energies

MDPI

Review

A Comprehensive Review on the Hydrogen–Natural Gas–Diesel Tri-Fuel Engine Exhaust Emissions

Hassan Sadah Muhssen [1,2], Máté Zöldy [3] and Ákos Bereczky [1,*]

[1] Department of Energy Engineering, Faculty of Mechanical Engineering, Budapest University of Technology and Economics, Műegyetem rkp. 3, 1111 Budapest, Hungary; hmuhssen@edu.bme.hu or hassan.wassit@yahoo.com
[2] Department of Mechanical Engineering, Faculty of Engineering, Wasit University, Wasit 52001, Iraq
[3] Department of Automotive Technologies, Faculty of Transportation Engineering and Vehicle Engineering, Budapest University of Technology and Economics, 1111 Budapest, Hungary; zoldy.mate@kjk.bme.hu
* Correspondence: bereczky@energia.bme.hu

Abstract: Natural gas (NG) is favored for transportation due to its availability and lower CO_2 emissions than fossil fuels, despite drawbacks like poor lean combustion ability and slow burning. According to a few recent studies, using hydrogen (H_2) alongside NG and diesel in Tri-fuel mode addresses these drawbacks while enhancing efficiency and reducing emissions, making it a promising option for diesel engines. Due to the importance and novelty of this, the continuation of ongoing research, and insufficient literature studies on HNG–diesel engine emissions that are considered helpful to researchers, this research has been conducted. This review summarizes the recent research on the HNG–diesel Tri-fuel engines utilizing hydrogen-enriched natural gas (HNG). The research methodology involved summarizing the effect of engine design, operating conditions, fuel mixing ratios and supplying techniques on the CO, CO_2, NO_x and HC emissions separately. Previous studies show that using natural gas with diesel increases CO and HC emissions while decreasing NO_x and CO_2 compared to pure diesel. However, using hydrogen with diesel reduces CO, CO_2, and HC emissions but increases NO_x. On the other hand, HNG–diesel fuel mode effectively mitigates the disadvantages of using these fuels separately, resulting in decreased emissions of CO, CO_2, HC, and NO_x. The inclusion of hydrogen improves combustion efficiency, reduces ignition delay, and enhances heat release and in-cylinder pressure. Additionally, operational parameters such as engine power, speed, load, air–fuel ratio, compression ratio, and injection parameters directly affect emissions in HNG–diesel Tri-fuel engines. Overall, the Tri-fuel approach offers promising emissions benefits compared to using natural gas or hydrogen separately as dual-fuels.

Keywords: emission; hydrogen enrichment; natural gas; HCNG; diesel engine

Citation: Muhssen, H.S.; Zöldy, M.; Bereczky, Á. A Comprehensive Review on the Hydrogen–Natural Gas–Diesel Tri-Fuel Engine Exhaust Emissions. *Energies* **2024**, *17*, 3862. https://doi.org/10.3390/en17153862

Academic Editors: Sławomir Wierzbicki and Kamil Duda

Received: 25 June 2024
Revised: 1 August 2024
Accepted: 2 August 2024
Published: 5 August 2024

1. Introduction

Numerous studies by medical experts have demonstrated that emissions from vehicle exhaust pipes contribute significantly to various health problems, including allergic respiratory conditions like rhinosinusitis and bronchial asthma, as well as cardiovascular and neurobehavioral issues. Moreover, compelling evidence suggests that prolonged exposure to fine particulate matter (PM 2.5) pollution is associated with increased rates of cardiopulmonary mortality [1–4]. In the early 19th century, increased automation prompted a surge in research aimed at enhancing power-generating machinery [5,6]. These engines typically utilize fuels derived from fossil fuels. Among the two main types of engines, namely spark ignition (SI) and compression ignition (CI) engines, CI engines are predominantly utilized for power generation in medium and heavy-duty applications due to their enhanced efficiencies and lower specific fuel consumption [6,7]. Due to their cost effectiveness, durability, reliability and high combustion efficiency, diesel engines are commonly used worldwide [8–11]. Despite this, diesel engines are considered one of the main sources of environmental pollution [12–14] and for a sustainable future, it is a key challenge to decrease their emissions [15]. The NO_x and particulate matter are the most important pollutants from diesel engines. Photochemical smog and acid rain are attributed to NO_x emissions. Particulate matter from diesel engines exhausts are consists of some chemical components such as trace elements, carbon, inorganic organic carbon components, etc. [16,17]. These particles are harmfully affecting the environment and human health. The use of alternative fuels has established itself as an attractive solution to resolve the inconsistency between the scarcity of oil resources and the necessity to increase energy while at the same time reducing pollutant emissions [4,18].

Scientists and researchers have conducted numerous experimental studies to identify alternative fuels that could address environmental concerns. Potential solutions include biodiesel derived from vegetable oils, bio alcohols such as methanol, propanol, ethanol, and butanol sourced from corn, biomass from organic material, biogas from sources like swamps and landfills, natural gas from fossil fuels, dimethyl ether (DME), hydrogen (H_2), oxyethylene ethers (OMEs) and other options yet to be discovered in the future [19–27]. Among these alternative fuels, natural gas (NG) is extremely favorable and appealing in transportation [19–21]. First of all, NG is available at attractive prices in wide regions of the world [28–30]. In addition to natural gas and oil fields, NG industries produce this gas from gradually more demanding resources, for example, coal-bed methane, sour gas, tight gas, shale gas, methane gas hydrate [28], Power-to-Gas (P2G) and Power-to-Methane (P2M) technologies [31,32]. Moreover, even though NG's major component is methane (CH_4), which is a greenhouse gas, natural gas is still favorable as a gaseous fuel due to low C/H. So, it helps in reducing CO_2 emissions owing to its higher hydrogen to carbon ratio than that of entirely fossil fuels. Moreover, NG could significantly decrease NO_x emissions besides producing almost zero particulate matter and smoke [33,34], which is difficult to achieve with conventional diesel engines. Third, by reason of its high methane percentage, NG is not prone to detonation under normal operation conditions. So the NG can be applied in CI engines that have a high compression ratio (CR) and release a brake thermal efficiency (BTE) higher than this one from normal fuel engines [28,29]. Despite the benefits of using NG, there are some drawbacks, such as slow-burning and poor lean combustion ability. These issues result in engines exhibiting longer burn times, lower power output, high cyclic variation, decreasing in-cylinder pressure (ICP) and heat release rate (HRR) [28,29]. Additionally, with NG, the HC and CO emissions increase more than 100 times, and the engine suffers from a low BTE at low and medium loads. Whereas at high engine loads the BTE is similar or slightly higher as compared with regular diesel fuel mode [28].

By reason of the disadvantages of using an NG–diesel fuel mode, many researchers looked for alternative fuels such as hydrogen (H_2) and others. Using H_2 as a secondary fuel in compression ignition (CI) engines is considered an effective contributor to the reduction of smoke, CO and CO_2 and HC levels, which could reach up to more than 50% under optimal operation conditions [35]. High H_2 amounts have a tendency to cause an obvious influence on combustion, like a sharp increase in HRR and BTE due to the high flame speed of hydrogen. Conversely, the combustion of H_2 with a higher energy content causes an increase in in-cylinder temperature, which leads to significant growth of NO_x, especially under conditions of high load and high in-cylinder pressure [35].

Therefore, to leverage the benefits offered by diesel, hydrogen, and natural gas and remove their disadvantages if used individually, scientists and researchers combined them together in HNG–diesel Tri-fuel mode. Blending natural gas with hydrogen presents a promising alternative to conventional fossil fuels, with potential reductions in overall pollutant emissions [29]. This approach offers a viable path toward sustainable transportation, especially considering the increasingly stringent emission standards in European countries. Hydrogen-enriched natural gas (HNG) combines the benefits of both hydrogen and natural gas while mitigating their drawbacks, leading to improved fuel efficiency, reduced emissions of CO, HC, and CO_2, and enabling lean burn [29,36,37]. Hydrogen's unique characteristics, including its wide flammability range, contribute to enhanced efficiency and lower toxic emissions, along with improved ignition delay and flame stability [29,36]. Adding hydrogen to natural gas can also result in shorter burning times, wider flammability limits, and expanded exhaust gas recirculation (EGR) capabilities while maintaining low cyclic variations and NO_x emissions [29,36]. Hythane is a mixture of hydrogen and natural gas that has been patented by Frank Lynch of Hydrogen Components Inc. USA for its energy content, enabling engine use without modifications [29,38]. Demonstrations of Hythane's efficacy have been conducted across North America, Europe, and Asia [29,39]. Even a small fraction of hydrogen (5–30% by volume) added to natural gas in internal combustion engines offers significant benefits, leveraging specific physical and chemical properties [29,38]. The hydrogen fraction in HNG is determined by the partial pressures of the two fuels in the tank, with the Wobbe index serving as a key indicator of gas composition's impact on engine performance. A constant Wobbe index ensures minimal fluctuations in air–fuel ratio and combustion rate despite changes in gas composition [29,38,40].

Despite the benefits of using a mixture of HNG (hydrogen and natural gas), scientific research on this gas mixture is still limited due to obstacles to the use of hydrogen. These obstacles are represented by the difficulty of producing hydrogen gas, the danger of storing it, and the limitations of its delivery pipelines and distribution stations, all of which cause its high price and limited availability. Therefore, scientific research on the use of HNG with other fuels is still continuing and increasing whenever one of the obstacles to the use of hydrogen is passed or overcome.

Numerous scientists and researchers have demonstrated through their studies that HNG-ICE (hydrogen-enriched natural gas internal combustion engine) stands as a globally eco-friendly fuel option for internal combustion vehicles. The potential for HNG-ICE appears promising, provided there is a cost-effective means to produce hydrogen and a supply infrastructure akin to that of crude oil products, ensuring seamless overall engine performance [4].

Recently, some numerical and experimental scientific studies have been conducted on HNG–diesel Tri-fuel engines. These studies analyzed the effect of the fuel mixture ratio (hydrogen, natural gas, and diesel), fuel supplying methods and conditions, engine details and operation conditions on the combustion, performance and exhaust emissions of HNG–diesel engines. Out of these studies, there are a few that have discussed the exhaust emissions, and it showed that switching from NG-diesel and H_2-diesel dual-fuel mode to HNG–diesel Tri-fuel mode plays a vital role in exhaust emissions behavior.

Due to the novelty of the HNG–diesel subject and continued studies on it, the previous literature studies like [4,6,29,38,41–43] on the HNG–diesel fuel mod are excellent and continue, but they have not included almost studies and do not cover all aspects, especially the exhaust emissions. In addition, there is a conflict in some of the results reached by researchers in terms of the effect of a certain variable (or parameter) on the exhaust emissions and pollutants in HNG–diesel Tri-fuel engines. Therefore, this is the apt time to prepare a review study summarizing the latest findings of the HNG–diesel engine studies in terms of exhaust emissions. This study will be a summary showing the interrelationship between the influences and emotions results in HNG–diesel engines to serve as a guide for researchers and companies to use.

The primary goal of this research is to review studies related to the use of hydrogen-enriched natural gas (HNG) in diesel engines, besides the studied cases on dual-fuel modes, even though diesel–H_2 or diesel–NG. This literature review primarily examines how engine operating parameters and fuel blends affect emission characteristics. The research methodology includes summarizing the impacts of engine design, operating conditions, fuel mixing ratios, and supply techniques on CO, CO_2, NO_x, and HC emissions individually.

2. Emission Characteristics

Engine emissions that have been analyzed in this review study are carbon monoxide (CO), carbon dioxide (CO_2), Hydrocarbons (HC), nitrogen oxide (NO_x), and particulate matter (PM). Engine details, operation conditions, and fuel blends for the cited studies have been detailly included in Table 1.

2.1. Carbon Monoxide (CO)

Carbon monoxide (CO) is a product of incomplete combustion of hydrocarbon fuel. The mean causes of CO generation are suddenly too low reaction mixture temperature, sudden lag of oxidant [44], too short reaction time, incomplete combustion [45,46], rich air–fuel mixture, lean mixture at low combustion temperature, high C/H of fuel [44,47,48], and trapped of fuel in crevices [28,49]. Moreover, HRR [50], air–fuel mixture homogeneity [51], air–fuel ratio [45,46,52], charge temperature [53], and injection configuration inside the combustion chamber also directly affect CO emissions [54]. Figure 1 displays the findings of some researchers regarding the results of CO emissions relative to some influences like HNG–diesel fuel blends, engine speed, load and compression ratio, exceeds air ratio, diesel injection pressure and timing, and intake temperature.

Arslan and Kahraman [55] found that CO emissions were significantly higher at 1500 rpm for [diesel + (500 g/h) NG], [diesel + (500 g/h) (90%NG + 10%H_2)], and [diesel + (500 g/h) (80%NG + 20%H_2)] compared to other mixtures. At 30.61 Nm load, CO emissions for [diesel + (500 g/h) NG] were 13% and 27% difference from [diesel + (500 g/h) (90%NG + 10%H_2)] and [diesel + (500 g/h) (80%NG + 20%H_2)], respectively. The lowest CO emissions were observed with [diesel + (250 g/h) NG]. Higher CO emissions were linked to a 500 g/h flow rate and 20% H_2 addition due to ignition delay and incomplete combustion. This trend was consistent at both 1500 and 1750 rpm, though other emissions were lower.

Zhou et al. [56] observed that CO emissions, a result of incomplete CH_4 combustion in diesel-CH_4 operation, increased sharply to 3.1, 4.5, 5.7, 5.6, and 4.1 times higher than diesel fuel across five loads. Despite improved combustion efficiency at 70–90% engine loads, CO emissions remained high. In contrast, CO emissions from diesel–H_2 operation decreased to 0.6, 0.5, 0.6, 0.6, and 0.7 times those of diesel due to hydrogen's substitution of carbon content. Akansu et al. [57] noted that adding hydrogen to methane reduces incomplete combustion and CO emissions, which decrease further with higher hydrogen fractions. For five loads, CO emissions for 100%CH_4, 30%H_2–70%CH_4, 50%H_2–50%CH_4, 70%H_2–30%CH_4, and 100% H_2 were 4.58, 3.92, 3.50, 2.94, and 0.74 times that of diesel. Using 70%H_2–30%CH_4 reduced CO emissions by 35.8% compared to diesel–CH_4. Schefer et al. [58] observed that adding hydrogen to methane increased OH, H, and O radicals, enhancing lean-burn methane combustion stability and reducing CO emissions.

Alrazen et al. [59] found that CO emissions are mainly controlled by the excess air factor (λ). In the CNG–diesel mode, CO is a byproduct of incomplete combustion, and its mass fraction decreased from 0.0379 to 0.019 as λ increased from 1.2 to 2.4. In H_2–diesel mode, the CO mass fraction fell from 0.000338 to 0.000284 over the same λ range. The primary factor in reducing CO emissions is substituting hydrogen for diesel's carbon content. Adding hydrogen to CNG extends the flammability range and reduces incomplete combustion, thus lowering CO emissions. In H_2–CNG-diesel mode, the CO mass fractions for various H_2–CNG mixtures (0.037 for CNG, 0.022 for 70%CNG + 30%H_2, 0.0155 for 50%CNG + 50%H_2, 0.009 for 30%CNG + 70%H_2, and 0.0003 for H_2) decrease with increasing λ. Schefer et al. [58] observed that hydrogen addition increases OH, H, and O radicals, enhancing lean-burn methane combustion stability and reducing CO emissions.

Arat et al. [60] studied the effects of using 30% H_2 in HCNG at various CNG rates on a Tri-fuel diesel engine. They found that CO emissions were significantly higher in the Tri-fuel mode. A 75% diesel, 7.5% H_2, and 17.5% CNG mixture increased CO emissions by 12%, while a 50% diesel, 15% H_2, and 35% CNG mixture increased CO emissions by 8%. The rise in CO emissions was primarily due to incomplete combustion of CNG, especially with lower hydrogen content and the amount of pilot diesel used. Reducing the pilot diesel fuel could help lower CO emissions.

Tangöz et al. [61] conducted an experimental study on the effects of compression ratio (CR) and excess air factor (λ) on the performance and exhaust emissions of an HCNG–diesel engine. They found that when λ exceeds 1.1, CO values are below the Euro-VI standard (4.0 g/kWh) [62]. Increasing λ significantly reduced CO values at a constant CR. The lowest CO values were observed with a CR of 12.5 and λ below 1.1. The impact of higher H_2 content in the HCNG mixture on CO emissions was unclear [61].

Benbellil et al. [63] found that CO emissions are higher for HNG–diesel than diesel alone. Increasing H_2 content in HNG at low to intermediate loads raises CO emissions due to lean mixtures reducing combustion efficiency. A 50% H_2 mixture increases CO by 55%, 26%, and 34% at 80%, 60%, and 20% engine loads, respectively. Higher hydrogen content improves combustion efficiency, converting CO to CO_2 and reducing CO emissions. Combining hydrogen with methane extends methane's flammability range and enhances lean-burn combustion stability, lowering CO emissions.

Quadri et al. [64] observed the highest CO emissions in pure diesel mode compared to HCNG mixtures. CO emissions decreased with HCNG mixtures due to lower carbon content and better combustion efficiency, increasing oxygen fraction and promoting CO oxidation. At 220 bar, improved fuel–air mixing reduced CO emissions but at 240 bar, emissions increased due to poor mixing and insufficient combustion time. The highest CO emissions (0.075% by volume) were at 200 bar in pure diesel mode, while the lowest (0.023%) were at 220 bar with a 20% HCNG substitution.

De Simio and Iannaccone [65] found that CO emissions were higher for tri-fuel cases compared to full diesel (FD), but advancing diesel pilot injections reduced CO emissions. Early pilot injection (SOPI) led to stable, low CO levels similar to those with advanced injection timing and conventional combustion. Advancing injection timing increases combustion temperatures, improving oxidation but potentially causing incomplete combustion if too advanced, as seen in DF–CNG points with 50 Nm at earlier SOPI.

Tutak et al. [66] showed that adding H_2 to compressed natural gas (CNG) speeds up combustion in dual-fuel engines, improving efficiency by reducing heat loss. However, exceeding 20% hydrogen energy fraction increases cylinder pressure and combustion instability. Switching from pure diesel to gaseous fuel significantly reduced CO emissions from 1.8% to 0.08%, with H_2 replacing CNG having no significant impact on CO emissions.

Mansor et al. [67] reported peak CO increases of 800% for 40% diesel, 660% for 50% diesel, and 520% for 60% diesel in diesel-CH_4 operations. CO emissions rise with higher methane content due to its slow flame speed but decrease with more hydrogen, which enhances methane's flammability and reduces CO emissions. Diesel combustion results in low CO_2 and CO emissions, further reduced by adding hydrogen. Higher diesel fractions might reduce CO emissions due to CO_2 dissociation into CO at high hydrogen content.

