WOJSKOWA AKADEMIA TECHNICZNA
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ROZPRAWA DOKTORSKA

„Badania układu zasilania dwupaliwowego do modernizacji silników spalinowych pojazdów ciężarowych”

mgr inż. Janusz Chojnowski

Rozprawa doktorska stanowiąca zbiór powiązanych tematycznie artykułów naukowych w dyscyplinie inżynieria mechaniczna

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Z wyrazami szacunku,

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Ja, niżej podpisany, oświadczam, że rozprawa doktorska pt.: „*Badania układu zasilania dwupaliwowego do modernizacji silników spalinowych pojazdów ciężarowych*”

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Mam świadomość, że złożenie nieprawdziwego oświadczenia skutkować będzie niedopuszczeniem do dalszych czynności postępowania w sprawie nadania stopnia doktora lub cofnięciem decyzji o nadaniu mi stopnia doktora oraz wszczęciem postępowania dyscyplinarnego/karnego.

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Wykaz najistotniejszych skrótów i oznaczeń:

Symbol	Oznaczenie	Jednostka
BEV	(ang. *Battery Electric Vehicle)* – pojazd elektryczny akumulatorowy	
CDF	(ang. *Conventional Dual Fuel*) – konwencjonalne spalanie dwupaliwowe	
CH_3OH	metanol	
CH_4	metan	
CO	tlenek węgla	
CO_2	dwutlenek węgla	
CO_2-*eq*	emisja gazów cieplarnianych przedstawiona w formie ekwiwalentu dwutlenku węgla	
CR	(ang. *Common Rail*) system elektronicznego wtrysku paliwa w silnikach o ZS	
CNG	(ang. *Compressed Natural Gas*) – sprężony gaz ziemny	
DMC	dopuszczalna masa całkowita	[kg]
DPF	(ang. *Diesel Particulate Filter*) – filtr cząstek stałych	
EDC	(ang. *Electronic Diesel Control*) – elektroniczny sterownik silnika o ZS	
EEV	(ang. *Enhanced Environmentally friendly Vehicle*) – nieoficjalny, ale szeroko uznawany standard, stosowany jako „miękki” cel dla producentów pojazdów przed wprowadzeniem normy EURO VI	
EHVA	(ang. *Electro-Hydraulic Variable Valve Actuator*) – system elektrohydraulicznej niezależnej kontroli położenia zaworów silnikowych	
FAME	(ang. *Fatty Acid Methyl Esters)* – estry metylowe kwasów tłuszczowych, Biodiesel B100	
FCEV	(ang. *Fuel Cell Electric Vehicles*) – pojazd elektryczny z ogniwem paliwowym (najczęściej wodorowym)	
G_e	godzinowe zużycie paliwa	[kg/h]
GHG	(ang. *GreenHouse Gases*) – gazy cieplarniane	
GMP	górne martwe położenie tłoka	

H_2	wodór	
HC	węglowodory	
HCNG	(ang. *Hydrogen to Compressed Natural Gas*) – paliwo będące mieszaniną gazu ziemnego i wodoru	
HRR	(ang. *Heat Release Rate*) – współczynnik uwalniania ciepła	[kW]
HVO	(ang. *Hydrotreated Vegetable Oil*) – hydrorafinowany olej roślinny	
	koń mechaniczny	[KM]
	kilowat	[kW]
	kilowatogodzina	[kWh]
LC	liczba cetanowa	
LNG	(ang. *Liquefied Natural Gas*) – skroplony gaz ziemny	
LO	liczba oktanowa	
LPG	(ang. *Liquefied Petroleum Gas*) – skroplony gaz propan-butan	
LTC	(ang. *Low-Temperature Combustion*) – spalanie niskotemperaturowe	
MAP	(ang. *Manifold Absolute Pressure*) – ciśnienie bezwzględne w kolektorze dolotowym silnika	[bar]
MBCS	(ang. *Model Based Control System)* – system sterowania z wykorzystaniem modeli referencyjnych	
M_o	moment obrotowy	[Nm]
n	prędkość obrotowa	[obr./min]
Ne	moc silnika	[kW, KM]
NCBiR	Narodowe Centrum Badań i Rozwoju	
NH_3	amoniak	
	niutonometr	[Nm]
	normalne metry sześcienne	[Nm^3]
NMHC	(ang. *Non-Methane Hydrocarbons*) – węglowodory niemetanowe	
NO_2	dwutlenek azotu	
NO_x	tlenki azotu	
O_2	tlen	
OBD	(ang. *On-Board Diagnostics*) – system autodiagnostyki pojazdu	
OŹE	odnawialne źródła energii	
PD	(niem. *Pumpe-Düse-System*) – pompowtryskiwacz	

PM	(ang. *Particulate Matter*) – cząstki stałe	
PPM	(ang. *Parts Per Million*) – liczba cząstek na milion	
PRR	(ang. *Pressure-Rise Rate*) – szybkość wzrostu ciśnienia w cylindrze	
RCCI	(ang. *Reactivity Controlled Compression Ignition*) – niskotemperaturowe spalanie dwupaliwowe kontrolowane reaktywnością paliw	
SAF	(ang. *Sustainable Aviation Fuel*) – zrównoważone syntetyczne paliwo lotnicze	
SCR	(ang. *Selective Catalytic Reduction*) – selektywna redukcja katalityczna (szkodliwych składników spalin)	
	tonokilometr	[TKM]
$\mathbf{U_b}$	niepewność standardowa	
UE	Unia Europejska	
UHC	(ang. *Unburned HydroCarbons*) – niespalone węglowodory	
USA	(ang. *United States of America*) – Stany Zjednoczone Ameryki Północnej	
VVA	(ang. *Variable Valve Actuation*) – system niezależnego, zmiennego sterowania zaworami silnika	
TTW	(ang. *Tank-to-Wheel*) – intensywność emisji gazów cieplarnianych wynikająca z procesów transformacji energii w pojeździe	
WTW	(ang. *Well-to-Wheel)* – intensywność emisji gazów cieplarnianych w całym cyklu życia pojazdu, z uwzględnieniem wszystkich etapów, od wydobycia surowców po zużycie paliwa lub energii w samym pojeździe	
$\boldsymbol{Wz}$	współczynnik zastąpienia paliw w dwupaliwowym systemie zasilania	
ZI	zapłon iskrowy	
ZS	zapłon samoczynny	
$\boldsymbol{\Delta_{gr}}$	błąd graniczny	
$\boldsymbol{\lambda}$	„lambda”, współczynnik nadmiaru powietrza	
$\boldsymbol{\eta}$	sprawność ogólna silnika	

I. STRESZCZENIE ROZPRAWY DOKTORSKIEJ

1. Wstęp i motywacja

Analizy przeprowadzone w 2022 roku wykazują, że silniki spalinowe generują około 25% globalnej pracy mechanicznej, wykorzystanej do napędzania różnorodnych urządzeń roboczych i środków transportu [1]. W kontekście transportu zasadniczą rolę odgrywają silniki tłokowe o zapłonie samoczynnym (ZS). Dane wskazują, że na świecie funkcjonuje około 450 milionów pojazdów wykorzystujących ten rodzaj napędu [2]. Ich popularność na rynku jest wynikiem kombinacji czynników takich jak niezawodność, konstrukcyjna prostota oraz wysoka sprawność ogólna (η), która przekłada się niskie zużycie jednostkowe paliwa.

Pojazdy z napędem spalinowym w zdecydowanej większości korzystają z paliw pochodzących z przetwarzania ropy naftowej. Światowe wydobycie tego surowca osiąga dziennie poziom około 90 milionów baryłek [3], z czego około 65% przeznacza się na produkcję paliw wykorzystywanych w transporcie [4]. Prognozy dotyczące przyszłego zużycia energii wskazują, że w nadchodzących dekadach paliwa kopalne będą nadal odgrywać istotną rolę na skalę globalną. Przewiduje się, że ogólnoświatowa konsumpcja ropy naftowej będzie wzrastać w tempie około 1% rocznie [1]. Masowe wykorzystanie paliw kopalnych skutkuje wzrostem emisji dwutlenku węgla (CO_2) do atmosfery, ponieważ niemal każdy atom węgla zawarty w tych paliwach ostatecznie przekształca się w cząsteczkę tego gazu. Szacuje się, że tłokowe silniki spalinowe odpowiadają za około 10% globalnej emisji CO_2 ze źródeł przemysłowych [5]. Problemem jest również emisja dotycząca substancji toksycznych, wynikająca z procesu produkcji i logistyki paliw kopalnych. Rozwiązaniem tych problemów mają być konsekwentnie rozwijane od kilku dekad normy emisji poszczególnych składników spalin, z wyraźnym naciskiem na redukcję CO_2, tlenków azotu (NO_x) i cząstek stałych (ang. *Particulate Matter,* PM) [6].

W obliczu globalnych wyzwań związanych z ochroną klimatu nasilają się działania mające na celu ograniczenie emisji gazów cieplarnianych (ang. *GreenHouse Gases*, GHG) oraz redukcję przedostawania się zanieczyszczeń toksycznych do atmosfery. W tym kontekście, poza implementacją odpowiednich regulacji prawnych, kluczowego znaczenia nabiera opracowanie i zastosowanie zrównoważonych strategii dotyczących wykorzystania odnawialnych źródeł energii (OŹE) oraz implementacji technologii przetwarzania energii charakteryzujących się niskim poziomem lub zerową emisją szkodliwych substancji [4]. Takie podejście wymaga przyjęcia perspektywy holistycznej, która umożliwia kompleksową analizę

emisji zanieczyszczeń na wszystkich etapach cyklu życia pojazdów – począwszy od pozyskiwania surowców, przez produkcję nośników energii i urządzeń do jej przetwarzania, aż po etap użytkowania i ostateczną utylizację zużytych pojazdów i wszystkich powiązanych z nimi zużytych komponentów [7]. W takim kontekście szczególne znaczenie zyskuje koncepcja retrofittingu, rozumiana jako proces modernizacji istniejących systemów, urządzeń lub konstrukcji, mający na celu poprawę efektywności energetycznej, dodanie nowych funkcjonalności i wydłużenie okresu użytkowania [8,9]. W odniesieniu do pojazdów wyposażonych w silniki o ZS retrofitting w zasilanie dwupaliwowe może przyczynić się do wydłużenia resursu pojazdu przy jednoczesnej redukcji emisji GHG oraz innych szkodliwych substancji zawartych w spalinach [10]. Takie działania mogą w synergiczny sposób przyczynić się do walki z globalnym ociepleniem i być jedną z dróg do zrównoważonego rozwoju gospodarki w ramach globalnych wysiłków na rzecz zwalczania zmian klimatycznych oraz stanowić ważny element strategii zrównoważonego rozwoju gospodarczego.

W związku z powyższym w niniejszej pracy podjęto tematykę potencjału modernizacji tłokowych silników o ZS do zasilania dwupaliwowego. Poruszono szerokie spektrum zagadnienia, zaczynając od aspektów ekonomicznych, bezpieczeństwa i gospodarczych, z uwzględnieniem analiz atrakcyjność rynkowej. Wytypowano również najbardziej perspektywiczne paliwa do rozwiązań dwupaliwowych. Istotną częścią pracy jest opis efektów prac nad oryginalnym rozwiązaniem nowoczesnej instalacji dwupaliwowej dla serii silników stosowanych w pojazdach ciężarowych, wypełniający tym samym aspekt analizy i zastosowania wyników własnych badań naukowych autora. Całość podjętych i zrealizowanych działań wypełnia definicję oryginalnie rozwiązanego problemu naukowego.

Motywacją do podjęcia pracy była chęć wykorzystania warunków dokonującej się transformacji źródeł napędu pojazdów do znalezienia i wykorzystania niszy, w której beneficjentami mogą być wszyscy ludzie związani na co dzień z szeroko pojętym sektorem motoryzacyjnym oraz branżą polskiego transportu drogowego.

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z mobilnym stanowiskiem diagnostyczno-montażowym”. Realizatorem projektu NCBiR było konsorcjum składające się z Wydziału Inżynierii Mechanicznej Wojskowej Akademii Technicznej oraz firmy Gaslux sp. z o.o., która była liderem projektu.

2. Analiza stanu zagadnienia

2.1. Imperatywy ekologiczne i ich ograniczenia

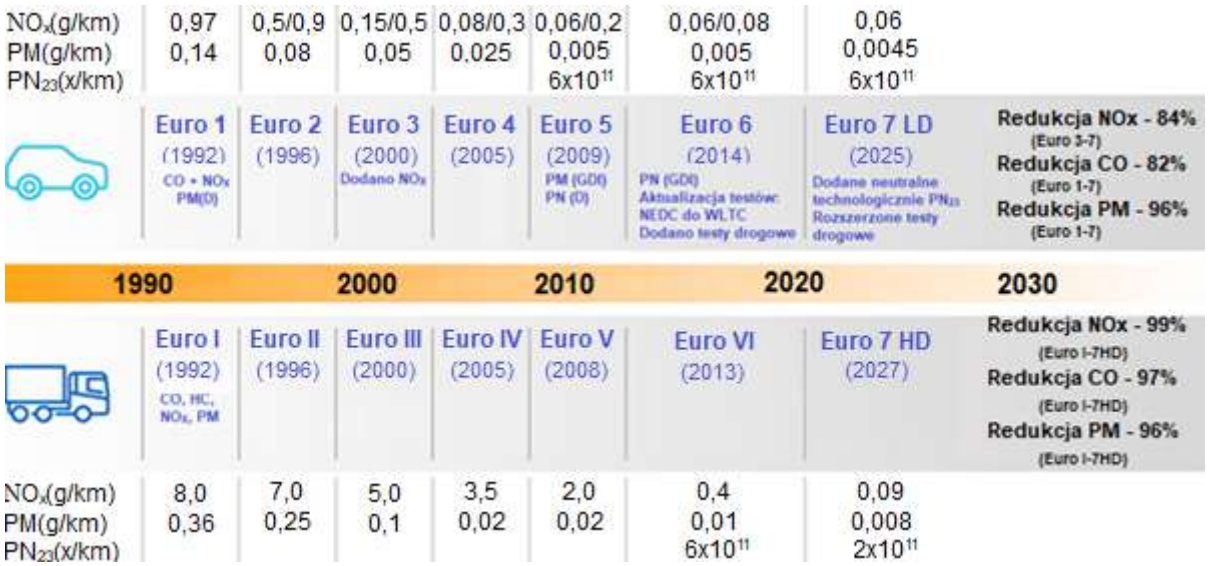
Aktualna globalna struktura wykorzystania źródeł energii (tzw. miks energetyczny) stanowi złożone połączenie różnorodnych źródeł energii, w tym paliw kopalnych dominujących w światowej produkcji energii, takich jak ropa naftowa, węgiel i gaz ziemny (rys. 1) [11]. Ze względu na ich negatywny wpływ na środowisko, głównie poprzez wpływ na emisję GHG, wzrasta potrzeba poszukiwania i implementacji bardziej zrównoważonych oraz ekologicznych rozwiązań.

Rys. 1. Światowy miks energetyczny w 2022 roku uwzględniający energię elektryczną, cieplną oraz transport [12]

Przyczyny i powody dążenia do zmiany globalnego miksu energetycznego są wielowymiarowe. Zmiany klimatyczne – spowodowane emisją GHG związaną z wykorzystywaniem paliw kopalnych – stanowią poważne zagrożenie dla stabilności ekologicznej, gospodarczej i społecznej na całym świecie. Zmniejszenie tej emisji jest kluczowe dla ograniczenia efektu globalnego ocieplenia i jego katastrofalnych skutków, takich jak ekstremalne zjawiska pogodowe, podnoszenie się poziomu oceanów czy utrata bioróżnorodności [13]. Kolejnym problemem jest brak równomiernego dostępu do zasobów energetycznych, co wymusza konieczność transportu surowców i uzależnia kraje ubogie w paliwa kopalne od ich zakupów w różnych, także geopolitycznie niestabilnych regionach. To skłania do poszukiwania bardziej niezależnych i odnawialnych źródeł energii [1]. Rozwój technologii wykorzystujących OŹE, takich jak energia słoneczna, wiatrowa czy geotermalna, zapewnia możliwość dywersyfikacji źródeł energii i zwiększenia bezpieczeństwa energetycznego. Sprzyjają temu postęp technologiczny i spadające koszty produkcji energii z OŹE [12], a presja społeczna i regulacje prawne także odgrywają w tym procesie istotną rolę.

Coraz większa świadomość ekologiczna społeczeństw oraz wprowadzanie przez rządy celów dotyczących redukcji emisji CO_2 i zwiększenia udziału energii odnawialnej w miksie energetycznym napędzają zmiany w kierunku bardziej zrównoważonych i ekologicznych systemów energetycznych.

Unia Europejska (UE) w zakresie regulacji emisji spalin wprowadziła normy emisyjne Euro dla pojazdów samochodowych [14]. Te normy, zaczynając od Euro 1 w 1992 roku, stopniowo zaostrzały limit emisji dla nowych pojazdów (rys. 2).

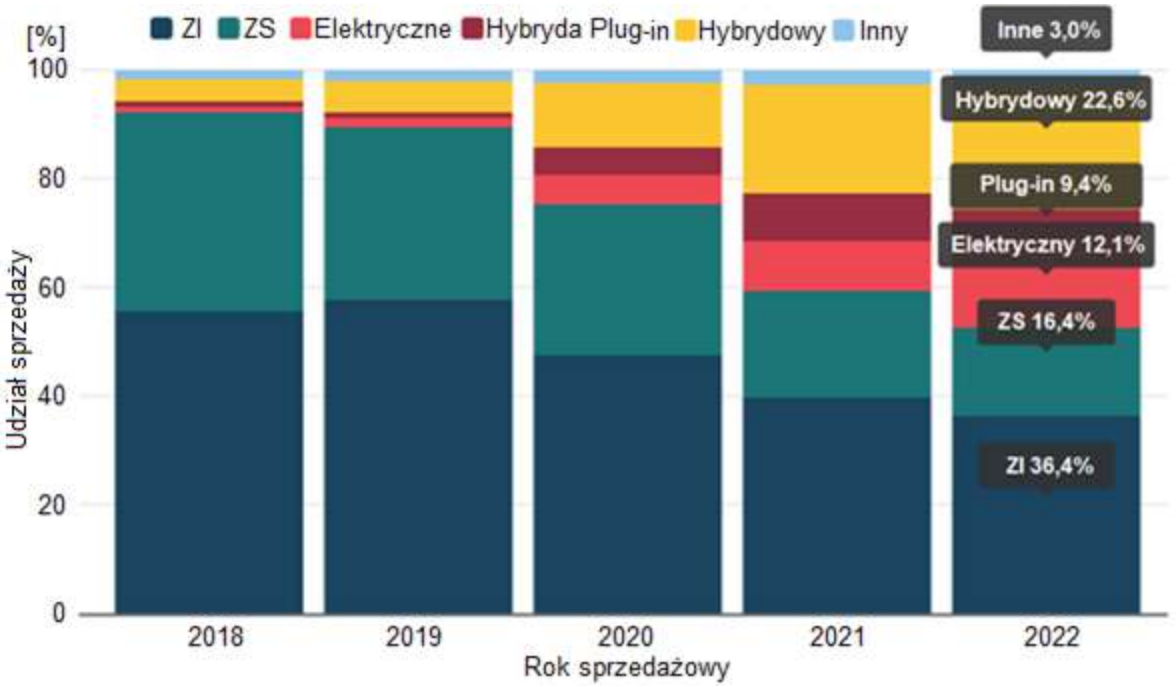


Rys. 2. Ewolucja ograniczeń emisyjnych dla pojazdów samochodowych w UE.
Opracowanie na podstawie [12]

UE stawia również na promocję pojazdów elektrycznych (ang. *Battery Electric Vehicle*, BEV) i hybrydowych jako część swojej strategii „Zielonego Ładu", mającej na celu osiągnięcie neutralności klimatycznej do 2050 roku [15]. W Stanach Zjednoczonych Ameryki (*ang. United States of America,* USA) kluczową rolę w transformacji ekologicznej odgrywa Agencja Ochrony Środowiska (*ang. Environmental Protection Agency,* EPA) [16] oraz Kalifornijska Rada Zasobów Powietrznych (*ang. California Air Resources Board,* CARB) [17]. Standardy EPA, znane jako Tier, zostały wprowadzone w latach 90. i były sukcesywnie zaostrzane [16]. Kalifornia posiada także unikalne uprawnienia do ustalania własnych, bardziej rygorystycznych norm w USA, czego przykładem może być program *Low-Emission Vehicle III* [18]. W USA promowany jest rozwój pojazdów z napędami alternatywnymi także poprzez różnego rodzaju ulgi podatkowe i subwencje [19]. W Azji Chiny i Japonia są na czele, jeśli chodzi o działania zmierzające do redukcji emisji szkodliwych składników spalin. Chiny, borykające się z poważnym zanieczyszczeniem powietrza, wprowadziły normy „China 6", które są jednymi z najbardziej rygorystycznych na świecie [20]. Japonia z kolei ma długą

historię regulacji ekologicznych, z normami emisji JP i silnym naciskiem na rozwój pojazdów z napędem hybrydowym i BEV [21]. Kraje takie jak Indie, Brazylia czy Rosja również wprowadzają własne normy emisji, często wzorując się na regulacjach europejskich czy amerykańskich [22], jednakże stopień ich rygorystyczności i skuteczności wdrażania różni się w zależności od kraju. Globalne dążenie do redukcji emisji szkodliwych składników spalin z pojazdów jest procesem złożonym i niejednorodnym, wymagającym współpracy międzynarodowej oraz innowacji technologicznych. Różnorodność podejść w poszczególnych regionach świata świadczy o kompleksowości wyzwań związanych z ochroną środowiska.

Wyznacznikiem dokonujących się zmian może być struktura sprzedaży nowych samochodów osobowych w UE (rys. 3).



Rys. 3. Udział sprzedaży nowych pojazdów osobowych w UE w latach 2018–2022 – podział według źródła zasilania. Opracowanie na podstawie [23]

Obserwuje się wyraźny trend spadku sprzedaży samochodów osobowych wyposażonych w silniki spalinowe, co jest efektem szeregu czynników, w tym m.in. wprowadzenia rygorystycznych norm emisyjnych Euro czy ujawnienia skandali związanych z manipulacją wynikami pomiarów emisji przez niektóre przedsiębiorstwa motoryzacyjne. Wydarzenia te w znaczący sposób przyczyniły się do podważenia zaufania konsumentów do tradycyjnych technologii napędowych. W tym samym czasie rośnie sprzedaż samochodów hybrydowych i BEV. Zjawisko to można przypisać rosnącej świadomości ekologicznej społeczeństwa oraz postrzeganiu tych technologii jako bardziej przyjaznych dla środowiska. Istotnym czynnikiem przyczyniającym się do tej zmiany są też korzyści finansowe, takie jak niższe opłaty podatkowe

czy możliwość skorzystania z dotacji rządowych, które są oferowane jako zachęta do inwestycji w pojazdy o mniejszym wpływie na środowisko [24,25,26]. Mimo zauważalnej zmiany preferencji konsumentów i dynamicznego rozwoju rynku pojazdów elektrycznych oraz hybrydowych samochody z napędem spalinowym wciąż dominują na rynku. Jednocześnie ich udział systematycznie maleje, co wskazuje na długoterminową transformację w kierunku bardziej zrównoważonej mobilności.

Trendy elektryfikacji i hybrydyzacji, które przedstawiono na rys. 3, stanowią fundamentalne elementy strategii dekarbonizacji sektora transportu. Mimo iż przynoszą one szereg korzyści, takich jak zminimalizowanie emisji lokalnych, wysoka efektywność energetyczna i zmniejszenie poziomu hałasu, napotykają również na istotne ograniczenia, które sprawiają, że nie są w stanie w pełni zaspokoić potrzeb rynku. Elektryfikacja boryka się z wyzwaniami związanymi z relatywnie niską gęstością energii akumulatorów elektrochemicznych, co przekłada się na ograniczony zasięg pojazdów na jednym ładowaniu. Problem stanowią również: długi czas ładowania oraz kwestie związane z wpływem na środowisko produkcji i utylizacji akumulatorów. Proces elektryfikacji jest szczególnie ograniczony w transporcie ciężkim i długodystansowym, na przykład w pojazdach ciężarowych operujących na trasach międzynarodowych, w statkach dalekomorskich czy w samolotach [27]. Hybrydyzacja – podobnie jak w przypadku elektryfikacji – ma uzasadnienie w pojazdach osobowych, ale wysoka energochłonność transportu ciężkiego i dalekobieżnego minimalizuje korzyści związane z tą technologią [28]. W związku z powyższym środowiska naukowe [29,30] oraz branże zainteresowane [31,32,33,34] typują, że w najbliższej przyszłości w napędach wykorzystywanych w transporcie energochłonnym dominować będą technologie wodorowe (głównie w formie transformacji wodoru w energię elektryczną w ogniwach paliwowych) oraz stosowanie paliw pochodzenia niekopalnego, tj. paliw syntetycznych i biopaliw. Na rys. 4 pokazano prognozę doboru paliw i technologii dla wybranych napędów energochłonnych.

		Dalekomorski		Blisko-morski i śródlądowy		Lotniczy kontynentalny		Lotniczy międzykontynentalny		Kołowy dalekobieżny		Sprzęt rolniczy	
Główne źródło napędu	Silnik spalania wewnętrznego												
	Akumulatory chemiczne												
	Ogniwa wodorowe												
Napęd hybrydowy													
Paliwo	LNG/BioLNG*												
	CNG/BioCNG*												
	Hydrorafinowany olej roślinny (HVO)*												
	Metanol – CH_3OH*												
	Wodór – H_2*												
	Amoniak – NH_3*												
	Paliwa syntetyczne*												
	**Zrównoważone paliwo lotnicze – SAF*												
Okresy wykorzystania		2024 2030	2030 2050	2024 2030	2030 2050	2024 2030	2030 2050	2024 2030	2030 2050	2024 2030	2030 2050	2024 2030	2030 2050

*aktualnie problemy z ograniczoną dostępnością i ceną
**SAF (ang. *sustainable aviation fuel*) to forma paliwa syntetycznego

Wykonalne	W fazie badań i rozwoju	Poważne bariery w aplikacji

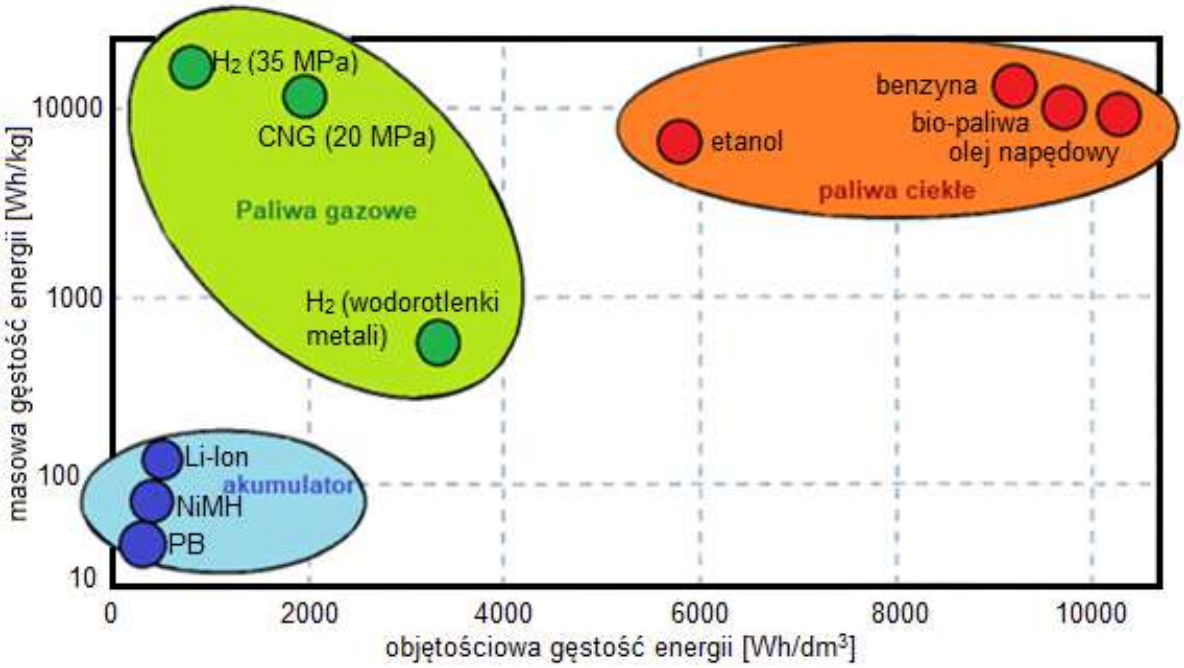
Rys. 4. Prognoza wykorzystania technologii i paliw przyszłości dla energochłonnych form transportu [34]

Zestawienie zaprezentowane powyżej ma swoje uzasadnienie w aspektach dotyczących efektywności ekonomicznej i logistycznej, ograniczeń fizykochemicznych, infrastrukturalnych oraz ilości dostępnych niezbędnych zasobów [35,36].

W kontekście przewidywanego wzrostu znaczenia pojazdów hybrydowych i BEV w strukturze transportu kluczowym wyzwaniem staje się zapewnienie odpowiedniej ilości i dostępności niskoemisyjnej energii elektrycznej. Realizacja tego celu jest ściśle powiązana z koniecznością dokonania znacznych inwestycji w OŹE oraz rozbudowę infrastruktury energetycznej [37]. Jednocześnie zasoby pierwiastków ziem rzadkich, kluczowych dla produkcji akumulatorów i ogniw paliwowych, są ograniczone i geograficznie skoncentrowane,

co może rodzić np. utrudnienia w łańcuchach dostaw [38]. Kwestia produkcji wodoru, szczególnie poprzez metodę elektrolizy wymagającą znaczących ilości wody słodkiej, wpisuje się w szerszy kontekst globalnych wyzwań związanych z dostępem do wody [39].

Z fizykochemicznego punktu widzenia niska gęstość energii wodoru oraz akumulatorów elektrochemicznych – w porównaniu z wartościami uzyskiwanymi w przypadku paliw węglowodorowych (rys. 5) [40] – wpływa negatywnie na zasięg BEV i pojazdów wodorowych (ang. *Fuel Cell Electric Vehicles* – FCEV). Ponadto wykorzystujące wodór ogniwa paliwowe w FCEV są wrażliwe na temperaturę pracy, co ogranicza ich sprawność w zależności od pory roku czy miejsca działania [41]. Wymagana do prawidłowego funkcjonowania ogniw wysoka czystość paliwa oraz ich wysoka podatność na uszkodzenia mechaniczne stanowią kolejne wyzwania techniczne [41]. Wodór w stanie gazowym ma bardzo niską gęstość energii na jednostkę objętości (rys. 5), co sprawia, że efektywne przechowywanie i transport tego paliwa są trudne do realizacji. W logistyce wodoru stosuje się zbiorniki wysokociśnieniowe lub schładza H_2 do stanu ciekłego przy temperaturze niższej niż -253°C, co wymaga użycia drogich zbiorników kriogenicznych [42]. Dodatkowo ze względu na małe rozmiary cząsteczek wodoru istnieją wyzwania związane z uszczelnianiem i zapobieganiem wyciekom, co jest kluczowe dla zapewnienia bezpieczeństwa [29].



