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# Exergy, energy, and emissions analyses of binary and ternary blends of seed waste biodiesel of tomato, papaya, and apricot in a diesel engine

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ABSTRACT

Biodiesel is considered as a renewable biofuel-based substitute for fossil diesel as the properties of biodiesel are similar to those of normal diesel fuel. However, biodiesel has some properties which a negative effect on engine combustion. Binary and ternary fuel blends (blends of biodiesels with the opposite properties) are the best environmentally friendly alternative in compression ignition engines. In this paper, the effects of binary and ternary blends of tomato, papaya, and apricot seed biodiesels on the energy and exergy balance as well as emissions on a compression ignition diesel engine were experimentally and theoretically investigated. The obtained results reveal that the maximum and minimum exergy efficiency are related to the biodiesel of tomato-papaya blend at about 29.63% and pure diesel at about 28.46% respectively. Also, the obtained results show that, compared to the tomato biodiesel-diesel blend, using the binary blends decreases the percentage of heat loss exergy by 5.5% and 3.3% on average for tomato-papaya biodiesel -diesel and tomato-apricot biodiesel-diesel, respectively. The results address that the energy percentage of exhaust emissions at the speed of maximum torque and maximum power averaged 30% while this fraction for exergy was about 13%. The emission results show that the minimum oxygen monoxide emissions, which is 0.3% less than that of diesel, is related to the tomato-apricot-papaya biodiesel-diesel ternary blend.

## 1. Introduction

Diesel engines are widely used in various sectors such as agriculture, transportation, and construction or manufacturing industries due to improved fuel economy, durability, reliability, and specific power output. However, they are known to be the major sources of nitrogen oxides (NOx) and particulate matter (PM) emissions; Therefore, significant exploration is focused on numerous feedstocks (edible/nonedible oils along with their methyl esters (biodiesel)) as a probable source of fuel for diesel usage sectors [1,2]. Nevertheless, biodiesel has some cons' such as low heating value, higher viscosity, higher density, and poor cold-weather flow properties such as cloud point, pour point, and cold filter plugging point. These disadvantages lead to poor atomization of fuel, injector clogging, narrow spray patterns of fuel in the combustion chamber, and incomplete combustion [3,4]. The different approaches of the researchers who are working in the area of alternate fuels for engine application with the aim of the partial and complete elimination of diesel are tabulated in Table 1. The suitability of neat biodiesel in diesel engines is also too low as it affects the engine performance and running life due to high viscosity. One of the best possible ways to overcome the above stated problems is to use binary and ternary fuel blends, which possess properties as per the American Society for Testing Materials (ASTM) specification for biodiesel. Using a high viscosity methyl ester in various proportions with another low viscosity biodiesel has the ability to give a stable solution and creates a fuel that is feasible to completely replace diesel in compression ignition (CI) engines [5]. The low viscous biodiesel can be a solubilizer and improve the properties of the blend [6,7]. The major advantages of this blend are that it can be used in CI engines without any major tweaking and that it produces less harmful gases as emissions to the environment.

The second-generation biofuels (or advanced biofuels) are derived from non-edible crop feedstocks and industrial wastes etc. [8]. One of these sources is tomato seed waste. Almost 40 M tons of tomatoes are processed annually in the global food industries (Tomato [9]). During

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**Table 1**
Overview of elimination of conventional fuel from CI engines.

| | | |
|---|---|---|
| Elimination of conventional fuel from diesel engines | Partial elimination | Biodiesel + diesel blends |
| | | Biodiesel + alcohol + diesel blends |
| | | Biodiesel1 + biodiesel 2 + diesel blends |
| | | Biodiesel + alcohol blends |
| | Complete elimination | Neat biodiesel |
| | | Biodiesel + oil blends |
| | | Other mixed feedstock |

the industrial processing of tomatoes, three to seven percent of the raw materials are lost as waste [10,11]. Although tomato seed comprises nearly 72 % of tomato waste by weight, it is not suitable for human consumption and is commonly used as animal feed or fertilizers [12–15]. The oil content of tomato seed is registered in the range of 13.3–19.3 %, and can be used as a source of biodiesel production [16–19]. Another source of biodiesel production is Papaya Seed Oil (PSO) which is non-edible and converting the waste product (seeds) of the fruit into biodiesel is a sensible option. Papaya fruit can weigh from 200 g to more than 3000 g, and its seed content can be approximately 15 %–20 % of the wet weight of the fruit. Since the seeds are not consumed, 15 %–20 % of the biomass i.e., the amount of seeds is discarded. These seeds contain 30 %–34 % oil with nutritional and functional properties, which can be utilized as the commercial feedstock for biodiesel synthesis. Therefore, the evaluation of papaya seed oil as biodiesel feedstock will contribute to the development of regional communities and their overall economy [20–22]. Prunus armeniaca L. belonging to the family Rosaceae grows over the five continents of the world and its production level exceeds two million tons. Apricot yields 22–38 % of kernels, which contain up to 54.2 % oil. In addition, this biodiesel is produced from the apricot feedstock that is consistently available, economically viable, and locally available. [22–24], could be used as another source for converting to biodiesel. Moreover, large amounts of fruit seeds are discarded yearly at processing plants. This not only wastes a potentially valuable resource but also aggravates an already serious disposal problem [3,25,26].

The performance of diesel engines on biodiesel blends has been investigated by many researchers with promising results. These research works mainly contain reported information about ignition delay, combustion, efficiency, and pollutant emissions. Moreover, the analysis of the second law of thermodynamics is one of the recent tools utilized in energy conversion systems [27]. Exergy or energy availability is a key parameter in the second law analysis. Exergy is defined as the maximum theoretical useful work that can be obtained as a system interacts in an equilibrium state. In general, exergy analysis is used as an environmental assessment tool to account for waste and determine real energy efficiency. Unlike energy, exergy can be destructed. Exergy destruction, which is called irreversibility, is one of the main reasons for the low efficiency of energy conversion systems like internal combustion engines [28].

There are a vast number of studies for the first and second law analysis of internal combustion engines and a few of the recent studies are reviewed below. Asokan et al., [29] made a PSO ternary blend by mixing PSO biodiesel (10 % vol) with watermelon biodiesel (10 % vol). They reported that the emissions of oxygen monoxide (CO), hydrocarbon (HC), and smoke were less than for diesel. In another study, Anwar et al., [30] compared PSO biodiesel blends with Apricot Seed Oil (ASO) biodiesel and found ASO blends have better engine performance while emitting higher exhaust emissions. Exergy analysis of direct ignition (DI) diesel engine fueled with several biodiesel/diesel blends containing different amounts of expanded polystyrene and novel soluble hybrid Nano catalyst was carried out by Aghbashlo et al., [31]. Gnanamani et al., [32] studied diesel-cotton seed oil blends that were tested on an engine running with a direct fuel injection mode of operation. The experiments were conducted 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. Habibian et al. [33] investigated the combustion process in a DI diesel engine (OM314) with biodiesel fuel in different Blends (B20, B40, B100) of soybean at full load and 1200 rpm by a thermodynamic model using both thermodynamics first and second laws of thermodynamics. The results of the analysis of energy and availability balance showed that the efficiency of the first and second laws for pure biodiesel fuel was more than of the other two fuels and total availability. Indicated work availability, heat loss availability, burned fuel availability and irreversibility for 20 % biodiesel fuel were more than of two other fuels. Nabi and Rasul [34] compared diesel engine performance, emissions, energy, and exergy parameters of waste cooking and macadamia (Macadamia integrifolia) biodiesels and a reference diesel. They found out that without a significant reduction in engine performance, a significant reduction in total unburnt hydrocarbon (THC), carbon monoxide (CO), and particulate matter (PM) emissions with a penalty of increased nitrogen oxides (NOx) emissions were realized with all two biodiesel blends. Notwithstanding the great number of studies in energy and exergy analysis in internal combustion engines, there are no attempts in exergy analysis of diesel engines in which binary and ternary fuels are used to reduce pollution production. Also, although the exergy and pollution characteristics of alternative fuels, especially biodiesels, were investigated in different studies, the effects of ternary biodiesel are not established. In other words, diesel engine cycles for three biodiesels and diesel blend cases have not been studied theoretically and experimentally. Therefore, for the first time, the first and second laws of thermodynamics were employed in this study in order to analyze the quantity and quality of energy and exergy in an indirect ignition diesel engine, which was running with the binary and ternary biodiesel and diesel blends.