Mansor et al. [68] noted that CO formation indicates incomplete combustion due to poor mixing, slow flame speed, and dissociation. Pure diesel had a CO mass fraction of 0.004 while adding methane (60D–40CH_4) raised it to 0.025. Reducing methane and adding hydrogen decreased CO peaks to 0.024, 0.015, 0.013, and 0.006 for 28%, 20%, 12%, and 0% methane, respectively. Higher intake temperatures increased CO peaks but shortened formation duration. Methane's slower flame speed causes higher CO, mitigated by hydrogen's faster flame speed, with CO peaks initially rising with methane but dropping with more hydrogen.

Zhao et al. [69] found that increasing the proportion of H_2 reduces CO production in a naturally aspirated engine. Higher H_2 levels lead to lower CO emissions as the engine power increases, primarily because H_2 combustion only produces water, unlike diesel and methane, which emit CO and CO_2. Replacing some natural gas with hydrogen can thus reduce greenhouse gas emissions.

Ekin et al. [70] showed that adding H_2 to an NG–diesel dual-fuel engine significantly decreases CO emissions by 86% in Mode 1 and 80% in Mode 2. Hydrogen's high combustion efficiency and rapid flame speed enhance the overall combustion process, reducing incomplete combustion and CO emissions.

Lounici et al. [71] observed that CO emissions are generally reduced by enriching NG with H_2 due to improved fuel utilization and combustion.

Reddy et al. [72] found that H_2-enriched CNG–diesel dual-fuel engines produce lower CO emissions across all engine compression ratios (CRs). CO emissions decrease as CR increases, with significant reductions noted at higher H_2 percentages in the HCNG mixture. The CO emission values of 42.6, 41, 40, 43.5 and 38.3 g/kW-h, respectively, for 5%, 10%, 15% and 20%H_2 in HCNG mixture at CR of 16.5 compared to 109.3 g/kW-h with the diesel–CNG dual-fuel mode.

Luo et al. [73] noted that switching from 100% diesel to a mix of 80%diesel and 20%H_2 slightly decreases CO emissions while switching to 50%diesel and 50%NG slightly increases CO due to NG's slower flame speed. Adding H_2 to a diesel-NG mix at 100% engine load reduces CO emissions by 17.1%.

Pichayapat et al. [74] reported a 12.97% reduction in average CO emissions with a diesel-HCNG dual-fuel mode compared to pure diesel, except at low engine speeds where emissions were similar.

Cameretti et al. [75] found that CO emissions increased from 226 ppm in a H_2–diesel scenario to 860 ppm with CH_4–diesel due to incomplete CH_4 combustion, which is more prevalent with less H_2. H_2 improves combustion efficiency and reduces carbon-based emissions, but less H_2 leads to higher CO emissions due to less efficient combustion.

Yin et al. [76] investigated diesel–CH_4–H_2 Tri-fuel engine combustion and emissions. They studied three scenarios: (1) 1200 rpm, 0–60% H_2, 108 kPa, 7 BTDC; (2) 1500 rpm, 0–60% H_2, 108 kPa, 12 BTDC; (3) 1200 rpm, 0–60% H_2, 120 kPa, 7 BTDC. CO emissions stemming from incomplete combustion were highest in Case 3 due to lower combustion temperatures. Increasing the hydrogen energy fraction (HEF) reduced CO emissions, especially in Case 2, due to lower methane content and higher in-cylinder temperatures enhancing CO oxidation.

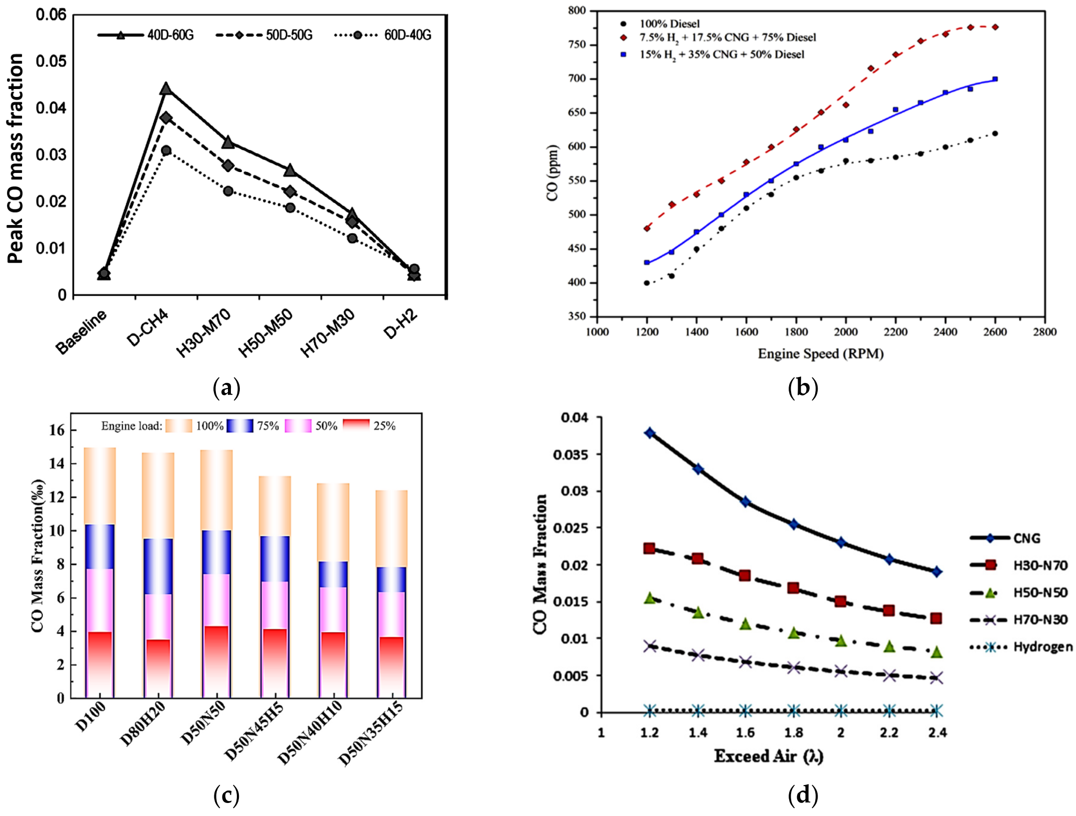


Figure 1. *Cont.*

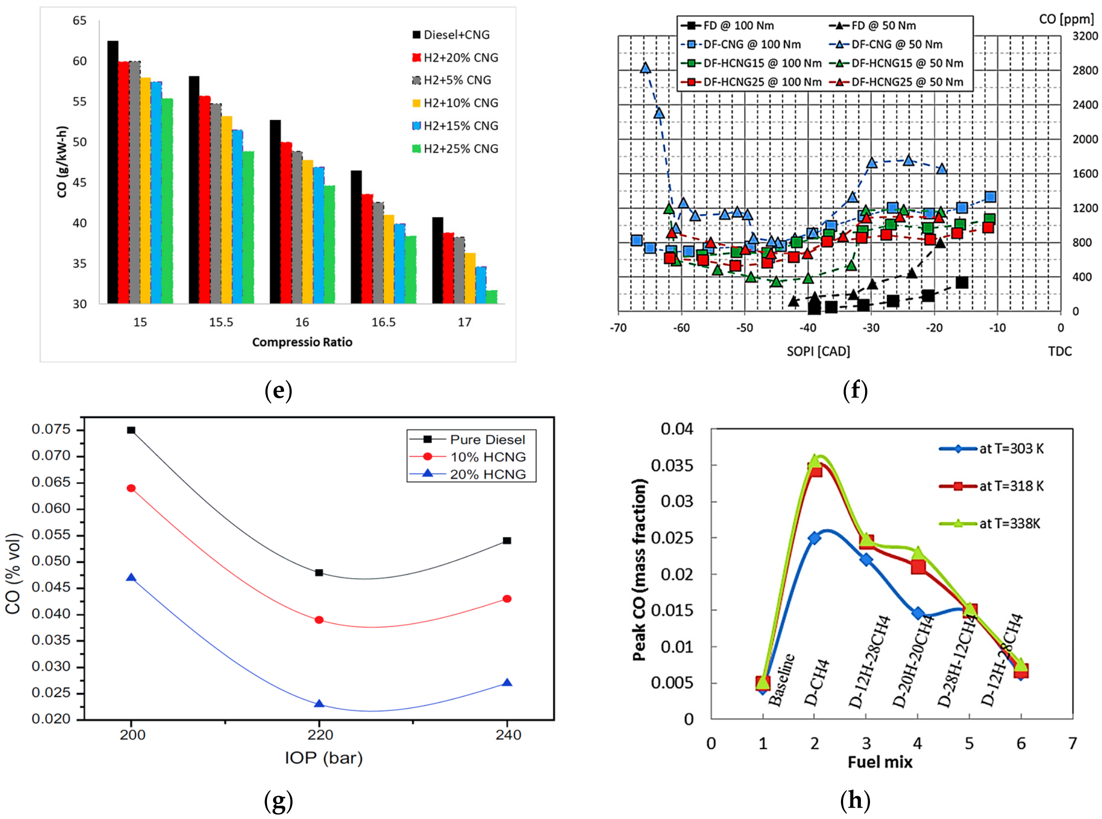


Figure 1. CO emissions identified in previous experimental and numerical studies. (**a**) Peak CO emissions versus diesel–hydrogen–methane mixture combinations [67]. (**b**) CO emissions versus engine speed for various diesel–CNG-H_2 fuel blends [60]. (**c**) CO emissions with a range of engine loads and fuel mixtures [73]. (**d**) CO emissions peak with various exceeds air ratios and fuel mixtures [59]. (**e**) CO emissions are related to engine compression ratios and fuel blends based on [72]. (**f**) CO emissions related to SOPIs at 100 and 50 Nm in full diesel (FD) and dual-fuel (DF) modes with different MFs [65]. (**g**) CO emissions related to injection operation pressures (IOPs) and HCNG substitutions [64]. (**h**) CO emissions with various values of Ti and fuel blends [68].

Summary of CO Findings

The summary of carbon monoxide (CO) emissions, their causes, and solutions highlights several key points:

- Different fuel mixtures affect CO emissions differently. Higher NG content can increase CO levels due to longer ignition delays, while hydrogen reduces CO emissions by substituting carbon, improving combustion efficiency, flame speed, and oxidation rates. The impact of fuel mixtures is shown in Table 1 and Figure 1a;
- CO emissions generally rise with engine speed (Figure 1b) and decrease with engine load (Figure 1c) under stable conditions. Variations are summarized in Table 1;
- Increasing the excess air factor (λ) reduces CO emissions by improving combustion, as shown in Table 1 and Figure 1d;
- Higher compression ratios (CRs) enhance fuel-air mixing and reduce CO emissions, although excessively high CRs can worsen mixing and increase CO emissions. Effects are detailed in Table 1 and Figure 1e;
- Injection Timing and Pressure: Advanced injection timing improves combustion and reduces CO, though too early timing can increase CO. Increased diesel injection pressure initially decreases then increases CO due to its effect on air–fuel mixing and combustion, as shown in Table 1 and Figure 1f,g;
- CO emissions vary with fuel composition. Hydrogen-enriched fuels typically produce lower CO emissions compared to pure diesel or NG;

- Techniques like low-temperature combustion (LTC) and adjusting intake temperature can reduce CO emissions by improving fuel-air mixing and combustion efficiency.

In summary, reducing CO emissions involves optimizing combustion processes, enhancing fuel properties, and refining engine design. Strategies like hydrogen enrichment, advanced injection timing, and improved engine conditions are crucial but require more comprehensive research, especially for HNG–diesel Tri-fuel engines.

2.2. Carbon Dioxide (CO_2)

In an IC engine, carbon dioxide (CO_2) is primarily formed through the combustion process when hydrocarbon fuels react with oxygen in the air. This combustion reaction results in the oxidation of carbon (C) in the fuel, forming CO_2, and the oxidation of hydrogen (H) in the fuel, forming water vapor (H_2O). These reactions occur within the combustion chamber of the engine at high temperatures and pressures [77]. The main significant influencers on the CO_2 emissions are air–fuel ratio [78,79], fuel injection timing [45,52,80–82], fuel injection pressure [81,83], in-cylinder temperature besides the carbon content of the fuel [28], compression ratio [84,85], engine speed [52,80,82,86], engine load [45,52,80–82,84,87], and fuel composition and share percentage [45,52,80,82,84,86,87]. Figure 2 illustrates the findings of some researchers regarding the results of CO_2 emissions relative to some influences, including HNG–diesel fuel blends, engine speed and load, exceeds air ratio, and diesel injection timing.

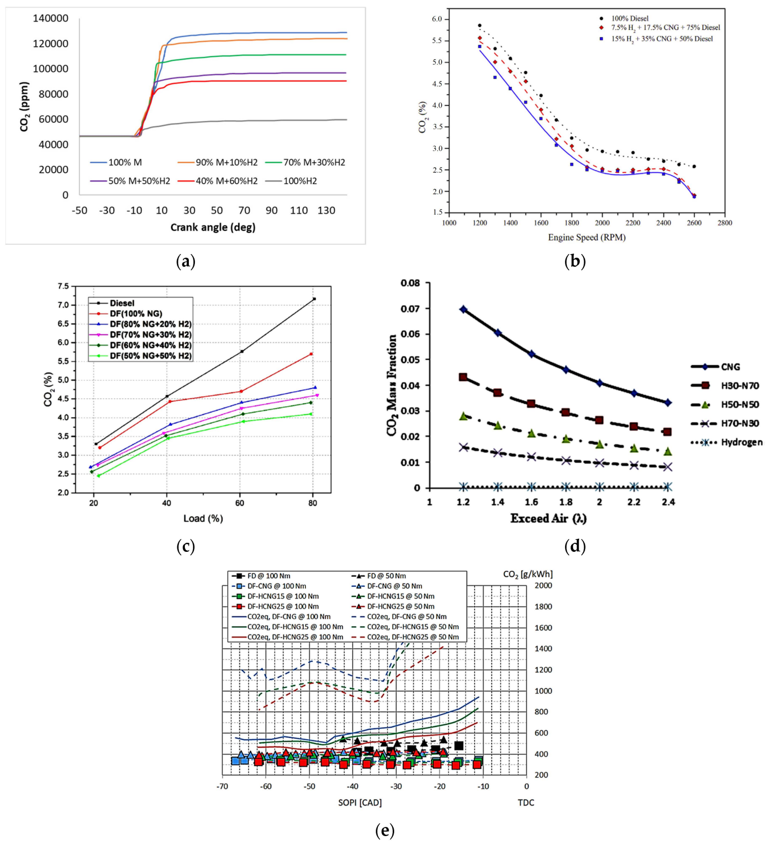


Figure 2. CO_2 emissions results reported in accomplished studies. (**a**) CO_2 emissions versus hydrogen percentages in diesel–hydrogen–methane mixture [75]. (**b**) CO_2 emissions versus engine speed for various diesel–CNG–H_2 fuel blends [60]. (**c**) CO_2 emissions variation related to engine load and various fuel mixtures [63]. (**d**) Peak CO_2 emissions are relevant to various exceeds air ratios and gaseous fuel mixtures [59]. (**e**) CO_2 emissions related to SOPIs at 100 and 50 Nm in full diesel (FD) and dual-fuel (DF) modes with different MFs [65].

Arslan and Kahraman [55] reported that CO_2 emissions increased with rising fuel consumption and engine load. However, CO_2 levels were lowest at both 1500 and 1750 rpm and all load conditions for a diesel (500 g/h) + (90%NG + 10%H_2) mixture, indicating better combustion with this fuel mix.

Alrazen et al. [59] found that in the CNG–diesel mode, CO_2 decreased progressively with higher excess air (λ) levels, from 0.0695 at λ 1.2 to 0.0331 at λ 2.4. For diesel–H_2 operation, CO_2 decreased from 0.000494 to 0.000476 at these λ levels. CO_2 reductions were notable with CNG, 30%H_2–70%CNG, 50%H_2–50%CNG, 70%H_2–30%CNG, and pure H_2 added to diesel, with 70%H_2–30%CNG showing the greatest reduction. Hydrogen addition to CNG consistently reduced CO_2 emissions and formation regions due to lower carbon content.

Arat et al. [60] observed that fuel mixtures of 75% diesel—17.5% CNG—7.5% H_2 and 50% diesel—35% CNG—15% H_2 reduced CO_2 by 11.4% and 16.7%, respectively, compared to diesel alone. This reduction is attributed to methane's incomplete combustion and lower carbon content when using H_2 and CNG.

In Benbellil et al. [63], at low loads (20–40%), CO_2 emissions from NG and diesel are similar due to limited gaseous fuel use. At higher loads, increasing NG reduces CO_2 emissions by up to 20% at 80% engine load compared to diesel. This is because NG, mainly composed of CH_4, has a lower carbon content. Adding hydrogen (H_2) to NG further decreases CO_2 emissions, with a 50% H_2 mixture reducing emissions by 43% compared to diesel and 28% compared to pure NG. The study shows a linear decrease in CO_2 with higher H_2 proportions in HNG mixtures.

De Simio and Iannaccone [65] found that CO_2 emissions are always lower in dual-fuel cases than in a diesel mode, though equivalent CO_2 can be higher. The lowest values were obtained at medium and low loads using the LTC combustion regime. Without post-treatment, equivalent CO_2 was close to diesel at 100 Nm for [PD + (75%CNG + 25%H_2)]. The method of H_2 production is critical for effective CO_2 reduction.

Tutak et al. [66] realized that switching from diesel to gaseous fuel reduced CO_2 emissions by 6.47% to 9.16%. Replacing CNG with H_2 further decreased CO_2 emissions, with a 21% H_2 energy fraction leading to a 20% reduction due to substituting carbon fuel with hydrogen.

Zhao et al. [69] observed that CO_2 production decreases as the hydrogen fraction increases. In naturally aspirated engines, the lambda parameter decreases with increasing power. CO_2 emissions are complex but highest when using 100% CH_4. Higher hydrogen content significantly reduces CO_2 emissions, which is crucial for low-carbon shipping, as hydrogen produces only water upon burning. Replacing some natural gas with hydrogen cuts greenhouse emissions.

Lounici et al. [71] found that using natural gas in dual-fuel engines reduces CO_2 emissions, and adding hydrogen, a carbon-free fuel, further lowers CO_2 emissions. However, better combustion can sometimes increase CO_2 emissions by converting CO to CO_2, but overall emissions remain similar to natural gas dual-fuel.

Luo et al. [73] detected that switching from 100% diesel to a mix of 80% diesel and 20% hydrogen significantly reduces CO_2 emissions, while a mix of 50% diesel and 50% natural gas slightly increases them. Increasing hydrogen content in HNG–diesel Tri-fuel significantly decreases CO_2 emissions, especially at high engine loads.

Cameretti et al. [88] showed that using hydrogen and methane blends in a diesel engine significantly reduces CO_2 emissions due to hydrogen's carbon-free nature and high combustion efficiency.

