EXERGY ANALYSIS IN DIESEL ENGINE WITH BINARY BLENDS

by

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Investigation on Diesel engine with minimized fuel consumption rate and increased output power is not the meaningful procedure if irreversibility in the thermodynamic system is ignored. This current procedure is aimed to signify the importance of exergy analysis in Diesel engine performance on the perspective of Second law of thermodynamics analysis. In this study, diesel-cotton seed oil blends were tested on engine running with direct fuel injection mode of operation. The experiments were conducted with diesel (D), 5% cotton seed oil-95% diesel (CB5), 10% cotton seed oil-90% diesel (CB10), and 15% cotton seed oil-85% diesel (CB15) for estimation of brake power, energy rate, and exergy rate in the fuel and exhaust, heat release rate, exergy destruction, ideal efficiency (I law), and actual (II law) efficiency. The results outcome that an increase in trend was observed in the fuel exergy and thermal exergy loss with engine speed for D, CB5, CB10, and CB15. The loss of exergy, heat release rate, percentage of exergy and exergy transferred through exhaust gases decreased for CB5, CB10, and CB15 compared to diesel.

Key words*: exergy, energy, diesel, cotton seed, exergy destruction*

Introduction

The superior characteristics of Diesel engines in the aspect of improved fuel efficiency have led to significant investigation on this engine than the gasoline engine. Running the Diesel engine with biodiesel-diesel combinations minimize the fuel cost to reasonable extend [1]. The optimum performances of any thermodynamic system rely on the execution of exergy analysis. Though most of the investigations on internal combustion (IC) engine focus on biodiesel blends for optimum performance with reduced emissions, the key factor in all such cases are how to enhance the efficiency of fuel. One of the promising tasks in Diesel engine is that whether the performance and emission studies are experimented based on energy or exergy analysis. Though the biodiesel can be an alternative energy source which is being combined with diesel for investigation, most research on biodiesel blends outcome that research conceded so far was on the basis of energy point of view only and not on the exergy point of view. The major portion of fuel exergy after combustion is destroyed on account of many reasons which have to be considered for effective analysis. Experimental analysis was performed for the CI engine both by quantitative and qualitative methods based on thermodynamic First and Second law, respectively [2]. The combustion, heat transfer and exhaust gases were closely related to losses in exergy [3]. The exergy analysis on palm and cotton oil-diesel blends on the performance of Diesel engine was illustrated with the potential importance of exergy study [4]. Operation of the

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Diesel engine at 3/4 of the peak load, 1900 rpm, and 2:3 combination of biodiesel from waste cooking oil and neat diesel resulted in least thermal exergy loss through the engine system and optimum fuel consumption efficiency [5]. The experimental study on palm oil and cotton seed oil biodiesel blends with diesel resulted in the estimation of ultimate power supplied by such engines could be achieved by perspective emission and exergy analysis [6]. The particle swam optimization could be a measure of exergoeconomic performance of steam power stations [7]. The exergetic efficiency was in direct relation with the cetane number and delay period and the exergy destruction decreased with decrease in engine speed [8]. The 15% castor oil biodiesel in diesel resulted in optimum engine performance with improved thermal and exergetic efficiency [9]. The internal losses in any thermodynamic systems could not be evaluated by First law of thermodynamics [10, 11]. The energy loss in thermal power equipments could not be evaluated based on only the First law of thermodynamics [12]. The experimental investigation on the CI engine both by quantitative and qualitative methods based on thermodynamic First and Second law, respectively indicated that the exergy is an important measure of engine performance [13]. The energetic and exergetic efficiency were found to be higher in simarouba bio diesel blends than in diesel [14]. The qualitative analysis of thermodynamic system in engines operated with diesel rather than quantitative computations was utilized to examine the possibilities for irreversibility in the different engine parts. This showed that one half of the chemical exergy possessed by the fuel lost through the system boundary due to unaccounted elements and nearly 15% of input exergy of the fuel was lost in maintaining the uniform temperature distribution in the engine components or through the emissions. The implementation of analysis of thermodynamics on engine with the concept of active modular method improved the targeted outcome and reduced CO_2 emission in gasoline engine [15]. The irreversibility in combustion produced 41%-42% destruction in input exergy when the start of injection was 300-340 CA° [16]. The optimum exergetic efficiency was resulted at 1800 rpm with fuels such as canola, hazelnut, and pure diesel [17]. The exergetic efficiency of 91.7% was yielded during transesterification of waste cooking canola oil biodiesel [18]. The multi generation systems with biogas yielded 46.53% increase in exergetic efficiency compared to the simple systems. The conclusion of previous studies on energy and exergy analysis in Diesel engines indicated that exergy analysis has an inconceivable effect on improvement in overall thermodynamic system analysis of the engine. Based on the author's knowledge, this present research is focused on the energy and exergy analysis of binary blends in Diesel engine as a means of emphasizing the earlier research outcomes based on Second law of thermodynamics and also to perform Second law analysis with cotton seed oil biodiesel in Diesel engine as an alternative source of energy.