## 2. Materials and methods

In this section, engine and measuring devices' specifications, biodiesel fuel, experiment conditions, thermodynamic modeling materials, and test methods are explained.

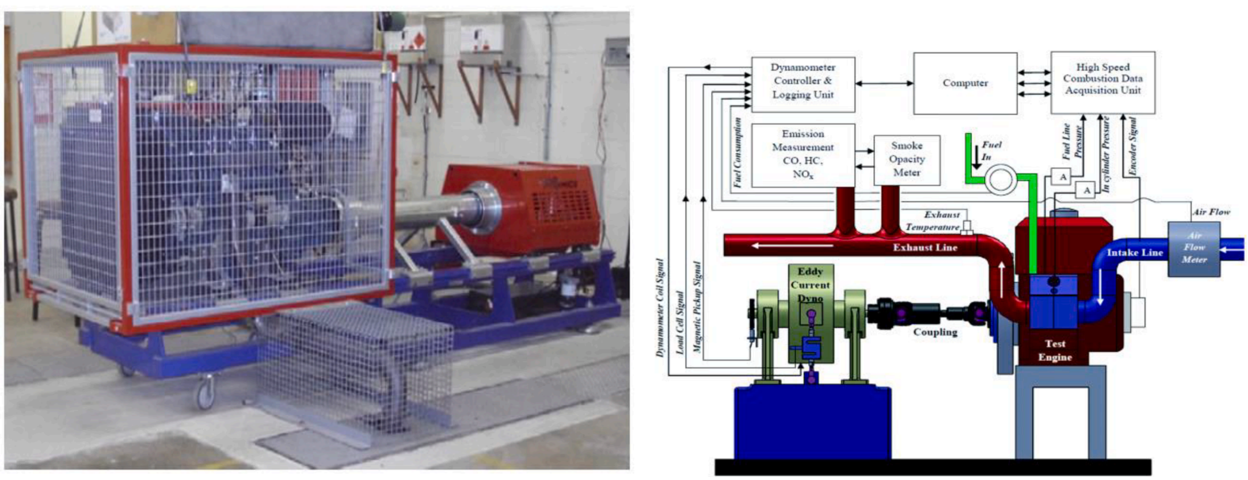
### 2.1. Engine and measuring devices' specifications

Engine performance and emission tests were carried out according to ISO 8178 standard engine test procedure. The experiments were carried out based on short-term tests in the form of multiple (factorial) factor designs. The tests were done in the thermodynamics laboratory of Central Queensland University (Australia). The four-cylinder indirect ignition (IDI) Kubota V3300 diesel engine was used in this study. Table 2 indicates its technical features and characteristics [35]. To examine the performance, the engine was coupled to the water-cooled operation and low-inertia GW63 eddy-current dynamometer, whose technical specifications are max. power 63 kW, max. torque 250 ± 4 %FS Nm, and max. speed 9000 ± 1 rpm (Fig. 1a). In addition, the Dina Engine fuel consumption measurement system with an accuracy of ± 0.001 kg was used to measure fuel consumption and the MAHA-MGT5 device measured the emissions of the engine. In addition, the exhaust gas temperature data in terms of degrees Kelvin was automatically recorded on the computer by an AVL DISMOKE 480 BT device with an accuracy of ± 5 K. This machine can determine CO, Carbon dioxide ($CO_2$), and HC values using infra-red technology, and also the amounts of oxygen ($O_2$) and $NO_x$ gases using chemical sensors. A schematic of the set-up of these devices is shown in Fig. 1b. The accuracy and uncertainty values of the experimental data, which were calculated based on the method of Holman and Gajda [36], are encapsulated in Table 3.

Also, the errors related to the equipment in measuring the required parameters have been recognized and measured as summarized in Table 4 [37]. In the data analysis, a 95 % confidence interval was

**Table 2**
Kubota V3300 engine specifications.

| Specifications of Kubota V3300 –T | |
|---|---|
| Kind of engine | Vertical, 4 cycle liquid cooled diesel |
| Number of cylinders | 4 |
| Cylinder diameter | 98 |
| Compression ratio | 22.6 |
| Total displacement (L) | 3.32 |
| Stroke (mm) | 110 |
| System of Combustion | E-TVCS |
| System of Intake | Naturally aspirated |
| Output gross intermittent (kW/rpm) | 54.5/2600 |
| No load low idling speed (rpm) | 700–750 |
| No load high idling speed (rpm) | 2800 |
| Net continuous (kW/rpm) | 44.1/2600 |
| Net intermittent (kW/rpm) | 50.7/2600 |
| Crankshaft rotation direction | Counter-clockwise (viewed from flywheel side) |
| Governing | Centrifugal fly weight high speed governor |
| Kind of fuel Diesel | No-2-D based on ASTM D975 |
| Capacity of starter (V kW) | 12–2.5 |
| Capacity of Alternator (V A) | 12–60 |
| Dry weight with SAE flywheel and housing (kg) | 272 |




(a) (b)

**Fig. 1.** (a) Kubota V3300 Combustion System; and (b) schematic of engine and dynamometer.

**Table 3**
Accuracy and uncertainty of the experimental measurement.

| Measurement | Accuracy (%) | Uncertainty |
|---|---|---|
| Engine torque ± 0.1 Nm | ±4% FS | ± 0.04 |
| Engine speed | ±1rpm | ± 0.05 |
| Engine fuel consumption | ±0.001Kg/s | ± 0.48 |
| Airflow measurements | ±0.04 L/s | ± 0.1 |
| Water temperature | ±1 °C | ±1.12 |
| Exhaust gas temperature | ±1 °C | ±0.31 |
| NO emission | ±5ppm | ±0.58 |
| HC emission | ±4ppm | ±0.77 |
| CO emission | ±0.02 %vol | ±0.8 |
| CO2 emission | ±0.3 %vol | ±0.27 |
| O2 emission | ±0.02 %vol | ±0.2 |

considered for the difference in means.

### 2.2. Biodiesel fuel and experiment conditions

Three kinds of methyl esters were used in this study. First, biodiesel was obtained from tomato waste seed oil which was converted to biodiesel by the double-stage transesterification method. A double-glazed glass reactor with 8 L capacity was used at Tarbiat Modares University New Research Centre, Renewable Energy Laboratory, and Biodiesel Division, Iran [19]. Second, biodiesel produced from PSO was sourced from the Reliable Convenience Store, an approved local supplier of vegetable oils in Central Queensland, Australia [22], and third biodiesel was produced from Apricot (Prunus armeniaca L. species) seed oil was purchased from a local producer named Chromium Group Pty ltd. of Eumundi, Queensland, Australia [22]. In the previous research of the authors, the constituents of the biodiesels have been given [19,20,38].

