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Research Article

Impact of biodiesel blends on performance, emissions and waste heat recovery of diesel engine driven cogeneration system

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ABSTRACT

Due to the rapidly increasing energy demand, the world needs to focus more on identifying alternative energy sources like biofuels and energy conservation techniques that enhance the efficiency of various systems. A cogeneration (CHP) system is one of the most emerging techniques for achieving the goal of energy conservation by providing useful power (electricity) and heating simultaneously. So the current study proposes a diesel engine-driven CHP system that is fueled with different blends of biodiesel. The objective of the current study is to investigate the impact of Eureka Sativa oil biodiesel on waste heat recovery, performance, and emission characteristics of diesel engine driven combined heating and power generation system. The cogeneration unit is developed by connecting the exhaust pipe of a single-cylinder, four-stroke diesel engine with a heat exchanger. The pure diesel, along with 10%, 15%, 20%, and 25% by volume of biodiesel, was used as fuel for the cogeneration unit. The AVL Di-Gas 444N multi-gas analyzer was utilized to evaluate the engine exhaust gas emissions. Diesel fuel has the highest brake thermal efficiency and the lowest brake specific fuel consumption (BSFC). B20 has the highest brake thermal efficiency and the lowest BSFC among all blends of biodiesel. Also, B20 has better emission characteristics than all other blends of biodiesel. The exhaust gas temperature and waste heat recovery increase with the percentage of biodiesel in the blends. The B25 has the highest overall efficiency (38.49%) among all blends, which is 1.93 % lower than pure diesel. However, result analysis revealed that B20 is the best fuel among all biodiesel blends in terms of engine performance and emission formation. Whereas B25 is a better fuel in terms of WHR and overall cogeneration unit efficiency.

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INTRODUCTION

There is a critical need to find alternative and clean energy sources as well as energy conservation measures to improve the efficiency of various systems in order to ensure a sustainable future due to the rapidly expanding energy demand, limited energy supplies, and rising emissions. To increase the effectiveness of I.C engine, new control for combustion systems [1], thermal insulation [2], and WHR (waste heat recovery) [3] have been proposed. Since around 66% of total fuel energy goes with cooling water and exhaust gas, WHR technologies retain the ability to significantly reduce IC engine environmental impact while increasing fuel efficiency and engine performance. In a cogeneration system, a prime mover and a heat recovery system work together to provide the majority of the power and heating demands [4-6]. The main benefit of a combined heating and power (CHP) system is its potential to improve fuel utilization efficiency because less fuel is needed to develop the same amount of mechanical and thermal energy in a single cogeneration unit than it would to do so using conventional technologies [7]. Governments are promoting the utilization of CHP system not just in commercial sectors but also in residential ones due to the lower fuel consumption and emissions connected with cogeneration systems. Electric and thermal power may be supplied to homes, hospitals, workplaces, and hotels using micro-CHP systems (<50 kW) [8].

Because of their durability, greater efficiency at smaller power ranges, and lower cost compared to alternative prime movers like fuel cells, Stirling engines, and micro-turbines, internal combustion engines are appropriate for small-scale cogeneration applications. In an IC engine, heat can be recovered from engine cooling water, lubricating oil, and exhaust gas [9]. Additionally, by connecting the engine to a generator, electrical energy may be generated. Depending on the required output frequency, conventional CHP units operate at a set engine speed. Power output is controlled by altering the engine load to meet user demand. By regulating the speed of the ICE to its optimal efficiency, larger efficiency and smaller fuel consumption for part load operation might be obtained [10]. Fu et al. [11] emphasize the significance of conducting an exergetic and energetic study of the IC engine under aligning characteristics for the sake of determining the appropriate operating condition of the engine. They deduced that operating factors like engine speed and load have an impact on how different types of energy are distributed. Additionally, they came to the conclusion that recovering both exhaust gas and cooling water energy might greatly increase engine fuel efficiency. Agnese Magno et al. [12] investigate how a biodiesel blend affects the exhaust emissions and energy distribution of a small diesel engine. Using a common-rail engine with three cylinders and 1028 cc, the experiment was conducted. Pure diesel and a 20% by volume biodiesel blend were used in the tests (B20). It was discovered that the energy flows and the braking fuel conversion efficiency are not considerably impacted by the addition of 20% by volume biodiesel to diesel fuel. However, under typical operating situations, a biodiesel blend enables the reduction of combustion noise and pollutant emissions.

The need for cleaner fuels, including biodiesel, has been prompted by the emission reduction target [13–15]. In comparison to diesel fuel, the biodiesel molecule's oxygen concentration promotes better combustion, resulting in fewer release of carbon monoxide (CO), hydrocarbons (HC), and particulate matter (PM) [16,17]. On the other hand, using biodiesel as fuel typically results in higher NOx emissions, according to the literature [18,19]. Jena and Misra [20] investigated the impact of oxygenated fuels on the exergy and energy efficiency of a diesel engine from an energetic perspective. They came to the conclusion that additional study should be done on the energy potential of biodiesel. A thorough examination on energy balance at various test sites using alternative fuels was also recommended by Abedin et al. [21]. When compared to diesel fuel, biodiesel is more environmentally benign since it produces less HC (hydrocarbons), CO (carbon monoxide), and PM [22]. Particle emissions analysis has demonstrated that burning biodiesel produces smaller particles than burning diesel fuel [23]. Because NOx emissions depend on several parameters [24,25], a clear pattern was not discovered for them. The use of EGR (exhaust gas recirculation), in particular, has a significant impact on NOx emissions and can cause biodiesel combustion to produce NOx emissions at levels comparable to diesel fuel [25]. From an energy standpoint, Jena et al. [26] investigated how oxygenated fuels affected the energy and efficiency of a CI engine. Their findings suggested that adding more oxygen to the fuel improves combustion and decreases irreversibility. However, they came to the conclusion that more thorough research is required to fully understand the energy potential of biodiesel. A thorough analysis of energy balance at several test sites was also recommended by Abedin et al. [27] to determine the viability of alternative fuels.

A heterogeneous catalyst (CaO) was used by Aparna Singh et al. [28] to optimize the performance and emission characteristics of a CI engine powered by Jatropha biodiesel. Findings indicate that a diesel engine running on B20 performed almost as well as one running on diesel. Elkelawy et al. [29] investigated the performance, combustion, and emissions characteristics of a DI-diesel engine running on mixtures of algal biodiesel, diesel, and n-pentane. The performance and emissions of the engine will suffer as a result of the addition of biodiesel to the diesel. Pentane added to biodiesel blends generally enhances engine performance and emission characteristics. Medhat Elkelawy et al. [30] investigated the combustion and emission characteristics of a CI engine while introducing nanoparticles to diesel-biodiesel blends. The efficiency of the thermal brake is increased while the diesel engine is operating in the presence of nanofluid emulsions. Moreover, CO and HC emissions are significantly reduced by nanofluid emulsions. All nanofluid combustion results in rising NOx emissions. An experimental investigation on the performance, emission, and combustion behaviour of diesel, WCO biodiesel, and cyclohexane blends in a DI-CI engine was conducted by Medhat Elkelawy et al. [31]. The findings demonstrate that the addition of cyclohexane significantly enhances both engine performance and emissions. Using the life cycle analysis of biodiesel fuels made from soybean, Jatropha, Calophyllum inophyllum, and microalgae conducted a comparative study by Sudhakar Uppalapati et al. [32]. The findings suggest that GHG emissions were strictly controlled throughout the manufacturing of soybean biodiesel, but microalgae-based biodiesel had low net energy value, net renewable energy, and energy ratio throughout its life cycle. The flexibility, thermodynamics, and techno-economic analysis of a coal-fired CHP system were performed by Jiajia Li et al. [33]. In comparison to the base case, the author found that the maximum power production is improved by 6.35% above the rated power and that 27.26 tonnes of coal may be saved every cycle. In order to anticipate the performance and emission characteristics of a micro-tri-generation system powered by a CI engine and using diesel, Karanja oil, and Karanja biodiesel, an artificial neural network model was built by Kamal Kishor Khatri et al. [34]. Results indicate that Karanja biodiesel has favourable performance and emission characteristics in simulations and tests.

