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Assessment of waste tyre pyrolysis oil and oxy-hydrogen gas usage in a diesel engine in terms of energy, exergy, environmental, and enviroeconomic perspectives

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ABSTRACT

Nowadays, alternative fuel research is increasing day by day due to the restrictions imposed on internal combustion engines. Using waste tyres as fuel will positively contribute to the national economy and environmental impact. Using oxy-hydrogen (HHO) gas, will help improve engine performance and exhaust emissions. This study focuses on the energy, exergy, environmental, and enviroeconomic impacts of using diesel, WTPO, and HHO gas. Experiments were carried out at constant 3000 rpm with torque values of 1.6 Nm, 3.2 Nm, 6.4 Nm, 7.9 Nm, and 12.8 Nm. The experiments were initially conducted with 100% base diesel fuel, followed by tests with diesel/WTPO blends (various proportions of WTPO at 20%, 40%, and 60% by volume), diesel-HHO gas (at different flow rates of 5 L/min and 10 L/min) dual-fuel application, and finally, diesel/WTPO-HHO dual-fuel application. When the results are examined, the highest first (approximately average 24% increase) and second law efficiencies (approximately average 13% increase) compared to basic diesel fuel were achieved with Diesel+10 L/min HHO dual fuel. The addition of WTPO to diesel fuel caused a decrease in both first and second-law efficiencies; however, using a Diesel/WTPO blend with HHO gas contributed to a maximum average increase of approximately 17% and 7% in first and second-law efficiencies, respectively. Furthermore, the maximum reduction in exhaust energy (approximately an average 12%) was achieved with a diesel blend containing 60% WTPO fuel by volume. Adding WTPO fuel to diesel fuel has contributed to a decrease in environmental and enviroeconomic impact. The maximum decrease in environmental and enviroeconomic impact (averaging 9%) was achieved with a diesel blend containing 20% WTPO fuel by volume.

1. Introduction

The global demand for energy and the need to mitigate environmental impacts have driven significant interest in alternative fuels for internal combustion engines. Diesel engines, known for their high thermal efficiency and durability, are widely used in various sectors, including transportation, agriculture, and power generation. However, their reliance on petroleum-based fuels contributes to greenhouse gas emissions and air pollution, which are major concerns in climate change and public health [1–5].

Waste tyre pyrolysis oil (WTPO) has emerged as a promising alternative fuel due to its potential to convert waste tyres, a significant environmental hazard, into valuable energy resources. Waste tire pyrolysis is a process that converts used tires into valuable products such as oil, gas, and char. This method is gaining attention as a sustainable solution to manage the increasing volume of waste tires while producing alternative fuels [6]. The oil obtained from tire pyrolysis, known as Tyre Pyrolysis Oil (TPO), has properties similar to traditional fuels but usually requires further processing to reduce impurities [7]. Additionally, integrating HHO (hydroxide gas) into the pyrolysis process is being investigated to improve the quality and efficiency of the fuels produced [8]. Pyrolysis of waste tires typically yields 40–60% liquid oil, 10–20% gas, and 20–35% char, with the exact ratios being affected by the reactor type, temperature, and heating rate [9]. WTPO contains alkanes and aromatic hydrocarbons but also contains contaminants such as sulfides,

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Nomenclature

$\dot{E}_{exh.}$	Exhaust energy (kW)
$\dot{E}_{fuel}$	Fuel energy (kW)
$\dot{E}_{lost}$	Lost energy (kW)
EE_{CO2}	Environmental impact of CO_2 (tons CO_2/year)
EC_{CO2}	Enviroeconomic impact of CO_2 (Euro/year)
ec_{CO2}	Price of one ton of CO_2 (Euro tons/CO_2)
$\dot{m}_{air}$	Mass flow rate of air (kg/h)
$\dot{m}_{exh.}$	Mass flow rate of exhaust (kg/h)
$\dot{m}_{fuel}$	Mass flow rate of fuel (kg/h)
Q_{fuel}	Lower heating value (kJ/kg)
S_x	Entropy production (kW/K)
T_{exh}	Exhaust temperature (K)
T_0	Ambient temperature (K)
$T_{surface}$	Surface temperature of the engine (K)
t_{op}	Engine's operating time (hours/year)
$\dot{W}_e$	Net engine power (kW)
$\dot{X}_{dest.}$	Exergy destruction (kW)
$\dot{X}_{exh.}$	Exhaust exergy (kW)
$\dot{X}_{fuel}$	Fuel exergy (kW)
$\dot{X}_{lost}$	Lost exergy (kW)
$\dot{X}_{\dot{W}_e}$	Net engine power (kW)
$\dot{X}_{II}$	Second law efficiency
x_{tm}	Thermomechanical specific exergy
x_{ch}	Chemical specific exergy
ε_{fuel}	Chemical exergy factor
η_{first}	First law efficiency
τ	Torque (Nm)

Abbreviations

BSEC	Brake specific energy consumption
C	Carbon
CO	Carbon monoxide
CO_2	Carbon dioxide
D	Diesel
DHHO5	Diesel fuel + 5 L/min HHO gas
DHHO10	Diesel fuel + 10 L/min HHO gas
DW20	20% Waste tyre pyrolysis oil + 80% Diesel fuel
DW40	40% Waste tyre pyrolysis oil + 60% Diesel fuel
DW60	60% Waste tyre pyrolysis oil + 40% Diesel fuel
GHG	Greenhouse gas
HC	Hydrocarbon
HHO	Oxy-hydrogen
ICE	Internal combustion engine
LHV	Lower heating value (kJ/kg)
MFR	Mass flow rate (kg/h)
N	Engine speed (1/min)
NO_x	Nitrous oxide
SI	Sustainability index
WTPO	Waste tyre pyrolysis oil

nitrides, and polyaromatic hydrocarbons (PAHs). These contaminants contribute to poor fuel properties such as low cetane number, low flash point, and high density [10]. The addition of n-butanol (a similar pyrolysis product) to waste plastic oil has been found to reduce thermal efficiency and increase hydrocarbon (HC) and carbon monoxide (CO) emissions, especially at low engine loads. Optimization of blend ratios is crucial to balance performance and emissions [11]. TPO has a high calorific value (approximately 40–44 MJ/kg), making it suitable for use as a boiler and furnace fuel, but it cannot be used directly in commercial automobile engines due to high emissions and impurities [7]. Pyrolysis offers a sustainable method for waste tire disposal, reducing environmental pollution and providing an alternative fuel source [12]. Waste tire pyrolysis is a promising technology for converting waste tires into valuable fuels and by-products. The process produces significant amounts of oil with high calorific value but requires further processing to reduce impurities for wider applications. This technology's environmental and economic benefits make it a viable solution for sustainable management of waste tires. The oil produced is chemically complex, containing aliphatic, aromatic and hetero atom fractions, and has properties similar to kerosene or light fuel oil [13]. WTPO possesses properties similar to conventional diesel, making it a viable candidate for use in diesel engines without substantial modifications [14–17]. Tyre pyrolysis oil (TPO) has been successfully burned in test furnaces and engines, showing similar power output and combustion efficiency to diesel fuel but with higher emissions (NO_x, CO, SO_x, HC) [6,7,18]. TPO can be used as a fuel additive or blended with other fuels to improve their properties and reduce emissions [18,19]. WTPO-diesel blends show similar performance to conventional diesel in terms of torque and power output, indicating that WTPO can be used efficiently in diesel engines without any modifications [20]. Studies have shown that WTPO can achieve comparable performance to diesel fuel while reducing the volume of waste tyres and associated disposal issues [20,21]. Additionally, using WTPO can contribute to a circular economy by recycling waste materials into valuable products, aligning with global sustainability goals [22]. Alongside WTPO, integrating hydrogen or hydrogen-rich gases, such as oxy-hydrogen, offers further opportunities to enhance engine performance and reduce emissions. Hydrogen, with its high energy content and clean-burning characteristics, can improve combustion efficiency and reduce pollutants such as carbon monoxide (CO) and unburned hydrocarbons (HC) [23–26]. However, previous studies have noted that hydrogen addition can lead to increased nitrogen oxide (NO_x) emissions due to higher combustion temperatures [27–31]. This necessitates a balanced approach to fuel formulation that leverages the benefits of hydrogen while mitigating its drawbacks. The researchers basically stated that HHO gas is an important factor in increasing the combustion efficiency of tyre oils obtained as a result of pyrolysis and it is useful in reducing fuel consumption values [32]. Another study focussed on the use of HHO and biogas mixtures together. It has been stated that the use of waste tyre oils added to diesel fuel at the rate of 20% by volume with biogas and HHO gases supplied from the intake manifold increases NOx emissions, reduces fuel consumption and increases thermal efficiency [33]. In addition, it is stated that the addition of HHO from the intake manifold is effective on the combustion efficiency, decreases the amount of fuel consumption and reduces emissions in the use of pyrolytic oils with low thermal degree or in the engine where combustion in the engine causes various problems [34–37].

The use of hydroxy gas (HHO) as an additional fuel in internal combustion engines has been investigated to improve engine performance and reduce exhaust emissions. HHO, produced by water electrolysis, consists of hydrogen and oxygen and is known for its strong oxidizing properties. The addition of HHO generally reduces carbon monoxide (CO), hydrocarbons (HC), and particulate matter (PM) emissions in diesel engines [38–40]. HHO also reduces CO and HC emissions in gasoline engines, providing up to 20% reduction for CO and 50% reduction for NO_x [41]. While HHO reduces CO and HC emissions, it often increases NO_x emissions in diesel engines due to higher combustion temperatures [42]. In gasoline engines, HHO can reduce NOx emissions by about 50% [41]. HHO can improve brake thermal efficiency and reduce specific fuel consumption in diesel and gasoline engines [43,44]. The addition of HHO contributes to better combustion characteristics by improving cylinder pressure and heat release rates [45]. Integration of HHO gas into internal combustion engines generally reduces harmful emissions such as CO, HC and PM, while increasing NO_x emissions. Adding HHO generally improves engine performance measures such as brake thermal efficiency and specific fuel consumption. However, HHO flow rates must be carefully managed to avoid adverse effects on volumetric efficiency at low engine speeds. Overall, research results indicate that HHO holds promise as an additional fuel to improve engine performance and reduce unavoidable emissions. Recent research has explored various blends and dual-fuel strategies to optimize engine performance and emissions. For example, studies using Fischer-Tropsch fuels have demonstrated that low-cetane number fuels can enhance thermal efficiency and reduce exhaust emissions in diesel engines when combined with exhaust gas recirculation (EGR) systems [46,47]. Similarly, biodiesel blends have been reported to reduce CO, total hydrocarbons (THC), and NO_x emissions while increasing brake thermal efficiency (BTE) and lowering brake-specific fuel consumption (BSFC) [48–54]. These findings suggest that combining alternative fuels with advanced combustion strategies can provide a path toward cleaner and more efficient diesel engine operation.

The present study investigates WTPO and oxy-hydrogen gas use in a diesel engine from an energy, exergy, environmental, and enviroeconomic perspective. By examining the combined effects of these alternative fuels, this research aims to provide a comprehensive understanding of their potential to enhance engine performance while minimizing environmental impact. This study is innovative in its comprehensive evaluation of the combined use of waste tyre pyrolysis oil (WTPO) with oxy-hydrogen gas (HHO) in a diesel engine by considering multiple perspectives, including energy, exergy, environmental, and environmental economic. Unlike previous studies that focused on the separate use of WTPO or HHO, this research investigates the synergistic effects of their combined use on engine performance and emissions. By integrating these two alternative fuels, the study not only investigates the improvements in combustion efficiency and reductions in harmful emissions but also provides a detailed exergy analysis that provides deeper insights into the energy degradation and losses in the engine system. In addition, including an environmental economic analysis that evaluates the economic impacts of emission reductions adds a unique dimension to the study by providing a holistic approach that relates technical performance to environmental and economic impacts. This multi-faceted analysis distinguishes the present study from previous studies and provides valuable insights into the sustainable use of alternative fuels in diesel engines.

The innovative aspect of this study lies in the comprehensive evaluation of waste tyre pyrolysis oil (WTPO) and oxy-hydrogen (HHO) gas as dual fuel in diesel engine, filling a critical gap in the literature. While previous studies focused on individual alternative fuels or dual fuel applications, this study is pioneering in combining WTPO and HHO gas with diesel to explore their synergistic effects from energy, exergy, environmental and enviroeconomic perspectives. By integrating first and second law efficiencies into a holistic environmental impact assessment, the study provides a novel, multi-dimensional analysis highlighting the potential to optimize diesel engine performance while addressing environmental concerns. This approach advances the understanding of alternative fuel applications and offers an innovative dual-fuel strategy that utilizes waste-derived and green energy sources and contributes to the future of sustainable internal combustion engine technologies.

Table 1
Properties of fuel mixtures used in experiments.