Table 5 shows the physical and chemical properties of pure Diesel (D) and pure biodiesel produced from Tomato (T), Papaya (P), and Apricot (A) as well as the blends of Tomato, Apricot, and Papaya (TAD), TPD, and Tomato, Apricot, Papaya, and Diesel (TAPD) based on the ASTM standard. In this study, the engine was operated at different engine speeds (ranging from 1200 to 2400 rpm) and full load since The full load engine characteristics and, thus the performance characteristics constitute bases for the calculations related to the traffic flow on the roads [39] Only B20 blends (Biodiesel 20 % and 80 % diesel by volume) were investigated in this study because B20 is the most popular biodiesel blend. In addition, comparisons of the effect of binary and ternary fuels

**Table 4**
Total measurement uncertainty of each parameter.

| Computed performance parameter | Measured Variables | Instruments used | % uncertainty of measuring instruments | Calculation | Total % measuring uncertainty |
|---|---|---|---|---|---|
| BP | Load, rpm | Load sensor, load indicator, speed measuring unit | 0.19, 0.1, 1.1 | $\sqrt{0.19^2+0.1^2+1.1^2}$ | 1.2 |
| Load | Load | Load sensor, load indicator | 0.19, 0.1 | $\sqrt{0.19^2+0.1^2}$ | 0.21 |
| Fuel flow | Fuel flow | Fuel measuring unit, fuel flow transmitter | 0.054, 1.3 | $\sqrt{0.054^2+1.3^2}$ | 1.3 |
| IMEP | Pressure, speed | pressure sensor, speed measuring unit | 0.1, 1.1 | $\sqrt{0.1^2+1.1^2}$ | 1.7 |
| BSFC | SFC (diesel), BP | Fuel measuring unit, fuel flow transmitter, load sensor, load indicator, speed measuring unit | 0.054, 1.3, 0.19, 0.1, 1.1 | $\sqrt{0.054^2+1.3^2+0.19^2+0.1^2+1.1^2}$ | 1.7 |

**Table 5**
Specifications of studied biodiesels according to ASTM standard.

| Property Name | unit | D | T | P | A | TAD | TPD | TAPD | Limit range | Test Method STM |
|---|---|---|---|---|---|---|---|---|---|---|
| Molecular Formula | | $C_{14}H_{25}$ | $C_{17}H_{31}O_2$ | $C_{18}H_{33}O_2$ | $C_{17}H_{32}O_2$ | $C_{17}H_{32}O_2$ | $C_{18}H_{32}O_2$ | $C_{17}H_{32}O_2$ | | |
| Yield % of the Biodiesel | | – | 90.18 % | 91.3 % | 92 % | – | – | – | | |
| Kin. Viscosity @ 40 °C | cSt | 3.23 | 5 | 3.53 | 4.26 | 4.25 | 4.22 | 4.23 | 1.9–6 | D445 |
| Density – @15 °C | Kg/m3 | 827.2 | 883 | 840 | 855 | 841.1 | 833.6 | 840.64 | max 900 | D4052 |
| Flash Point (Closed Cup) | °C | 68.5 | 190 | 112 | 105 | | | | min 130 | D93 |
| TAN | mg KOH/g | 0.05 | 0.74 | 0.42 | 0.25 | | | | max 0.8 | D974 |
| Cetane Index | 0 | 48 | 47.7 | 48.29 | 50.45 | | | | min 47 | D976 |
| Lower Calorific Value | J/g | 45,300 | 36,666 | 38,490 | 39,640 | 38,174 | 37,808 | 38,460 | | |
| Oxygen Content | wt% | 1 | 11.29 | 11.27 | 11.21 | 11.92 | 11.83 | 11.7 | | |
| Carbon Content | wt% | 87 | 75.41 | 75.9 | 75.23 | 76.01 | 77.87 | 76.02 | | |
| Hydrogen Content | wt% | 12 | 11.57 | 11.72 | 11.72 | 11.72 | 11.86 | 11.69 | | |

would become very difficult if other blends were used for the study. All conditions at which the engine has been tested are shown in Table 6. Engine performance data (including power, fuel consumption, exhaust gas temperature, and outlet water temperature) and exhaust emissions were measured on binary and ternary fuels in different operating conditions. Additionally, energy and exergy analyses were performed by evaluating fuel energy, power, heat loss, gas exhaust loss, and irreversibility. To have a comprehensive comparison of the effects of the combination of B20 binary and ternary fuels on diesel engine performance, it was compared with the performance of pure diesel. The experiments were performed on the test engine while it was fueled with the following five different fuel types:

A) Pure diesel (D).
B) 80 % diesel and 20 % tomato biodiesel (TD).
C) 80 % diesel and 20 % binary blend biodiesel of tomato (10 %) and papaya (10 %) (TPD).
D) 80 % diesel and 20 % binary blend biodiesel of tomato (10 %) and apricot (10 %) (TAD).
E) 80 % diesel and 20 % ternary blend biodiesel of tomato (6.66 %), apricot (6.66 %) and papaya (6.66 %) (TAPD).

**Table 6**
All engine test conditions.

| Biodiesel type | D | TD | TPD | TAD | TAPD |
|---|---|---|---|---|---|
| Percentage of Biodiesel | 20 % | 20 % | 20 % | 20 % | 20 % |
| Percentage of Load | 100 % | 100 % | 100 % | 100 % | 100 % |
| Engine Speeds | 1200<br>1400<br>1600<br>1800<br>2000<br>2200<br>2400 | 1200<br>1400<br>1600<br>1800<br>2000<br>2200<br>2400 | 1200<br>1400<br>1600<br>1800<br>2000<br>2200<br>2400 | 1200<br>1400<br>1600<br>1800<br>2000<br>2200<br>2400 | 1200<br>1400<br>1600<br>1800<br>2000<br>2200<br>2400 |

At each tested engine speed, the percentage of biodiesel was fixed at 20 % based on volume. At every operating condition, five types of fuel were tested. These are indicated as "D", "TD'', "TPD'', "TAD", and "TAPD" on the various figures showing graphs of results. The ISO 8178 standard engine test procedure was used to carry out tests of engine emissions and performance and results were based on short-term tests by applying a full factorial experiment based on the complete randomized block design.

### 2.3. Thermodynamic modeling

Fig. 2 shows an engine schematic form in order to describe the theoretical thermodynamic model. The input and output energies to and from the engine can be represented by taking into account the engine as a control volume. Energy is supplied to the engine as the chemical energy of the fuel and the energy of intake air and leaves as output power, energy in the exhaust gas, and heat transfer [40,41]. For the energy and exergy analysis of the engine control volume Fig. 2, it is assumed that the entire engine setup operates at steady-state conditions; the intake air and exhaust gas behave as mixtures of ideal gases, and potential and kinetic energy changes are minor or ignored.

To calculate the energy and exergy distribution of the different fuels, several tools have been used in different processes or systems which approximately follow a common work pattern for all of them. In research related to internal combustion engines, these analyses usually consider three stages. Firstly, combustion products must be known. For this purpose, the general form of the reaction equation of fuel combustion for each test operation was determined as follows [42–44].

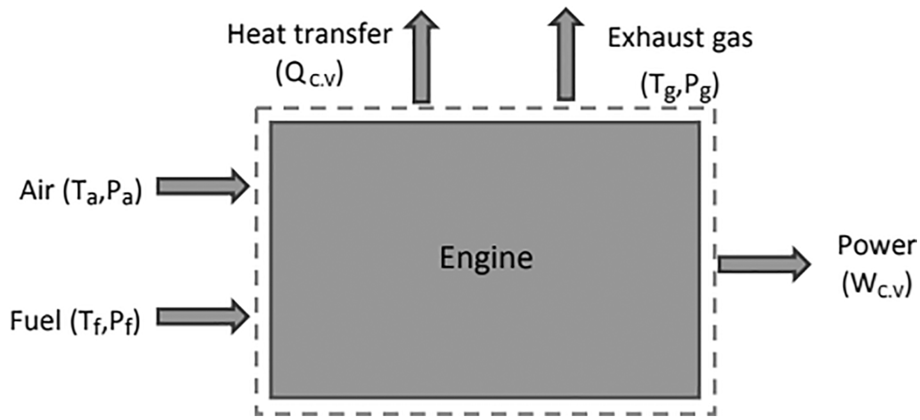



**Fig. 2.** Engine control volume.

$$a\,C_{14}H_{25} + b\,C_xH_yO_z + c\,C_7H_{17} + d\,(O_2 + 3.76N_2) \rightarrow e\,CO + f\,CO_2 + g\,NO + h\,NO_2 + i\,O_2 + j\,N_2 + k\,HC + m\,H_2O$$

The diesel fuel used in the test engine is a commercial product and its chemical formula is C14.4H24.9. The molecular formulas of pure Tomato, Papaya, and Apricot biodiesel used in the test and their binary and ternary blends are approximated as the formulas in Table 3. Some simplifying assumptions were made for determining the reaction equation. The combustion equation is written based on 1 mol of fuel (i.e., a + b + c = 1). The molar fractions of combustion products of CO, $CO_2$, NO, HC and $O_2$ are measured on a dry basis with a gas analyzer. The other coefficients of the equation are obtained from a stoichiometric balance for each chemical element involved in the process. Further, for energy analysis, the first law of thermodynamics has been conducted by considering energy rate balances on a per mole of fuel basis for the engine control volume shown in Fig. 2 represented as the Eq. (1).