The above literature shows the feasibility of various biodiesel as alternative fuels. The literature discusses the emission characteristics and performance analysis of compression ignition engines running on different biodiesel blends. But very few studies are present in the literature that show the investigation of performance and emission analysis, along with waste heat recovery, of a combined heating and power generation system for different blends of Eureka Sativa oil biodiesel. So the current research proposes a diesel engine-driven CHP system that is fueled with different blends of Eureka Sativa oil biodiesel. And the main aim of the current study is to investigate the impact of Eureka Sativa oil biodiesel on waste heat recovery, performance, and emission characteristics of a combined heating and power generation system driven by a diesel engine. In the present study, a heat exchanger is coupled with the exhaust gas pipe of a single-cylinder, four-stroke, small (7 hp) diesel engine to develop a combined heating and power generation unit. 10%, 15%, 20%, and 25% by volume of Eureka Sativa biodiesel, along with pure diesel, are used as fuel for diesel engines to evaluate the performance of the CHP system. An AVL DiGas 444N multi-gas analyzer is used to measure the emission characteristics of the unit. The present research shows that B20 has the best engine performance and emission characteristics of any biodiesel blend. Although B25 is a better fuel in terms of WHR and overall cogeneration unit efficiency. Hence, in the course of this work, the optimal biodiesel blend relative to performance, emissions, and waste heat recovery will be determined among all evaluated blends, making this work novel research.

ENERGY ANALYSIS

In an IC engine, the combustion process produce heat energy due to burning of the fuel. A portion of total fuel energy is transformed into useful work, and the remaining portion is dissipated into the atmosphere through cooling water and exhaust gas in accordance with the first law of thermodynamics. Due to frictional losses and engine heat radiation, the remaining portion is lost. According to the second law of thermodynamics, about one-third of the fuel's energy has to be released into the atmosphere in the form of waste heat. This waste heat can also be used to enhance the thermal efficiency of the I.C. engine.

The energy balance of the engine and its surrounding can be expressed as follow [35]:

$$\dot{Q}_{Fuel} = BP + \dot{Q}_c + \dot{Q}_{ex} + \dot{Q}_{loss} \tag{1}$$

The total fuel energy ($\dot{Q}_{Fulel}$) is obtained with the help of fuel mass flow rate ($\dot{m}_f$) and calorific value of fuel (CV_f):

$$\dot{Q}_{Fuel} = \dot{m}_f \times CV_f \tag{2}$$

BP is the brake power it is calculated by using engine speed (N) and torque (T_b) [12]:

$$BP = 2.\pi.N.T_b \tag{3}$$

where N is expressed in rev/s and T_b is expressed in N-m.

The heat rate of heat lost in cooling water ($\dot{Q}_c$) is calculated with the help of following formula [12]

$$\dot{Q}_c = \dot{m}_{cw}\, C_{p_{cw}}(T_{cwo} - T_{cwi}) \tag{4}$$

Where $\dot{m}_{cw}$ is the cooling water mass flow rate; $C_{p_{cw}}$ the cooling water specific heat. T_{cwi} and T_{cwo} cooling water inlet and outlet temperature, respectively.

The available rate of heat in the exhaust gas of the I.C engine can be calculated with help of following equation [21]

$$\dot{Q}_{ex} = (\dot{m}_a + \dot{m}_f)\, C_{p_{eg}}(T_{eg} - T_0) \tag{5}$$

Where $\dot{m}_a$ and $\dot{m}_f$ are the mass flow rate of the intake air and fuel, $C_{p_{eg}}$ is the exhaust gas specific heat [21]. And T_0 is the environment temperature.

The rate of heat recovery from the exhaust gas is obtained with the help of following formula [12]

$$\dot{Q}_R = \dot{m}_w\, C_{p_w}\, (T_{wo} - T_{wi}) \tag{6}$$

Where $\dot{m}_w$ mass flow rate of water is enter into the heat recovery system, C_{p_w} is specific heat of water entering into heat recovery system. T_{wi} and T_{wo} are temperature of water at the inlet and outlet of the heat recovery system.

The brake thermal efficiency, (BTE) is calculated with help of following formula [12]

$$BTE = \frac{B.P}{\dot{m}_f \, CV_f} \tag{7}$$

The overall efficiency (η) of a cogeneration system is calculated as follow

$$\eta_{Overall} = \frac{B.P + \dot{Q}_R}{\dot{m}_f \, CV_f} \tag{8}$$

Measuring Devices

Airflow flow measurement

The mass flow rate of air is measured with the help of an air box. At the entrance of the air box, an orifice is attached, and the exit of the orifice is connected with the inlet manifold of the diesel engine with the help of a pipe, as shown in Figure 4. A U-tube manometer with a proper scale is also connected to the air box. During engine operation, pressure inside the air box decreases due to the suction stroke of the engine. The pressure difference was measured with the help of a manometer, and the mass of air per second sucked by the engine was obtained by the equation (9).

$$\text{Mass flow of air}(\dot{m}_a) = \rho_a C_d A_o \sqrt{\frac{2\, g\, h_w \rho_w}{\rho_a}} \tag{9}$$

Where ρ_a, ρ_w, A_o, h_w and C_d are density of air, density of water, cross sectional area of orifice , manometer reading and coefficient of discharge respectively.

Fuel flow measurement

The fuel flow rate was measured with the help of the burette method. In this method, a properly scaled glass burette, as shown in Figure 4, was connected to the fuel tank and engine via a tee valve. The fuel line was initially linked to both the engine and the burette so that diesel could be filled into the burette without interrupting the delivery of fuel to the engine. The tee valve was adjusted so that diesel began to flow from the burette to the engine in order to measure fuel consumption. The time taken to consume a certain amount of fuel is recorded using a stopwatch. The volume flow rate of fuel consumed can be calculated by dividing the time taken by the amount of fuel consumed.

Water flow measurement

The rotameters are used to measure the flow rate of cooling water in Figure 1. Rotameters are based on the variable area principle. In a narrowed measuring glass tube, the float is free to travel up and down. The float will take up a position where the buoyancy forces and weight are balanced as it moves upward. The instantaneous flow rate is measured by the vertical location of the float, as represented by the scale. To achieve the correct flow rate, the rotameter valves must be opened gradually and carefully. The measurement tube may be damaged if the float jumps abruptly, which must be avoided. A line on the scale is given for alignment, and it should correspond with the white line on the measuring tube.

Figure 1. Photo of rotameter.

Temperature measurement

The temperature of exhaust gas, cooling water, and water flow through the heat exchanger was measured with the help of a k-type thermocouple, as shown in Figure 2a. The working principle of a thermocouple is based on the Seeback Effect. To measure the temperature of any fluid, the probe of the thermocouple is fixed in the fluid whose temperature is to be measured, and the other end of the thermocouple is connected to the temperature indicator through a wire. The temperature indicator shows the temperature reading on a digital screen, as shown in Figure 2b.

Emission measurement

The emission characteristics of the diesel engine were measured by using AVL DiGas 444N exhaust gas analyzer. The AVL DiGas 444N exhaust gas analyzer is shown in Figure 3. The AVL DiGas 444N is a device that measures the relative quantities of various gaseous components like CO_2, NO_x, CO and HC in exhaust gases from diesel engines. In this device, a small diameter plastic tube is connected to an

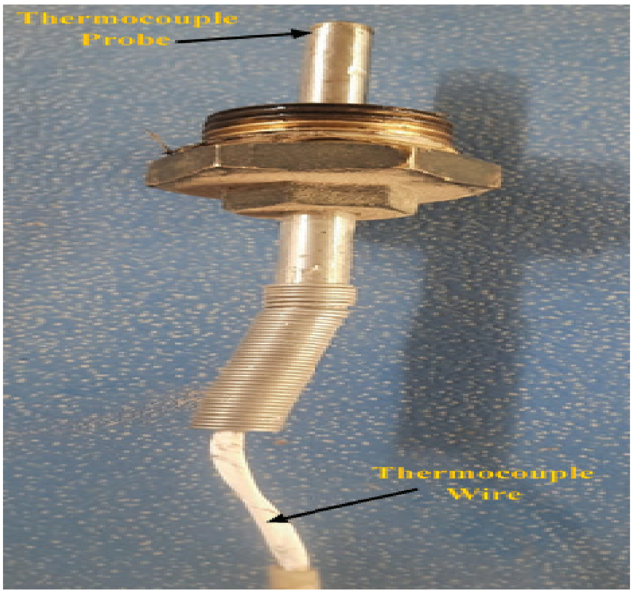

Figure 2a. Thermocouple.