Rys. 5. Gęstość energetyczna paliw ciekłych, gazowych oraz akumulatorów elektrochemicznych [40]

W kontekście infrastrukturalnym zarówno sieć szybkich ładowarek elektrycznych dla BEV, jak i zaplecze dystrybucji wodoru są obecnie zdecydowanie mniej rozwinięte niż rozwiązania stosowane w przypadku tradycyjnych paliw węglowodorowych [43]. Istotnym

ograniczeniem w kontekście transportu kołowego jest również infrastruktura drogowa przegotowana pod maksymalne wymiary, naciski osi i masę całkowitą pojazdów ciężarowych [44]. Zastosowanie w nich ciężkich akumulatorów bądź zbiorników wodoru o dużej objętości w znaczący sposób ogranicza ładowność masową i objętościową pojazdu [45].

Wysokie koszty początkowe inwestycji w pojazdy elektryczne, infrastrukturę ładowania lub stacje wodorowe mogą zmniejszać ich atrakcyjność dla konsumentów i przedsiębiorstw w kontekście efektywności ekonomicznej i logistycznej tych technologii [42]. Ponadto kwestie związane z wydajnością operacyjną, takie jak czas ładowania akumulatorów lub tankowania wodoru, wpływają na logistykę, zwłaszcza w transporcie ciężkim i długodystansowym [45].

W efekcie mimo że elektryfikacja, hybrydyzacja i technologie wodorowe mają duży potencjał ekologiczny, w szczególności w pojazdach osobowych, ich pełna implementacja w sektorach transportu energochłonnego oraz pokrewnych sektorach gospodarki może być ograniczona [29,34,42]. Sam proces dekarbonizacji wedle wielu naukowców [4,42,46], jak również poprzez strategie rynkowe dużych producentów pojazdów ciężarowych [47,48,49,50] będzie wymagał kompleksowego podejścia. W nim swoje miejsce będą miały również pojazdy zasilane wysokosprawnymi silnikami tłokowymi spalania wewnętrznego. Nieocenionymi zaletami napędów wykorzystujących silniki spalinowe są ponad 100-letnia historia ich rozwoju (w tym optymalizacji procesów roboczych i sterowania) oraz w odróżnieniu np. od napędów wykorzystujących trakcyjne akumulatory elektrochemiczne, możliwość poddania ich całkowitemu i bezpiecznemu procesowi recyklingu [4]. Koniecznym elementem determinującym możliwość wykorzystania silników tłokowych w przyszłości będzie ograniczenie użycia paliw kopalnych i sukcesywne zastępowanie ich mało- i docelowo zeroemisyjnymi substytutami [42].

2.2. Idea retrofittingu w zasilanie dwupaliwowe

Retrofitting w branży motoryzacyjnej jest procesem modernizacyjnym, który pozwala na adaptację istniejących pojazdów do nowych standardów technicznych i ekologicznych. Jego głównym celem jest nie tylko poprawa parametrów eksploatacyjnych, takich jak ogólnie pojęta wydajność i bezpieczeństwo, lecz także zwiększenie ekologiczności pojazdów poprzez zmniejszenie emisji szkodliwych substancji oraz przedłużenie ich całkowitego czasu eksploatacji [8,9,51,52,53]. Takie działania wpisują się w globalne tendencje zrównoważonego rozwoju.

Pojazdy wyposażone w silniki o ZS stanowią szczególny cel dla retrofittingu ze względu na ich powszechne użycie w transporcie i istotny wpływ na zanieczyszczenie powietrza. Tradycyjne silniki o ZS znane są z wysokiej sprawności i trwałości, ale niestety również z emisji znacznych ilości NO_x i PM, które są szkodliwe dla zdrowia ludzi i środowiska [54]. Retrofitting tych silników, poprzez wdrażanie zaawansowanych systemów emisji, takich jak filtry cząstek stałych (ang. *Diesel Particulate Filter,* DPF) i systemy selektywnej redukcji katalitycznej (ang. *Selective Catalytic Reduction,* SCR), czy połączenie w system hybrydowy z napędem elektrycznym, może znacząco zmniejszyć emisję szkodliwych substancji [52]. Wpisuje się to również w kluczowy aspekt retrofittingu dotyczący zwiększenia użyteczności pojazdu – poprawa klasy jakości emisji spalin może zdeterminować o np. możliwości wjazdu do stref czystego transportu czy ograniczania opłat drogowych [55]. Dodatkowo wymiana starych komponentów na nowocześniejsze może wydłużyć żywotność pojazdu, zmniejszyć zużycie paliwa bądź pozwolić na użycie jego tańszego rodzaju, co jest ekonomicznie korzystne i zgodne z zasadami zrównoważonego rozwoju. Ponieważ każdorazowo produkcja nowego pojazdu – w szczególności wykorzystującego akumulatory elektrochemiczne – wiąże się z dużą emisją dwutlenku węgla, przedłużenie życia istniejących pojazdów może mieć znaczący wpływ na jego redukcję. Potwierdzeniem zasadności tej drogi są programy rządowe i działania producentów pojazdów [52,56,57,58].

Zastosowanie zasilania dwupaliwowego w używanych silnikach o ZS stanowi obiecujące rozwiązanie w obliczu rosnących wymagań efektywności energetycznej i presji ekologicznej. Podstawę tej technologii stanowią systemy zasilania pojazdów, które umożliwiają współspalanie dwóch rodzajów paliw o zróżnicowanych właściwościach fizykochemicznych. Typowo w rozwiązywaniach samochodowych wykorzystuje się ON (lub jego zamienniki) jako paliwo o wysokiej reaktywności oraz paliwo niskoreaktywne – najczęściej gaz ziemny w formie sprężonej (ang. *Compressed Natural Gas,* CNG) lub skroplonej (ang. *Liquefied Natural Gas,* LNG) albo skroplony gaz propan-butan (ang. *Liquefied Petroleum Gas,* LPG) [53]. Dzięki optymalizacji procesu spalania i możliwości wykorzystania paliw o mniejszej zawartości cząstek węgla silniki dwupaliwowe mogą znacząco przyczynić się do poprawy jakości powietrza i ograniczenia wpływu transportu na środowisko naturalne [58,64]. Jedną z głównych zalet dwupaliwowości jest redukcja emisji CO_2. Możliwość zwiększenia sprawności ogólnej stanowi kolejny istotny atut zasilania dwupaliwowego, co szerzej opisano w podrozdziale 2.4.3. pracy. Duża elastyczność paliwowa układów zasilania dwupaliwowego, umożliwiająca wykorzystanie różnorodnych źródeł energetycznych takich jak: oleje roślinne, biogaz czy

wodór, otwiera nowe perspektywy dla zrównoważonego rozwoju i wykorzystania lokalnych zasobów energetycznych [53,54]. Koszty eksploatacyjne silników dwupaliwowych mogą być niższe w porównaniu z kosztami tradycyjnych silników o ZS, głównie dzięki zastosowaniu tańszych paliw alternatywnych i redukcji kosztów wynikających z norm emisji spalin.

Mimo wielu zalet implementacja systemów dwupaliwowych wiąże się z wyzwaniami. Poza badaniami badawczo-rozwojowymi niezbędne są również analizy dotyczące rynku, uzasadniające podjęcie tematyki oraz odnajdujące potencjalne grupy obiorców mogące być beneficjentami nowoczesnej technologii dwupaliwowej.

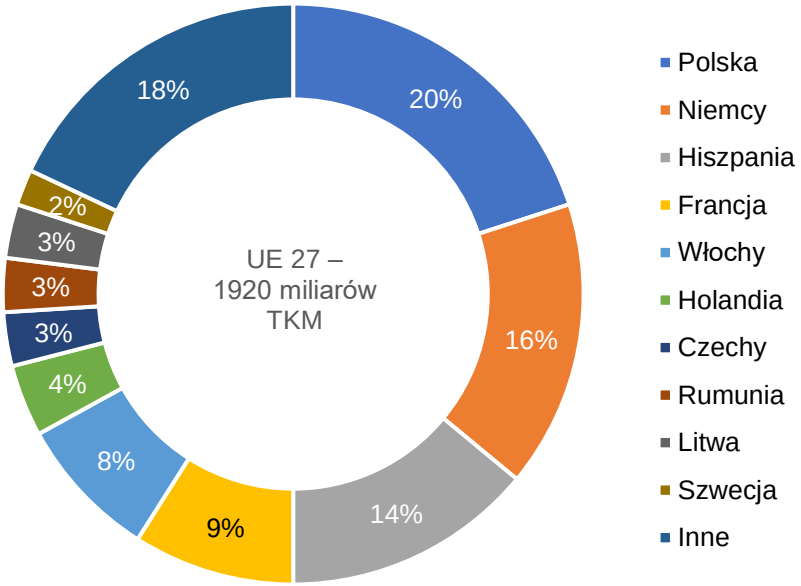
2.3. Analiza rynku, potencjału aplikacyjnego i bezpieczeństwa

Niefabryczne instalacje zasilania dwupaliwowego to najczęściej konstrukcje bazujące na instalacjach IV generacji zasilania LPG [54] – szerzej ten temat zostanie opisany w podrozdziale 2.4.1. Rynek nowych i używanych pojazdów osobowych z silnikami o zapłonie iskrowym (ZI) z pośrednim wtryskiem benzyny, będący największym i typowym dla producentów niefabrycznych instalacji zasilania LPG, od kilku lat zmniejsza się, a w dłuższej perspektywie nastąpi przyśpieszenie tego procesu. Wynika to z dokonującej się elektryfikacji pojazdów osobowych (co potwierdzają deklaracje producentów [60–62]) oraz naturalnego rozwoju układów zasilania paliwem w silnikach o ZI w kierunku wykorzystania wtrysku bezpośredniego i mieszanego [54], co utrudnia unifikację rozwiązań technicznych, a co dotychczas miało miejsce w przypadku najbardziej popularnych instalacji LPG dla silników o ZI z wtryskiem pośrednim [63]. Sytuacja ta stanowi szczególny problem dla firm zajmujących się samochodowymi instalacjami zasilania gazowego i ich podwykonawców. Polska branża instalacji gazowych jest jedną z największych na świecie [64]. Szacuje się, że po polskich drogach porusza się ok. 3,5 miliona pojazdów zasilanych paliwem gazowym [65], a na polskim rynku działa ponad 200 podmiotów w tej branży. Wszystkie te podmioty poszukują nowych perspektyw rynkowych [64].

W przeciwieństwie do rozwoju silników o ZI rozwój systemów zasilania silników o ZS otworzył szersze możliwości stosowania gazowo-olejowych układów zasilania dwupaliwowego. Rozwój wysokociśnieniowych wtryskiwaczy takich jak pompowtryskiwacze (niem. *Pumpe-Düse-system*, PD) czy układy *Common Rail* (CR) wykorzystujące elektroniczne systemy sterowania umożliwiły precyzyjne kontrolowanie procesu spalania w silnikach o ZS dzięki strategii wielofazowego wtryskiwania paliwa [66]. Możliwości wtryskiwania

wieloetapowego można również wykorzystać do efektywnego współspalania paliw przy dużym stopniu wymiany paliwa podstawowego, tj. ON [53] na paliwo gazowe w układach dwupaliwowych. Znane jeszcze z lat 90. XX wieku systemy „diesel-gaz", charakteryzujące się niskim współczynnikiem wymiany paliw, bazujące na układach zasilania silników o ZS z mechanicznym wtryskiem, nie zdobyły popularności rynkowej [53]. Poza upowszechnieniem się elektronicznie sterowanego, wysokociśnieniowego, wieloetapowego wtrysku ON w silnikach o ZS aspektem technicznym predysponującym do zasilania dwupaliwowego jest popularyzacja i rozpowszechnienie się nowoczesnych systemów turbodoładowania (turbosprężarki ze zmienną geometrią łopat, sterowalne systemy upustu i recyrkulacji spalin) oraz zawansowanych systemów redukcji emisji szkodliwych składników spalin. Otwiera to nowe perspektywy wykorzystania instalacji dwupaliwowych w silnikach o ZS będących aktualnie w eksploatacji [53,54,66].

Drogowy transport towarów odgrywa ogromną rolę w gospodarce i jest kluczowym elementem handlu krajowego i międzynarodowego w Europie [68]. W sektorze drogowego transportu ciężkiego UE silniki o ZS stanowią główne źródło napędów 96% pojazdów [25,67]. Polska jest liderem branży transportu drogowego w UE i odpowiada za 19,8% całkowitej liczby tonokilometrów (TKM) w UE, wyprzedzając Niemcy – 16,0% i Hiszpanię – 14,1% (rys. 6) [69].

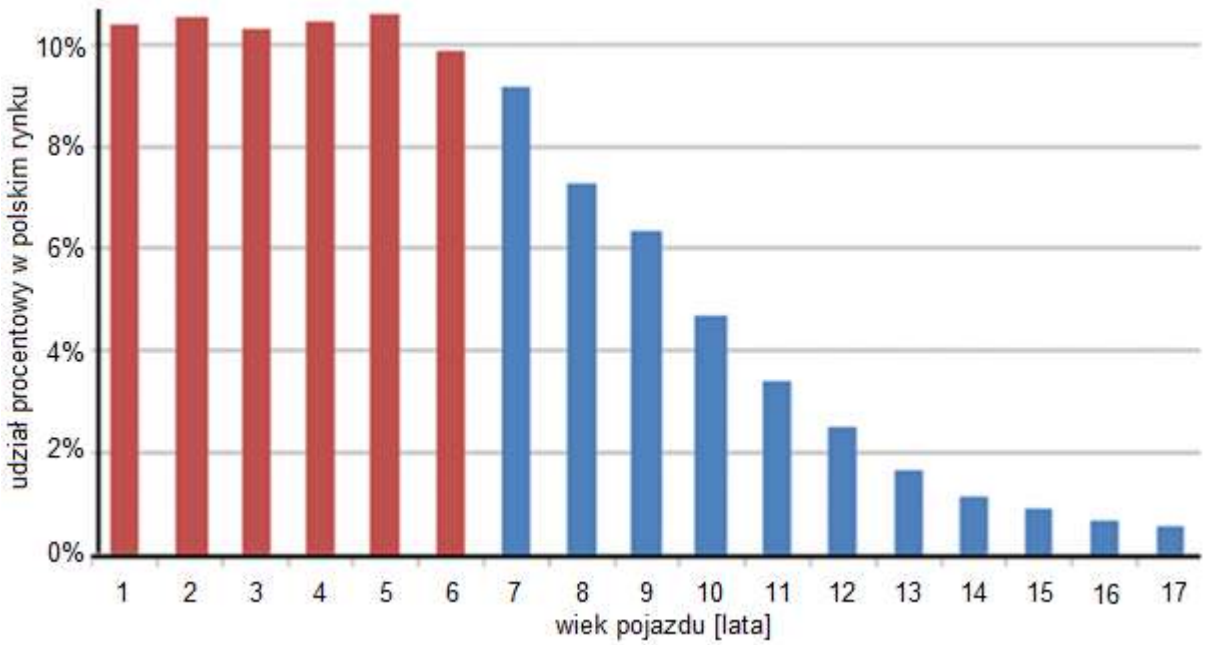


Rys. 6. Udział w rynku transportu drogowego w UE dla 10 krajów o największej wykonanej liczbie tonokilometrów w 2022 r. [69,70]

W 2022 roku po UE poruszało się ponad 6,2 mln pojazdów ciężarowych [71]. Ten energochłonny sektor, ze względu na swoją znaczącą wielkość i istotność w strukturze

gospodarczej, powinien być odpowiednio przygotowany na nadchodzące transformacje związane z procesem dekarbonizacji. Jednocześnie stanowi on pole zainteresowania dla przedsiębiorstw specjalizujących się w niefabrycznych instalacjach zasilania gazowego.

Ekonomicznie uzasadniony czas eksploatacji pojazdów ciężarowych jest zróżnicowany w zależności od warunków użytkowania, przy czym dla transportu międzynarodowego wynosi on około 5–7 lat, a dla krajowego i lokalnego 10–15 lat, z dłuższym okresem dla pojazdów specjalistycznych [72]. W ostatnich latach zauważalny jest znaczący wzrost liczby zarejestrowanych ciągników siodłowych w Polsce. Rozkład wiekowy polskiej floty pojazdów ciężarowych zaprezentowano na rys. 7 [73]. Warto podkreślić, że pojazdy o wieku do sześciu lat stanowią obecnie znaczącą większość floty, co ma implikacje dla potencjalnej możliwości adaptacji instalacji dwupaliwowej [66].



Rys. 7. Procentowy udział pojazdów ciężarowych na rynku polskim z podziałem na wiek pojazdu [66,69]

Pojazdy, których dalsza eksploatacja nie opłaca się w UE, często są eksportowane do krajów Europy Wschodniej, gdzie konkurują z nowymi pojazdami z Chin i Turcji. Polska, która w latach poprzednich importowała używane pojazdy, obecnie stała się eksporterem, sprzedając ciężarówki przed osiągnięciem technicznego końca ich żywotności do krajów takich jak Tunezja czy Pakistan [72].

Rynek transportu drogowego w Polsce charakteryzuje się wysokim poziomem fragmentacji ze względu na wielkość firm – mikroprzedsiębiorstwa odgrywają znaczącą rolę, mimo iż ich samodzielny wkład w całkowitą pracę przewozową, mierzoną w TKM, pozostaje

ograniczony. Mniejsze podmioty gospodarcze mają jednak istotny udział w rynku przewozów ładunków [65]. Jednocześnie dostrzegalny jest trend wzmacniania pozycji większych przedsiębiorstw transportowych, co może sygnalizować kierunek zmian w zakresie inwestycji w nowoczesne technologie transportowe, w tym systemy dwupaliwowe. Analiza struktury floty ciągników siodłowych ukazuje, że dominuje posiadanie ograniczonej liczby pojazdów przez większość firm, co może być barierą w adaptacji zaawansowanych technologicznie rozwiązań. Niemniej jednak obserwuje się wzrost liczby pojazdów w segmencie przedsiębiorstw posiadających flotę w zakresie od 21 do 50 pojazdów [65,73], co sugeruje, że przedsiębiorstwa te mogą wykazywać większą gotowość do inwestowania w nowoczesne technologie, w tym w rozwiązania dwupaliwowe. Szczegółową analizę struktury rynku i możliwości adaptacji używanych pojazdów ciężarowych w zasilanie dwupaliwowe zawarto w publikacji [A4].

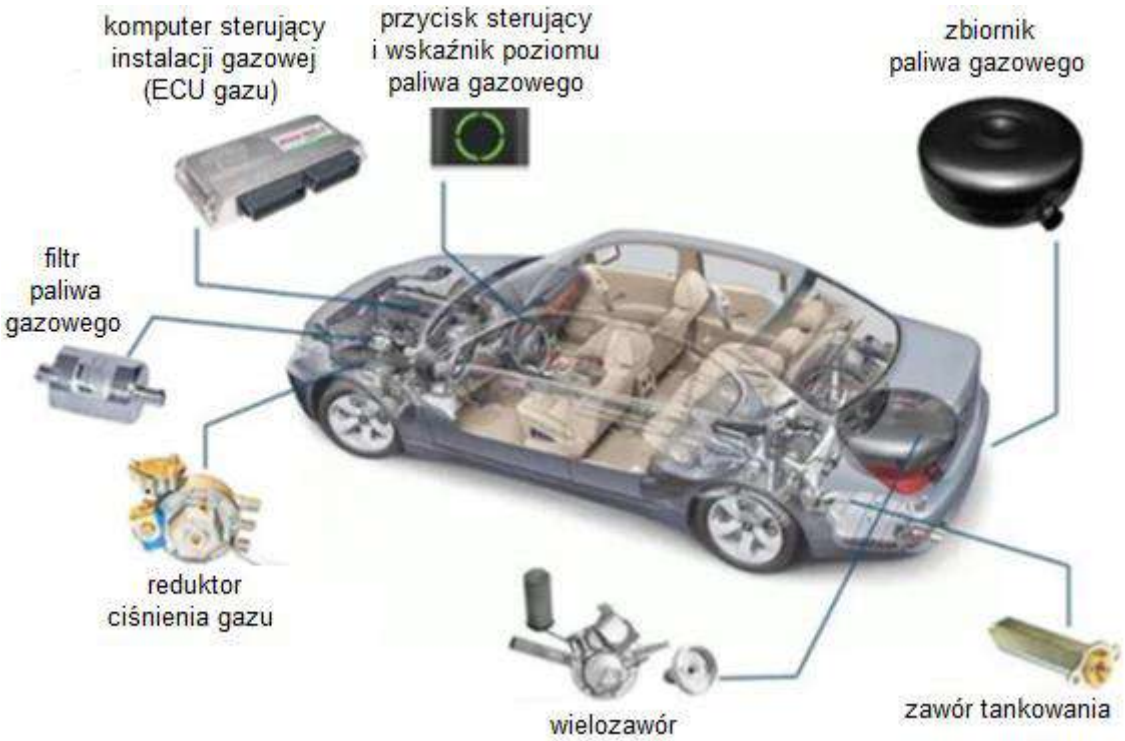
W ramach innych badań przeprowadzonych przez autora, opublikowanych w artykule [A12], dokonano wielokryterialnej analizy atrakcyjności rynkowej nowo wprowadzanych do sprzedaży w UE w 2023 roku ciągników siodłowych Volvo FH. Analiza ta obejmowała modele z napędem konwencjonalnym wyposażonym w silnik o ZS zasilanym ON i HVO (spełniającym normę emisji spalin Euro VI), pojazd z rozwiązaniem dwupaliwowym oraz BEV. Wyniki wskazują, że pomimo wyższej ceny zakupu, całkowity koszt eksploatacji ciągnika siodłowego BEV jest niższy w porównaniu z kosztem pojazdów spalinowych, przy czym system dwupaliwowy znajduje się pomiędzy napędem o ZS i elektrycznym. Zauważono, że BEV charakteryzują się mniejszą efektywnością w transporcie długodystansowym. Z powodów nieefektywności logistycznej, w związku z ograniczeniami szczegółowo opisanymi w podrozdziale 2.1., ich zastosowanie zostaje zawężone do obszarów miejskich i transportu międzymiastowego. Jednym z istotnych wniosków pracy jest stwierdzenie, że potencjalne korzyści płynące z użytkowania pojazdów spalinowych, w tym wariantów dwupaliwowych wykorzystujących biopaliwa, mogą przewyższyć zalety napędu elektrycznego w regionach, gdzie produkcja energii elektrycznej charakteryzuje się wysoką emisją CO_2. Problem ten jest szczególnie istotny dla rynku polskiego, na którym średnia emisja ekwiwalentu dwutlenku węgla (CO_2-*eq*) na każdą wyprodukowaną kilowatogodzinę (kWh) energii elektrycznej wynosi około 650 gramów, w porównaniu ze średnią unijną na rok 2023, która wynosi 213 gramów CO_2-*eq*/kWh [68]. W świetle przeprowadzonych analiz pojazdy dwupaliwowe bezwzględnie wypełniają lukę na rynku transportu dalekobieżnego oraz zdają się być efektywną alternatywą dla transportu bliskiego do momentu zmiany struktury miksu energetycznego.

W literaturze przedmiotu często podkreśla się, że percepcje społeczne dotyczące bezpieczeństwa pojazdów mogą znacząco wpływać na decyzje konsumentów [74]. Stereotypy te, zakorzenione w obawach przed zwiększonym ryzykiem wybuchu lub pożaru w wyniku kolizji bądź awarii, często potęgowane są przez medialne doniesienia o jednostkowych incydentach, co prowadzi do przesadzonej percepcji ryzyka [75,76]. W ramach badań przeprowadzonych przez autora w artykule [A3] dokonano analizy bezpieczeństwa rozwiązań instalacji zasilania gazowego w pojazdach samochodowych. Artykuł podkreśla wysokie bezpieczeństwo instalacji zasilania paliwami gazowymi w samochodach, akcentując, że zarówno aspekty prawne, jak i techniczne zapewniają ich bezpieczne użytkowanie. Wskazano przy tym, że paliwa gazowe są bezpieczniejsze niż konwencjonalne podczas kolizji czy pożarów, pod warunkiem że instalacje nie są przedmiotem rażącego zaniedbania w eksploatacji i konserwacji – o czym świadczą również dane statystyczne [77]. Stoi to w opozycji do sytuacji pojazdów elektrycznych, które wprowadzają odmienny zakres wyzwań, w szczególności w odniesieniu do pożarów akumulatorów, które mogą być trudniejsze do opanowania niż tradycyjne pożary paliw węglowodorowych [77].

2.4. Systemy spalania dwupaliwowego dla silników o ZS

2.4.1. Założenia ogólne budowy samochodowych instalacji gazowych

Samochodowe instalacje gazowe to najczęściej alternatywne systemy zasilania silników o ZI i ZS ściśle współpracujące z oryginalnym systemem zasilania. Niezależnie od typu podstawowego układu zasilania instalacje gazowe muszą spełniać wysokie standardy efektywności działania i bezpieczeństwa zgodne z wymaganiami określonymi np. w dokumentach homologacyjnych, dyrektywach i regulaminach Europejskiej Komisji Gospodarczej Organizacji Narodów Zjednoczonych. W literaturze przedmiotu [53,78,79,80] wskazuje się, że różne instalacje gazowe wykorzystują podobny zestaw komponentów podstawowych, w tym elementy takie jak: wlewy, zbiorniki, przewody oraz armatura, a także specyficzne komponenty umieszczone w komorze silnika, np. reduktor ciśnienia gazu (rys. 8).



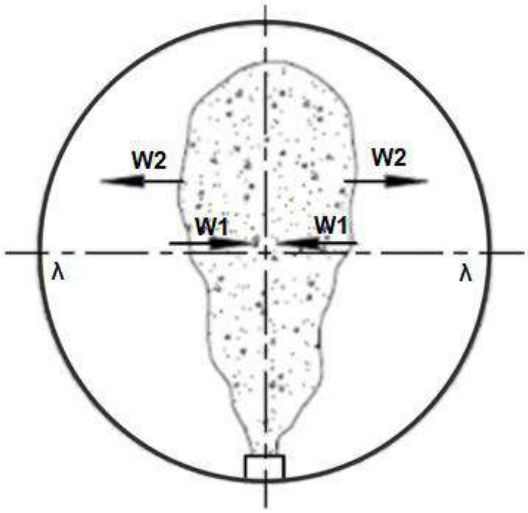
Rys. 8. Przykładowa lokalizacja komponentów w samochodowej instalacji gazowej [81]

Wyróżnia się również kategorie systemów gazowych w kontekście ich montażu i funkcjonalności: systemy fabryczne, systemy montowane dodatkowo jako uzupełnienie układu zasilania fabrycznego (co wpisuje się w koncepcję retrofittingu) oraz systemy jednopaliwowe – przykładem tych ostatnich mogą być autobusy miejskie wykorzystujące CNG w zmodyfikowanych silnikach o ZS realizujących spalanie ZI. Niefabryczne systemy zasilania dwupaliwowego do silników o ZS najczęściej są konstrukcyjnie podobne do odpowiednich generacji typowych instalacji zasilania LPG dla silników o ZI. Ogólna struktura i podział instalacji gazowych w pojazdach zostały szczegółowo przedstawione w literaturze, gdzie zaznacza się zarówno ich uniwersalne komponenty, jak i specyficzne aspekty konstrukcyjne odpowiadające różnym typom pojazdów i silników czy koncepcjom realizowanego spalania [78,80].

2.4.2. Konwencjonalne spalanie dwupaliwowe

W silniku dwupaliwowym konwencjonalnego spalania (ang. *Conventional Dual Fuel*, CDF) do cylindra dostarczana jest mieszanina paliwa niskoreaktywnego w fazie lotnej z powietrzem o składzie określonym przez współczynnik nadmiaru powietrza (λ), najczęściej w przedziale 1,0 - 1,5 [53,82].

Mieszanina gazowa – przy stosowanym w silniku stopniu sprężania – nie może ulec samozapłonowi w suwie sprężania z uwagi na różnice fizykochemiczne paliwa niskoreaktywnego, takie jak wysoka liczba oktanowa (LO) oraz wyższa temperatura samozapłonu (dla przykładu gaz ziemny ma prawie dwukrotnie wyższą od ON temperaturę samozapłonu) [53,82]. Mieszanina zapalana jest od dawki paliwa wysokoreaktywnego (najczęściej ON, wykorzystywanego jako paliwo bazowe w silnikach o ZS) wtryskiwanego do cylindra w suwie sprężania, przy kącie wyprzedzenia wtrysku standardowym dla silnika o ZS, w okolicach górnego martwego położenia tłoka (GMP); dawkę paliwa rozpoczynającą zapłon ładunku znajdującego się już w cylindrze nazywa się również dawką inicjującą. Wskutek wysokiej temperatury czynnika i obecności aktywnych centrów zapłonu paliwo wysokoreaktywne ulega samozapłonowi [54]. Najlepsze warunki samozapłonu ma paliwo wysokoreaktywne znajdujące się na zewnętrznej powierzchni strugi, gdzie pojawiają się warunki intensywnego parowania i mieszania z powietrzem oraz występuje wysokie stężenie tlenu [54,80]. Od powierzchni strugi płomień przenosi się zarówno do wnętrza strugi (W1 – rys. 9) w strefę bogatej mieszaniny obu paliw, jak i na zewnątrz (W2 – rys. 9) w strefę jednorodnej mieszaniny paliwa niskoreaktywnego i powietrza [53]. Ze względu na bogatszą mieszaninę w strefie tej panują wyższe temperatury ułatwiające zapłon i całkowite spalanie nawet ubogich mieszanin gazowych. Na zewnątrz strugi prędkość spalania gazu zmniejsza się w miarę oddalania się płomienia od strugi [53]. Schemat procesu pokazano na rys. 9.



