International Journal of Engineering

Journal Homepage: www.ije.ir

Exergy and Energy Analysis of Diesel Engine using Karanja Methyl Ester under Varying Compression Ratio

M. K. Rath, S. K. Acharya*, P. P. Patnaik, S. Roy

Department of Mechanical Engineering, ITER, SOA University, Bhubaneswar-751030, India

PAPER INFO

Paper history:
Received 26 October 2013
Received in revised form 01 April 2014
Accepted in 17 April 2014

Keywords:
Karanja methyl ester
Exergy
Biofuels
Variable compression ratio
Diesel engine
Energy

ABSTRACT

In the present study, there is a need to reduce the consumption of conventional fuel and related fossil fuels. So, in order to promote the use of renewable sources such as biofuels there is a requirement for the effective evaluation of the performance of engines based on the second law of thermodynamic. In this paper, the analysis of energy, exergy, mean gas temperature and exhaust gas temperature of a compression ignition (CI) engine using diesel and Karanja methyl ester blends are done. The results are calculated at various compression ratios under full load and at different engine loads at a compression ratio of 18:1. It is observed that the exergy efficiency, mean gas temperature, brake thermal efficiency increases with increase in compression ratio as well as load. The exhaust gas temperature and destruction of exergy decreases with increase in compression ratio and increases with increase in load for all blends of fuel.

doi: 10.5829/idosi.ije.2014.27.08b.12

NOMENCLATURE

BSFC	Brake specific fuel consumption, kg/kWhr
BTE	Brake Thermal Efficiency, %
E_x	Exergy consumption, kW
LCV	Lower calorific value of fuel, kJ/kg
$\dot{m}$	Mass flow rate of fuel, kg/s
$\dot{m}_a$	Mass flow rate of intake gas, kg/s
$\dot{Q}in$	Rate of fuel energy input to the engine, kW
$\dot{Q}_1$	Rate of heat energy taken away by cooling water, kW
$\dot{Q}_2$	Rate of heat energy taken away by exhaust gas, kW
$\dot{Q}_3$	Rate of unaccounted loss of heat energy, kW
$\dot{W}_e$	Rate of energy converted to electric power, kW
$\dot{Q}_{out}$	Rate of energy rejected to the environment, kW
T	Temperature, K

Greek Symbols

η_{ex}	Exergy efficiency
δE_x	Exergy destruction
η_{th}	Thermal efficiency

Subscripts

H	high temperature (used for the combustion chamber)
In	input
L	low temperature (used for the ambient)
Out	output
w	cooling water

*Corresponding Author's Email: saroj.acharya76@gmal.com (S.K. Acharya)

Please cite this article as: M. K. Rath, S. K. Acharya, P. P. Patnaik, S. Roy, Exergy and Energy Analysis of Diesel Engine using Karanja Methyl Ester under Varying Compression Ratio. International Journal of Engineering (IJE) Transactions B: Applications Vol. 27, No. 8, (2014): 1259-1268

1. INTRODUCTION

From the economic point of view diesel fuel is very important for its wide range of usage such as in transportation, power generation, road vehicles, agriculture etc [1, 2]. The substitution of these fuels is of great interest because it will reduce their dependence on petroleum based energy and will create jobs for rural areas where agriculture is the main activity. Most of the studies on engines using biodiesel have evaluated their performance based on their brake power, brake thermal efficiency (BTE), brake specific fuel consumption (BSFC) and emissions analysis [3-5]. By analysing such parameters thermal pollution is ignored and the real performance of the engines with respect to the second law of thermodynamics is overlooked. In this paper, emphasis is given to use the exergy concept to evaluate the performance of a CI engine using Karanja biodiesel under varying compression ratio. The exergy of an isolated system is a property that is not conserved. It can be generated, destroyed and stored. Exergy is destroyed mostly as low temperature heat transfer as well as during chemical reactions [6]. This unutilized exergy normally interacts with its surroundings and can lead to undesirable environmental processes in non-equilibrium situations that can cause environmental problem. The exergy content of a natural material input can be interpreted as a measure of its quality or potential usefulness, or its ability to perform useful work [7]. The deviation between the efficiency predicted by the first law of thermodynamics and the observed efficiency of the engine is an evidence of the defective exploitation of the fuel, which is due to the presence of internal irreversibilities in the engine. First law of thermodynamics cannot identify and quantify the sources of losses (irreversibilities) in the engine. It simply provides the overall efficiency and hence fails to clarify the reasons of deviation between the ideal and actual performance of the engine [8, 9]. On the other hand, second law of thermodynamics offers a new perspective for the analysis of the performance of the engine based on the concept of exergy. The objective of this paper is to analyze the energy, destruction of exergy and exergy efficiency of a CI engine using pure diesel & Karanja methyl ester in different blends (20%, 40%, 60%) with diesel at variable loads and compression ratios.

