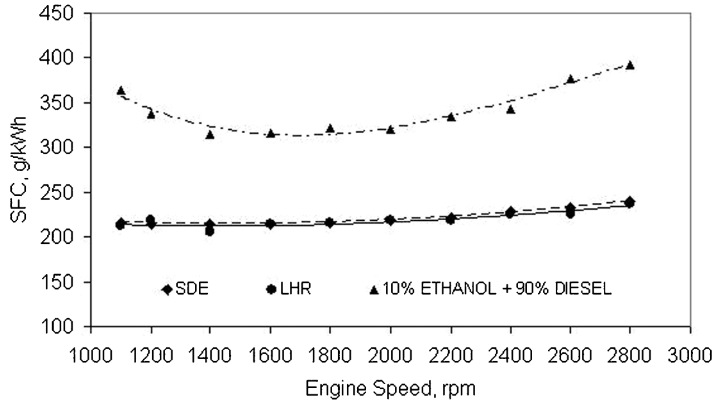

Fig. 5. The change of specific fuel consumption depending on engine speed at full load.

heating values of the diesel fuel and ethanol were 46.2 MJ/kg and 25.182 MJ/kg, respectively. Because of the lower heating values of the ethanol, compared with the diesel fuel, it appears to have lower exhaust gas temperatures, as can be seen in Fig. 7. The heating value of ethanol–air mixture is lower at low, medium and high speeds according to SDE diesel–air mixture. This explains the reason why the engine power and torque decreases at allowed speeds. The lower density of ethanol decreases the blended (LHReth) momentum and consequently penetration depth in-cylinder. Last two factors deteriorating sufficient combustible mixture formation in-cylinder lead to decrease in the power and the torque at all engine speeds in case of LHReth. The reason of the increase in the engine power and torque at higher speeds may be explained with the increase of turbulence in cylinder. The reason of increase in the power and torque in LHR diesel engine, compared to the SDE diesel condition, is the increase of exhaust gas energy. The increase of exhaust gas energy leads to improvement of volumetric efficiency because of the increased turbocharger outlet pressure.

The changes of specific fuel consumption versus engine speed are shown in Fig. 5. The specific fuel consumption is lower than by 1 % during all operating range of the SDE engine in the case of the use of LHR. Similarly, the specific fuel consumption increases approximately 54 % during all operating range of the SDE engine in the case of the use of LHReth. Huang et al. [16] explained that for the blend of Z5E25D70, the SFC were increased from 8.5% to 20.2%; for the blend of Z5E30D65, the BSFCs were increased from 11.1% to 31.5% (Z5 – 5 % n-butonol as a solvent, E5 - 5% ethanol, D70 – 70% diesel fuel). Bo et al. [17] show that BSFC can be

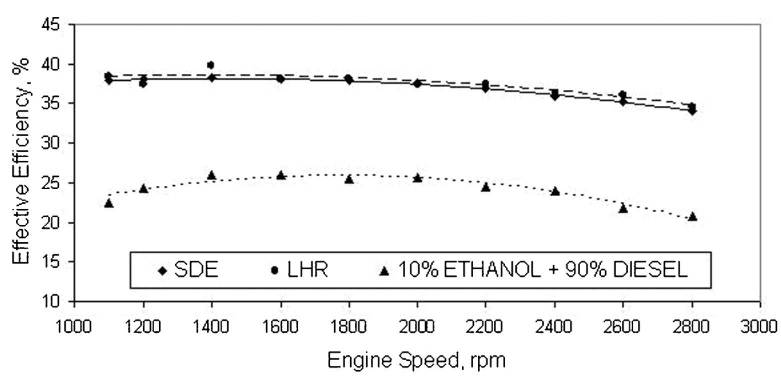

Fig. 6. The change of effective efficiency depending on engine speed at full load.

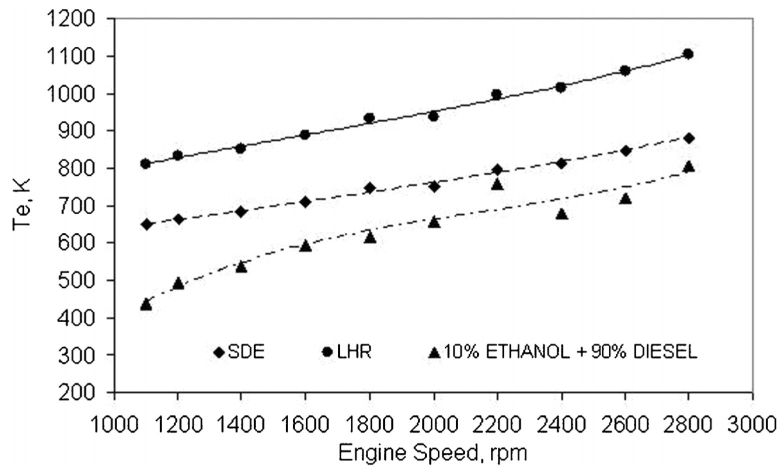

Fig. 7. The change of T_e, K (exhaust turbine inlet) depending on engine speed at full load.

decreased by about 10%. This can be attributed to the rate of engine power output increasing more than that of the fuel consumption rate when the engine speed was increased. In addition, the ethanol appeared to have larger SFC than diesel fuel, primarily owing to their lower heating values. Hence, for the ethanol, a larger fuel consumption rate must be supplied to attain the same engine power output as that of the diesel fuel. The positive effect of increased in-cylinder temperatures due to heat insulation decreases the specific fuel consumption in LHR engine.