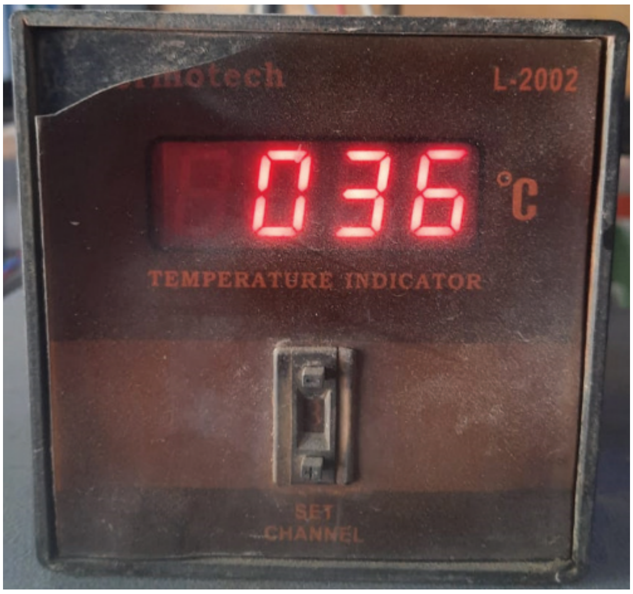

Figure 2b. Temperature indicator.

Figure 3. AVL DiGas 444N gas analyzer.

emission probe. And the CO_2, NO_x, CO, and HC emission levels were measured by inserting the emission probe of the gas analyzer into the engine exhaust pipe.

Experimental Setup

A water cooled single-cylinder, four-stroke diesel engine is used as the prime mover for the combined heating and power generation system. Table 1 illustrates the technical specifications of the engine used in this study. The engine output shaft is connected to the eddy current dynamometer (model: AG80; maker: SAG) for variation of engine load. The load sensor of the dynamometer gives a load signal to the digital indicator, and as a consequence, the load is shown digitally in kilograms. The fuel consumption is the amount of fuel consumed per unit time it is measured with help of fuel burette and stop watch. The cylinder and the cylinder liner are separated by a water jacket. A centrifugal pump is used to circulate water to cool the engine. The flow rate of the cooling water was measured using rotameters. All readings are taken at steady-state conditions. Throughout the test, the compression ratio and speed of the engine were maintained at 17.5:1 and 1500 rpm, respectively. The all performance measuring devices such as fuel burette, eddy current dynamometer, load controller and rotameter are shown in the Figure 4. To extract all data from the experimental setup for the engine performance study, the data acquisition system and IC engine software 9.0 was used.

The emission characteristics of the diesel engine were measured by using an exhaust gas analyser. The CO_2, NO_x, CO, and HC emission levels were measured by inserting the

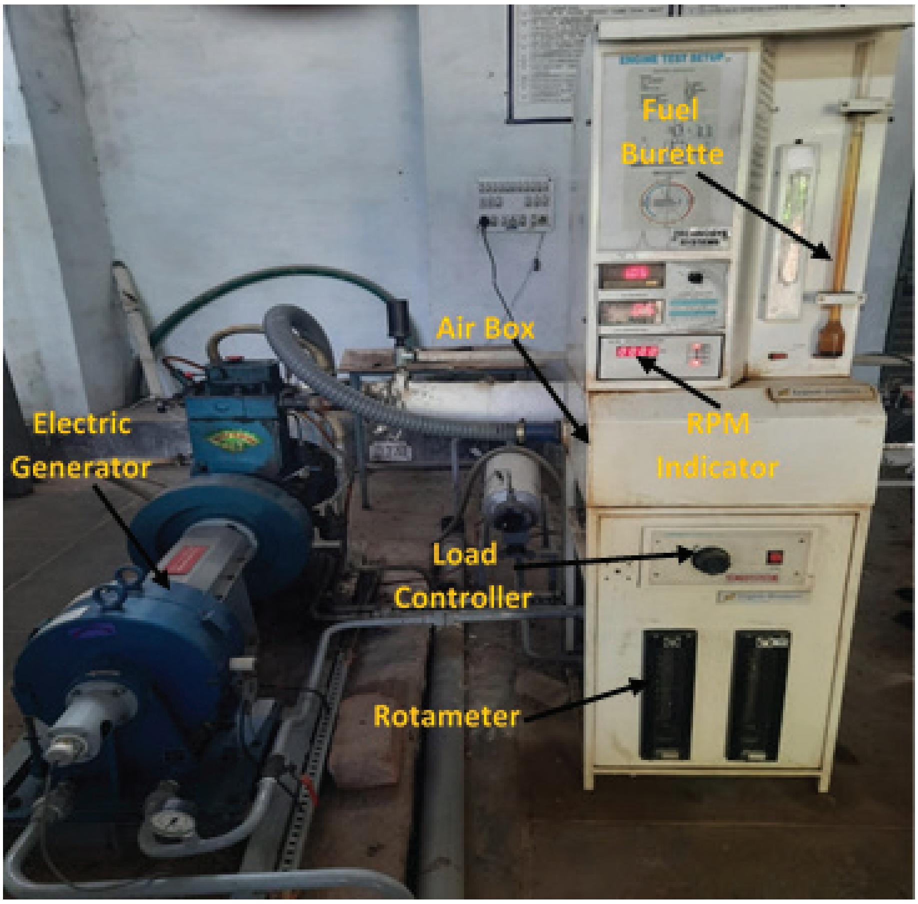

Figure 4. Diesel engine test rig with measuring device.

emission probe of the gas analyzer into the engine exhaust pipe. The AVL DiGas 444N exhaust gas analyzer is shown in Figure 3.

A double pipe heat exchanger was attached to the engine exhaust to extract the heat from the exhaust gas, and water was used as the heat transfer fluid for this heat exchanger. The double pipe heat exchanger was with help of two concentric pipe. The diameter of inner and outer pipes are 38 mm and 76 mm respectively as shown in line diagram Figure 5a .The setup is equipped with thermocouples at various points to measure the temperatures of the water and exhaust gases, as illustrated in Figure 5b.

The cogeneration (CHP) system was developed by connecting a diesel engine with a double pipe heat exchanger.

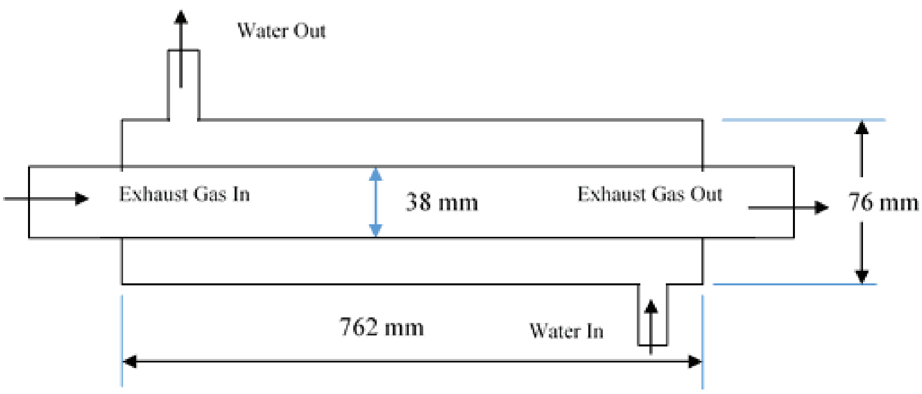

Figure 5a. Line diagram of heat exchanger.