Figure 1. Flow diagram of test engine set-up

Experimental set-up

The experimental procedure consists of preparation of biodiesel followed by testing for energy and exergy in DI Diesel engine. Initially the transesterification is employed for the extraction of cotton seed oil biodiesel from the raw sample with methyl-alcohol and potassium hydroxide (KOH) catalyst [19, 20]. This is done by heating the oil to 60 ° C followed by mixing. Furthermore, the test experiment was conducted on a 3.5 kW naturally aspirated DI Engine. Figure 1 shows the flow diagram of test engine set-up and the hydraulic dynamometer was coupled to engine shaft to vary the load. For the measurement of the CO, CO_2, NO_x, and HC exhaust emissions, the AVL digas 444 five gas analyzer was incorporated. For safety considerations, effective intake pressure was maintained at 1.01325 bar.

In this experimental work, CB5, CB10, and CB15 with reference to diesel were tested. All types of fuels were prepared in Centre for Energy Research lab, Sathyabama Institute of Science and Technology. The properties of fuel samples are presented in tab 1.

Table1. Thermophysical properties of the tested fuels

Property	CB5	CB10	CB15	Diesel	ASTM method	Instrument
Specific gravity at 15 °C	0.87	0.88	0.9	0.83	ASTM D4052	Hydrometer
Color	1	1	1	1	ASTM D1500	Distillation unit
Flash point °C	112	118	125	63	ASTM D93	Pensky-Martens flash point apparatus
Pour point °C	8	8.2	8.4	–3	ASTM D97	Pour point apparatus
Kinematic viscosity at 40 °C in mm^2/s	31.34	29.56	27.42	2.98	ASTM D445	Say bolt viscometer
Cetane number	42	38	37	48	ASTM D976	Ignition quality tester
Calorific value in kJ/kg	39300	39150	38850	43000	ASTM D240	Calorimeter

American Petroleum Institute gravity (API) from specific gravity of fuel:

$$API = \left(\frac{141.5}{sg}\right) - 131.5 \tag{1}$$

where *sg* is the specific gravity.

It was observed that Cetane number of diesel is higher than cotton seed oil biodiesel and viscosity of diesel is less than biodiesel blends. The energy and exergy analysis were prepared for four-stroke Diesel engine fuelled with biodiesel blends.

The experiment was conducted under controlled room temperature and pressure. The different parameters measured with the degree of accuracy and instruments incorporated are represented in tab. 2. The test for energy and exergy analysis was carryout out with engine

Table 2. Instrumentation with uncertainty analysis

Quantity to be measured	Instrument		Uncertainty error
Load control and torque	Hydraulic dynamometer		±0.5%
Mass-flow rate of air to the engine	Air box meter		±0.3%
Mass-flow rate of fuel	Calibrated glass tube and stop watch		±0.5%
Temperature of exhaust, T_{in}, T_{out}	*K*-type thermocouples		±1%
Engine speed	Tachometer type movistrobe		±1%
Composition of exhaust gases [%]	AVL digas 444 five gas analyzer	CO_2	±0.06%
		CO	±0.04%
		NO	±0.46%
		HC	±0.45%

speed of 500 rpm, 1000 rpm, 1500 rpm, 2000 rpm, 2500 rpm, and 3000 rpm. To control uncertainty in the measurement parameters, the test was conducted completely for each fuel sample with sufficient time interval of 30 minutes. The test engine name plate details are given in tab. 3.

Table 3. Engine specification

Type of engine	Direct injection diesel engine
The number of stroke	four-stroke
Bore	80 mm
Stroke	70 mm
Displacement volume	230 cc
Compression ratio	20:1
Maximum speed	3500 rpm
Maximum power	3.5 kW

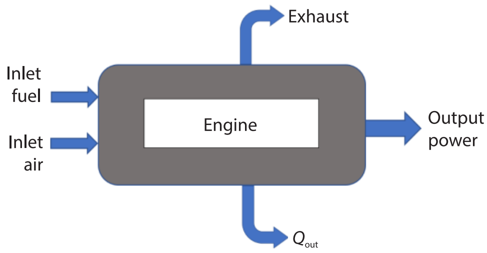

Figure 2. Flow diagram

Energy and exergy analysis

The following section illustrates the assumptions made for First law of thermodynamics based energy analysis and Second law of thermodynamics based exergy analysis. The flow diagram is represented in fig. 2.