2. BASIC THEORY OF ENGINE ENERGY AND EXERGY BALANCE

2. 1. Energy and Exergy Balance System and Equations

Internal Combustion (IC) engine energy balance is also called thermal balance, which is the research of IC engine energy distribution from the angle of system integration. A commonly used method is to build an energy balance system including all kinds of energy in IC engine [10]. Then, the proportions of various kinds of energy in each of the subsystems can be studied. Therefor, by optimizing its energy distribution, energy utilization efficiency of IC engine can be improved. By the in-cylinder energy balance, total fuel chemical energy can be divided into effective work, heat transfer loss, friction loss, exhaust gas energy and unburned product chemical energy. The heat transfer loss is taken away by cooling water. IC engine energy balance can be effectively analysed by considering the stable energy balance equation and the concept of control volume as shown in Figure 1.

There are two groups of energy considered in the control volume out of which the energy input into the control volume includes fuel chemical energy and intake enthalpy. Energy outflow from control volume consists of effective output power, exhaust gas energy, cooling water energy, unburned fuel energy and heat transfer from the surface of IC engine by convection and radiation. The second law analysis in this paper is done by considering the engine as a heat engine (Figure 2) and it works between two temperatures reservoirs, the hot side at combustion temperature T_H and the cold side at ambient temperature T_L.

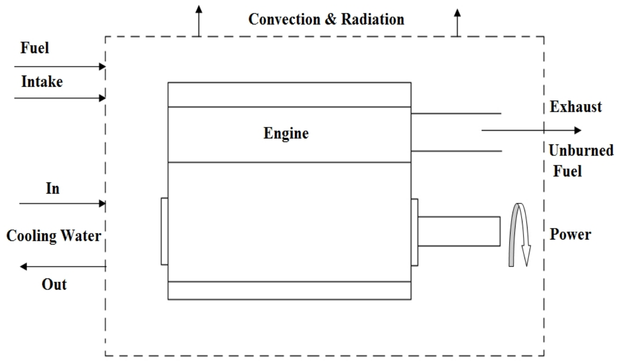

Figure 1. Internal Combustion engine energy balance system (control volume)

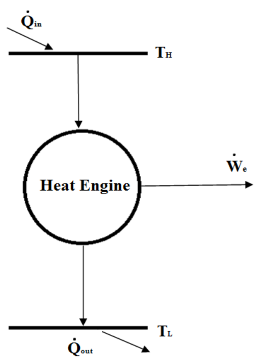

Figure 2. Heat engine

The stable energy balance equation gives the relationship among various kinds of energy, as follows. Heat supplied to the engine $\dot{Q}_{in}$ at T_H is evaluated from the mass flow rate of fuel and fuel low calorific value.

$$\dot{Q}_{in} = \dot{m}\mathrm{LCV} \tag{1}$$

One part of the heat supplied $\dot{Q}_{in}$ is converted into electrical power $\dot{W}_e$ (engine loads) and the other part $\dot{Q}_{out}$ is rejected to the environment. By applying the first law of thermodynamics to this heat engine system, we obtain:

$$\dot{Q}_{in} = \dot{W}_e + \dot{Q}_{out} \tag{2}$$

Equation (2) is about the quantity of the power streams involved in the system. However, it does not give any information about their quality. Quality is the message given by the second law of thermodynamics:

$$\frac{\dot{Q}_{in}}{T_H} - \frac{\dot{Q}_{out}}{T_L} + \dot{S}_{gen} = 0 \tag{3}$$

where,
$\dot{S}_{gen}$= The entropy generation rate of the system relative to environment.
Equations (2) and (3) follows the exergy (E_x) function that gives information not only on the quantity of the powers but also on their quality [11].
Exergy is also called available energy and is used evaluate the energy quality. The exergy that is considered in this paper indicates the maximum recoverable energy after combustion [12]. For the sake of simplicity, chemical exergy is not taken into account. The exergy consumption accounting equation for engine is given by:

$$E_x = \dot{Q}_{in}\left(1 - \frac{T_L}{T_H}\right) = \dot{W}_e + \delta E_x \tag{4}$$

where,
E_x= Exergy consumption rate, kW
T_H= The hot side temperature or combustion chamber temperature, K.
δE_x= The rate of destruction in exergy, kW.
The destruction of exergy is calculated using Equation (4) by putting the value of input energy, cold side and hot side temperature and rate of energy converted to electric power. The Gouy-Stodola version of destruction of exergy is:

$$\delta E_x = T_L\dot{S}_{gen} \tag{5}$$

The brake thermal efficiency (η_e) of the current system which is the ratio of rate of energy converted to electric power to the input fuel energy rate is given by:

$$\eta_e = \frac{\dot{W}_e}{\dot{Q}_{in}} \tag{6}$$

The exergy efficiency (η_{ex}) is the ratio of rate of energy converted to electric power to the exergy consumption rate and is given by:

$$\eta_{ex} = \frac{\dot{W}_e}{E_x} \tag{7}$$

From Equation (4), the determination of the combustion chamber temperature is necessary to evaluate the exergy consumption and the exergy efficiency.
$\dot{Q}_{out}$ is given by:

$$\dot{Q}_{out} = \dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3 \tag{8}$$

$$\dot{Q}_1 = \dot{m}_w \cdot C_w \cdot (T_{2w} - T_{1w}) \tag{9}$$

where,
$\dot{m}_w$ = Mass flow rate of cooling water, kg/s
C_w = Specific heat capacity of cooling water,
T_{2w} = Cooling water temperature at outlet, K
T_{1w} = Cooling water temperature at inlet, K
The rate of heat energy taken away by the exhaust gas can be written as the difference between exhaust gas enthalpy and intake gas enthalpy.