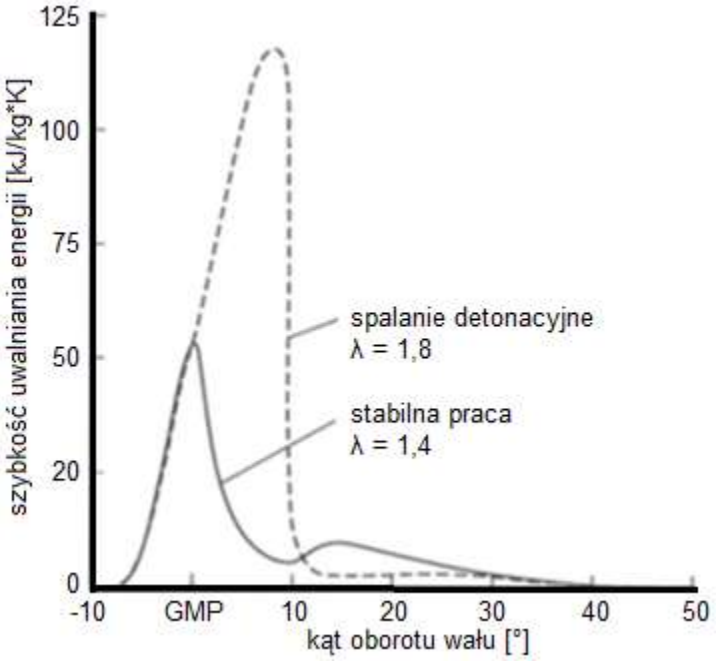
Rys. 9. Schemat procesu spalania gazu ziemnego i ON w silniku CDF:
λ –współczynnik nadmiaru powietrza mieszaniny paliwo niskoreaktywne – powietrze zasysanej do cylindra;
W1 – szybkość propagacji płomienia w strudze paliwa wysokoreaktywnego;
W2 – szybkość propagacji płomienia w mieszaninie gazowej [53]

W dwupaliwowych układach zasilania paliwo niskoreaktywne dostarczone razem z powietrzem w trakcie suwu ssania „zastępuje” ilość niezbędnej dawki wtrysku głównego paliwa wysokoreaktywnego. Zamianę paliwa podstawowego, wysokoreaktywnego na paliwo niskoreaktywne definiuje się jako współczynnik zastąpienia (*Wz*) zgodnie z ze wzorem (1) [53,82,83]:

$$W_Z = \frac{Ge_{SF} - Ge_{DF}}{Ge_{SF}} * 100\%, \tag{1}$$

gdzie: Ge_{SF} – godzinowe zużycie paliwa wysokoreaktywnego w trybie jednopaliwowym [kg/h],
Ge_{DF} – godzinowe zużycie paliwa wysokoreaktywnego podczas zasilania dwupaliwowego [kg/h].

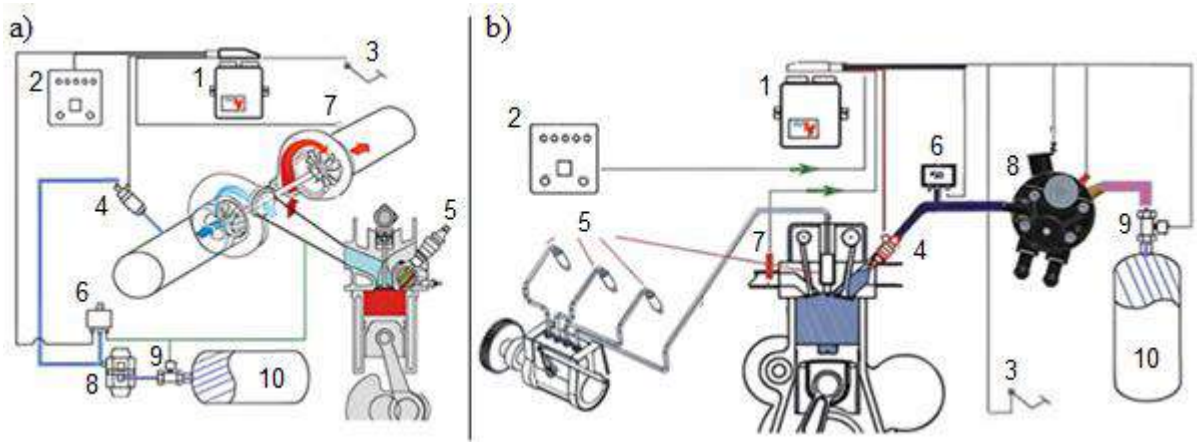
Całościowo proces spalania dwupaliwowego konwencjonalnego ma charakter „kinetyczny” – z wyraźnym występowaniem płomienia, co łączy systemy ZI i ZS. Na zmiany prędkości spalania deflagracyjnego istotnie wpływa skład mieszaniny palnej w cylindrze. Przy ubogich mieszaninach w okolicach ścianek komory spalania może dochodzić do zmniejszenia prędkości spalania lub nawet zaniku płomienia, co może prowadzić do zwiększonej emisji niespalonych węglowodorów (ang. *unburned hydrocarbons,* UHC). Z kolei duża ilość ubogiej mieszanki paliwowo-powietrznej po przeciwległej stronie do regionów spalania dawki inicjującej może podlegać szybkim reakcjom samozapłonu wspomaganym wysokim ciśnieniem i temperaturą, prowadząc do spalania detonacyjnego. Zbyt małe dawki paliwa wysokoreaktywnego mogą wywołać turbulentne spalanie mieszanki gazowo-powietrznej, co może w określonych warunkach, np. wysokiej temperatury, powodować występowanie spalania detonacyjnego podobnego do tego w silnikach o ZI. Na pojawianie się efektu spalania detonacyjnego zasadniczy wpływ ma również temperatura mieszanki [53,54,84]. Efekty zmian współczynnika nadmiaru powietrza są stosunkowo mniej poważne i głównie regulują wynikową temperaturę spalania (rys. 10), która przedstawia zmiany w szybkości uwalniania ciepła z silnika dwupaliwowego przy zwiększającym się stężeniu paliwa gazowego w układzie dolotowym silnika, powodując spalanie detonacyjne przy stałej objętości dawki inicjującej paliwa wysokoreaktywnego [53,54,84]. Więcej na temat przyczyn spalania detonacyjnego w CDF zaprezentowano w pracy [A5].



Rys. 10. Przebieg uwolnienia energii w funkcji obrotu wału korbowego silnika CDF dla dwóch różnych mieszanin propan-powietrze przy stałej dawce paliwa wysokoreaktywnego ON. Wz = 80%, nconst.= 1000 obr./min, temperatura powietrza w kolektorze dolotowym = 40°C. Opracowanie na podstawie [53]

Popularne na rynku rozwiązania niefabryczne samochodowego zasilania dwupaliwowego dla silników o ZS bazują wyłącznie na konwencjonalnym spalaniu dwupaliwowym. Systemy te, zwane potocznie jako „diesel-gaz" lub „gazodiesel", mają wiele różnych komercyjnych nazw [85,86]. Zasada działania tych systemów oparta jest na założeniach instalacji LPG dla silników o ZI, III lub IV generacji [80]. W każdym z nich paliwo wysokoreaktywne (typowo ON) wtryskiwane jest do komory spalania silnika z wykorzystaniem fabrycznego sytemu zasilnia (oraz bez modyfikacji oprogramowania sterownika silnika o ZS (ang. *Electronic Diesel Control*, EDC)), dodatkowa porcja paliwa niskoreaktywnego (zazwyczaj LPG lub CNG) dostarczana jest zaś z wykorzystaniem wtryskiwaczy gazowych wprost do układu dolotowego silnika. W niektórych rozwiązaniach bazujących na instalacji LPG III generacji stosuje się pojedynczy wtryskiwacz bądź wtryskiwacze pracujące w trybie *full group* (wszystkie wtryskiwacze działają jednocześnie), np. tuż przed turbosprężarką (rys. 11a) [87]. Obecność paliwa gazowego wpływa na pracę regulatora dawkowania ON, zmniejszając ilość dostarczanego paliwa. Dodatkowo, gdy to możliwe, system ten może emulować pozycję pedału przyspieszenia. Daje to możliwość jego aplikacji również do starszych silników o ZS z wtryskiem mechanicznym ON. W zależności od wariantu i silnika, w którym dokonuje się instalacji systemu bazującego na III generacji instalacji zasilania LPG, maksymalny współczynnik zastąpienia paliw wynosi ok. 20–40% [53]. W rozwiązaniach bazujących na instalacjach LPG IV generacji (rys. 11b)

wykorzystujących wtrysk sekwencyjny paliwa gazowego (ciśnienie pracy układu zazwyczaj w przedziale 1,5–3,0 bar) tuż przed zaworami, w zależności od wariantu i silnika, w którym dokonuje się aplikacji tego systemu, *Wz* maksymalny wynosi ok. 20–60%, ze średnią wymianą paliw w pełnym zakresie obciążeń silnika poniżej 50% [53,86]. Wysokie wartości *Wz* są wykonalne w szczególności w aplikacjach z układami CR, gdzie istnieje pewna niewielka możliwość zmniejszania dawki ON, poprzez emulowanie wskazań czujnika ciśnienia na listwie w paliwowej wtryskiwaczy ON i emulację pozycji pedału przyspieszenia jednocześnie. Popularność tych rozwiązań wynika z kompromisu rynkowego, w którym względnie uniwersalna konstrukcja instalacji, mała liczba modyfikacji mechanicznych koniecznych do wykonania w silniku, a co za tym idzie niska cena modyfikacji w zasilanie dwupaliwowe determinują zasadność ekonomiczną i możliwości wprowadzenia systemu do sprzedaży.



Rys. 11. Schemat niefabrycznych instalacji CDF: a) na bazie instalacji LPG III generacji [87]; b) na bazie instalacji LPG IV generacji [88]: 1 – sterownik gazowy, 2 – przełącznik, 3 – pedał przyspieszenia, 4 – wtryskiwacz gazu, 5 – wtryskiwacz ON, 6 – czujnik ciśnienia i temperatury gazu, 7 – czujnik temperatury spalin, 8 – reduktor cieśnienia gazu, 9 – elektrozawór odcinający przepływ gazu, 10 – zbiornik gazu

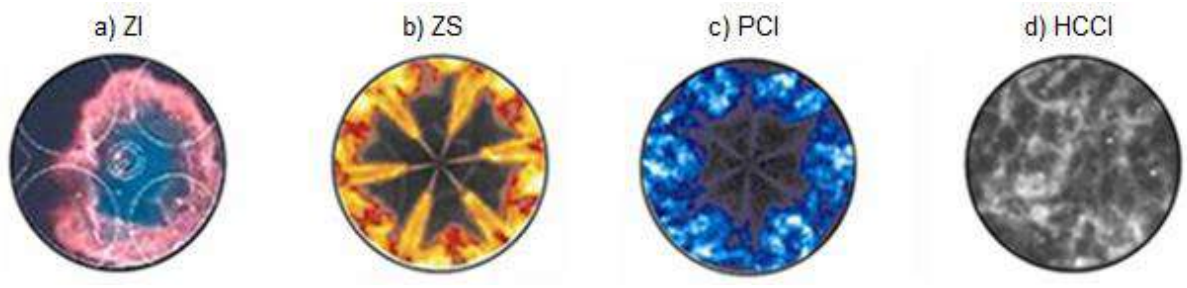
W rozwiązaniach fabrycznych zasilania dwupaliwowego stosowanych np. w jednostkach pływających oraz w niektórych pojazdach ciężarowych wykorzystywane są zaawansowane systemy CDF. W systemach tych zasilanie w paliwo niskoreaktywne może odbywać się w sposób pośredni i bezpośredni. Charakteryzują się one wysokim *Wz* na poziomie 95–99% w pełnym zakresie obciążeń, a sterowanie wykonywane jest w całości przez jeden ECU silnika. Fabryczne silniki CDF bazują na konstrukcjach silników o ZS, ale zostają one poddane poważnym modyfikacjom mechanicznym i funkcjonalnym w celu osiągniecia efektywnej pracy i ograniczenia emisji szkodliwych składników spalin [53,82]. Niemniej w celu intensyfikacji dekarbonizacji sektora transportu morskiego prowadzone są dalsze prace nad zwiększeniem sprawności silników z rozwiązaniem dwupaliwowym przy jednoczesnym minimalizowaniu emisji szkodliwych składników spalin.

2.4.3. Niskotemperaturowe spalanie dwupaliwowe kontrolowane reaktywnością paliw

Spalanie niskotemperaturowe (ang. *Low-Temperature Combustion*, LTC) to technologia o znaczącym potencjale, przyczyniająca się do rozwoju silników spalinowych i charakteryzująca się wyższą całkowitą sprawnością. Kluczowymi warunkami koniecznymi do realizacji procesu LTC są homogeniczna mieszanka paliwowo-powietrzna oraz wysoki współczynnik λ w przedziale ok. 1,5–2,0 w komorze spalania. Takie uwarunkowania umożliwiają znaczące obniżenie emisji NO_X i PM poprzez utrzymanie niskiej lokalnej temperatury spalania i wysokiego namiaru tlenu [89,90]. Opisane warunki nie mają miejsca w tradycyjnych silnikach o ZI i ZS czy CDF. Silniki o ZI korzystają z jednorodnych mieszanek paliwowo-powietrznych, lecz działanie w warunkach znaczącego nadmiaru powietrza jest ograniczone przez mechanizmy deflagracyjne rozprzestrzeniania się płomienia. Z kolei silniki o ZS charakteryzują się możliwością osiągnięcia wysokiej wartości nadmiaru powietrza, jednak mieszanka paliwowo-powietrzna jest lokalnie bogata w obszarach inicjacji spalania [91], o czym pisano w podrozdziale 2.4.2.

Mimo obiecujących perspektyw wdrożenie technologii LTC napotyka na liczne problemy technologiczne, które wymagają dalszych badań i rozwoju [85,86]. Aby możliwe było spalanie w niskiej temperaturze w silniku tłokowym, konieczne jest spełnienie wielu warunków umożliwiających bezpłomieniowe spalanie mieszanki. Ten rodzaj spalania można osiągnąć poprzez objętościowy zapłon samoczynny mieszanki. Zapłon samoczynny z ładunkiem jednorodnym (ang. *Homogeneous Charge Compression Ignition,* HCCI) jest podstawowym systemem realizującym LTC, w największym stopniu spełniającym założenia tego sposobu spalania. W tej koncepcji całkowicie odparowana, jednorodna mieszanka paliwowo-powietrzna jest sprężana do punktu samozapłonu. W przypadku HCCI (rys. 12d) zapłon następuje w wielu miejscach wewnątrz cylindra bez wyraźnego rozprzestrzeniania się płomienia, co zapewnia wysoką efektywność spalania i do 99% redukcji NO_x i PM w porównaniu z typowym silnikiem o ZS [90]. Niska temperatura spalania powoduje jednak wzrost emisji UHC i tlenku węgla (CO). W związku z mniejszą konwekcją ciepła przez ściany cylindra zwiększa się sprawność silnika. W HCCI występują problemy z częściowym brakiem samozapłonów przy niskich obciążeniach oraz zbyt dużym przyrostem ciśnienia (ang. *Pressure Rise Rate*, PRR) przy dużych obciążeniach [91]. Chociaż problemy te są w pewnym stopniu wspólne dla wszystkich koncepcji LTC, w HCCI intensyfikują się ze względu na bardziej stochastyczny charakter spalania. Powoduje to również problemy związane z dużą zmiennością pomiędzy cyklami, dodatkowo wzmacniane podczas stanów przejściowych pracy silnika [91].

Inną koncepcją obecną w nowoczesnych silnikach o ZS i stanowiącą krok w stronę realizacji LTC są systemy, w których przygotowuje się część ładunku spalin i reformowanego w nich paliwa przed kolejnym cyklem pracy w cylindrze (ang. *Premixed Compression Ignition*, PCI) [91,92]. Mechanizm ten realizowany jest poprzez wydłużony suw ssania, kilkuetapowy wtrysk ON, wysokie wskaźniki otwarcia zaworu EGR oraz zarządzanie temperaturą spalin w ich obiegu wtórnym, co przyczynia się do redukcji NO_x i PM. Na rys. 12 zestawiono obrazy z komór spalania dla ZI, ZS, PCI i HCCI [93].

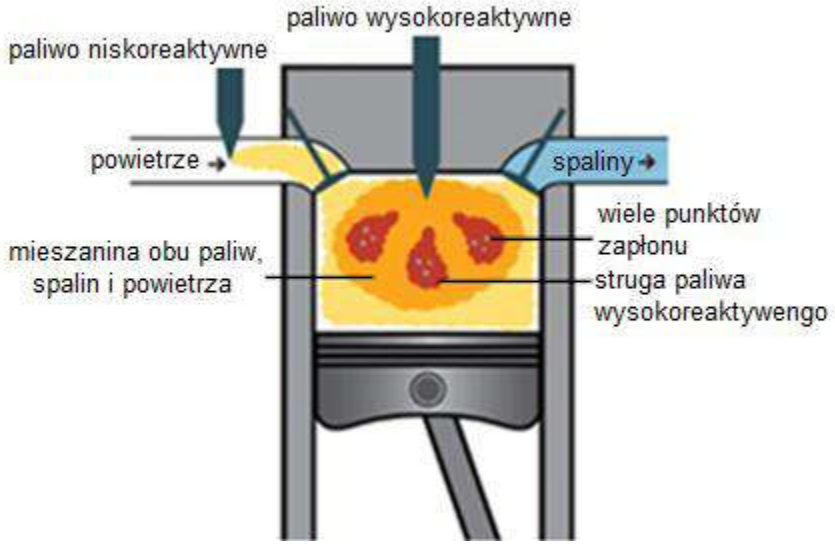


Rys. 12. Wizualizacja procesów spalania w systemach *ZI, ZS, PCI, HCCI* [91]

PCI tylko częściowo wykorzystuje zalety LTC, a problemy z utrzymaniem i kontrolą samozapłonu niskotemperaturowego powodują konieczność ograniczenia zakresu pracy silnika do obszarów, w których zachodzi proces spalania w sposób umożliwiający prawidłowy przebieg samozapłonu. Problem ten nie występuje jednak w przypadku modyfikacji kinematyki chemicznej w komorze spalania paliwami o różnej reaktywności.

Spalanie dwupaliwowe kontrolowane reaktywnością paliw (ang. *Reactivity Controlled Compression Ignition*, RCCI) to idea realizacji LTC o znacząco poszerzonym zakresie prawidłowego działania silnika. W koncepcji tej realizowane jest rozwarstwienie reaktywności ładunku wzdłuż osi symetrii cylindra [94]. Współczynnik uwalniania ciepła z paliwa (ang. *Heat Release Rate*, HRR) w RCCI jest powiązany z kinetyką chemiczną oraz kontrolowany głównie poprzez uwarstwienie reaktywności, które jest uzyskiwane przez strategię wtrysku paliwa nisko- i wysokoreaktywnego oraz pracę systemu EGR (realizującego obieg wtórny spalin bądź tzw. ujemne nałożenie otwarcia zaworów silnika) [59,95]. Systemy dwupaliwowe CDF od RCCI odróżnia niewystarczające rozwarstwienie reaktywności w cylindrze oraz niepełna homogenizacja paliwa z powietrzem, co skutkuje wyższymi lokalnymi temperaturami w porównaniu z temperaturami, jakie osiąga LTC [53,94]. RCCI definiuje się jako strategię spalania w silniku dwupaliwowym, w której mieszanka w cylindrze tworzy się z co najmniej dwóch paliw o różnej reaktywności, podczas gdy paliwo o niskiej reaktywności jest jednorodnie mieszane z powietrzem. Dostosowanie proporcji paliw i strategii wtrysku paliwa,

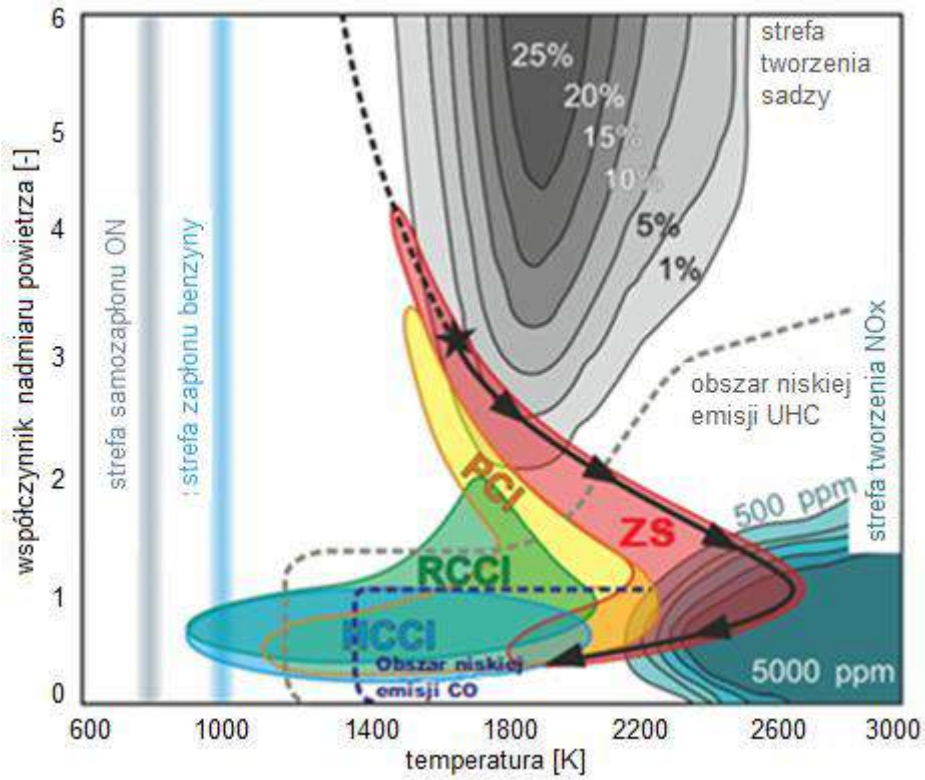
np. poprzez wtryski wielokrotne i sterowanie czasami wtrysku, tworzy kontrolowane gradienty reaktywności w komorze spalania, co umożliwia kontrolowanie i optymalizację faz spalania oraz czasu ich trwania [90,96]. Przygotowanie paliwa w klasycznym trybie RCCI obejmuje w pierwszej kolejności wprowadzenie paliwa o niskiej reaktywności do cylindra w celu wytworzenia dobrze wymieszanego ładunku paliwa o niskiej reaktywności, powietrza i recyrkulowanych gazów spalinowych. Paliwo o wysokiej reaktywności jest następnie wtryskiwane przed zapłonem wstępnie zmieszanego paliwa poprzez pojedyncze lub wielokrotne wtryski bezpośrednio do komory spalania. Paliwo o wysokiej reaktywności jest zwykle wtryskiwane na początku suwu sprężania silnika oraz może – ale nie musi – być wtryskiwane w okolicach GMP w celu zwiększenia ciśnienia w cylindrze, co ma doprowadzić do samozapłonu paliwa wysokoreaktywnego i docelowo zwiększyć ciśnienie w cylindrze w celu zainicjowania samozapłonu przygotowanej wcześniej mieszanki znajdującej się już w cylindrze [90,94]. Proces spalania w RCCI definiuje się jako „kinetyczny" – bezpłomieniowy [91]. Schemat działania silnika RCCI zobrazowano na rys. 13.



Rys. 13. Schemat działania silnika RCCI [97]

Różnice pomiędzy opisanymi powyżej sposobami LTC oraz konwencjonalnymi ZS polegają na ich charakterystycznych „drogach" spalania, co zaprezentowano na rys. 14. Warto zwrócić uwagę również na problematykę spalania niecałkowitego. W celu ograniczenia szczytowych wartości ciśnienia spalania oraz charakterystyki jego narastania wydłuża się czas mieszania gazów w cylindrze i opóźnia spalanie. Powoduje to przerwanie przebiegu spalania ze względu na spadek temperatury wynikający z postępującego rozszerzania się w przestrzeni

cylindra w suwie pracy. Szczegółową analizę różnych metod spalania LTC przedstawiono w pracach [91,94,98].



Rys. 14. Zależność współczynnika nadmiaru powietrza i temperatury spalania dla systemów HCCI, RCCI, PCI i ZS z zaznaczeniem obszarów tworzenia substancji toksycznych [91]

Prace badawczo-rozwojowe nad RCCI stanowią istotny obiekt zainteresowania dla środowiska naukowego zajmującego się silnikami spalinowymi [95,96,99,100,101,102]. W pracy [90] wykazano eksperymentalnie 56% całkowitej sprawności cieplnej silnika pracującego pojazdu ciężarowego w trybie RCCI w stanie ustalonym oraz dokonano przeglądu postępów w badaniach w tej materii, by ostatecznie wyznaczyć metodykę postępowania dotyczącą modyfikacji mechanicznych i sterowania w celu osiągnięcia średniej 50% sprawności ogólnej silnika. Z kolei w badaniu [94] przedstawiono argumenty dotyczące opłacalności komercyjnej silników RCCI, a także omówiono ich emisję CO_2-*eq* WTW (ang. *Well-to-Wheel* – sumaryczny wpływ środowiskowy od źródła pozyskania energii do energii napędzającej koła pojazdu).

Prace [95,96,99,100,101,102] również demonstrują stanowiska badawcze dotyczące RCCI. Duża część stanowisk badawczych opiera się na zmodyfikowanych, opracowanych komercyjnie konstrukcjach silników o ZS. Wynika to z aspektów ekonomicznych oraz faktu, że istniejące silniki o ZS doskonale nadają się do retrofittingu w RCCI. Niemniej w celu osiągnięcia efektywnego systemu RCCI konieczna jest głęboka modyfikacja aspektów

mechanicznych silnika i zastosowanie zaawansowanego systemu sterowania. Wśród najważniejszych modyfikacji należy wymienić:

- system niezależnego, zmiennego sterowania zaworami silnika (ang. *Variable Valve Actuation*, VVA),
- zmieniony bądź modyfikowany stopień sprężania z wykorzystaniem VVA,
- zmienioną geometrię denka tłoka,
- modyfikację kąta rozpylania wtryskiwacza paliwa wysokoreaktywnego,
- system realizujący zasilanie paliwem niskoreaktywnym,
- precyzyjne i szybkie systemy mierzące w czasie rzeczywistym ciśnienie w cylindrze i emisję spalin,
- całkowicie na nowo zaprojektowany system sterowania.

Część zagadnień dotyczących modyfikacji mechanicznych umożliwiających realizację RCCI została opisana w publikacjach [A6, A7]. Należy podkreślić, że wszystkie wymienione powyżej modyfikacje nie są niezbędne w realizacji RCCI. Rozszerzają jednak możliwości tego systemu w sposób poszerzający nie tylko zakresy operacyjne pracy silnika, ale również w kontekście wielopaliwowości czy możliwości wykorzystania zmiennych trybów pracy takich jak zmiana trybu pracy z RCCI na CDF i klasyczny ZS [94].

Do chwili obecnej technologia RCCI nie została skutecznie skomercjalizowana i spopularyzowana. Rozpoczęcie badań aplikacyjnych nad potencjalnym wykorzystaniem tej innowacyjnej technologii w sektorze pojazdów ciężarowych miało miejsce w 2012 roku. Prace te, zainicjowane przez TNO Automotive we współpracy z OEM Group Japan, zostały udokumentowane w literaturze [103]. Efektem tych działań była demonstracja zastosowania dwupaliwowego systemu zasilanego CNG/ON, stanowiąca pełnoskalowy eksperyment, który – zgodnie z raportami – osiągnął wymagane limity emisji określone w normach EURO VI dla NO_x oraz PM. Równolegle prace prowadzone przez CMT Motores Térmicos we współpracy z Volvo Trucks zaowocowały znaczącym postępem w rozwoju układu napędowego zgodnego z normami EURO VI, poprzez częściową integrację technologii RCCI w silnikach o spalaniu mieszanym z CDF [104,105]. Na płaszczyźnie komercyjnej działalność na rzecz promowania RCCI prowadzi firma Arena Red, oferująca retrofittingowe zestawy do jednostek pływających, a od niedawna także zestawy przeznaczone dla pojazdów dostawczych. Rozwój technologii sterowania na podstawie ciśnienia spalania w cylindrze (ang. *Combustion Pressure Based Control*, CPBC), realizowany przez tę firmę, doprowadził do możliwości częściowego skomercjalizowania RCCI, chociaż problemem pozostają wysokie koszty modyfikacji [97].

Efekty modyfikacji przeprowadzonych przez Arena Red zostały zaprezentowane w pracach [106,107,108]. Silniki zainstalowane na jednostkach pływających oraz pojazdach ciężarowych, które zostały opisane w tych publikacjach, mają współczynnik zastąpienia paliw do 99%. Prace nad komercjalizacją technologii RCCI bazujące na aktualnych silnikach CDF i klasycznych silnikach o ZS prowadzone są również przez firmy specjalizujące się w silnikach do zastosowań dalekomorskich oraz elektroenergetycznych, m.in. przez firmę Wärtsilä [109], co podkreśla rosnące zainteresowanie i potencjalne możliwości wykorzystania tej technologii na szerszą skalę.

2.5. Podsumowanie analizy stanu zagadnienia

Obecna struktura globalnego miksu energetycznego zdominowana jest przez paliwa kopalne, co generuje negatywny wpływ na środowisko z powodu emisji gazów cieplarnianych i innych zanieczyszczeń. W związku z rosnącym zapotrzebowaniem na bardziej zrównoważone i przyjazne dla środowiska rozwiązania obserwuje się zmianę w kierunku wykorzystania alternatywnych źródeł energii w sektorze transportowym. Elektryfikacja i hybrydyzacja pojazdów są uznawane za kluczowe strategie w procesie dekarbonizacji transportu, jednakże napotykają one szereg ograniczeń, takich jak niska gęstość energii, ograniczona efektywność logistyczna czy braki infrastrukturalne. W konsekwencji technologie te znajdą główne zastosowanie w transporcie lekkim i wśród samochodów osobowych. Technologie wodorowe oraz stosowanie paliw niekopalnych, takich jak paliwa syntetyczne i biopaliwa w silnikach spalinowych o wyższej sprawności ogólnej, wskazywane są jako potencjalne rozwiązania dla napędów energochłonnych w przyszłości.

Proces retrofittingu, mający na celu dostosowanie pojazdów do nowych standardów środowiskowych przez redukcję emisji i przedłużenie ich żywotności, wpisuje się w obecne trendy ekologiczne. Polska branża drogowego transportu ciężkiego oraz producenci niefabrycznych samochodowych instalacji gazowych mogą być beneficjentami rozwoju zasilania dwupaliwowego dla silników o ZS, a większość pojazdów ciężarowych jest technologicznie predysponowana do wykonania retrofittingu. Istnieje więc potencjał modyfikacji używanych silników o ZS do spalania dwupaliwowego CDF i RCCI, oferujących redukcję emisji szkodliwych substancji i zwiększenie efektywności energetycznej. W sektorze niefabrycznych instalacji dwupaliwowych istnieją obecnie rozwiązania, które – mając na celu niskie koszty aplikacyjne – ograniczają maksymalny *Wz* do ok. 50%. Technologia RCCI, choć obiecująca w związku wysoką sprawnością silnika do ok. 50% i niemal całkowitym

z wyeliminowaniem NO_x i PM w składzie spalin, wymaga zaawansowanych modyfikacji mechanicznych i innowacyjnego sterowania, co stanowi wyzwanie dla jej komercjalizacji. Prace nad tą technologią prowadzone są intensywnie zarówno w sektorze badawczym, jaki przez przemysł, co świadczy o potencjale i zainteresowaniu tą metodą realizacji spalania. Demonstracje zastosowania RCCI w różnych sektorach, w tym morskim oraz w silnikach używanych po procesie retrofittingu podkreślają zasadność kontynuowania prac nad zagadnieniem.