Fuel Mixture	Density (kg/m^3) at 15 °C	Fire point °C	Flash point °C	Kinematic viscosity (cSt) at 40 °C	Calorific value (MJ/kg)
Diesel (D)	840	67	55	3.38	42.5
WTPO (W)	975	55	78	26.4	44.45
DW20	918	60	63	11.8	43.3
DW40	918	63	63	11.9	43.25
DW60	921	66	65	12.1	43.21

Table 2
Test matrix, fuel mixture ratios and abbreviations were used in the experiments.

Engine load (Nm)	Engine speed (rpm)	Fuel Mixture	Diesel (D) (%)	WTPO (W) (%)	HHO (lt/min)
1.6, 3.2, 6.4, 7.9, and 12.8	3000	D	100	0	0
		DW20	80	20	0
		DW40	60	40	0
		DW60	40	60	0
		DHHO5	100	0	5
		DHHO10	100	0	10
		DW20HHO5	80	20	5
		DW20HHO10	80	20	10
		DW40HHO5	60	40	5
		DW40HHO10	60	40	10
		DW60HHO5	40	60	5
		DW60HHO10	40	60	10

2. Material and methods

2.1. Fuel preparation

The mixture of WTPO and diesel fuel used in the engine experiments was obtained from a local dealer selling the products commercially. The purchased WTPO was first passed through a coarse filter. After that, it was heated in a heater with a magnetic stirrer at a temperature of 70 °C and pressurised through a diesel filter and all particles remaining in it were removed. Fuel mixtures were prepared volumetrically. For this purpose, pre-engine experiments were carried out to determine the blend ratios. Table 1 shows the ratios, abbreviations in the graphs and physical properties of the fuel blends prepared for the experiments.

Detailed data on fuel mixture ratios and engine parameters determined based on the data obtained as a result of preliminary experiments are given in Table 2.

The HHO generator used in the engine experiments was purchased from UCR, a company that commercially assembles automobiles. The general structure of the HHO generator is shown in Fig. 1. Detailed data on the HHO generator are shown in Table 3.

The production capacity of the HHO generator varies with current. This relationship means that the amount of HHO gas produced can be increased or decreased by adjusting the electrical current supplied to the generator. Such adjustments are typically made via the control panel, which allows operators to fine-tune current levels to the desired output (Fig. 1). However, additional components are integrated in the experimental setup to ensure both safety and precision in the operation of the system. In particular, a precision natural gas valve was installed to accurately control the gas flow and a flow brand flow meter was installed to measure the flow. This flowmeter plays a crucial role in regulating the gas flow directed to the intake manifold, allowing the correct amount of HHO gas to be delivered. By utilising these components, the system can maintain optimum performance while avoiding potential hazards associated with improper gas flow. This careful regulation is essential to ensure efficient and safe operation of the HHO generator in a variety of applications. A spark control valve is also installed along the line to prevent sparks from entering the intake manifold.

During operation, HHO gas is used as dual fuel at constant mass flow rates of 5 lt/min and 10 lt/min, respectively, across all loads. The energy ratio of the mass-controlled HHO gas in dual fuel mode for fuel mixture

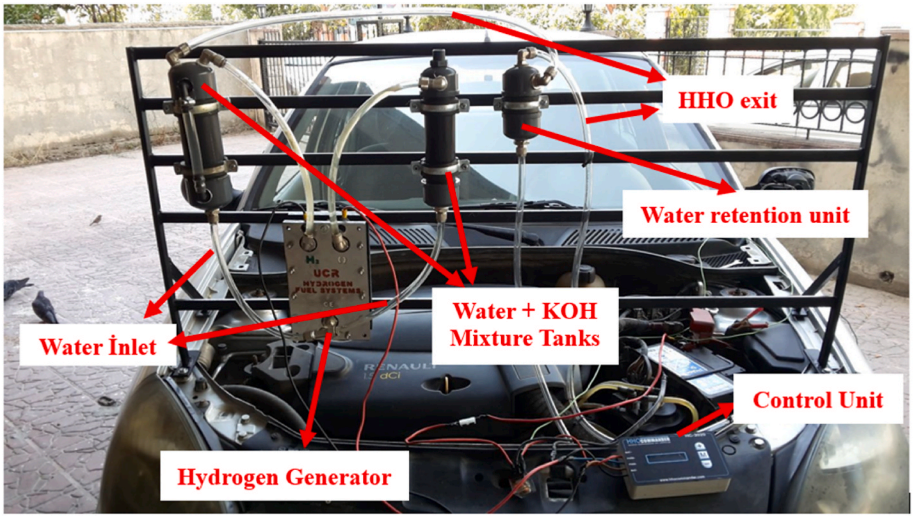

Fig. 1. UCR Brand Hydrogen Generator system and components.

Table 3
HHO generator specifications.

Property	Unit (Description)	Value
Maximum production	L/min	30
Electrodes (Anode-cathode)	Titanium Alloy	60 (cm) * 40 (cm)
Maximum voltage	V	24
Maximum current	A	30
Reactor volume	L	2.3
Maximum operating temperature	^{o}C	65

and varying loads is presented in Table 4. Using reference [55] to calculate the energy contribution of HHO gas in the WTPO + HHO dual fuel application, the density of HHO is taken as 0.538 kg/m^3 and its LHV is taken as 13328 kJ/kg. In addition, the LHV of the dual fuel (LHV_{DF}) was used for energy calculation in the WTPO + HHO dual fuel application. The LHV_{DF} is presented in Eq. (1) [56]. Here, m represents the mass of gas and liquid fuel.

$$LHV_{DF} = \left(\frac{m_{HHO}}{m_{HHO} + m_{WTPO}}\right) x\ LHV_{HHO} + \left(1 - \frac{m_{HHO}}{m_{HHO} + m_{WTPO}}\right) x\ LHV_{WTPO} \tag{1}$$

2.2. Experimental setup

The experiments of diesel-WTPO-HHO fuel mixtures were performed on a single-cylinder, four-stroke, naturally aspirated 186 FAG model diesel engine. Technical specifications of the specified engine are presented in Table 5.

The mixing chamber is positioned 40 cm before the intake manifold of the engine to ensure optimum mixing of the HHO + air mixture. Thus, there is a continuous supply of HHO + air mixed fuel behind the intake valves of the engine. Emission values including CO, CO_2, HC, smoke and NOx were monitored in all engine experiments using a Mobydic Combi instrument and the measurement ranges are detailed in Table 6.

Fuel consumption values were measured in mass using a precision balance during the study. Variations in exhaust gas emissions were monitored in real-time with a K-type thermocouple. Fig. 2 illustrates the schematic view of the experimental setup.

Using high-accuracy measurement tools, combined with the rigorous uncertainty analysis provided by the Kline-McClintock method, ensured that the experimental data were precise and reliable. This meticulous approach to data accuracy allowed researchers to draw meaningful conclusions from the experiments and provided a solid foundation for further research and development in engine optimization and emissions control. By understanding and quantifying the total uncertainties, the researchers could identify the confidence levels of their experimental results and make informed decisions regarding the implications of their findings. This approach also highlighted potential areas for improvement in future experiments, guiding enhancements in measurement techniques and experimental design. The overall uncertainties of the experimental data were determined using the Kline-McClintock method [57,58]. The accuracy of the measurement tools and the total

Table 4
The energy ratio of the mass-controlled HHO gas in dual fuel mode.

	Contribution of HHO in terms of energy (%)				
	1.6 Nm	3.2 Nm	6.4 Nm	7.9 Nm	12.8 Nm
DHHO5	9.4	7.5	4.2	3.9	3.0
DHHO10	16.6	15.3	8.8	8.2	6.2
DW20HHO5	9.2	7.7	4.0	3.7	2.9
DW20HHO10	17.7	14.4	7.8	7.4	5.8
DW40HHO5	8.6	7.2	4.0	3.9	2.7
DW40HHO10	17,1	14.6	8.1	7.9	5.4
DW60HHO5	8.3	6.7	3.8	3.5	2.6
DW60HHO10	16.3	13.8	7.7	7.2	5.3

Table 5
Technical specifications of the 186 FAG model diesel engine.

Specification	Details
Bore x stroke	86 × 70 mm
Compression ratio	18:1
Total displacement	418 cm^3
Maximum power	7 kW (3600 rpm)
Cooling system	Air cooled
Intake system	Natutally-aspirated
Type of fuel injection	Direct-injection
Fuel delivery advance angle	21 ± 1 (°CA) BTDC
Pressure of injection	19.6 ± 0.49 Mpa
Injector nozzle number	4

Table 6
Technical properties of the devices used to measure exhaust emissions.

Emissions	Unit	Meas.	Range	Accuracy
HC	ppm	0	20000	±12
CO	%	0	10	±0.06
CO_2	%	0	20	±0.5
NO_x	ppm	0	5000	±5%
Smoke	%	0	20	±1

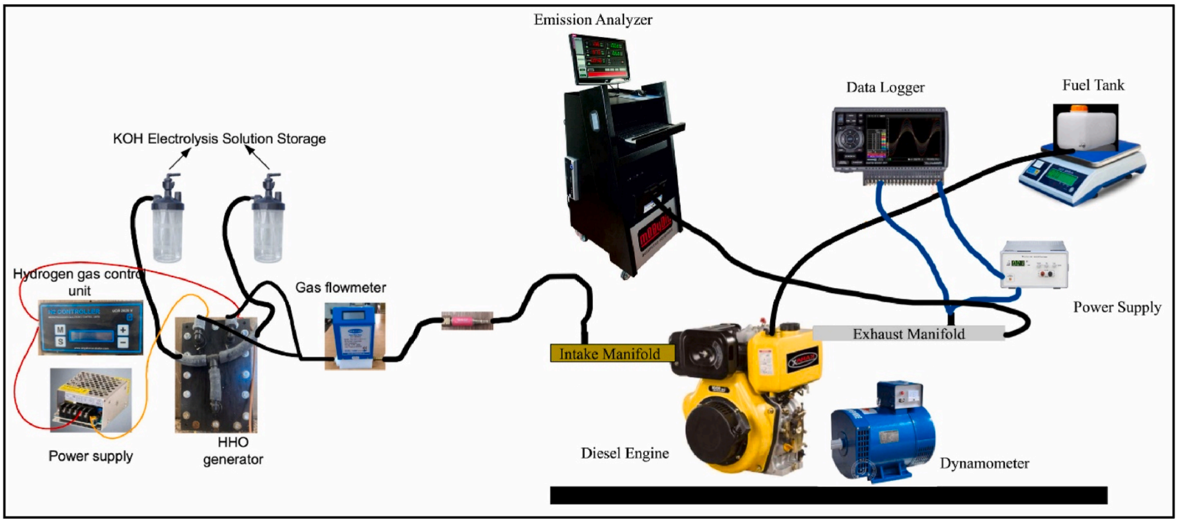

Fig. 2. Schematic visual of the engine and test setup.

uncertainties calculated are given in Table 7.

The diesel-WTPO-HHO gas experiments were conducted repeatedly for each fuel mixture, maintaining a constant shaft speed of 3000 rpm and varying engine loads at 1.6 Nm, 3.2 Nm, 6.4 Nm, 7.9 Nm, and 12.8 Nm. Prior to commencing the experiments, the engine underwent a warm-up phase using diesel fuel. Once warmed up, the pilot fuel was initiated. Subsequently, the liquid fuel supply to the engine was gradually reduced, while HHO gas was introduced into the engine. All measurements were recorded upon achieving the desired power output of the engine. The collected data were then analyzed and discussed in the following section.

3. Theoretical methodology

The present study focuses on the energy, exergy, sustainability, environmental, and enviroeconomic analyses of blends containing diesel, WTPO, and HHO fuels. Table 8 presents the parameters used for the analyses. The emission values included in the analyses are presented in Section 4.1.

Assumptions for energy and exergy analyses are outlined as follows [59–61]:

1. The engine runs in a state of steady operation.
2. The final products of combustion reach a state of chemical equilibrium.
3. The gases entering and exiting the system are assumed to be ideal gases.
4. The kinetic and potential energies of the masses (gas and liquid) entering and exiting the engine are disregarded.
5. It is presumed that the properties of the flow within the control volume remain constant throughout the duration of the analysis.
6. The reference ambient temperature is 301 K, and an ambient pressure of 101 kPa is assumed.

Table 7
Measurement tools accuracy and total uncertainties.