The effective efficiency variations of all conditions are presented in Fig. 6. According to SDE, LHRe shows an increment of average 1% depending on engine speed at full load in effective efficiency. LHReth shows a decrement of average 35% depending on engine speed at full load in effective efficiency. Effective efficiency changes in SFC are affected. Mixture of lower heating value is lower than diesel fuel. This situation worsens motor performance.

The exhaust gas temperature variations of all conditions before the turbine inlet are presented in Fig. 7. In LHReth condition, the exhaust gas temperature before the turbine inlet decreases by 16.9 %. In LHR diesel condition, the exhaust gas temperature before the turbine inlet increases by 25 % at all engine speeds.

In general, the heat from exhaust gases also increases along with the increase in engine speed. Upon used diesel engine fuel in low heat rejection engines, the heat stemming from exhaust gases due to insulation increases the available energy of the exhaust gases. This condition also increases the amount of the air delivered to the cylinders by elevating output pressure of the turbo compressor.

Results of the SDE, LHRe and LHReth engines tests are shown in Figs. 8–9. Figures 8–9 represent the variation of the energy balance at 1400 rpm and 2800 rpm (full load). In each figure, the results were presented by taking SDE, LHRe and LHReth into account at the injection advance of 18° CA. At all speeds, the reduction in heat rejection mostly resulted in an increase in exhaust energy. The exhaust energy increased approximately by 11 % in LHRe engine, but decreased approximately by 60 % in LHReth engine at 1400 rpm, compared to the standard engine. However, the exhaust gas energy increased nearly by 8 % in LHRe, but decreased by 48 % in LHReth engine 2800 rpm, compared to the standard engine. In the LHRe configuration, because of the higher exhaust gas energy, there is an increase of volumetric efficiency. Furthermore, the inlet cycle may charge more than standard air/fuel mixing the delivery rate. As a result, the exhaust gas energy increased.

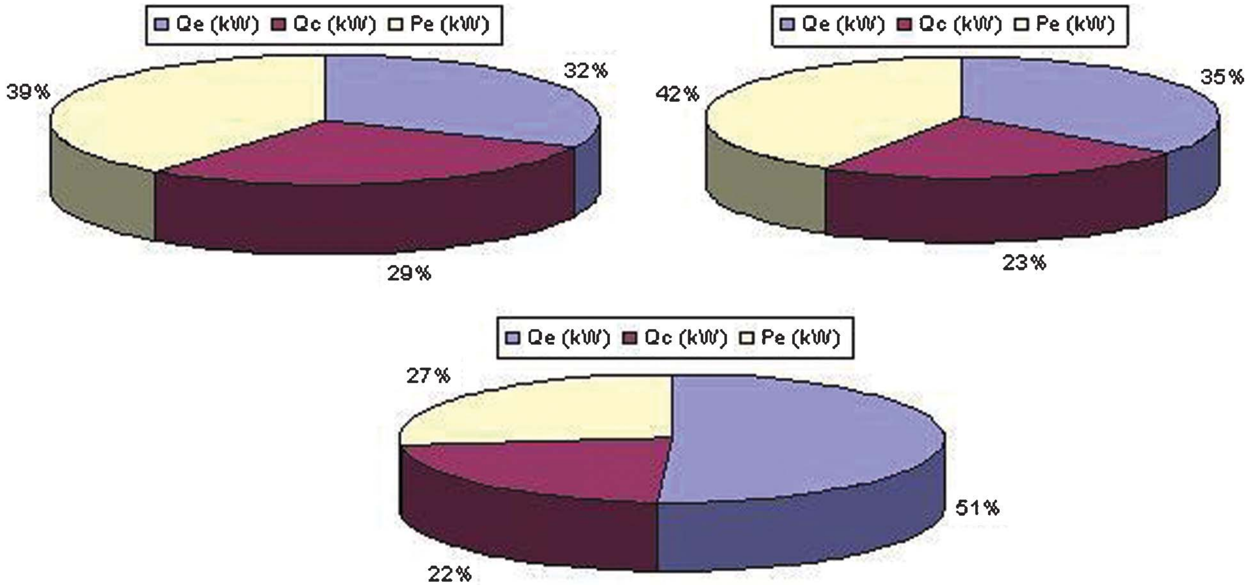

Fig. 8. Comparison of energy balance on 1400 rpm at full load.