Assumptions:
- All fuel type gas mixture is ideal gas.
- Kinetic energy and potential energy of fuel mixture and exhaust is negligible.
- Reference test conditions are $P_0 = 1$ atmosphere, $T_0 = 25$ °C.
- Combustion is complete.
- Engine is steady flow open system.

Mass balance for engine is given:

$$\sum \dot{m}_{\text{in}} = \sum \dot{m}_{\text{out}} \tag{2}$$

where $\dot{m}_{\text{in}}$ and $\dot{m}_{\text{out}}$ are the mass-flow rate at entry and exit of the engine (control volume), respectively.

Energy balance was estimated from the following steady-state, steady flow energy balance equation:

$$\sum \dot{m}_{\text{in}} h_{\text{in}} + Q = \sum \dot{m}_{\text{out}} h_{\text{out}} + W + Q_{\text{conv}} + Q_{\text{rad}} \tag{3}$$

where Q and W are the input heat energy and net output work, respectively, h_{in} and h_{out} – the specific enthalpy at inlet and exit, respectively, and $Q_{\text{conv},}$ and Q_{rad} – the heat loss due to convection and radiation, respectively.

The energy possessed by the fuel, E, before combustion based on the lower calorific value, C_{u}, is:
- Fuel energy

$$E = \dot{m}_{\text{f}} C_{\text{u}} \tag{4}$$

The energy loss after combustion is calculated:

– Energy loss

$$Q_{\text{loss}} = E_{\text{fuel}} - \left(W + Q_{\text{exh}}\right) \tag{5}$$

Energy available in the emission of burnt gases:

$$Q_{\text{ex}} = \sum \dot{m}_{\text{out}} h_{\text{ou}} = \dot{m}_{\text{CO}} \Delta h_{\text{CO}} + \dot{m}_{\text{CO}_2} \Delta h_{\text{CO}_2} + \dot{m}_{\text{NO}} \Delta h_{\text{NO}} + \dot{m}_{\text{NO}_2} \Delta h_{\text{NO}_2} \tag{6}$$

where h_{ou} is the enthalpy difference between the burnt gases and the surrounding atmosphere.

Irreversibility in any thermodynamic process is the reason of energy destruction resulted in the necessity of exergy analysis.

$$\sum \dot{m}_{\text{in}} \varepsilon_{\text{in}} = \sum \dot{m}_{\text{out}} \varepsilon_{\text{out}} + E\dot{x}_{\text{heat}} + E\dot{x}_{\text{work}} + E\dot{x}_{\text{dest}} \tag{7}$$

where ε_{in} is the specific exergy at inlet, ε_{out} – the specific exergy at outlet, $E\dot{x}_{\text{heat}}$ – the loss of exergy due to heat transfer rate, $E\dot{x}_{\text{work}}$ – the exergy of output power, and $E\dot{x}_{\text{dest}}$ – the thermal exergy lost(destructed) the system boundary.

The fuel exergy before combustion:

$$E\dot{x}_{\text{in}} = \dot{m}_{\text{fuel}} \varepsilon_{\text{fuel}} \tag{8}$$

The exergy of the fuel per unit mass $\varepsilon_{\text{fuel}}$ is the product of lesser calorific value of the fuel and the chemical exergy factor calculated:

$$\varepsilon_{\text{fuel}} = H_u^{\Phi} \tag{9}$$

The inlet exergy, Φ, of the fuel based on chemical energy:

$$\Phi = 1.0401 + 0.1728_{\text{hc}} + 0.0432_{\text{oc}} + 0.2169_{\text{ac}} (1.216901_{\text{hs}}) \tag{10}$$

where h, c, o, and a are the mass fraction of hydrogen, carbon, oxygen, and sulphur content of the fuel, respectively. While $E\dot{x}_{\text{work}}$ is the exergy rate produced at engine shaft as useful work output, which is calculated based on the rate of energetic work.

$$E\dot{x}_{\text{work}} = \dot{W} \tag{11}$$

The thermal exergy leaving to the environment, estimated:

$$E\dot{x}_{\text{heat}} = \sum \left(1 - \frac{T_0}{T_{\text{cw}}}\right) \dot{Q}_{\text{loss}} \tag{12}$$

where T_{cw} is the bulk mean of the outlet temperature, $T_{\text{cw,out}}$, and inlet temperature, $T_{\text{cw,in}}$, while T_0 is the reference temperature.