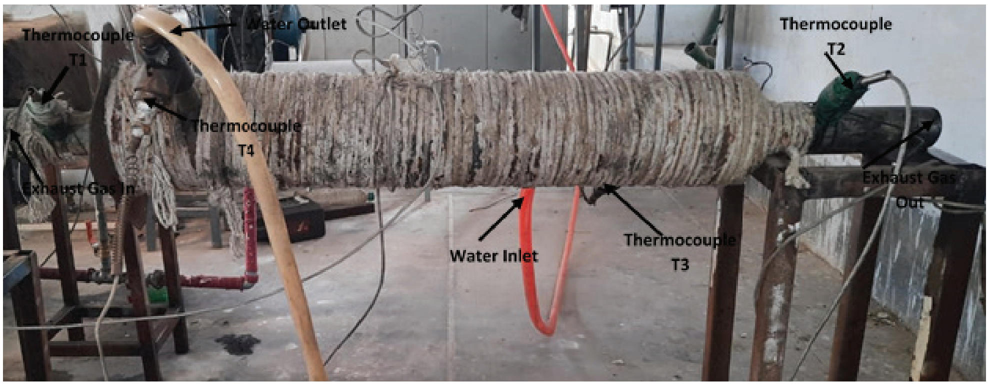

Figure 5b. Photograph of heat exchanger.

Table 1. Engine specification

Description	Specifications
Engine Type	Four-Stroke, Water-Cooled, Diesel engine
Maximum Torque	5.2 kW & 7 BHP
Compression Ratio	17.5/1
Combustion Volume	660 c.c
Cylinder bore x stoke	87.5 mm x 110mm
Injector Pressure	220 bar
Method of Loading	Eddy Current Dynamometer
Inlet Valve Open /Close	4.5 ^{0} BTDC/35.5 ^{0} ABDC
Fuel Delivery	23 deg. BTDC
Exhaust Valve Open/Close	35.5 ^{0} BBDC/4.5^{0}ATDC

The double pipe heat exchanger is integrated with the diesel engine with the help of pipes and valves, as shown in Figures 6a and 6b. As exhaust gas passes through the inner pipe of the heat exchanger, it will heat the water flowing through the annulus between the inner and outer pipes of the double pipe heat exchanger. The combined heating and power generation (cogeneration) mode operates by keeping valve V1 closed and valve V2 open, as indicated in Figure 6a.

Thermophysical Properties of Eureka Sativa (Taramira) Biodiesel

In this study, Eureka sativa (Taramira) seed oil was utilized, and it was purchased from a local supplier in Bikaner, Rajasthan, India. Methanol and KOH were also bought from Friends Chemical Agency in Bikaner, India, while the

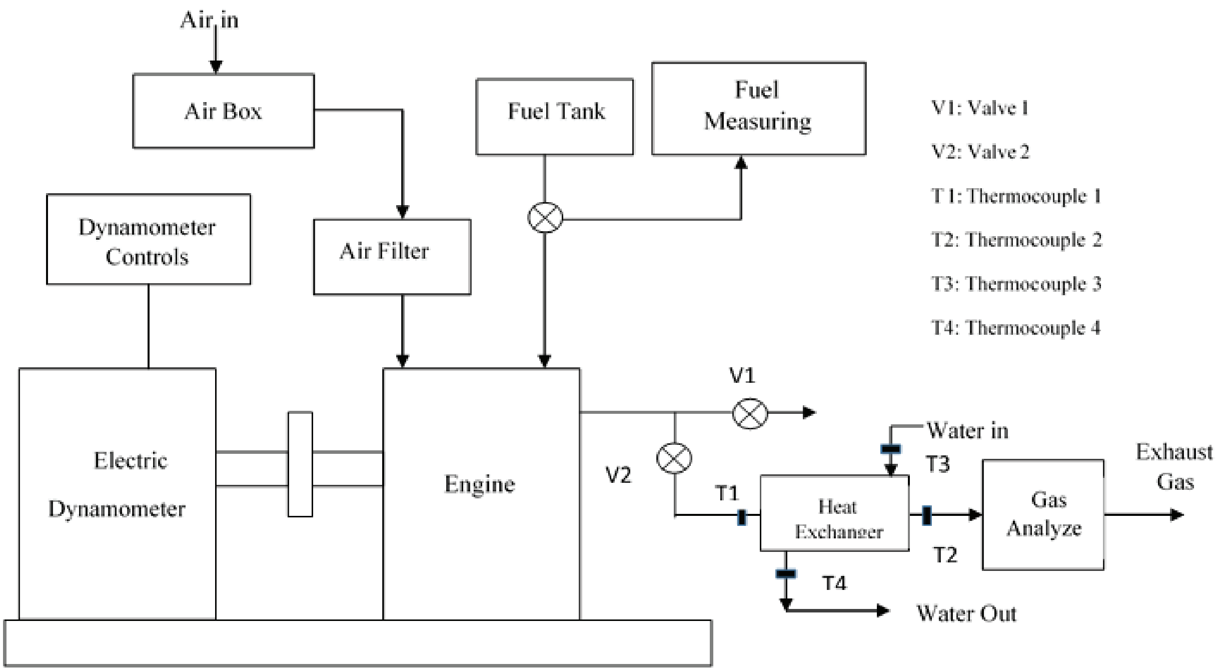

Figure 6a. Schematic diagram of experimental setup.

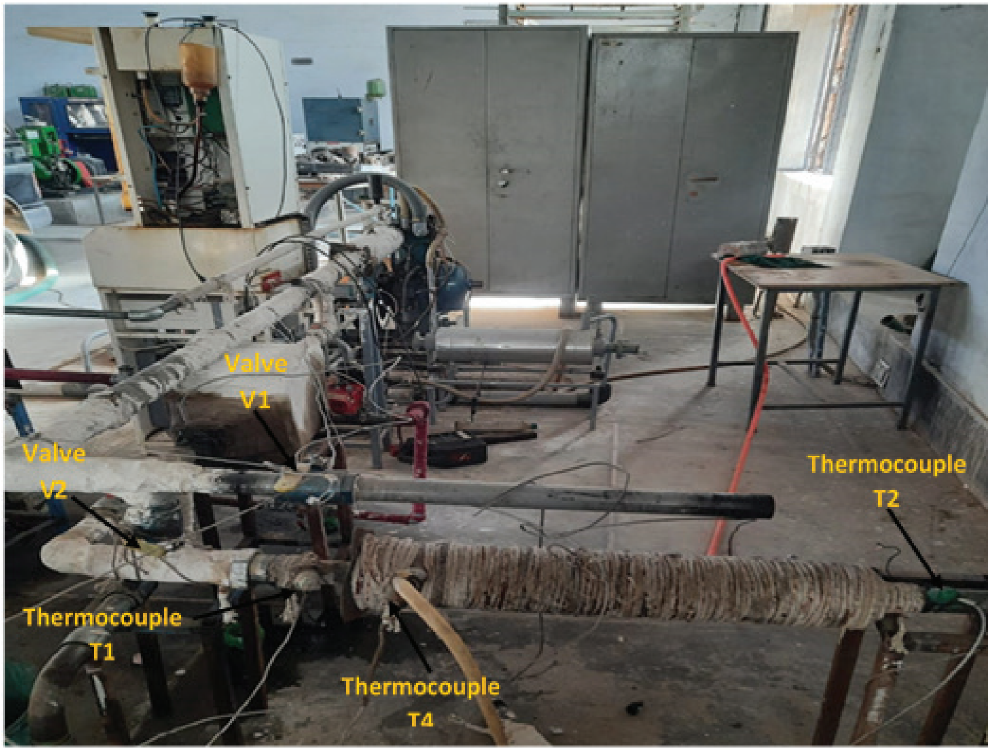

Figure 6b. Photo of experimental setup.