$$\dot{Q}_2 = \dot{H}_2 - \dot{H}_1 \tag{10}$$

where,
$\dot{H}_2$ = Rate of exhaust gas enthalpy, kW
$\dot{H}_1$= Rate of intake gas enthalpy, kW
The calculation formula of rate of exhaust gas enthalpy rate is:

$$\dot{H}_2 = (\dot{m} + \dot{m}_a) \cdot C_{p\_ex} \cdot T_{ex} \tag{11}$$

where,
C_{p_ex} = Specific heat capacity of the exhaust gas,
T_{ex} = Exhaust gas temperature, K
The calculation formula of rate of intake gas enthalpy rate is:

$$\dot{H}_1 = (\dot{m} + \dot{m}_a) \cdot C_{p\_in} \cdot T_L \tag{12}$$

where,
C_{p_in} = Specific heat capacity of the intake gas,
T_L = The cold side at ambient temperature, K
The brake specific fuel consumption (BSFC) kg/kWh is given by:

$$\mathrm{BSFC} = \frac{\dot{m}}{\dot{W}_e} \tag{13}$$

The percentage of each kind of energy in terms of total energy is given by:

$$\eta_f = \eta_e + \eta_w + \eta_{exh} + \eta_u = 100\% \tag{14}$$

η_f= Total percentage of fuel chemical energy
η_w is the percentage of cooling water energy in total fuel energy and it is calculated by:

$$\eta_w = \frac{\dot{Q}_1}{\dot{Q}_{in}} \tag{15}$$

η_{exh}, which is the percentage of exhaust energy in total fuel energy is given by:

$$\eta_{exh} = \frac{\dot{Q}_2}{\dot{Q}_{in}} \tag{16}$$

The percentage of unaccounted loss of energy in total fuel energy (η_u) is calculated by:

$$\eta_u = \frac{\dot{Q}_3}{\dot{Q}_{in}} \tag{17}$$

According to the second law of thermodynamics, the evaluation indicator of energy is not only the quantity but also the quality. Waste heat energy is a kind of low-grade energy compared to effective mechanical energy since it requires various kinds of thermodynamic cycles to be converted into effective work. In accordance with Carnot principle, the maximum efficiency of thermodynamic cycles cannot be higher than Carnot cycle efficiency. Thus, only part of waste heat energy can be recovered. Therefore, exergy analysis is more useful to evaluate the total heat energy characteristics. Exergy is also called available energy and is used to evaluate the energy quality.

2. 2. Determination of Mean Gas Temperature or Combustion Temperature

Mean gas temperature is the average maximum combustion temperature at a particular load and compression ratio. The combustion temperature T_H is calculated from the measured exhaust temperature. The engine real cycle is approximated to be an ideal diesel cycle. The T-S and P-V diagram for ideal diesel cycle is shown in Figures 3 and 4, respectively. T_3 represents the combustion temperature T_H and T_4 represents the exhaust gas temperature.

Due to the isentropic expansion of the gas (considered perfect gas) between states 3 and 4, the relationship between T_3 and T_4 is established as:

$$T_3 = T_4\left(\frac{r_c}{r_{of}}\right)^{\gamma-1} \tag{18}$$

γ is the specific heat ratio (C_p/C_v) and its value is 1.4, since the gas considered is perfect. r_c is the engine compression ratio, which is the ratio of the maximum volume (V_1) to the minimum volume (V_2) formed in the engine cylinder.

$$r_c = \frac{V_1}{V_2} \tag{19}$$

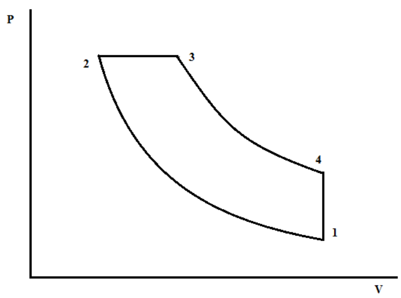

Figure 3. P-V plot of an ideal diesel cycle

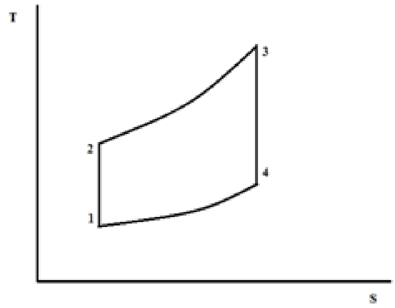

Figure 4. T-S plot of an ideal diesel cycle

r_{of} is the cutoff ratio of the engine given by:

$$r_{of} = \frac{V_3}{V_2} \tag{20}$$

where, V_3 and V_2 are the engine cylinder volumes after and before combustion, respectively.
The cutoff ratio varies with respect to brake thermal efficiency (η_e) as follows:

$$\eta_e = 1 + \frac{1-r_{of}^{\gamma}}{r_c^{\gamma-1}\,(r_{of}-1)} \tag{21}$$

As the values of T_4 and η_e are determined experimentally and the values of r_c and γ are known, r_{of} and T_3 (T_H, combustion chamber temperature) can be calculated using Equations (18) and (21).