Thermal barrier coatings of the combustion chamber reduced heat transfer to the coolant. Compared to SDE, the reduction rates in heat transfer to the coolant were 15 % in LHRe engine and 22 % in LHReth engine at 2800 rpm. Low heat rejection ceramics have a much lower thermal conductivity than metals so that the energy flow to the coolant will be reduced, and the higher combustion temperatures will lead to more expansion work. The increase of the combustion temperature causes the brake power to rise from 32 to 34 % at medium load and from 37 to 38 % at high load for LHRe. It can be seen that the values of the brake power are slightly higher (except for low load) for the LHRe case as compared to the standard case (without coating).

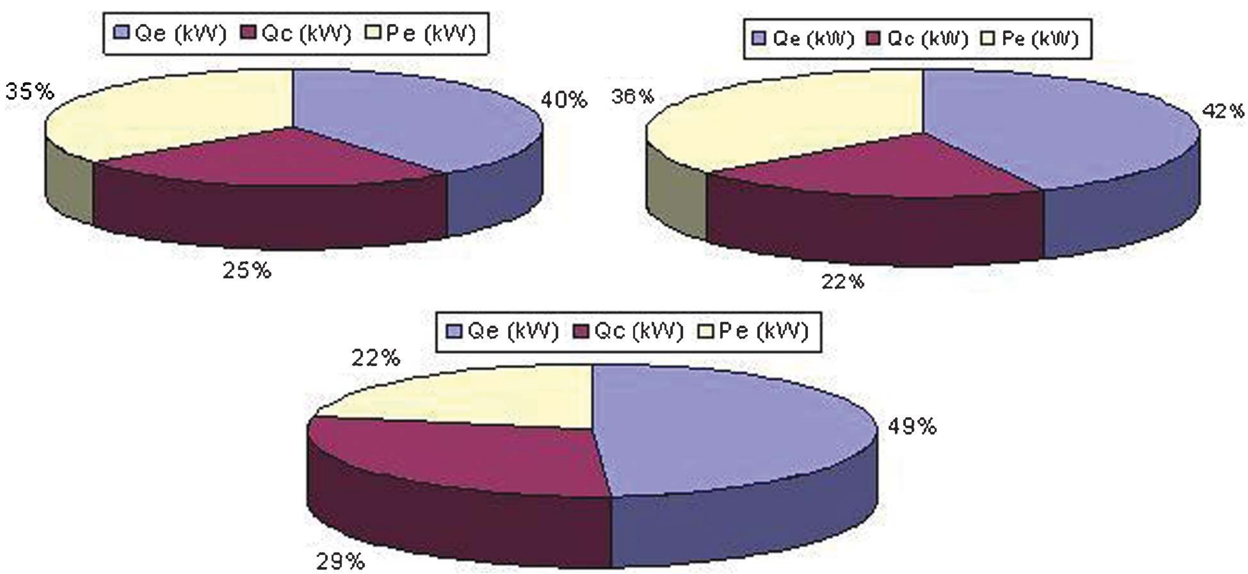

Fig. 9. Comparison of energy balance on 2800 rpm at full load.

4. CONCLUSIONS

Turbulence is directly proportional to engine speed. Engine speed increases, turbulence will increase. This situation increases the homogeneity of mixture. As a result of burning improves. An increased temperature in the cylinder shortens the ignition delay. It will make a positive impact on engine performance.

The engine power and torque were increased mainly due to the increased exhaust gas temperatures before the turbine inlet in LHR engine. The in-cylinder combustion temperatures are lowered due to the lower heating value of the ethanol fuel, less heat will be transferred to the engine parts, so the intake air temperatures decreases. This increases the volumetric efficiency when the ethanol was used as fuel. With the application of the thermal barrier coating the exhaust gas temperature before the turbine inlet becomes higher, this increases the turbine efficiency so the volumetric efficiency improves for the diesel fuel in LHR engine. However, the combined effects of the increased intake air temperature and lower exhaust gas temperature before the turbine inlet decrease the volumetric efficiency for the ethanol fuel in LHR engine. As the turbocharger did not compensate, it decreased intake air mass.

- The engine power increases by 2 % at all speeds in LHR diesel engine condition. In LHReth condition, the engine power decreases by 22.5 % at all speeds.
- The engine torque increases by 2.5 % at all engine speeds in LHR diesel engine condition. In LHReth condition, the engine torque decreases by 23 % at all engine speeds.
- The specific fuel consumption is lower than by 1 % during all operating range of the SDE engine in the case of the use of LHR. Similarly, the specific fuel consumption increases approximately 54 % during all operating range of the SDE engine in the case of the use of LHReth.
- In LHReth condition, the exhaust gas temperature before the turbine inlet decreases by 16.9 %. In LHR diesel condition, the exhaust gas temperature before the turbine inlet increases by 25 % at all engine speeds.
- The exhaust energy increased approximately by 11 % in LHRe engine, but decreased approximately by 60 % in LHReth engine at 1400 rpm, compared to the standard engine.

Recommendations for future works: Fuel injection timing and pressure can be changed to obtain an optimization of performance. Compression ratio can be changed to obtain an optimization of performance.

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