Table 2. Thermophysical properties of biodiesel [12, 36, 37]

Fuel Properties	Diesel	Biodiesel
Density (kg/m^3)	834.4	871
Calorific Value (MJ/kg)	42.9	35
Kinematic Viscosity (mm^2/s)	2.692	5.71
Specific Gravity	0.8344	0.871
Pour Point (°C)	32 to -57	-2.97
Flash Point (°C)	52 to 96	197.3
Cetane Number	51.8	59.08

petroleum diesel utilised in this study was provided by a nearby filling station. Eureka Sativa (Taramira) seeds oil was transformed into biodiesel by an alkali-catalyzed transesterification process. KOH was used as the catalyst amounting 1.0% on mass basis and 20% methanol was treated with the used Eureka Sativa (Taramira) seed oil [36,37]. The base catalyzed transesterification procedure was carried out in the water bath shaker. The chemical and physical properties of Eureka Sativa biodiesel is presented in the Table 2.

Methodology

The prime objective of the current study is to examine the effects of various biodiesel blends made from Eureka sativa oil on the waste heat recovery, performance, and emissions of diesel engine powered cogeneration (CHP) systems. To achieve the objective of the present research, first of all, a CHP system was developed by connecting a heat exchanger with the exhaust pipe of a diesel engine, as shown in Figure 6a and 6b. Biodiesel was prepared from Eureka sativa oil using the transesterification process as described in the above section. After this, four diesel and biodiesel blends, namely B10, B15, B20, and B25, were prepared by adding 10%, 15%, 20%, and 25% by volume of biodiesel to diesel, respectively. The cogeneration (CHP) mode were operated by keeping valve V1 closed and valve V2 open, as indicated in Figure 6a. The experimental investigation of the performance of the CHP system was conducted separately for diesel and for each blend of biodiesel. Experimental tests were prepared for each fuel at 0, 20, 40, 60, 80, and 100 percent loads at constant engine speed (1500 rpm) and compression ratio (17.5). For each fuel, the engine load varied from zero to hundred percent, and all required data such as brake power (BP), brake thermal efficiency (BTE), brake specific fuel consumption (BSFC), emissions, and thermocouple readings were recorded. The amount of fuel burned in 60 seconds was recorded in order to measure BSFC and BTE at each load condition. The AVL DiGas 444N multi-gas analyzer was used to record the PPM of different gases in exhaust for each load condition. Different graphs were plotted for engine performance, emissions, and waste heat recovery for each fuel and load condition. After plotting the graphs, the results were analyzed to determine the expected outcome.

This research also included an uncertainty analysis of the overall system. The accuracy of each used instrument is

Table 3. Uncertainty of various parameters

Parameter	Range of Experiment	Accuracy	Uncertainty (%)
Sensor engine speed	0–5500	±1 rpm	±1.0
Load display	0–50 kg	±0.1 kg	±0.4
Thermocouple	0-1200°C	±1 °C	±1.0
Rota meter System	0–250 cc	±0.1 cc	±0.9
CO_2	0 to 20% by Vol.	±0.01 %	±0.08
CO	0 to 15% by Vol.	±0.01 %	±0.06
HC	0 to 30,000 ppm	±1 ppm	±1.2
NO_x	0 to 5000 ppm	±1 ppm	±0.7

listed in Table 3. The uncertainty of the overall system was calculated by using the principle of propagation of error by Holman [38].

$$\text{Total uncertainty} = \sqrt{\begin{array}{l}[(\text{uncertainty of brake power})^2 + (\text{uncertainty of BSFC})^2 \\ + (\text{uncertainty of BTE})^2 + (\text{uncertainty of HC})^2 \\ + (\text{uncertainty of CO})^2 + (\text{uncertainty of } CO_2)^2 \\ + (\text{uncertainty of } NO_x)^2 + (\text{uncertainty Thermocouple})^2]\end{array}}$$

$$\text{Total uncertainty} = 1.75\%$$

RESULTS AND DISCUSSION

On the test rig, the experimental investigations are conducted using the biodiesel blends and diesel described above. In this part, the findings of the engine performance, emission characteristics, and waste heat recovery from exhaust gas are thoroughly analyzed.

Engine Performance

Brake thermal efficiency

Figure 7 depicts the fluctuation of BTE with load for various blends of Eureka Sativa biodiesel. The BTE rises with load for all biodiesel-diesel blends, reaching its maximum values at 80% of the test engine's full load. Due to higher friction losses and heat losses under full load conditions, the BTE decreases at full load [39,40]. Maximum values of BTE are 31.87%, 25.97%, 26.06%, 27.97% & 26.87% for B0, B10, B15, B20, and B25 respectively. The accompanying graph illustrates how diesel has a greater BTE than all blends of biodiesel and diesel at all loads due to biodiesel's lower calorific value (CV) [41]. Additionally, Figure 7 demonstrates that the BTE rises as biodiesel content rises, reaching a maximum value of 27.97% at B20. This is possibly because there is oxygen available in the biodiesel molecules (the methyl ester of the oil), which participates in burning and improves combustion efficiency [42]. The test findings also show that the BTE is lower at B25 (26.87%) than at B20 because of the leaning effect brought on by too much oxygen in the fuel [42].

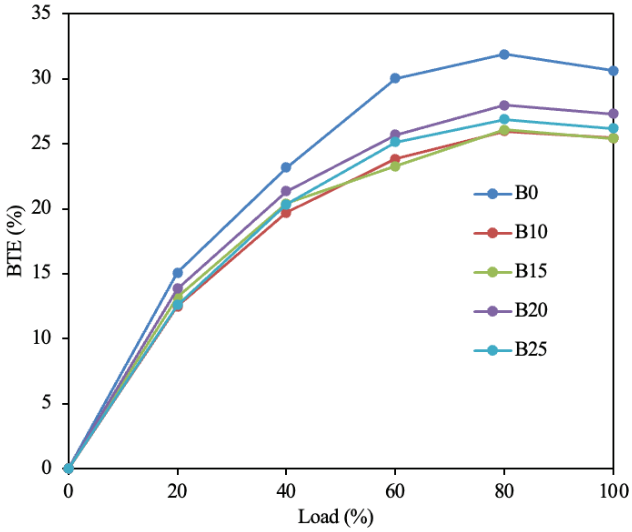

Figure 7. BTE variation with engine load for different blends of biodiesel.

Brake power

Figure 8 illustrates the distinction in BP produced by various blended fuels with a change in load. Since applied torque is a function of load and increases as load increases, BP grows as load increases [43,57]. Therefore, the maximum B.P for diesel and all biodiesel blends is observed at full load, and it is 4.75 kW for B0, 4.28 kW for B10, 4.68 kW for B20, and 4.64 kW for B25. It is clear from the accompanying graph that diesel has more brake power than biodiesel, since the brake power of a diesel engine significantly depends on calorific value and that diesel has a larger calorific value than biodiesel [44]. However, the B20 (4.68 kW) has the highest maximum brake power among all the biodiesel blends due to the effective combustion that takes place in this blend as a result of the additional oxygen molecules that are present alongside the biodiesel molecules [42].