Cameretti et al. [75] found that using hydrogen in a medium-speed marine engine reduces CO_2 emissions by up to 54%, meeting strict greenhouse gas requirements.

Summary of CO_2 Findings

The key points of carbon dioxide (CO_2) emissions findings could be summarized as:

- Blending hydrogen with diesel or natural gas (NG) lowers CO_2 emissions compared to using pure fuels, as illustrated in Table 1 and Figure 2a;
- Hydrogen in fuel blends improves combustion efficiency and produces only water vapor, eliminating CO_2 emissions from carbon-based fuels;
- CO_2 emissions vary with engine load and speed. At low loads, hydrogen and NG blends reduce CO_2 more than diesel, while at higher loads, these blends may increase CO_2 emissions due to greater fuel consumption, as shown in Table 1 and Figure 2c. Engine speed generally reduces CO_2, as indicated in Table 1 and Figure 2b;
- Low-temperature combustion (LTC) and optimized combustion parameters (air–fuel ratio, injection timing) can further reduce CO_2 emissions. Advancing diesel injection timing and using leaner mixtures lower CO_2, as summarized in Table 1 and shown in Figure 2d,e.

Overall, hydrogen-enriched fuel blends present a promising method for reducing CO_2 emissions from internal combustion engines. However, there is a need for more comprehensive studies on these effects and strategies, as current research is limited.

2.3. Hydrocarbons (HC)

Hydrocarbon emissions stem from unburned fuels due to insufficient temperatures near the cylinder wall, where the air–fuel mixture is cooler than at the cylinder center [89–91]. These emissions comprise various species like alkanes, alkenes, and aromatics, usually measured as CH_4 equivalents [90,92]. Diesel engines typically produce low hydrocarbon levels, especially under light loads due to lean air–fuel mixing. In lean mixtures, slow flame speeds can prevent complete combustion during the power stroke, leading to higher hydrocarbon emissions [90,93]. Factors such as fuel type, engine tuning, and design influence diesel hydrocarbon content, with irregular operating conditions further affecting emissions. Rapid engine speed changes, improper injection, large nozzle cavity volumes, and injector needle bounce can cause unburned fuel to enter the exhaust [90,94]. If the exhaust temperature exceeds 600 °C and oxygen is present, hydrocarbons continue reacting, reducing tailpipe emissions compared to those exiting the cylinder [90,95]. Hydrocarbons are emitted not only from vehicle exhaust but also from the crankcase, fuel system, and atmospheric venting during fuel handling [90,95]. Crankcase and evaporative emissions contribute 20–35% and 15–25%, respectively, while tailpipe emissions account for 50–60% of total hydrocarbon emissions [90,96]. Figure 3 illustrates some researchers' findings on HC emissions, considering various factors such as HNG–diesel fuel blends, engine speed, load and compression ratio, excess air ratio, diesel injection pressure and timing, and intake temperature.

Arslan and Kahraman [55] observed that HC emissions decreased with increasing engine load. The highest HC emissions occurred with [diesel (500 g/h) + (90%NG + 10%H_2)] at 1500 rpm, while the second highest was with [diesel (500 g/h) + (80%NG + 20%H_2)] due to incomplete combustion of the increased NG. Increasing the speed to 1750 rpm reduced HC emissions, with a 67% decrease for [diesel (250 g/h) + (80%NG + 20%H_2)] at 40.81 Nm load due to better combustion.

Zhou et al. [56] realized that HC emissions for CH_4, 70%CH_4 + 30%H_2, 50%CH_4 + 50%H_2, 30%CH_4 + 70%H_2, and H_2 were 12.0, 10.3, 9.0, 7.7, and 0.8 times higher than diesel, respectively. At 10, 30, and 50% loads, these values averaged 6.7, 4.8, 2.6, 1.6, and 0.8. Diesel–CH_4–H_2 fuels produced 14.6–76.8% lower HC emissions than diesel–CH_4 fuels, with greater reductions at higher loads. Adding H_2 to CH_4 improved HC emissions by enhancing CH_4 combustion efficiency and reducing unburned CH_4.

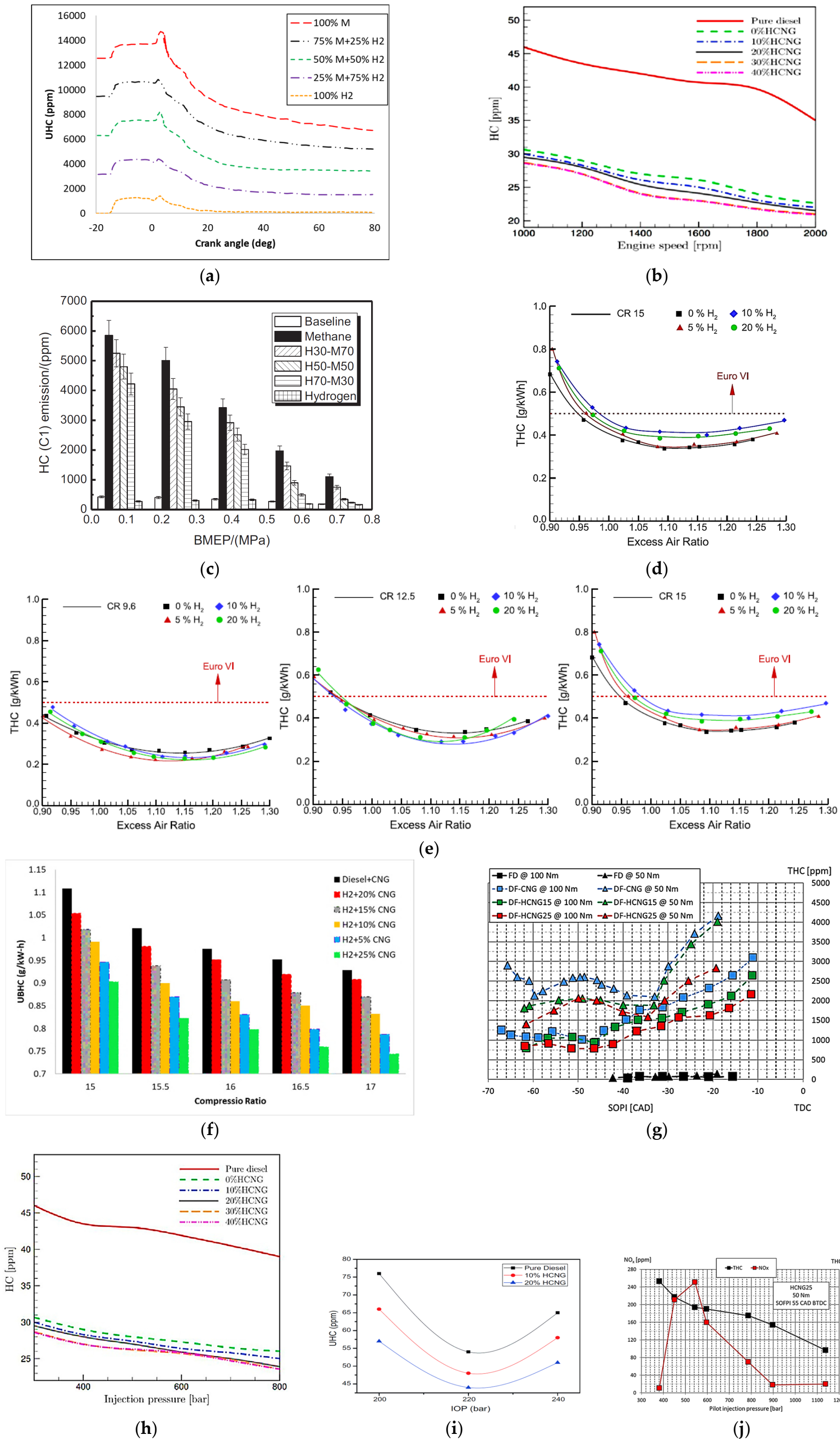


Figure 3. HC results in previously accomplished studies. (**a**) UHC pollutant versus hydrogen percentages in diesel-hydrogen-methane mixture [88]. (**b**) HC emissions are related to engine speeds and various fuel mixtures [97]. (**c**) Total HC (C1) emission related to engine BMEP with various hydrogen-methane adding [56]. (**d**) THC emission values versus excess air ratios and H_2-CNG blends [61]. (**e**) THC emissions values vs. excess air ratios with different compression ratios and H_2-CNG blends [61]. (**f**) UBHC emissions related to engine compression ratios and fuel blends, according to Reddy et al. [72]. (**g**) THC emissions related to SOPIs at 100 and 50 Nm in full diesel (FD) and dual-fuel (DF) modes with different MFs [65]. (**h**) HC emissions with 1200 rpm engine speed and various diesel injection pressures [97]. (**i**) UHC emissions related to injection operation pressures (IOPs) and HCNG substitutions [64]. (**j**) Trade-off between THC and NO_x in low-temperature combustion (LTC) regime [65].

Zareei et al. [97] reported that diesel engines emit more HC than HCNG engines due to incomplete combustion. HCNG with hydrogen increases combustion temperatures, reducing HC emissions. Higher engine speeds and injection pressures further decrease HC emissions, with 40% hydrogen content in HCNG showing significant reductions compared to pure diesel and hydrogen-free CNG.

Tangöz et al. [61] detected a decrease in THC emissions between λ of 1.1 and 1.15, then an increase beyond 1.15. THC emissions were below the Euro-VI standard [62], which is 0.5 g/kWh for a gaseous fuel engine. Emissions were 0.26–0.39 g/kWh for various diesel–CNG and diesel–H_2–CNG blends at λ of 1.15. Higher compression ratios increased THC emissions due to more unburned gas in the combustion chamber. The effect of hydrogen on THC emissions remains unclear.

Benbellil et al. [63] found that adding H_2 to NG in dual-fuel (DF) mode increases HC emissions at low engine loads compared to pure NG. At 20% load, HC emissions were 15% higher with a 50% H_2 mix than with pure NG. However, as engine load increases, HC emissions decrease, with a 50% H_2 mix reducing HC emissions by 45% at 80% load.

Quadri et al. [64] noticed that increasing hydrogen in HCNG reduces unburned hydrocarbons (UHC) due to higher brake thermal efficiency (BTE). At 220 bar, 10% and 20% HCNG mixes showed minimal UHC due to faster combustion, though UHC slightly increased at 240 bar.

De Simio and Iannaccone [65] found that a very lean air/gaseous fuel mixture increased THC emissions despite excess oxygen. Increasing injection advance raises combustion temperature, improving oxidation until LTC mode stabilizes emissions. Higher H_2 content reduced THC emissions more effectively than CNG substitution, likely due to more complete combustion from H_2's lower ignition energy at both medium and low loads.

Tutak et al. [66] realized that switching from diesel to compressed natural gas (CNG) increased THC emissions by 2.5 times due to the "crevice effect", where unburnt gaseous fuel remains in small combustion chamber gaps. Adding hydrogen (H_2) to CNG speeds up gas combustion in a dual-fuel engine, reducing heat loss and improving efficiency. With a 19% H_2 energy fraction, THC emissions drop to about 100 ppm, comparable to conventional engines. However, exceeding 19% H_2 causes a slight increase in THC emissions due to combustion instability and knocking.

Ekin et al. [70] found that HC emissions were significantly influenced by fuel composition and injection timing. Adding hydrogen to a natural gas–diesel dual-fuel mixture greatly reduced HC emissions. This reduction is due to hydrogen's higher flame speed and wider flammability limits, which enhance combustion and reduce unburned hydrocarbons. In Mode1 ([25% diesel + (65%NG + 10%H_2)]), an 89% reduction in HC emissions was achieved with a 50% natural gas and 25% hydrogen mixture at 14° CA BTDC. In Mode2 ([25% diesel + (75%NG + 20%H_2)]), a 76% reduction was achieved with 75% natural gas and 15% hydrogen. The improvements are due to hydrogen ensuring more complete combustion and reducing areas where combustion does not occur.

Lounici et al. [71] observed that THC emissions follow the same trend as CH_4 emissions in NG-diesel dual-fuel mode since THC emissions are mainly CH_4. The pilot fuel's contribution to THC emissions is minimal. Adding H_2 to NG reduces THC emissions by improving combustion and decreasing NG presence in H_2–NG blends. However, the reduction in mixture proportions makes the impact of H_2 enrichment on THC reduction less clear.

Reddy et al. [72] found that diesel–CNG dual-fuel enriched with H_2 emits slightly lower UBHC emissions than diesel–CNG dual-fuel alone due to an improved burn-up rate. UBHC emissions decrease with increasing compression ratio (CR) in both engine operations. For example, UBHC emissions at a CR of 16.5 are 0.8, 0.85, 0.88, 0.92, and 0.76 g/kW-h for 5%, 10%, 15%, 20%, and 25% H_2 in HCNG mixture, respectively, compared to 0.952 g/kW-h for diesel–CNG dual-fuel mode.

Pichayapat et al. [74] realized that the average HC emissions significantly decreased by 15.84% across all engine speeds (from 800 rpm to 4000 rpm) when operating under diesel-HCNG dual-fuel mode compared to running on diesel alone.

Cameretti et al. [88] found that reducing hydrogen (H_2) in the fuel mixture increased hydrocarbon (HC) emissions due to more unburned methane. With 100% hydrogen and diesel, HC emissions were lower as the only carbon source was diesel. Higher methane content led to incomplete combustion, lower temperatures, and thus higher HC emissions.

Cameretti et al. [75] observed that increasing hydrogen in the methane-hydrogen blend decreased unburned hydrocarbon (UHC) emissions in a medium-speed marine engine. This reduction is due to hydrogen's higher reactivity and burning velocity, improving combustion and reducing unburned fuel.

Summary of HC Findings

Based on extensive studies on hydrocarbon (HC) emissions in combustion processes across various fuels and conditions, several key conclusions emerge:

- In Table 1 and as could be seen in Figure 3b,c, HC emissions decrease with increased engine speed and load, improving combustion efficiency;
- As can be noted in Table 1 and seen in Figure 3a, higher natural gas (NG) content often increases HC emissions, but hydrogen (H_2) addition generally reduces them by promoting a more complete combustion;
- Better combustion efficiency, aided by faster flame propagation and enhanced oxidation, lowers HC emissions. Hydrogen enrichment and combustion stability are crucial for this;
- From Figure 3h, higher injection pressures usually reduce HC emissions by improving fuel atomization, though excessive pressure can affect air–fuel mixing and lead to varying HC levels (Figure 3i). Optimal injection timing enhances combustion efficiency and controls HC levels (Figure 3g);
- Higher CRs generally reduce HC emissions by improving fuel-air mixing (Figure 3f), though excessively high CRs can worsen mixing and increase emissions (Figure 3e);
- HC emissions generally decrease with a higher excess air factor (λ) (Figure 3d), indicating better combustion. However, in some engine designs and fuel mixtures, further increasing λ can slightly raise HC emissions;
- Adding hydrogen reduces HC emissions (Figure 3a), especially at low engine loads, though its effect depends on operating conditions and fuel composition;
- Different combustion modes and techniques, such as low-temperature combustion and dual-fuel operations, affect HC emissions. Advanced injection timing and techniques improve efficiency and reduce HC formation (Figure 3j);
- Higher exhaust gas temperatures and leaner mixtures generally lower HC emissions by promoting better combustion.

In summary, controlling HC emissions involves optimizing fuel composition, injection timing, compression ratio, and combustion efficiency. Effective strategies include hydrogen enrichment, advanced combustion techniques, and fine-tuning engine conditions. Further research is needed to enhance combustion efficiency and reduce HC emissions.

2.4. Nitrogen Oxide (NO_x)

Nitrogen oxide (NO_x) is a grouped emission comprising approximately 90% nitrogen monoxide (NO) and the rest of nitrogen dioxide (NO_2). The formation of NO in the combustion involves five typical mechanisms, namely thermal NO, prompt-NO, oxidation of HCN, oxidation of HNCO, and the N_2O and NNH mechanisms [98]. According to Wei and Geng [28], in the thermal mechanism, the formation of NO is affected by in-cylinder temperature and O_2 concentration. NO formation takes place when the temperature is higher than about 1800 K and the formation rate increases together with increasing of in-cylinder temperature. In the prompt mechanism, nitrogen monoxide (NO) is generated through a process instigated by intermediate hydrocarbon fragments, notably CH and

CH_2, which arise from fuel combustion. These fragments interact with nitrogen (N_2) within the combustion chamber, resulting in the formation of compounds containing both carbon and nitrogen (C–N). Following this, these C–N-containing compounds undergo reactions with oxygen (O_2) via diverse pathways, ultimately resulting in the generation of NO [28]. Figure 4 depicts some research findings regarding NO_x emissions, taking into account several variables, including blends of HNG–diesel fuel, engine speed, load and compression ratio, excess air ratio, diesel injection pressure and timing, as well as intake temperature.