Parameters	Device	Accuracy
Time measurement	Digital chronometer, s	±1 %
Exhaust gas temperature	Thermocouple, °C	±1 %
Gas flow	Mass flow meter, l/min	±1 %
Engine speed	Incremental encoder, rpm	±1 %
Fuel mass	Precision scale, g	±0.1 %
Calculated Results		Uncertainty
BSEC		±1.54 %
Engine torque		±1.22 %
Power		±1.17 %

3.1. Energy analysis

The 1st thermodynamic's law asserts that the energy gained by a system during interaction equals the energy absorbed by the surroundings. This principle of energy balance is represented by Eq. (2) [58,62]. Also, Eq. (3) is derived from the energy balance (Eq. (2)) mentioned down applied to the ICE [63].

$$\sum E_{in} = \sum E_{out} \tag{2}$$

$$\dot{E}_{fuel} = \dot{W}_e + \dot{E}_{exh.} + \dot{E}_{lost} \tag{3}$$

Where, E_{in} represents the energy entering the system, while E_{out} represents the energy leaving the system. Additionally, $\dot{E}_{fuel}$, $\dot{W}_e$, $\dot{E}_{exh.}$, and $\dot{E}_{lost}$ are defined as fuel energy (kW), net engine power (kW), exhaust energy (kW), and lost energy (kW), respectively.

$$\dot{E}_{fuel} = \dot{m}_{fuel} \; x \; \mathcal{Q}_{fuel} \tag{4}$$

The product of the fuel mass flow rate (MFR) ($\dot{m}_{fuel}$) and its lower heating value ($\mathcal{Q}_{fuel}$) provides the $\dot{E}_{fuel}$, as seen in Eq. (4).

$$\dot{W}_e = 2\pi N\tau \tag{5}$$

In Eq. (5), which is the expanded form of $\dot{W}_e$, N and τ represent the engine speed and torque, respectively [56].

$$\dot{E}_{exh.} = \sum_{k=1}^{n} m_k x h_k = m_{CO} x h_{CO} + m_{CO2} x h_{CO2} + \ldots \tag{6}$$

Eq. (6) defines the expanded form of $\dot{E}_{exh.}$. Here, m_k and h_k denote the MFR and the specific enthalpy of exhaust products, respectively [61].

$$\eta_{thermal} = \dot{W}_e / \dot{E}_{fuel} \tag{7}$$

The ratio of the engine net power to the engine input energy gives the thermal efficiency ($\eta_{thermal}$) of the engine. The $\eta_{thermal}$ of the engine is calculated using Eq. (7) [64].

Table 8
Experimental results were used in the energy and exergy analysis of test fuels.

Test Fuels	Load (Nm)	$\dot{m}_f$ (kg/h)	$\dot{m}_{air}$ (kg/h)	T_o °C	T_{exh} °C	$T_{surface}$ °C
D	1.6	0.55	16.38	28	138	50
	3.2	0.67	16.33	28	189	55
	6.4	1.21	16.32	28	261	68
	7.9	1.36	16.12	28	440	101
	12.8	1.72	16.01	28	551	125
DW20	1.6	0.56	16.34	28	133	47
	3.2	0.68	16.28	28	187	53
	6.4	1.25	16.24	28	253	66
	7.9	1.38	16.13	28	436	100
	12.8	1.73	15.88	28	539	125
DW40	1.6	0.59	16.20	28	130	45
	3.2	0.70	16.16	28	185	51
	6.4	1.29	16.14	28	250	65
	7.9	1.41	16.01	28	432	99
	12.8	1.89	15.80	28	529	124
DW60	1.6	0.61	16.12	28	121	41
	3.2	0.75	16.03	28	162	48
	6.4	1.32	15.98	28	236	62
	7.9	1.43	15.89	28	416	95
	12.8	1.91	15.65	28	511	121
DHHO5	1.6	0.49	16.22	28	141	60
	3.2	0.62	16.19	28	192	68
	6.4	1.14	16.16	28	274	69
	7.9	1.24	15.99	28	451	121
	12.8	1.61	15.89	28	569	136
DHHO10	1.6	0.51	16.02	28	159	68
	3.2	0.56	15.93	28	212	71
	6.4	1.05	15.87	28	294	79
	7.9	1.13	15.74	28	468	135
	12.8	1.52	15.63	28	576	145
DW20HHO5	1.6	0.50	16.18	28	139	66
	3.2	0.61	16.06	28	192	70
	6.4	1.21	15.97	28	256	75
	7.9	1.33	15.87	28	444	130
	12.8	1.70	15.76	28	541	144
DW20HHO10	1.6	0.47	16.08	28	141	67
	3.2	0.60	16.01	28	199	71
	6.4	1.19	15.83	28	259	79
	7.9	1.27	15.76	28	453	131
	12.8	1.63	15.63	28	555	144
DW40HHO5	1.6	0.54	15.99	28	133	62
	3.2	0.65	15.84	28	188	70
	6.4	1.22	15.76	28	256	75
	7.9	1.23	15.67	28	436	126
	12.8	1.81	15.56	28	537	139
DW40HHO10	1.6	0.49	15.91	28	139	63
	3.2	0.59	15.80	28	192	72
	6.4	1.14	15.71	28	267	77
	7.9	1.18	15.61	28	439	129
	12.8	1.76	15.49	28	544	140
DW60HHO5	1.6	0.56	15.87	28	126	60
	3.2	0.70	15.78	28	168	70
	6.4	1.28	15.63	28	239	76
	7.9	1.38	15.59	28	420	126
	12.8	1.86	15.46	28	516	136
DW60HHO10	1.6	0.52	15.83	28	132	62
	3.2	0.63	15.71	28	174	79
	6.4	1.21	15.60	28	246	88
	7.9	1.31	15.52	28	429	129
	12.8	1.82	15.41	28	521	139

3.2. Exergy analysis

Exergy denotes the potential to obtain useful work from fuel energy while elucidating the energy degradation. The sum of the exergy leaving ($\dot{X}_{out}$) the system and the exergy destruction ($\dot{X}_{dest.}$) within the system is equal to the exergy entering ($\dot{X}_{in}$) the system, calculated using Eq. (8).

$$\dot{X}_{in} = \dot{X}_{out} + \dot{X}_{dest.} \tag{8}$$

$$\dot{X}_{fuel} = \dot{X}_{\dot{W}_e} + \dot{X}_{exh.} + \dot{X}_{lost} + \dot{X}_{dest.} \tag{9}$$

The expanded form of Eq. (8) is presented as Eq. (9). The $\dot{X}_{in}$ is defined as fuel exergy ($\dot{X}_{fuel}$). $\dot{X}_{fuel}$ comprises net engine power ($\dot{X}_{\dot{W}_e} = \dot{W}_e$), exhaust exergy ($\dot{X}_{exh.}$), lost exergy ($\dot{X}_{lost}$), and $\dot{X}_{dest.}$.

$$\dot{X}_{fuel} = \dot{m}_{fuel}\ x\ \mathcal{Q}_{fuel}\ x\ \varepsilon_{fuel} \tag{10}$$

Eq. (10) is the expanded form of $\dot{X}_{fuel}$, where ε_{fuel} represents the chemical exergy factor. The ε_{fuel} is calculated using Eq. (11) [65–67].

$$\varepsilon_{fuel} = 1.0401 + 0.1728\ HxC^{-1} + 0.0432\ OxC^{-1} + 0.2169\ \alpha xC^{-1}\left[1 - 2.0628HxC^{-1}\right] \tag{11}$$

The terms H, C, O, and α in Eq. (11) represent the mass fractions of hydrogen, mass fractions of carbon, mass fractions of oxygen, and sulfur of fuel, respectively.

$$\dot{X}_{exh.} = \sum_{k=1}^{n} \dot{m}_k x\left[x_{tm,k} + x_{ch,k}\right] \tag{12}$$

$$x_{tm,k} = \sum\nolimits_{k=1}^{n}\left[\left(\overline{h}_k - \overline{h}_{k,0}\right) - T_{amb.}\left(\overline{s}_k - \overline{s}_{k,0}\right)\right] \tag{13}$$

$$x_{ch,k} = \overline{R}T_{amb.}\sum\nolimits_{k=1}^{n}\left[ln\frac{x_{env,k}}{x_{exh,k}}\right] \tag{14}$$

$\dot{X}_{exh.}$ is a function of the MFR of exhaust gases released into the environment and the specific exergy of the exhaust products ($x_{tm} + x_{ch}$), and $\dot{X}_{exh.}$ can be calculated using Eq. (12). Where, x_{tm} and x_{ch} represent the thermomechanical and chemical-specific exergies of the components contained within the exhaust, respectively. Additionally, $x_{tm,k}$ can be calculated with Eq. (13), and $x_{ch,k}$ can be calculated with Eq. (14). Where, $\overline{h}$, $\overline{s}$, $T_{amb.}$, $\overline{R}$, x_{env}, and x_{exh} represent enthalpy, entropy, ambient temperature, specific gas constant, the molar fraction of component i in the reference surrounding, and the exhaust gas, respectively [68,69].

$$\dot{X}_{lost} = \sum \dot{E}_{lost}\ x\left[1 - \frac{T_{amb.}}{T_s}\right] \tag{15}$$

$\dot{X}_{lost}$ encompasses all heat transfer from the engine surface to the surroundings, and it can be calculated using Eq. (15) [70]. Here, T_s symbolizes the temperature of the engine surface.

$$\dot{X}_{II} = \frac{\dot{X}_{\dot{W}_e}}{\dot{X}_{fuel}} \tag{16}$$

The division of the net power of the engine by the exergy fuel ratio gives the second law efficiency ($\dot{X}_{II}$), which is calculated using Eq. (16) [71].

$$S_x = \frac{\dot{X}_{dest.}}{T_{amb.}} \tag{17}$$

Entropy production is a function of exergy destruction and can be calculated using Eq. (17) [72,73].

$$\text{Sustainability index (SI)} = \frac{1}{1 - \dot{X}_{II}} \tag{18}$$

The sustainability index (SI) can be calculated using Eq. (18) to determine the sustainability of diesel, WTPO, and HHO fuel blends [73–75].

3.3. Environmental and enviroeconomic analysis

In this paper, annual CO_2 emissions amounts were assessed through the testing of the diesel, WTPO, and HHO fuel blends in a diesel engine across various engine loads for environmental analysis. Eq. (19) can be utilized to calculate the quantity of CO_2 emissions discharged into the environment throughout the test engine's operational period [76,77].

$$EE_{CO2} = m_{CO2}\ x\ \dot{W}_e\ x\ t_{op} \tag{19}$$

Here, EE_{CO2}, m_{CO2}, and t_{op} respectively symbolize the amount of CO_2 emitted into the environment during the engine's operation (tons CO_2/year), the amount of CO_2 emitted per unit of engine power (tons CO_2/kWh), and the engine's operating time (hours/year). It is accepted that the engine used in the experimental study operates 330 days a year, with 8 h per day [76].

$$EC_{CO2} = ec_{CO2}\ x\ EE_{CO2} \tag{20}$$

Using Eq. (20), the environmental-economic analysis (EC_{CO2}, Euro/year) of the amount of CO_2 emitted into the environment during the engine's operation can be conducted [77]. In this equation, ec_{CO2} (Euro tons/CO_2) represents the price of one ton of CO_2, which is 62.47 Euros for the February 2024 [78]. EE_{CO2} is derived from Eq. (18).

4. Results and discussions

4.1. Emission analysis

Fig. 3 presents variations in HC, CO, and CO_2 emissions for diesel, WTPO, and HHO fuel blends at different loads. Gradually increasing the load from 1.6 Nm to 12.8 Nm for all test fuels decreased HC and CO emissions while causing an increase in CO_2 emissions. It is reported in the literature [79] that increasing engine load leads to increased thermal energy and cylinder temperatures, resulting in better combustion