Brake specific fuel consumption (BSFC)

The variation in BSFC with load is shown in Figure 9 which shows that BSFC for diesel as well as all blends of

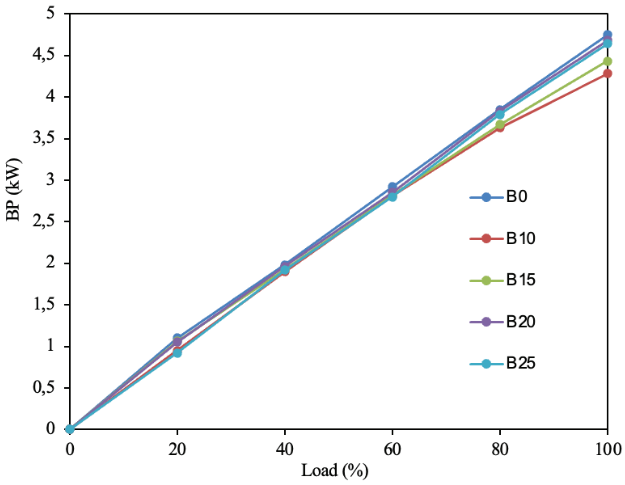

Figure 8. Brake power variation with engine load for different blends of biodiesel.

biodiesel decrease with load. The lowest values of BSFC are obtained at 80% of the test engine's maximum load, for all biodiesel-diesel blends. This is because greater friction and heat losses under full load conditions result in a decrease in net power [45]. The minimum values of BSFC are 0.250, 0.33, 0.32, 0.290, and 0.315 (kg/kW h) for B0, B10, B15, B20, and B25, respectively. Figure 9 shows that, at all loads, diesel has a lower BSFC than all blends of biodiesel. This trend was seen because, because biodiesel-diesel blends had a lower CV value than diesel, more biodiesel-diesel blends were needed to maintain the same level of power [45,58]. In addition, Figure 6 demonstrates that the lowest value of BSFC, attained at B20 among all biodiesel-diesel blends, is 0.290 kg/kW h. This is likely a result of the oxygen that is present in biodiesel (i.e., the methyl ester of the oil), which further improves the fuel's combustion efficiency by taking part in the combustion process and raising the power output, requiring less fuel to produce the same amount of power [46]. But the results also show that B25 has a higher BSFC than B20, which is due to the fact that the mixture gets leaner as the mass percentage of oxygen in biodiesel exceeds a specific threshold, which causes incomplete combustion and increases the amount of fuel needed to provide the same amount of power [45].

Emission Characteristics

NOx emission

A rise in NO_x emissions is seen with a rise in load for diesel and all blends of biodiesel, as depicts in Figure 10. Because NO_x formation is mainly due to high combustion chamber temperature and excess oxygen availability during combustion. As the load rises, the combustion chamber temperature rises, producing more NO_x [47]. The maximum NO_x emission levels for B0, B10, B15, B20, and B25 at full load are 1212, 1320, 1343, 1282, and 1431 ppm, respectively. The emission of NO_x increased with the proportion of biodiesel in the blend because of the oxygen that is present in biodiesel (i.e., the methyl ester of the oil), and the in-cylinder temperature increased with the proportion of biodiesel in the blend [4,49]. Although B20 had the lowest NO_X emission of any biodiesel blend, it was still 5.78% higher than diesel. Because efficient combustion occurs as a result of the proper air-fuel mixture ratio for B20, the available oxygen in cylinder would be less as compared to other blends, which results in reduced NOx emissions. Ulusoy et al. [48] and Patel et al. [49] reported similar findings.

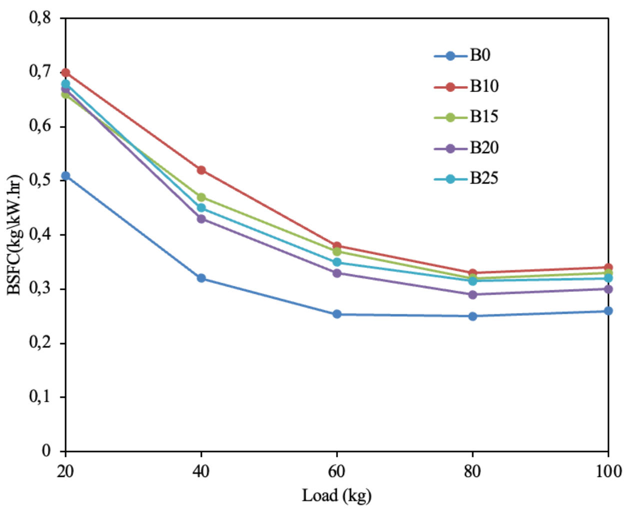

Figure 9. BSFC variation with engine load for different blends of biodiesel.

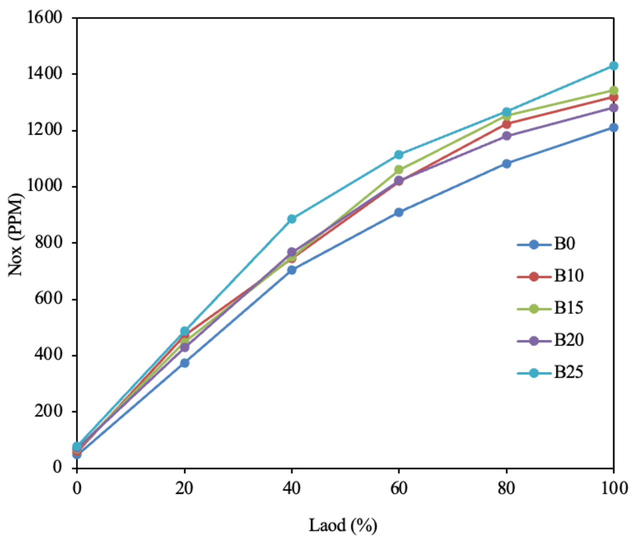

Figure 10. No_x variation with engine load for different blends of biodiesel.

CO_2 emission

With an increase in load, the CO_2 content of the emissions rises. This may be understood by the fact that the amount of fuel consumed rises as the load does. Naturally, more CO_2 is formed when increased fuel is used to provide the engine with additional heat energy [50,51]. Furthermore, as seen in Figure 11, when an engine burns a higher proportion of biodiesel, less CO_2 is released into the atmosphere. This could be the result of biodiesel fuel having a lower carbon to hydrogen ratio than diesel, as shown by Murat Karabektas et al. [52]. The B20 has the highest CO_2 emissions of any biodiesel blend, which is 1.69% lower than diesel's highest CO_2 emissions. The B20 (8.11%) exhaust emissions contain greater carbon dioxide concentrations as a result of efficient, nearly complete combustion as discussed in above section [53] (Shrivastava et al.).

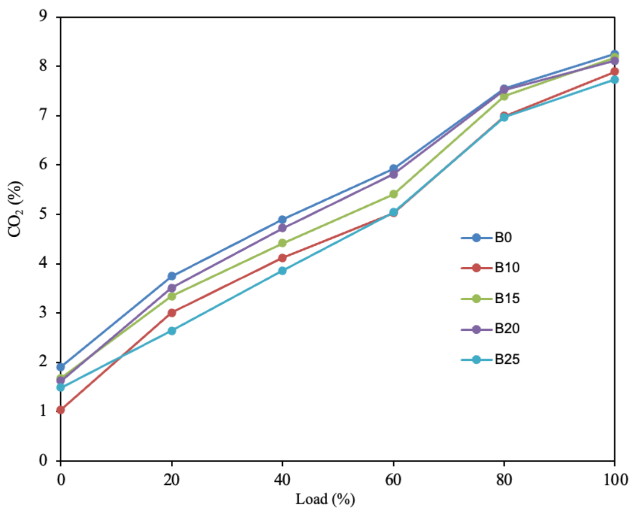

Figure 11. CO_2 emission variation with engine load for different blends of biodiesel.

CO emission

The formation of CO is due to the incomplete combustion of fuel. As seen in Figure 12, the emission of CO rises for diesel as well as biodiesel as the load rises from 0% to 100%. Since result is taken at constant rpm, the engine nearly always inhales a fixed amount of air, but the quantity of fuel increases with each escalation in load [4]. As a result, the air-to-fuel ratio falls, which raises the CO level due to incomplete combustion. Among all biodiesel blends, B20 had the lowest CO emissions due to efficient combustion a result of the proper air-fuel mixture ratio for B20. A similar study was reported by Chaudhari et al. [54].

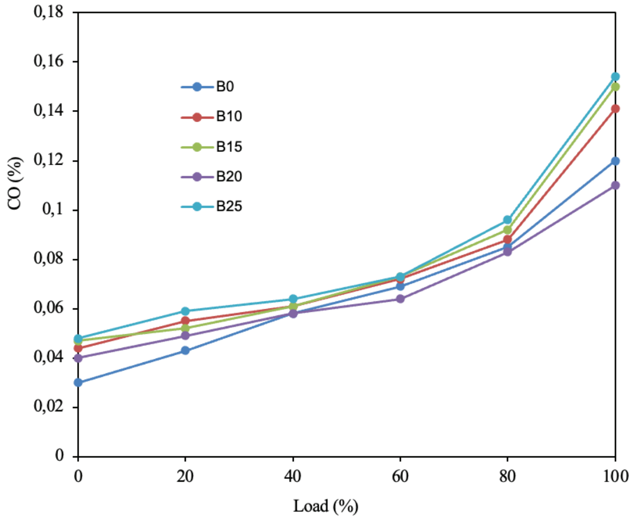

Figure 12. CO emission variation with engine load for different blends of biodiesel.

