The Effect of Ethanol-Gasoline Blends on Thermal Balance of an SI Engine

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Abstract
The thermal balance of a four stroke, four-cylinder SI engine operating on ethanol-gasoline blends (0, 5, 10, 15 and 20%) was established. The thermal balance was respect of useful work, heat lost through exhaust, heat lost to the cooling water and unaccounted losses (heat lost by lubricating oil and radiation). The results indicate that as the percentage of ethanol in the ethanol-gasoline blends increased, the percentage of useful work increased, while the other losses decreased as compared to neat gasoline fuel. The maximum useful work and minimum heat losses were obtained at 20% ethanol-gasoline blend (E20). It can be concluded that an ethanol-gasoline blend ratio of 20:80 is an optimum blend as far as thermal balance is concern leaving the other parameters intact.

Keywords: SI engine - Thermal balance – Ethanol-gasoline blends - Gasohol

1. Introduction
The majority energy used today is obtained from fossil fuels. Due to continuous increase in the cost of fossil fuels, demand for clean energy has also been increased. Therefore, alternative fuel sources are sought. Some of the most important fuels are biogas, natural gas, vegetable oil and its esters, alcohols and hydrogen. Ethyl alcohol, which is one of the renewable energy sources, is obtained from biomass. It has been tested intensively in the internal combustion engines [1]. The use of alcohols as a fuel for spark ignition engines has some advantages compared to the gasoline. Ethanol owes better anti-knock characteristics than gasoline. The engine thermal efficiency can be improved with increasing compression ratio. Ethanol burns with lower flame temperatures and luminosity owing to decrease the peak temperature inside the cylinder, so the *NOx* emissions are lower. Ethanol has high latent heat of vaporization. The latent heat cools the intake air, so that the increased charge density increases volumetric efficiency. However, the oxygen content of ethanol reduces its heating value compared to gasoline which a disadvantage for ethanol that reduces the vehicle ranges per liter of fuel tank capacity [2–4].

Palmer [5] used various blend rates of ethanol–gasoline fuels in engine tests. Results indicated that adding 10% ethanol, increases the engine power output by 5% and the octane number can be increased by 5% for each 10% ethanol added.

The effects of ethanol and gasoline blends on spark ignition engine emissions were investigated by Hseih and his colleagues [6]. In their study, test fuels were prepared using 99.9% pure ethanol and gasoline blended with the volumetric ratios of 0–30% ethanol (E0, E5, E10, E20 and E30). These percentages represent the ratios of ethanol amount in total blends. In the experiments performed at different throttle openings and engine speeds, nearly the same torque values were obtained when used different ratios of ethanol–gasoline blends compared pure gasoline. Only the torque values obtained using E5 and E30 blends were lower than that of pure gasoline (E0) especially at high engine speeds (after 4000 rpm) and partly open throttle in 20%. It is reported that, this arose from the original fuel injection system strategies which prepare rich fuel mixtures. Therefore, the leaning effect of ethanol to increase the air fuel equivalence ratio (k) to higher value, and make the burning closer to stoichiometric. As a result, the better combustion can be achieved and the higher torque output can be acquired. The effect of alcohol and gasoline blends contained by mass ratios of 1.25%, 2.5%, 3.75% and 5% oxygen on engine emissions were investigated experimentally by Taylor et al. [7]. Methanol, ethanol, i-propanol and n-propanol were used as fuel. When using alcohol and gasoline blends contained by mass ratio of 5% oxygen, the HC and CO emissions were decreased by

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40% and 75%, respectively. The effects of ethanol and gasoline blends on Thermal balance of a single cylinder diesel engine were investigated by Ajav and others [8]. The results obtained indicated that the thermal balance of the engine operating on 5 and 10% ethanol-diesel blends and fumigated ethanol was not significantly different at the 5% level of significance when compared to diesel. However, in the case of 15 and 20% ethanol-diesel blends, the thermal balance was significant.