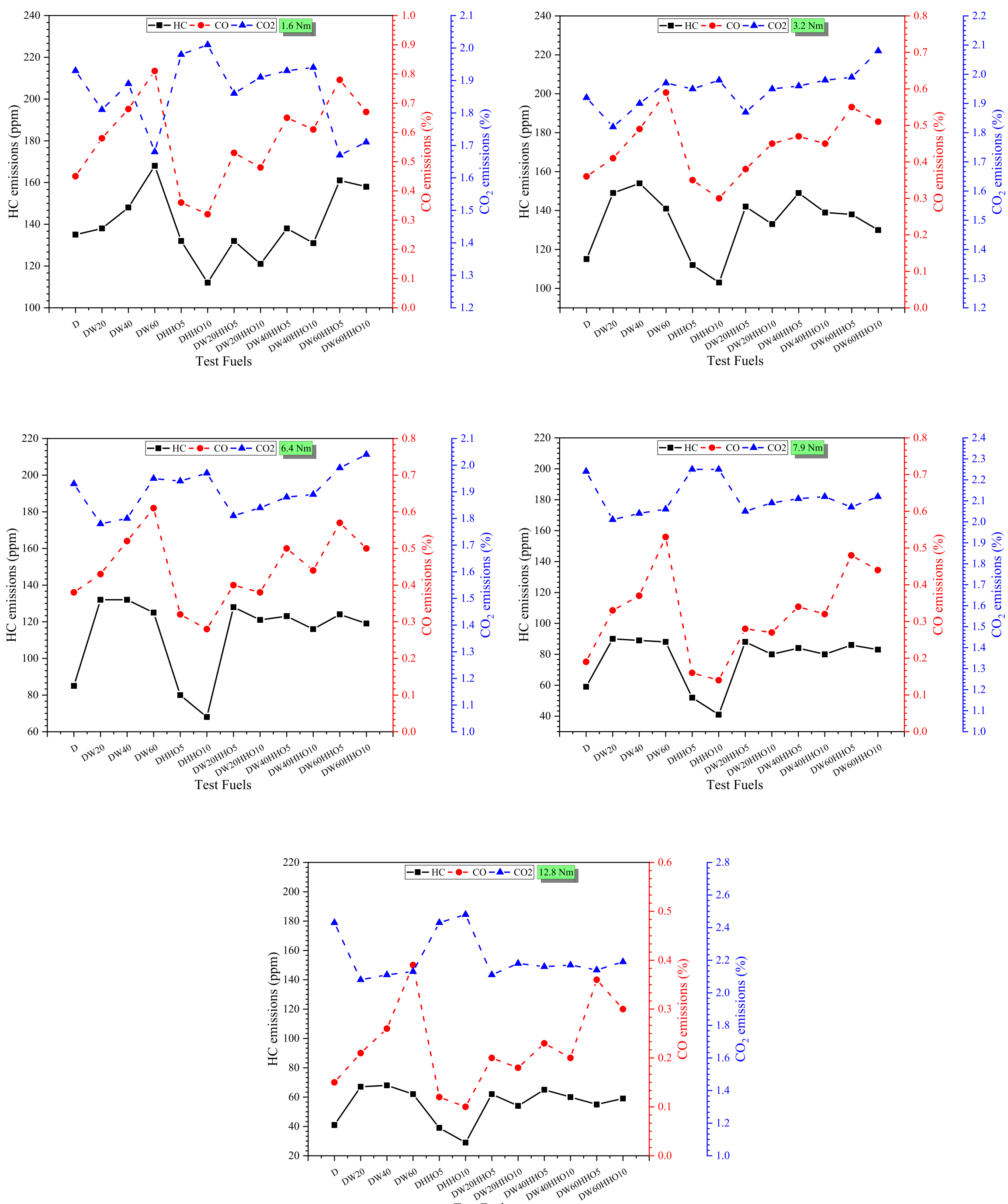

Fig. 3. Variations of HC, CO, and CO_2 emissions under different loads for diesel, WTPO, and HHO test fuels.

efficiency and, hence, reduced HC and CO emissions. Therefore, the decrease in these two emissions with increasing loads is consistent with the literature. In diesel engines, HC emissions may increase due to incomplete combustion, leakage from the exhaust valve, fuel properties, and non-homogeneous mixture [80]. The present study obtained the maximum HC emissions with the DW60 fuel (168 ppm) at an engine load of 1.6 Nm. The increase in HC emissions compared to diesel fuel is approximately 24%. Under low load conditions, a diesel engine exhibits lower combustion temperatures. Additionally, residual gas temperatures are also low. Diesel and WTPO fuels, which have high density, evaporate more slowly and to a lower extent at low temperatures. This results in decreased combustion stability and reduces combustion efficiency. Consequently, HC emissions increase. The minimum HC emissions at the same engine load were achieved with the DHHO10 fuel, resulting in

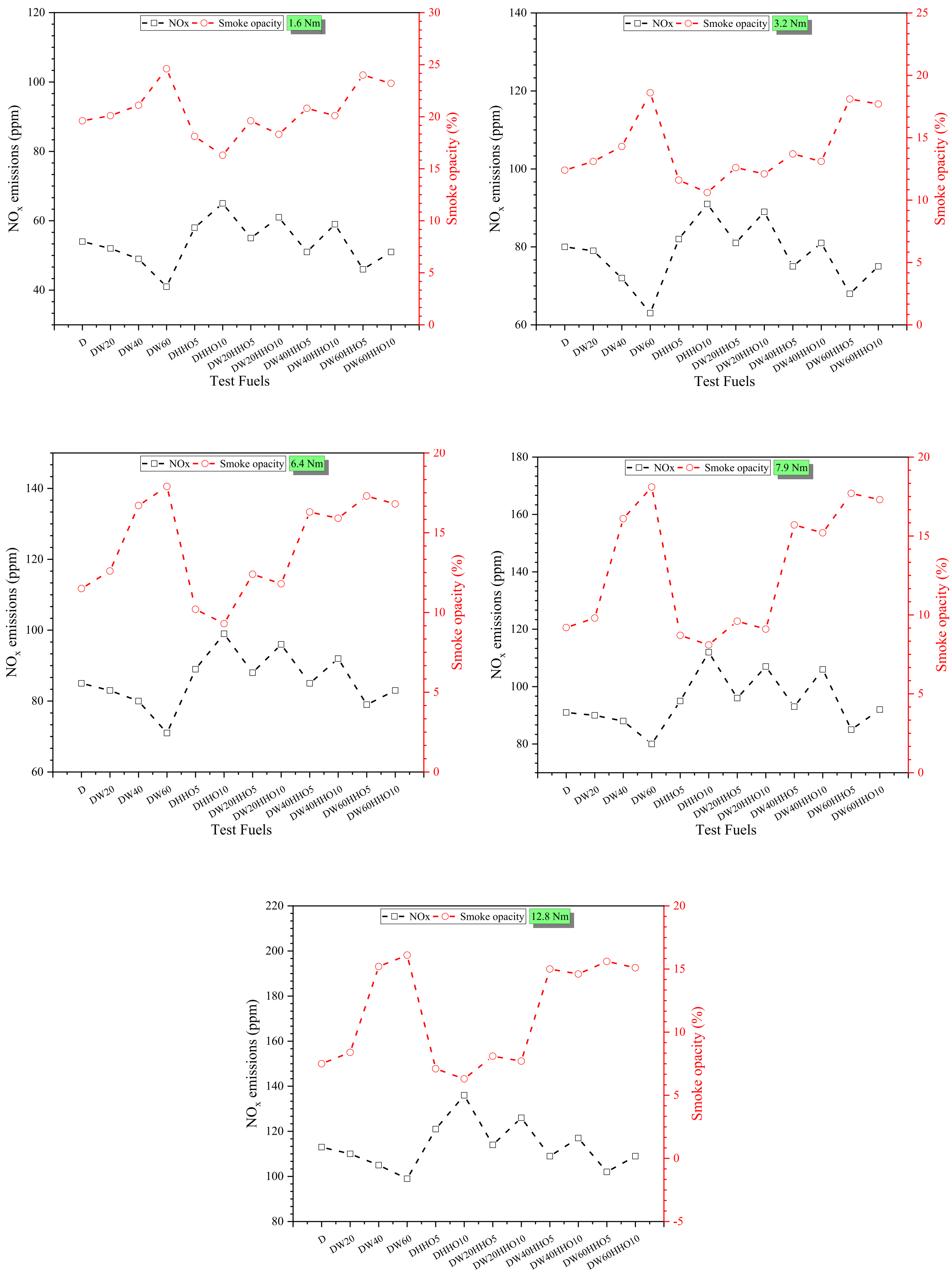

Fig. 4. Variations of NO_x and smoke opacity under different loads for diesel, WTPO, and HHO test fuels.

approximately a 17% reduction in HC emissions compared to diesel. The lowest HC emissions under all load conditions were obtained with the DHHO10 fuel at 12.8 Nm, with 29 ppm. Approximately 29% less HC emissions were produced compared to diesel fuel. The highest HC emission value under the same load conditions was 68 ppm obtained with the DW40 fuel, resulting in approximately 66% more HC emissions compared to diesel fuel. The higher density of WTPO compared to diesel fuel caused it to atomize over a longer period under the current temperatures, leading to some fuel being expelled through the exhaust without burning. On the other hand, running diesel-waste tyre-containing fuels with HHO fuel contributed to reducing HC emissions. Compared to the DW60 fuel, the DW60HHO10 fuel exhibited 10 ppm less HC emissions at an engine load of 1.6 Nm. It can be said that the oxygen and hydrogen in HHO improve combustion, reduce fuel consumption, and, therefore, contribute to a reduction in HC emissions.

When all carbon (C) in the fuel cannot find sufficient oxygen, one portion remains unburned while another is expelled as CO emissions. Additionally, low temperatures within the cylinder can cause the conversion reaction of CO emissions to CO_2 emissions to freeze, leading to an increase in CO emissions [81]. The most significant evidence of this phenomenon is the higher CO emissions at low loads compared to lower CO emission values at high loads, as observed in Fig. 4. Furthermore, an increase in cylinder temperatures accelerates the conversion reactions of CO to CO_2, thereby reducing CO emissions. The highest CO emissions were obtained with the DW60 fuel (0.81% vol.) at an engine load of 1.6 Nm, resulting in an approximately 80% increase in CO emissions compared to diesel operation. Under the same engine load conditions, the lowest CO emissions were achieved with the DHHO10 fuel (0.32% vol.), representing approximately a 29% reduction in CO emissions compared to diesel fuel. Karagöz and others [15], obtaining similar results with the use of WTPO, indicated that increasing the concentration of WTPO in diesel fuel lowers combustion temperatures, thereby reducing the oxidation of CO emissions, resulting in an increase in CO emissions. Running WTPO, which contributed to high CO emissions, with HHO fuel contributed to a decrease in CO emissions. Compared to the DW60 fuel, the DW60HHO10 fuel exhibited approximately a 17% reduction in CO emissions at a motor load of 1.6 Nm. The lowest CO emissions under all load conditions were obtained with the DHHO10 fuel (0.1% vol.) at a motor load of 12.8 Nm. While diesel fuel resulted in 0.15% vol. CO emissions at the current engine load, the DHHO10 fuel achieved approximately a 33% reduction. Similar trends of decreasing CO emissions were observed in diesel-WTPO-HHO studies. Compared to the DW60 operation, the DW60HHO10 fuel reduced CO emissions by approximately 23%. The oxygen content in HHO fuel increases the O_2 amount within the cylinder, converting CO emissions to CO_2 emissions.

On the other hand, as observed in Fig. 3, while CO_2 emissions increased, CO emissions decreased, and vice versa. The improvement in combustion efficiency and reduction in CO emissions due to the DHHO10 fuel generally resulted in the highest CO_2 emissions. The lowest CO_2 emissions were observed at 1.6 Nm, while the highest CO_2 emissions were observed at 12.8 Nm. At lower loads, lower cylinder temperatures and combustion stability increased both HC and CO emissions while reducing CO_2 emissions. The increase in load resulted in increased fuel injection, increasing the percentage of carbon (C) within the cylinder. The increased oxygen content due to HHO injection contributed to the increase in CO_2 emissions. The addition of WTPO to diesel fuel contributed to the reduction of CO_2 emissions, a greenhouse gas. The addition of WTPO, which reduces combustion temperatures, slows down the conversion reactions of CO to CO_2, leading to a decrease in CO_2 emissions. Additionally, an increase in WTPO content results in poorer combustion performance, leading to higher HC and CO emissions and lower CO_2 emissions. In obtaining similar results, Tudu and colleagues [82] reported an increase in CO_2 due to the blend's lower oxygen content and weak air/fuel mixture. The lowest CO_2 emission at an engine load of 1.6 Nm was obtained with the DW60 fuel (1.68% vol.), representing approximately a 13% decrease compared to diesel fuel. Similarly, when operated with HHO, the DW60HHO10 fuel exhibited approximately an 11% reduction in CO_2 emissions compared to diesel fuel.

Fig. 4 presents variations in NO_x and smoke pollutants for diesel, WTPO, and HHO fuel blends under varying load conditions. As seen under all load conditions, a trade-off relationship exists between NO_x and smoke pollutants. While NO_x emissions increase, smoke emissions decrease, and vice versa. It is possible to find literature supporting the trade-off relationship between NO_x and smoke emissions [83]. NO_x emissions occur at high oxygen levels and high operating temperatures [84]. Therefore, it can be said that increasing the engine load from 1.6 Nm to 12.8 Nm has contributed to the increase in NO_x emissions. The presence of O_2 in HHO fuel has increased the oxygen content within the cylinder. Additionally, the combustion of hydrogen in HHO fuel may lead to high cylinder temperatures. These two phenomena could explain the increase in NO_x emissions with HHO fuel blends. The highest NO_x emissions were found with the DHHO10 fuel in the between 1.6 Nm and 12.8 Nm load range, while the lowest was found with the DW60 fuel. At an engine load of 12.8 Nm, the DHHO10 fuel exhibited approximately a 20% increase in NO_x emissions compared to diesel fuel, with 136 ppm NO_x emissions. Under the same load conditions, the DW60 fuel exhibited approximately a 12% reduction in NO_x emissions compared to diesel fuel, with 99 ppm NO_x emissions. Similarly, at a load of 1.6 Nm, the DHHO10 fuel exhibited approximately a 20% higher NO_x emission formation compared to diesel fuel. Under the same load conditions, the DW60 fuel produced approximately a 24% decrease in NO_x emissions compared to diesel fuel.

Hürdoğan and colleagues [20] similarly reported that adding WTPO, which lacks oxygen in its content, results in lower NO_x emission levels than diesel fuel.