3. EXPERIMENT

3. 1. Experimental Setup

The experimental set up is shown in Figure 5. The setup used during the course of experimental investigations was a single cylinder, four stroke, 3.5 kW variable compression ratio (VCR) diesel engine test rig connected to eddy current dynamometer. Software package “Enginesoft LV” was employed for online performance and combustion analysis. Piezo sensor and crank angle sensor which measures the combustion pressure and the corresponding crank angle, respectively were mounted onto the engine head. The output shaft of the eddy current dynamometer was fixed to a strain gauge type load cell for measuring applied load to the engine. ‘K’ type thermocouples were used to measure gas temperature at the engine exhaust, calorimeter exhaust, water inlet of calorimeter, water outlet of calorimeter, engine cooling water outlet and ambient temperature. The fuel flow was measured by the use of 50 ml burette and stopwatch with level sensors.

A computerized data acquisition system was used to collect, store and analyse the data during the experiment by using various sensors. Details of engine specification are given in Table 1.

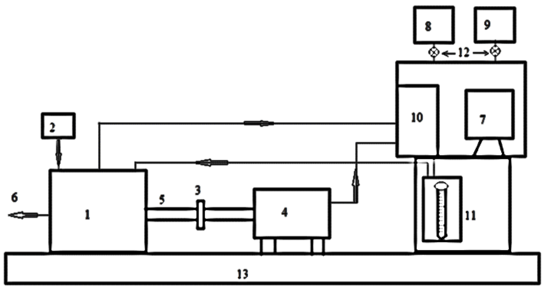

Figure 5. Actual engine setup

1. VCR Engine 8. Diesel Tank
2. Air filter 9. Biodiesel tank
3. Coupling 10. Data System
4. Dynamometer 11. Burette
5. Shaft 12. Valves
6. Exhaust 13. Bed
7. Computer

TABLE 1. Technical specification of engine.

Power	3.5kW
Speed type	constant
Engine	Make kirloskar
No of cylinder	single
No.of strokes	four
Max speed	1500rpm
Bore	87.5mm
Stroke	110mm
Compression ratio	Variable(12-18)
Swept vol	661.45cc
Cooling type	Water cooled

TABLE 2. Fuel properties of diesel and biodiesel blends.

Properties	**Diesel**	**B20**	**B40**	**B60**
Calorific value(kJ/kg)	42000	40935	39900	38871
Density(Kg/m^3)	830	848	860	868
Viscosity at 40^{O}C(cSt)	1.38	2.41	2.84	3.34

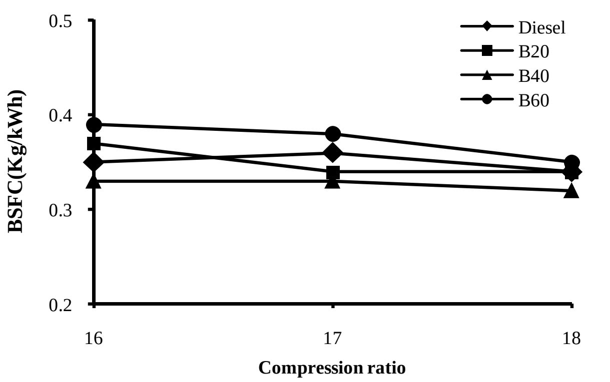

Figure 6. Variation of brake-specific fuel consumption with Compression ratio at full load.

3. 2. Experimental Procedure The VCR engine was started using standard diesel runs of 30 minutes. When the engine warmed up, the readings were taken. The tests were conducted at the rated speed of 1500 rpm. Fuel consumption was measured with the help of a measuring burette attached to the data acquisition system. Firstly, the baseline data was generated by conducting the tests with diesel fuel. Then, the tests were conducted using blends of neat Karanja methyl ester and diesel (B20, B40, B60) under compression ratios (CR) of 16:1, 17:1 and 18:1 at 100% (12 kg) load. The same engine operating parameters of performance and combustion were measured for different loads at a compression ratio of 18 using B20, B40 and B60 blends. The recordings were stored in a personal computer for further processing of results. The properties of the diesel fuel and the blends of Karanja oil methyl ester with diesel are summarized in Table 2.

3. 3. Preparation of Karanja Oil Methyl Ester For preparation of Karanja methyl ester from the neat Karanja oil firstly the acid value of the neat Karanja oil is measured using titration technique. The acid value of the oil is found to be 6.3 mg KOH/g. The acidity of oil is reduced by the esterification method wherein the acid catalysed reaction of the oil with methanol is done. Samples are taken from the products of reaction at an interval of 30 minutes and the acid value is measured. The reaction is stopped when acid value of the oil comes down below 5 mg KOH/g. The products of reaction are then allowed to settle in a separating funnel. After 24 hours, the triglyceride comes to the bottom of the funnel and untreated alcohol floats at the top. The triglyceride of Karanja oil is then collected from the bottom of the separating funnel. The triglyceride of the oil is then transesterified to produce Karanja oil methyl ester (KOME) or biodiesel of Karanja oil on the same reactor. Then, reaction is carried out at a temperature of 60°C with continuous stirring. The products of the reaction is kept in the separating funnel and allowed to settle down. Glycerol settles at the bottom of the separating funnel and is collected. The top part of the funnel contains Karanja oil methyl ester. The Karanja oil methyl ester is then water washed twice to remove the untreated methanol. The dissolved water is removed by heating the collected Karanja oil methyl ester. Finally, the Karanja oil methyl ester is blended with diesel fuel at ratios of 20%, 40% and 60% on volume basis and is referred to as B20, B40 and B60, respectively.