$$\frac{\dot{Q}_{C.V}}{\dot{n}_f} - \frac{\dot{W}_{C.V}}{\dot{n}_f} = \overline{h}_P - \overline{h}_R$$

Where $\dot{Q}_{C.V}$ and $\dot{W}_{C.V}$ represent the rate of heat transfer and power on the shaft (brake power) and $\dot{n}_f$ represents the molar flow rate of fuel. Also $h_P$ and $h_R$ are the specific molar enthalpies of the products and reactants. Moreover, the energy loss due to the exhaust gases can be calculated as the difference between the energy input rates and outputs of power and heat transfer. A detailed explanation of the method of calculation of this parameter can be found in Michael and Howard, [45] and Kotas [27]. Thirdly, the exergy rate balance for the engine control volume at a steady state should be written and solved as follows:

$$\dot{X}_{fuel} + \dot{X}_{air} - \dot{X}_{ex} - \dot{W}_{CV} - \sum \dot{Q}_{c.v}\left(1 - \frac{T_0}{T_j}\right) - \dot{X}_{dest} = 0$$

where$\dot{X}_{fuel}$, $\dot{X}_{air}$ and are the exergy transfer rate of fuel, air intake, and $X_{ex}$ exhaust gas and $\dot{W}_{C.V}$ (Output power) is equal to the exergy rate of work generated by the shaft. The fifth term represents the exergy that transfers heat from the engine control volume to the environment, where $T_j$ is the boundary temperature from where heat transfer occurs, and this is considered the cooling water temperature [45]. Finally, $\dot{X}_{dest}$ is the exergy destroyed in the control volume.

## 3. Results and discussion

In this section, the effects of engine speed on energy and exergy, energy and exergy analysis as well as engine emission analysis are discussed.

### 3.1. Energy vs engine speed

For the five fuel combinations in this study, changes in this parameter versus engine speed are shown in Fig. 3a. It is obvious that the fraction of output work energy has a parabolic trend for all types of fuel. This means that they experience an upward trend with increasing engine speed, but after hitting a peak between 1600 rpm and 1800 rpm, they see a significant fall. Higher mechanical friction loss and lower volumetric efficiency of the engine at higher speeds can be the main reasons for this trend as supported by other researchers [46]. Considering the type of fuel illustrates that the blends of (TD, TPD, TAD, and TAPD) all had higher thermal efficiency than that of pure diesel, and only at high speed did they all have roughly the same level. At the same time, the blends all reach a maximum at lower speeds than for diesel. This can be attributed to the higher cetane numbers for biodiesel blends and the blends having shorter ignition delays. Finally, the highest levels of thermal efficiency were achieved by TPD, TD, TAD, TAPD, and D, with 32.12 %, 31.75 %, 31.36 %, 31.27 %, and 30.8 %, respectively. This result indicates that the use of binary blends of biodiesel mixtures can improve the energy or thermal efficiency in comparison to a blending of single biodiesel and diesel and these improvements in energy efficiency can be attributed to changes in fuel specification parameters such as viscosity, cetane number, and oxygen content. It should be noted that lower energy and exergy rates can be found using the binary biodiesel blends (Nabi et al., 2018); [47], therefore it is a good suggestion to evaluate different biodiesel blends to find the best based on energy analysis. The thermal efficiency of a heat engine is the fraction of heat that is converted into work by the combustion of fuel. Furthermore, higher thermal efficiency means low specific fuel consumption and less fuel for a given power. Therefore, by using tomato biodiesel and its binary and ternary blends we can achieve higher power or energy by consuming less amount of fuel compared to pure diesel.

Fig. 3b compares the trend of heat loss energy for the five different fuels versus engine speed. As demonstrated in this graph, energy heat loss demonstrated a downward trend with increasing engine speed. It decreases on an average of 7 % for all fuel combinations. This possibly occurred because the fuel mixture or combustion product would have less time for heat transfer at a higher speed. Looking at the results for the five types of fuel investigated in this study shows that, although using a mixture of TD increases the percentage of heat loss energy, the use of binary or ternary blends of biodiesel can decrease this component of losses. The results in Fig. 3b also show that the heat loss energy varies from 47 % down to 41 % for TD over the tested engine speed range, while the equivalent outcome for D varies between 41 % and 39 %, with this outcome being because of an increase in fuel consumption and combustion energy. However, when the engine was fueled with TPD, this heat loss energy experienced a reduction in comparison to TD and D. Moreover, the result for the TAPD ternary blend showed a lower

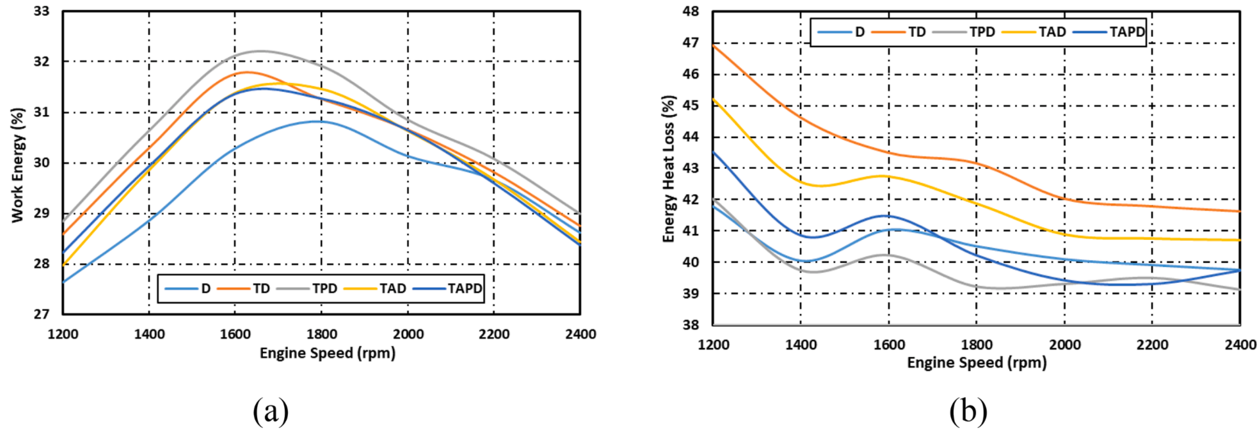



**Fig. 3.** (a) Output power and (b) Heat transfer energy versus engine speed for different fuels.

percentage of heat transfer energy loss compared to TD and D. To improve the thermal efficiency of an internal combustion engine, it is essential to reduce all losses including heat loss, exhaust loss, pumping loss, and friction loss. Particularly, a reduction in the heat loss from the in-cylinder gas to the combustion chamber walls is regarded as being an effective means of increasing not only the output energy but also the piston work, thus improving the thermal efficiency of an internal combustion engine. Therefore, by using the ternary blend we can reduce this main part of losses in comparison to pure diesel and its blending with tomato biodiesel. According to the thermodynamic models, the energy rate of heat transfer depends on several factors such as specific enthalpy of inlet mixture and combustion products, enthalpy change, and molar fraction of inlet fuel [45,48,49]. Therefore, predicting the trend of heat loss energy based on fuel type would be difficult because it is associated with various parameters. Hoseinpour et al. [50] stated that the fuel type does not show an apparent effect on the energy losses by heat transfer in the engine.