Smoke emissions occur in rich mixture regions near the injector nozzle where oxygen is scarce [85]. As observed in Fig. 4, blending WTPO with diesel fuel has significantly increased smoke emissions. One of the main reasons for this increase is the higher density and aromatic carbon content of WTPO compared to diesel fuel. Additionally, higher fuel consumption means a higher percentage of carbon within the cylinder, which is another important parameter leading to increased smoke emissions. Similar findings were reported by Murugan et al. [14], increasing the amount of WTPO was associated with an increase in aromatic carbon content, which resulted in increased smoke emissions. The use of HHO in diesel and WTPO blended fuels has contributed to decreasing smoke emissions. Because HHO increases the amount of O_2 in the cylinder, another important factor is the increase in combustion efficiency with HHO. The lowest smoke opacity percentage under all load conditions was achieved with the DHHO10 fuel (6.3%) at 12.8 Nm, representing approximately a 16% reduction compared to diesel fuel. Under the same load conditions, the highest smoke opacity percentage was obtained with the DW60 fuel (16.1%), exhibiting approximately a 115% increase compared to diesel fuel. The highest smoke opacity percentages were obtained with the DW60 fuel (24.6%) at an engine load of 1.6 Nm.

4.2. Energy analysis

Fig. 5 presents variations in energy analysis parameters for diesel, WTPO, and HHO fuel blends under different loads. As the load increased from 1.6 Nm to 12.8 Nm, it was observed that the engine input energy (fuel energy), brake power, exhaust energy, and loss energy increased. The increased need for thermal energy to meet the shaft power resulted in a higher engine input energy. For all loads, the highest fuel energy was found with the DW60 fuel, while the lowest fuel energy was found with DHHO10 (except for the 1.6 Nm engine load). Adding WTPO to diesel fuel led to increased injection and fuel energy requirements to meet the shaft power. However, adding HHO to diesel and diesel-WTPO blends had further reduced the fuel energy required to meet the shaft power. The increase in O_2 content and high diffusivity of HHO in the for approximately 65% and 58% of the fuel energies, respectively. The use of HHO in diesel-WTPO blends slightly reduced the lost energy ratio. At 12.8 Nm, the lost energy of

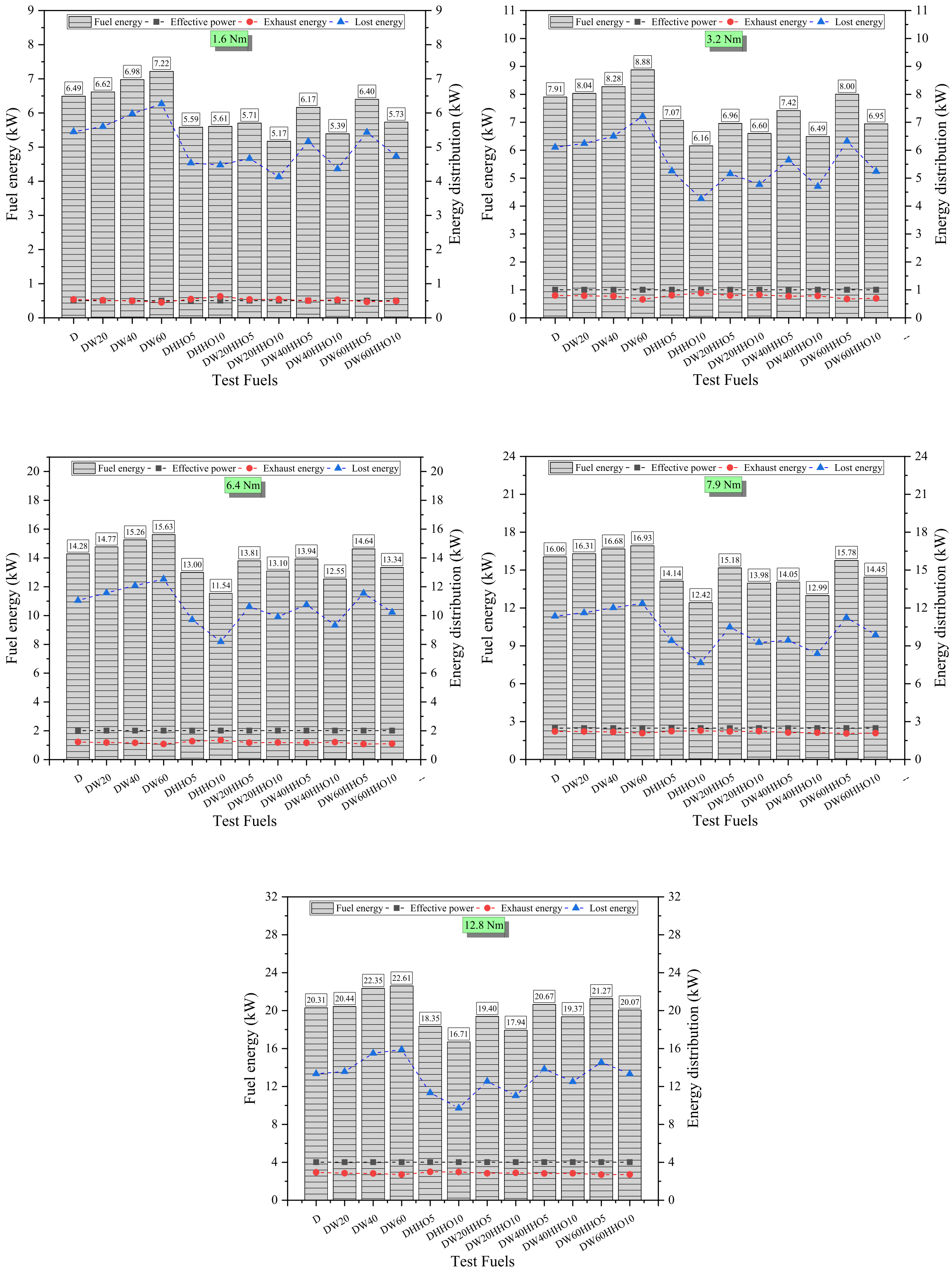

Fig. 5. Variations of fuel energy, exhaust energy, and lost energy parameters of energy analysis under different loads for diesel, WTPO, and HHO test fuels.

combustion zone have contributed to enhancing combustion efficiency, thereby reducing the fuel energy requirement. The highest fuel energy, 22.6 kW, was obtained with the DW60 fuel at 12.8 Nm, representing approximately an 11% increase compared to diesel fuel. Under the same load conditions, the addition of HHO to the DW60 fuel reduced the fuel energy to 20.07 kW. Thus, the fuel energy requirement of DW60HHO10 was approximately 11% lower than that of DW60 fuel. The fuel energy requirement decreased by approximately 1.5% with DW60HHO10

Fig. 5. Variations of fuel energy, exhaust energy, and lost energy parameters of energy analysis under different loads for diesel, WTPO, and HHO test fuels.

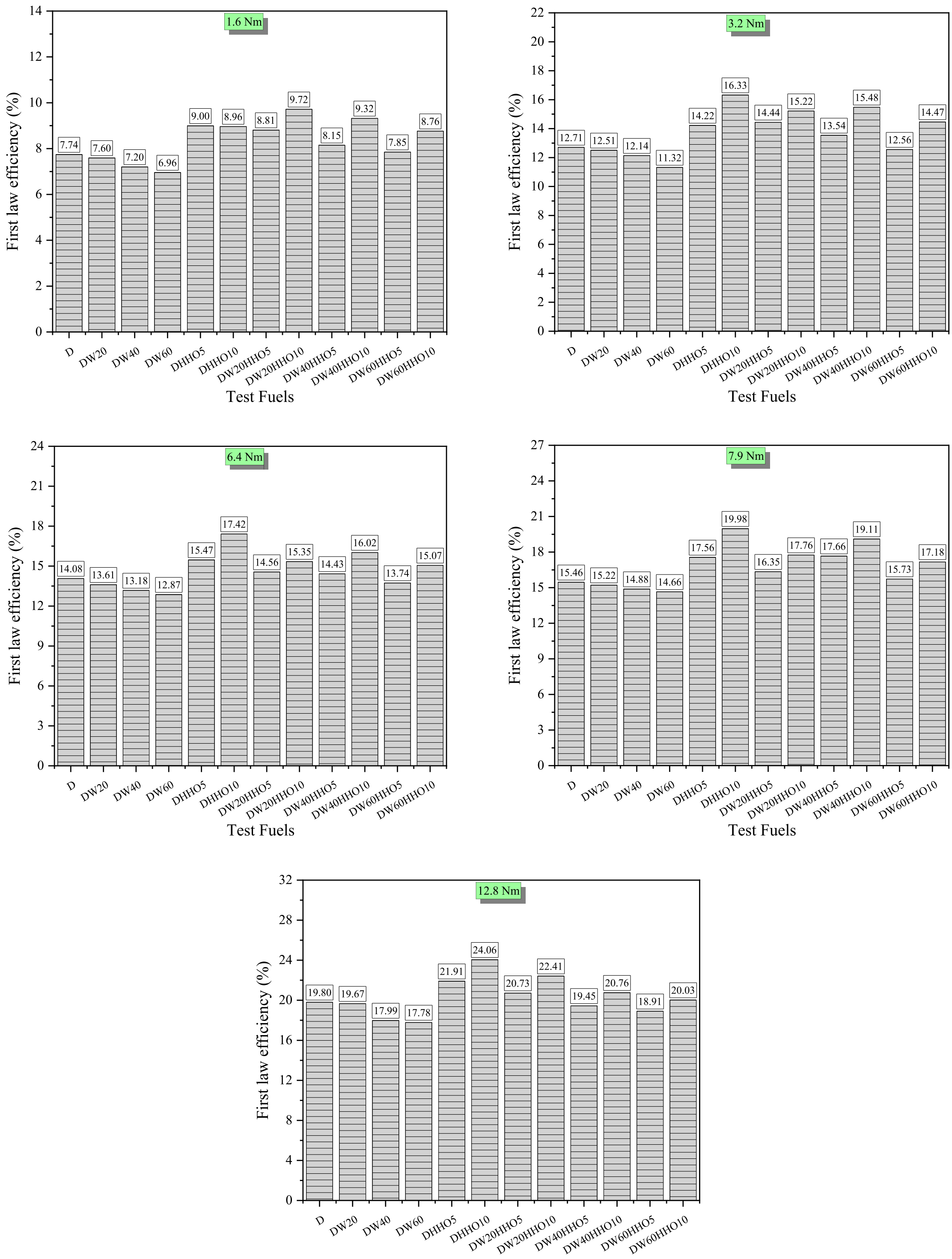

Fig. 6. Variations of first law efficiency under different loads for diesel, WTPO, and HHO test fuels.

compared to diesel fuel.

It was observed that the exhaust energy loss increased with the change in engine load. However, it was determined that the exhaust energy values of diesel, diesel-WTPO, and HHO fuel blends were similar trends under all loads. Generally, the highest exhaust energy was obtained with the DHHO5 fuel (3 kW) at 12.8 Nm, while the lowest exhaust energy loss was achieved with the DW60HHO5 fuel (2.7 kW). Under the same load conditions, while the exhaust energy of DHHO5 increased by approximately 2% compared to diesel fuel, the exhaust energy of DW60HHO5 decreased by approximately 8%.

attributing this to WTPO's weaker mixing with air, reduction in combustion pressure, and inertia within the combustion chamber.