4. RESULTS AND DISCUSSION

4. 1. Brake Specific Fuel Consumption Figure 6 shows the variation of brake specific fuel consumption (BSFC) with change in compression ratio for diesel and

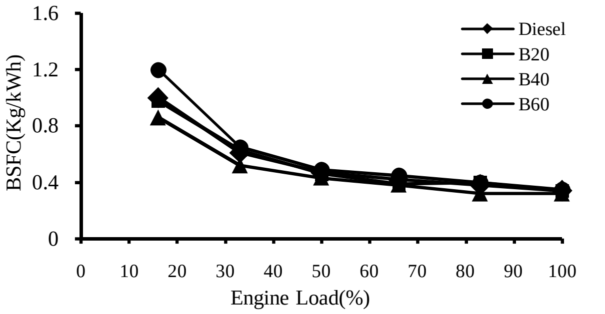

Figure 7. Variation of brake-specific fuel consumption with engine load at Compression ratio 18:1

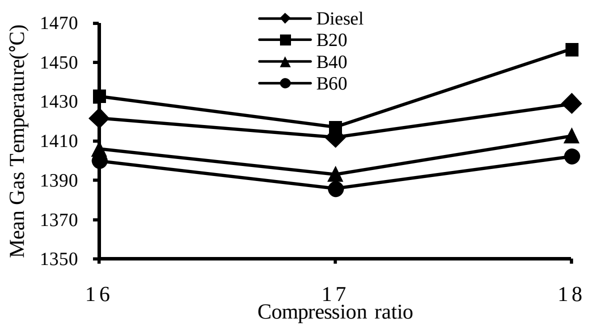

Figure 8. Variation of Mean gas temperature with Compression ratio at full load

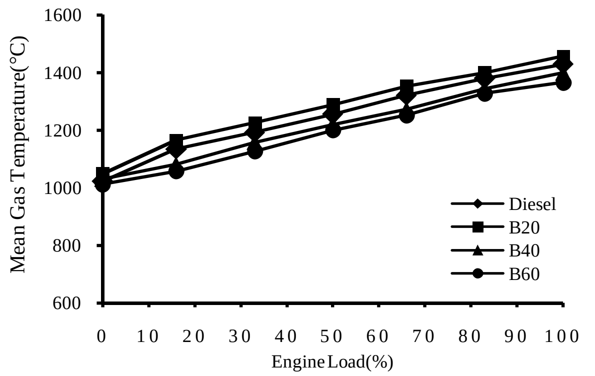

Figure 9. Variation of Mean gas temperature with Engine load at Compression ratio (18:1)

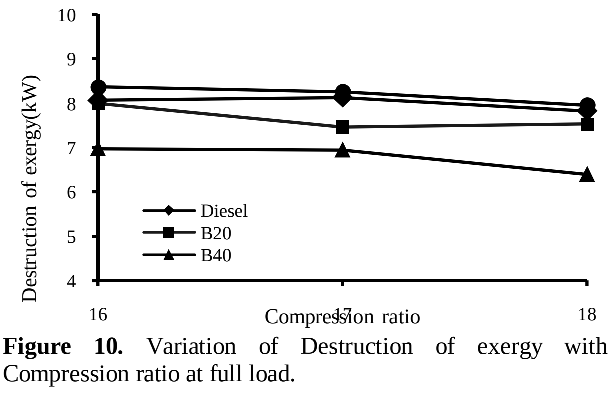

Figure 10. Variation of Destruction of exergy with Compression ratio at full load.

different blends of Karanja oil methyl ester. Brake specific fuel consumption decreases with increase in compression ratio at full load due to increased air density at higher compression ratios that leads to better atomization. BSFC for B20 and B40 is observed to be lower than diesel due to better combustion of oxygenated fuel.

The variation of BSFC at different loads is shown in Figure 7. It seen that the BSFC decreases with increase in load. It may be due to decrease in pumping losses with increasing load. Brake specific fuel consumption is found higher for B60 compared to B40 at full load. The BSFC for B60 and B40 are found to be 0.35kg/kWh and 0.32kg/kWh, respectively.

4. 2. Mean Gas Temperature (MGT)

Figure 8 shows the variation of mean gas temperature of combustion chamber at different compression ratios. The MGT at a CR of 18 is found to be higher in comparison to the MGT at a CR of 16 for all tested fuels. At a compression ratio of 18, the mean gas temperature is found to be higher as compared to mean gas temperature at a compression ratio of 16. This is because of higher compression ratios in which there is better atomization of fuel that leads to improved fuel air mixing. The MGT is found to be 1456.7°C, 1429°C, 1412.72°C and 1402.2°C for B20, diesel, B40 and B60, respectively.