Podsumowując: rozwój i implementacja systemów niefabrycznych realizacji spalania dwupaliwowego stanowi atrakcyjną alternatywę mającą na celu wpisanie się w nowoczesne strategie dekarbonizacyjne. Niezbędne jest podjęcie działań dotyczących:

- wytypowania paliw przyszłości do zastosowań dwupaliwowych,
- zbadania empirycznego wpływu wyższych współczynników zastąpienia dla instalacji CDF na aspekty eksploatacyjne pojazdu,
- kontynuowania prac badawczych i rozwojowych dotyczących RCCI w celu uatrakcyjnienia komercyjnego tej technologii.

3. Cele i zakres rozprawy

Analiza stanu zagadnienia będąca przedmiotem dostępnych badań i publikacji umożliwiła sprecyzowanie tematyki pracy doktorskiej. Implementacja systemów niefabrycznych spalania dwupaliwowego stanowi alternatywną, atrakcyjną ścieżkę strategii dekarbonizacyjnych transportu. Zidentyfikowano potencjał aplikacyjny technologii uzasadniający kontynuowanie prac w tym zakresie. Planując realizację pracy, dokonano wyboru działań, które ukierunkowane były na cele utylitarne i nadrzędny cel naukowy.

Cele utylitarne dotyczą zbadania możliwości aplikacyjnych niefabrycznych systemów dwupaliwowych, w których skład wchodziły:

- badania atrakcyjności rynkowej i gotowości do retrofittingu,
- przeglądy rozwiązań technologicznych i dotyczących niezbędnych modyfikacji mechanicznych dla silników dwupaliwowych,
- wytypowanie paliw niskoemisyjnych do nowoczesnych rozwiązań dwupaliwowych,
- opracowanie narzędzi i wskazanie kluczowych obszarów badań i rozwoju nad retrofittingiem w RCCI.

Cel naukowy:

- eksperymentalna ocena skutków zwiększenia współczynnika zastąpienia w układzie dwupaliwowym CDF - opracowanie koncepcji i badania weryfikacyjne instalacji CDF CNG/ON o średnim współczynniku zastąpienia przekraczającym 50% w pełnym zakresie obciążeń silnika realizowanych na hamowni silnikowej, podwoziowej oraz w warunkach drogowych.

Streszczenie rozprawy zostało podzielone na siedem rozdziałów. W rozdziale pierwszym dokonano wprowadzenia do zagadnienia oraz omówiono motywacje autora do podjęcia prac nad wybraną tematyką. Rozdział drugi zawiera szeroko zakrojoną analizę stanu zagadnienia ze szczególnym naciskiem na kwestie: imperatywów ekologicznych i ich ograniczeń, idei retrofittingu, analiz rynkowych i bezpieczeństwa, omówiono również systemy spalania dwupaliwowego dla silników o ZS – CDF i RCCI. W pracy wyodrębniono niniejszy rozdział trzeci celem przedstawienia głównych celów oraz układu pracy doktorskiej. W rozdziale czwartym dokonano analizy wykorzystania paliw niskoemisyjnych w nowoczesnym zasilaniu dwupaliwowym o ZS, ze szczegółowym omówieniem wytypowanych paliw oraz zasadności

wykorzystania ich w systemach dwupaliwowych. W rozdziale piątym, będącym efektem badań własnych autora i jednocześnie stanowiącym najważniejszy rozdział dysertacji, zaprezentowano wyniki prac dotyczących opracowania i badań instalacji CDF CNG/ON o średnim *Wz* przekraczającym 50% w pełnym zakresie obciążeń silnika realizowanych na hamowni silnikowej, podwoziowej oraz warunkach drogowych. W rozdziale piątym wyodrębniono trzy etapy pracy zgodnie z miejscem ich realizacji tj.: badania na hamowni silnikowej, badania na hamowni podwoziowej oraz badania drogowe. Chronologicznie dla etapu pierwszego: omówiono szczegóły budowy stanowiska pomiarowego oraz metodyki pomiarów. Zaprezentowano pierwotną i rozwojową koncepcję ww. instalacji. Wykonano badania podstawowe. Omówiono system minimalizujący efekty spalania detonacyjnego w zakresie dużych obciążeń. Poddano analizie wyniki badań gazów odlotowych/spalin. W etapie drugim: omówiono koncepcję i realizacje mobilnej stacji diagnostyczno-montażowej z hamownią podwoziową do badań pojazdów ciężarowych. Opisano wyniki badań nad kluczowymi elementami prac w warunkach hamowni podwoziowej gwarantujące możliwie bezinwazyjną i poprawną realizację przeprowadzonych badań. Ostatecznie opisano efekt optymalizacji parametrów eksploatacyjnych po modyfikacji technicznej pojazdu w zasilanie dwupaliwowe. W etapie trzecim: przedstawiono metodykę realizacji badań drogowych pojazdu oraz dokonano analizy wyników. Rozdział piąty kończy się syntetycznymi wnioskami z badań. W rozdziale szóstym uzyskane rezultaty badań i analiz zestawiono z działaniami dotyczącymi przyszłości retrofittingu silników w RCCI oraz omówiono inne miejsca transferu technologii dwupaliwowej. Wnioski końcowe zaprezentowano w rozdziale siódmym.

4. Analiza możliwości wykorzystania paliw niskoemisyjnych w zasilaniu dwupaliwowym o ZS

Paliwa silnikowe w szerszym wymiarze to odrębne i bardzo rozległe zagadnienie. W rozdziale tym przekrojowo opisano prace autora dotyczące wytypowania paliw niskoemisyjnych do stosowania w silnikach dwupaliwowych pojazdów samochodowych. Zakres opracowania dotyczy w szczególności wpływu paliwa na proces spalania, teoretyczną emisję toksycznych składników spalin oraz możliwości ich wykorzystania w perspektywie transformacji energetycznej. Wytypowane przez autora paliwa poddano szerszej analizie oraz częściowym badaniom laboratoryjnym. Rozdział zakończony jest krótką analizą zasadności użycia wytypowanych paliw w nowoczesnych układach dwupaliwowych w porównaniu z innymi, klasycznymi systemami transformacji energii chemicznej w nich zawartych.

Zgodnie z informacjami w rozdziale drugim różne paliwa do układów dwupaliwowych dzięki swoim właściwościom fizykochemicznym mogą okazać się korzystne z punktu widzenia sterowania procesem spalania w silniku dwupaliwowym oraz emisji składników spalin. Z kolei prowadzenie prac analitycznych mających na celu dobranie paliw do układów dwupaliwowych uzasadnia konieczność poszukiwania nowych, niskoemisyjnych źródeł energii dla silników spalinowych. Stawiane przez prawodawstwo bardzo restrykcyjne wymagania środowiskowe mogą doprowadzić do szybkiego wprowadzenia i spopularyzowania nowych typów paliw na rynku.

W procesie spalania stosowanym w silnikach tłokowych kluczowe znaczenie przypisuje się interakcjom trzech pierwiastków chemicznych: wodoru, węgla oraz tlenu. Reakcje chemiczne są odpowiedzialne za transformację energii chemicznej zawartej w paliwach w energię cieplną, która następnie jest przekształcana w wyniku przemian gazowych w silnikach na energię mechaniczną. Głównymi produktami procesu spalania są dwutlenek węgla i woda, co towarzyszy wyzwoleniu energii. W celu minimalizacji emisji CO_2 preferuje się wykorzystywanie paliw o niskim stosunku zawartości węgla do wodoru. Z uwagi na rosnące regulacje ograniczające emisję gazów cieplarnianych wodór przedstawia się jako najbardziej atrakcyjny pod względem ekologii składnik paliwowy. Jednakże jego aplikacja jako źródło energii napotyka na specyficzne wyzwania, omówione częściowo w podrozdziale 2.1. Pomimo tych trudności wzbogacenie paliw węglowodorowych w wodór przyczynia się do zauważalnej redukcji emisji dwutlenku węgla, co jest istotne dla ochrony środowiska. W artykule [A2]

dokonano przeglądu paliw o niskiej emisji CO_2 dla zastosowań dwupaliwowych. W tabeli 1 przedstawiono zbiorcze podsumowanie omawianej pracy.

Tabela 1. Zbiorcze porównanie najważniejszych parametrów wybranych paliw do silników dwupaliwowych*, Opracowanie na podstawie [A2]

Paliwo: wysokoreaktywne niskoreaktywne	Proporcja C/H		Masowa zawartość wodoru [%]	Masowa zawartość węgla [%]	Przybliżony poziom emisji CO_2 [g/MJ]	Emisje główne:
	% masy	molowa				
Olej Napędowy	7,26	0,59	12,94	86,93	69–75	HC, CO, CO_2, NO_x
Biodiesel B20	6,63	0,56	12,79	84,87	ok. 73	HC, CO, CO_2, NO_x
FAME B100	6,32	0,53	12,17	76,96	ok. 71	HC, CO, NO_x
HVO	5,49	0,46	15,40	84,60	ok. 71	HC, CO, NO_x
Benzyna silnikowa E5	9,00	0,73	10,00	90,01	69–70	HC, CO, CO_2, NO_x
Etanol E100	4,00	0,33	13,00	52,00	62-64	HC, NO_x
Metanol M100	3,00	0,25	12,50	37,49	ok. 70	HC, NO_x
LPG	4,65	0,39	17,69	82,30	63-61	HC, NO_x
CNG	3,00	0,25	25,00	75,00	56–55,35	CH_4, NO_x
HCNG20**	2,77	0,23	25,75	74,25	55–54,3	CH_4, NO_x
HCNG40**	2,52	0,20	26,50	73,50	52,8-53,6	CH_4, NO_x
HCNG50**	2,15	0,18	27,50	72,50	50,4-51,1	NO_x

* Ze względu na różny skład chemiczny CNG w poszczególnych krajach i z poszczególnych źródeł w tabeli paliwa na bazie gazu ziemnego są czyste – 100% metan.
** HCNG to mieszanina H_2 z CNG, liczba po HCNG mówi o procentowej objętościowej zwartości H_2 w CNG, składy CNG i H_2 są w proporcjach objętościowych zgodnie z [110], a elementy zależne od innych, np. stosunek C/H czy zawartość właściwa wodoru w HCNG20, zostały obliczone przez autora.

W artykule [A2] przedstawiono problem z parametryzacją właściwości fizykochemicznych paliw gazowych. Rozbieżności między różnymi paliwami są duże, a problem ten w szczególności dotyczy CNG oraz jego mieszanin z wodorem (ang. *Hydrogen to Compressed Natural Gas*, HCNG). W tabeli 1 można również zaobserwować powody typowego wykorzystania CNG jako paliwa niskoreaktywnego, zastępczego dla ON w typowych popularnych układach dwupaliwowych. Wynika to głównie z właściwości tego paliwa, ograniczających występowanie spalania detonacyjnego oraz z mniejszej proporcji C/H. Można też zauważyć, że paliwa z domieszkami H_2 dodatkowo poprawiają tę proporcję.

W HCNG emisje CO_2 zmniejszają się jeszcze bardziej wraz ze wzrostem zawartości wodoru w paliwie.

Zdaniem autora wysokoreaktywnym substytutem ON w zasilaniu dwupaliwowym, mogącym odgrywać rolę paliwa przyszłości, jest HVO. Przekonanie to powstało w wyniku dogłębnej analizy dostępnych paliw ciekłych, mogących odgrywać rolę paliwa wysokoreaktywnego w silniku dwupaliwowym w ramach prac nad artykułem [A2]. Ogólny obraz dostępnej wiedzy wskazuje na istotność dalszych prac z dwupaliwowym zasilaniem HCNG i HVO. Dostępne informacje na temat emisji wynikających ze spalania tych paliw w systemach wykorzystujących pojedyncze paliwo wskazują na znaczące korzyści związane z obniżeniem emisji gazów GHG. Zastosowanie tych paliw przynosi dodatkowe pozytywne efekty, takie jak zmniejszenie emisji WTW oraz potencjał do częściowego rozwiązania problemów związanych z infrastrukturą, opisanych w podrozdziale 2.1. Produkcja HVO i HCNG z wykorzystaniem zielonego wodoru, pochodzącego z nadwyżek produkcji energii elektrycznej generowanej przez OŹE, jak wiatr czy słońce, adresuje jeden z kluczowych problemów związanych z energią odnawialną: jej zmienność i trudności z akumulacją energii. Nadwyżki z produkcji energii, generowane w okresach intensywnego nasłonecznienia czy silnego wiatru, często przekraczają bieżące zapotrzebowanie, co jest dużym problemem całego sektora energetyki [111]. Wykorzystanie tych nadwyżek do produkcji zielonego wodoru, który następnie może być używany do produkcji biopaliw takich jak HVO i HCNG, stanowi rozwiązanie umożliwiające magazynowanie energii w postaci chemicznej, zwiększając efektywność energetyczną całego systemu i jednocześnie częściowo rozwiązując problematyczne przechowywanie wodoru opisane w podrozdział 2.1.

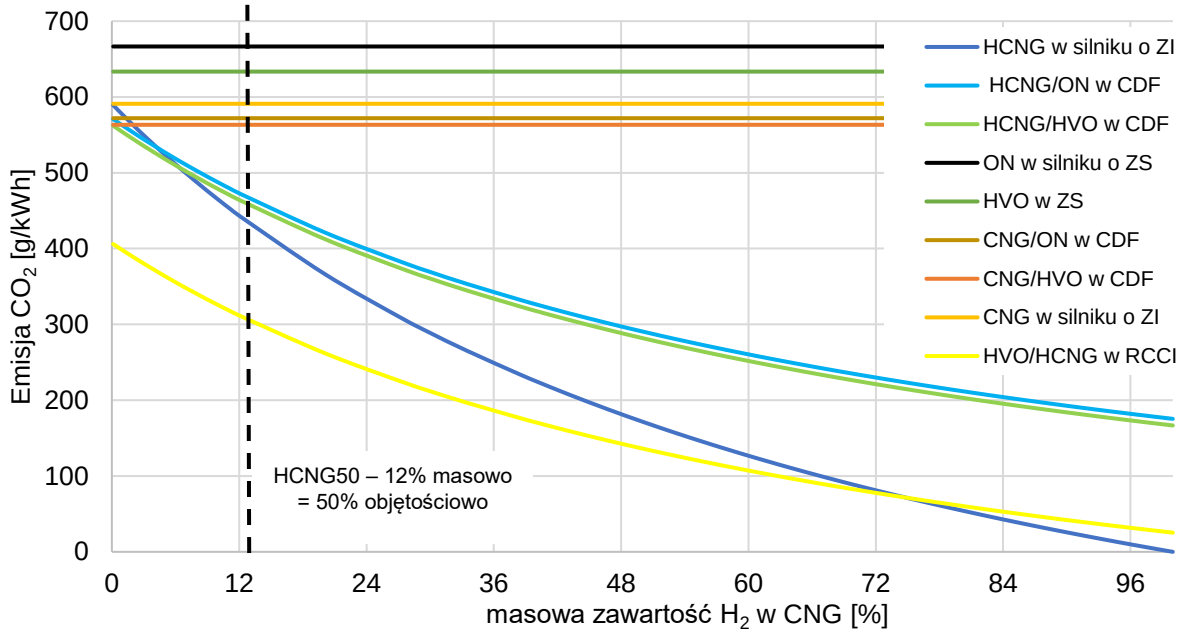
Dodatkowo HCNG w proporcji objętościowej 50% CNG, 50% H_2 może być dystrybuowane bezinwazyjnie w istniejącym systemie dystrybucji gazu ziemnego (bez ryzyka występowania korozji wodorowej i rozszczelnienia) [112,113], co ponownie wskazuje wysokie prawdopodobieństwo spopularyzowania się tego paliwa. HVO zyskuje aktualnie popularność rynkową, będąc dostępnym w sprzedaży detalicznej w krajach Beneluxu, Czechach i Skandynawii. W procesie produkcji obu paliw może być użyta biomasa odpadowa np. z procesów produkcji żywności. W produkcji HCNG można wykorzystać biometan pochodzący z procesów np. fermentacji anaerobowej czy hydrotermalnej gazyfikacji biomasy [114]. Zwiększa to ogólną efektywność wykorzystania zasobów, ale również przyczynia się do redukcji odpadów. Dla najbardziej ekologicznych konfiguracji HCNG całościowe poziomy emisji WTW są ujemne [70,114]. W wariantach ekologicznych wskaźnik WTW dla HVO jest

ok. 90% mniejszy niż dla ON. Więcej na temat HVO jako paliwa wysokoreaktywnego dla silników przedstawiono w publikacji [A1], gdzie między innym paliwo to poddano badaniom fizykochemicznym i reologicznym, wykazując wieloaspektową przewagę nad paliwem Biodiesel B100 FAME (ang. *Fatty Acid Methyl Esters* – estry metylowe kwasów tłuszczowych z olejów roślinnych).

Zasadność wykorzystania HVO i HCNG w układach dwupaliwowych zamiast bezpośrednio w silnikach o ZS i ZI uzasadniono w pracy [A11]. W ramach realizacji tej publikacji przygotowano model numeryczny wykorzystujący maksymalne sprawności ogólne silników η, które wytypowane zostały w ramach prac przeglądowych autora dla różnych silników i systemów spalania. Wykorzystując ww. model wraz z danymi z tabeli 2, stworzono zależność jak na rys. 15:

Tabela 2. Zestawienie sprawności ogólnej i współczynnika zastąpienia paliw dla wytypowanych systemów spalania w pojazdach samochodowych wykorzystanych dla obliczeń na rys. 15 [A11]

Rodzaj spalania	η [%]	*Wz* średni [%]
ZI	34	-
ZS	42	-
CDF	38	75
RCCI	50	95



Rys. 15. Teoretyczna emisja CO_2 w g/kWh w relacji do zawartości masy H_2 w CNG dla wytypowanych konfiguracji spalania i użytych w nim paliw

Zgodnie z zależnościami przedstawionymi na rys. 15 zwiększanie współczynnika zastąpienia, poprawianie sprawności ogólnej silnika oraz zwiększanie zawartości wodoru zmniejsza emisję teoretyczną CO_2, co wskazuje na zasadność dalszych prac rozwojowych nad systemami zasilania dwupaliwowego. Widać to w szczególności na zaprezentowanym przykładzie silnika RCCI. Spalanie HCNG z masową zawartością H_2 do 12% (odpowiadającym mniej więcej paliwu HCNG50) w silnikach o ZI i CDF jest porównywalne. Szczegółowe i bardziej obszerne analizy dla innych konfiguracji zawarto w artykule [A11].

5. Badania własne

W niniejszym rozdziale, stanowiącym wynik oryginalnego wkładu autora w zakresie badań nad systemem CDF CNG/ON o średnim współczynniku zamiany paliwa przekraczającym 50% dla pełnego spektrum obciążenia silnika przeznaczonego do pojazdów ciężarowych, zidentyfikowano trzy główne fazy badawcze. Uczyniono to na podstawie miejsca ich realizacji, wyróżniając: badania na hamowni silnikowej, badania na hamowni podwoziowej oraz badania drogowe.

W pierwszym etapie skupiono się na szczegółowym opisie badanego silnika, przedstawiając równocześnie detale konstrukcyjne stanowiska pomiarowego, metodologię badań oraz analizę niepewności pomiarowych. Zaprezentowano wstępną oraz rozwiniętą koncepcję opracowywanej i badanej instalacji CDF, gdzie omówiono również celowość oraz efekty działań badawczych dotyczących określenia charakterystyk pracy wtryskiwaczy ON. Kolejnym krokiem było opracowanie projektu, a następnie implementacja systemu minimalizującego efekty spalania detonacyjnego przy wysokim stopniu zastąpienia paliwa bazowego i w szerokim zakresie obciążeń silnika. Następnie przeprowadzono badania mające na celu określenie stanu technicznego (poziomu zużycia) silnika oraz wykonano wstępną optymalizację parametrów eksploatacyjnych silnika po jego modyfikacji technicznej na zasilanie dwupaliwowe – tzw. „strojenie”. Pierwszy etap zakończono analizą wyników odnoszących się do emisji spalin.

W drugim etapie skoncentrowano się na opracowaniu i realizacji koncepcji mobilnej stacji diagnostyczno-montażowej wyposażonej w hamownię podwoziową, przeznaczonej do badań i implementacji dwupaliwowych układów zasilania w używanych pojazdach ciężarowych. Przedstawiono wyniki badań skupiających się na kluczowych aspektach prac realizowanych na hamowni podwoziowej w celu zapewnienia możliwie najmniejszej inwazyjności oraz poprawności i jakości przeprowadzanych badań. Zakończeniem etapu było omówienie rezultatów optymalizacji parametrów eksploatacyjnych pojazdu po jego modyfikacji do zasilania dwupaliwowego.

W trzecim etapie przedstawiono metodologię przeprowadzania badań drogowych oraz dokonano analizy uzyskanych wyników, z uwzględnieniem specyfiki eksploatacyjnej pojazdu w realnych warunkach jego użytkowania.

Rozdział kończy się syntetycznymi wnioskami z badań.

Badania i rozwój instalacji CDF o wysokim współczynniku zastąpienia realizowano w ramach projektu NCBiR: Mazowsze/0123/19-00 „Innowacyjna ekologiczna instalacja CNG do silników wysokoprężnych ograniczająca emisję szkodliwych składników spalin wraz z mobilnym stanowiskiem diagnostyczno-montażowym". Realizatorem projektu było konsorcjum składające się z Wydziału Inżynierii Mechanicznej Wojskowej Akademii Technicznej oraz lidera firmy Gaslux sp. z o.o. Autor pracy był członkiem zespołu realizującego ten projekt.

5.1. Badania na hamowni silnikowej

5.1.1. Opis badanego silnika i stanowiska pomiarowego

Jako obiekt badań wybrano używany silnik Volvo serii D13 [115] z podstawowym układem zasilania paliwem wykorzystującym wtryskiwacze PD. Uzasadnieniem podjętej decyzji były duże nasycenie rynku transportu drogowego pojazdami wykorzystującymi tę konstrukcję oraz potencjalna możliwość transferu technologii do sektora silników zaburtowych, tj. serii Volvo Penta, wykorzystujących w większości wtryskiwacze PD [116]. Badania i obliczenia wykonano dla silnika D13C460 stosowanego w ciągnikach siodłowych Volvo serii FH trzeciej generacji w latach 2009–2014 [117]. Przebieg pojazdu ciężarowego, z którego silnik będący obiektem badań został wymontowany i zainstalowany na hamowni silnikowej, to 780 600 km. Całkowity czas pracy silnika wyniósł 11 800 godzin, a od początku eksploatacji zużył on 229 100 dm^3 ON. W tabeli 3 zestawiono wybrane dane techniczne silnika D13C460.

Tabela 3. Dane techniczne silnika D13C460 [115]

Liczba cylindrów	6
Pojemność skokowa	12,8 dm^3
Skok i średnica tłoka	158 mm x 131 mm
Stopień sprężania	17,8:1
Układ doładowania	turbosprężarkowy z wewnętrznym systemem upustu ciśnienia spalin (ang. *Wastegate*)
Moc maksymalna	338 kW (460 KM)
Maksymalny moment obrotowy	2300 Nm
Zakres obrotów optymalnych	1150–1300 obr./min
Zakres obrotów ekonomicznych	1000–1500 obr./min
Pełny zakres obrotowy	600–2100 obr./min
Spełniana norma emisji spalin	Euro V EEV (ang. *Enhanced Environmentally friendly Vehicle*) wyposażony w SCR

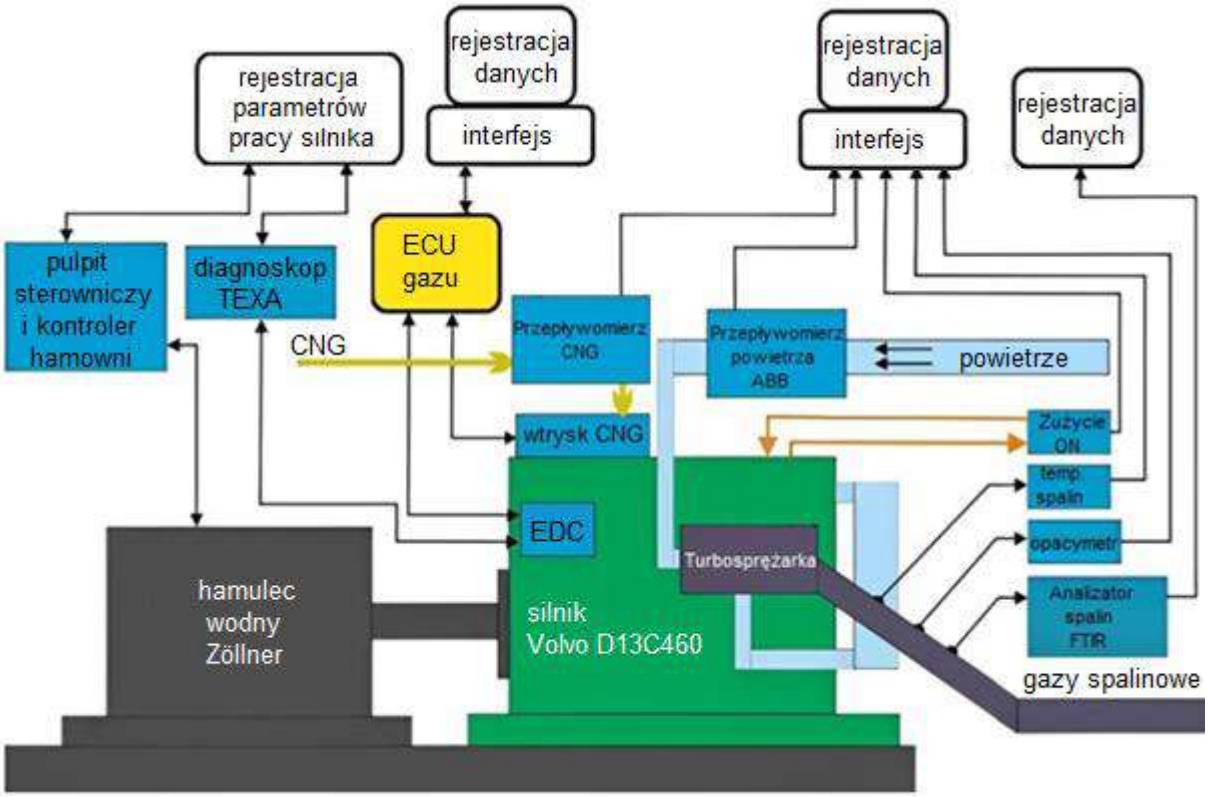
Badanie spalinowego silnika tłokowego stosowanego w pojazdach ciężarowych, posiadającego dużą pojemność skokową oraz wysoki generowany moment obrotowy w niskim zakresie prędkości obrotowej pracy silnika, wymaga wykorzystania w stanowisku badawczym odpowiedniego hamulca dynamometrycznego mogącego wygenerować wysokie wartości obciążenia. Stanowisko badawcze (rys. 16) zostało również wyposażone w systemy rejestrujące podstawowe parametry pracy silnika, takie jak temperatury czy ciśnienia w poszczególnych podzespołach silnika. Wykorzystano precyzyjne systemy mierzące zużycie paliwa nisko- i wysokoreaktywnego oraz powietrza, a także zaawansowany, wieloskładnikowy pomiar zawartości poszczególnych składników spalin i ich zadymienia. W tabeli 4 zaprezentowano wykorzystywane w badaniach na hamowni silnikowej oprzyrządowanie badawcze.

Tabela 4. Zestawienia urządzeń pomiarowych na stanowisku badawczym

Nazwa urządzenia /zmierzona ilość	Producent/model/typ urządzenia	Zakres pomiarowy	Dokładność
Hamulec dynamometryczny wodny: • Moment obrotowy – M_o • Prędkość obrotowa – n	Zöllner PS1-3812/AE	M_o = 0–7000 Nm n = 0–3000 obr./min Ne = 0–1250 kW	± 1 Nm ± 1 obr./min ± 1 kW
Waga paliwowa: • Zużycie ON – G_{eON}	AVL 733S Fuel Balance	0–200 kg/h	± 0,005 kg/h
Przepływomierz wirowy: • Zużycie CNG – G_{eCNG}	SwirlMaster FSS450	1,2–104,6 kg/h	± 0,6% zmierzonej ilości
Termiczny przepływomierz masowy: • Przepływ powietrza w układzie dolotowym – Q_s	SensyMaster FMT430	100–6000 Nm^3/h	± 1,0 Nm^3/h
Dymomierz: • Zadymienie spalin – współczynnik ekstynkcji promieniowania świetlnego – k	AVL Opacimeter 4390	0,001–10 m^{-1}	± 0,002 m^{-1}
Analizator spalin – pomiar stężenia określonych składników toksycznych w spalinach: • dwutlenek węgla (CO_2) • tlenek węgla (CO) • tlenki azotu (NO_x) • dwutlenek azotu (NO_2) • podtlenek azotu (N_2O) • węglowodory (HC) • woda (H_2O) • metan (CH_4) • etan (C_2H_6)	Protea atmosFIR (system monitorowania emisji spalin wykorzystujący spektroskopię fourierowską w falach podczerwonych)	CO_2 (0,01–23)% CO (1,0–11000)ppm NO_x (1,0–6000) ppm NO_2 (0,8–342) ppm N_2O (0,6–326) ppm HC (1,0–5000) ppm H_2O (0,25–25)% CH_4 (1–5000) ppm C_2H_6 (1–820) ppm	± 0.1% zmierzonej wartości (dla poszczególnych składników dokładność może się różnić ze względu na dokładność wyznaczenia

• propan (C_3H_8) • formaldehyd CH_2O • eten (C_2H_4) • tlen (O_2)		C_3H_8 (0,8–550) ppm CH_2O (0,8–219)ppm C_2H_4 (0,6–880) ppm O_2 (0,1–21)%	charakterystyki widmowej przez producenta urządzenia)
Termopary: • pomiar temperatury	NiCr – NiAl (K)	-50–1100°C	± 1°C
Ciśnieniomierz: • ciśnienie w układzie dolotowym	TESTO 400	-100–200 hPa	0,3 Pa + 1% zmierzonej ilości
Stacja pogodowa (warunki otoczenia): • ciśnienie atmosferyczne • wilgotność powietrza • temperatura powietrza	 Termohigrobarometr LAB-EL LB-706B	 800–120hPa 0–100 % -50–120°C	 ± 0,1 hPa ± 0,1% ± 0,1°C

W trakcie realizacji badań posiłkowano się również danymi opisującymi pracę silnika ze sterownika EDC, z wykorzystaniem diagnoskopu TEXA komunikującego się ze sterownikiem poprzez linię CAN i danych z dwupaliwowego układu zasilania, wykorzystując interfejs z układem FTDI (ang. *Future Technology Devices International Ltd*) w celu połączenia ECU gazu z komputerem osobistym rejestrującym dane. Istotne dane pomiarowe z tych systemów, takie jak moment obrotowy, generowana moc, odczyty temperatur i ciśnień, zostały zweryfikowane co do poprawności wskazań z aparaturą pomiarową wymienioną w tabeli 4. Na rys. 16 zaprezentowano schemat stanowiska badawczego.