HC emission

As indicated in Figure 13, unburned hydrocarbon emission were also made for various loads. For diesel and all of its blends, it can be seen that the amount of unburned hydrocarbon (HC) rises as the load increases, which may be because the load increases fuel consumption [45]. The maximum HC emission levels for B0, B10, B15, B20, and B25 at full load are 108, 113, 119, 102, and 123 ppm, respectively. The graph shows that B20 has the lowest HC emission (102 ppm) due to efficient fuel combustion, which is consistent with the findings of Shrivastava et al. [53].

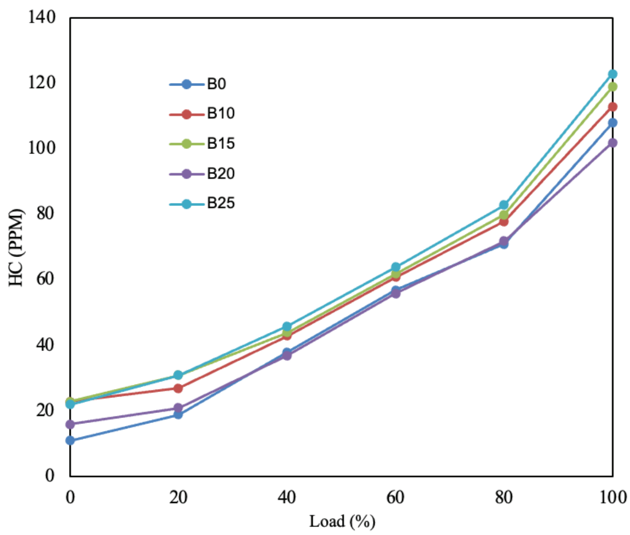

Figure 13. HC emission variation with engine load for different blends of biodiesel.

Waste Heat Recovery Parameter

Exhaust gas temperature

Figure 14 illustrates how engine loads and various biodiesel-diesel blends affect the temperature of exhaust gases. For diesel as well as biodiesel blends, the EGT increases with load because additional power is needed to handle greater loading, and additional power requires more fuel [55]. The EGT rises as a result of the burning of more fuel because there is more heat available in the combustion chamber [56, 57]. The maximum exhaust gas temperatures observed at 100% load are 265 °C, 298 °C, 314, 340 °C, and 365 °C for pure diesel, B10, B15, B20, and B25, respectively. Above readings shows that the EGT rises as the percentage of biodiesel in the blends increases, and the highest maximum EGT is 365 °C at B25, which is 100 °C greater than the EGT of pure diesel. The unburned fuel in the premixed combustion phase and the protracted porous nature of biodiesel both increase the temperature of exhaust gases [58].

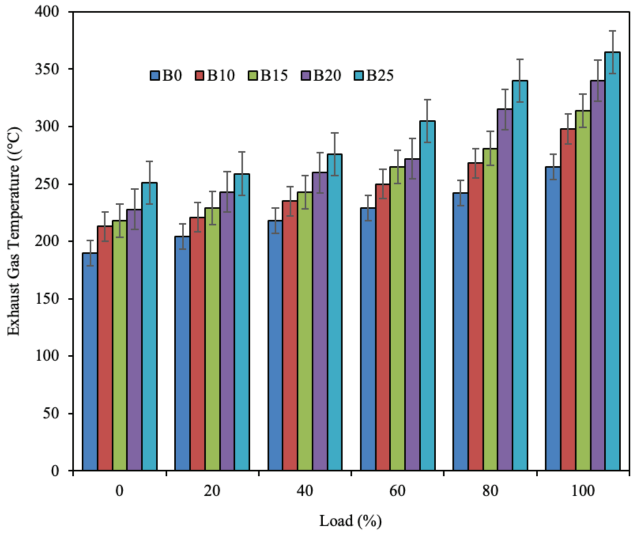

Figure 14. Variation of exhaust gas temperature with load for different blends of biodiesel.

Waste heat recovery

The variation in waste heat recovery from the exhaust gas of a diesel engine with engine load for diesel as well as all blends of biodiesel is depicted in Figure 15. The rate of heat extracted from the exhaust gas rises with engine load because the EGT rises with engine load, as discussed in the above section. The EGT also rises with the proportion of biodiesel in the blend, and the increased EGT increases heat recovery [59,12]. The maximum rates of heat recovered noticed at full load conditions are 1.11 kW, 1.38 kW, 1.52 kW, 1.66 kW, and 1.80 kW for pure diesel, B10, B15, B20, and B25, respectively. Figure 16 displays the CHP system's total useful energy output. The total useful energy output produced by the CHP system rises with engine load, as can be shown in Figure 13. Diesel, B10, B15, B20, and B25, respectively, have maximum total energy outputs of 5.86 kW, 5.66 kW, 5.95 kW, 6.34 kW, and 6.44 kW under full load conditions [12].

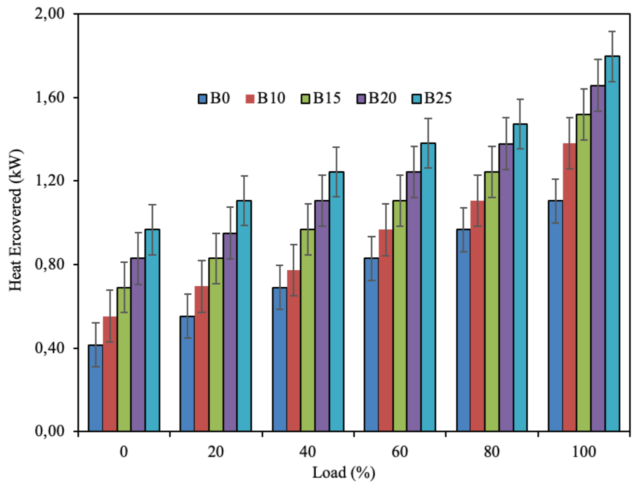

Figure 15. Variation of heat recovered with load for different blends of biodiesel.

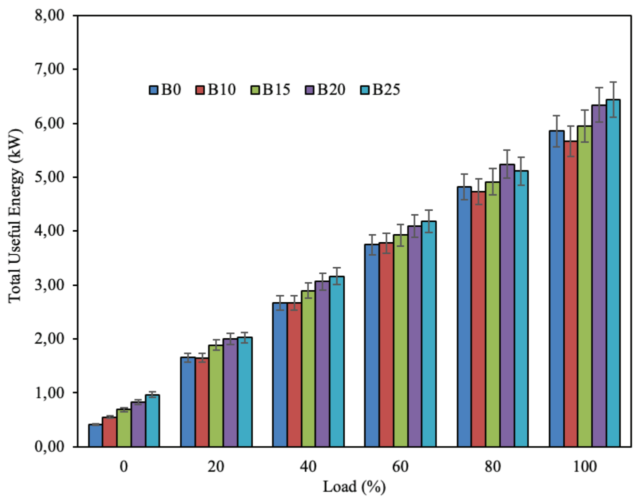

Figure 16. Variation of total output energy with load for different blends of biodiesel.