The variation of mean gas temperature with load is shown in Figure 9. Mean gas temperature increases with an increase in load at a compression ratio of 18:1 for diesel and the blends of Karanja methyl ester (B20, B40 and B60). This is due to the fact that with increase in load the heat release rate increases.

4. 3. Destruction of Exergy

Figure 10 shows the variation of destruction of exergy for different compression ratio at full load and it is observed that destruction of exergy decreases with increase in compression ratio. Destruction of exergy is due to losses in energy due to energy conversion reaction and heat transfer. Destruction in exergy is found to be more for B60 as compared to diesel, B20 and B40 because of its higher viscosity which leads to incomplete combustion. The destruction of exergy is caused by irreversibilities due to combustion of fuel, frictional losses and heat transfer losses. Destruction in exergy is found to be 7.96 kW, 7.83 kW, 7.54 kW, 6.40 kW for B60, diesel, B20, B40, respectively.

Figure 11 shows the variation of destruction of exergy for various loads. It is found that destruction of exergy increases with increase in load for all fuels tested due to increase in fuel richness and ignition delay causing more energy and heat transfer loss.

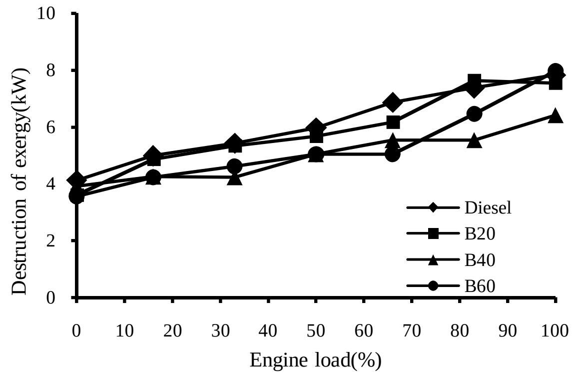

Figure 11. Variation of Destruction of exergy with Engine load at Compression ratio (18:1)

4. 4. Exergy Efficiency Figure 12 shows the variation of exergy efficiency with compression ratio for B20, B40, B60 and diesel. Exergy efficiency increases with an increase in compression ratio. This is caused because with increasing in compression ratio, the effective work increases due to the increase in air density and air temperature which leads to better atomization and combustion of fuel. Exergy efficiency is measured to be highest for B40 as compared to other fuels. The reason may be due to the presence of inherent oxygen which leads to better combustion of the B40 blend. Exergy efficiency for B40, B20, diesel and B60 are found to be 34.82%, 31.46%, 30.33%, and 30.05%, respectively.

The variation of exergy efficiency with engine load is shown in Figure 13. It is observed that the exergy efficiency increases with an increase in load due to increase in the power output and decrease in the BSFC at higher loads.

4. 5. Exhaust Gas Temperature Figure 14 shows the variation of exhaust gas temperature with compression ratio for all tested fuels. It can be observed that exhaust gas temperature decreases with an increase in compression ratio.

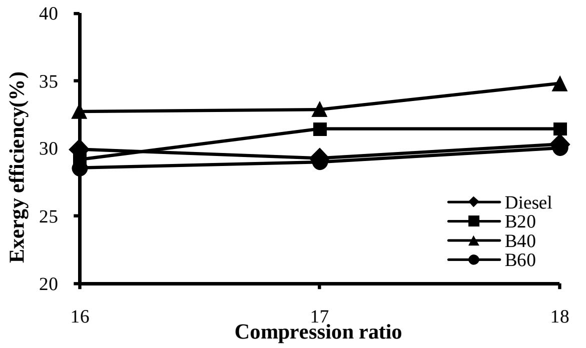

Figure 12. Variation of Exergy efficiency with Compression ratio at full load

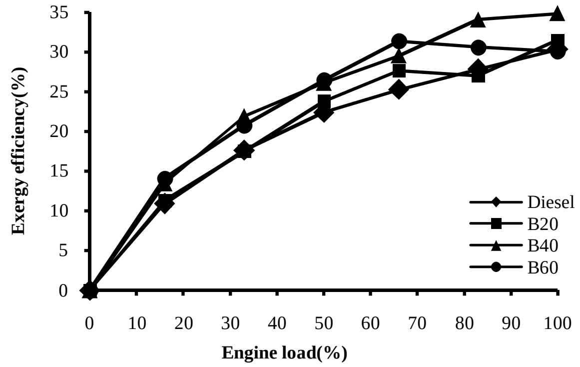

Figure 13. Variation of Exergy efficiency with Engine load at Compression ratio (18:1)

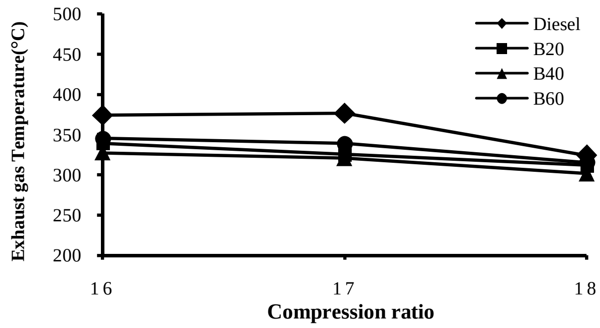

Figure 14. Variation of exhaust gas temperature with compression ratio at full load