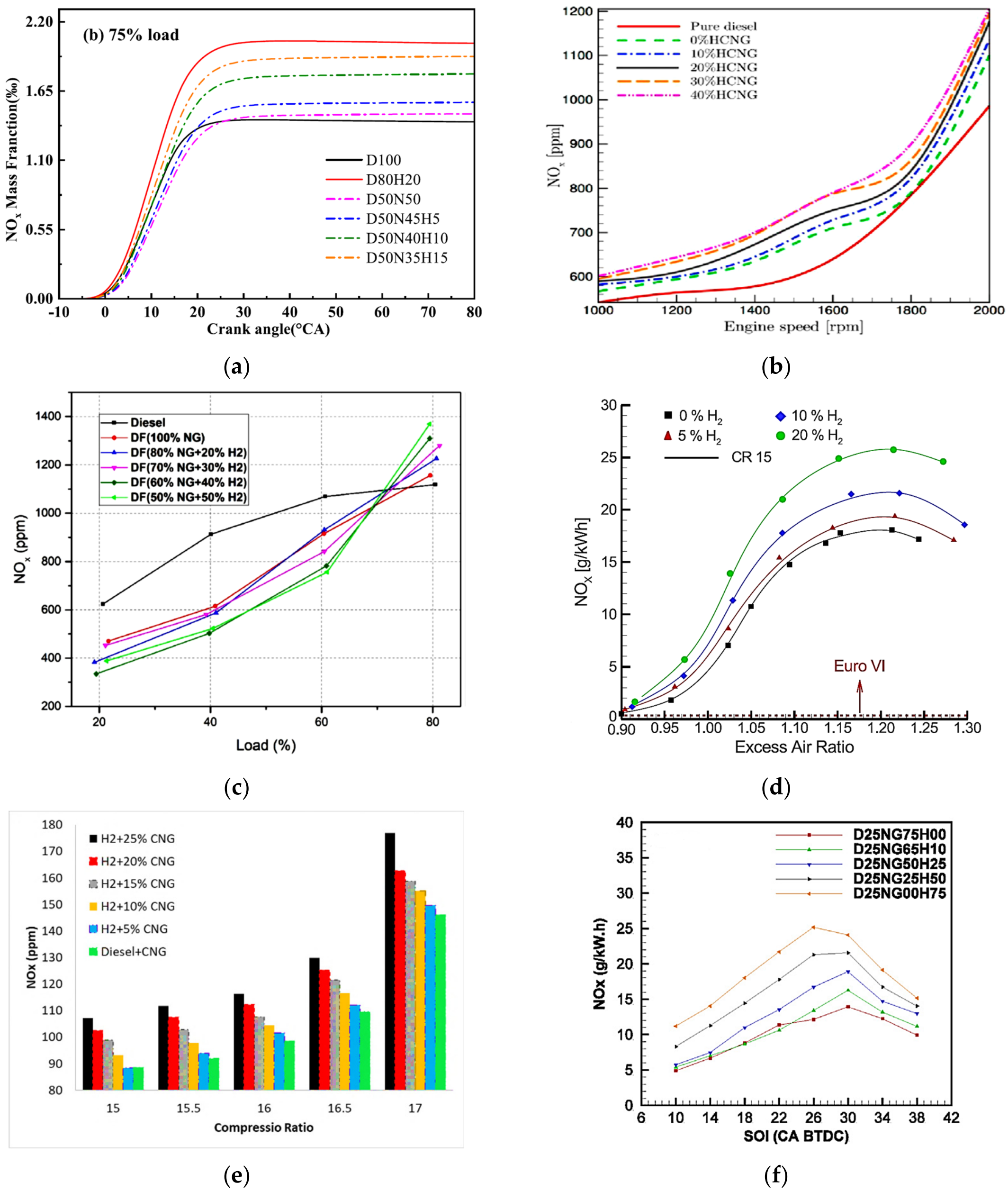
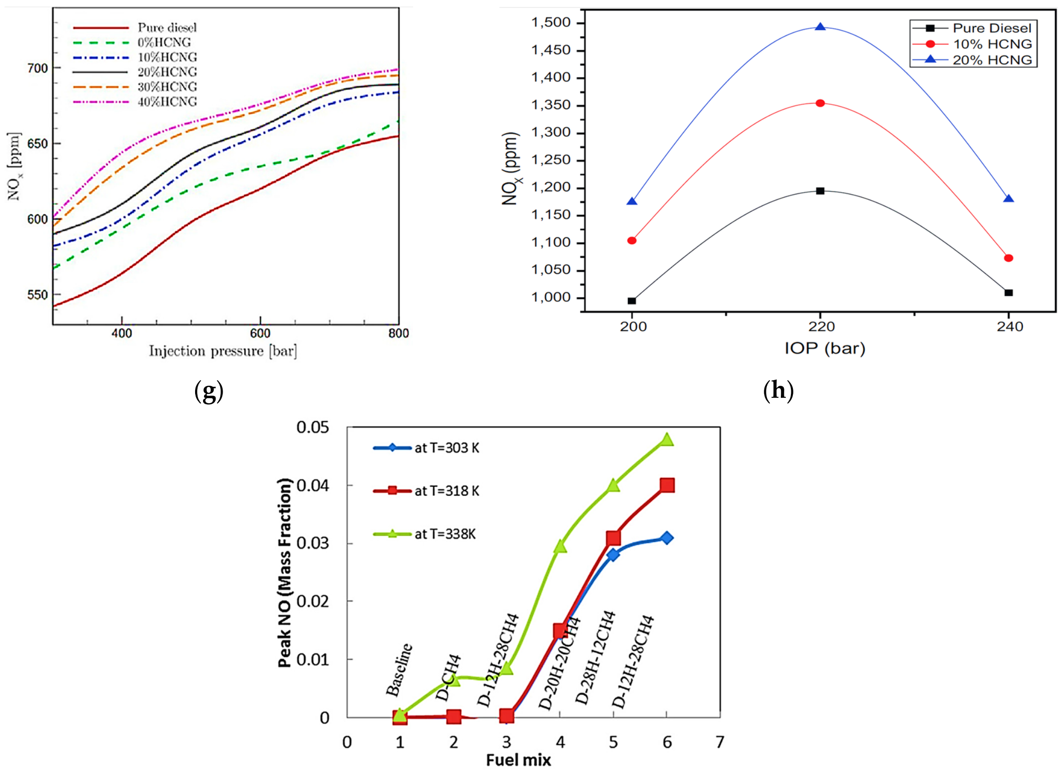


Figure 4. *Cont.*



(**g**) (**h**)

(**i**)

Figure 4. NO_x findings from literature involved studies. (**a**) NO_x emissions with 75% engine load and various fuel blends [73]. (**b**) The amount of engine NO_x emissions at various engine speeds and fuel mixtures [97]. (**c**) NO_x emissions variation at several engine loads and fuel mixtures [63]. (**d**) NO_x emissions related to versus values of excess air ratio and HCNG fuel blends [61]. (**e**) NO_x emission variation related to compression ratio values and used fuel mixtures based on [72]. (**f**) NO_x emission related to SOI at various fuel blends [70]. (**g**) NO_x emissions with 1200 rpm engine speed and various diesel injection pressures [97]. (**h**) NO_x emissions related to injection operation pressures (IOPs) and HCNG substitutions [64]. (**i**) NO_x emissions of various values of Ti and fuel blends [68].

Arslan and Kahraman [55] found that low temperatures resulted in low NO_x emissions at 1500 and 1750 rpm and no-load conditions. The lowest NO_x emissions were observed with [diesel + (500 g/h) NG] at high engine loads, with a slight decrease in NO_x levels as rpm increased. High NO_x emissions persisted for [diesel + (500 g/h) NG], [diesel + (500 g/h) (90%NG + 10%H_2)], and [diesel + (500 g/h) (80%NG + 20%H_2)], especially at high loads, with a 13% increase at 1750 rpm for [diesel + (500 g/h) (80%NG + 20%H_2)].

Zhou et al. [56] reported that a diesel–CH_4 mixture decreased NO_x by 8.0% and 8.9% at 10% and 30% loads but increased by 10.7% and 32.6% at 70% and 90% loads compared to diesel alone. With diesel–H_2, NO_x emissions rose due to higher in-cylinder pressure and combustion temperatures, increasing by 16.8%, 38.1%, and 50.2% at 50%, 70%, and 90% loads. Law and Kwon [99] noted that CH_4 lowers H_2–air flame temperatures, reducing NO_x formation. Zhou et al. [56] found that [diesel + (30%H_2 + 70%CH_4)] emitted 14.1%, 16.5%, and 46.4% less NO_x than diesel, [diesel + CH_4], and [diesel + H_2], respectively, across five loads. Increasing H_2 in H_2–CH_4 mixtures raised NO_x levels due to higher burning rates.

Alrazen et al. [59] found that [diesel + (30%H_2 + 70%CNG)] had lower NO emissions than [diesel + (50%H_2 + 50%CNG)], [diesel + (70%H_2 + 30%CNG)], and diesel-H_2 at all excess air ratios (λ). At λ = 1.2, NO increased with H_2 addition: 1.67*10^−6 (CNG), 0.00088 (30%H_2 + 70%CNG), 0.0034 (50%H_2 + 50%CNG), 0.01 (70%H_2 + 30%CNG), and 0.0131 (H_2). Similar trends were observed at λ = 1.4, 1.8, 2, 2.2, and 2.4. H_2 increased the air–fuel mixture burning rate, raising NO emissions, with peak NO values rising as the H_2 fraction increased. Methane can lower H_2–air flame temperatures, reducing NO formation.

Arat et al. [60] found that using a 30% H_2 mixture in H_2–CNG fuels significantly reduces NO_x emissions. Fuel mixtures of [(75% diesel) + (17.5% CNG + 7.5% H_2)] and [(50% diesel) + (35% CNG + 15% H_2)] resulted in 28% and 47.2% NO_x reduction, respectively, due to the slower flame speed and longer ignition delay of CNG.

Zareei et al. [97] noted that nitrogen oxides are a major concern for CNG and HCNG fuels compared to diesel. Adding hydrogen to CNG reduces carbonyl emissions but increases NO_x emissions by 16.25% compared to diesel and 9.95% compared to regular CNG due to higher combustion temperatures. Higher injection pressures (40:1 ratio) also correlate with increased NO_x emissions.

Tangöz et al. [61] realized that the maximum NO_x values of 18, 19.8, 21.5, and 25.5 g/kWh for CNG and various H_2–CNG mixtures (5%, 10%, 20% H_2) at λ of 1.15 and CR of 9.6. Higher CRs (12.5 and 15) and λ ratios (1.16 and 1.21) led to maximum NO_x values exceeding the Euro VI standard (0.4 g/kWh) [62] for a gas engine. NO_x emissions increased with more hydrogen due to higher flame temperature and speed but decreased with λ ratios above 1.15 for all fuels. For (10% H_2 + 90% CNG), NO_x values were 4, 21.5, and 16.7 g/kWh at λ of 0.96, 1.15, and 1.3 for CRs of 9.6, 12.5, and 15, respectively. Adding hydrogen to CNG increases NO_x emissions.

In Benbellil et al. [63], NO_x emissions in dual-fuel operation were up to 60% lower than in diesel and pure NG or HNG modes at low and moderate loads. At 20% engine load, NO_x emissions were 620, 470, and 390 ppm for diesel, pure NG, and 50% H_2 blend modes, respectively. This reduction is due to the lower combustion rate from the slow flame speed of CNG, leading to longer ignition delay and lower combustion temperature. However, NO_x emissions were higher in DF mode, especially at high loads and with H_2 addition, due to improved combustion. Higher CH_4 content in H_2 mixtures led to greater NO_x reduction.

Quadri et al. [64] found that increasing injection opening pressure to 220 bar raised NO_x emissions by 20.1%, 22.6%, and 27.06% for pure diesel, [pilot diesel + (90%CNG + 10%H_2)], and [pilot diesel + (80%CNG + 20%H_2)], respectively, compared to 200 bar. This rise is due to the higher adiabatic flame temperature of H_2. At 240 bar, NO_x emissions decreased by 15.4%, 20.8%, and 20.96% for the same mixtures compared to 220 bar.

De Simio and Iannaccone [65] observed significant NO_x reductions at low engine load for Tri-fuel compared to diesel mode. At medium load, NO_x levels remained high despite a reduction trend due to LTC, especially with higher H_2 content. Advanced injection in diesel mode increased temperature peaks and NO_x emissions, similarly in Tri-fuel, but LTC and higher advance injection reduced NO_x close to zero at low load.

Tutak et al. [66] reported a 27% reduction in NO_x emissions when switching from diesel to CNG. Adding H_2 increased NO_x emissions due to higher combustion pressure and temperature, crucial factors in NO_x formation. The increase was highest with 19% H_2 content.

Mansor et al. [67] results show that NO emissions are higher in diesel-CH_4 than in diesel operation. Increasing H_2 in H_2–CH_4 mixtures raises NO emissions, peaking with diesel–H_2. CH_4 lowers combustion temperatures. NO emissions increased by 1500%, 400%, and 100%, with 40%, 50%, and 60% diesel, respectively. Lower NO emissions occur with [diesel + (70%CH_4 + 30%H_2)] than with diesel + H_2. Diesel's lower combustion temperature restrains NO formation, while higher H_2 content increases it. The equivalence ratio affects NO, peaking at stoichiometric values. The highest ratios are with (50%H_2 + 50%CH_4), but temperature is the dominant factor.

Mansor et al. [68] observed that at Ti = 303 K, NO_x emissions were under 0.0002 for baseline, [60%diesel + 40%CH_4], and [60%diesel + (28%CH_4 + 12%H_2)]. Above 20% H_2, NO_x exceeded 0.01, peaking at 0.033 for [60%diesel + 40%H_2]. With 12% and 20% CH_4 at 318 K, NO_x was 0.001 to 0.004. At Ti > 338 K, only baseline fuel had NO_x < 0.001; others ranged from 0.005–0.048. NO_x rises linearly with H_2 content. Ti and H_2 significantly affect

NO_x due to higher injected charge energy. NO_x emissions increase with Ti, largest at 338 K. H_2's higher specific energy and flame speed influence combustion more than CH_4.

Zhao et al. [69] observed that increasing H_2 levels raises NO emissions while lowering NO_2 emissions. NO_x concentration rises with engine output but remains mostly unaffected by varying H_2 proportions. As output power increases, differences in NO emission concentrations from different mixture ratios decrease. Higher temperatures increase NO emissions. H_2 affects NO_x emissions by (1) raising cylinder temperature, which boosts NO_x production, and (2) shortening combustion time and producing water vapor, which suppresses NO_x production.

Ekin et al. [70] results show that NO_x emissions increased by 12% and 11% in a dual-fuel engine using hydrogen and natural gas for Mode 1 ([25%diesel + (50%NG + 25%H_2)]), PDIT 14° BTDC) and Mode 2 ([25%diesel + (75%NG + 15%H_2)]), PDIT 10^0 BTDC), respectively. The rise is due to higher cylinder temperatures from hydrogen combustion. To reduce NO_x, the study suggests optimizing the hydrogen-to-natural gas ratio and adjusting diesel injection timing. Mitigation methods include Exhaust Gas Recirculation (EGR), various diesel injection strategies, and injecting water vapor. These strategies aim to maintain high performance while reducing NO_x emissions [70].

Lounici et al. [71] findings indicate that NO_x emissions are lower in the NG–diesel dual-fuel mode than in conventional diesel at low to moderate loads. Adding H_2 to NG slightly increases NO_x emissions due to better combustion, but the effect is minimal due to the low charge temperature at these loads.

Reddy et al. [72] observed that NO_x emissions increase with higher cylinder compression ratios (CR). Due to in-cylinder temperature differences, CNG–diesel dual-fuel results in lower NO_x emissions compared to hydrogen addition. At a CR of 16.5, NO_x emissions were 112.3, 116.3, 121.6, 125.2, and 130.1 ppm for 5%, 10%, 15%, and 20% H_2 in HCNG, respectively, compared to 109.3 ppm for the diesel–CNG dual-fuel mode.

Luo et al. [73] found that switching from 100% diesel to dual-fuel (80% diesel + 20% H_2, 50% diesel + 50% NG) or Tri-fuel (50% diesel + 35% NG + 15% H_2) modes increases NO_x emissions, except at 100% load. At 75% load, NO_x emissions increased by 43.7%, 2.8%, and 34.5% for the respective fuel mixtures. Higher H_2 content raises NO_x emissions due to increased in-cylinder temperatures and faster combustion, enhancing NO_x formation during the premixed combustion stage. At high loads, NO_x is concentrated at the bottom and middle of the combustion chamber, while at low loads, it is mainly at the front.

Pichayapat et al. [74] findings indicated that switching from diesel to diesel–HCNG fuel reduces average NO_x emissions by 1.16%. NO_x emissions rise at 1000–2500 RPM but decrease at 2600–4000 RPM due to faster combustion reducing nitric oxide in the exhaust.

Cameretti et al. [88] results of the CFD analysis show that NO_x levels drop as hydrogen in the fuel mix decreases. NO_x levels were recorded as 2.66, 2.16, 1.50, 1.96, and 0.45 units for various hydrogen and CNG mixtures, with higher hydrogen content causing higher combustion temperatures and NO_x production.

Cameretti et al. [75] Numerical analysis on a marine engine shows NO_x emissions rise by up to 76% when hydrogen replaces methane due to higher combustion temperatures and thermal diffusivity. This increase breaches NO_x limits, indicating the need to balance hydrogen content for optimal emissions reduction.

Yin et al. [76] reported that Increasing hydrogen energy fraction (HEF) from 0 to 10, 20, 40, and 60% raises NO_x emissions in all scenarios. Higher HEF increases peak combustion temperatures and residence times, leading to more NO_x emissions.

Summary of NO_x Findings

The reviewed studies provide a thorough understanding of NO_x emissions and their dependency on factors such as fuel composition, engine load, combustion temperature, and equivalence ratio. Here is a summary:

- Natural Gas (NG) and Diesel Blends: Mixing NG with diesel reduces NO_x emissions at low to moderate loads but increases them at higher loads, especially with more NG (Figure 4a,c);
- Hydrogen (H_2) Addition: Adding hydrogen raises NO_x emissions due to higher combustion temperatures and faster-burning rates, varying with hydrogen percentage and blend components (Figure 4a);
- NO_x emissions rise with engine load and speed, particularly with significant hydrogen content. However, they vary based on fuel composition and combustion characteristics at different loads (Figure 4b,c). One study shows a decrease in NO_x with increasing speed;
- Higher combustion temperatures, especially with hydrogen-rich fuels, increase NO_x emissions. Longer residence times also raise NO_x levels with higher hydrogen fractions;
- Adjusting hydrogen-to-natural gas or diesel ratios and diesel injection timing can mitigate NO_x emissions. Lowering peak in-cylinder temperatures through combustion parameter adjustments is effective (Figure 4f);
- Higher injection pressures generally increase NO_x due to higher combustion temperatures (Table 1, Figure 4e,g). However, very high pressures may decrease NO_x by reducing combustion efficiency (Figure 4h);
- Adjusting the air excess ratio affects NO_x formation, with peak NO_x values increasing and then decreasing as the ratio rises (Figure 4d);
- Many studies compare NO_x emissions with Euro VI standards, indicating a need for further reductions;
- Higher intake temperatures increase combustion temperatures, leading to higher NO_x emissions (Figure 4i).

Overall, while NG and hydrogen blends offer environmental benefits, managing NO_x emissions is challenging. Optimizing fuel composition, engine parameters, and combustion characteristics is crucial for lower NO_x levels and maintaining engine performance. Recommended strategies include water vapor injection, EGR, and various diesel injection techniques to improve emissions, including NO_x. Further studies are needed to combine these techniques for more effective NO_x reduction.

2.5. Particulate Matter (PM)

In internal combustion (IC) engines, particulate matter (PM) emissions primarily consist of smoke and soot particles. Smoke is composed of visible particulates larger than 1 micron in diameter, while soot consists of finer particles, typically less than 1 micron in size [100]. These emissions are formed during the combustion process when fuel-rich regions or incomplete combustion occur due to insufficient oxygen availability or improper mixing of fuel and air [77,101]. Moreover, high sulfur content also can contribute to the formation of PM emissions [102]. Soot particles are predominantly composed of carbonaceous materials that aggregate to form larger particles that can be emitted as smoke. The formation of PM emissions is influenced by various factors such as engine operating conditions (e.g., load, speed), fuel type and quality, combustion chamber design, combustion timing and techniques, and after-treatment systems [103]. Effective control measures include optimizing combustion parameters, improving fuel quality, employing particulate filters or other exhaust treatment technologies, and using alternative fuels [103,104]. Figure 5 illustrates the conclusions drawn by various researchers on the impact of different factors such as HNG–diesel fuel blends, engine speed, load, compression ratio, and diesel injection timing on PM emissions.