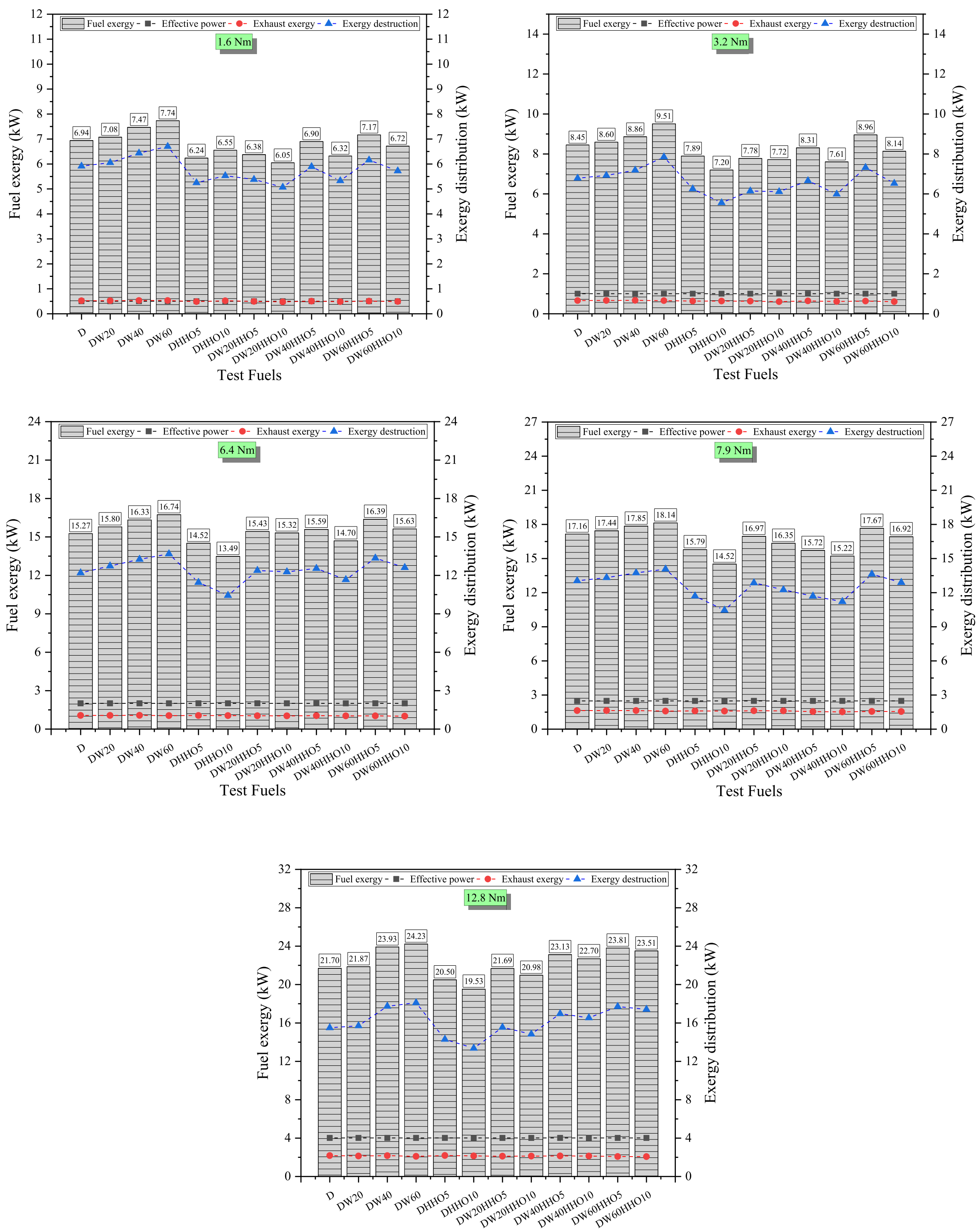

Fig. 7. Variations of fuel exergy, exhaust exergy, and exergy destruction parameters of exergy analysis under different loads for diesel, WTPO, and HHO test fuels.

On the other hand, adding WTPO to diesel fuel increased the lost energy ratio. It could be argued that the increased fuel input to meet the shaft power increased lost energy. The high WTPO density and its aromatic compounds reduce combustion efficiency. This scenario led to increased fuel consumption and consequently increased lost energy. The highest lost energy was found with the DW60 fuel under all loads, while the lowest lost energy was found with the DHHO10 fuel. The addition of HHO fuel to both diesel and diesel-WTPO blends contributed to reducing fuel consumption and consequently reducing lost energy. At 12.8 Nm, the lost energy of DW60 fuel (15.89 kW) constituted approximately 70% of the fuel energy. Under the same conditions, the lost energy of diesel (13.35 kW) and DHHO10 fuel (9.72 kW) accounted for approximately 65% and 58% of the fuel energies, respectively. The use of HHO in diesel-WTPO blends slightly reduced the lost energy ratio. At 12.8 Nm, the lost energy of DW60HHO10 fuel (13.34 kW) accounted for approximately 66% of the fuel energy. Under the same load conditions, the lost energy percentages of DW20HHO5 (12.54 kW), DW20HHO10 (11.03 kW), DW40HHO5 (13.83 kW), DW40HHO10 (12.51 kW), and DW60HHO5 (14.55 kW) fuels were approximately 65%, 62%, 67%, 64%, and 68 %of the energy share, respectively.

Fig. 6 presents variations in first-law efficiency for diesel, WTPO, and HHO fuel blends at different loads. The addition of WTPO to diesel fuel resulted in a decrease in the efficiency of the first law. One of the factors contributing to the decrease in first-law efficiency is the higher fuel consumption of DW20, DW40, and DW60 fuels compared to diesel fuel. WTPO having higher viscosity than diesel fuel and higher density, causes the fuel injected at the end of the compression stroke to evaporate more slowly and reduces combustion efficiency. This results in higher fuel consumption and a decrease in first-law efficiency. However, adding HHO to DW20, DW40, and DW60 fuels increased the first law efficiency. Wu and colleagues [79] also reported that increasing the amount of WTPO increases the mixture density and viscosity, increasing ignition delay and negatively affecting engine performance. Similarly, they reported improvements in performance using WTPO in conjunction with hydrogen. The highest first-law efficiency is achieved with the DHHO10 fuel under all load conditions (except for the 1.6 Nm engine load), while the lowest is obtained with the DW60 fuel. Parameters such as unstable combustion phase, residual gases from the previous cycle, fluid friction, LHV of the mixture, and fuel consumption may be among the reasons for the first law efficiency results at 1.6 Nm. The highest first law efficiency, approximately 24%, is achieved with the DHHO10 fuel at 12.8 Nm, representing an increase of about 21% compared to diesel fuel (19.8%). At the same loads, the first law efficiency of the DW60 fuel is 17.7%, while with the addition of HHO10, it increases to 20%. The improvement in combustion efficiency and the reduction in fuel consumption due to HHO addition are among the most significant factors contributing to the increase in the first law efficiency.

4.3. Exergy analysis

Fig. 7 presents variations in exergy analysis parameters for diesel, WTPO, and HHO fuel blends at different loads. The fuel exergy increased for all fuel blends as the engine load was increased from 1.6 Nm to 12.8 Nm. The primary reason for this was the increased fuel mass flow rate. While the augmentation of WTPO to diesel fuel increased fuel exergy at all test loads, adding HHO to diesel and diesel-WTPO blends contributed to a decrease in fuel exergy. At a 1.6 Nm engine load, the highest fuel exergy was achieved with the DW60 fuel (7.74 kW), showing an increase of approximately 11.5% compared to diesel fuel. At the same loads, the lowest fuel exergy was found with the DW20HHO10 fuel (6.05 kW), indicating a decrease of approximately 13% compared to diesel fuel. At 12.8 Nm, the fuel exergy of the DW60 fuel (24.2 kW) showed an increase of approximately 11.6% compared to diesel fuel. At the same loads, the lowest fuel exergy was obtained with the DHHO10 fuel (19.5 kW), representing a decrease of approximately 10% compared to diesel fuel.

The exhaust exergy increased for all fuel blends as the load was increased from 1.6 Nm to 12.8 Nm. At higher loads, increased fuel input into the system can lead to a wider range of combustion processes and, consequently, higher EGT. This is one of the reasons for the increase in exhaust exergy. Similarly, the exhaust energy loss increases as the load increases, as reported by Yarlagadda and Ravuru [81]. The lowest exhaust exergy for all test fuels was found with the DW20HHO10 fuel at 1.6 Nm, resulting in approximately a 7% decrease compared to diesel fuel. At the same loads, the highest exhaust exergy was achieved with the DW60 fuel (0.53 kW), showing an increase of approximately 2% compared to diesel fuel. The highest exhaust exergy for all test fuels was found with the DHHO5 fuel at 12.8 Nm, showing an increase of approximately 0.5% compared to diesel fuel. At the same loads, the lowest exhaust exergy was obtained with DW60HHO5 and DW60HHO10 fuels (2.08 kW), showing approximately 5% lower exhaust exergy than diesel. For 12.8 Nm, the exhaust exergy of diesel fuel (2.18 kW) accounts for 10% of the fuel exergy. Under the same load conditions, the percentage of fuel exergy for DW60HHO5 and DW60HHO10 fuels with low exhaust exergy is approximately 8.7% and 8.8%, respectively.

For all test fuels, exergy destruction has the highest share of fuel exergy. Fig. 7 shows the exergy destruction of different engine loads and fuel blends. Along with the increase in fuel exergy during load changes, exergy destruction also increased. The lowest exergy destruction at 1.6 Nm was achieved with HHO-added fuels, while the highest exergy destruction at the same engine load was observed with diesel-WTPO blends. At 1.6 Nm, the exergy destruction for DHHO5, DHHO10, DW20HHO5, DW20HHO10, DW40HHO5, DW40HHO10, and DW60HHO10 fuels was 5.25 kW, 5.53 kW, 5.38 kW, 5.07 kW, 5.89 kW, 5.33 kW, and 5.72 kW, respectively. Compared to the exergy destruction of diesel fuel (5.92 kW), representing approximately 11%, 6.6%, 9%, 14%, 0.5%, 10%, and 3.5%, respectively. Additionally, while diesel fuel exergy destruction accounted for approximately 85% of the fuel exergy, the shares of exergy destruction in the fuel exergies for DHHO5, DHHO10, DW20HHO5, DW20HHO10, DW40HHO5, DW40HHO10, and DW60HHO10 fuels were approximately 84%, 84.4%, 84.3%, 83.8%, 85.3%, 84.3%, and 85%, respectively. The highest exergy destruction for all test conditions was achieved with the DW60 fuel (18.1 kW) at 12.8 Nm, showing approximately 17% more exergy destruction compared to diesel fuel. At 12.8 Nm, the lowest exergy destruction was achieved with the DHHO10 fuel (13.3 kW), showing approximately 14% less exergy destruction compared to diesel fuel. At the same loads, diesel fuel exergy destruction accounted for 71% of the fuel exergy, while DHHO10 fuel exergy destruction accounted for approximately 68%. As a result, adding WTPO to diesel fuel leads to an increase in fuel consumption and, consequently, an increase in fuel exergy. This situation can result in intense chemical reactions and high cylinder temperatures. The literature shows high fuel consumption increases cylinder temperatures and chemical reactions, leading to higher exergy destruction and entropy generation [86–88]. Additionally, Bayramoğlu and Nuran [89], obtaining similar results, have reported that WTPO is more dominant in exergy destruction compared to diesel, attributing this to WTPO's weaker mixing with air, reduction in combustion pressure, and inertia within the combustion chamber.

In Fig. 8, variations in entropy generation are presented for diesel, WTPO, and HHO fuel blends in the between 1.6 Nm and 12.8 Nm load range. Adding WTPO to diesel fuel increased entropy generation, whereas the addition of HHO to blends containing both diesel and WTPO reduced entropy generation. The highest entropy generation was achieved with the DW60 fuel under all loads, while the lowest entropy generation was found with diesel-HHO fuels. The increase in fuel consumption due to WTPO resulted in higher combustion temperatures and intense chemical reactions, leading to higher exergy destruction. This increase in exergy destruction, in turn, led to irreversibility and higher entropy generation. Similarly, Yarlagadda and Ravuru [65] found that the addition of WTPO to diesel fuel resulted in higher entropy generation compared to diesel. Also, Eq. (17) reveals that entropy generation is

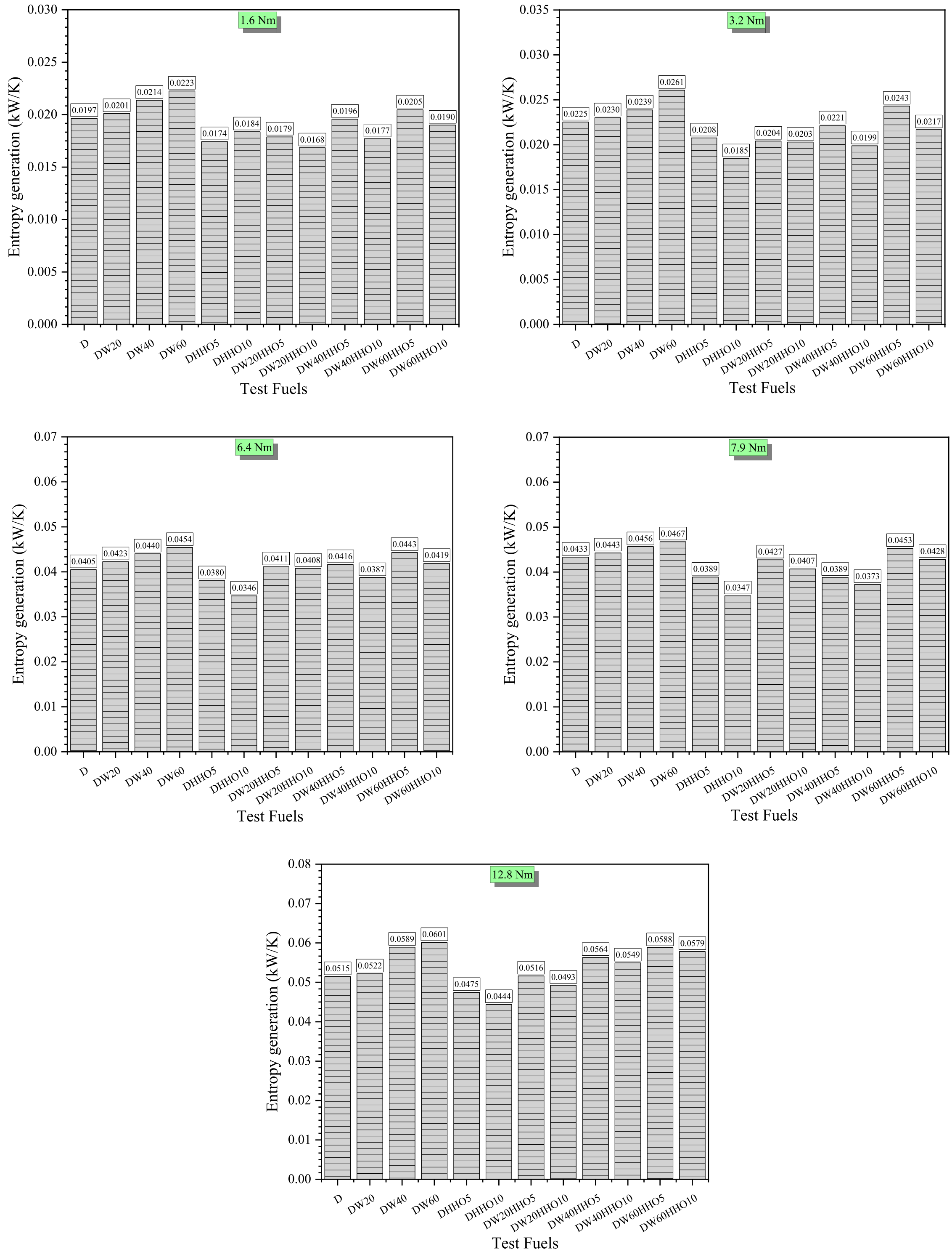

Fig. 8. Variations of entropy generation under different loads for diesel, WTPO, and HHO test fuels.