Fig. 4 shows that the fraction of exhaust gas emission, after reaching a peak at 1400 rpm, experienced a significant decrease at 1600 rpm which was followed by a steady increase for the rest of the studied speed range. All fuel combinations have their highest thermal efficiency at about 1400 rpm which is associated with completed combustion and a higher in-cylinder temperature and can lead to increasing exhaust gas emission while operating at high speeds resulting in a higher percentage of the products of uncompleted combustion which can result in increasing the fraction of wasted energy. Having the lowest ratio of exhaust gas energy at 1600 rpm could occur because of having the highest level of output work at about this point (it can be seen in Fig. 3a). Actually, there is a direct relationship between the ratio of exhaust gas

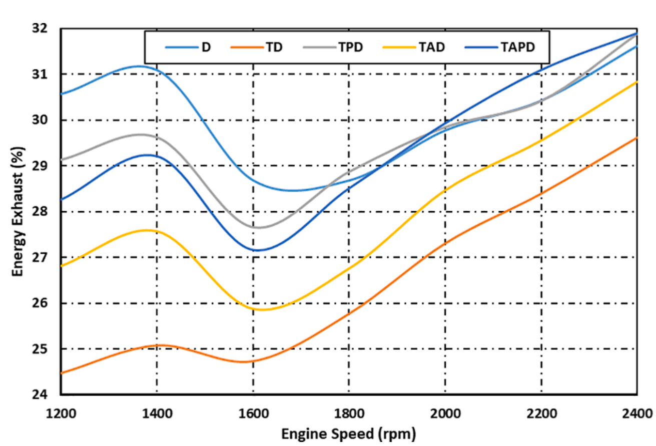



**Fig. 4.** Exhaust gas energy versus engine speed for different fuels.

energy and combustion product. More completed combustion would happen at this speed, thus less amount of energy wastes as exhaust gases leading to the lowest ratio of exhaust gas energy. Considering the type of fuel demonstrates that the highest fraction of energy wasted by exhaust gas emission was generally for pure diesel which could be because of having the highest levels of CO and HC in the combustion product as these have high enthalpy and play a significant role in the fraction of exhaust emission [51,54]. However, comparing results for TD with those for binary and ternary biodiesel blends shows that TD has a lower fraction in terms of exhaust emission. The main reason behind this is TD has a lower heat value than that of the other biodiesels and their blends. Generally, it can be seen that, in comparison with diesel, using TD and binary as well as ternary blends (TAD, TAPD, TPD) decreases this exhaust gas energy by 6 %, 3.5 %, 2 %, and 1.5 %, respectively.

### 3.2. *Exergy vs engine speed*

Fig. 5a shows output power exergy versus engine speed. Obtained results show that the results of exergy efficiency are similar to those of energy efficiency. Exergy efficiency increases with an increase in speed from 1600 rpm to 1800 rpm and then it experiences a considerable decrease. This means that, for all fuel blends, energy and exergy efficiency have a similar trend with different values, while exergy efficiency is lower than energy efficiency. The results are generally in good agreement with those of other researchers [50]. According to the type of fuel examined in this study, while the binary blend of TPD had the best performance based on exergy efficiency throughout this speed range, those of another binary blend (TAD) and a ternary blend (TAPD) were very similar to TD. These results can be because of the reasons explained earlier for energy efficiency. It is noteworthy that the exergy efficiency of all of the fuel blends (TD, TPD, TAD, TAPD) was generally higher than that of diesel. The maximum exergy efficiency of TPD, TD, TAD, TAPD and D was 29.63 %, 29.08 %, 28.97 %, 28.94 % and 28.46 %, respectively. These maximum efficiencies occurred at 1600 rpm except for D which was at 1800 rpm. Karagoz et al. [52] represented that using biodiesel blends and other additives improved the exergy efficiency of a CI engine.

The fraction of the heat transfer exergy for various fuel combinations at the examined range of engine speed has been shown in Fig. 5b. The results demonstrate that the exergy percentage of heat transfer for all the studied fuels reduces at 1400 rpm, which can be because, at this speed (the speed of maximum tuque), the engine produces the highest power with the least amount of specific fuel consumption. However, after 1400 rpm the variation in this power follows an erratic movement, and it does not actually demonstrate a clear picture. As explained in the previous section, the exergy rate of heat transfer depends on various factors, so they can have complicated effects on these parameters and result in this

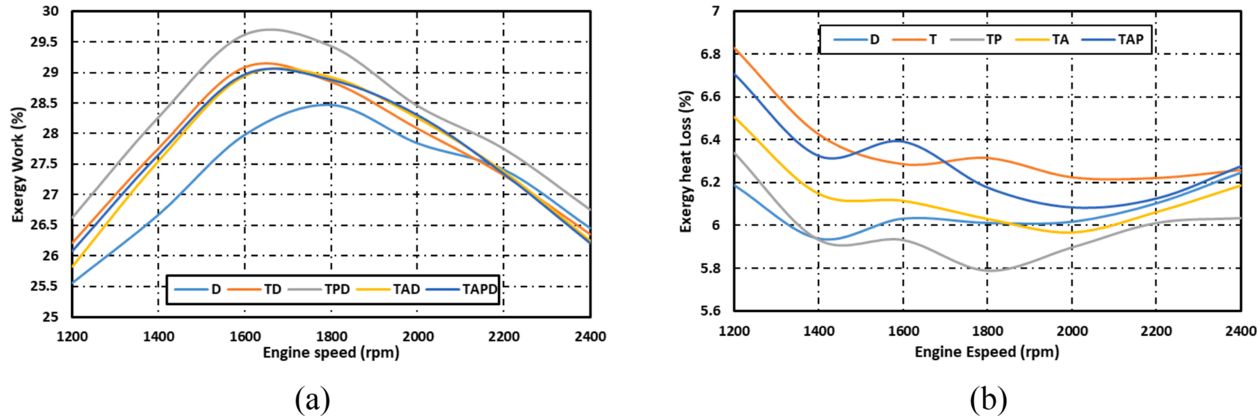



**Fig. 5.** (a) Output power and (b) heat transfer exergy versus engine speed for different fuels.

variable or unstable movement. This type of variable trend for heat loss exergy has also been stated by Iortyer and Bwonsi et al. (2017). Having regarding the type of fuel shows that, although in comparison with D, TAPD and TD had a higher fraction of heat loss exergy, the use of binary blends of TPD and TAD can reduce this heat loss exergy at high engine speed greater than for the pure diesel. For TPD this fall was on average 3 % at engine speed more than 1400 rpm, while TAD had a heat loss exergy a bit less or even at the same level as D. Also, the results obtained show that, compared to TD, using the binary blends decreases the percentage of heat loss exergy by 5.5 % and 3.3 % on average for TPD and TAD, respectively. Energy and exergy analyses were performed for a diesel engine using biodiesel, diesel, and bioethanol blends by Sayin Kul and Kahraman [54]. The results of this study represented that, although the exergy rate through heat transfer took its lowest value for D at low engine speeds, at high engine speeds it is not possible to determine which fuel blend had the lowest value of this factor.

Variation of the exergy fraction of the exhaust gas versus speed is depicted in Fig. 6a. It indicates that the exergy percentage of the exhaust gas increases with speed increase on an average of 34.8 %. The higher exhaust gas temperature and fuel consumption due to increasing engine speed will raise the percentage of the exhaust gas exergy. At high speeds, power losses due to friction increase, and this led to greater fuel consumption for the specific torque and this corresponds to greater exergy being dissipated in the exhaust gas. Another point about this figure is that all fuel blends experience a slight decrease at 1600 rpm which can be because of having higher efficiency and more completed combustion at this point. Shadidi et al. [55] represented that, when a CI engine runs on petrodiesel and biodiesel blends containing mixed with a nano-catalyst, the fuel exergy and exhaust exergy increase with increasing engine speed. A comparison made between different fuel combinations demonstrates that, although at lower speed the use of binary and ternary blends slightly decreases the exergy percentage of the exhaust gas compared to pure diesel, the mixture of TD shows a lower level for exhaust gas exergy at all speeds. This was largely due to the lower heating value in cases of TD which, as mentioned in Table 1, is 9000 kJ/kg lower than that of diesel. Nabi et al. [47] showed that the energy and exergy were less with three biodiesel blends compared to the reference diesel. They stated that lower exergy and energy with the three biodiesel blends could be associated with less LHV of the biodiesel blends. Generally, based on this parameter, it can be seen that binary and ternary blends increase the fraction of exhaust gas exergy on average by 9.4 % compared to TD at all the examined speeds.