Rys. 16. Schemat stanowiska badawczego z silnikiem Volvo D13C460 [83]

W badaniach użyto dwóch paliw:

- ON – paliwo zgodne z PN-EN590 [118],
- CNG – paliwo zgodne z ISO 15403-2:2010 [119].

Każde z paliw pochodziło z jednej partii i miało parametry normatywne zgodne z ich kartą charakterystyki. Wybrane parametry zestawiono w tabeli 5:

Tabela 5. Wybrane właściwości paliw wykorzystanych do badań

Parametr	ON	CNG
Gęstość w 15°C, [kg/m3]	831	0,655
Liczba oktanowa (LO) [-]	30	110
Liczba centanowa (LC) [-]	51	0
Temp. samozapłonu [°C]	180–230	650
Stechiometryczny stosunek powietrza do paliwa	14,6	17,05
Wartość opałowa [MJ/kg] [MJ/m³]	43	36
Wartość opałowa mieszanki λ=1 [MJ/m³]	3,66–3,83	3,4
Dolna granica palności [% obj. gazu w powietrzu]	0,6	5

Dokonano analizy niepewności toru pomiarowego na podstawie metod statystycznych, zgodnie z wytycznymi Głównego Urzędu Miar [120]. Wyliczona w ten sposób niepewność nosi nazwę niepewności typu A. Druga składowa niepewności pomiaru to niepewność typu B. Jej wyznaczanie polega na analizie niepewności zastosowanych przyrządów oraz innych przyczyn mogących wpływać na wynik pomiaru. Niepewność standardowa została wyznaczona na podstawie błędu granicznego. Stąd zależność na niepewność standardową typu B (U_B) przyjmuje postać:

$$U_B = \frac{\Delta gr}{\sqrt{3}}, \qquad (2)$$

gdzie: Δgr – błąd graniczny, U_B – niepewność standardowa.

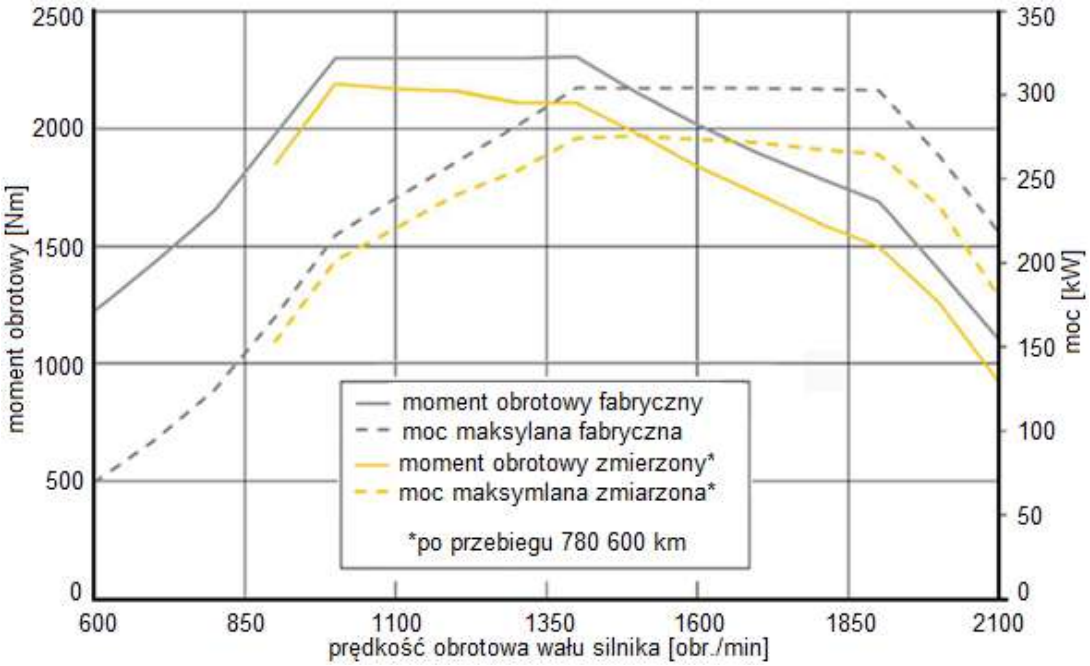
W zależności od wzajemnej relacji między wartością niepewności A i B określona została niepewność złożona, a następnie niepewność rozszerzona. Niepewności rozszerzono ze współczynnikiem $k = 2$ - zgodnie z własnościami rozkładu t – studenta: poziom ufności $p = 0,9$ [121].

Poprawne porównanie wyników uzyskanych w różnych dniach pomiarowych było możliwe po zredukowaniu ich zgodnie z zależnościami przedstawionymi w [123]. Badania i wyniki użytecznych pomiarów silnika zostały przekształcone na warunki normalne zgodnie z PN-ISO15550:2009 – Silniki spalinowe tłokowe – Określenie i metoda pomiaru mocy silnika – Wymagania ogólne [122], które zalecają korygowanie mocy silnika z uwzględnieniem

warunków otoczenia poprzez pomnożenie wartości określonej bezpośrednio przez obliczone współczynniki korekcyjne.

5.1.2. Ocena stanu technicznego silnika przed modyfikacją

W kontekście przeprowadzania modyfikacji silników na zasilanie dwupaliwowe konieczność implementacji szybkich i ekonomicznych metod oceny stanu technicznego tychże silników ma szczególne znaczenie. Istotne jest dokładne określenie stanu faktycznego silnika przed przystąpieniem do jakichkolwiek zmian, co ma kluczowe znaczenie dla zapewnienia optymalnej efektywności i niezawodności niefabrycznego systemu zasilania dwupaliwowego oraz możliwości liczbowej oceny efektów wprowadzonych zmian. Jednakże tradycyjne metody oceny, takie jak sprawdzanie stopnia sprężania (w przypadku większości silników wykorzystywanych w pojazdach ciężarowych konieczny jest demontaż wtryskiwaczy ON) czy analiza z wykorzystaniem analizatora spalin, mimo że są skuteczne, okazują się zarówno czasochłonne, jak i kosztowne. W komercyjnych zastosowaniach, gdzie czas i koszty operacyjne są krytycznymi czynnikami, powyższe metody mogą nie być ekonomicznie uzasadnione. Szybkie badanie określające stan silnika może się więc składać z odczytu wskazań systemu autodiagnostyki (ang. *On-Board Diagnostics*, OBD) pojazdu oraz wyznaczenia charakterystyki przebiegu mocy i momentu obrotowego silnika pojazdu. W ramach weryfikacji stanu technicznego silnika wyznaczono charakterystyki zewnętrzne silnika będącego obiektem badawczym, określone podczas jego zasilania ON i przedstawiono oryginalne charakterystyki oraz porównano z nimi charakterystyki zewnętrzne (rys. 17).



Rys. 17. Fabryczne i zmierzone po przebiegu 780 600 km charakterystyki zewnętrzne silnika Volvo D13C460

Analiza charakterystyk zewnętrznych badanego silnika dostarcza pewnych informacji na temat stanu technicznego i pozwala na ocenę możliwości adaptacji silnika do zasilania dwupaliwowego CNG/ON. Zmierzone M_o oraz N_e, są niższe o ~9% od wartości fabrycznych w zmierzonym zakresie, co jest wyznacznikiem ogólnego zużycia silnika (wynikającego z zużycia elementów takich jak wtryskiwacze PD czy połączenia tribologicznego: tłok, pierścienie, cylinder). Systemy dwupaliwowe wymagają, aby silnik był sprawny technicznie. Silnik o przebiegu 780 600 km, pomimo spadku M_o, nie może być uznany za wyeksploatowany. Silniki ciągników siodłowych poddawane są naprawie głównej zwykle przy przebiegu ok. 1,5–2,0 mln km [123]. Zgodnie z publikacją [124] badany obiekt znajduje się więc w „obszarze 2" krzywej „wannowej", a zaobserwowany spadek wartości momentu obrotowego i mocy można uznać za niewpływający negatywnie na decyzję dotyczącą zaadoptowania systemu dwupaliwowego, co dodatkowo potwierdza brak błędów w systemie OBD.

5.1.3. Założenia koncepcyjne oraz rozwiązania techniczne konwencjonalnej instalacji dwupaliwowej (CDF) o wysokim współczynniku zastąpienia paliw

Założeniem elementarnym w trakcie realizacji prac nad instalacją zasilania było wykorzystanie w niej CNG jako paliwa niskoreaktywnego oraz ON jako paliwa wysokoreaktywnego. Koncepcja realizacji projektu NCBiR: Mazowsze/0123/19-00 swoje uzasadnienie ekonomiczne miała w przepisach – zgodnie z § 1 ust. 2 zd. 1 nr 8 Niemieckiej federalnej ustawy o opłatach za korzystanie z dróg szybkiego ruchu, w okresie 1 stycznia 2019 – 31 grudnia 2023 roku pojazdy ciężarowe używające LNG lub CNG do napędu z przewagą gazu ziemnego jako głównego paliwa zwolnione były z opłat drogowych w systemie „Toll Collect" [55]. W związku z powyższym wysoki potencjał komercjalizacyjny, powiązany bezpośrednio z dostępnością stacji tankowania na terenie EU, był kluczowy i determinujący przy doborze paliw.

W celu skonstruowania instalacji CDF realizującej *Wz* ponad 50% w pełnym zakresie obciążeń opracowano założenia koncepcyjne zakładające pierwotnie:

A. Modyfikację sygnałów prądowych sterowania pompowtryskiwaczy

W procedurze tej sygnały wtryskiwaczy PD wysyłane z EDC zostają transformowane w ECU instalacji gazowej w sposób docelowo skracający czas otwarcia i zmieniający strategię ich pracy (najczęściej na wtrysk pojedynczy ciągły, wykorzystywany jako dawka inicjująca). Sterownik gazowy wysyła również do EDC odpowiedź informującą o poprawnej realizacji

pierwotnych instrukcji dla pompowtryskiwacza, co nie wywołuje błędów jego pracy dotyczącej ingerencji w aspekt sterowania. Dzięki tej metodzie można ograniczyć dawkowanie wtryskiwacza do wartości dawek minimalnych ograniczonych przez właściwości konstrukcyjne wtryskiwacza i możliwe jest przekroczenie w znaczny sposób wartości współczynnika zastąpienia, w porównaniu z metodami opisanymi w podrozdziale 2.4.2. W celu zastosowania tej metody niezbędne są badania empiryczne mające na celu wyznaczenie charakterystyk pracy wtryskiwaczy PD oraz przetransformowanie otrzymanych wyników na zależność czasu otwarcia wtryskiwaczy do ich dawkowania paliwa. W tabeli 6 zaprezentowano wyniki badań:

Tabela 6. Zestawienie wybranych efektów badań charakterystyk dawkowania wtryskiwaczy PD

Prędkość obrotowa [obr./min]	Moc zmierzona [kW]	Masa paliwa na godzinę [kg/h]	*Czas otwarcia wtryskiwacza [ms]	Wyliczona pojedyncza dawka wtrysku [mg]
800	3	36	1,48	17
800	59	362	2,85	85
800	105	1232	4,91	160
1000	54	516	2,30	81
1000	100	955	3,24	135
1000	150	1432	3,80	168
1000	200	1910	4,73	223
1200	50	398	2,00	54
1200	150	1194	3,21	142
1200	250	1989	4,32	227
1400	50	341	1,91	54
1400	150	1023	2,76	123
1400	251	1801	3,60	226
1800	50	265	1,64	50
1800	150	796	2,36	104
1800	250	1705	4,53	208

**Czas pomniejszony o opóźnienie początku wtrysku*

Przeprowadzone badania oraz kalkulacje wykazały dużą linearyzację działania wtryskiwaczy, tj. zależności dawki paliwa do czasu otwarcia wtryskiwacza w zakresie eksploatacyjnych prędkości obrotowych silnika. Opracowane wyniki zostały zaimplementowane w ECU instalacji zasilania gazowego w celu realizacji modyfikowania sygnału (rys. 18). W takcie badań zauważono dzielenie wtrysku przez EDC w zakresie prędkości obrotowej biegu jałowego (600–700 obr./min). Uniemożliwiało to efektywne realizowanie modyfikacji sygnału w tym zakresie. Na rys. 18 zaprezentowano zdjęcie wskazań oscyloskopu obrazujące przykładowe przebiegi oryginalnego oraz zmodyfikowanego sygnału EDC.

Rys. 18. Porównanie sygnałów kontroli wtryskiwaczy PD: oryginalny EDC (żółta linia) i zmodyfikowanego (niebieska linia/wykres). Pomiary wykonane z użyciem oscyloskopu Rigol MSO2202A

B. Kluczowe modyfikacje mechaniczne

W celu realizacji wartości *Wz* >50% system dwupaliwowy został wyposażony w modyfikacje mechaniczne, w skład których wchodziło zastosowanie:

- armatury gazowej (reduktora ciśnienia gazu i wtryskiwaczy) umożliwiającej utrzymywanie ciśnienia roboczego układu w przedziale 5,0–8,0 bar na wyjściu (typowo 1,5–3,0 bar). Modyfikacja ta przyczynia się do poprawy rozpylenia gazu, lepszego jego wymieszania oraz efektywnej pracy instalacji w warunkach wysokiego ciśnienia doładowania i przy dużym obciążeniu silnika, a w konsekwencji zapotrzebowaniu na paliwo w ok. 50–60 kg/h;
- dwóch wtryskiwaczy gazowych na cylinder z montażem w pobliżu zaworów dolotowych (rys. 19b) na specjalnym adapterze montowanym pomiędzy głowicą a kolektorem dolotowym (rys. 19a, 19b) – w celu przesunięcia wtryskiwaczy maksymalnie w stronę zaworów dolotowych. W silnikach pojazdów ciężarowych często stosuje się kolektor dolotowy zbiorczy (bez podziału na drogi gazowe dla poszczególnych cylindrów). Utrudnia to wykorzystanie w nich typowych w układach przyłączy gazowych (tzw. „wkrętek”) montowanych w kolektorach sekcyjnych dla poszczególnych cylindrów;
- wspólnej listwy paliwowej wtryskiwaczy gazowych, celem utrzymywania jednolitego ciśnienia gazu dla wszystkich wtryskiwaczy (rys. 19);

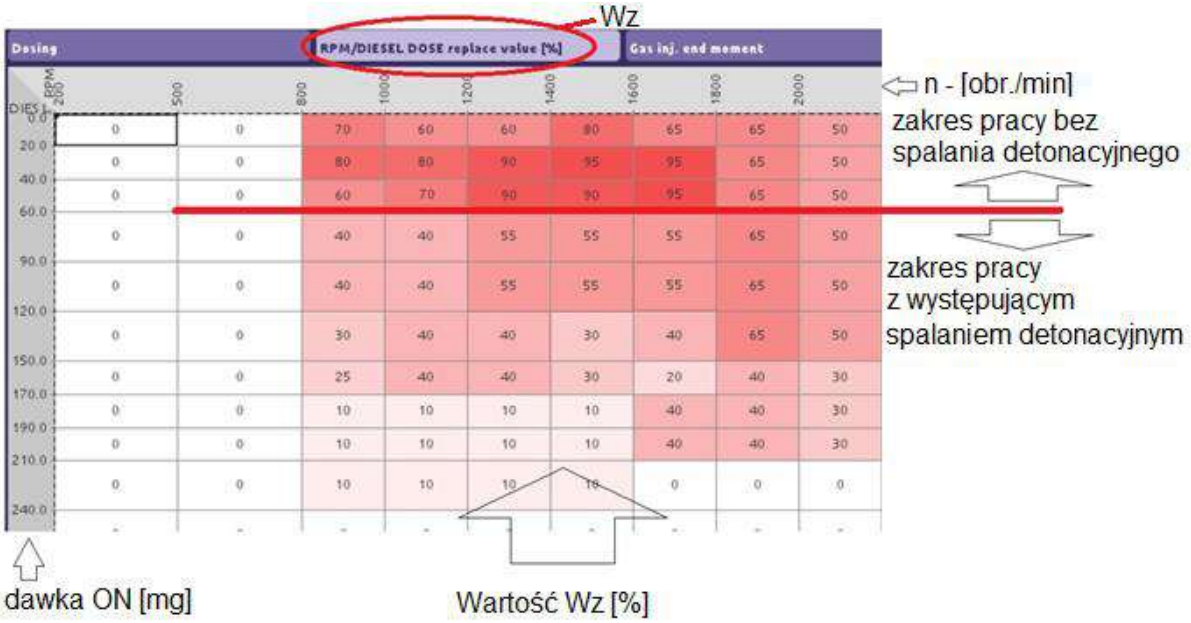
- czujnika nadmiaru powietrza w spalinach (tzw. „sonda lambda”), celem możliwości wprowadzenia korekt dla wtryskiwaczy gazowych w przypadku zidentyfikowania mieszanek zbyt bogatych;
- czujnika temperatury spalin – celem możliwości wprowadzenia korekt czasu otwarcia wtryskiwaczy gazowych w przypadku wykrycia podwyższonej temperatury gazów spalinowych np. przy dawkach zbyt „ubogich”;
- otwartego, w pełni programowalnego ECU instalacji gazowej, pracującego równolegle z EDC.

a) b)

Rys. 19. Adapter i listwa paliwowa: a) widok ze zdemontowanym kolektorem i wtryskiwaczami; b) widok z założonym kolektorem dolotowym i oraz zamontowanymi dwoma wtryskiwaczami gazowymi na cylinder

C. System ograniczający występowanie spalania detonacyjnego

Po wdrożeniu zaprezentowanych powyżej modyfikacji mechanicznych i sterowania dla zrealizowania CDF badanej instalacji w wersji pierwotnej zrealizowano możliwość stosowania *Wz* w zakresie od 5% do 95%. Wymiana na poziomie 95% skutkowała sporadycznie generowaniem błędów w systemie OBD, co powodowało wyłączenie danego wtryskiwacza PD z pracy do ponownego włączenia silnika. W kolejnym kroku przeprowadzono badania wstępne w celu sprawdzenia zachowania się badanej instalacji w warunkach dużych wartości współczynnika zastąpienia i przy dużych wartościach generowanego momentu obrotowego. Celem badań było zdefiniowanie obszarów występowania spalania detonacyjnego. Efekty przeprowadzonych badań zaprezentowano na rys. 20 i 21:

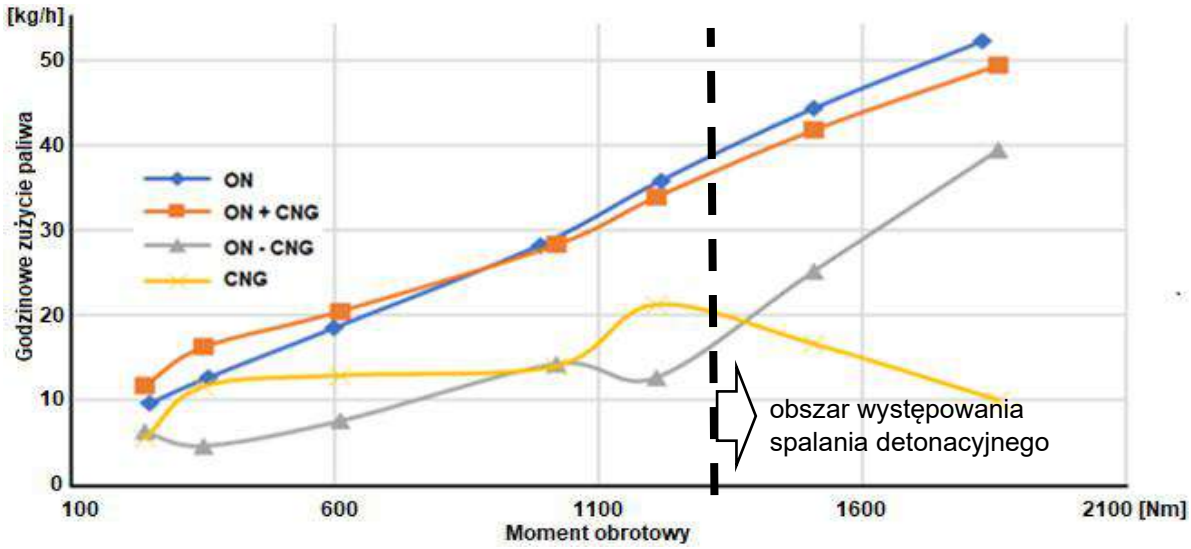


Rys. 20. Widok z oprogramowania ECU gazu z mapą współczynnika zastąpienia paliw w funkcji obrotów wału i dawki ON wyliczonej przez EDC z zaznaczonymi strefami spalania detonacyjnego

Na powyższym rysunku można zaobserwować tendencję do występowania spalania detonacyjnego wraz ze wzrostem wyliczonej dawki ON. Uniemożliwiło to przy pracy w zakresie prędkości obrotowych optymalnych (1150–1300 obr./min, tabela 3) efektywne realizowanie współczynnika zastąpienia powyżej:

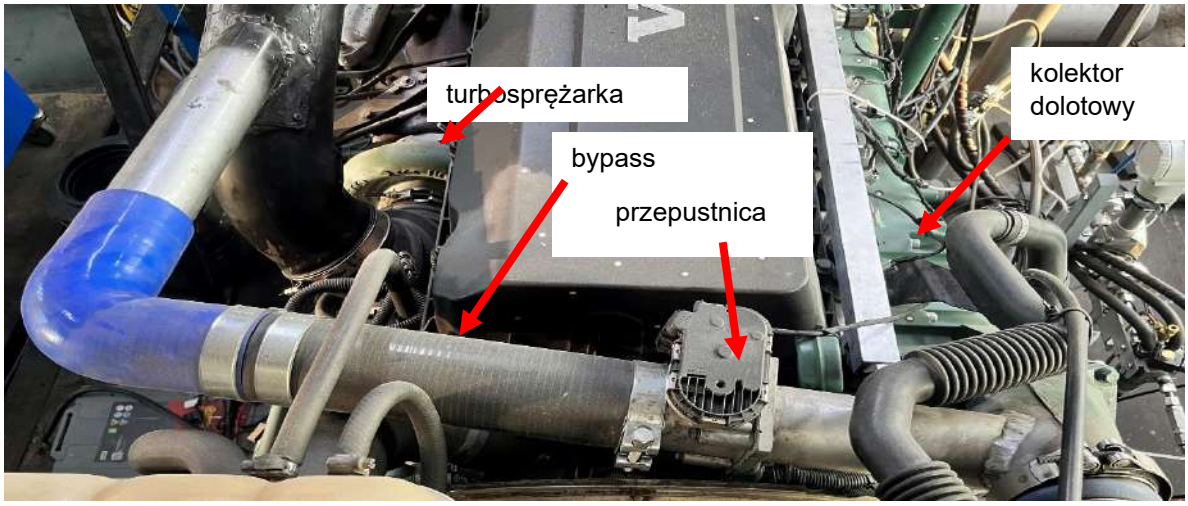
- 55% w zakresie 90–120 mg wyliczonej dawki ON,
- 40% w zakresie 150–170 mg wyliczonej dawki ON,
- 10% w zakresie 190–240 mg wyliczonej dawki ON.

Wskazuje to na poważne ograniczenia wymiany wraz ze wzrostem generowanego momentu obrotowego, które zobrazowano na rys. 21.



Rys. 21. Zależność *Ge* dla: ON, CNG+ON, ON – CNG w funkcji generowanego *Mo* przy $n_{const.}$=1300 obr./min

Zaobserwowano wyraźny spadek godzinowego zużycia gazu ziemnego (Ge_{CNG}) wraz ze wzrostem generowanego momentu obrotowego, wynikający z konieczności minimalizacji występowania spalania detonacyjnego i jednocześnie uniemożliwiającego uzyskanie wysokiego współczynnika zastąpienia przy dużym obciążaniu silnika (zakres wyliczonej dawki ON 90-240 mg). Więcej na temat powodów występowania spalania detonacyjnego oraz szerszy zakres wyników zawarto w publikacji [A5]. W artykule tym zaprezentowano również koncepcję i realizację systemu, którego celem było zminimalizowanie występowania spalania detonacyjnego poprzez zmniejszanie ciśnienia doładowania oraz zautomatyzowane zmniejszanie współczynnika zastąpienia paliw po jego wykryciu. W systemie tym zastosowano obejście (tzw. *bypass*) przepływu powietrza dolotowego z przepustnicą, mogące zmniejszać ciśnienie doładowania poprzez częściowy „upust" powietrza za chłodnicą powietrza doładowanego i przekierowanie go do sekcji sprężającej turbosprężarki jak na rys. 22. W celu uzyskania poprawności działania tego układu w oprogramowanie ECU wkomponowano moduł emulujący wskazania czujnika ciśnienia bezwzględnego (ang. *Manifold Absolute Pressure Sensor, MAP sensor*) w kolektorze dolotowym. Zastosowano również akustyczny czujnik spalania detonacyjnego, którego odczyty, dzięki modyfikacji oprogramowania ECU instalacji zasilania gazowego, wymuszały na systemie zwiększanie współczynnika zastąpienia paliw w sposób zautomatyzowany.



Rys. 22. *Bypass* i przepustnica systemu minimalizującego efekty spalania detonacyjnego

Zastosowanie omówionego w tym podrozdziale systemu minimalizującego występowanie spalania detonacyjne umożliwiło w zakresie pracy obrotów optymalnych (1150–1300 obr./min, tabela 3) efektywne realizowanie współczynnika zastąpienia paliw powyżej:

- 80% w zakresie 90–120 mg wyliczonej dawki ON,
- 75% w zakresie 150–170 mg wyliczonej dawki ON,

- 40% w zakresie 190–240 mg wyliczonej dawki ON.

Na tak przygotowanej instalacji CDF wykonano wstępną optymalizację parametrów eksploatacyjnych pracy układu tzw. „strojenia” na badanym silniku w celu „odtworzenia” możliwie zbliżonego do fabrycznego przebiegu momentu obrotowego, przy jak najwyższych wartościach *Wz*. W trakcie badań zaobserwowano możliwość osiągania wartości momentu obrotowego i mocy wyższych niż wartości fabryczne (rys. 17). Zauważono problem niestabilnej pracy w stanach nieustalonych wynikający z braku aproksymacji pomiędzy komórkami mapy wymiany gazowej (rys. 20).

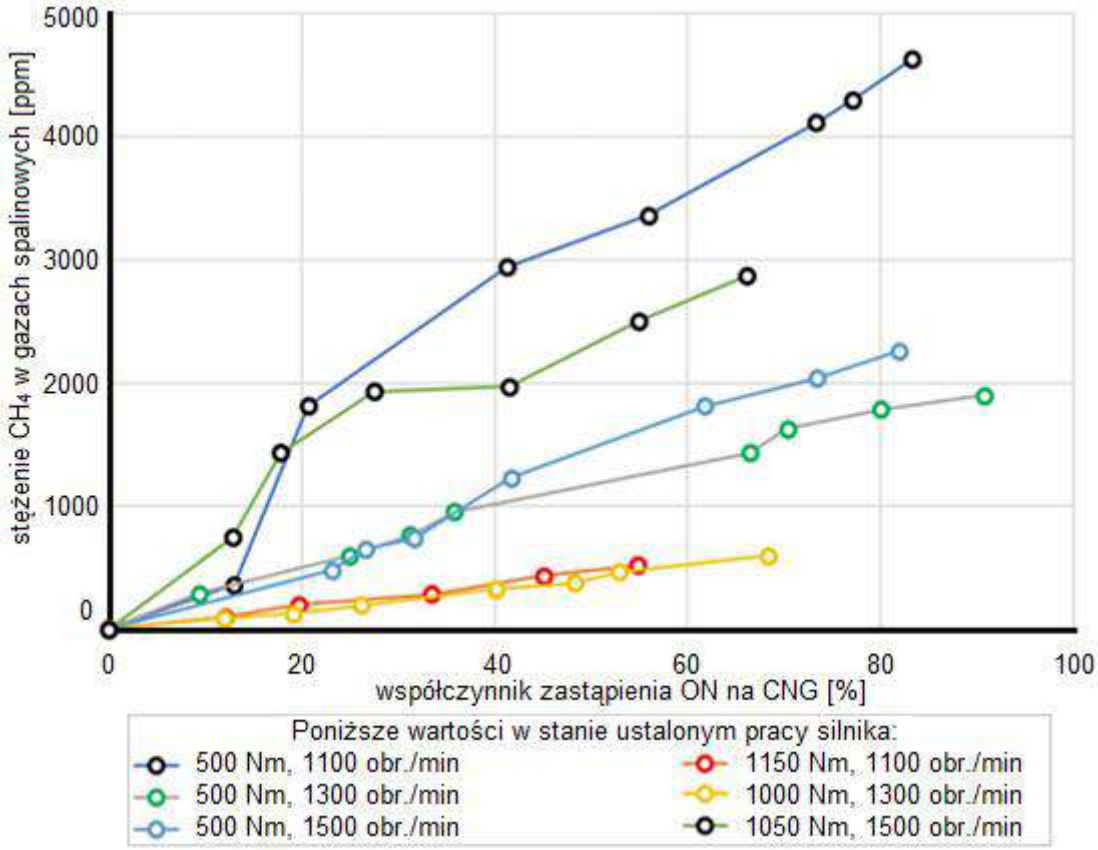
5.1.4. Badania i analiza składu spalin

W kolejnym kroku dla zoptymalizowanej instalacji CDF CNG/ON przeprowadzono badania wpływu współczynnika zastąpienia na skład i zadymienie spalin. Pełen zakres badań i wyników omówiono w artykule [A8]. Metodyka przeprowadzanych badań polegała na wytypowaniu punktów pracy silnika odpowiadających najczęściej występującym warunkom pracy silnika wykorzystywanego w ciągniku siodłowym, realizującego dalekobieżny transport drogowy, tj. zakres prędkości obrotowych ekonomicznych (1100–1500 obr./min, tabela 3). Badania przeprowadzono dla dwóch wariantów obciążenia. W wybranych poszczególnych punktach pracy *Wz* paliw zmieniano w oprogramowaniu skokowo o 10%, począwszy od 10%. W każdym punkcie pracy silnika ilość CNG była korygowana za pomocą sterownika gazowego w taki sposób, aby wytworzony przez silnik moment obrotowy był zgodny z M_o na jednopaliwowym zasilaniu ON. Badania prowadzono do momentu osiągnięcia *Wz*, w którym zidentyfikowano występowanie spalania detonacyjnego. Próbki spalin „surowych” pobierano przed układem SCR. Miało to na celu m.in. umożliwienie dalszej oceny zakresu modyfikacji systemu oczyszczania spalin. W tabeli 7 zestawiono wybrane wyniki przeprowadzonych badań.