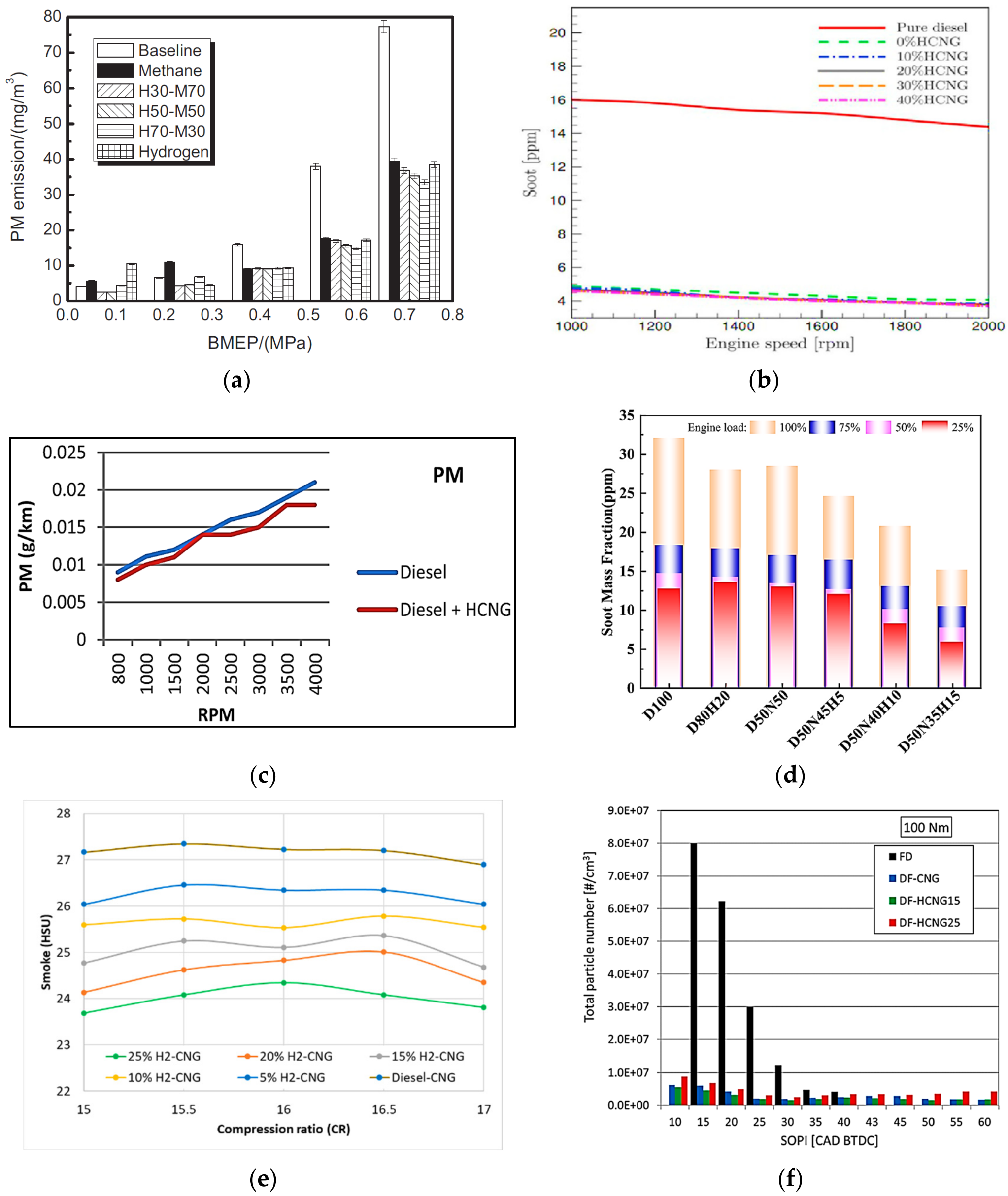


Figure 5. PM, PN, Smoke, and Soot emissions findings from literature-involved studies. (**a**) Effect of H_2-CH_4 addition on particulate emission with various BMEP [56]. (**b**) Soot emissions in various fuel blends and engine speeds [97]. (**c**) Comparison of particulate matter (PM) emissions between diesel and diesel blended with HCNG [74]. (**d**) Soot emission related to fuel blends and engine loads [73]. (**e**) Smoke emission related to compression ratios and used fuel blends based on [72]. (**f**) Impact of SOPI on total particle number at 100 Nm in diesel, dual, and Tri-fuel modes with various blends [65].

Reddy et al. [72] reported that increasing hydrogen in fuel reduces smoke opacity due to hydrogen's cleaner combustion. Smoke opacity values for 5%, 10%, 15%, and 20% hydrogen additions are 26.33, 25.78, 25.37, 25, and 24.08 HSU, respectively, at a 16.5 compression ratio, compared to 27.2 HSU for diesel–CNG dual-fuel.

De Simio and Iannaccone [65] reached the fact that without advanced diesel injection, most fuel in full diesel (FD) mode burned through diffusion combustion, unlike in dual-fuel (DF) mode where gas was the primary fuel. With earlier injections, pre-mixed combustion was dominant, reducing particle numbers (PN) to DF mode levels. Thus, both particle number and mass in FD mode were much higher than in DF mode near the top dead center (TDC). In DF mode, particle concentration was less affected by diesel injection timing due to minimal diesel use, reducing diffusion flame.

Ekin et al. [70] results show that by including hydrogen in the gas mixture, 78% and 84% improvement were achieved in soot emissions, respectively, with Mode1 (([25%diesel + (50%NG + 25%H_2)]) and PDIT of 14^0 BTDC), and Mode2 (([25%diesel + (75%NG + 15%H_2)]) and PDIT of 10^0 BTDC) conditions.

Luo et al. [73] found that switching from 100% diesel to 80% diesel–20% H_2, 50% diesel–50% NG dual-fuel, or diesel–NG–H_2 ternary fuel reduces soot emissions at medium and high loads but increases soot emissions at low loads for 80% diesel–20% H_2 and 50% diesel–50% NG. Gas fuels affect soot formation through thermal effects, reaction rate changes, and chemical interactions. Net soot formation is influenced by temperature-dependent rates of soot initiation and oxidation. At low loads, reduced soot oxidation can lead to higher soot emissions. Increasing the H_2 ratio in 50% diesel–50% NG reduces soot, with 50% diesel–35% NG–15% H_2 lowering soot by 52.3%, 56.5%, and 54.5% at 25% load compared to 100% diesel, 80% diesel–20% H_2, and 50% diesel–50% NG.

Zareei et al. [97] detected that NG fuel and its H_2 blends produce significantly less soot than pure diesel. Diesel, with its lower self-ignition temperature and higher cetane number, has a shorter ignition delay than HCNG, leading to higher soot levels due to insufficient mixing time. HCNG, with 40% hydrogen, cut soot emissions by 73.24% compared to diesel and by 7.09% compared to CNG. Tests at 1200 and 2400 rpm with various injection pressures, maintaining a 40:1 AFR, showed that higher injection pressures reduce soot emissions for all fuels.

Zhou et al. [56] found that adding hydrogen (H_2) and methane (CH_4) slightly increases particulate emissions at low loads due to reduced local oxygen pressure and incomplete diesel combustion. This trend is similar to hydrocarbon emissions. Replacing diesel with H_2 and CH_4 reduces soot precursor formation and diesel particulates. At medium to high loads (50%, 70%, 90%), particulate emissions decreased by 48.6% and 48.7% for diesel-H_2 and diesel-CH_4, respectively. Combining H_2 and CH_4 further reduces particulates, with the best ratio at 70% H_2–30% CH_4. Hydrogen addition shifts methane oxidation to a lower carbon pathway, reducing soot formation.

Pichayapat et al. [74] findings indicate that found that PM emissions dropped by 9.14% on average at speeds from 800 to 4000 rpm when using diesel with HCNG instead of pure diesel. The study concludes that adding HCNG to diesel engines with minimal pilot diesel significantly reduces emissions. It suggests HCNG is viable for diesel engines with proper engine selection and HCNG diesel dual-fuel system design. Results show that the HCNG–diesel Tri-fuel system can reduce emissions below EURO IV standards and conventional diesel levels, with PM emissions specifically dropping by 9.14% across the 800 to 4000 rpm range.

Summary of Particulate Matter (PM) Emissions

Particulate matter (PM) emissions can be summarized as follows:

- Adding hydrogen (H_2) to diesel–CNG fuel blends lowers smoke opacity (see Figure 5e) due to hydrogen's cleaner combustion. Smoke opacity decreased from 26.33 HSU at 5% H_2 to 24.08 HSU at 20% H_2, compared to 27.2 HSU for diesel–CNG dual-fuel [72];
- In full diesel (FD) mode without advanced injection, higher particle emissions occur due to diffusion combustion. Dual-fuel (DF) mode, with gas as the primary fuel, results in lower particle numbers (see Figure 5f). Early diesel injection in FD mode reduces particle numbers to levels similar to DF mode, but particle mass and numbers are highest near top dead center (TDC) injection timing [65];
- Blending diesel with hydrogen and methane affects soot emissions differently at various loads. At medium and high loads, D80$H_2$0 and D50N50 blends reduce soot emissions, but at low loads, they increase due to reduced soot oxidation rates (see Figure 5d). Increasing hydrogen content in D50N50 decreases soot emissions, with D50N35H15 reducing soot by over 50% at 25% load compared to pure diesel [73];
- Hydrogen–methane blends significantly reduce particulate emissions, especially at medium to high loads, with reductions of over 48% (see Figure 5a). Higher injection

pressures also reduce soot emissions consistently across different fuels and engine speeds. HCNG blends, particularly with 40% hydrogen, significantly reduce soot compared to pure diesel and CNG [56];

- The relationship between PM and engine rpm is complex, influenced by various factors, as shown by differing behaviors of soot (see Figure 5b) and PM emissions (see Figure 5c).

Overall, integrating HCNG as a supplementary fuel in diesel engines, with proper system design, reduces particulate emissions below EURO IV standards, effectively mitigating soot and other emissions under varied engine loads and conditions.

2.6. Summary of Emission Characteristics

Table 1 summarizes recent studies on H_2–NG–diesel fuel mixtures and their effects on engine exhaust emissions, detailing engine specifics, operating conditions, fuel combinations, and emission outcomes. The review focuses on how hydrogen enrichment in natural gas affects emissions in HCNG–diesel Tri-fuel engines. Previous studies indicate that natural gas with diesel increases CO and HC emissions but decreases NO_x and CO_2, while hydrogen with diesel reduces CO, CO_2, and HC but increases NO_x. Combining hydrogen and natural gas with diesel in Tri-fuel engines improves heat release, in-cylinder pressure, and combustion, reducing CO, CO_2, HC, and NO_x compared to dual-fuel modes. Emissions in HNG–diesel Tri-fuel engines are influenced by engine power, speed, load, air–fuel ratio, excess air, fuel blends, compression ratio, intake temperature, fuel injection pressure, and timing. Table 1 presents these research findings:

- CO Emissions: Lowest in diesel and diesel–H_2 modes, highest in diesel-NG. Tri-fuel mode sees CO decrease with more H_2, load, power, and air–fuel ratio (λ) but increase with intake temperature and NG. CO emissions vary with compression ratio (CR) and fuel injection pressure;
- CO_2 Emissions: Unstable in diesel mode, decrease in tri-fuel mode with more hydrogen and λ, but increase with load and rpm. CR has a minor effect; CO_2 generally rises with output power;
- HC Emissions: Lowest in diesel and diesel-H_2 modes, highest in diesel-NG. Tri-fuel mode sees HC decrease with more H_2, load, power, and λ but increase with intake temperature and NG. HC emissions vary with CR and fuel injection pressure;
- NO_x Emissions: Increase with load, H_2, power, and intake temperature across all fuels, generally rising with λ and rpm. CR and injection pressure affect NO_x variably, with some studies showing lower NO_x at intermediate CR. High hydrogen content (70% H_2) reduces NO_x at intermediate to high loads, whereas NG-only blends result in the lowest NO_x emissions;
- Studies show that hydrogen in fuel blends significantly reduces soot and particulate emissions, improving combustion efficiency and reducing smoke opacity. Specific hydrogen, methane, and natural gas ratios in dual and ternary fuel systems significantly cut particulate matter, especially at medium to high loads, though low loads may see increased emissions due to reduced soot oxidation.

Overall, hydrogen-enriched fuel blends offer a promising solution to achieve lower emissions and meet stringent environmental standards.

Table 1. Summary of latest studies that investigated the use of HCNG as a mixture with diesel in various diesel engines. ▼: lowest, ▲: highest, ⇩: decreasing, ⇔: intermediate, ⇧: increasing, ↓: slightly decreasing, ↑: slightly increasing, ⇓: medium decreasing, ⇑: medium increasing, ⟱: highly decreasing, ⟰: highly increasing, DFM: diesel fuel mode, PD: pilot diesel fuel, rpm: revolution per minute, Fs: fuels, IP: injection pressure, NG: natural gas, CNG: compressed natural gas, H_2: hydrogen, λ: Excess air factor, S: stroke, CR: compression ratio, C: cylinder, DS: displacement, NM: Not mentioned, WC: water-cooled, φ: equivalent ratio, AC: air-cooled, T_i: intake temperature, LTC: low-temperature combustion, SOPI: start of pilot injection, ADI: advance diesel injection, OP: output power, IT: initial temperature, CO: carbon monoxide, CO_2: carbon dioxide, HC: Hydrocarbons, NO_x: nitrogen oxide, PM: particulate matter, PN: particulate number.

Ref	Engine Details	Engine Operation Conditions	Fuel Number ($Fn_{(1,2,...)}$) and Combinations	Engine Exhaust Emissions Results				
				CO	CO_2	HC	NO_x	Smoke, PM, PN, Soot
[55]	4S, DF, 3C, WC, CR = 22.8:1, DS = 1028 m^3	1. Speed (rpm) (rpm) = 1500, 1750 2. Loads (Nm) = 0.0, 10.2, 20.41, 30.61, and 40.81	1. DFM 2. PD + (250 g/h) NG 3. PD + (500 g/h) NG 4. PD + (250 g/h) [90% NG + 10%H_2] 5. PD + (500 g/h) [90% NG + 10%H_2] 6. PD + (250 g/h) [80% NG + 20%H_2] 7. PD + (500 g/h) [80% NG + 20%H_2]	▲ for F3, F5, and F7 with all loads and rpm_s. ⇔ for F2, F4, and F6, while the ▼ for F1 with all rpm_s and loads.	⟰ with ⇧ of load for all fuels and rpm. Approximately the same for all fuels, rpm, and loads except F5 show the ▼ value, ▲ for F1 with 1750 rpm.	⟰ with ⇧ of load with all fuels and rpm. ▲ for F3, F5, and F7, ⇔ for F2, F4, F6. ▼ for F1 at all rpm_s and loads.	⟰ with ⇧ of load for all fuels and rpm, but ↓ at full load and rpm of 1500 for all fuels.	NM
[56]	4S, DF, 4C, CR = 19:1, DS = 4334 cm^3	1. Speed (rpm) = 1800 2. Load % = 10, 30, 50, 70, and 90	1. DFM 2. PD + CH_4 3. PD + (30%H_2 + 70%CH_4) 4. PD + (50%H_2 + 50%CH_4) 5. PD + (70%H_2 + 30%CH_4) 6. PD+ H_2	▼ for F6 with all loads. ▲ for F2 with all loads. ▲ for all fuels at a load of 50%. ⇓ with ⇧ of H_2 for all loads.	NM	⟱ with ⇧ of load for all Fs. ⇓ with ⇧ of H_2 with all loads. ▲ for F2, and F7. ▼ for F1 and F6 with all loads.	⟰ with ⇧ of load for all fuels. For F3 to F6, ⇑ with ⇧ of H_2. ▲ for F6 at load 50 to 90%. ▼ for F3 at 10,30, 70% load and F1 at 90% load.	PM ⟰ with ⇧ of load for all Fs, except for F6, ↓ with ⇧ load from 10 to 30% and then ⇑. ▼ for F3 and F4 at 10% load, and significantly ▲ for F1 at load 50 to 90%. Approximately the same with F2 to F6 at 50% load. At 70 and 90% load, same for F2 and F6, less and continue ⇩ with ⇧ of H_2 for F3 to F5.
[59]	4S, DI DF, 1C, CR = 20.36:1, SV = 0.45 LIT	1. Speed (rpm) = 2000 2. Load (Nm) = 20.18 3. λ= (1.2), (1.4), (1.8), (2), (2.2) and (2.4)	1. PD + CNG 2. PD + (30%H_2 + 70%CNG) 3. PD + (50%H_2 + 50%CNG) 4. PD + (70%H_2 + 30%CH_4) 5. PD+ H_2	⇓ with ⇧ of λ for all fuels. ⟱ with ⇧ of H_2. ▲ for F1 and ▼ for F5 with all λ.	⇓ with ⇧ of λ for all fuels. ⟱ with ⇧ of H_2. ▲ for F1 and ▼ for F5 with all λ.	NM	⟱ with ⇧ of λ for all fuels. ⟰ with ⇧ of H_2. ▲ for F5 and ▼ for F1 with all λ.	NM
[60]	4S, DI Diesel, 4C, WC, CR = 17.5:1, DS = 3567 CC	1. Speed (rpm) = 1200–2600 with an interval of 100 rpm at WOT 2. Load % = 100 3. φ = 0.74, λ = 1.35, AFR = 21.04, For 30 HCNG, AFR = 16.4	1. DFM 2. (75%PD) + (17.5%CNG + 7.5%H_2) 3. (50%PD) + (35%CNG + 15%H_2)	⟰ with ⇧ of rpm for all fuels. ▲ for F2, ⇔ for F3, and ▼ for F1 with all rpm.	⟱ with ⇧ of rpm for all Fs. ↓ with ⇧ of H_2. ▲ for F1, ⇔ for F2, and▼ for F3 with all rpm.	NM	⟱ with ⇧ of rpm for all fuels. ⇓ with ⇧ of H_2. ▲ for F1, ⇔ for F2, and▼ for F3 with all rpm.	NM