Energy is supplied to the engine in the form of chemical energy of the fuel and leaves the engine in the form of exhaust, cooling water, brake power and heat transfer. Heat losses must decrease to improve the engine efficiency. Therefore, it is very important to know the fractions of the heat losses mechanisms. This is the main aim of the thermal balance investigation. There are many reports about the effects of the ethanol on the performance and emission characteristics. However, it is apparent that a little information is available on thermal balance of electronically controlled SI engines operating on ethanol in the existing related literature. The objective of the study reported in this paper is therefore to establish and evaluate the thermal balance of SI engine.

2. Materials and Methods

2.1. Engine and Instrumentation

Experiments were conducted on a conventional spark ignition engine with 4-cylinder, 4-stroke, and electronically controlled electromechanical fuel injectors. Specifications of the engine are presented in Table 1. The engine was coupled to a SCHENCK. Make WT190 type hydraulic dynamometer to measure the power. At four various points, temperatures were measured using a temperature measuring device. Temperature measuring points are presented in Table 2. For measuring the fuel flow rate, a PIERBURG type, flow meter was used. The experimental setup is shown in Fig. 1.

Table 1- Technical specifications of the test engine

Engine type	SOHC, fuel injected
Number of cylinders	4
Compression ratio	9.7
Bore (mm)	71
Stroke (mm)	83.6
Displacement volume (cc)	1323
Max. power (kW)	64 at 5200 rpm
Cooling system	Water-cooled

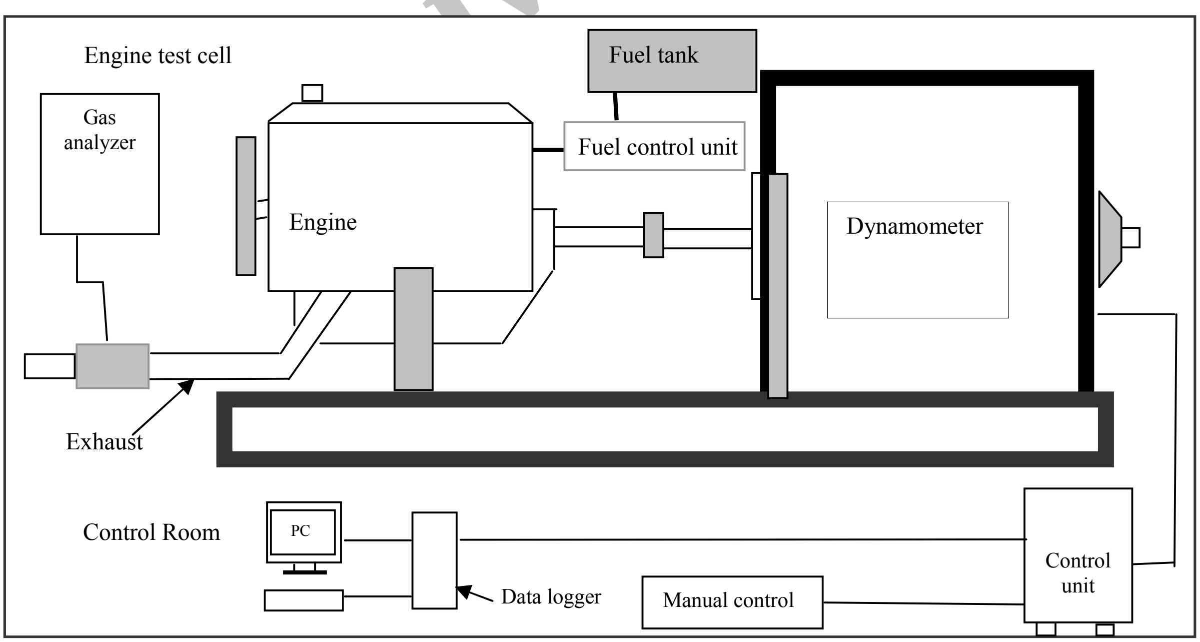

Fig. 1- Schematic diagram of experimental apparatus.