Tabela 7. Zestawienie *Wz* dla najniższych stężeń wybranych zmierzonych szkodliwych składników spalin

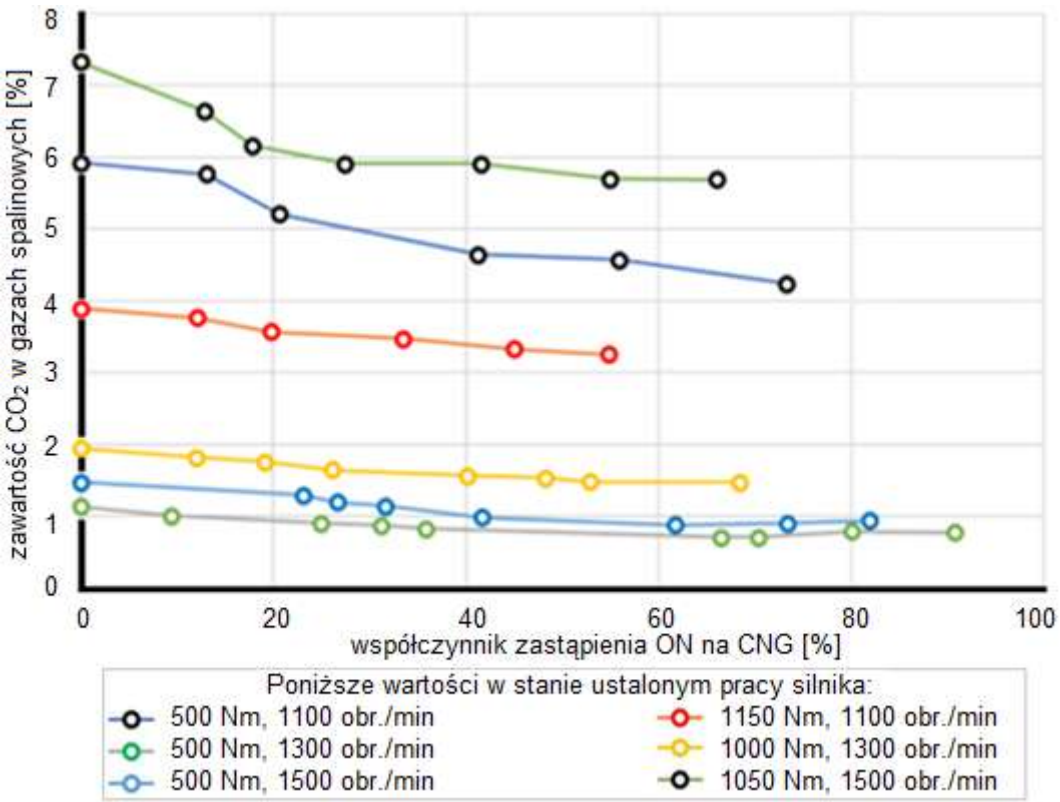
Wz dla najniższych wartości stężeń poszczególnych składników w spalinach [%]							M_o [Nm]	*n* [obr./min]
k_{min}	CO	NO	NO_x	NH_3	C_2H_6	CH_4		
80	20	70	80	70	50	20	500	**1500**
60	15	30	30	60	20	15	1050	
80	20	85	85	60	20	20	500	**1300**
60	15	40	40	70	15	15	1000	
60	20	70	50	80	30	20	500	**1100**
40	15	40	30	50	15	15	1150	

W trakcie badań zaobserwowano złożone interakcje wpływające na skład spalin. Zwiększanie współczynnika zastąpienia nie prowadzi do spadku temperatury i zadymienia spalin. Zaobserwowano natomiast wzrost stężenia CO, NO_x, niemetanowych węglowodorów (ang. *non methane hydrocarbon,* NMHC) oraz duży wzrost stężenia CH_4 w spalinach (rys. 23), podczas gdy zawartość CO_2, etanu (C_2H_6) i amoniaku (NH_3) uległa obniżeniu.



Rys. 23. Stężenie metanu w gazach spalinowych dla poszczególnych punktów pracy silnika (różna prędkość obrotowa, różny moment obrotowy) w funkcji *Wz*

Określenie optymalnej regulacji instalacji CDF okazało się trudne ze względu na różny wpływ współczynnika zastąpienia paliw. Mimo to badania pozwoliły na określenie wpływu udziału CNG w spalaniu dwupaliwowym w kontekście składu gazów spalinowych. Użycie paliwa CNG skutkowało wzrostem udziału toksycznych związków w spalinach, najbardziej HC, z zawartością tych składników znacznie przewyższającą wartości dla silnika zasilanego wyłącznie ON. Odnotowano spadek zawartości CO_2, korzystny z punktu widzenia emisji gazów cieplarnianych (rys. 24). Największy osiągnięty współczynnik zastąpienia bez występowania spalania detonacyjnego wynosił około 90% w badanym zakresie prędkości i obciążeń silnika, co można uznać za zadowalający wynik.



Rys. 24. Procentowa zawartość CO_2 w spalinach dla poszczególnych punktów pracy silnika (różna prędkość obrotowa, różna moment obrotowy) w funkcji *Wz*

Uzyskane wyniki wskazują na potencjalne korzyści płynące ze zwiększania *Wz* w testowanej instalacji CDF, ale jednocześnie podkreślają wyzwania związane ze zwiększeniem stężeń niektórych zanieczyszczeń, co wymaga optymalizacji i ewentualnych zmian konstrukcyjnych. Kluczowe w kontekście redukcji emisji CH_4 jest zminimalizowanie czasu współotwarcia (przekrycia) zaworów dolotowego i wydechowego, obecnego w silnikach o ZS. Zmiany tej można dokonać poprzez dobór faz rozrządu z wykorzystaniem np. zmodyfikowanego wałka rozrządu, systemu zmiennych faz rozrządu czy VVA oraz optymalizacji procesu wtrysku paliwa CNG. Kolejnym z sugerowanych sposobów ograniczenia emisji metanu jest wykorzystanie specjalnie do tego przeznaczonego bądź zmodyfikowanie fabrycznego reaktora katalitycznego dla tego składnika spalin. Poprawę składu spalin można uzyskać również poprzez wykorzystanie w systemie dwupaliwowym biopaliw z domieszką wodoru, które omówione zostały w rozdziale czwartym. Brak powyższych modyfikacji najprawdopodobniej uniemożliwi spełnienie aktualnych norm emisji spalin. W dalszym zakresie prac niezbędne jest przeprowadzanie badań z dobranym odpowiednio układem zmniejszającym obecność szkodliwych składników w gazach spalinowych, np. zmodyfikowanym układem fabrycznym SCR lub dołożonym reaktorem katalitycznym.

5.1.5. Podsumowanie pierwszego etapu badań własnych – hamownia silnikowa

W ramach prac na hamowni silnikowej zbudowano stanowisko wykorzystujące używany silnik Volvo D13C460 i określono jego stan techniczny jako umożliwiający proces retrofittingu w zasilanie dwupaliwowe. Opracowano założenia koncepcyjne instalacji CDF zasilanej CNG/ON przy wysokim współczynniku zastąpienia oraz wykonano i wdrożono do testów jego wstępną wersję. Na początku badań określono charakterystyki działania pompowtryskiwaczy, w efekcie czego możliwe stało się osiągnięcie *Wz* w zakresie 5–90%. Następnie opracowano koncepcję oraz wdrożono szereg modyfikacji mechanicznych instalacji, docelowo minimalizując efekty spalania detonacyjnego przy dużych obciążeniach silnika. Wykonano wstępne „strojenie" instalacji dwupaliwowej i zaobserwowano możliwość osiągnięcia wartości momentu obrotowego i mocy wyższych niż znamionowe (fabryczne). Zauważono problem niestabilnej pracy w stanach nieustalonych, wynikający z braku aproksymacji pomiędzy komórkami mapy współczynnika zastąpienia. Finalnie przeprowadzono pomiar oraz ocenę emisji poszczególnych składników spalin, które ujawniły zależność pomiędzy wzrostem współczynnika zastąpienia paliw a obniżeniem stężenia CO_2 – dzięki zastosowaniu CNG, mającego inną proporcję C/H w porównaniu do ON. Odnotowano kilkukrotne zwiększenie emisji CH_4, których głównym powodem jest współotwarcie zaworów dolotowego i wydechowego (przedmuch powietrza zawierającego paliwo metanowe podczas tzw. przepłukania cylindra). Jako potencjalne rozwiązania tego problemu zaproponowano: modyfikację faz rozrządu, dodanie przeznaczonego do pracy z dużym stężeniem metanu w spalinach reaktora katalitycznego w układzie wydechowym lub użycie paliw z dodanym wodorem. W związku z powyższym w kolejnym etapie prac w celu poszerzenia zakresu badawczego konieczne było wykonanie badań instalacji CDF zainstalowanej w kompletnym pojeździe oraz w warunkach pracy dynamicznej.

5.2. Badania na hamowni podwoziowej

Badania, które nie były możliwe do wykonania na hamowni silnikowej, ale były możliwe na hamowni podwoziowej, to przede wszystkim te, które wymagały oceny całego pojazdu w warunkach jak najbardziej zbliżonych do rzeczywistych, tj:

- testy z kompletnym układem napędowym pojazdu,
- sprawdzenie poprawności działania pojazdu i jego podsystemów z dodanym systemem zasilnia dwupaliwowego,

- sprawdzenie wpływu modyfikacji na komponenty układu napędowego (przekazywania mocy, sprzęgła, opon itd.),
- wykonanie „strojenia” układu dwupaliwowego w warunkach pracy dynamicznej.

Samochody ciężarowe, ze względu na specyfikę ich zastosowania, generują na kołach napędowych wysokie wartości momentu siły, które są niezbędne do transportowania ciężkich ładunków przy zachowaniu odpowiedniej dynamiki, a także w zróżnicowanych warunkach drogowych, takich jak zmiany elewacji terenu czy różnice w prędkości i kierunku wiatru. Wymuszenie pracy silnika w skrajnych warunkach podczas badań drogowych bez ładunku jest niemożliwe lub bardzo trudne [125]. Problem ten wynika z trudności w generowaniu dużych wartości momentu obrotowego przy niskich prędkościach obrotowych silnika, co jest związane z ograniczeniami w generowaniu odpowiedniego ciśnienia doładowania. W odpowiedzi na te wyzwania zastosowanie hamowni podwoziowej umożliwia wymuszenie i utrzymanie optymalnych warunków pracy układu napędowego przez określony czas, co ma zasadnicze znaczenie dla precyzyjnej analizy i potencjalnej optymalizacji parametrów pracy silnika w zasilaniu dwupaliwowym. Ponadto w aspekcie retrofittingu pojazdów ciężarowych istotne jest znalezienie efektywnego i uzasadnionego ekonomicznie podejścia do procesu modyfikacji. Proces ten, w którego skład wchodzą: wstępna weryfikacja stanu technicznego pojazdu oraz niezbędna optymalizacja parametrów eksploatacyjnych dla instalacji dwupaliwowej, musi uwzględniać pełen zakres operatywny silnika w trakcie zasilania dwupaliwowego. Istotną wartością dodaną dla potencjalnych klientów wykonujących modyfikację jest możliwość zredukowania kosztów ekonomicznych i czasowych związanych z przetransportowaniem floty pojazdów do miejsca przeprowadzania modyfikacji. Realizacja modyfikacji na terenie parku maszyn klienta stanowi zatem znaczące udogodnienie, które może wpłynąć na decyzję o wyborze takiej formy retrofittingu i zdeterminować ostateczną decyzję.

W związku z powyższym w tym podrozdziale zaprezentowano proces konceptualizacji i realizacji mobilnej hamowni podwoziowej jako narzędzia do wykonywania retrofittingu w zasilanie dwupaliwowe. W dalszej części omówiono wyniki badań, w których opisano kluczowe aspekty prac realizowanych na hamowni podwoziowej w celu zapewnienia możliwie najmniejszej inwazyjności i najwyższej poprawność przeprowadzanych na niej działań, tj. badań, „strojenia” czy pomiarów. Następnie zaprezentowano rezultaty optymalizacji parametrów eksploatacyjnych badanej instalacji CDF z wykorzystaniem hamowni podwoziowej. Podrozdział ten zakończono syntetycznym podsumowaniem.

5.2.1. Opracowanie projektu i realizacja mobilnej hamowni podwoziowej dla pojazdów ciężarowych

Projektowanie i budowa mobilnej hamowni dla pojazdów ciężarowych przebiegały w kilku etapach. Autor był koordynatorem prac dotyczących powstania hamowni mobilnej. Działania rozpoczęły się od etapu zebrania wymagań ze względu na planowane wykorzystanie urządzenia. Na ich podstawie stworzono projekt koncepcyjny. Obejmował on ogólny układ urządzenia, typy wykorzystywanego osprzętu oraz ich funkcjonalności. Procesy projektowe i budowa zostały szerzej omówione w artykule [A10]. Po zdefiniowaniu założeń i zaprojektowaniu hamowni przeprowadzono jej budowę. W efekcie zbudowano mobilną hamownię dla pojazdów ciężarowych (rys. 25) na konstrukcji kontenera sześciostopowego i posiadającą system mocowania zgodny z normą ISO 1161, tj. standardowy dla kontenerów morskich. Pozwala to na transport hamowni przy użyciu ustandaryzowanych elementów montażowych [126].

Rys. 25. Widok ogólny mobilnej hamowni podwoziowej: a) widok z przodu z rampami; b) widok z tyłu; c) widok z pojazdem ciężarowym na hamowni; d) widok hamowni w trakcie transportu na pojeździe ciężarowym

Jednym istotnych założeń projektu hamowni było pozostawienie jej w pełni otwartej architektury *hardware* i oprogramowania, jak również możliwość projektowania indywidualnych trybów operacyjnych. W związku z tym komputer hamowni ma kilka złącz dla sygnałów cyfrowych i analogowych, wykorzystując wsparcie kontrolera USB, Bluetooth oraz

PWM (z ang. *Pulse Width Modulation* – modulacja szerokości impulsów). Ponadto oprogramowanie wyposażono w pełni konfigurowalny i kompatybilny protokół CAN (ang. *Controller Area Network*). Pomiary mogą być korygowane zgodnie z normami (tabela 8) dzięki wbudowanej stacji pogodowej uczestniczącej w pracy hamowni i realizacji na niej „testów specjalnych". Nadrzędnie ma to doprowadzić do ułatwionego rozszerzenia możliwości urządzenia poprzez przeprowadzanie zautomatyzowanych testów diagnostycznych pod obciążeniem. W tabeli 8 zaprezentowano parametry użytkowe i dane techniczne opracowanej hamowni podwoziowej.

Tabela 8. Parametry użytkowe i dane techniczne opracowanej mobilnej hamowni podwoziowej

	wartość/nazwa/typ
Tryby pracy	obciążeniowy
Typ hamulca	2x elektrowirowy Frenelsa FF16, 3300 Nm
Maks./min szerokość toru pomiarowego	5700 mm /1000 mm
Maks. prędkość liniowa na obwodzie rolki	200 km/h
Średnica rolki	320 mm
Układ docisku osi napędowej do rolek hamowni	wyciągarka elektryczna z odczytem siły naciągu
Wymiary zew. urządzenia	6058 mm x 2545 mm x 2870 mm
Masa własna	5574 kg
System transportu hamowni	Znormalizowany kontener sześciostopowy, system mocowania ISO 1161, manipulacja za pomocą HDS (hydrauliczny dźwig samochodowy), demontowalny dach
Tryby operacyjne	stały moment, stała prędkość, dynamiczny: rampowy, możliwość programowania cyklu obciążenia
Tolerancje temperaturowe (praca) [przechowywanie]	(-10 do +35) [-40 do +50] °C
Sterownik	OEM DynoRevolt na Raspberry Pi 4, kompatybilny z OBD2, złącze HDMI, 4xUSB 3.0, 4x wejście analogowe HP, 4x wejście analogowe, 3x lowside PWM
Software	OS Debian Linux 5.4.14, OEM dyno software DynoRevolt „Dyno2 RPI 2.17.0" w pełni konfigurowalne, obsługa sygnałów CAN, obsługa do 166 sygnałów silnikowych za pomocą złącza OBD pojazdu (chip ELM 327, STN1110) analogowych, cyfrowych PWM, Bluetooth, USB i OBD
Normy pomiarowe	DIN 70020, EC 95-1, ISO 1585, JIS D1001, SAE J1349

Kolejnym krokiem działań w ramach pracy doktorskiej było określenie, zbadanie i przenalizowanie istotnych aspektów bezinwazyjnego i poprawnego przeprowadzania badań układów dwupaliwowych na hamowni podwoziowej.

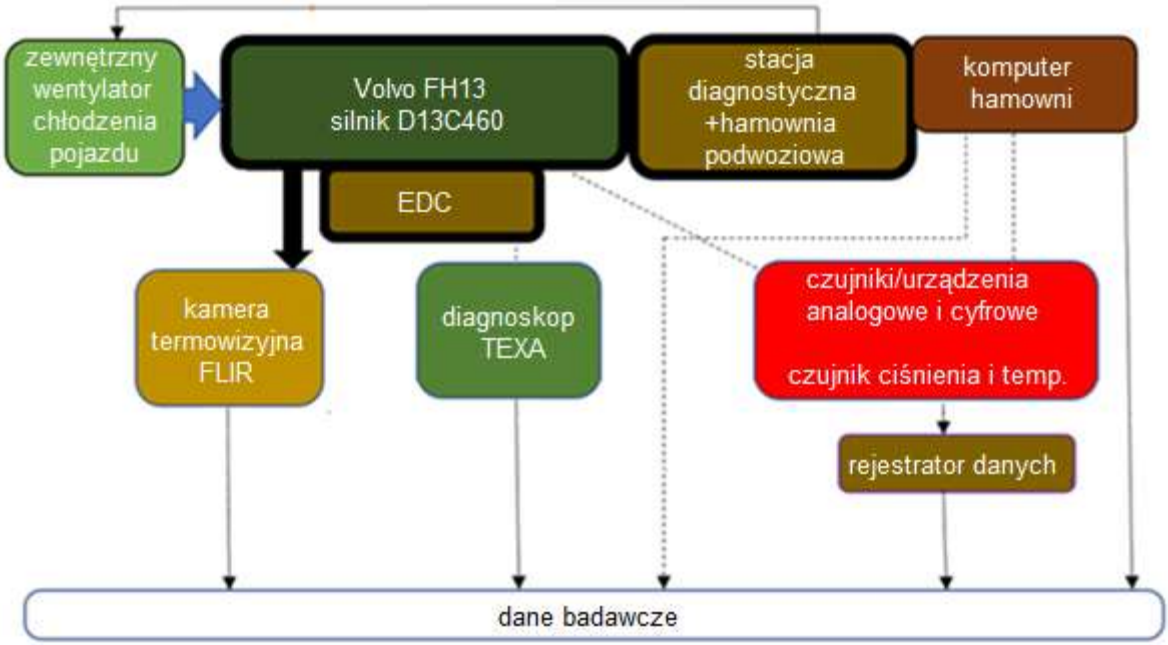
5.2.2. Analiza kluczowych aspektów bezinwazyjnego i poprawnego przeprowadzania badań na hamowni podwoziowej

Metodyka badań dotycząca treści zawartych w powyższym podtytule opracowana została empirycznie i obejmowała podejście do rozwiązywania aspektów związanych z:

- doborem przełożeń w skrzyni biegów – osiągnięcie pełnego zakresu prędkości obrotowej pracy silnika pojazdu na jednym przełożeniu;
- wpływem wartości wymuszonego obciążenia (docisku) osi napędowej pojazdu do rolek hamowni – kryteria optymalizacyjne: minimalizacja różnicy pomiarów prędkości obrotowej wału korbowego wyliczonego na hamowni i w EDC, temperatura opon;
- doborem trybu pracy hamowni odpowiednim do wybranych zadań – wybór trybów statycznych, dynamicznych i kombinowanych;
- wpływem poprawności doboru zakresu analizy danych ruchu swobodnego kół na rolkach hamowni, tzw. „wybiegu" na kalkulację strat mechanicznych w układzie napędowym.

Zaprezentowane powyżej aspekty mają wpływ na poprawność realizacji badań i „strojenia" układów dwupaliwowych oraz oddziałują pośrednio na możliwość przyspieszonego zużycia pojazdu. Zostały one szczegółowo omówione w artykule [A9].

Poniżej zaprezentowano najistotniejsze zdaniem autora fragmenty, tj. badania nad wpływem docisku osi (mającej bezpośredni wpływ na przyspieszoną degradację opon) oraz doboru trybu pracy na przebieg ciśnienia doładowania (mający wpływ na możliwość poprawnej optymalizacji pracy instalacji przy dużym obciążeniu silnika przy niskich prędkościach obrotowych wału korbowego). Testy przeprowadzono na ciąganiu siodłowym Volvo FH z silnikiem D13C460 (tabela 3. podrozdział 5.1.1.) z 12-stopniową, zautomatyzowaną skrzynią biegów „iShift" [127]. Przed przeprowadzeniem testów ustawienie pojazdu na hamowni było każdorazowo zabezpieczane zgodnie z zasadami BHP. Płyny eksploatacyjne samochodu były rozgrzewane do roboczej temperatury pracy, a bieżnik opon na początku każdego testu miał temperaturę 50°C. W celu zapewnienia powtarzalności testów monitorowano w czasie rzeczywistym szereg parametrów pracy silnika, takich jak temperatury, ciśnienia w odpowiednich podzespołach oraz prędkość obwodową kół pojazdu. Na rys. 25 zaprezentowano schemat ideowy stanowiska badawczego:



Rys. 26. Schemat ideowy stanowiska badawczego z hamownią podwoziową

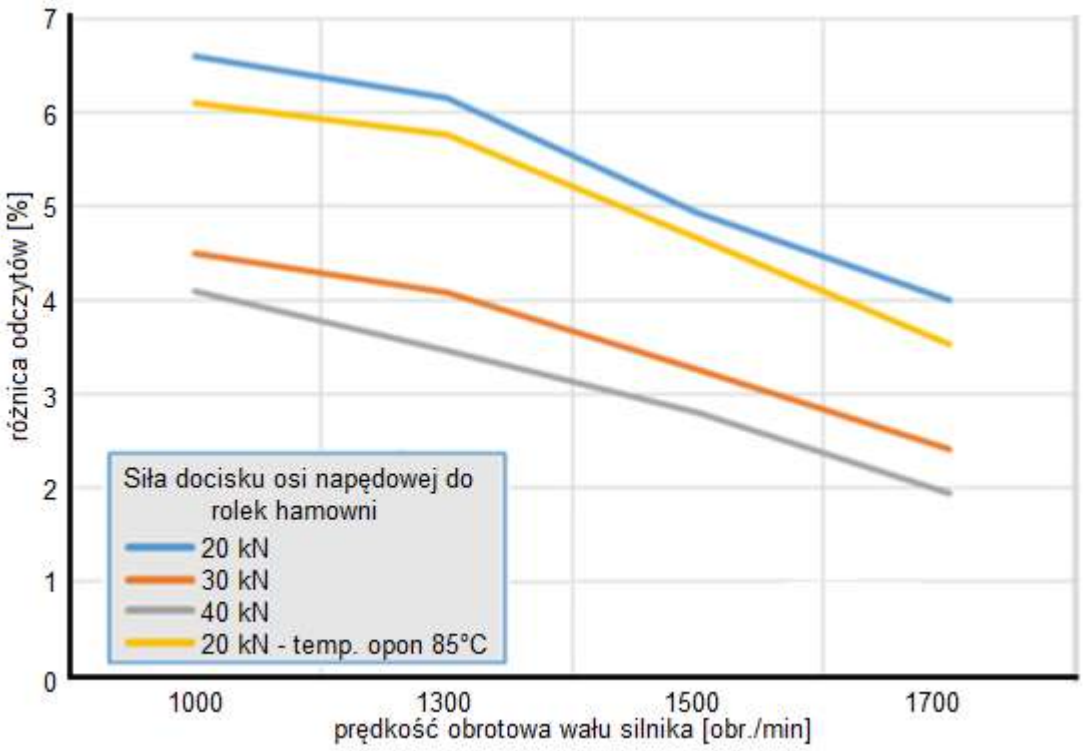
W trakcie badań zidentyfikowano niepożądany „poślizg" kół napędowych na rolkach hamowni. Stwarza to wielowymiarowy problem, na który składają się:

- zależność „poślizgu" od obciążenia osi napędowej na rolkach,
- powstający błąd różnicy odczytów prędkości obrotowej wału silnika na hamowni i rzeczywistych, zgodnie ze wzorem (3):

$$\text{różnica odczytów}=\frac{wyliczony\ n\ silnika\ (dane\ z\ EDC)}{wyliczony\ \mathrm{n}\ silnika\ (dane\ z\ rolek\ hamowni)} * 100\% \qquad (3)$$

- wpływ na przegrzewanie i degradację opon oraz zmiana ich promienia dynamicznego.

Na rys. 27 zaprezentowano efekt badań:

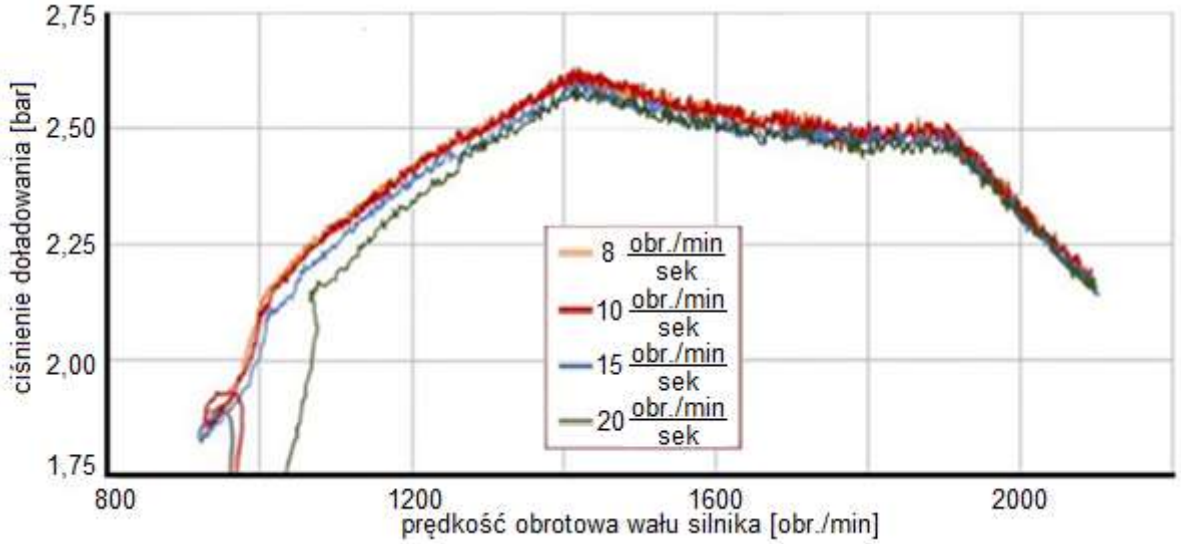


Rys. 27. Wykresy różnicy odczytów wynikających z „poślizgu" opony na rolkach hamowni dla pełnego obciążenia silnika

Na rys. 27 przedstawiono powiązanie zmniejszenia względnego błędu synchronizacji odczytów prędkości obrotowej wraz ze wzrostem docisku osi napędowej oraz zmniejszeniem generowanego momentu obrotowego (w celu zestawienia osi x na rys. 26 z generowanym momentem obrotowym zalecane jest przeanalizowanie rys. 17, podrozdział 5.1.3.). Wzrost docisku osi natomiast generował wykładniczy wzrost temperatury opon w trakcie testu. Gwałtowne nagrzewanie się opony doprowadza do jej szybszej degradacji oraz w skrajnych przepadkach do uszkodzenia. Ogranicza to np. maksymalny czas trwania realizowanego testu na hamowni. W powyższym przykładzie po uwzględnieniu zjawiska nagrzewania się opon uznano optymalną siłę docisku osi wynoszącą 30 kN. Więcej przykładów, badań i wniosków dotyczących tego problemu zaprezentowano w pracy [A9].

Inną istotną kwestią w trakcie realizacji badań na hamowni podwoziowej dotyczącą funkcjonalności układów dwupaliwowych jest dobór parametru związanego ze strategią i szybkością narastania prędkości obrotowej silnika, czyli tzw. „wybiegu". Wpływa on bezpośrednio na obciążenie silnika i determinuje możliwość wytworzenia maksymalnego ciśnienia doładowania, co z kolei skorelowane jest z możliwością wygenerowania dużych wartości dawkowania przez wtryskiwacze ON. Brak pokrycia tego obszaru pracy silnika w trakcie „strojenia" może prowadzić do niepoprawnej pracy instalacji CDF np. w trakcie

ruszania pojazdem z ładunkiem. Zależność ciśnienia doładowania od prędkości „wybiegu” na hamowni podwoziowej zaprezentowano na rys. 28.



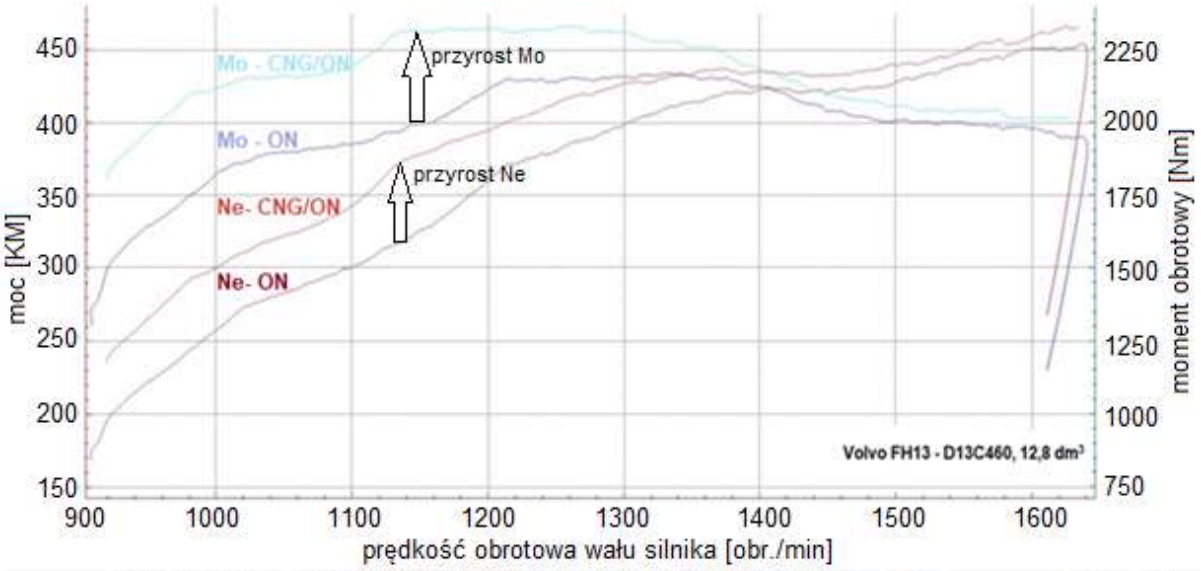
Rys. 28. Wpływ prędkości „wybiegu” na przebieg ciśnienia doładowania w funkcji prędkości obrotowej. Badania zrealizowano przy pełnym obciążeniu silnika

Powyższy wykres pokazuje, że kinetyka „wybiegu” 10 $\frac{\text{obr./min}}{sekunda}$ spełnia wymagania dotyczące ciśnienia doładowania. Hamowanie układu napędowego pojazdu przy niższych wartościach narastania przyspieszenia niepotrzebnie wydłuża test, co z kolei powoduje obciążenie termiczne opon i może prowadzić do intensywniejszego ich zużycia. Wyższe wartości nie pozwalają na osiągnięcie pełnego ciśnienia doładowania, co uniemożliwia pełne i poprawne „strojenie” instalacji.