directly related to exergy destruction: An increase in exergy destruction increases entropy generation, while a decrease in exergy destruction decreases entropy generation. Therefore, the reduced exergy destruction with the addition of HHO to diesel-WTPO blends resulted in decreased entropy generation. The highest entropy generation was achieved with the DW60 fuel (0.0601 kW/K) at 12.8 Nm, indicating an increase in entropy generation of approximately 17% compared to diesel fuel. The entropy generation of HHO-added DW60HHO10 fuel (0.0579 kW/K) decreased by approximately 4% compared to DW60. The lowest entropy generation under all load conditions was achieved with the DW20HHO10 fuel (0.0168 kW/K) at 1.6 Nm, indicating a decrease in entropy generation of approximately 15% compared to diesel fuel. Overall, the DHHO10 fuel exhibited lower entropy generation.

In Fig. 9, variations in the second law efficiency and sustainability index (SI) are presented for diesel, WTPO, and HHO fuel blends at different loads. Similar to the first law efficiency, when WTPO was blended with diesel fuel, the second law efficiency decreased, while the addition of HHO to blends containing both diesel and WTPO increased

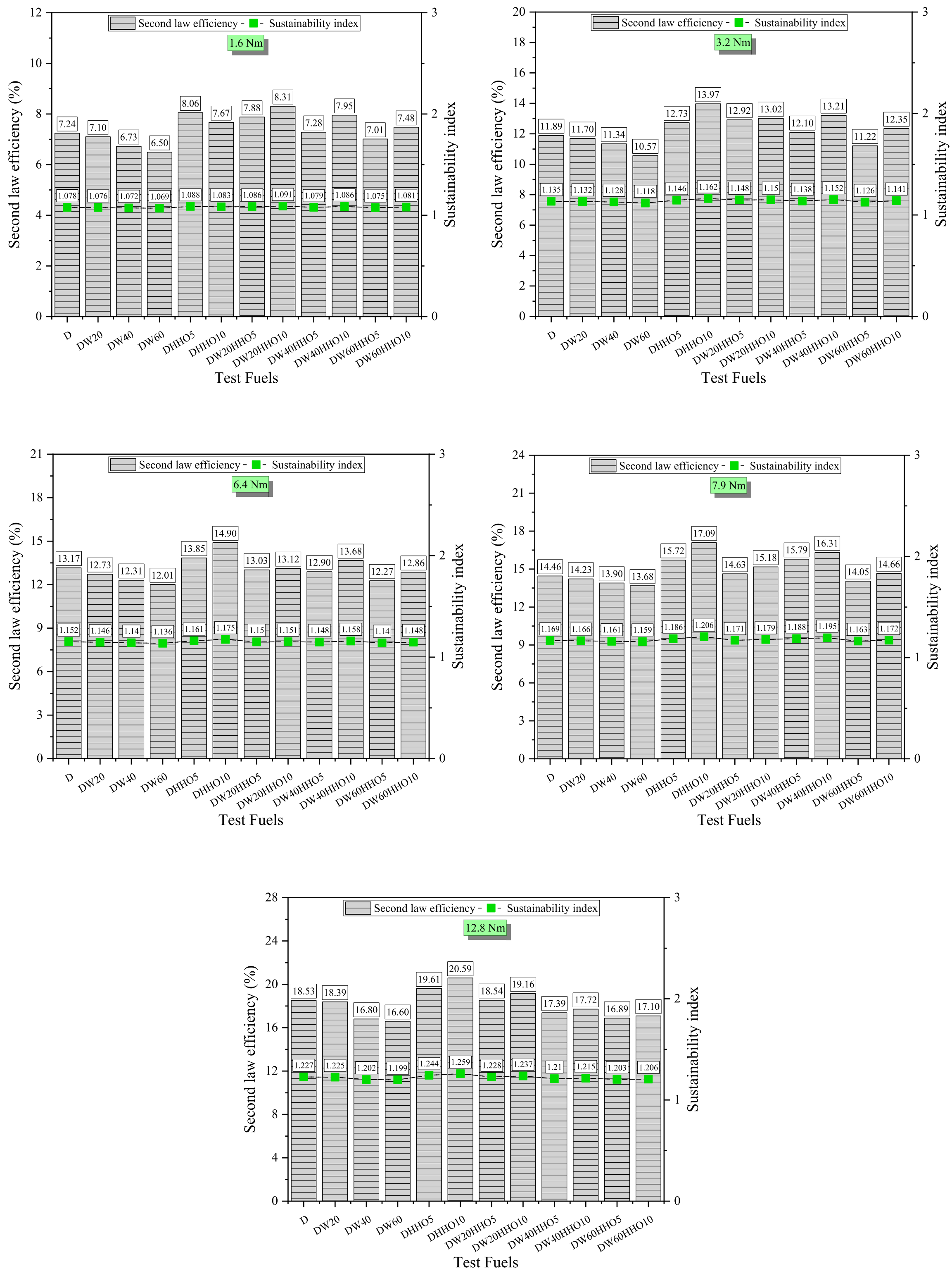

Fig. 9. Variations of second law efficiency and sustainability index under different loads for diesel, WTPO, and HHO test fuels.

the second law efficiency. Some studies in the literature report an increase in second-law efficiency with the addition of WTPO [90,91]. For example, Karagoz and colleagues [90] reported that increasing the amount of WTPO in diesel fuel decreases second-law efficiency but still results in higher second-law efficiency compared to diesel fuel. The difference between the current study and the study by Karagoz et al. is mainly due to variations in the physical and chemical properties of WTPO fuel. The WTPO used in this study, with its higher density and viscosity characteristics, results in poorer combustion performance. This leads to decreases in both first-law and second-law efficiencies. The lowest second law efficiency was achieved with the DW60 fuel (6.4%) at 1.6 Nm, representing a decrease of approximately 10% compared to diesel fuel. At the same loads, the highest second-law efficiency was obtained with the DW20HHO10 fuel (8.3%), showing an increase of approximately 15% compared to diesel fuel. The highest second law efficiency was achieved with the DHHO10 fuel (20.5%) at 12.8 Nm, indicating an increase of approximately 11% compared to diesel fuel. At the same loads, the lowest second law efficiency was found with the DW60 fuel (16.5%), showing a decrease of approximately 11% compared to diesel fuel. Additionally, the addition of HHO to the DW60 fuel (DW60HHO5 and DW60HHO10) increased the second law efficiencies by approximately 2% and 4%, respectively.

The SI of all test fuels had increased with the increase in engine load from 1.6 Nm to 12.8 Nm. A more stable combustion phase and lower specific fuel consumption at higher loads contributed to the increase in SI. Adding WTPO to diesel fuel has led to a decrease in SI values. The main reason for this situation is the increase in fuel consumption and lower second-law efficiency. When Eq. (18) is examined, an increase in second-law efficiency results in higher SI, while a decrease in second-law efficiency leads to lower SI. Therefore, the decreased second law efficiency due to higher specific fuel consumption has resulted in lower SI values for diesel-WTPO blends. Adding HHO to both diesel and diesel-WTPO blends contributed to an increase in SI values. The enhancement of second-law efficiencies through the addition of HHO was considered a major factor in boosting SI. The highest SI was attained with the DHHO10 fuel (1.26), indicating approximately 2.4% greater sustainability compared to diesel fuel. Conversely, at the same loads, the lowest SI was recorded with the DW60 fuel (1.99), representing approximately a 3.3% decrease compared to diesel fuel. Despite this, the addition of HHO to the DW60 fuel marginally improved sustainability.

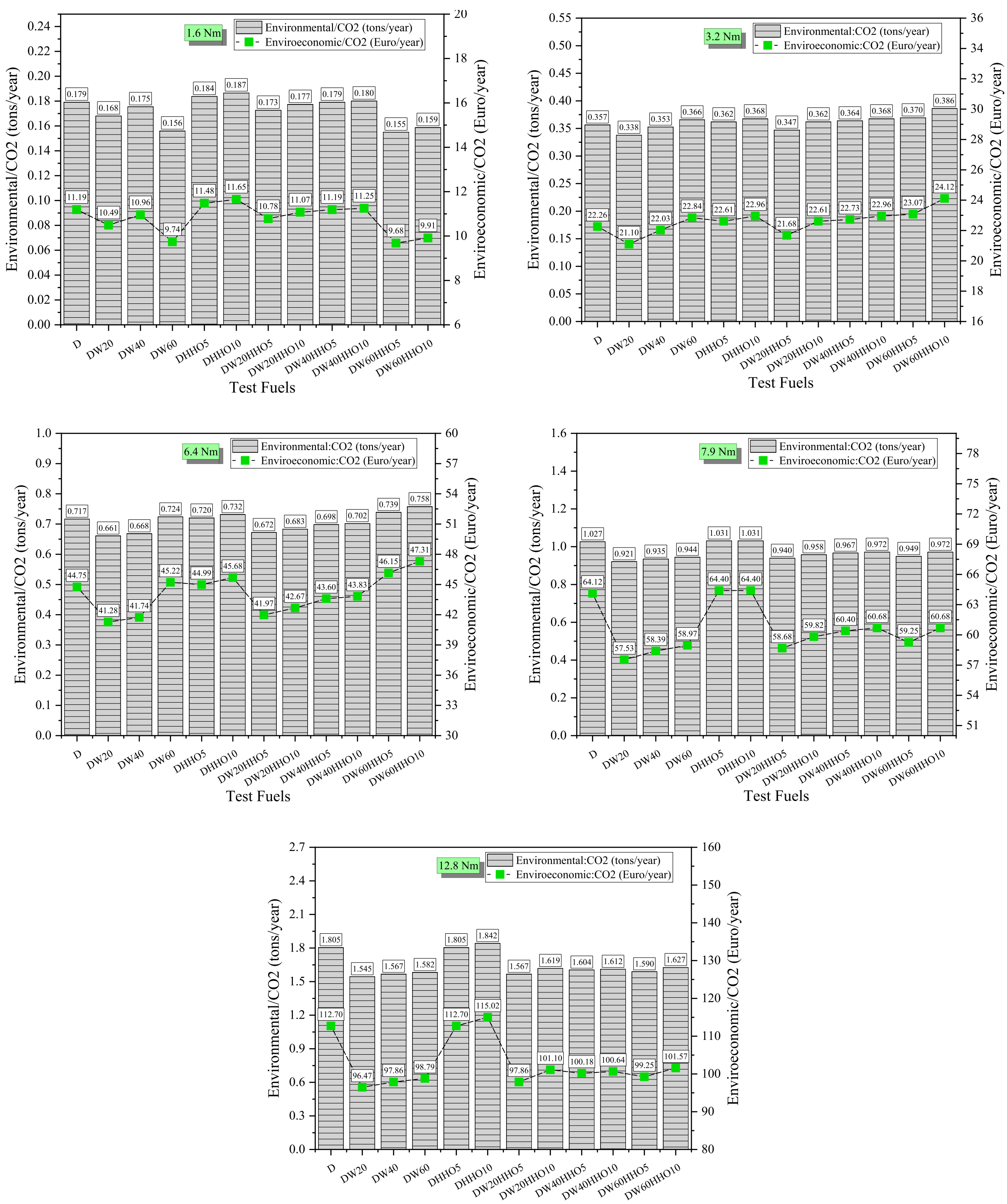

Fig. 10. Variation of the annual in CO_2 amount emitted from the test engine and enviroeconomic factor under different loads for diesel, WTPO, and HHO test fuels.