Table 2- Points of thermocouples

T1	Inlet water to engine
T2	Outlet water from engine
T3	Inlet air
T4	Outlet exhaust gases to manifold

2.2. Testing procedure

The engine was operated with ethanol-gasoline blends having 5, 10, 15 and 20% ethanol on volume basis (E5, E10, E15 and E20). At the beginning, the engine was run at a three-quarter opening position of the throttle valve in order to attain near maximum speed. After the engine reached the steady-state conditions, the first experiment was conducted with pure gasoline as a basis for comparison. The engine was gradually loaded by the hydraulic dynamometer and test matrix consisted of nine speeds ranging from 1000-5000 rpm with 500 rpm interval for each fuel operation. For each speed condition, the engine was run for at least three minutes, and the temperatures for the various points were recorded. The experiments were replicated three times to record the average values. The thermal losses through the various points were calculated as follows: The total heat supplied by the fuel (Q) was calculated by the following formula:

$$Q = \dot{m}_f \, CV \ , \tag{1}$$

Where, $\dot{m}_f$ is the fuel consumption (kg/s) and *CV* is the lower calorific value of the fuel, (kJ/kg).
The brake power delivered by the engine (P_b) and absorbed by the dynamometer was:

$$P_b = 2.\pi.N.T.10^{-3}, \tag{2}$$

where, *N* is the crankshaft rotational speed (rev/s) and *T* is the torque (Nm). The heat rejected to the coolant water (Q_w) was determined by:

$$Q_w = \dot{m}_w .C_w .(T_2 - T_1)\,, \tag{3}$$

Where m_w is the water flow rate (kg/s), C_w is the specific heat of water (kJ/kg ºC), T_1 is the inlet water temperature (ºC) and T_2 is the outlet water temperature (ºC).

The sensible enthalpy loss is considered for the exhaust flow in this study. The heat lost through the exhaust gases (Q_e) was calculated considering the heat necessary to increase the temperature of the total mass (fuel + air), m_e (kg/s), from the outside conditions T_3 (ºC) to the temperature of the exhaust T_4 (ºC). This heat loss is also known as sensible heat, and to calculate it, it is necessary to estimate the mean specific heat of the exhaust gases (C_e), which, in this case, is assumed to be the value for air with a mean temperature of the exhaust [7].

$$Q_e = \dot{m}_e .C_e .(T_4 - T_3). \tag{4}$$

The unaccounted heat losses are the heat rejected to the oil plus convection and radiation heat losses from the engine external surfaces. The unaccounted heat losses (Q_u) are given as

$$Q_u = Q - (Q_w + Q_e + P_b) \tag{5}$$

4. Results and Discussion

The thermal balance of the test engine operating on gasoline and ethanol-gasoline blends was established at different engine speed conditions. The thermal balance analysis and evaluation was carried out regarding useful work, heat lost to cooling water, heat lost through the exhaust and unaccounted losses. The engine thermal balance for the gasoline alone and ethanol-gasoline blends at various speeds (1000, 2000, 3000, 4000 and 5000 rpm) used in this work are presented in Fig.2-6. It can be seen from this figure that as the percentage of ethanol in the ethanol-gasoline blends is increased, the percentage of useful work is also increased, while the other losses are decreased compared to neat gasoline fuel operation. That the maximum useful work and minimum heat losses were obtained at E20.