The exhaust exergy rate is calculated:

$$E\dot{x}_{\text{out}} = \sum \dot{m}_i \left(\varepsilon_{\text{tm}} + \varepsilon_{\text{chem}}\right) \tag{13}$$

where the exhaust specific exergy, ε_{tm}, for an open system based on thermal energy interaction is calculated:

$$\varepsilon_{\text{tm}} = \left(h - h_0\right) - T_0 \left(s - s_0\right) \tag{14}$$

where s is the total entropy per unit mass of the emitted gases, h – the total enthalpy per unit mass, T_0 – the environment (reference) temperature and the h_0, s_0 represents the total enthalpy per unit mass and total entropy per unit mass of the reference state, respectively. While εchem is the exhaust gases exergy based on the fuel chemical energy and can be estimated:

$$\varepsilon_{f\,\text{chem}} = \text{R}T_0 \ln \frac{x_\text{i}}{x_\text{r}} \tag{15}$$

where R is the constant of gas, x_i and x_r are the emitted gases and gases in the surrounding environment molar fractions. The dead state ambient consists of 0.034% CO_2, 0.0007% CO, 0.0002% SO_2, 3.030% H_2O, 0.00005% H_2, 20.35% O_2, 75.670% N_2, and 0.914% rest of the gases if any.

The exergetic efficiency calculated as output exergy to the exergy possessed by the fuel sample:

$$\eta_e = \frac{Ex_\text{work}}{Ex_\text{fuel}} \tag{16}$$

Result and discussion

The DI engine was run with speed ranges from 500-3000 rpm for the fuel samples such as D, CB5, CB10, and CB15. The loss of exergy, heat release rate, percentage of exergy and exergy transferred through exhaust gases were calculated for carrying out the energy analysis. First, the engine was operated with pure diesel.

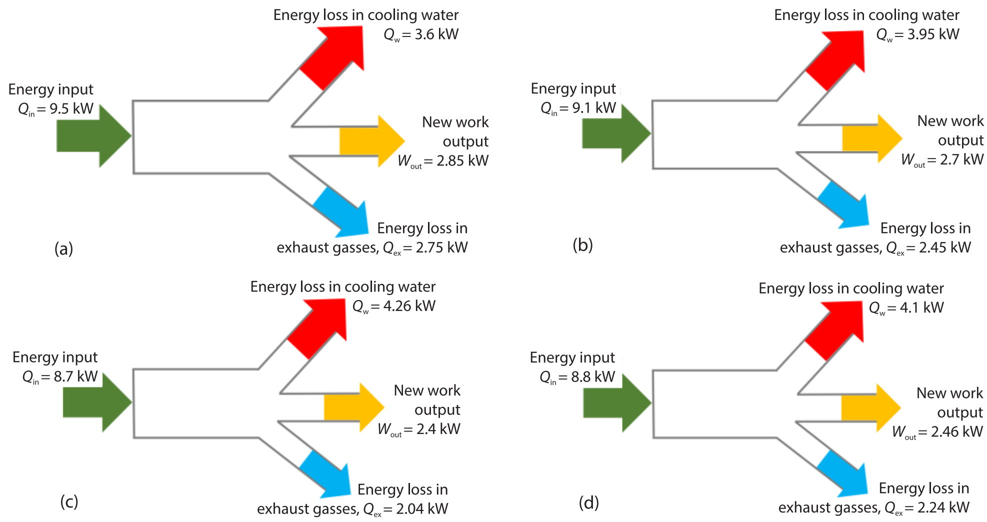

Figure 3. Energy balance of DI diesel engine at 2500 rpm based on First law of thermodynamics; (a) D, (b) CB5, (c) CB10, and (d) CB15