Fig. 6b illustrates the fraction of destructed exergy or the irreversibility versus engine speed. The results prove the trend of destructed exergy experienced a sharp decrease as engine speed increased to 1800 rpm and then remained roughly at the same level. This is due to the fact that exergy efficiency increases with an increase in speed from 1600 rpm to 1800 rpm. However, after these speeds, exergy efficiency decreases while exhaust gas follows an increasing trend, so this behavior can keep the destructed exergy at the same level at a higher speed. It has been reported by Meisami et al. [49] that, with an increase in load and speed, the in-cylinder gas temperature also increases and the exergy destruction consequently decreases. Moreover, it is indicated in the literature that exergy destruction due to the combustion process decreases as the temperature of the gasses in the engine cylinder increases [56,57]. In addition, having regard to fuel type shows that the mixture of TD did not have a clear effect on exergy losses or irreversibility and the binary blend of TAD had the same percentage for this, however, the binary blend of

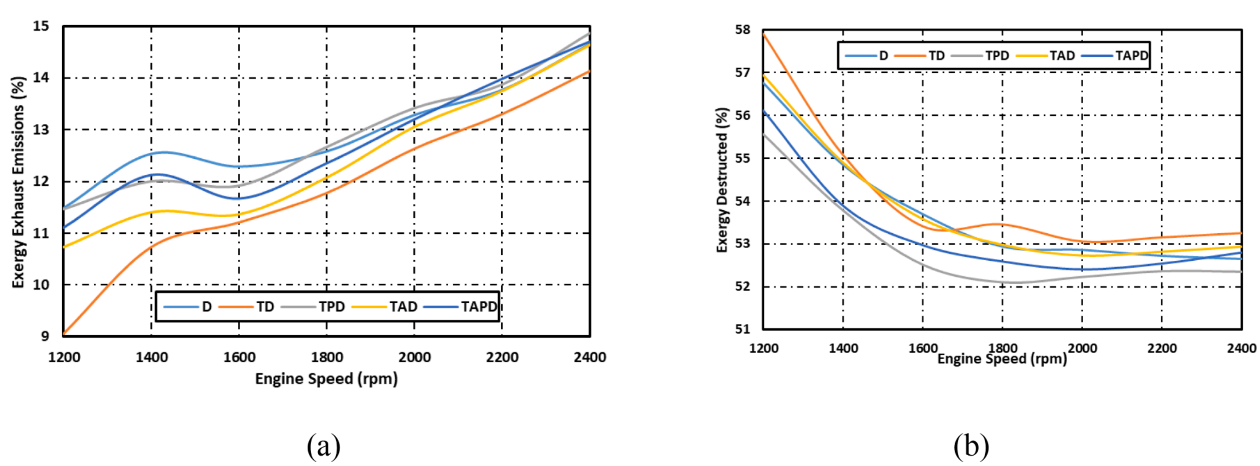



**Fig. 6.** (a) Exhaust gas emission and (b) destructed exergy versus engine speed for different fuels.

TPD and the ternary blend of TAPD reduced the fraction of irreversibility by approximately 7 % on average throughout the engine speed range compared to results for TD and pure diesel. Generally, the lowest value of the irreversibility at all engine speeds was for TPD. Therefore, the use of binary blends can reduce the level of wasted exergy which has the largest fraction in the exergy distribution. The obtained result shows that we can generally improve the performance of the engine using the binary and ternary blends instead of blending diesel with only one type of biodiesel. Therefore, by blending some type of oil or biodiesel we can achieve some remarkable results based on the performance of diesel engines. At the same time, biodiesel production from mixed oils can be considered a sustainable approach to industrial biofuel production. It can be suggested to work on this issue (evaluating the combination of other types of oils) to find the best oil blending base on energy and exergy performance.

Generally, fuel type does not show a clear effect on exergy losses or irreversibility, it can be because of the many parameters that affect irreversibility [50]. However, based on engine performance we should have less destructed exergy and this study shows that using the ternary blending of some biodiesel as fuel in a diesel engine can lead to this good indication.

### 3.3. Energy-Exergy analysis

The energy and exergy distribution for all fuels at engine speeds of 1400 and 2400 rpm is shown in Fig. 7. These are the speeds at which the maximum torque and power of the engine respectively occurred and play an important role in the assessment of the emission standard and engine performance. In this figure, the blue bars illustrate output power energy and exergy. A comparison between exergy and energy of output power for the five different fuels proves that the percentage of output power exergy (exergy efficiency) is slightly lower than the percentage of output power energy. Energy efficiency was 29.27 % on average while exergy efficiency was 26.90 %. In other words, the obtained results show that one-third of the total inlet energy (almost 30 %) at the two given speeds is transformed to output power and this was nearly-one-fourth of total exergy. According to its definition, exergetic efficiency is directly related to engine power and inversely related to fuel exergy. Since the chemical exergy of each inlet fuel type ($e_{fi}$) is always higher than the lower heating value (LHV), fuel exergy would be higher than fuel energy