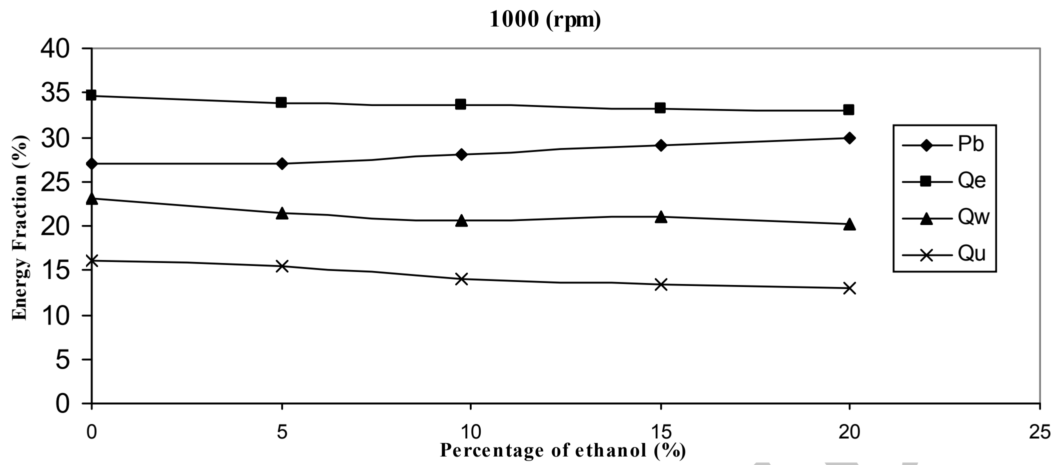

Fig. 2- Thermal balance of engine operating on 0, 5, 10, 15 and 20% ethanol- gasoline blends at 1000 (rpm)

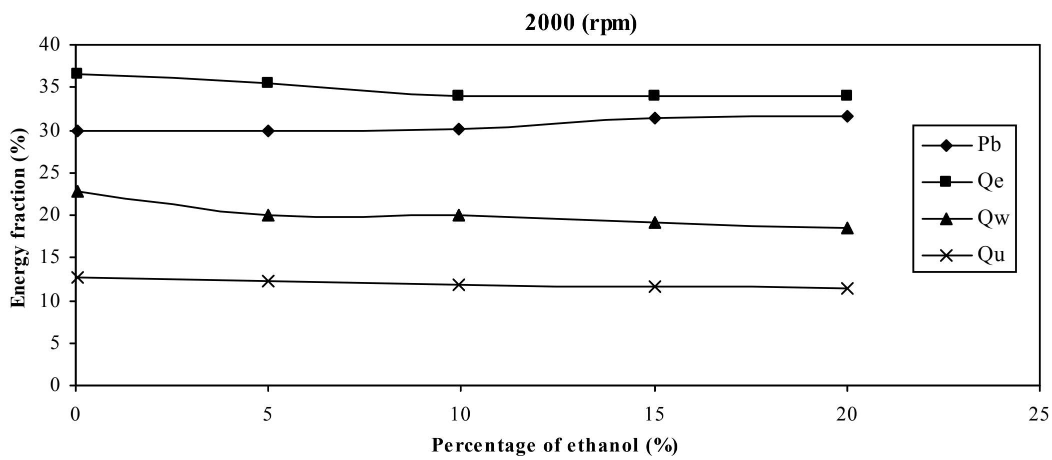

Fig. 3- Thermal balance of engine operating on 0, 5, 10, 15 and 20% ethanol- gasoline blends at 2000 (rpm)

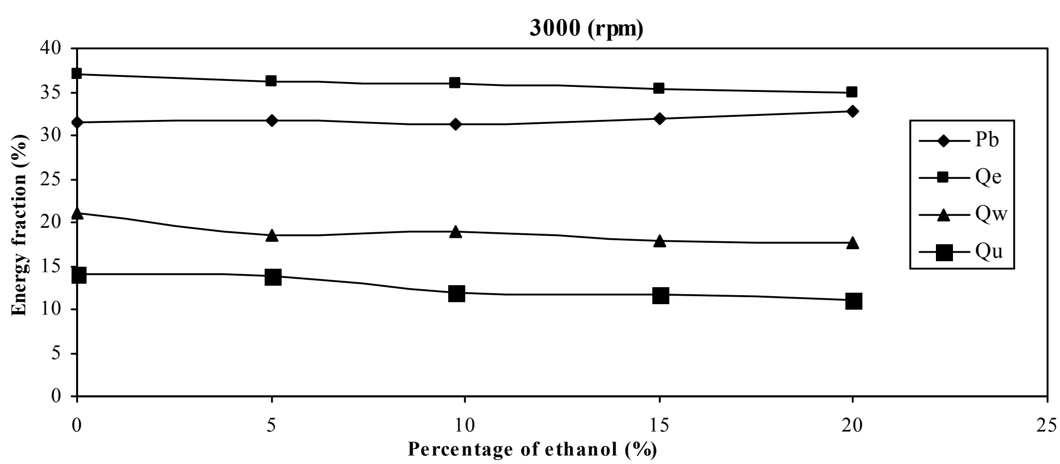

Fig. 4- Thermal balance of engine operating on 0, 5, 10, 15 and 20% ethanol- gasoline blends at 3000 (rpm)