4.4. Environmental and enviroeconomic analysis

In heat engines, emissions are released into the environment as a result of chemical reactions. The damage inflicted on the environment by emitted pollutants can vary depending on their type. For instance, CO_2 emissions can contribute to the formation of greenhouse gases and, thus, global warming [75]. The CO_2 gas produced in ICEs can vary depending on the type of fuel used, engine operating parameters, and combustion efficiency [58]. Reducing CO_2 emissions from internal combustion engines could contribute to mitigating global warming, a significant environmental threat. This study calculates the annual CO_2 emissions amount and the annual cost of CO_2 emissions for different engine loads and fuel blend ratios. Fig. 10 presents variations in environmental and enviroeconomic parameters for diesel, WTPO, and HHO fuel blends at different loads. As the load increased from 1.6 Nm to 12.8 Nm, the amount of CO_2 emitted from the engine increased. The lowest CO_2 emission was achieved at 1.6 Nm with the DW60HHO10 fuel (0.155 CO_2 tons/year), resulting in approximately a 13% decrease in annual CO_2 emission compared to diesel fuel. The highest CO_2 emission was obtained at 12.8 Nm with the DHHO10 fuel (1.84 CO_2 tons/year), resulting in approximately a 2% increase in annual CO_2 emission compared to diesel fuel. Generally, the use of WTPO in diesel fuel reduces the annual CO_2 emission amount. The addition of HHO to both diesel and diesel-WTPO blends slightly increases CO_2 emissions. The average annual CO_2 emissions amount for the test fuels D, DW20, DW40, DW60, DHHO5, DHHO10, DW20HHO5, DW20HHO10, DW40HHO5, DW40HHO10, DW60HHO5, and DW60HHO10 are 0.81 CO_2 tons/year, 0.72 CO_2 tons/year, 0.73 CO_2 tons/year, 0.75 CO_2 tons/year, 0.82 CO_2 tons/year, 0.83 CO_2 tons/year, 0.73 CO_2 tons/year, 0.76 CO_2 tons/year, 0.76 CO_2 tons/year, 0.76 CO_2 tons/year, 0.76 CO_2 tons/year, and 0.78 CO_2 tons/year, respectively.

In all test conditions, the lowest annual CO_2 emission cost was achieved with diesel-WTPO blends, while the addition of HHO to both diesel and diesel-WTPO blends slightly increased annual CO_2 costs. The lowest annual CO_2 emission cost was obtained with the DW60 fuel (9.74 CO_2 Euro/year) at 1.6 Nm, resulting in approximately a 13% decrease compared to diesel fuel. The highest annual CO_2 emission cost was obtained with the DHHO10 fuel (115.02 CO_2 Euro/year) at 12.8 Nm, indicating approximately a 2% increase compared to diesel fuel. The average annual CO_2 costs for the test fuels D, DW20, DW40, DW60, DHHO5, DHHO10, DW20HHO5, DW20HHO10, DW40HHO5, DW40HHO10, DW60HHO5, and DW60HHO10 are 51 CO_2 Euro/year, 45 CO_2 Euro/year, 46 CO_2 Euro/year, 47 CO_2 Euro/year, 51 CO_2 Euro/year, 52 CO_2 Euro/year, 46 CO_2 Euro/year, 47 CO_2 Euro/year, 48 CO_2 Euro/year, 48 CO_2 Euro/year, 47 CO_2 Euro/year, and 49 CO_2 Euro/year, respectively. On the contrary, Singh and Paul reported [92] in their study that the addition of diesel fuel increases combustion stability and, as a result, leads to an increase in CO_2 emissions and an enhanced enviroeconomic impact. The primary reason for the differing results in the two studies is that the WTPO used in the experiments has different physicochemical properties. The high density and high viscosity of WTPO cause a decrease in combustion stability and, consequently, in combustion efficiency. This situation results in a decrease in CO_2 emissions and, thus, a reduction in the enviroeconomic impact.

Table 9
Summary of similar studies.

Study	Fuels Used	Engine Type	Parameters Analyzed	Key Findings	Advantages of Current Study
Current study	Waste tyre pyrolysis oil (WTPO), oxy-hydrogen gas	Diesel engine	Enhanced energy efficiency, reduced emissions, and improved enviroeconomic benefits	Enhanced energy efficiency, reduced emissions, and improved enviroeconomic benefits	Comprehensive analysis covering energy, exergy, environmental, and enviroeconomic perspectives
M. Vergel-ortega et al. [93]	Biodiesel blends	Single-cylinder diesel engine	Performance, emissions	The use of biodiesel significantly reduces carbon monoxide (CO), total hydrocarbon (THC) and NOx emissions. It increases the thermal efficiency of the engine and reduces specific fuel consumption.	The reported reduction in carbon monoxide (CO), total hydrocarbon (THC), and NOx emissions with biodiesel demonstrates that alternative fuels can significantly improve environmental performance. This is particularly relevant to our study as we also aim to identify fuel combinations that enhance environmental outcomes.
Koten [41]	Hydrogen	Multi cylinder diesel engine	Thermal efficiency, NO_x emission	Increasing hydrogen addition in diesel engines results in increased brake specific fuel consumption and brake thermal efficiency, but also increases exhaust temperature and NOx emissions at high loads.	The reported increase in brake specific fuel consumption (BSFC) and brake thermal efficiency (BTE) with hydrogen addition underscores the potential of hydrogen as an effective additive to enhance combustion efficiency. This aligns with our findings, where the introduction of oxy-hydrogen gas similarly enhances the thermal efficiency of the diesel engine due to improved combustion characteristics.
Haonan Yuan et al. [94]	Fischer-tropsch fuels	Direct injection diesel engine with EGR	Performance, emissions, EGR	Low cetane number Fischer-tropsch fuels show potential for higher thermal efficiency and reduction in exhaust emissions in commercial diesel engines with EGR.	The findings that low cetane number FT fuels can achieve higher thermal efficiency and reduced exhaust emissions, especially when coupled with exhaust gas recirculation (EGR), highlight the importance of fuel properties and combustion control strategies in optimizing engine performance and environmental impact. This is particularly relevant to our research, which also focuses on enhancing thermal efficiency and reducing emissions through alternative fuel use.
Sujeet Kesharvani et al. [95]	Ethanol blended fuel with algae biodiesel	Diesel engine	Performance, emission	Ethanol blended fuel with algae biodiesel improves combustion and performance, while reducing greenhouse emissions compared to diesel fuel.	At the end of the study; "future research should include new technologies such as waste heat recovery systems that increase fuel economy and reduce the environmental impact of diesel engines." the necessity of this study was emphasized with the sentence.

5. Conclusions

Table 9 presents a comparison of the current experimental study with previous studies in the literature.

In the present paper, energy, exergy, and sustainability analyses were conducted for diesel, WTPO, and HHO fuel blends. The CO_2 gas generated as a result of combustion contributes to greenhouse gas formation and is one of the causes of global warming. Therefore, the environmental impact and eco-economic aspects of the CO_2 gas produced from the combustion of diesel, WTPO, and HHO fuel blends were evaluated. The overall assessment of the analyses is as follows:

- Adding WTPO to diesel fuel increases HC and CO emissions while reducing CO_2 emissions. This is because the physical and chemical properties of WTPO increase the density and aromatic compounds of the mixture compared to diesel fuel, leading to more intense chemical processes and increased fuel consumption. Using HHO in both diesel and diesel-WTPO operations has contributed to a decrease in HC and CO emissions. This is attributed to the contribution of H_2 and O_2 in HHO to improving combustion efficiency.
- There is a trade-off relationship between NO_x and smoke opacity. The physical and chemical properties of WTPO increase the density of diesel blends, resulting in richer mixture zones and denser smoke opacity. The addition of HHO to both diesel and diesel- WTPO blends increases NO_x emissions while reducing smoke opacity. Maximum NO_x emissions are obtained with the DHHO10 fuel (showing a 20% increase compared to diesel at 12.8 Nm), while minimum NO_x emissions are achieved with the DW60 fuel (showing a 24% decrease compared to diesel at 1.6 Nm). In terms of smoke emissions, the situation is reversed. Maximum smoke opacity is achieved with the DW60 fuel (showing a 26% increase compared to diesel at 1.6 Nm), while minimum smoke opacity is achieved with the DHHO10 fuel (showing a 16% decrease compared to diesel at 12.8 Nm).
- Under all load conditions, the highest fuel energy (22.6 kW) and exergy (24.2 kW) are achieved with the DW60 fuel containing 60% WTPO, while the addition of HHO to the diesel-WTPO mixture reduces fuel energy and exergy. The highest decrease in fuel energy occurs in the DW40HHO10 study, averaging approximately 14%, while the highest decrease in fuel exergy happens in the DW40HHO10 study, averaging approximately 6%.
- Compared to diesel fuel, the maximum increase in energy loss is achieved with the DW60 operation, averaging approximately 15%. However, when the DW60 fuel is used in a dual fuel mode with 10 L/min HHO gas, the energy loss rate decreases by averaging approximately 10% compared to diesel fuel.
- In terms of exhaust energy, compared to diesel fuel, the maximum decrease occurs with the DW60 fuel, averaging approximately 12%. Conversely, the maximum increase in exhaust energy (averaging 8%) is achieved with the diesel-HHO dual fuel application. This increase is attributed to HHO gas raising combustion temperatures and promoting a more stable combustion phase, leading to higher exhaust gas temperatures and consequently increased exhaust energy loss.
- The lowest exergy destruction at all loads is achieved with the Diesel+10 L/min HHO dual fuel application, which shows an average reduction in exergy destruction of approximately 14.6% compared to diesel fuel. Conversely, adding WTPO to diesel fuel increases exergy destruction. However, using the WTPO-HHO dual fuel application decreases exergy destruction. The DW40HHO10 study exhibits an approximate average reduction in exergy destruction of about 7% compared to diesel fuel.
- The highest exergy efficiency at all loads is achieved with the Diesel +10 L/min HHO dual fuel application, showing an average increase in exergy efficiency of approximately 13.2% compared to diesel fuel. Conversely, adding WTPO to diesel fuel results in a decrease in exergy efficiency. However, the WTPO-HHO dual fuel application tends to increase exergy efficiency. The DW40HHO10 study exhibits an approximate average increase in exergy efficiency of about 6.6% compared to diesel fuel.
- Adding WTPO to diesel fuel reduces sustainability, whereas using both diesel and diesel/WTPO blends with HHO gas increases sustainability. The sustainability ranking from highest to lowest is as follows: DHHO10 > DHHO5 > DW40HHO10 > DW20HHO10 > DW20HHO5 > DW40HHO5 > D100 > DW60HHO10 > DW20 > DW60HHO5 > DW40 > DW60.
- Mixing diesel fuel with WTPO contributes to a reduction in environmental and enviroeconomic impact. One of the primary reasons for this is the significant reduction in CO_2 emissions facilitated by WTPO. The lowest environmental impact at all loads is achieved with diesel-WTPO fuel blends, with an average maximum reduction in environmental and enviroeconomic impact of 9% compared to diesel fuel observed with the DW20 fuel.
- Operating diesel fuel in a dual fuel mode with HHO increases environmental and enviroeconomic impact compared to running diesel, whereas using diesel/WTPO blends with HHO contributes to reducing environmental and enviroeconomic impact. Compared to diesel fuel, the maximum average reduction in the use of HHO dual fuel mode is achieved with the DW20HHO5 fuel blend, at a rate of approximately 6.6%.

  In general, the use of WTPO in diesel fuel results in low performance and high emission values. This adversely affects energy, exergy, and SI. The current study shows that despite WTPO initially exhibiting poorer energy and exergy analysis results, its performance improves when used in the HHO dual fuel mode. However, this improvement remains lower compared to diesel-HHO dual fuel operation. Therefore, the fuel properties of WTPO fuel need to be improved. The use of nano additive materials can improve fuel properties. Future studies could explore the use of nanoparticle additives to enhance the performance of blends containing high levels of WTPO in a dual fuel mode with HHO.

In light of the findings from this study, the integration of waste tyre pyrolysis oil and oxy-hydrogen gas as alternative fuels in diesel engines presents a promising pathway towards sustainable energy solutions. Future research should focus on optimizing the pyrolysis process to maximize energy yield and minimize the environmental footprint. Additionally, scaling up the application of oxy-hydrogen gas in commercial diesel engines could further validate its potential in reducing emissions and improving engine performance. Collaboration between academia, industry, and government bodies will be crucial to advancing this research and realizing its full potential in contributing to global sustainability goals.

CRediT authorship contribution statement

Salih Özer: Writing – review & editing, Writing – original draft, Visualization, Validation, Supervision, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Halil Erdi Gülcan:** Writing – review & editing, Writing – original draft, Visualization, Validation, Investigation, Formal analysis, Data curation, Conceptualization. **Samet Çelebi:** Writing – review & editing, Writing – original draft, Visualization, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Usame Demir:** Writing – review & editing, Writing – original draft, Validation, Supervision, Software, Methodology, Investigation, Formal analysis, Data curation, Conceptualization.

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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