Table 1. *Cont.*

Ref	Engine Details	Engine Operation Conditions	Fuel Number ($Fn_{(1,2,...)}$) and Combinations	Engine Exhaust Emissions Results				
				CO	CO_2	HC	NO_x	Smoke, PM, PN, Soot
[97]	4S, Diesel, 3C, CR = 17:1, DS = 3.2 L	1. Speed (rpm) = 1200 and 2400 2. Load % = 100 3. AFR = 40, 45, 50, 55, 60, and 65 4. IP (bar) = 300 to 800	1. DFM 2. PDF + CNG 3. PD + (90%CNG + 10%H_2) 4. PD + (80%CNG + 20%H_2) 5. PD + (70%CNG + 30%H_2) 6. PD + (60%CNG + 40%H_2)	NM	NM	⇓ with ⇧ of rpm for all Fs. ⇓ with ⇧ of H_2 for all rpm. ▼ for F6 and ▲ for F1 with all rpm.	⇑ with ⇧ of rpm for all Fs. ⇑ with ⇧ of H_2 for all rpm. ▼ for F1 and ▲ for F6 at all rpm.	Soot ↓ with ⇧ of rpm for all Fs. ⟱ with F2 to F6 in compare with F1 for all rpm. ↓ with ⇧ of H_2 for F2 to F6 with all rpm. ▼ for F6 and ▲ for F1 at all rpm.
[61]	4S, Modified Diesel, 4C, CR = 17.5:1, DS = 3.9 L	1. Speed (rpm) = 1500 2. Load % = 100 3. λ = 0.9, 1.1, 1.15, 1.2, 1.25, and 1.3 4. CR = 9.6, 12.5, and 15	1. PD + CNG 2. PD + (95%CNG + 5%H_2) 3. PD + (90%CNG + 10%H_2) 4. PD + (80%CNG + 20%H_2)	⟱ with ⇧ of λ from 0.9 to 1.05 and almost stable for λ from 1.05 to 1.3 with all CRs. Close values and same behavior for all Fs. ▼ with CR of 12.5 for all Fs.	NM	⇑ with ⇧ of CR for all fuels. ▼ for F1 and F2, ⇔ for F4, and ▲ for F3 with all λ_s when CR = 15.	▼ for F1 for all cases. ⟰ with ⇧ of H_2 for all CRs. ⟰ with ⇧ of λ until λ = 1.15, then ⇓ with ⇧ of λ for all Fs and CRs. ▼ for all Fs and λ_s with CR = 12.5.	NM
[63]	4S, DI Diesel, 1C, AC, CR = 18:1, DS = 630 cm^3	1. Speed (rpm) = 1500 2. Load % = 20, 40, 60, and 80	1. DFM 2. PD + CNG 3. PD + (80%CNG + 20%H_2) 4. PD + (70%CNG + 30%H_2) 5. PD + (60%CNG + 40%H_2) 6. PD + (50%CNG + 50%H_2)	▼ for F1 at all loads. ⟱ with ⇧ of load for F3-F6, and ⇑ with ⇧ of load for fuels 1 and 2. ▲ for F5 and ▼ for F1 at loads of 20 and 40%. ▲ for F2 and ▼ for F6 at loads of 60 and 80%. ⟱ with ⇧ of H_2 at loads of 60 and 80%. ⟰ with F2 to F6 compared with F1.	⟰ with ⇧ of load for all fuels. ⇓ with ⇧ of H_2 for all loads. ▼ for F6, and ▲ for F1 with all loads. ▼ with F2 than F1, but ▲ with F2 than other Fs.	▼ for F1 at all loads. ⇑ and then ⇓ with F2 to F6 when ⇧ load. ▲ for F5 and ▼ for F1 at loads of 20%. ▲ for F3 and ▼ for F6 at load of 40%. ▲ for F2 and ▼ for F6 at loads of 60 and 80%. ⇓ with ⇧ of H_2 at loads of 60 and 80%.	⟰ with ⇧ of load for all fuels. ▼ for F5 and ▲ for F1 at loads of 20 and 40%. ▲ for F1 and ▼ for F6 at load of 60%. ▲ for F6 and ▼ for F1 and ⇑ with ⇧ of H_2 at 80% loads.	NM
[65]	4S, Diesel, 4C, CR = 18:1, DS = 1910 cm^3	1. Speed (rpm) = 2000 rpm 2. Load (Nm) = 50, and 100 3. AFR = 14.5, 15.9, 16.5, 17.4 4. SOPI [CA] BTDC = 15, 35, 55 5. IP (bar): 380, 450, 550, 620, 800, 900, 1000, 1150	1. DFM 2. PD + CNG 3. PD + (85%CNG + 15%H_2) 4. PD + (75%CNG + 25%H_2)	Non-linear relation with SOPI. ▲ for F2 and ▼ for F4 with a load of 100 Nm. All values ⇓ with ⇧ of load and ⇧ of H_2.	Non-linear relation with SOPI. All values ⇓ with ⇧ of load and ⇧ of H_2.	Non-linear relation with SOPI. All values ⟱ with ⇧ of load and ⇧ of H_2.	Non-linear relation with SOPI. ⟰ with ⇧ of ADI and then ⇩ when reach LTC for all Fs. All values ⟱ with ⇧ of load and ⇑ with ⇧ of H_2.	PN ▲ for F1, ⇔ with F4, ▼ for F2 and F3. With advancing SOPI, ⟱ for F1, and ↓ with other Fs. ⟱ for F2 to F4 in compare with F1 with all SOPIs.
[64]	4S, DI Diesel, 1C, WC, CR = 17.5:1, DS = 661 cm^3	1. Speed (rpm) = 1500 2. Load % = 25, 50, 75, and 100 3. IP (bar) = 200, 220, and 240	1. DFM 2. PD + (90%CNG + 10%H_2) 3. PD + (80%CNG + 20%H_2)	▲ at 200 bars, ⇔ at 240 bars, and ▼ at 220 bars for all Fs. ▼ for F3, ⇔ for F2, and ▲ for F1 at all IP. ⟱ with ⇧ H_2 for all IP_s.	NM	▼ at 220 bars, ⇔ at 240 bars, and ▲ at 200 bars for all fuels. ▼ for F3, ⇔ for F2, and▲ for F1 at all IP_s. ⟱ with ⇧ H_2 for all IP_s.	▲ at 220 bars and ▼ at 200 and 240 bars for all fuels. ▼ for F1, ⇔ for F2, and ▲ for F3 at all IP_s. ⟰ with ⇧ H_2 for all IP_s.	NM

Table 1. *Cont.*

Ref	Engine Details	Engine Operation Conditions	Fuel Number ($Fn_{(1,2,\ldots)}$) and Combinations	Engine Exhaust Emissions Results				
				CO	CO_2	HC	NO_x	Smoke, PM, PN, Soot
[66]	4S, DI Diesel, 1C, AC, CR = 17:1, DS = 573 cm^3	1. Speed (rpm) = 1500 2. Load: IMEP (Mpa) = 0.70	1. DFM 2. 10%PD + [90%NG] 3. 10%PD + [88%NG + 2%H_2] 4. 10%PD + [82%NG + 8%H_2] 5. 10%PD + [78%NG + 12%H_2] 6. 10%PD + [71%NG + 19%H_2] 7. 10%PD + [69%NG + 21%H_2]	↓ with ⇧ of H_2 for F2 to F7. ▲ for F1 and ▼ for F7. ⤋ with F2 to F7 compared with F1.	⇩ with ⇧ of H_2, except at F7 that slightly ▲ than F6. ▲ for F1 and ▼ for F6. ⤋ with F2 to F7 compared with F1.	▼ for F1 and F6. ⇔ for F4, F5, and F7. ▲ for F2 and F3.	⤊ with ⇧ of H_2 for F2 to F7. ▲ for F1 than F2. ▼ for F2 and ▲ for F7.	NM
[67]	4S, DI Diesel, 1C, CR = 19.3:1, DS = 406 cm^3	1. Speed (rpm) = 1500 2. Load % = 100 3. φ= 0.833, 0.88, 0.928, 0.941, 0.935, 0.893	1. DFM 2. 40%PD + [60%((100%CH_4), (70%CH_4 + 30%H_2), (50%CH_4 + 50%H_2), (70%CH_4 + 30%H_2), (100%H_2))] 3. 50% PD + [50%((100%CH_4), (70%CH_4 + 30%H_2), (50%CH_4 + 50%H_2), (70%CH_4 + 30%H_2), (100%H_2))] 4. 60% PD + [40%((100%CH_4), (70%CH_4 + 30%H_2), (50%CH_4 + 50%H_2), (70%CH_4 + 30%H_2), (100%H_2))]	▼ for F1 and PD + H_2, while ▲ for PD + CH_4 in all mixings. ⤋ with ⇧ of H_2 and ⇧ of PD in all mixings. ▲ for 40%DF + 60% CH_4.	NM	NM	▼ for F1 and PD + CH_4, while ▲ for PD + H_2 in all mixings. ⇑ with ⇧ of H_2 and ⇩ of PD in all mixings. ⤊ with ⇧ of H_2 for ▼ PD, ⇑ with ⇧ of H_2 for ⇔ PD, and ↑ with ⇧ of H_2 for ▲ PD. ▲ for 40%DF + 60%H_2.	NM
[68]	4S, DI Diesel, 1C, CR = 19.3:1, DS = 406 cm^3	1. Speed (rpm) = 1500 2. Load % = 100 3. φ = 0.833, 0.88, 0.928, 0.941, 0.935, 0.893 4. T_i (K) = 303, 318, 338	1. DFM 2. 60%PD + [40%CH_4] 3. 60% PD + [28%CH_4 + 12%H_2] 4. 60% PD + [20%CH_4 + 20%H_2] 5. 60% PD + [12%CH_4 + 28%H_2] 6. 60% PD + [40%H_2]	▼ for F1 and F6, while ▲ for F2 in all T_i values. ⤋ with ⇧ of H_2 in all T_i values. ⇑ for T_i of 303 to 318 K and ↑ with T_i of 318 to 338 K for all Fs.	NM	NM	▼ for F1 and F2, while ▲ for F6 in all T_i values. ⤊ with ⇧ of H_2 in all T_i values. ↑ for T_i of 303 to 318 K and ⇑ with T_i of 318 to 338 K for all Fs.	NM
[69]	4S, DI Diesel, 1C, CR = 13.5:1, DS = 11.9 L	1. Speed (rpm) = 1000 2. Load % = 100 3. OP (W) = 1200, 1600, 2000, 2400, 2800, 3200, 3600 4. T_i (K) = 800, 900, 1000	Gas–fuel replacement 90% 1. PD + (100%CH_4) 2. PD + (95%CH_4 + 5%H_2) 3. PD + (90%CH_4 + 10%H_2) 4. PD + (85%CH_4 + 15%H_2)	⤋ with ⇧ OP with all Fs. ▲ for F1 and ▼ for F4 at all OPs. ⇓ with ⇧ of H_2 at all OPs.	Not stable with OP, but almost ⇧ together. ▲ for F1 than other Fs. ▲ for F4 than F2 and F3 at OP of 1200 to 2400 W. ▼ for F4 than all Fs at OP of 2400 to 3600 W.	NM	⤊ with ⇧ of OP with all Fs. ▲ for F4 and ▼ for F1 at all OPs. ⇑ with ⇧ of H_2 at all OPs.	NM
[70]	4S, DI Diesel, 1C, R = 16.25:1, DS = 2.44 L	1. Load = partial (25%-405 bar BMEP) 2. Speed (rpm) = 910 3. SOPI (BTDC) = 10°, 14°, 18°, 22°, 26°, 30°, 34°, 38°, 42°, 46°, 50° 4. Various AFR according to mixing rate	Mode 1 1. 25%PD + [75%NG + 0%H_2] 2. 25%PD + [65%NG + 10%H_2] 3. 25%PD + [50%NG + 25%H_2] 4. 25%PD + [25%NG + 50%H_2] 5. 25%PD + [0%NG + 75%H_2] Mode 2 1. 25%PD + [75%NG + 0%H_2] 2. 25%PD + [75%NG + 5%H_2] 3. 25%PD + [75%NG + 10%H_2] 4. 25%PD + [75%NG + 15%H_2] 5. 25%PD + [75%NG + 20%H_2] 6. 25%PD + [75%NG + 25%H_2]	⤋ with ⇧ of H_2 and ⤊ with ⇧ of CH_4. ▼ for Mode 1.	⇓ with ⇧ of H_2 and ⇑ with ⇧ of CH_4. ▼ for Mode 1.	⤋ with ⇧ of H_2 and ⤊ with ⇧ of CH_4. ▼ for Mode 1.	↑ with ⇧ of H_2, but ⇓ when H_2 exceeded 50% in the gas mixture. ▲ for Mode 1 at ADI = 14° BTDC and F3. ▲ for Mode 2 at ADI = 10° BTDC and F4. Generally, ▲ for Mode 1 than Mode 2.	Soot ⤋ with ⇧ of H_2 for all operation conditions. ▼ for Mode 2.

Table 1. *Cont.*

Ref	Engine Details	Engine Operation Conditions	Fuel Number ($Fn_{(1,2,\ldots)}$) and Combinations	Engine Exhaust Emissions Results				
				CO	CO_2	HC	NO_x	Smoke, PM, PN, Soot
[71]	4S, DI Diesel, 1C, AC, CR = 18:1, DS = 630 cm^3	1. Speed (rpm) = 1500 2. Load % = 30, 40, 50, and 70	At 30% load: 1. DFM 2. PD + NG 3. PD + (84%NG + 16%H_2) 4. PD + (80%NG + 20%H_2) 5. PD + (70%NG + 30%H_2) At 40% load: 1. DFM 2. PD + NG 3. PD + (88.6%NG + 11.4%H_2) 4. PD + (80%NG + 20%H_2) 5. PD + (70%NG + 30%H_2) At 50 and 70% load: 1. DFM 2. PD + NG 3. PD + (90%NG + 10%H_2) 4. PD + (80%NG + 20%H_2) 5. PD + (70%NG + 30%H_2)	▼ for F1 with all loads. ▲ for F2 from 30 to 60% load. ▲ for F3 at 70% load. ⤊ with F2 to F5 compared with F1.	⤊ with ⇧ of load for all Fs. ▲ for F1 for all loads. ▼ for F5 at 30 and 70% loads, F3 at 40 and 50% loads. ⇓ with F2 to F5 compared with F1.	▼ for F1 of ▲ for F2 at all loads. ⇓ with ⇧ of H_2 for F3 to F5 with all loads. ▲ at 50% load, ⇔ at 70% load, and ▼ at 30% load for all Fs, except F4, the ▲ at 70% load.	⤊ with ⇧ of load for all Fs. ▲ for F1 with all loads. ▼ for F5 at 30% load, F2 and F3 at 40 to 67% loads, and for F4 at 70% load. ⇓ with F2 to F5 compared with F1.	NM
[72]	4S, DI Diesel, 1C, WC, CR = 16.05:1, DS = 553 cm^3	1. Speed (rpm) = 1500 2. Load: Rated power = 3.5 kW 3. CR = 15, 15.5, 16, 16.5, 17	1. PD + 100%CNG 2. PD + (95%CNG + 5%H_2) 3. PD + (90%CNG + 10%H_2) 4. PD + (85%CNG + 15%H_2) 5. PD + (80%CNG + 20%H_2) 6. PD + (75%CNG + 25%H_2)	⤋ with ⇧ of CR for all Fs. ▲ values for F1 and ▼ values for F6 with all CRs. ⇓ with ⇧ of H_2 for Fs of F2 to F4 with all CRs. ▼ for F5 than F1 but ▲ than Fs of F2, F3, F4 and F6 with all CRs.	NM	⇓ with ⇧ of CR for all Fs. ▲ values for F1 and ▼ values for F6 with all CRs. ⇑ with ⇧ of H_2 for Fs of F2 to F5 with all CRs.	⤊ with ⇧ of CR for all Fs. ▲ values for F6 and ▼ values for F1 with all CRs. ⇑ with ⇧ of H_2 for all CRs.	Smoke ⇓ with ⇧ of H_2 for all CRs. ▲ values for F1 and ▼ values for F6 with all CRs. Slightly affected by CR.
[73]	4S, DI Diesel, 6C, WC, CR = 17:1, DS = 11.5 L	1. Speed (rpm) = 1800 2. Load % = 25, 50, 75, 100 3. Rated power = 235.6 kW	1. DFM 2. 50%PD + [50%NG] 3. 50%PD + [45%NG + 5%H_2] 4. 50%PD + [40%NG + 10%H_2] 5. 50%PD + [35%NG + 15%H_2] 6. 80%PD + [20% H_2]	⤊ with ⇧ of load for all Fs. ▲ for F2 and ▼ for F6 at 25% load. ▲ for F1 and ▼ for F6 at 50% load. ▲ for F1 and ▼ for F5 at 75 and 100% loads. ⇓ with ⇧ of H_2 for Fs of F3 to F5 with all loads. Values with F2 are very close to values with F1 with all loads.	⤊ with ⇧ of load for all Fs. ▲ for F2 and ▼ for F6 at 25% load. ▲ for F1 and ▼ for F6 at 50% load. ▲ for F1 and ▼ for F6 at 75% load. ▲ for F1 and ▼ for F5 at 100% load. ⇓ with ⇧ of H_2 for Fs of F3 to F5 with all loads.	NM	⤊ with ⇧ of load for all Fs. ▲ for F5 and ▼ for F1 at 25 and 50% loads. ▲ for F6 and ▼ for F1 at 75% load. ▲ for F6 and ▼ for F2 at 100% load. ⇑ with ⇧ of H_2 for Fs of F2 to F4 with all loads. For gaseous fuels, F6 causes ▼ values at 25% load, ⇔ values at 50% load, and ▲ values at 75 and 100% loads.	Soot ⤊ with ⇧ of load for all Fs. ⇓ with ⇧ of H_2 in F3 to F5 with all loads. ▲ for F1 with loads 50 to 100%, and ▼ for F5 with all loads. ▲ for F6 at load 25%. At 100% load, ▼ for F2 than F1, and ▲ than other Fs.
[74]	4S, DI Diesel, 4C, WC, CR = 17.4:1, DS = 2494 cm^3	1. Speed (rpm) = 800, 1000, 1500, 2000, 2500, 3000, 3500, 4000 2. Load % = 100 3. Maximum power = 106 Kw at 3400 rpm	1. DFM 2. PD + HCNG	⤊ with ⇧ rpm with all Fs. ▲ for F1 and ▼ for F2 at all rpm. ⇓ with F2 in compare with F1.	NM	⤊ with ⇧ rpm with all Fs. ▲ for F1 and ▼ for F2 at all rpm. ⇓ with F2 in compare with F1.	⤊ with ⇧ rpm with all Fs. ▲ for F1 at rpm of 800. ▲ for F2 at rpm of 1000 to 2500, and ▲ for F1 at rpm of 2500 to 4000.	PM ⤊ with ⇧ rpm with all Fs. ▲ for F1 and ▼ for F2 with all rpm. ↓ with F2 in compare with F1.