From fig. 3(a), it was evident that 29.67% useful work output was produced at 2500 rpm. The power available at engine shaft was 30% of peak load as depicted in fig. 3(a). The energy wasted with the emission of gases and engine cooling was 29% and 62.1%, respectively. While 29.67% input energy of fuel was transferred to useful power output, 43.41% of thermal energy was observed by the engine cooling system and 26.92% of input energy was lost in exhaust gases when running the engine with peak load for the fuel sample CB5 as shown in fig. 3(b). For CB10, 27.95%, 47.6%, and 25.45% are the useful power output, heat transfer through cooling water and exhaust gases, respectively as shown in fig. 3(c). Lowest power output of 27.5% was depicted for CB15 fuel sample as shown in fig. 3(d). The aforementioned trends were due to lower heating value of bio diesel blends compared to pure diesel. Exhaust temperature of gases, O_2 level in the fuel, calorific value and viscosity were the influencing factors for increase in energy loss through exhaust gases for CB5, CB10, and CB15 with reference to pure diesel. Due to the inferior Cetane number of cotton seed bio diesel blends there was increase in ignition delay period, resulted in reduction in thermal efficiency based on brake power as compared to pure diesel at a particular speed, say 2500 rpm as revealed in fig. 4.

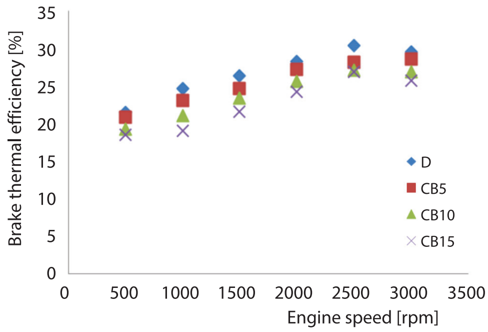

Figure 4. Influence of engine speed on brake thermal efficiency for the fuel samples

Fraction of the energy available at the engine was destructed to the thermodynamic losses due to heat loss in the process. Detailed exergy analyses need to be performed in order to account for the destruction losses. Exergy supplied at inlet, useful attainable power, exergy loss through the engine cooling system, exergy carried by the burnt gases after combustion and exergy destruction on account of the environmental conditions need to be subjected to a comprehensive analysis as given in fig. 5.

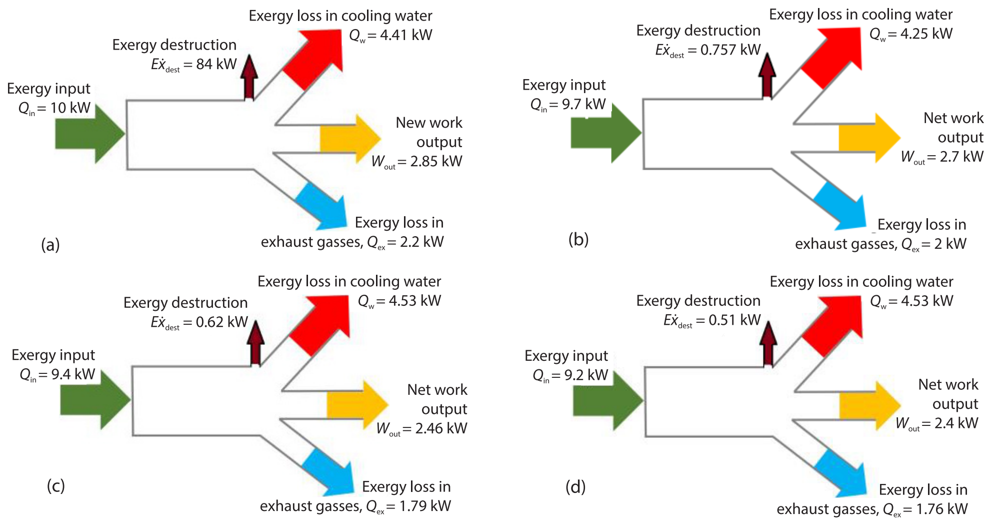

Figure 5. Exergy balance of DI diesel engine at 2500 rpm based on Second law of thermodynamics

When operating the engine with conventional diesel 28.5% of maximum attainable work was produced while 41.1%, 22% exergy lost through cooling water and exhaust gases, respectively, at maximum power input as shown in fig. 6. The volatility, cetane number, kinematic viscosity, and calorific value of biodiesel are the thermophysical properties for this loss in exergy. The bio diesel blends such as CB5, CB10, and CB15 yielded less useful work in comparison neat diesel due to lesser calorific value. The exergy work output, exergy loss through engine cooling, and exhaust gases were 27.85%, 43.82%, and 20.54% as shown in fig. 7, respectively when the engine is operated with CB5 at 2500 rpm.