Oba opisane aspekty podkreślają złożoność problemu badań przeprowadzanych na hamowni podwoziowej. Selekcja i szczegółowa analiza kluczowych elementów umożliwiających bezinwazyjne i precyzyjne wykonanie testów stanowią proces optymalizacyjny obejmujący wiele składowych. Należy zauważyć, że używana w eksperymentach hamownia nie może być wykorzystywana do badań homologacyjnych, a wyniki uzyskane przy jej pomocy powinny być traktowane jako porównawcze ze względu na to, że opierają się na zestawieniu danych wejściowych i wynikowych. Niemniej, w kontekście regulacji systemów zasilania dwupaliwowego takie podejście jest adekwatne i wystarczające. Ważne jest również uwzględnienie, iż niektóre błędy opisane w artykule [A9] mogą być skutecznie zniwelowane.

5.2.3. Efekty optymalizacji parametrów eksploatacyjnych pojazdu z badaną instalacją CDF w warunkach dynamicznych

Po zdefiniowaniu optymalnej strategii działania na hamowni, wypełniającej wymogi niezbędne do poprawnej regulacji opracowywanej instalacji CDF, wykonano „strojenie" układu dwupaliwowego na badanym pojeździe w warunkach pracy dynamicznej. Celem „strojenia" było uzyskanie stabilnej pracy przy jak najwyższym współczynniku zastąpienia paliw. Schemat stanowiska badawczego, na którym wykonano badania, zaprezentowano w poprzednim podrozdziale (rys. 25). Na rys. 29 przedstawiono zestawienie porównawcze charakterystyk zewnętrznych momentu obrotowego i mocy dla jednopaliwowego trybu pracy na ON oraz CDF przeprowadzonych na Volvo FH z silnikiem D13C460 po wykonanej regulacji instalacji dwupaliwowej.



paliwo	moc maksymalna przy obr./min	moment obrotowy maks. przy obr./min	czas testu	*n* wału	wart. korekcji	temp. tła	ciśnienie tła	wilg. tła	maks. *n* wału	typ testu	nr testu
ON	454 KM | 1637 obr./min	2165 Nm | 1334 obr./min	113 sek.	obliczony	0,0%	16°C	1024 hPa	16%	1641 obr./min	rampa	93
CNG/ON	466 KM | 1626 obr./min	2323 Nm | 1259 obr./min	112 sek.	obliczony	0,0%	17°C	1024 hPa	15%	1634 obr./min	rampa	98

Rys. 29. Widok z oprogramowania hamowni podwoziowej: zestawienie efektów „strojenia" dla trybu jednopaliwowego i CDF – wykresy zewnętrzne *Mo* i *Ne*

Na rys. 29 widać wyraźny wzrost momentu obrotowego i mocy maksymalnej w pełnym zakresie prędkości obrotowej silnika. Odnotowano ~7% wzrost momentu obrotowego w zakresie obrotów optymalnych silnika – co ma szczególnie istotny i pozytywny wpływ, z punktu widzenia praktycznego zwiększając dynamikę pojazdu. Zaobserwowane wzrosty maksymalne M_o i N_e przewyższają wartości fabryczne. W trakcie badań zidentyfikowano możliwość dalszego zwiększania tych wartości poprzez realizowanie dodatkowej dawki CNG funkcją korekty gazowej w oprogramowaniu sterownika instalacji. Obserwacja ta może się przełożyć np. na zaprogramowanie trybu „zwiększonej mocy"

użytecznej w momentach większego zapotrzebowania na siłę napędową pojazdu, np. w trakcie pokonywania stromych pojazdów. Należy mieć na uwadze, że zwiększanie w ten sposób dawki CNG zwiększa jednocześnie emisję CH_4.

5.2.4. Podsumowanie drugiego etapu badań własnych – hamownia podwoziowa

W ramach badań dotyczących diagnostyki i optymalizacji pracy pojazdów ciężarowych w kontekście retrofittingu w zasilanie dwupaliowe opracowano i zrealizowano projekt mobilnej hamowni podwoziowej. Urządzenie wpisuje się w potencjalne zapotrzebowanie rynkowe dotyczące modyfikacji w zasilanie dwupaliwowe, a jego mobilność dodaje benefitów w postaci możliwości „strojenia" instalacji wprost na parku maszynowym klienta, gdzie dokonać można weryfikacji poprawności montażu i strojenia układu dwupaliwowego. Projekt ten na tle typowych rozwiązań charakteryzuje się wprowadzeniem otwartej architektury oprogramowania, która zapewnia szerokie możliwości adaptacji do różnorodnych potrzeb badawczych. Proces weryfikacji urządzenia obejmował szczegółową analizę poprawności pomiarów, skupiając się na strategiach prawidłowego nieinwazyjnego badania pojazdów. W tym kontekście szczególną uwagę poświęcono kilku kluczowym aspektom: doborowi zakresu prędkości obrotowej silnika i odpowiednich przełożeń w skrzyni biegów, analizie wpływu wymuszonego obciążenia (docisku) osi napędowej pojazdu do rolek hamowni, poprawnej selekcji trybu pracy hamowni dostosowanego do specyficznych wymagań badawczych oraz doboru zakresu analizy danych dotyczących strat mechanicznych w układzie napędowym. Wykonano regulację nastaw opracowanej instalacji CDF wysokiej wymian paliw na hamowni podwoziowej w warunkach dynamicznych. Efektem „strojenia" było osiągnięcie istotnego przyrostu wartości momentu obrotowego i mocy w porównaniu z wartościami uzyskiwanymi w ramach standardowych pomiarów na ON oraz wartościami fabrycznymi. Wyniki te podkreślają potencjał zastosowania mobilnej hamowni podwoziowej do prac rozwojowych i komercyjnych, szczególnie w kontekście optymalizacji systemów napędowych pojazdów ciężarowych, co otwiera nowe perspektywy dla badań nad ogólnie pojętą efektywnością i redukcją emisji spalin.

W celu kompleksowego zbadania pojazdu oraz wpływu wykorzystania w nim badanej instalacji CDF, jak również określenia faktycznych wartość współczynnika zastąpienia paliw w zmiennych warunkach rzeczywistych przeprowadzono badania drogowe.

5.3. Badania drogowe

Testy drogowe, w przeciwieństwie do tych wykonywanych w kontrolowanych warunkach laboratoryjnych na hamowni silnikowej i podwoziowej, oferują unikalny wgląd w rzeczywiste działanie i problemy systemów dwupaliwowych. Realizacja badań drogowych umożliwia zidentyfikowanie subtelnych aspektów i niedoskonałości, które mogą pozostać niewykryte podczas analizy laboratoryjnej na hamowni silnikowej oraz podwoziowej. Badania drogowe pozwalają na obserwacje pracy pojazdu w złożonych i dynamicznie zmieniających się warunkach.

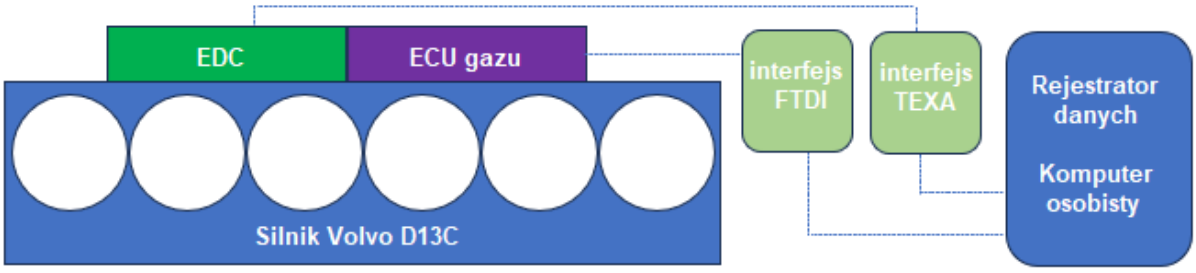
W tym podpunkcie przedstawiono opracowany program badań drogowych mających na celu eksperymentalną weryfikację poprawności działania badanej instalacji CDF w warunkach rzeczywistych. Następnie przeprowadzono analizę kluczowych wyników wyselekcjonowanych przez autora dysertacji. W dalszej części dokonano omówienia problemów technicznych zidentyfikowanych w trakcie realizacji badań (wraz z ich rozwiązaniami), które miały wpływ na efektywność i poprawność działania opisywanej instalacji CDF.

W podsumowaniu zestawiono tabelarycznie i omówiono najistotniejsze wyniki przeprowadzonych testów drogowych.

5.3.1. Program badań drogowych

Proces optymalizacji systemów dwupaliwowych dla efektywnej pracy w rozmaitych scenariuszach praktycznych wymaga przeprowadzenia rozbudowanych iteracyjnych cykli testowych. W ramach tych działań zastosowano metodykę wykonywania testów drogowych, które posłużyły do weryfikacji, identyfikacji oraz eliminacji wszelkich defektów w funkcjonowaniu opracowanego systemu CDF o dużym współczynniku zastąpienia paliw. Uwagę poświęcono również stabilności pracy silnika i instalacji zasilania gazowego oraz możliwości długotrwałego utrzymania osiągów bez negatywnego wpływu na niezawodność systemu.

W tym celu określono kluczowe parametry, które były niezbędne do oceny efektywności pracy instalacji. Dane te były rejestrowane bezpośrednio z elektronicznych jednostek sterujących EDC oraz ECU systemu gazowego, co zostało szczegółowo opisane w podrozdziale 5.1.1. Na rys. 30 przedstawiono schemat układu pomiarowego.



Rys. 30. Schemat ideowy układu pomiarowego użytego w badaniach drogowych

Na podstawie badań wstępnych pojazdu bez obciążenia podczas jazdy samym ciągnikiem bez naczepy stwierdzono, że układy pomiarowe oraz instalacja CDF umożliwiają prowadzenie badań zgodnie z założonym programem.

Testy drogowe przeprowadzono przy zastosowaniu różnych poziomów obciążenia, tj. 100%, 60% i 30% dopuszczalnej masy całkowitej (DMC), która wynosiła 40 000 kg dla zestawu pojazdów składającego się z ciągnika siodłowego i naczepy. Zilustrowano to na rys. 31.

Rys. 31. Zespół pojazdów wykorzystany w badaniach drogowych:
ciągnik siodłowy z badaną instalacją CDF oraz naczepa

W celu zapewnienia powtarzalności badań w poszczególnych etapach wytypowano trzy trasy drogowe, na których były wykonywane badania:

- trasa nr 1 – z 100% DMC pojazdu: WAT – Wiskitki – Radziejowice – WAT – o długości ~100 km, charakteryzująca się dużym natężeniem ruchu o dużej zmienności, gdzie

istniała uzasadniona konieczność częstej zmiany biegów, wpływająca na dużą zmienność warunków pracy silnika i instalacji CDF,

- trasa nr 2 – 60% DMC pojazdu: WAT – Płońsk – WAT – o długości ~150 km, charakteryzująca się średnim natężeniem ruchu, profilem terenu o dużej zmienności, co generowało częstą zmianę warunków pracy silnika i instalacji CDF,
- trasa nr 3 – 30% DMC pojazdu: WAT – Białystok – WAT – o długości ~400 km, charakteryzująca się małym natężeniem ruchu oraz możliwością długookresowej jazdy ze stałą prędkością zbliżoną do maksymalnej – 89 km/h.

Podczas badań każda trasa była powtarzalna kilkukrotnie (3–4 razy) w zbliżonych warunkach natężenia ruchu. Początek testu dla danej trasy był realizowany w tej samej porze dnia roboczego.

Zaprezentowany powyżej program badań drogowych wypełnia założenia testowania instalacji CDF w warunkach najbardziej zbliżonych do rzeczywistej pracy ciągnika siodłowego w warunkach zmiennego ruchu drogowego, różnej masy całkowitej pojazdu, zmiennego profilu drogi oraz długotrwałej pracy.

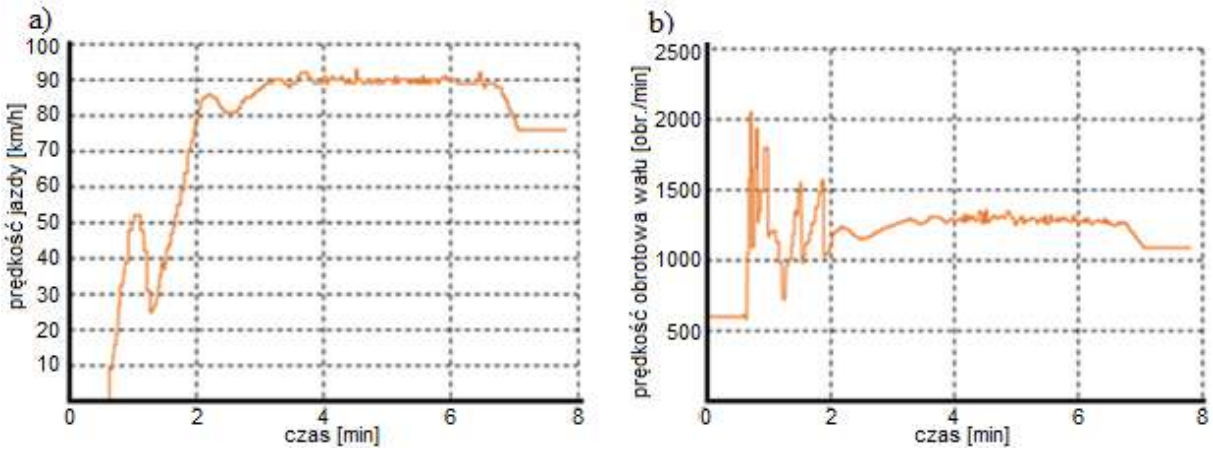
5.3.2. Analiza wybranych wyników testów drogowych

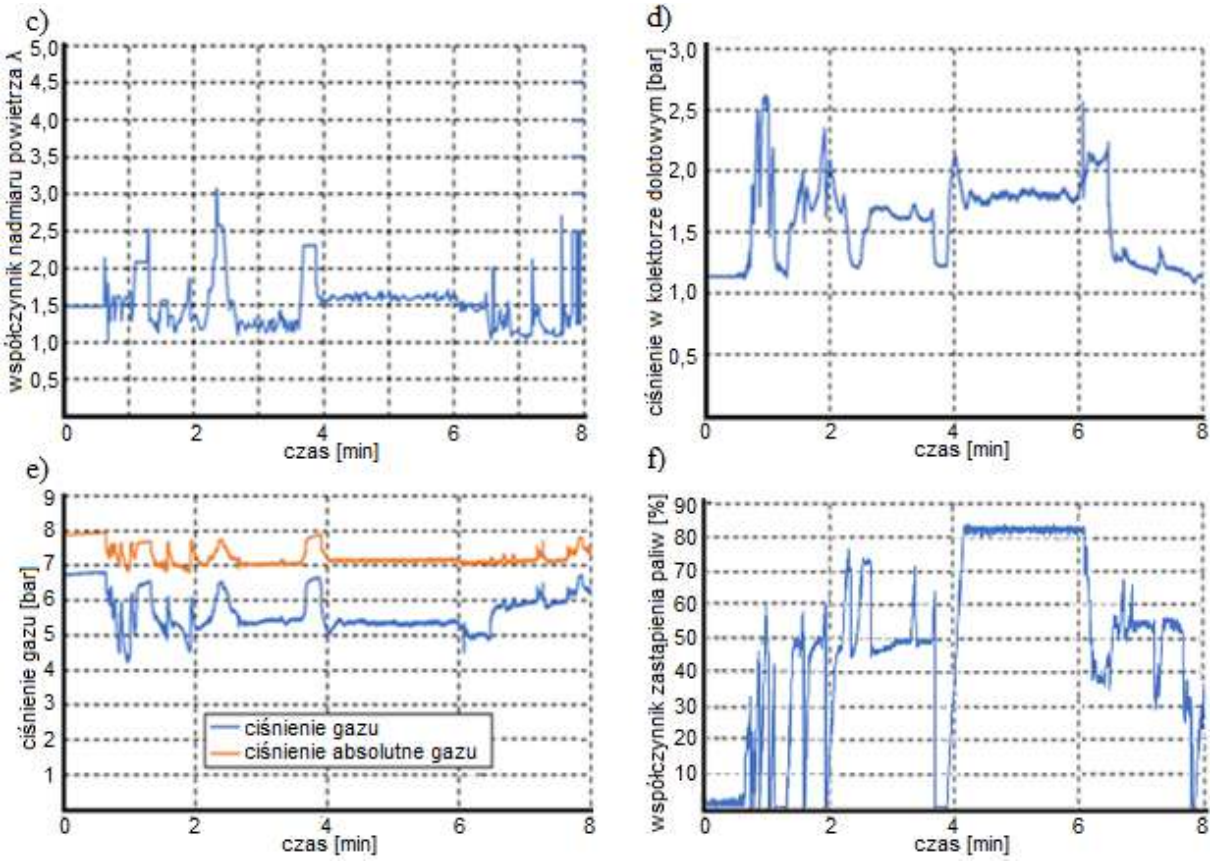
Po przeprowadzeniu badań drogowych przystąpiono do analizy uzyskanych danych. W prezentowanym fragmencie (rys. 32), dotyczącym ośmiominutowego odcinka z trasy Wiskitki – węzeł drogi krajowej (DK) nr 719 o długości 7,2 km, wykonanego podczas pierwszego przejazdu na tej trasie, skupiono uwagę na ocenie kluczowych parametrów pracy pojazdu oraz badanej instalacji do zasilania dwupaliwowego. Ten wybrany segment analizy, uznany przez autora za szczególnie interesujący i reprezentatywny, wytypowano na szczegółowe rozpatrzenie różnych stanów pracy instalacji dwupaliwowej.

Rys. 32. Analizowany odcinek testowy Wiskitki – węzeł DK nr 719 o długości 7,2 km – [źródło: Google Maps]

Na rys. 33 zaprezentowano wybrane wyniki badań. Analiza wyników skupia się na zestawieniu i ocenie parametrów rzeczywistej:

- prędkości jazdy (rys. 33a),
- prędkości obrotowej wału silnika (rys. 33b),
- współczynnika nadmiaru powietrza (rys. 33c),
- ciśnienia w kolektorze dolotowym (rys. 33d),
- ciśnienia CNG na wyjściu z reduktora ciśnienia (rys. 33e),
- współczynnika zastąpienia (rys. 33f).





Rys. 33. Wybrane wyniki badań drogowych z odcinka Wiskitki – węzeł DK nr 719 dla: a) prędkości jazdy, b) prędkość obrotowej wału silnika, c) współczynnika nadmiaru powietrza, d) ciśnienia w kolektorze dolotowym, e) ciśnienia CNG na wyjściu z reduktora ciśnienia, f) współczynnika zastąpienia

Na podstawie przeprowadzonych badań, w trakcie gwałtownego rozpędzania pojazdu na wzniesieniu (0–2 minuta) (rys. 33a) i jednocześnie uzyskiwania wysokich prędkości obrotowych silnika (rys. 33b) oraz wartości ciśnienia doładowania (rys. 33d), zaobserwowano spadek ciśnienia absolutnego gazu poniżej zakładanych konstrukcyjnie 5 barów (rys. 33e) oraz asynchronizację czasów otwarcia wtryskiwaczy gazowych, co ma swoje odzwierciedlenie we wskazaniach współczynnika nadmiaru powietrza λ (rys. 33c). W okresie tym, powiązanym z wysokim zapotrzebowaniem na M_o, zauważono również dużą zmienność *Wz* (rys. 33f), z największym zagęszczeniem w zakresie od 40–60%. Na rys. 33f można również zaobserwować częste zmiany biegów powiązane z koniecznością generowania jak największej siły napędowej. Każdorazowo zmiana biegów powoduje stopniowe zwiększanie *Wz*, co zmniejsza średnie wartości wymian. W okresie tym odnotowano również epizodyczne występowanie spalania detonacyjnego.

Na początku i końcu okresu od 2 do 4 minuty zaobserwowano kilkusekundowe hamowanie silnikiem pojazdu przy prędkości maksymalnej bez realizowanych wtrysków paliw, co skutkuje skokiem wartości wskaźnika λ (rys. 33c). Ponowne rozpoczęcie generowania mocy

silnika nie powoduje już tak wyraźnych spadków ciśnienia gazu jak w przypadku początkowego rozpędzania, co ma powiązanie z mniejszym zakresem zmiany prędkości obrotowej silnika (rys. 33b). W okresie tym można zauważyć częściowe ustabilizowanie współczynnika zastąpienia, co wiąże się z dłuższym czasem pomiędzy zmianami przełożeń w skrzyni biegów wraz ze zwiększaniem prędkości pojazdu. W trakcie tego okresu można również zauważyć, jak duże opory ruchu (60% DMC pojazdu), jazda na wzniesieniu oraz prędkości blisko maksymalnych (rys. 33a) wymuszają pracę instalacji w *Wz* ok. 50%.

Po osiągnieciu prędkości maksymalnej 89 km/h (rys. 33a) oraz wjechaniu na płaski odcinek drogi zaobserwowano wyraźne ustabilizowanie pracy pojazdu oraz instalacji CDF, powodujące zmniejszenie intensywność zmian *Wz*, co obrazuje okres od 6 do 8 minuty. W trakcie jednostajnej jazdy w zakresie obrotów optymalnych (rys. 33b) wskaźnik λ utrzymywał się na poziomie ok. 1,6. Ciśnienie absolutne gazu oscylowało w okolicach 5,5 bar, a instalacja CDF bez występowania spalania detonacyjnego realizowała *Wz* ok. 85%. Wskazuje to spełnienie wymagań konstrukcyjnych instalacji gazowej oraz wysoką sprawność jej działania w stanach ustalonych.

W okresie od 6 do 8 minuty można zaobserwować okresy częściowego wytracania prędkości pojazdu w warunkach niebędących procesem pełnego hamowania silnikiem, tj. w trakcie realizowania zasilania paliwami (ok. 6,5–7 min. testu) można w nim zauważyć nieuzasadnione zmniejszenie *Wz* do wartości ok. 60%. W czasie od 7 do 8 minuty testu pojazd poruszał się na nadbiegu (bieg 12 przekładni pojazdu) w sposób jednostajny w zakresie prędkości poniżej obrotów optymalnych, ok. 1050–1100 obr./min (rys. 33b). Zmniejszone zapotrzebowanie na siłę napędową w warunkach małego ciśnienia doładowania oraz poza zakresem obrotów optymalnych generowało nieuzasadnione zmiany w *Wz*.

Powyższa oraz szersza analiza wszystkich wyników badań drogowych wymusiła konieczność modyfikacji badanej instalacji CDF.

5.3.3. Rozpoznane i rozwiązane niedoskonałości w pracy instalacji CDF

W wyniku przeprowadzonych badań drogowych zidentyfikowano, a następnie rozwiązano problemy dotyczące:

- chwilowych spadków ciśnienia poniżej 5 bar oraz nierównomiernego rozkładu ciśnienia gazu w szynie zasilającej wtryskiwacze CNG, prowadzących w warunkach dużego obciążenia do niewydolności ilościowej układu – problem rozwiązano poprzez

wykorzystanie reduktora ciśnienia gazu o większej wydajności cieplnej i modyfikację listwy paliwa gazowego na możliwość jej zasilania z obu jej stron,

- niestabilnej pracy silnika podczas gwałtownej zmiany obciążenia oraz w trakcie częściowego hamowania silnikiem, czego efektem były duże gwałtowne zmiany współczynnika zastąpienia paliw i występowanie epizodycznego spalania detonacyjnego – wprowadzono aproksymację *Wz* między komórkami mapy zasilania gazowego w sterowniku, zmieniono wartości na mapie sterowania przepustnicą *bypassu* oraz strategię zmiany *Wz* po zidentyfikowaniu spalania detonacyjnego,
- występującej niestabilnej pracy silnika w momencie przełączania instalacji na pracę dwupaliwową – zaprogramowano funkcjonalność zmniejszania stopienia narastania *Wz* podczas przełączania instalacji trybów pracy instalacji CDF,
- współpracy instalacji CDF z ogranicznikiem prędkości jazdy (89 km/h) podczas zjazdu ze wzniesienia, gdy pojazd zwiększał prędkość ponad maksymalną (efekt szarpania pojazdu) – dokonano modyfikacji w logice sterowania wtryskami CNG,
- przegrzewania się sterownika układu zasilania CNG – co wymusiło zmianę jego miejsca montażu.

Wyżej wymienione modyfikacje i poprawki dotyczące poprawności działania instalacji CDF wykonywano w trakcie realizowania programu badań drogowych.

5.3.4. Podsumowanie trzeciego etapu badań własnych – badania drogowe

Na rys. 33 zaprezentowano docelową mapę współczynnika zastąpienia paliw w funkcji wyliczonej dawki ON i prędkości obrotowej silnika. Mapa ta została wykorzystana w ostatnich przejazdach testowych w ramach badań drogowych.

DOSE \ RPM	200	500	800	1000	1200	1400	1600	1800	2000
0.0	0	0	0	0	0	0	0	0	0
20.0	0	0	80	85	90	90	80	60	35
40.0	0	0	90	90	90	90	80	65	40
60.0	0	0	80	75	80	80	75	60	40
90.0	0	0	70	70	75	75	70	35	30
120.0	0	0	55	60	60	55	30	20	0
150.0	0	0	50	50	50	50	20	10	0
170.0	0	0	40	45	45	40	10	0	0
190.0	0	0	20	20	20	20	0	0	0
210.0	0	0	0	0	0	0	5	0	0

n - [obr./min]

dawka ON [mg]

Rys. 34. Widok z oprogramowania ECU gazu – docelowa mapa współczynnika zastąpienia paliw

W etapie prac dotyczących badań drogowych mających na celu eksperymentalną weryfikację poprawności działania badanej instalacji CDF w warunkach rzeczywistych opracowano i zrealizowano plan badań drogowych wypełniający założenia różnych scenariuszy użytkowania ciągnika siodłowego w dalekobieżnym transporcie drogowym. W trakcie realizacji badań oraz analizy wyników zidentyfikowano i wyeliminowano problemy w pracy pojazdu z opracowaną instalacją CDF. Intensyfikacja występowania problemów związana była w większości z pracą instalacji w stanach nieustalonych oraz w momentach generowania dużych wartości momentu obrotowego. Wyeliminowanie błędów pozwoliło na wypracowanie rozwiązań technicznych instalacji pracującej w sposób niepowodujący utrudnień w pracy silnika ciągnika siodłowego. W tabeli 9 zaprezentowano zbiorczo najważniejsze wyniki dotyczące zrealizowanych badań drogowych.

Tabela 9. Wyniki badań drogowych

Nr trasy	1		2		3	
Przebieg trasy	WAT – Płońsk – WAT		WAT – Wiskitki – Radziejowice – WAT		WAT – Białystok– WAT	
Natężenie ruchu	średnie		duże		małe	
Profil terenu	zróżnicowany		zróżnicowany		teren płaski	
Masa własna pojazdu	100% DMC ~40 ton		60% DMC ~26 ton		30% DMC ~16 ton	
	Wartości średnie dla przejazdu					
	prędkość [km/h]	*Wz* [%]	prędkość [km/h]	*Wz* [%]	prędkość [km/h]	*Wz* [%]
Przejazd 1	45,20	50,15	57,23	59,24	81,80	79,80
Przejazd 2	48,30	55,40	65,22	66,84	87,40	82,30
Przejazd 3	50,20	60,12	61,43	67,84	86,20	84,20
Przejazd 4	55,20	66,21	55,81	68,50	---	---
Średnia	**49,70**	**57,95**	**59,91**	**65,58**	**85,10**	**82,10**

W tabeli 9 przedstawiono średnie wartości prędkości jazdy i współczynnika zastąpienia paliw dla poszczególnych warunków jazdy. Można zaobserwować powiązanie zmniejszania średniego współczynnika zastąpienia paliw wraz ze wzrostem DMC pojazdu i zwiększania udziału jazdy niejednostajnej (wyniki dla trasy nr 1 i 2). W trakcie typowej pracy ciągnika siodłowego, tj. w trakcie jazdy autostradowej w okolicach prędkości maksymalnej pojazdu, faktyczny współczynnik zastąpienia omawianej instalacji dwupaliwowej wyniósł ponad 80%. W efekcie badań drogowych poprawiono działanie opracowanej instalacji CDF, zarówno pod względem operatywności pojazdu, jak i poprzez efektywne zwiększanie współczynnika zastąpienia paliw. Zaprezentowane w tym podrozdziale wyniki badań drogowych potwierdziły

empirycznie możliwość osiągnięcia zakładanej średniej wymiany paliw powyżej 50% w opracowanej instalacji dwupaliwowej spalania konwencjonalnego z wykorzystaniem CNG jako paliwa niskoreaktywnego i ON jako paliwa wysokoreaktywnego.

5.4. Wnioski z badań własnych

Prace nad instalacją CDF dla serii silników Volvo D13 wykazały:

- efektywność metody modyfikacji sygnałów prądowych sterowania wtryskiwaczy PD w celu osiągnięcia wysokich wartości współczynnika zastąpienia paliw w układach dwupaliwowych konwencjonalnego spalania (ponad 50%),
- sporadyczne wyłączenie z pracy wybranych wtryskiwaczy PD przy *Wz* na poziomie 95%,
- osiągnięcie rzeczywistych wysokich współczynników zastąpienia paliw w układach dwupaliwowych konwencjonalnego spalania ze zminimalizowanym efektem spalania detonacyjnego (zakres *Wz* od 5% do 90%),
- możliwość odtworzenia fabrycznych przebiegów momentu obrotowego (lub lepszych) przy wysokich wartościach współczynnika zastąpienia,
- zwiększanie współczynników zastąpienia w badanej instalacji CDF wykazało korelację w obniżeniu emisji CO_2 przy jednoczesnym zwiększeniu stężenia CH_4 w spalinach. Możliwa jest eliminacja problemu poprzez dodanie dedykowanego systemu oczyszczania spalin ze szkodliwych składników (np. reaktor katalityczny utleniający CH_4), zmianę faz rozrządu lub wykorzystanie do zasilania biopaliw z domieszkami wodoru. Konieczne są dalsze badania w tym zakresie,
- empirycznie potwierdzenie w warunkach badań drogowych odzwierciedlających faktyczną pracę pojazdu ciężarowego możliwość osiągnięcia współczynnika zastąpienia w zakresie ~56–82%,
- w przypadku opracowywania instalacji CDF dla danego typoszeregu silników (marki) niezbędne są badania w trzech etapach (tj. badania na hamowni silnikowej, badania na hamowni podwoziowej i badania drogowe). W przypadku komercyjnego montażu opracowanej instalacji CDF proces retrofittingu obejmuje tylko etapy pracy na hamowni silnikowej oraz testy drogowe.