Table 1. *Cont.*

Ref	Engine Details	Engine Operation Conditions	Fuel Number ($Fn_{(1,2,\ldots)}$) and Combinations	Engine Exhaust Emissions Results				
				CO	CO_2	HC	NO_x	Smoke, PM, PN, Soot
[88]	4S, DI Diesel, 1C, WC, CR = 16.5:1, DS = 522 cm^3	1. Speed (rpm) = 1500, 2000 2. Premixed ratio (RP %) = 86.8 3. φ = 0.184, 0.203, 0.212, 0.221 4. In-cylinder initial temperature (K) = 346.4 5. In-cylinder initial pressure (bar) = 1.591	1. PD + 100%CNG 2. PD + (75%CNG + 25%H_2) 3. PD + (50%CNG + 50%H_2) 4. PD + (25%CNG + 75%H_2) 5. PD + 100% H_2	NM	⤊ with ⇧ of CH_4 and ⤋ with ⇧ of H_2. ▲ for F1, and ▼ for F5.	⤊ with ⇧ of CH_4 and ⤋ with ⇧ of H_2. ▲ for F1, and ▼ for F5.	⤋ with ⇧ of CH_4 and ⤊ with ⇧ of H_2, except the opposite behavior with F2 and F3. ▲ for F5, and ▼ for F1.	NM
[75]	4S, DI Diesel, 6C, WC, CR = 13.4:1, DS = 8796 cm^3	1. Speed (rpm) = 1000 2. Premixed ratio (RP %) = 88 3. φ = 0.5 4. IP (bar) = 1700 5. Intake pressure (bar) = 1.85 6. Gross BMEP (bar) = 10	1. PD + 100%CNG 2. PD + (90%CNG + 10%H_2) 3. PD + (70%CNG + 30%H_2) 4. PD + (50%CNG + 50%H_2) 5. PD + (40%CNG + 60%H_2) 6. PD + 100% H_2	Not clear behavior with ⇧ of H_2 and ⇩ of CH_4.	⤋ with ⇧ of H_2 and ⤊ with ⇧ of CH_4. ▲ for F1, and ▼ for F5.	⤋ with ⇧ of H_2 and ⤊ with ⇧ of CH_4. ▲ for F1, and ▼ for F5.	⤊ with ⇧ of H_2 and ⤋ with ⇧ of CH_4. ▲ for F6, and ▼ for F1.	NM
[76]	4S, DI Diesel, 1C, WC, CR = 15.6:1, DS = 1402 cm^3	1. Speed (rpm) = 1200, 1500 2. IT (K) = 298 3. λ = 1.9 4. IP (bar) = 1000 5. SOPI (BTDC) = 7^0, 12^0 6. In-cylinder initial pressure (bar) = 1.08, 1.2	1. 5% PD + 95%CNG 2. 5% PD + [95% (90%CNG + 10%H_2)] 3. 5% PD + [95% (80%CNG + 20%H_2)] 4. 5% PD + [95% (70%CNG + 30%H_2) 5. 5% PD + [95% (60%CNG + 40%H_2) 6. 5% PD + [95% (50%CNG + 50%H_2) 7. 5% PD + [95% (40%CNG + 60%H_2) 6. 5% PD + 95% H_2	⤋ with ⇧ of H_2 and ⤊ with ⇧ of CH_4 for all tested cases. ▲ with ▲ rpm and ▲ SOPI for all Fs and In-cylinder initial pressures. ▼ with ▼ rpm, In-cylinder initial pressure, and SOPI for all Fs.	NM	NM	⇑ with ⇧ of H_2 and ⇓ with ⇧ of CH_4 for all tested cases. ▲ with ▲ In-cylinder initial pressure for all Fs, rpm and SOPI. ▼ with ▲ rpm and SOPI for all Fs and In-cylinder initial pressure.	NM

Based on the data provided in Table 1 and Figures 1–5, an overall summary of individual exhaust gas components and the factors influencing them in compression ignition (CI) engines using diesel, NG–diesel, and HNG–diesel as fuels is provided in Table 2. According to Table 2, it could be summarized that the exhaust emissions of CI engines vary significantly depending on the type of fuel used, which is diesel, natural gas–diesel (NG–diesel), or hydrogen–natural gas–diesel (HNG–diesel). Diesel tends to produce higher levels of carbon monoxide (CO) and particulate matter (PM), while NG–diesel results in higher hydrocarbon (HC) emissions. HNG–diesel generally shows a balanced performance but with a notable decrease in CO and PM emissions as the hydrogen content in the blend increases. Emission levels are also influenced by various engine parameters such as injection pressure, compression ratio, and engine load, with notable trends including decreasing CO_2 and HC emissions with advanced pilot injection timing and increasing nitrogen oxide (NO_x) emissions with higher initial temperatures and engine speeds. There is a noticeable gap in research on the effects of advanced pilot injection timing, intake temperature, initial temperature, and in-cylinder initial pressure on emissions for all fuel types.

Table 2. Generalized summary of exhaust emissions and the influences affecting them for compression ignition engines fueled by diesel, NG–diesel, and HNG–diesel. ▼: lowest, ▲: highest, ⇔: intermediate, ⇧: increasing, ↓: slightly decreasing, ↑: slightly increasing, ⇓: medium decreasing, ⇑: medium increasing, ⟱: highly decreasing, ⤊: highly increasing, rpm: revolution per minute, IP: injection pressure, NG: natural gas, HNG: hydrogen–natural gas, λ: Excess air factor, CR: compression ratio, Ti: intake temperature, SOPI: start of pilot injection, IT: initial temperature.

Exhaust Component	Summary of Findings
Carbon monoxide (CO)	▼ with diesel, ▲ with NG-diesel, ⇔ with HNG–diesel but ⟱ with ⇧ of H_2 in the HNG blend. ⇓ and then ↑ with advancing SOPI and ⇧ of λ, CR, IP, and in-cylinder initial pressure. ⇑ and then ↑ with ⇧ of Ti and IT. ⇑ to ⤊ with ⇧ of engine rpm and load.
Carbon dioxide (CO_2)	▲ with diesel, ⇔with NG-diesel, ▼ with HNG–diesel and ⟱ with ⇧ of H_2 in HNG blend. ⇓ with advancing SOPI and ⇧ of λ. ⤊ with ⇧ of engine load, and⟱ with ⇧ of engine rpm. There is a lack of investigation on the effects of CR, IP, Ti, IT, and in-cylinder initial pressure on CO_2 emissions.
Hydrocarbons (HC)	▲ with NG–diesel, ▼ with diesel and HNG–diesel, and ⇓ to ⟱ with ⇧ of H_2 in the HNG blend. ⇓ with advancing SOPI and ⇧ of IP, rpm, and CR. ⇓ and then ↑ or inverse behavior with ⇧ of λ according to another influences. There is a lack of investigation on the effects of Ti, IT, and in-cylinder initial pressure on the HC emissions.
Nitrogen oxide (NO_x)	▼ with NG-diesel, ▼ to ⇔ with diesel, ▲ with HNG–diesel and ⇑ with ⇧ of H_2 in the HNG blend. ⤊ and then ↓ with continue ⇧ of λ. ⤊ and then ⇓ with continue advancing SOPI. ⇑ to ⤊ with ⇧ of CR, IP, engine load and rpm. ↑ to ⇑ with continue ⇧ of Ti.
Particulate matter (PM)/Smoke/Soot	▼ with diesel, ▲ with NG-diesel, ⇔ with HNG–diesel and ⇓ to ⟱ with ⇧ of H_2 in an HNG blend. ↓ to ⟱ with continue advancing SOPI, slightly affected by CR. ⇑ to ⤊ with ⇧ of engine load, and ↑ to ⇑ with ⇧ of engine rpm. There is a lack of investigation on the effects of λ, Ti, IT, and in-cylinder initial pressure on the PM, smoke and soot emissions.

3. Conclusions

This review explores the impact of enriching natural gas with hydrogen on emissions in HCNG-diesel Tri-fuel engines. It shows that while using natural gas or hydrogen with diesel affects emissions, combining them in the Tri-fuel mode reduces CO, CO_2, HC, and PM emissions but increases NO_x emissions compared to dual-fuel modes. The study identifies operational parameters like engine power, speed, load, air–fuel ratio, and fuel blends influencing emissions. It notes that CO emissions decrease with higher hydrogen content, load, power, and advanced diesel injection timing but increase with engine speed. CO_2 emissions generally decrease with more hydrogen and lambda but increase with load. HC emissions mostly decrease with more hydrogen, lambda, advancing diesel injection timing, engine speed, and load but vary with intake temperature, PDIP, and CR. NO_x emissions typically rise with engine load, speed, hydrogen content, power, CR, and intake temperature but decrease with more natural gas. The conclusion emphasizes referring to Table 1 for engine specifications and emissions data. Table 2 provides a generalized summary of exhaust emissions and the influences affecting them for compression ignition engines fueled by diesel, NG-diesel, and HNG–diesel. Hydrogen incorporation significantly reduces soot and particulate emissions. Specific hydrogen ratios and optimized timings enhance reductions, showing hydrogen's potential as a supplementary fuel for cleaner diesel engines. Among reviewed studies, varied focuses were on engine loads, speeds, AFR, PDIP, PDIT, CR, and intake air temperatures, altering fuel blends. Many studies did not address all emissions comprehensively, indicating a need for more research on emissions reduction techniques in HNG–diesel Tri-fuel engines. Future research should adopt a multidisciplinary approach, integrating hydrogen enrichment, advanced injection timing, combustion optimization, and exhaust gas recirculation for reducing CO, CO_2, HC, NO_x, and PM emissions. Combining experimental studies with advanced simulations like CFD can provide deeper insights. Exploring novel techniques like water vapor injection and advanced diesel injection methods may offer additional emission reductions while maintaining engine performance. Collaboration among academia, industry, and regulatory bodies is crucial for developing sustainable solutions for internal combustion engine emissions.

Funding: This research was funded by National Laboratory for Renewable Energy, grant number RRF-2.3.1-21-2022-00009.

Acknowledgments: The authors would like to express thanks to the Budapest University of Technology and Economics for its scientific support across all disciplines and levels. The Ministry of Higher Education, University of Wasit-Iraq, and Stipendium Hungaricum are also acknowledged by the authors for providing financial support to the first author. Thanks also extended to Project No. RRF-2.3.1-21-2022-00009, titled National Laboratory for Renewable Energy, has been implemented with the support provided by the Recovery and Resilience Facility of the European Union within the framework of Program Széchenyi Plan Plus.

Conflicts of Interest: The authors declare no conflict of interest.

Abbreviations

AFR	Air fuel ratio	H_2	Hydrogen
ATDC	After top dead center	HC	Hydrocarbons
BP	Brake power	HCNG	Mixture of hydrogen and compressed natural gas
BSFC	Brake-specific fuel consumption	HNG	Mixture of hydrogen and natural gas
BT	Brake torque	HRR	Heat release rate
BTDC	Before top dead center	IMEP	Indicated mean effective pressure
BTE	Brake thermal efficiency	IP	Injection pressure
CA	Crank angle	IOP	Injection operating pressure
CH_4	Methane	LTC	Low-temperature combustion
CI	Compression ignition	Mix	Mixture
CNG	Compressed natural gas	NG	Natural gas
CO	Carbon monoxide	NO_x	Nitrogen oxides
CO_2	Carbon dioxide	PM	Particulate matter
CR	Compression ratio	SOPI	Start of pilot injection
DF	Dual-fuel	TDC	Top dead center
FC	Fuel consumption	Ti	Intake temperature
FD	Full diesel	UBHC	Unburned hydrocarbons
DFS	Diesel fuel system	λ	Excess air factor
PDF	Pilot diesel fuel		

References

1. Raub, J.A.; Mathieu-Nolf, M.; Hampson, N.B.; Thom, S.R. Carbon monoxide poisoning—A public health perspective. *Toxicology* **2000**, *145*, 1–14. [CrossRef] [PubMed]
2. Terzano, C.; Di Stefano, F.; Conti, V.; Graziani, E.; Petroianni, A. Air pollution ultrafine particles: Toxicity beyond the lung. *Eur. Rev. Med. Pharmacol. Sci.* **2010**, *14*, 809–821. [PubMed]
3. D'Amato, G.; Cecchi, L.; D'amato, M.; Liccardi, G. Urban air pollution and climate change as environmental risk factors of respiratory allergy: An update. *J. Investig. Allergol. Clin. Immunol.* **2010**, *20*, 95–102.
4. Mehra, R.K.; Duan, H.; Juknelevičius, R.; Ma, F.; Li, J. Progress in hydrogen enriched compressed natural gas (HCNG) internal combustion engines-A comprehensive review. *Renew. Sustain. Energy Rev.* **2017**, *80*, 1458–1498. [CrossRef]
5. Knothe, G.; Razon, L.F. Biodiesel fuels. *Prog. Energy Combust. Sci.* **2017**, *58*, 36–59. [CrossRef]
6. Singh, P.; Balasubramanian, D.; Venugopal, I.P.; Tyagi, V.V.; Goel, V.; Wae-Hayee, M.; Kalam, M.A.; Varuvel, E.G. A comprehensive review on the applicability of hydrogen and natural gas as gaseous fuel for dual fuel engine operation. *Energy Sources Part. A Recovery Util. Environ. Eff.* **2024**, *46*, 1559–1587. [CrossRef]
7. Goel, V.; Kumar, N.; Singh, P. Impact of modified parameters on diesel engine characteristics using biodiesel: A review. *Renew. Sustain. Energy Rev.* **2018**, *82*, 2716–2729. [CrossRef]
8. Bayraktar, H. An experimental study on the performance parameters of an experimental CI engine fueled with diesel–methanol–dodecanol blends. *Fuel* **2008**, *87*, 158–164. [CrossRef]
9. Wei, L.; Yao, C.; Wang, Q.; Pan, W.; Han, G. Combustion and emission characteristics of a turbocharged diesel engine using high premixed ratio of methanol and diesel fuel. *Fuel* **2015**, *140*, 156–163. [CrossRef]
10. Muhssen, H.S. Numerical Simulation of Mixing Characteristics of CNG-Air Mixer for Dual-Fuel Vehicle. 2016. Available online: http://psasir.upm.edu.my/id/eprint/67083/1/FK%202016%20146%20IR.pdf (accessed on 30 July 2024).
11. Zöldy, M. Improving heavy duty vehicles fuel consumption with density and friction modifier. *Int. J. Automot. Technol.* **2019**, *20*, 971–978. [CrossRef]
12. Torregrosa, A.; Broatch, A.; García, A.; Mónico, L. Sensitivity of combustion noise and NO_x and soot emissions to pilot injection in PCCI Diesel engines. *Appl. Energy* **2013**, *104*, 149–157. [CrossRef]
13. Tutak, W.; Lukács, K.; Szwaja, S.; Bereczky, Á. Alcohol–diesel fuel combustion in the compression ignition engine. *Fuel* **2015**, *154*, 196–206. [CrossRef]
14. Muhssen, H.S.; Bereczky, Á.; Zöldy, M. A Review on HCNG/Diesel Tri Fuel Engine Performance. In *Vehicle and Automotive Engineering 4*; Springer: Cham, Switzerland, 2023; pp. 268–288. [CrossRef]
15. Máté, Z.; Mária, S.C.; Péter, K.P.; Péter, B.; Ádám, T. Cognifive Sustainability. *Cogn. Sustain.* **2022**, *1*, 1–7.
16. Stayner, L.; Dankovic, D.; Smith, R.; Steenland, K. Predicted lung cancer risk among miners exposed to diesel exhaust particles. *Am. J. Ind. Med.* **1998**, *34*, 207–219. [CrossRef]
17. Cheung, K.; Ntziachristos, L.; Tzamkiozis, T.; Schauer, J.; Samaras, Z.; Moore, K.; Sioutas, C. Emissions of particulate trace elements, metals and organic species from gasoline, diesel, and biodiesel passenger vehicles and their relation to oxidative potential. *Aerosol Sci. Technol.* **2010**, *44*, 500–513. [CrossRef]
18. Muhssen, H.S.; Masuri, S.U.; Sahari, B.B.; Hairuddin, A.A. Design improvement of compressed natural gas (CNG)-Air mixer for diesel dual-fuel engines using computational fluid dynamics. *Energy* **2021**, *216*, 118957. [CrossRef]
19. Virt, M.; Arnold, U. Effects of Oxymethylene Ether in a Commercial Diesel Engine. *Cogn. Sustain.* **2022**, *1*, 3. [CrossRef]
20. Alahmer, A.; Rezk, H.; Aladayleh, W.; Mostafa, A.O.; Abu-Zaid, M.; Alahmer, H.; Gomaa, M.R.; Alhussan, A.A.; Ghoniem, R.M. Modeling and Optimization of a Compression Ignition Engine Fueled with Biodiesel Blends for Performance Improvement. *Mathematics* **2022**, *10*, 420. [CrossRef]
21. Goldbach, A.; Meier, H.F.; Wiggers, V.R.; Chiarello, L.M.; Barros, A.A.C. Combustion performance of bio-gasoline produced by waste fish oil pyrolysis. *Chem. Ind. Chem. Eng. Q.* **2022**, *28*, 1–8. [CrossRef]
22. Ramesohl, S.; Merten, F. Energy system aspects of hydrogen as an alternative fuel in transport. *Energy Policy* **2006**, *34*, 1251–1259. [CrossRef]
23. Hekkert, M.P.; Hendriks, F.H.; Faaij, A.P.; Neelis, M.L. Natural gas as an alternative to crude oil in automotive fuel chains well-to-wheel analysis and transition strategy development. *Energy Policy* **2005**, *33*, 579–594. [CrossRef]
24. Semelsberger, T.A.; Borup, R.L.; Greene, H.L. Dimethyl ether (DME) as an alternative fuel. *J. Power Sources* **2006**, *156*, 497–511. [CrossRef]
25. Dai, C.-C.; Tao, J.; Xie, F.; Dai, Y.-J.; Zhao, M. Biodiesel generation from oleaginous yeast Rhodotorula glutinis with xylose assimilating capacity. *Afr. J. Biotechnol.* **2007**, *6*, 2130–2134.
26. Kaparaju, P.; Serrano, M.; Thomsen, A.B.; Kongjan, P.; Angelidaki, I. Bioethanol, biohydrogen and biogas production from wheat straw in a biorefinery concept. *Bioresour. Technol.* **2009**, *100*, 2562–2568. [CrossRef] [PubMed]
27. Sukumaran, R.K.; Singhania, R.R.; Mathew, G.M.; Pandey, A. Cellulase production using biomass feed stock and its application in lignocellulose saccharification for bio-ethanol production. *Renew. Energy* **2009**, *34*, 421–424. [CrossRef]
28. Wei, L.; Geng, P. A review on natural gas/diesel dual fuel combustion, emissions and performance. *Fuel Process. Technol.* **2016**, *142*, 264–278. [CrossRef]
29. Arat, H.T.; Aydin, K.; Baltacioğlu, E.; Yaşar, E.; Baltacioğlu, M.K.; Conker, Ç.; Burgaç, A. A review of hydrogen-enriched compressed natural gas (HCNG)-fuel in diesel engines. *J. MacroTrends Energy Sustain.* **2013**, *1*, 115–122.