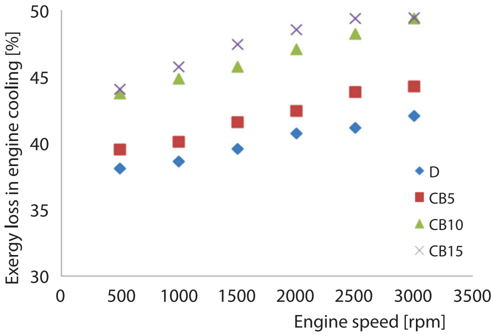

Figure 6. Influence of engine speed on exergy loss in engine cooling for the fuel samples

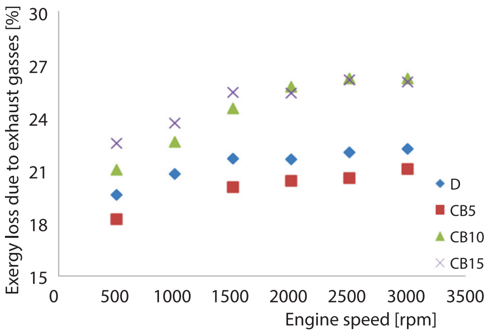

Figure 7. Influence of engine speed on exergy loss in exhaust gases for the fuel samples

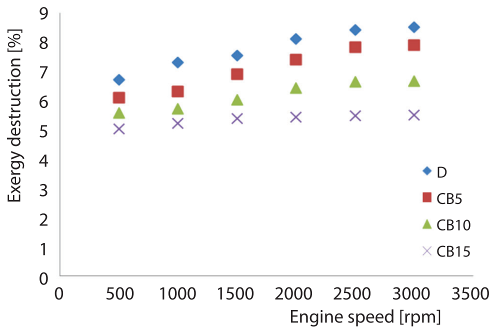

Figure 8. Influence of engine speed on exergy destruction the atmosphere for the fuel samples

The increase in cotton seed blend in diesel resulted in decrease in output exergy rate of the engine. It was inferred that increase in cotton seed blend resulted in decrease in exergy through exhaust emissions. The attained useful exergy in case of CB10 and CB15 is 26.17% and 26.09%, respectively. Similarly the exergetic heat loss through cooling water is 48.17% and 49.31% for CB10 and CB15 cotton seed blends, respectively. The increase in oxygen content of diesel-cotton seed oil bio diesel blends were the reason for less exergy destruction than with the conventional diesel.

The thermal exergy leaving the boundary of engine system was the most important portion of the total exergy losses and observed as 8.4%, 7.81%, 6.63%, and 5.49% for diesel, CB5, CB10, and CB15, respectively as depicted from fig. 8. This increase in trend of exergy destruction was due to increased O_2 content in the cotton seed blends. Since the exergy available at inlet depended on the chemical exergy of the fuel which in turn depended on the lower heating value of the fuel, it was found to be more for conventional diesel than the cotton seed blends. The fuel air mixing behavior, inertia produced in the combustion chamber and drop in pressure were the contributing factors for the thermal exergy destruction.

Conclusion

The effective utilization of available energy as potential method of research called exergization on the concert of direct injection diesel engine with fuel samples such as D, CB5, CB10, and CB15 were subjected to detailed experimental study. Both the First and Second laws of thermodynamics analysis were performed for calculation of brake thermal efficiency and exergetic efficiency, respectively. In addition, the useful output exergy, loss of exergy with the burnt gases, exergy heat utilized for engine cooling to maintain uniform temperature and the destruction of exergy findings were also the part of this experimental work. The experiments were conducted with engine speed range of 500-3000 rpm to account for the difference in fuel type. The fuel exergy was in direct proportion with engine speed for all the fuel samples. The exergy loss through exhaust was decreased for diesel-cotton seed blends than conventional diesel. Same trend was also depicted for thermal exergy destruction the environment. But the loss of exergy due to cooling of the engine was in increasing trend for cotton seed blends than conventional diesel. The 20% reduction in exergy loss through exhaust was arrived for CB15 compared to pure diesel. But the useful power output for CB15 was 8.46% lesser than diesel at an optimum engine speed of 2500 rpm. Addition of 5% cotton seed to the diesel consumed 6.58% increase in power for cooling the engine at the same speed than diesel. Form the findings, it was concluded that addition of biodiesel blends more than 5% was not the viable procedure of investigating engine performance. Though the results of literature were enormous with various biodiesel blends but the research should be on the view of qualitative manner than the quantitative procedure. These experimental findings could be a meaningful method of exergization of diesel engines with Second law analysis.

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