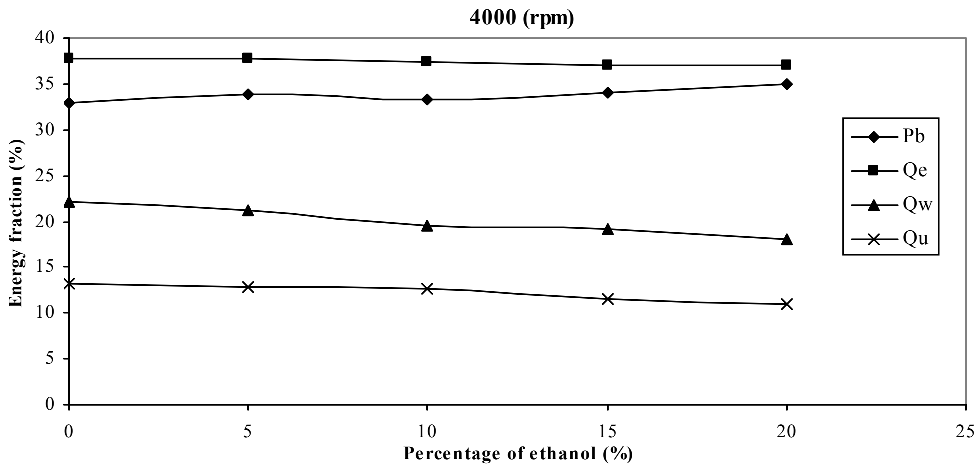

Fig. 5- Thermal balance of engine operating on 0, 5, 10, 15 and 20% ethanol- gasoline blends at 4000 (rpm)

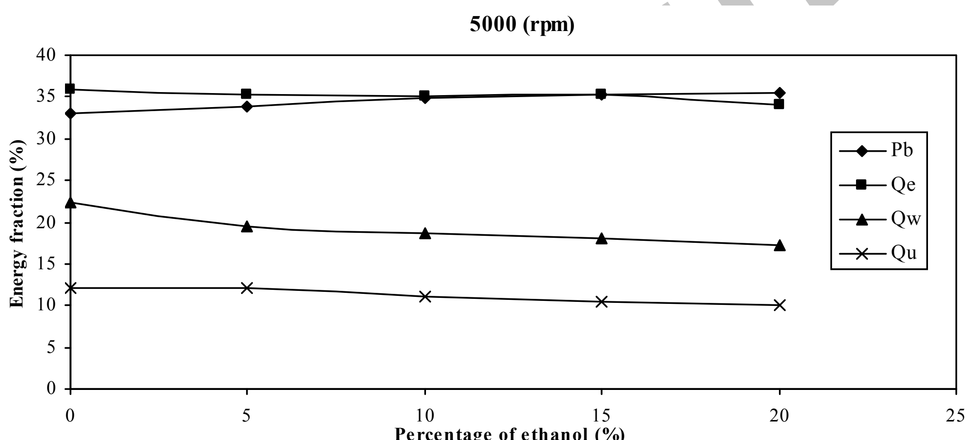

Fig. 6- Thermal balance of engine operating on 0, 5, 10, 15 and 20% ethanol- gasoline blends at 5000 (rpm)

This is due to the chemical structure of ethanol that contains an oxygen atom in its basic form. It therefore can be treated as a partially oxidized hydrocarbon. When ethanol is added to the blended fuel, it can provide more oxygen for the combustion Process and leads to the so-called ''leaning effect'', cooling effect of ethanol as well as more efficient combustion as compared to gasoline. Ethanol burns with lower flame temperatures and luminosity owing to the decrease of the peak temperature inside the cylinder, so both the exhaust gas temperature as well as the cooling water temperatures was lower in the case of ethanol-gasoline blend operations. There was less heat loss through these channels, and as such, more useful work was available at the engine crankshaft. Tables 3–7 shows the variation of useful work and heat losses with engine speed at different blends. It can be seen from the results, which useful work and heat losses increased with increased speed.