Overall efficiency

The efficiency of combined heating and power generation system is illustrated in Figure 17. The overall efficiency rises with load for all biodiesel-diesel blends, reaching its maximum values at 14 kg of load that is 80% of the test engine's full load. Maximum values of BTE are 40.42%, 38.49%, 37.46%, 34.79 % & 34.62% for B0, B25, B20, B15,

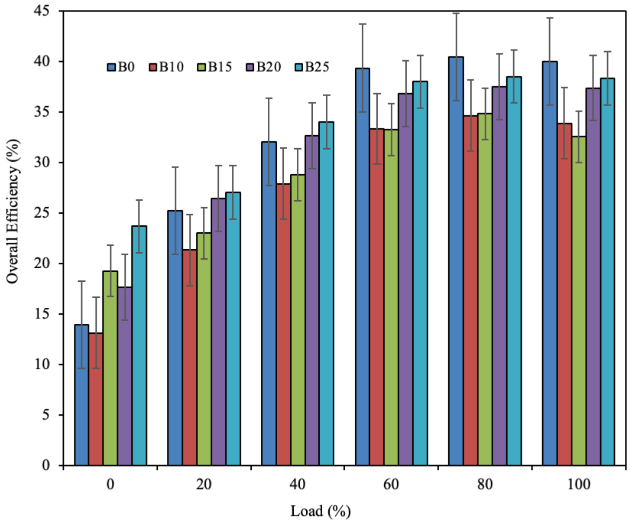

Figure 17. Overall efficiency of CHP system for different blends of biodiesel.

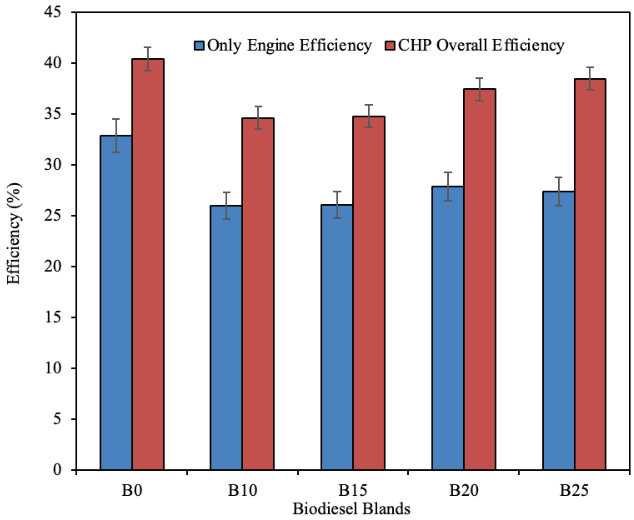

Figure 18. Comparison of engine efficiency and CHP overall efficiency for different blends.

Table 4. Measured parameter of cogeneration system

S.N.	Parameter	Diesel	B10	B15	B20	B25
1	Brake thermal efficiency (%)	31.87	25.97	26.06	27.97	26.87
2	Brake Power (kW)	4.75	4.28	4.43	4.68	4.64
3	BSFC (kg/kWh)	0.25	0.33	0.32	0.29	0.315
4	Exhaust Gas Temperature (°C)	265	298	314	340	365
5	Waste Heat Recovery (kW)	1.11	1.38	1.52	1.66	1.80
6	Overall (CHP system) Efficiency (%)	40.42	34.62	34.79	37.46	38.49
7	NOx ppm	1212	1320	1343	1282	1431
8	CO_2 (%)	8.25	7.89	8.18	8.11	7.73
9	CO (%)	0.12	0.141	0.15	0.11	0.154
10	HC ppm	108	113	119	102	123

and B10 respectively at 80% of the full load. It can also be observed in Figure 17 that the overall efficiency of the combined system is highest for pure diesel due to its lower fuel consumption and highest calorific value [12,57]. And the overall efficiency of biodiesel blends increases with a rise in the proportion of biodiesel because exhaust gas temperatures increase with an increase in the proportion of biodiesel in the blend, resulting in an increase in the amount of heat recovered, and increased recovered heat increases the overall efficiency of the CHP system [59].

Figure 18 compares the efficiency of the CHP system as a whole to that of the engine alone. For both diesel and biodiesel, a CHP system has a greater maximum efficiency than an engine. The maximum efficiencies for pure diesel, B10, B15, B20, and B25 increase by 7.55%, 8.65%, 8.73%, 9.59%, and 11.12%, respectively. The CHP system's overall efficiency rises as the proportion of biodiesel in the diesel and biodiesel blend rises [59]. This is owing to the fact that waste heat recovery rises as the fraction of biodiesel in the blend increases [12].

The readings of the above-discussed parameters for each load condition were taken after 10 to 15 minutes of load application to achieve steady state. For all parameters, three readings were taken for each load condition, and the average values of the measured parameters were taken in this study. All measured parameters regarding performance, emission and waste heat recovery of cogeneration system for pure diesel as well as different blends of Eureka sativa biodiesel are shown in the Table 4.

CONCLUSION

In this study, the effects of pure diesel and various biodiesel blends are investigated on the energy output and exhaust emissions of a CHP system prime movered by a diesel engine. Tests were done using pure diesel fuel and four different biodiesel blends that were 10%, 15%, 20%, and 25% by volume under steady state conditions and under various loads. The impact of Eureka Sativa biodiesel blends on performance, emission characteristics, and waste heat recovery of diesel engine is investigated. From the current investigation, the following inference can be made:

- The B.P. and BTE increase as the percentage of biodiesel in the blend increases, while the BSFC decreases. However, among all biodiesel blends, B20 (20% Eureka sativa oil biodiesel + 80% pure diesel) has the highest B.P. and BTE, while BSFC is the lowest, with the exception of pure diesel, which has 4.75 kW, 27.97%, and 0.290 kg/kW-h. The BTE of the B20 blend is 4.13% smaller than diesel fuel, while the BSFC of B20 is 0.04 kg/kW-h greater than diesel.
- The emissions of NO_x, CO_2, CO, and unburned HC rise with engine load for diesel and all blends of biodiesel. The emission parameters increased as the percentage of biodiesel increased, with Nox, CO, CO2, and HC all increasing, whereas the B20 blend had the most favorable emission characteristics of all blend combinations except diesel. In comparison to pure diesel, B20 emits 5.75% more NOx, 0.86% less CO2, 8.33% less CO, and 5.55% less HC.
- EGT increases with engine load as well as the proportion of biodiesel in the diesel and biodiesel blends. As a result, compared to 1.1 kW for pure diesel, waste heat availability for B25 rises to 1.80 kW.
- The overall efficiency of the CHP system rises with load for all biodiesel-diesel blends, reaching its maximum values at 80% of the test engine's full load. The maximum overall efficiency of a CHP system is 42.42 % for diesel fuel. The B25 has the highest overall efficiency (38.49%) among all tested blends which is 1.93 % lower than pure diesel. The maximum overall efficiency of the system due to waste heat recovery increased by 7.55% to 11.12% as the percentage of biodiesel increased from 0 to 25 % in the blends.
- Hence it can be concluded that B20 is the best fuel among all biodiesel blends in terms of engine performance and emission characteristics. Whereas B25 is a better fuel in terms of WHR and overall cogeneration unit efficiency.

Future Work

It can be observed that the efficiency of the overall CHP system is higher for pure diesel, even though the waste heat recovered increases with the increased proportion of biodiesel. This is because the thermal efficiency of only diesel engines is higher for pure diesel than all bends of biodiesel. So to increase the thermal efficiency of engines for biodiesel blends, an additive like n-pentane may be used, as described by Medhat Elkelawy et al. [29]. Because of this, the overall (CHP system) efficiency for biodiesel blends will rise, and it may surpass the overall efficiency for pure diesel. So the performance of CHP systems for different blends of Eureka Sativa biodiesel along with **additive** n-pentane may be considered a future scope of this research.

NOMENCLATURE

CHP	Combined heating and cooling
BSFC	Brake specific fuel consumption
WHR	Waste heat recovery
DI	Direct Injection
CI	Compression ignition
BP	Brake Power
BTE	Brake Thermal Efficiency
BSFC	Brake Specific Fuel Consumption
CV	Calorific Value
EGT	Exhaust Gas Temperature
PM	particulate matter
HC	hydrocarbons
ppm	parts per million
I.C Engine	Internal Combustion Engine
$\dot{Q}_{ex}$	Heat carried by exhaust
Cp	Specific Heat
$\dot{m}_a$	Mass Flow rate of air
$\dot{m}_f$	Mass Flow Rate of Fuel
KOH	Potassium hydroxide

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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