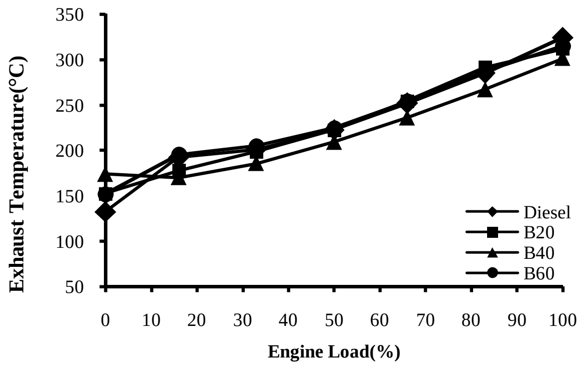

Figure 15. Variation of exhaust gas temperature with engine load at Compression ratio (18:1)

This phenomenon can be explained as follows, at a higher compression ratios, a larger part of fuel energy is converted to effective work. Exhaust gas temperature at a compression ratio of 18 for diesel, B60, B20 and B40 are 324.53, 315.55°, 312.12 and 301.63°C, respectively. Variation of exhaust temperature with engine load is shown in Figure 15. It is observed that at a higher compression ratio of 18, the exhaust temperature increases with increase in engine load. It is due to the fact that at higher loads there is increase in fuel richness and slightly inferior combustion. The maximum exhaust temperature is observed for B60 at full load due to the presence of inherent oxygen which leads to later stage combustion.

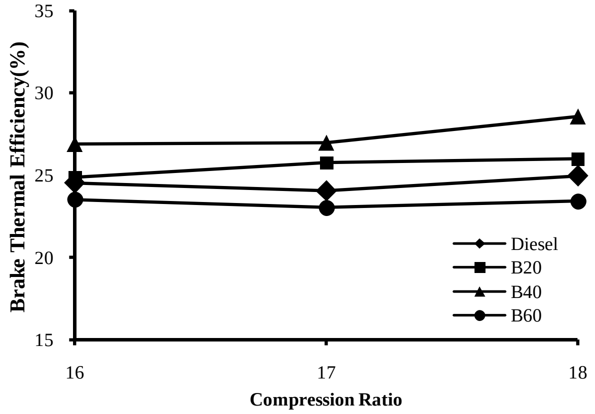

Figure 16. Variation of brake thermal efficiency with compression ratio at full load

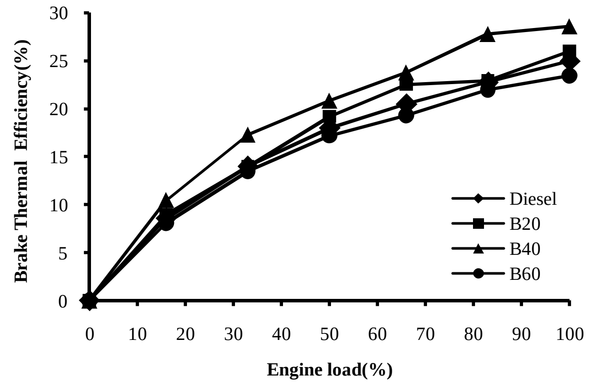

Figure 17. Variation of thermal efficiency with engine load at Compression ratio (18:1)

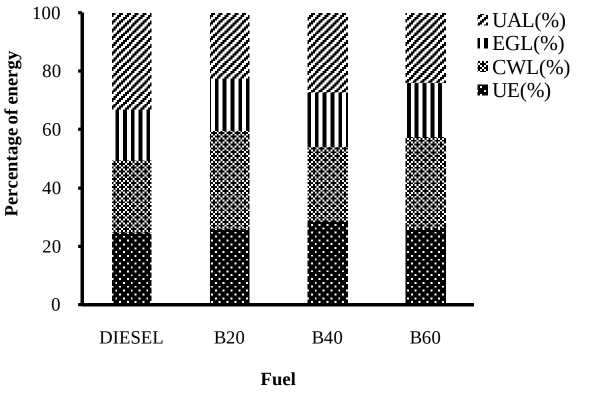

Figure 18. Percentage of energy Vs fuel at compression ratio (18:1)

4. 6. Brake Thermal Efficiency (BTE)

Figure 16 shows the variation of brake thermal efficiency with respect to compression ratio. It is observed that BTE is more at a compression ratio of 18 as compared to 16. This can be explained by observing that as the compression ratio increases fuel consumption decreases. It may be due to better mixing and spray characteristics of fuel, which leads to increased efficiency. The result shows that brake thermal efficiency for B40, B20, Diesel and B60 are 28.57%, 26%, 24.98% and 24.44%, respectively.

Figure 17 shows the brake thermal efficiency of the engine at different engine loads. Brake thermal efficiency increases with an increase in engine load due to a reduction in ignition delay, heat loss and increase in power output. The maximum brake thermal efficiency is found to be 28.57% for B40 at full load.