Table 3- Data of thermal balance at E0

N (rpm)	Q (kW)	P_b (kW)	Q_e (kW)	Q_w (kW)	Q_u (kW)
1000	17.82	4.81	5.18	3.90	3.70
1500	24.10	7.13	7.76	4.34	4.00
2000	37.34	11.20	11.78	6.60	4.96
2500	52.54	16.33	15.2	11.56	6.10
3000	67.80	21.30	25.57	12.64	7.58
3500	78.71	25.18	24.15	16.36	10.20
4000	91.25	30.00	32.15	18.40	10.50
4500	97.60	33.10	32.05	20.21	12.13
5000	103.20	34.50	35.5	20.5	12.50

Table 4- Data of thermal balance at E5

N (rpm)	Q (kW)	P_b (kW)	Q_e (kW)	Q_w (kW)	Q_u (kW)
1000	18.20	4.72	6.33	3.90	2.79
1500	25.97	7.24	9.89	4.42	3.65
2000	38.87	11.58	15.06	7.75	4.05
2500	53.29	16.30	17.77	10.39	6.63
3000	70.41	22.25	23.89	15.82	8.90
3500	80.96	26.00	28.35	15.75	9.50
4000	96.06	32.10	34.00	18.48	11.00
4500	98.00	32.50	35.05	19.03	12.06
5000	103.80	35.00	35.89	20.10	12.00

Table 5- Data of thermal balance at E10

N (rpm)	Q (kW)	P_b (kW)	Q_e (kW)	Q_w (kW)	Q_u (kW)
1000	18.34	4.78	6.34	4.00	2.50
1500	25.67	7.06	8.74	5.33	3.60
2000	38.80	10.86	11.80	8.00	5.00
2500	53.60	16.03	19.19	11.63	6.77
3000	68.71	21.50	24.60	13.91	8.75
3500	80.92	26.20	28.70	16.78	9.84
4000	93.24	31.00	33.77	17.79	10.67
4500	103.20	35.00	35.65	19.96	11.98
5000	106.27	37.00	39.01	19.3	12.09

Table 6- Data of thermal balance at E15

N (rpm)	Q (kW)	P_b (kW)	Q_e (kW)	Q_w (kW)	Q_u (kW)
1000	18.46	5.16	7.51	3.09	2.85
1500	25.21	8.06	9.80	4.49	3.00
2000	40.57	12.57	14.32	8.90	5.14
2500	54.00	16.85	18.56	9.24	6.56
3000	70.22	22.47	24.32	14.04	8.34
3500	81.34	27.00	28.08	16.16	10.09
4000	95.63	32.50	34.77	18.35	11.01
4500	99.70	34.00	37.50	20.04	13.26
5000	103.80	36.60	36.70	21.09	13.4

Table 7- Data of thermal balance at E20

N (rpm)	Q (kW)	P_b (kW)	Q_e (kW)	Q_w (kW)	Qu (kW)
1000	18.51	6.13	6.54	3.10	2.86
1500	28.49	9.68	9.17	5.57	3.74
2000	40.93	14.32	13.62	8.06	4.24
2500	55.74	19.62	19.70	10.67	6.80
3000	70.73	25.00	24.59	13.21	8.92
3500	84.01	30.33	28.01	15.40	10.10
4000	96.10	35.00	33.10	17.50	10.50
4500	102.56	38.02	34.5	18.30	10.80
5000	105.60	39.00	35.30	19.10	11.46

6. Conclusions

The present experimental study showed that ethanol addition has a significant effect on the percentage useful work and heat losses from the engine, whereas useful work increased, and heat losses decreased, with increased percentage of ethanol in the ethanol-gasoline blends. So that the maximum useful work and minimum heat losses were obtained at E20 .

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