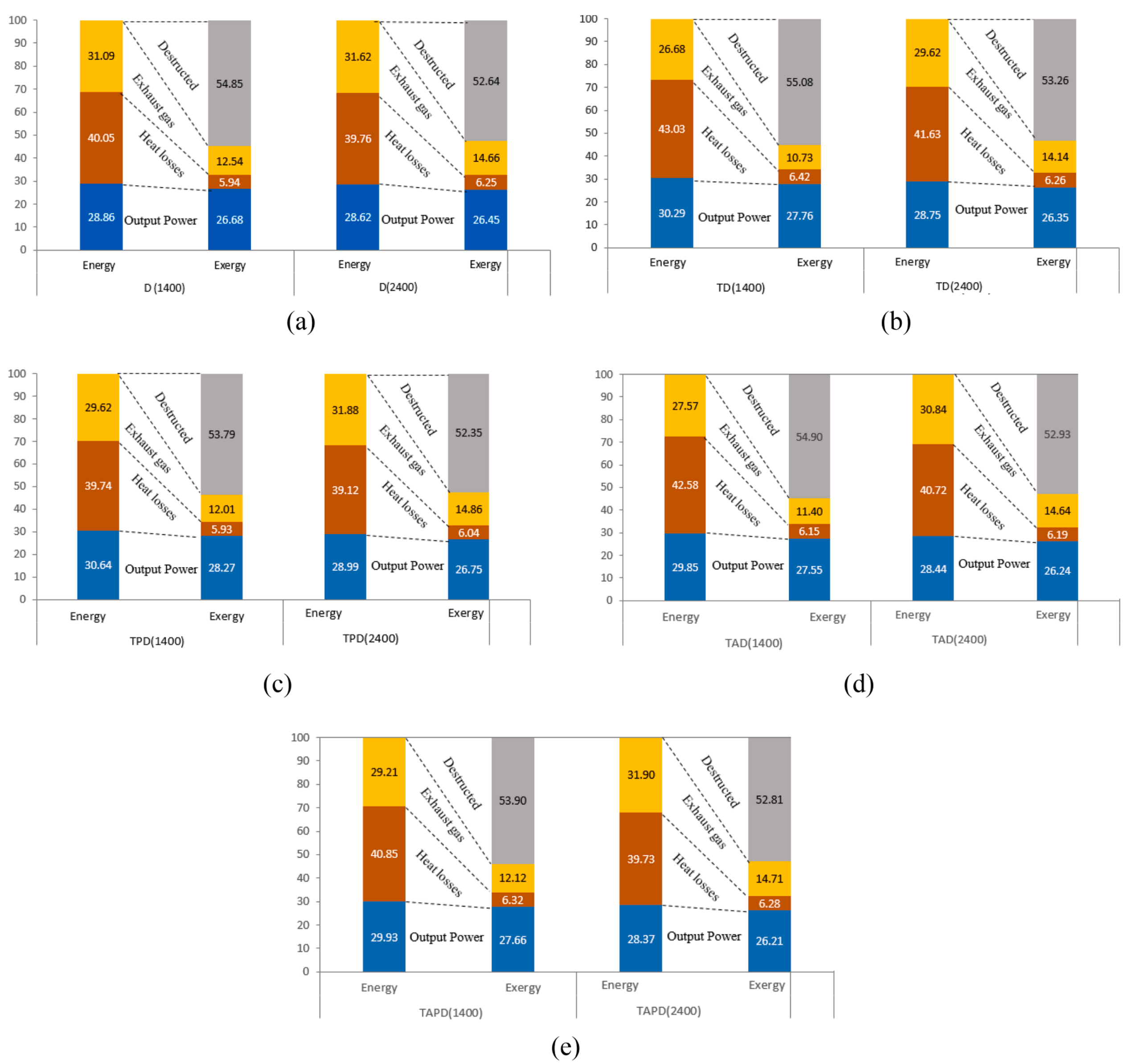



**Fig. 7.** Energy-exergy comparisons at 1400 rpm and 2400 rpm for (a) D, (b) TD, (c)TPD, (d) TAD, and (e) TAPD.

which can result in a lower exergy efficiency. Therefore, fuel quality or its availability always has a greater effect than its heating value. This result was proved by other research studies for various fuels under different working conditions [50,53]. Moreover, comparing this part of energy and exergy at the mentioned speeds shows that these fractions of output power are a bit higher for the speed of maximum torque for all the types of fuels studied, and it was about 1 % for pure diesel and nearly 5 % for TD and binary and ternary blends. This is because, at this speed, the engine which consumes the least amount of fuel can produce higher power. In addition, since biodiesels have a higher density than that of pure diesel, in a constant volume of fuel injection, more fuel enters the combustion chamber, thus increasing engine output power [19].

Brown bars in these graphs show the energy and exergy fraction of heat transfer. The obtained results indicate that about 40 % of the total energy for all fuel combinations was wasted as the heat transfer, while the fraction of heat transfer exergy was on average 6.2 % for all fuel types. Actually, the highest percentage of energy for all fuel combinations is wasted as heat transfer loss, but this part of exergy had the lowest fraction in the exergy distribution. Therefore, heat transfer exergy was considerably lower than heat transfer energy and it can be concluded that the main part of energy wasted by heat transfer is not capable of producing any work. Comparing this fraction at the two examined speeds shows that, although the amount of energy wasted as heat loss was a bit lower for the maximum speed of power, the percentage of heat loss exergy approximately had the same level at these speeds.

The orange bars in Fig. 7a depict the energy and exergy fractions of exhaust emission for different fuel combinations. It can be seen that the energy percentage of exhaust emission at the speed of maximum torque and maximum power was an average of 30 % while this fraction for exergy was about 13.12 %. It can be seen that the exergy fraction of exhaust gas decreased by 50 % in comparison with the energy fraction of energy wasted by exhaust emission. Also, the fraction of energy and exergy for the biodiesel blends wasted by exhaust emission at 2400 rpm on average increased by 9.9 % and 26.19 %, respectively, while those of pure diesel rose by 1.7 % and 16.9 %. The main reasons behind these results are having a higher exhaust gas temperature and lower combustion efficiency at a higher speed. Furthermore, comparing the fractions of heat transfer and exhaust emission shows that the percentage of exhaust gas energy is lower than that for heat transfer. For instance, regarding D at 1400 rpm, the results are 40.05 % for heat transfer and 31.09 % for exhaust emission, while their exergy percentages show an inverse behavior (5.94 % for heat transfer and 12.54 % for exhaust emission). So, the quality of energy which losses through exhaust gases is greater than that of heat transfer because exergy is defined as fuel quality or its availability.

Gray bars in Fig. 7a–e illustrate the percentage of destructed exergy. Apparently, it can be seen that the highest contribution in exergy distribution for the five fuel combinations at the two examined speeds was wasted by irreversibility or was for destructed exergy. It was at least half of inlet exergy (50 % or more). It can be concluded that the main part of exergy supplied to the engine is destructed due to irreversibility, while in the first law analysis and energy balance, losses are calculated only in the form of heat transfer and exhaust gas energy. This condition can result in having a lower exergy efficiency compared to energy efficiency. The performance values that were obtained agree well with those reported in the literature [31].

### 3.4. Emissions analysis

In using each fuel, increasing the engine speed from 1200 to 2400 rpm results in an increase in $CO_2$ emissions as shown in Fig. 8a. As illustrated in Table 1, the maximum torque occurs at 1400 rpm because of maximum performance efficiency. Furthermore, it is reported that higher efficiency in combustion leads to greater $CO_2$ emissions [58]. If the engine speed becomes more than 1400 rpm, the $CO_2$ emissions are reduced. An exception to this is a slight increase in $CO_2$ emissions from 1600 to 2400 rpm, probably due to a reduction in engine performance at maximum engine speed [59]. The figure shows that the $CO_2$ production of TD and TAD are very similar and are in the range of 1.5 to 2.5 % more than that of diesel, while TPD and TAPD increase $CO_2$ much less than do the other fuels, hence the average total $CO_2$ for all speeds of them only at about 0.25 % more than that of diesel. As can be seen in Table 3, TD and TAD have a higher density than that of other fuels. Moreover, density is mass per unit volume. If a fuel has a high density, it means much more mass is entering into the combustion chamber for the same volume, since new generation diesel injectors which supply fuel into the combustion chamber for power generation regulate the amount of fuel by volume, not by mass. Much more fuel entering into the engine cylinder means an increase in emissions. The increment of $CO_2$ emission values with biodiesel use can be explained by this reason [60].

Unanimous conclusions exist about the effects of engine speed on CO emissions. The higher the engine speed, the less the CO emission. The air/fuel ratio plays a key role in CO emissions. Reducing air in the fuel-air mixture causes unburnt fuel and poor combustion. In addition, one of the main reasons for CO production is the lack of a good mixing of air and fuel, as it causes the production of locally rich zones of fuel, where additional oxygen is needed to convert CO to $CO_2$ [61]. As demonstrated in Fig. 8b, CO emissions decrease in response to higher engine speeds, as is also observed regarding all the studied fuels. Moreover, CO drops sharply as engine speeds increase from 1200 to 1600 rpm, but then drops slightly as the engine speed goes from 1600 to 2400 rpm. This is because increasing the engine speed makes for better mixing of the air and fuel and optimizes the fuel/air equivalence ratio [62]. Furthermore, with an increase in engine speed, the turbulence of the mixture increases and its uniformity improves. Fig. 8b also shows that all biodiesel blends produce less carbon monoxide than that of pure diesel. The reduction of CO

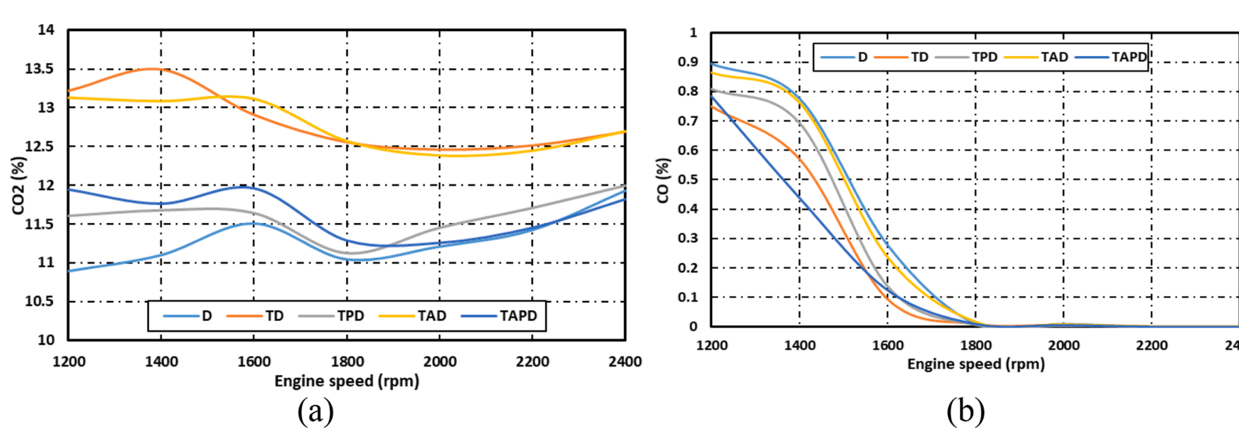



**Fig. 8.** (a) $CO_2$ and (b) CO versus engine speed for different fuels.

emissions probably happened because of the oxygen, which led to easier burning at higher temperatures in the cylinder. The minimum CO emission was related to TAPD. This is proved by the higher oxygen content in the shorter carbon chain length, which contributes to a cleaner and more complete combustion. Furthermore, there are methyl esters with longer chain lengths that have higher boiling and melting points, so they are less likely to be completely vaporized and burnt, thereby enhancing CO emissions. The shortest chain among all the biodiesel samples studied was obtained through TAPD ($C_{17}H_{32}O_2$ leading to a reverse connection with CO production [63,64].