30. Muhssen, H.S.; Masuri, S.U.; Sahari, B.; Hairuddin, A.A. Computational Fluid Dynamics Investigation of Air-Gas Pre Mixing Controller Mixer Designed for CNG-Diesel Dual-Fuel Engines. *CFD Lett.* **2019**, *11*, 47–62.
31. Kummer, K.; Imre, A.R. Seasonal and Multi-Seasonal Energy Storage by Power-to-Methane Technology. *Energies* **2021**, *14*, 3265. [CrossRef]
32. Imre, A.R. Seasonal Energy Storage with Power-to-Methane Technology. *Energies* **2022**, *15*, 712. [CrossRef]
33. Pirouzpanah, V.; Sarai, R.K. Reduction of emissions in an automotive direct injection diesel engine dual-fuelled with natural gas by using variable exhaust gas recirculation. *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.* **2003**, *217*, 719–725. [CrossRef]
34. Srinivasan, K.; Krishnan, S.; Qi, Y. Cyclic combustion variations in dual fuel partially premixed pilot-ignited natural gas engines. *J. Energy Resour. Technol.* **2014**, *136*, 012003. [CrossRef]
35. Dimitriou, P.; Tsujimura, T. A review of hydrogen as a compression ignition engine fuel. *Int. J. Hydrogen Energy* **2017**, *42*, 24470–24486. [CrossRef]
36. Mohammed, S.E.; Baharom, M.; Aziz, A.R.A. The effects of fuel-injection timing at medium injection pressure on the engine characteristics and emissions of a CNG-DI engine fueled by a small amount of hydrogen in CNG. *Int. J. Hydrogen Energy* **2011**, *36*, 11997–12006. [CrossRef]
37. Zhao, J.; Ma, F.; Xiong, X.; Deng, J.; Wang, L.; Naeve, N.; Zhao, S. Effects of compression ratio on the combustion and emission of a hydrogen enriched natural gas engine under different excess air ratio. *Energy* **2013**, *59*, 658–665. [CrossRef]
38. Ajmeri, K.H.; Sharma, S.; Shishodia, D.; Sharma, S.; Singh, S.; Molvi, I. Review on possibility to run the diesel engine on hcng fuel. *IJRAR-Int. J. Res. Anal. Rev. (IJRAR)* **2020**, *7*, 79–87.
39. Nelsson, C.; Hulteberg, C.; Saint-Just, J.; Kaiadi, M. HCNG–A Dead End or a Bridge to the Future. In Proceedings of the 18th World Hydrogen Energy Conference, Essen, Germany, 16–20 May 2010.
40. Nanthagopal, K.; Subbarao, R.; Elango, T.; Baskar, P.; Annamalai, K. Hydrogen enriched compressed natural gas (HCNG): A futuristic fuel for internal combustion engines. *Therm. Sci.* **2011**, *15*, 1145–1154. [CrossRef]
41. Szamrej, G.; Karczewski, M. Exploring Hydrogen-Enriched Fuels and the Promise of HCNG in Industrial Dual-Fuel Engines. *Energies* **2024**, *17*, 1525. [CrossRef]
42. Hairuddin, A.A.; Yusaf, T.; Wandel, A.P. A review of hydrogen and natural gas addition in diesel HCCI engines. *Renew. Sustain. Energy Rev.* **2014**, *32*, 739–761. [CrossRef]
43. Hairuddin, A.A.; Wandel, A.P.; Yusaf, T. Hydrogen and natural gas comparison in diesel HCCI engines-a review. In Proceedings of the 2010 southern region engineering conference (SREC 2010), Toowoomba, Australia, 10–12 November 2010; University of Southern Queensland: Toowoomba, Australia, 2010.
44. Liu, J.; Yang, F.; Wang, H.; Ouyang, M.; Hao, S. Effects of pilot fuel quantity on the emissions characteristics of a CNG/diesel dual fuel engine with optimized pilot injection timing. *Appl. Energy* **2013**, *110*, 201–206. [CrossRef]
45. Sayin, C.; Ilhan, M.; Canakci, M.; Gumus, M. Effect of injection timing on the exhaust emissions of a diesel engine using diesel–methanol blends. *Renew. Energy* **2009**, *34*, 1261–1269. [CrossRef]
46. Papagiannakis, R.G.; Hountalas, D.T.; Krishnan, S.R.; Srinivasan, K.K.; Rakopoulos, D.C.; Rakopoulos, C.D. Numerical evaluation of the effects of compression ratio and diesel fuel injection timing on the performance and emissions of a fumigated natural gas–diesel dual-fuel engine. *J. Energy Eng.* **2016**, *142*, E4015015. [CrossRef]
47. Kitamura, T.; Ito, T.; Senda, J.; Fujimoto, H. Mechanism of smokeless diesel combustion with oxygenated fuels based on the dependence of the equivalence ration and temperature on soot particle formation. *Int. J. Engine Res.* **2002**, *3*, 223–248. [CrossRef]
48. Gulcan, H.E.; Ciniviz, M. Experimental study on the effect of piston bowl geometry on the combustion performance and pollutant emissions of methane-diesel common rail dual-fuel engine. *Fuel* **2023**, *345*, 128175. [CrossRef]
49. Yousefi, A.; Birouk, M.; Guo, H. An experimental and numerical study of the effect of diesel injection timing on natural gas/diesel dual-fuel combustion at low load. *Fuel* **2017**, *203*, 642–657. [CrossRef]
50. Yang, B.; Wang, L.; Ning, L.; Zeng, K. Effects of pilot injection timing on the combustion noise and particle emissions of a diesel/natural gas dual-fuel engine at low load. *Appl. Therm. Eng.* **2016**, *102*, 822–828. [CrossRef]
51. Jamrozik, A.; Tutak, W.; Grab-Rogaliński, K. An experimental study on the performance and emission of the diesel/CNG dual-fuel combustion mode in a stationary CI engine. *Energies* **2019**, *12*, 3857. [CrossRef]
52. Guo, H.; Neill, W.S.; Liko, B. An experimental investigation on the combustion and emissions performance of a natural gas–diesel dual fuel engine at low and medium loads. In Proceedings of the Internal Combustion Engine Division Fall Technical Conference, Houston, TX, USA, 8–11 November 2015; American Society of Mechanical Engineers: New York, NY, USA, 2015; Volume 57274, p. V001T03A005.
53. Papagiannakis, R.G.; Hountalas, D.T. Experimental investigation concerning the effect of natural gas percentage on performance and emissions of a DI dual fuel diesel engine. *Appl. Therm. Eng.* **2003**, *23*, 353–365. [CrossRef]
54. Meng, L.; Zhang, C.; Bao, X.; Song, J.; Liu, Z. Effects of the substitution rate of natural gas on the combustion and emission characteristics in a dual-fuel engine under full load. *Adv. Mech. Eng.* **2017**, *9*, 1–8.
55. Arslan, E.; Kahraman, N. The effects of hydrogen enriched natural gas under different engine loads in a diesel engine. *Int. J. Hydrogen Energy* **2021**, *47*, 12410–12420. [CrossRef]
56. Zhou, J.; Cheung, C.; Leung, C. Combustion, performance and emissions of a diesel engine with H_2, CH_4 and H_2–CH_4 addition. *Int. J. Hydrogen Energy* **2014**, *39*, 4611–4621. [CrossRef]

57. Akansu, S.O.; Dulger, Z.; Kahraman, N.; Veziroğlu, T.N. Internal combustion engines fueled by natural gas—Hydrogen mixtures. *Int. J. Hydrogen Energy* **2004**, *29*, 1527–1539. [CrossRef]
58. Schefer, R.W.; Wicksall, D.; Agrawal, A.K. Combustion of hydrogen-enriched methane in a lean premixed swirl-stabilized burner. *Proc. Combust. Inst.* **2002**, *29*, 843–851. [CrossRef]
59. Alrazen, H.A.; Talib, A.A.; Ahmad, K. A two-component CFD studies of the effects of H2, CNG, and diesel blend on combustion characteristics and emissions of a diesel engine. *Int. J. Hydrogen Energy* **2016**, *41*, 10483–10495. [CrossRef]
60. Arat, H.T.; Baltacioglu, M.K.; Aydin, K.; Özcanli, M. Experimental investigation of using 30HCNG fuel mixture on a non-modified diesel engine operated with various diesel replacement rates. *Int. J. Hydrogen Energy* **2016**, *41*, 3199–3207. [CrossRef]
61. Tangöz, S.; Akansu, S.O.; Kahraman, N.; Malkoç, Y. Effects of compression ratio on performance and emissions of a modified diesel engine fueled by HCNG. *Int. J. Hydrogen Energy* **2015**, *40*, 15374–15380. [CrossRef]
62. European Commission. *No 595/2009 of the European Parliament and the Council of 18 June 2009 on Type-Approval of Motor Vehicles and Engines with Respect to Emissions from Heavy Duty Vehicles (Euro VI)*; European Commission: Brussels, Belgium, 2009.
63. Benbellil, M.A.; Lounici, M.S.; Loubar, K.; Tazerout, M. Investigation of natural gas enrichment with high hydrogen participation in dual fuel diesel engine. *Energy* **2022**, *243*, 122746. [CrossRef]
64. Quadri, S.A.P.; Rentala, G.S.; Sarap, R. Performance, combustion and exhaust emissions analysis of HCNG fuelled single cylinder diesel engine at different injection opening pressures. *World J. Eng.* **2018**, *15*, 710–718. [CrossRef]
65. De Simio, L.; Iannaccone, S. Gaseous and particle emissions in low-temperature combustion diesel–HCNG dual-fuel operation with double pilot injection. *Appl. Energy* **2019**, *253*, 113602. [CrossRef]
66. Tutak, W.; Jamrozik, A.; Grab-Rogaliński, K. Effect of natural gas enrichment with hydrogen on combustion process and emission characteristic of a dual fuel diesel engine. *Int. J. Hydrogen Energy* **2020**, *45*, 9088–9097. [CrossRef]
67. Mansor, M.R.A.; Abbood, M.M.; Mohamad, T.I. The influence of varying hydrogen-methane-diesel mixture ratio on the combustion characteristics and emissions of a direct injection diesel engine. *Fuel* **2017**, *190*, 281–291. [CrossRef]
68. Mansor, M.R.A.; Mohamad, T.I.; Sabah, O. Numerical investigation on combustion and emissions in a direct injection compression ignition engine fuelled with various hydrogen–methane–diesel blends at different intake air temperatures. *Energy Rep.* **2021**, *7*, 403–421. [CrossRef]
69. Zhao, R.; Xu, L.; Su, X.; Feng, S.; Li, C.; Tan, Q.; Wang, Z. A numerical and experimental study of marine hydrogen–natural gas–diesel tri–fuel engines. *Pol. Marit. Res.* **2020**. [CrossRef]
70. Ekin, F.; Ozsoysal, O.; Arslan, H. The effect of using hydrogen at partial load in a diesel-natural gas dual fuel engine. *Int. J. Hydrogen Energy* **2022**, *47*, 18532–18550. [CrossRef]
71. Lounici, M.S.; Boussadi, A.; Loubar, K.; Tazerout, M. Experimental investigation on NG dual fuel engine improvement by hydrogen enrichment. *Int. J. Hydrogen Energy* **2014**, *39*, 21297–21306. [CrossRef]
72. Reddy, S.; Dutta, M.; Reddy, K.V.K. Effect of compression ratio on performance of a hydrogen blended cng-diesel dual fuel engine. *J. Mech. Eng.* **2014**, *44*, 87–93. [CrossRef]
73. Luo, J.; Liu, Z.; Wang, J.; Xu, H.; Tie, Y.; Yang, D.; Zhang, Z.; Zhang, C.; Wang, H. Investigation of hydrogen addition on the combustion, performance, and emission characteristics of a heavy-duty engine fueled with diesel/natural gas. *Energy* **2022**, *260*, 125082. [CrossRef]
74. Pichayapat, K.; Sukchai, S.; Thongsan, S.; Pongtornkulpanich, A. Emission characteristics of using HCNG in the internal combustion engine with minimum pilot diesel injection for greater fuel economy. *Int. J. Hydrogen Energy* **2014**, *39*, 12182–12186. [CrossRef]
75. Cameretti, M.C.; De Robbio, R.; Palomba, M. Numerical Analysis of Dual Fuel Combustion in a Medium Speed Marine Engine Supplied with Methane/Hydrogen Blends. *Energies* **2023**, *16*, 6651. [CrossRef]
76. Yin, C.; Wang, C.; Shen, X.; Zhang, Z. Experimental and simulation study of diesel/methane/hydrogen triple-fuel combustion progression in a heavy-duty optical engine. *Int. J. Hydrogen Energy* **2024**, *62*, 562–578. [CrossRef]
77. Heywood, J.B. *Internal Combustion Engine Fundamentals*; Mcgraw-Hill: New York, NY, USA, 1988.
78. Zheng, J.; Wang, J.; Zhao, Z.; Wang, D.; Huang, Z. Effect of equivalence ratio on combustion and emissions of a dual-fuel natural gas engine ignited with diesel. *Appl. Therm. Eng.* **2019**, *146*, 738–751. [CrossRef]
79. Chaichan, M.T. Combustion of Dual Fuel Type Natural Gas/Liquid Diesel Fuel in Compression Ignition Engine. *IOSR J. Mech. Civ. Eng.* **2014**, *11*, 48–58. [CrossRef]
80. Sayin, C.; Canakci, M. Effects of injection timing on the engine performance and exhaust emissions of a dual-fuel diesel engine. *Energy Convers. Manag.* **2009**, *50*, 203–213. [CrossRef]
81. Agarwal, A.K.; Srivastava, D.K.; Dhar, A.; Maurya, R.K.; Shukla, P.C.; Singh, A.P. Effect of fuel injection timing and pressure on combustion, emissions and performance characteristics of a single cylinder diesel engine. *Fuel* **2013**, *111*, 374–383. [CrossRef]
82. Yousefi, A.; Guo, H.; Birouk, M. Effect of diesel injection timing on the combustion of natural gas/diesel dual-fuel engine at low-high load and low-high speed conditions. *Fuel* **2019**, *235*, 838–846. [CrossRef]
83. Yousefi, A.; Guo, H.; Birouk, M.; Liko, B. On greenhouse gas emissions and thermal efficiency of natural gas/diesel dual-fuel engine at low load conditions: Coupled effect of injector rail pressure and split injection. *Appl. Energy* **2019**, *242*, 216–231. [CrossRef]
84. Singh, R.; Maji, S. Performance and exhaust gas emissions analysis of direct injection cng-diesel dual fuel engine. *Int. J. Eng. Sci. Technol.* **2012**, *4*, 833–846.

85. Lal, S.; Mohapatra, S. The effect of compression ratio on the performance and emission characteristics of a dual fuel diesel engine using biomass derived producer gas. *Appl. Therm. Eng.* **2017**, *119*, 63–72. [CrossRef]
86. Lim, O.; Iida, N.; Cho, G.; Narankhuu, J. *The Research about Engine Optimization and Emission Characteristic of Dual Fuel Engine Fueled with Natural Gas and Diesel*; 0148-7191; SAE Technical Paper: Chicago, IL, USA, 2012.
87. Daukšys, V.; Račkus, M.; Zamiatina, N. Energy efficiency improvement adding various amounts of CNG in the naturally aspirated compression ignition engine. *Procedia Eng.* **2017**, *187*, 222–228. [CrossRef]
88. Cameretti, M.C.; De Robbio, R.; Mancaruso, E.; Palomba, M. CFD study of dual fuel combustion in a research diesel engine fueled by hydrogen. *Energies* **2022**, *15*, 5521. [CrossRef]
89. Demers, D.; Walters, G. Guide to exhaust emission control options. *BAeSAME Bristol.*. 1999. Available online: https://scholar.google.com/scholar?hl=hu&as_sdt=0,5&q=89.%09Demers,+D.;+Walters,+G.+Guide+to+exhaust+emission+control+options.+BAeSAME+Bristol+1999+&btnG= (accessed on 30 July 2024).
90. Reşitoğlu, İ.A.; Altinişik, K.; Keskin, A. The pollutant emissions from diesel-engine vehicles and exhaust aftertreatment systems. *Clean. Technol. Environ. Policy* **2015**, *17*, 15–27. [CrossRef]
91. Correa, S.M.; Arbilla, G. Carbonyl emissions in diesel and biodiesel exhaust. *Atmos. Environ.* **2008**, *42*, 769–775. [CrossRef]
92. Yamada, H.; Misawa, K.; Suzuki, D.; Tanaka, K.; Matsumoto, J.; Fujii, M.; Tanaka, K. Detailed analysis of diesel vehicle exhaust emissions: Nitrogen oxides, hydrocarbons and particulate size distributions. *Proc. Combust. Inst.* **2011**, *33*, 2895–2902. [CrossRef]
93. Zheng, M.; Mulenga, M.C.; Reader, G.T.; Wang, M.; Ting, D.S.; Tjong, J. Biodiesel engine performance and emissions in low temperature combustion. *Fuel* **2008**, *87*, 714–722. [CrossRef]
94. Payri, F.; Bermúdez, V.R.; Tormos, B.; Linares, W.G. Hydrocarbon emissions speciation in diesel and biodiesel exhausts. *Atmos. Environ.* **2009**, *43*, 1273–1279. [CrossRef]
95. Faiz, A.; Weaver, C.S.; Walsh, M.P. *Air Pollution from Motor Vehicles: Standards and Technologies for Controlling Emissions*; World Bank Publications: Washington, DC, USA, 1996.
96. Dhariwal, H. Control of blowby emissions and lubricating oil consumption in IC engines. *Energy Convers. Manag.* **1997**, *38*, 1267–1274. [CrossRef]
97. Zareei, J.; Haseeb, M.; Ghadamkheir, K.; Farkhondeh, S.; Yazdani, A.; Ershov, K. The effect of hydrogen addition to compressed natural gas on performance and emissions of a DI diesel engine by a numerical study. *Int. J. Hydrogen Energy* **2020**, *45*, 34241–34253. [CrossRef]
98. Glarborg, P.; Miller, J.A.; Ruscic, B.; Klippenstein, S.J. Modeling nitrogen chemistry in combustion. *Prog. Energy Combust. Sci.* **2018**, *67*, 31–68. [CrossRef]
99. Law, C.K.; Kwon, O. Effects of hydrocarbon substitution on atmospheric hydrogen–air flame propagation. *Int. J. Hydrogen Energy* **2004**, *29*, 867–879. [CrossRef]
100. Hidy, G. Aerosols. In *Encyclopedia of Physical Science and Technology*, 3rd ed.; Academic Press: Cambridge, MA, USA, 2003. [CrossRef]
101. Gupta, H.N. *Fundamentals of Internal Combustion Engines*; PHI Learning Pvt. Ltd.: Delhi, India, 2012.
102. Wallington, T.; Kaiser, E.; Farrell, J. Automotive fuels and internal combustion engines: A chemical perspective. *Chem. Soc. Rev.* **2006**, *35*, 335–347. [CrossRef]
103. Fiebig, M.; Wiartalla, A.; Holderbaum, B.; Kiesow, S. Particulate emissions from diesel engines: Correlation between engine technology and emissions. *J. Occup. Med. Toxicol.* **2014**, *9*, 6. [CrossRef] [PubMed]
104. Chen, Y.; Ma, J.; Han, B.; Zhang, P.; Hua, H.; Chen, H.; Su, X. Emissions of automobiles fueled with alternative fuels based on engine technology: A review. *J. Traffic Transp. Eng. (Engl. Ed.)* **2018**, *5*, 318–334. [CrossRef]