4. 7. Energy Balance Analysis

Figure 18 shows the percentage of energy with respect to different fuel at compression ratio 18 and 100% load. Total energy includes useful energy (UE), exhaust gas losses (EGL), cooling water losses (CWL) and unaccounted losses (UAL). The unaccounted loss is calculated by subtracting useful energy or brake power, exhaust energy losses and cooling water losses from input energy.

Exhaust gas loss increases as the blend % increases due to incomplete combustion, poor atomization. Cooling water loss also increases for higher blends due to more energy losses in combustion chamber. Result shows that cooling water loss and exhaust gas loss for B60 is 31.13% and 18.5%, respectively which is 26.7% and 5.1% higher compared to diesel.

5. CONCLUSION

From this study it was found that the BSFC decreases with increase in compression ratio and load for all blends of fuel. Mean gas temperature increases with an increase in compression ratio and load and was found to be maximum for B20. The destruction of exergy decreases with increase in compression ratio and increases with an increase in load for all blends of fuel. The destruction of exergy was measured to have a higher value for B60 at full load as compared to diesel, B20 and B40. The exergy efficiency was observed to increase with an increase in compression ratio and load for all blends of fuels tested. Exhaust temperature decreases with an increase in compression ratio and increases with an increase in load for all blends of fuel tested. Exergy efficiency was found to be the highest for B40 as compared to B20 at full load. Brake thermal efficiency increases with an increase in compression ratio and load for all blends of fuel.

6. REFERENCES

1. Barnwal, B. and Sharma, M., "Prospects of biodiesel production from vegetable oils in india", ***Renewable and Sustainable Energy Reviews***, Vol. 9, No. 4, (2005), 363-378.
2. Karekezi, S., "Poverty and energy in africa—a brief review", ***Energy Policy***, Vol. 30, No. 11, (2002), 915-919.
3. Nwafor, O. and Rice, G., "Performance of rapeseed oil blends in a diesel engine", ***Applied Energy***, Vol. 54, No. 4, (1996), 345-354.
4. Agarwal, A.K., "Biofuels (alcohols and biodiesel) applications as fuels for internal combustion engines", ***Progress in Energy and Combustion Science***, Vol. 33, No. 3, (2007), 233-271.
5. Rakopoulos, C. and Giakoumis, E., "Second-law analyses applied to internal combustion engines operation", ***Progress in Energy and Combustion Science***, Vol. 32, No. 1, (2006), 2-47.
6. Szargut, J., Morris, D.R. and Steward, F.R., "Exergy analysis of thermal, chemical, and metallurgical processes", (1987).
7. Adrian, B., Advanced engineering thermodynamics., John Wiley & Sons, Inc. (1988)
8. Caton, J.A., "On the destruction of availability (exergy) due to combustion processes—with specific application to internal-combustion engines", ***Energy***, Vol. 25, No. 11, (2000), 1097-1117.
9. Caton, J.A., Operating characteristics of a spark-ignition engine using the second law of thermodynamics: Effects of speed and load., SAE Technical Paper.(2000)
10. Michael, P. and Anthony, M., "Engine testing theory and practice", ***SAE International***, (1999).
11. Azoumah, Y., Blin, J. and Daho, T., "Exergy efficiency applied for the performance optimization of a direct injection compression ignition (CI) engine using biofuels", ***Renewable Energy***, Vol. 34, No. 6, (2009), 1494-1500.
12. Rath, M., Acharya, S. and Roy, S., "Thermodynamic analysis of compression ignition engine using neat karanja oil under varying compression ratio", ***International Journal of Ambient Energy***, (2014), 1-9. .

Exergy and Energy Analysis of Diesel Engine using Karanja Methyl Ester under Varying Compression Ratio

M. K. Rath, S. K. Acharya, P. P. Patnnaik, S. Roy

Department of Mechanical Engineering, ITER, SOA University, Bhubaneswar-751030, India

PAPER INFO

Paper history:
Received 26 October 2013
Received in revised form 01 April 2014
Accepted in 17 April 2014

Keywords:
Karanja methyl ester
Exergy
Biofuels
Variable compression ratio
Diesel engine
Energy

چکیده

در مطالعه حاضر، نیاز به کاهش مصرف سوخت رایج و سوخت های فسیلی وابسته به آن است. بنابراین، برای بهبود استفاده از منابع تجدیدپذیر از قبیل سوخت های زیستی نیاز به تخمین موثر از عملکرد موتورها بر پایه قانون دوم ترمودینامیک می باشد. در این مقاله، تحلیل انرژی، اکسرژی، دمای متوسط گاز و دمای گاز خروجی موتور احتراق تراکمی (CI) با استفاده از مخلوط متیل استر کارانجا و دیزل انجام شده است. نتایج در نسبت های مختلف تحت بارگذاری کامل و در بارگذاری های مختلف موتور در نسبت 18:1 محاسبه شده است. مشاهده شد که بازده اکسرژی، دمای گاز متوسط، بازده ترمز حرارتی با افزایش در نسبت تراکم همانند بارگذاری افزایش می یابد. دمای گاز خروجی و تخریب اکسرژی برای همه مخلوط های سوخت با افزایش در نسبت تراکم کاهش و با افزایش در بارگذاری افزایش می یابد.

doi: *10.5829/idosi.ije.2014.27.08b.12*