Fig. 9a shows that pure diesel has the lowest NOx emissions. It is generally accepted that biodiesels increase the formation of NOx because of their higher combustion chamber temperature [65]. It is also seen an increase in engine speed up to 2200 rpm causes an increase in $NO_x$ to about 440 to 570 ppm, but this then reduces to about 420 to 560 ppm at 2400 rpm. The two factors which affect increases in NOx between speeds of maximum torque (1400 rpm) and maximum power are exhaust gas temperatures and the rise in volumetric efficiencies [66]. At speeds from 2200 to 2400 rpm, it can be said that this trend is primarily due to there being a short amount of available time for NOx formation, which could be a result of the increment in the volumetric efficiency and flow velocity of the reactant mixture at higher engine speeds [67]. As can be seen in this figure, the amount of NOx produced for tomato biodiesel is the highest. To justify this phenomenon, it can be said that, according to Table 3, tomato biodiesel has the highest viscosity and density compared to other fuels. Increasing these two factors increases NOx production. Fuel viscosity has a significant effect on NOx emissions. Generally, the kinematic viscosity of biodiesel is greater than that of diesel fuel, which reduces fuel leakage during injection and leads to increased pressure as well as advanced injection timing [68]. The advance in injection timing facilitates increased fuel mass injected which in turn results in increased NOx emissions. The NOx emissions increase with increasing fuel density as well as decreasing cetane number [69–71]. The start of injection, the injection pressure, and the fuel spray characteristics are affected by the fuel density, which influences combustion as well as emissions. As mentioned before, in modern diesel engines fuel injection systems measure the fuel by volume. As a result, changes in the fuel density will greatly act upon the mass of fuel injected and corresponding NOx emissions [70].

It is evident from the analysis presented in Fig. 9b that combustion of the fuel with a proportion of biodiesel reduces the HC emissions under all conditions due to the oxygen which is contained in the biodiesel which thus increases the fuel combustion efficiency Shirneshan and Hosseinzadeh Samani [72]. As is shown in Fig. 9b, HC emissions decrease up to 15–20 % with an increase in engine speed due to better atomization and the swirling condition which improves the effect of mixing fuel and air molecules in the cylinder. These conditions can make the fuel mixture more homogeneous and reduce HC emissions [73]. The oxygen molecules in the structure of the cooking oil methyl esters help to provide better combustion conditions. This could be an important reason for less production of unburned hydrocarbon for biodiesel fuel blends [74,75]. Moreover, the cetane number of the fuel blend is improved by the higher proportion of biodiesel in the mixture, and it enhances the efficiency of combustion and then reduces the emission of HC pollutants [73]. From Table 3 it can be seen that, although tomato biodiesel has a lower percentage of oxygen content than that of the other fuels, it has the highest viscosity which, in turn, causes a larger amount of mass of fuel to enter the combustion chamber and gives better combustion quality, hence less HC is produced. From that table, it can also be seen that TAPD has a low volume percentage of oxygen and a relatively low density, which causes it to reduce HC to a lesser extent than other fuels. Another reason which might be mentioned for TAPD is that some researchers believe that, due to the weak substructure and the low volatility of biodiesels, HC emissions increase [76]. However, the HC emissions were boosted up at least 10 to 12 % when Apricot or Papaya was added to the tomato biodiesel fuel separately. This condition is a result of the higher cetane index of Apricot or Papaya compared to diesel and tomato biodiesel which causes a retarded combustion phase; the combustion temperature is therefore reduced under Apricot or Papaya combustion and the HC emissions thus increase [77]. This issue is related to incomplete combustion conditions during the stage of the power stroke. Moreover, the larger diffusion phase with diesel compared to Apricot or Papaya causes a reduction in HC emission [78].

In summarizing, it is noteworthy that, in evaluating the performance of a thermal engine fueled by different types of fuels or working under different operational systems, assessment of exergy distribution is significant because it can provide the researcher with a clear picture of the energy conversion processes and suggest the best way for reducing emissions, energy losses or performance optimization.

## 4. Conclusions

Although biodiesels are good alternatives to fossil fuels, some of their properties have adverse effects on engine combustion, which may increase emissions and reduce engine efficiency. These effects can be minimized by mixing biodiesel with different characteristics. Energy and exergy distribution can be the best parameters to compare performance. To calculate the energy and exergy distribution of the different fuels, several tools have been used in different processes. Considering the fuel type demonstrates that, at the maximum torque, the highest fraction of energy wasted by exhaust gas emission was generally for pure diesel at about 12.54 % while it is at about 10.73 % for TD. A comparison

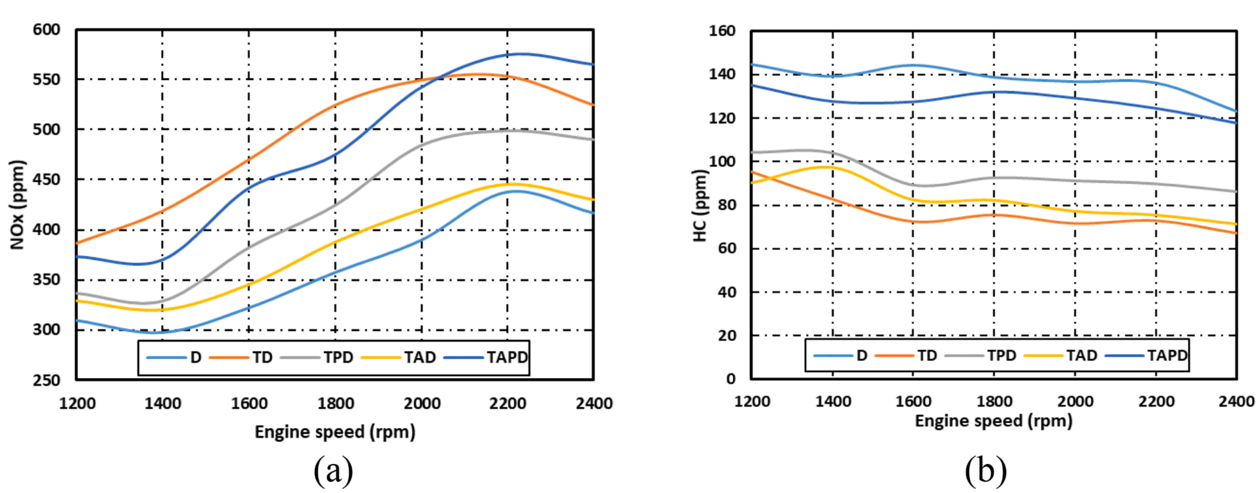



**Fig. 9.** (a) NOx and (b) HC versus engine speed for different fuels.

between exergy and energy of output power for the five different fuels studied proves that the percentage of output power exergy is about 2 % lower than the percentage of output power energy. Hence that energy efficiency was found to be 29.27 % on average while exergy efficiency was 26.90 %. The use of a dual biodiesel TPD blend was able to 1.59 % improve the engine output power efficiency percentage compared to pure diesel. Although the highest energy loss is in heat transfer, the highest exergy loss is in the exhaust fumes and research on reducing exergy losses of exhaust gases can increase the engine efficiency and better-applied results can be obtained to increase engine exergy efficiency.

## Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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