6. Analiza rezultatów badań własnych w kontekście dalszych działań dotyczących przyszłości retrofittingu silników o ZS w RCCI oraz potencjalny transfer technologii

Na podstawie przeprowadzonych badań wykazano, że istnieje możliwość opracowania instalacji CDF o wysokim współczynniku zastąpienia paliw dla używanych pojazdów ciężarowych w celu poddania ich procesowi retrofittingu. Decyzja determinująca podjęcie badań nad tego typu systemem spalania uzasadniona była m.in. możliwościami jego dalszej komercjalizacji. W przeprowadzonej analizie stanu zagadnienia (podrozdział 2.4.) wykazano, że dla implementacji niefabrycznego systemu CDF wymagany jest mniejszy stopień ingerencji technicznej w silnik pojazdu, niż ma to miejsce w przypadku systemu RCCI, co w sposób bezpośredni wpływa na koszty związane z modyfikacją. Jednym z wniosków wyciągniętych w trakcie badań nad opracowywaną instalacją CDF było zauważenie zwiększonej emisji metanu wzrastającej wraz ze zwiększaniem współczynnika zastąpienia paliw. Emisja CH_4 deprecjonuje korzyści ekologiczne wynikające ze zmniejszonej emisji CO_2, a w praktyce może to utrudnić lub nawet całkowicie uniemożliwić instalacji przejście przez proces homologacyjny. Sam „przedmuch" metanu do układu wydechowego pogarsza również sprawność ogólną silnika. Zaproponowane rozwiązania w postaci dobrania odpowiedniego układu neutralizującego obecność metanu czy zmiana faz rozrządu dodatkowo zwiększają koszt potencjalnej modernizacji. Co więcej, wymiana np. wałka rozrządu na taki o innym rozstawieniu krzywek, to modyfikacja głęboko ingerująca w konstrukcję silnika i może negatywnie wpłynąć na możliwość jego pracy w trybie jednopaliwowym. Z kolei dedykowany system „obróbki" spalin skupiony na usuwaniu metanu dodatkowo może pogorszyć sprawność silnika.

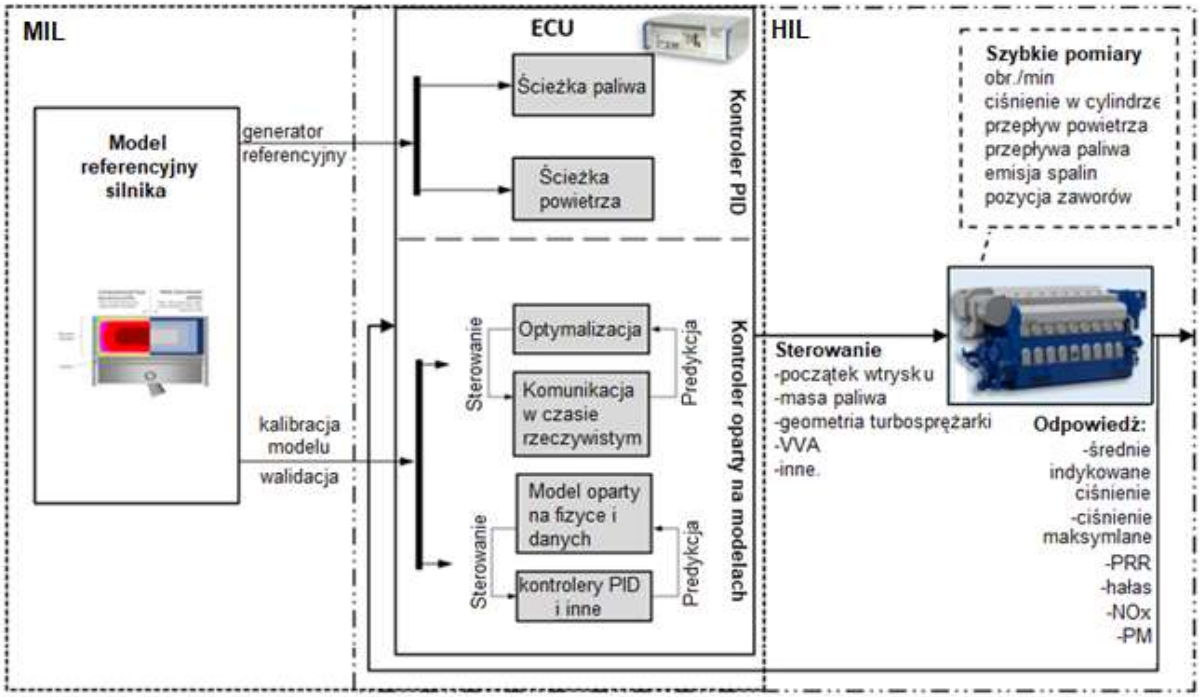
W związku z powyższymi problemami w tym rozdziale poddano analizie możliwość zaadaptowania wyników dotychczasowych prac, badań i analiz w kontekst wykorzystania ich dla retrofittingowych systemów RCCI. Samo niskotemperaturowe spalanie dwupaliwowe kontrolowane reaktywnością paliw dzięki wysokiej sprawności η >50% oraz niemal zerowej emisji NO_x i PM, opisane szerzej w podrozdziale 2.4.3., jest zdecydowanie bardziej perspektywicznym rozwiązaniem niż CDF i to pomimo zwiększonych kosztów początkowych. Wykazano również potencjał RCCI dotyczący redukcji CO_2 na tle rozwiązań konwencjonalnych, co opisano w rozdziale czwartym (rys. 15). Zdaniem autora dysertacji zwiększające się ograniczenia prawne dotyczące emisji spalin doprowadzą do kompromisu

kosztowo-ekologicznego, w którym retrofitting silników w RCCI może okazać się atrakcyjny ekonomicznie i zostać ostatecznie skomercjalizowany. Dzieje się to aktualnie w kontekście silników dla jednostek pływających [128]. Słuszne więc są dalsze prace dotyczące istotnych aspektów konstrukcyjnych i badawczych w kontekście ogólnie pojętego LTC, jak i samego RCCI [94].

Niezależna kontrola pracy zaworów silnika umożliwia optymalizację momentu i czasu trwania wtrysku paliwa, co jest szczególnie ważne w technologii RCCI, gdzie różne paliwa wymagają różnych czasów na stworzenie optymalnej mieszanki przed zapłonem. Pozwala również na kontrolę przepływu mieszanki paliwowo-powietrznej i spalin w zależności od obciążenia silnika i jego prędkości obrotowej, co przekłada się na lepszą efektywność i elastyczność działania silnika w warunkach niestabilnych i przejściowych. Dodatkowo VVA umożliwia zmianę efektywnego stopnia sprężania lub zastosowanie tzw. „ujemnego współotwarcia" zaworów dolotowego i wydechowego realizujących strategię EGR [91]. Wszystko to ma na celu sprawienie, by LTC odbywało się możliwie efektywnie i w jak najszerszym zakresie operatywnym silnika. Problemy RCCI dotyczące trudności samozapłonu przy niskim obciążeniu oraz zbyt dużych wartości PRR można wyeliminować również poprzez mieszane tryby pracy z CDF [129]. W przypadku niskich lub bardzo wysokich obciążeń silnik może przechodzić w tryb pracy zmiennej, mieszanej. Wykorzystanie trybów mieszanych i użycia VVA może zminimalizować efekty niepożądanej emisji CH_4. Dodatkowo jeśli dla danego typu silnika ograniczyć i zdefiniować zakres wykorzystywanych w nim paliw, samo VVA można zrealizować nie przez drogi i niespopularyzowany system pełnej niezależnej kontroli zaworów (EVHA – *Electro-Hydraulic Variable Valve Actuator*), a przez popularne systemy z napędem mechanicznym, analogiczne do takich jak Valvetronic [130] czy VVT-i [131], zyskujące popularność również w silnikach pojazdów ciężarowych. Może to stanowić realny do osiągnięcia kompromis technologiczno-kosztowy niezbędny do procesu retrofittingu.

W badaniach naukowych i opracowaniach dotyczących efektywnej realizacji RCCI często szczególny nacisk kładziony jest na problematykę sterowania. W przeciwieństwie do CDF, w którym najczęściej sterowanie odbywa się na zasadzie współpracy równoległej EDC i ECU instalacji gazowej, dla realizacji RCCI wymagane jest dużo bardziej kompleksowe podejście. Zobrazowanie tej problematyki zaprezentowano w pracy [129], gdzie omówiono techniki sterowania dla RCCI, m.in. wykorzystujące spalanie wielomodowe, ze szczególnym uwzględnieniem warunków przejściowych pracy silnika. Jednym z najczęściej proponowanych rozwiązań dla RCCI jest system sterowania z wykorzystaniem modeli referencyjnych opartych

na fizyce, zorientowanych na kontrolę (ang. *Model Based Control System*, MBCS). W publikacji [98] autorzy skupili się na podejściach quasi-wymiarowych i kinetycznych, podkreślając ich wysoką przewidywalność w takich aspektach jak szczegółowy opis składu mieszaniny, niejednorodności termicznej i przestrzennej oraz turbulencji w komorze spalania. Wykazano także iteracyjne ścieżki redukcji z tych modeli w kierunku zastosowań dla sterowania RCCI w czasie rzeczywistym oraz wykorzystanie zunifikowanych metodologii pracy modeli w pętli zamkniętej (ang. *Model-in-the-Loop,* MIL) i pracy silnika w pętli zamkniętej (ang. *Hardware-in-the-Loop,* HIL). Na rys. 35 zaprezentowano schemat ideowy działania MBCS:



Rys. 35. Ideowy schemat działania systemu sterowania RCCI z wykorzystaniem modeli referencyjnych zorientowanych na sterowanie [na bazie 132]

Opracowanie modeli referencyjnych dla systemów sterowania RCCI to kompleksowe zagadnienie, które wymaga szczególnego potraktowania. Jako istotne wymienia się kwestie takie jak:

- precyzyjne pomiary i analiza ciśnienia spalania w czasie rzeczywistym,
- odpowiednio zaprojektowane mechanizmy pre- i postprzetwarzania danych pomiarowych.

Bez wątpienia do stworzenia precyzyjnych modeli referencyjnych niezbędne będą kompleksowe badania laboratoryjne z wykorzystaniem stanowisk silnikowych i podwoziowych. Opracowana w ramach działań dotyczących podjętego tematu doktoratu

hamownia podwoziowa, dzięki w pełni otwartej architekturze, może zostać z powodzeniem wykorzystana do prac dotyczących modeli referencyjnych niezbędnych do efektywnego sterowania systemami RCCI.

Zaprezentowane w pracy technologie retrofittingu silników o zapłonie samoczynnym mogą znaleźć zastosowanie w wielu obszarach poza sektorem pojazdów ciężarowych, otwierając nowe możliwości dla ekologicznego zasilania dwupaliwowego. Jednym z potencjalnych zastosowań jest aplikacja w pojazdach takich jak sprzęt rolniczy i budowlany, który jest predysponowany do wykorzystania tej technologii ze względu na swoje specyficzne wymagania operacyjne. Ponadto technologia ta może zostać użyta w generatorach kogeneracyjnych prądu i ciepła, co pozwoli na efektywniejsze wykorzystanie zasobów energetycznych. Stabilizacja sieci elektroenergetycznych jest kolejnym obszarem, w którym retrofitting silników o ZS może przynieść korzyści. Dzięki swojej zdolności do szybkiego reagowania na zmiany w zapotrzebowaniu na energię w sieci elektrycznej generatory wykorzystujące silniki dwupaliwowe mogą stanowić filar w systemie elektroenergetycznym oraz w ekologiczny i synergiczny sposób wspierać OŹE. W kontekście zdecentralizowanych sieci energetycznych zmodyfikowane silniki o ZS mogą służyć jako lokalne źródła zasilania, zdolne do spalania wielu rodzajów paliw kopalnych i ekologicznych, co zapewniłoby energię w miejscach oddalonych od centralnych źródeł produkcji bądź pogrążonych w kryzysie wynikającym z wojny lub katastrofy naturalnej. Transport wodny i kolejowy to kolejne sektory, w których technologia retrofittingu może znaleźć zastosowanie. Zasadniczo wszystkie wysoko energochłonne gałęzie gospodarki, które obecnie korzystają z silnika o ZS, są punktem zainteresowania i potencjalnym miejscem wdrożenia nowoczesnych ekologicznych technologii dwupaliwowych.

7. Wnioski końcowe

Podsumowując przeprowadzone prace, analizy oraz badania, można wyszczególnić następujące kluczowe wnioski:

1. Analiza rynku
   - retrofitting silników o zapłonie samoczynnym w pojazdach ciężarowych do zasilania dwupaliwowego może stanowić jedną z możliwości realizacji strategii ekologicznych,
   - używane pojazdy ciężarowe dostępne na rynku są technologicznie przygotowane do procesu retrofittingu na nowoczesne zasilanie dwupaliwowe,
   - instalacje dwupaliwowe do zasilania gazowego charakteryzują się wyższymi poziomami bezpieczeństwa eksploatacji w porównaniu z rozwiązaniami konwencjonalnymi oraz pojazdami BEV,
   - pojazdy dwupaliwowe wykazują przewagę konkurencyjną nad pojazdami BEV i FCEV w transporcie dalekobieżnym,
   - technologie dwupaliwowe mogą wypełnić powstającą lukę rynkową dla firm rynku instalacji gazowych.

2. Potencjał paliw niskoemisyjnych do przyszłych zastosowań dwupaliwowych:
   - wyniki analiz wskazują na wysoki potencjał biopaliw z domieszkami wodoru, takich jak:
     – HVO (Hydrotreated Vegetable Oil) – jako paliwo wysokoreaktywne,
     – HCNG (Hydrogen Compressed Natural Gas) – jako paliwo niskoreaktywne,
   - paliwa te wpisują się w strategie dekarbonizacji, redukując problem przechowywania nadwyżek energii elektrycznej generowanej z OŹE oraz kwestie związane z magazynowaniem wodoru. Obecnie HCNG nie jest jeszcze szeroko komercjalizowane, HVO jest już dostępnym ekologicznym substytutem ON w kilku krajach UE.

3. Badania własne nad instalacją CDF CNG/ON dla serii silników Volvo D13:
   - wykazano możliwość średniej rzeczywistej wymiany paliw powyżej 50% w pełnym zakresie obciążeń oraz do 90% chwilowego zastąpienia. Możliwe jest osiągnięcie fabrycznych lub lepszych wartości momentu obrotowego podczas pracy dwupaliwowej,
   - zaobserwowano obniżenie emisji CO_2 przy jednoczesnym wzroście stężenia CH_4 w gazach spalinowych,

- konieczne jest kontynuowanie prac mających na celu neutralizację CH_4 w gazach spalinowych, m.in. poprzez modyfikację systemu oczyszczania spalin pojazdu,
- empirycznie potwierdzono, na podstawie badań drogowych, możliwość osiągnięcia rzeczywistego współczynnika zastąpienia paliw w zakresie 56–82%.

4. Przyszłość retrofittingu w zasilanie dwupaliwowe:
   - Konieczne jest skupienie się na realizacji spalania RCCI w związku z wyższą sprawnością ogólną silnika η >50%, niską emisją CO_2 i niemal zerową emisją NO_x i PM,
   - efektywne rynkowo może być wykorzystanie technologii mieszanych trybów spalania CDF i RCCI,
   - w celu efektywnej aplikacji RCCI konieczne jest poświęcenie się pracom badawczym dotyczącym procesu sterowania pracą silnika, z wykorzystaniem modeli referencyjnych skupionych na sterowaniu. Niezbędne w tym celu będą otwarte narzędzia diagnostyczno-badawcze, takie jak np. zaprezentowana w pracy hamownia z otwartą architekturą systemu operacyjnego.

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Dotychczasowy dorobek doktoranta legł u podstaw sformułowanego osiągnięcia naukowego w postaci cyklu powiązanych tematycznie publikacji naukowych będących podstawą Rozprawy doktorskiej zatytułowanej: „Retrofitting samochodowych silników o ZS w ekologiczne zasilanie dwupaliwowe".

Cykl publikacji podsumowujących przeprowadzone prace obejmuje: dziesięć artykułów opublikowanych w cenionych czasopismach naukowych o zasięgu międzynarodowym oraz dwa recenzowane materiały zaprezentowane na międzynarodowej konferencji naukowej. Sumaryczna liczba uzyskanych punktów MEiN to 930, artykuły zacytowano 57 razy oraz przeczytano łącznie 1602 razy. Całkowity sumaryczny *Impact Factor* cyklu publikacji przedkładanych jako osiągnięcie naukowe wynosi IF=12,8 (stan na 06.05.2024).

Do cyklu stanowiącego Rozprawę Doktorską wliczono następujące publikacje:

Nr	**Autorzy:**	**Tytuł:**	**Wydawnictwo:**	**Rok wydania**	**pkt MEiN**
A1	Chojnowski, J.; Nogas, P.	*Potencjał HVO jako biopaliwa wysokoreaktywnego w dwupaliwowych układach zasilania.*	Biuletyn WAT	2021	20
A2	Chojnowski, J.; Karczewski, M. Szamrej, G.;	*A Review of Low-CO2 Emission Fuels for a Dual-Fuel RCCI Engine*	Energies	2021	140
A3	Chojnowski, J.	*Safety in the use of car gas fuel installations*	Combustion Engines	2021	70
A4	Chojnowski, J.; Karczewski, M..	*Analysis of the market structure of long-distance transport vehicles in the context of retrofitting diesel engines with modern dual-fuel systems*	Combustion Engines	2021	70
A5	Chojnowski, J.; Karczewski, M.; Szamrej, G.	*The phenomenon of knocking combustion and the impact on the fuel exchange and the output parameters of the diesel engine operating in the dual-fuel mode (DIESEL-CNG),*	NAŠE MORE	2021	*
A6	Chojnowski, J.; Karczewski, M.; Szamrej, G.	*Solutions of CNG and LNG supply systems in modern land and marine CI engines working in dual-fuel (NG – DIESEL) mode*	NAŠE MORE	2021	*
A7	Chojnowski, J.; Karczewski, M.; Szamrej, G.	*A review of technical solutions for RCCI engines*	Combustion Engines	2021	70
A8	Chojnowski, J.; Karczewski, M.; Szamrej, G.	*Experimental assessment of the impact of replacing diesel fuel with CNG on the concentration of harmful substances in exhaust gases in a dual-fuel diesel engine.*	Energies	2022	140
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A10	Chojnowski, J.	*The concept of a mobile automated diagnostic and dynamometer station for heavy trucks.*	Combustion Engines	2023	70
A11	Chojnowski, J.; Karczewski, M.; Szamrej, G.	*Dual-fuel engines using hydrogen-enriched fuels as an ecological source of energy for industry and energy*	Combustion Engines	2024	70
A12	Chojnowski, J.; Dziubak, T.	*Multi-criteria analysis of semi-trucks with conventional and eco-drives on the EU market*	Energies	2024	140

*recenzowany materiał konferencyjny

Powyższe prace stanowią jednotematyczny cykl publikacji pt. „Retrofitting samochodowych silników o ZS w zasilanie dwupaliwowe". Poniżej zaprezentowano podział procentowy udziału współautorów w poszczególnych publikacjach oraz wyszczególniono obszary wykonawcze autora. Następnie zaprezentowano treść publikacji.

Oświadczenie o procentowym udziale współautorów w publikacjach naukowych

[A1] *Potencjał HVO jako biopaliwa wysokoreaktywnego w dwupaliwowych układach zasilania* Biuletyn WAT Biuletyn Vol. LXX, Nr 2, 2021 https://doi.org/10.5604/01.3001.0015.7011 (opublikowany 30 czerwca 2021) 20 pkt MEiN

Nazwisko i imię współautorów	*Udział procentowy*	*Podpis poświadczający*
Chojnowski Janusz	70 %	Janusz Chojnowski
Nogas Patrycja	30 %	Patrycja Nogas

Udział doktoranta w powstaniu publikacji obejmował:

- sformułowanie celów skupiających się na ocenie hydrorafinowanego oleju roślinnego jako biopaliwa wysokoreaktywnego w układach dwupaliwowych,
- kwerenda i analiza danych,
- pozyskanie próbek badawczych paliwa,
- współudział w badaniach laboratoryjnych,
- główny udział w opracowaniu wersji roboczej i końcowej artykułu od strony merytorycznej, edytorskiej i graficznej,
- wypracowanie korekt i odpowiedzi wynikających z procesu recenzji.

[A2] *A Review of Low-CO2 Emission Fuels for a Dual-Fuel RCCI Engine.* Energies 2021, 14, 5067. https://doi.org/10.3390/en14165067 (opublikowany 17 sierpnia 2021) 140 pkt MEiN

Nazwisko i imię współautorów	*Udział procentowy*	*Podpis poświadczający*
Chojnowski Janusz	35 %	Janusz Chojnowski
Szamrej Grzegorz	40 %	Grzegorz Szamrej
Karczewski Mirosław	25 %	

Udział doktoranta w powstaniu publikacji obejmował:

- opracowanie części artykułu dotyczącej paliw ciekłych, w szczególności HVO,
- udział w opracowaniu metodologii doboru paliw ciekłych do silników dwupaliwowych,
- udział w opracowaniu wersji roboczej i końcowej artykułu od strony językowej, edytorskiej i graficznej,
- kwerenda literatury w obszarze paliw ciekłych,
- weryfikacja danych literaturowych.

[A3] *Safety in the use of car gas fuel installations. Combustion Engines.* 2022, 189(2), 3-9. https://doi.org/10.19206/CE-142172 (opublikowany 17 września 2021) 70 pkt MEiN

Nazwisko i imię autora	*Udział procentowy*	*Podpis poświadczający*
Chojnowski Janusz	100%	Janusz Chojnowski

[A4] *Analysis of the market structure of long-distance transport vehicles in the context of retrofitting diesel engines with modern dual-fuel systems*. Combustion Engines. 2022, 188(1), 13-17. https://doi.org/10.19206/CE-142167 (opublikowany 17 września 2021) 70 pkt MEiN

Nazwisko i imię współautorów	*Udział procentowy*	*Podpis poświadczający*
Chojnowski Janusz	80 %	Janusz Chojnowski
Karczewski Mirosław	20 %	

Udział doktoranta w powstaniu publikacji obejmował:

- sformułowanie celów skupiających się na analizie rynku pojazdów ciężarowych dalekobieżnych w kontekście adaptacji do zasilania dwupaliwowego,
- kwerenda i analiza danych,

- opracowanie metodyki porównania,
- główny udział w opracowaniu wersji roboczej i końcowej artykułu od strony merytorycznej, edytorskiej i graficznej,
- wypracowanie korekt i odpowiedzi wynikających z procesu recenzji.

[A5] *The phenomenon of knocking combustion and the impact on the fuel exchange and the output parameters of the diesel engine operating in the dual-fuel mode (Diesel-CNG)*
Recenzowany materiał konferencyjny: II Międzynarodowa konferencja nauki i technologii morskiej "Naše more" 2021, Dubrownik, Chorwacja 17 – 18 września, 2021 https://www.nasemore.com/wp-content/uploads/2024/04/NASE%20MORE%202021_Conference%20proceedings.pdf

Nazwisko i imię współautorów	*Udział procentowy*	*Podpis poświadczający*
Chojnowski Janusz	60 %	Janusz Chojnowski
Szamrej Grzegorz	10 %	Grzegorz Szamrej
Karczewski Mirosław	30 %	

Udział doktoranta w powstaniu publikacji obejmował:

- sformułowaniem głównych założeń artykułu,
- współudział w badaniach badań wpływy stopnia zastąpienia ON/CNG na wstępowania spalania detonacyjnego w badanym układzie dwupaliwowym,
- udział w opracowaniu i badaniach systemu minimalizującego efekty spalania detonacyjnego,
- współopracowanie wyników badań,
- główny udział w opracowaniu wersji roboczej i końcowej artykułu od strony językowej, edytorskiej i graficznej,
- opracowanie korekt i odpowiedzi wynikających z procesu recenzji,
- prezentacja treści w ramach konferencji naukowej.

[A6] *Solution of CNG and LNG supply systems in modern land and marine CI engines working in dual-fuel (NG-Diesel Mode)*

Recenzowany materiał konferencyjny: II Międzynarodowa konferencja nauki i technologii morskiej „Naše more" 2021, Dubrownik, Chorwacja, 17 – 18 września, 2021 https://www.na-semore.com/wp-content/uploads/2024/04/NASE%20MORE%202021_Conference%20proce-edings.pdf

Nazwisko i imię współautorów	*Udział procentowy*	*Podpis poświadczający*
Chojnowski Janusz	30 %	
Szamrej Grzegorz	45 %	
Karczewski Mirosław	25 %	

Udział doktoranta w powstaniu publikacji obejmował:

- weryfikacja danych literaturowych,
- udział w opracowaniu wersji roboczej i końcowej artykułu od strony merytorycznej, edytorskiej i graficznej,
- prezentacja treści w ramach konferencji naukowej.

[A7] *A review of technical solutions for RCCI engines.* Combustion Engines. 2022,189(2), 36-46. https://doi.org/10.19206/CE-142551 (opublikowany 25 września 2021) 70 pkt MEiN

Nazwisko i imię współautorów	*Udział procentowy*	*Podpis poświadczający*
Chojnowski Janusz	35 %	
Szamrej Grzegorz	45 %	
Karczewski Mirosław	20 %	

Udział doktoranta w powstaniu publikacji obejmował:

- weryfikacja danych literaturowych,
- główny udział w opracowaniu wersji roboczej i końcowej artykułu od strony merytorycznej, edytorskiej i graficznej,
- prezentacja treści w ramach konferencji naukowej.

[A8] *Experimental Assessment of the Impact of Replacing Diesel Fuel with CNG on the Concentration of Harmful Substances in Exhaust Gases in a Dual Fuel Diesel Engine.* Energies 2022, 15, 4563. https://doi.org/10.3390/en15134563 (opublikowany 22 czerwca 2022) 140 pkt MEiN

Nazwisko i imię współautorów	*Udział procentowy*	*Podpis poświadczający*
Chojnowski Janusz	20 %	Janusz Chojnowski
Szamrej Grzegorz	20 %	Grzegorz Szamrej
Karczewski Mirosław	60 %	

Udział doktoranta w powstaniu publikacji obejmował:

- współuczestnictwo w realizacji badań emisji składników spalin na hamowni podwoziowej,
- obsługa aparatury pomiarowej,
- współudział w opracowaniu wyników oraz wersji roboczej i końcowej artykułu od strony merytorycznej, edytorskiej i graficznej,
- wypracowanie korekt i odpowiedzi wynikających z procesu recenzji.

[A9] *Influence of the Working Parameters of the Chassis Dynamometer on the Assessment of Tuning of Dual-Fuel Systems.* Energies 2022, 15, 4869. https://doi.org/10.3390/en15134869 (opublikowany 02 lipca 2022) 140 pkt MEiN

Nazwisko i imię współautorów	*Udział procentowy*	*Podpis poświadczający*
Chojnowski Janusz	50 %	Janusz Chojnowski
Karczewski Mirosław	50 %	

Udział doktoranta w powstaniu publikacji obejmował:

- analiza literatury w obszarze wykorzystywania hamowni podwoziowych w badaniach,
- współrealizacja badań na hamowni podwoziowej dotyczących bezinwazyjnego testów i „strojenia" pojazdów wyposażonych w instalacje dwupaliwowe,
- opracowanie wyników badań – w obszarze wyników parametrów roboczych pojazdu – moc użyteczna, prędkość obrotowa, starty mechaniczne,
- główny udział w opracowaniu wersji roboczej i końcowej artykułu od strony merytorycznej, edytorskiej i graficznej,

- opracowanie korekty i współudział w opracowaniu odpowiedzi wynikających z procesu recenzji.

[A10] *The concept of a mobile automated diagnostic and dynamometer station for heavy trucks.* Combustion Engines. 2023;193(2):122-128. https://doi.org/10.19206/CE-163320 (opublikowany 30 kwietnia 2023) 70 pkt MEiN

Nazwisko i imię autora	*Udział procentowy*	*Podpis poświadczający*
Chojnowski Janusz	100%	Janusz Chojnowski

[A11] *Dual-fuel engines using hydrogen-enriched fuels as an ecological source of energy for transport, industry and power engineering.*
Combustion Engines. https://doi.org/10.19206/CE-176800 (opublikowany 19.01.2024) 70 pkt MEiN

Nazwisko i imię współautorów	*Udział procentowy*	*Podpis poświadczający*
Chojnowski Janusz	40 %	Janusz Chojnowski
Szamrej Grzegorz	50 %	Grzegorz Szamrej
Karczewski Mirosław	10 %	

Udział doktoranta w powstaniu publikacji obejmował:

- współudział w kwerendzie literatury,
- opracowanie wstępu teoretycznego,
- pozyskanie części danych wejściowych do obliczeń symulacyjnych,
- opracowanie części artykułu dotyczącej sprawności silników spalinowych,
- udział w opracowaniu wersji roboczej i końcowej artykułu od strony, edytorskiej, graficznej i językowej,
- wypracowanie korekt i odpowiedzi wynikających z procesu recenzji.

[A12] *Multi-criteria analysis of semi-trucks with conventional and eco-drives on the EU market*
Energies. 2024; 17(5):1018. https://doi.org/10.3390/en17051018 (opublikowany 21.02.2024) 140 pkt MEiN

Nazwisko i imię współautorów	*Udział procentowy*	*Podpis poświadczający*
Chojnowski Janusz	75%	Janusz Chojnowski
Dziubak Tadeusz	25%	T Dziubak

Udział doktoranta w powstaniu publikacji obejmował:

- sformułowanie głównych celów skupiających się na multikryterialnej analizie atrakcyjności pojazdów ciężarowych dalekobieżnych ze względu na ich napęd: konwencjonalny, elektryczny i dwupaliwowy,
- kwerenda i analiza danych,
- opracowanie metodyki i modelu porównawczego,
- główny udział w opracowaniu wersji roboczej i końcowej artykułu od strony merytorycznej, edytorskiej i graficznej
- wypracowanie korekt i odpowiedzi wynikających z procesu recenzji.

[A1]

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Potencjał hydrorafinowanego oleju roślinnego (ang. HVO — Hydrotreated Vegetable Oil) jako biopaliwa wysokoreaktywnego w dwupaliwowych układach zasilania

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Streszczenie. W czasach intensywnych badań dotyczących zmniejszenia emisji gazów cieplarnianych do atmosfery z sektora motoryzacyjnego inżynierowie i badacze znajdują efektywne rozwiązania umożliwiające osiągnięcie ten cel. Emisję dwutlenku węgla można zmniejszyć poprzez zwiększenie sprawności silnika spalinowego lub zmniejszenie zawartości węgla w spalanym paliwie. Połączeniem obu tych założeń może być wykorzystanie nowoczesnych układów zasilania dwupaliwowego do silników o zapłonie samoczynnym (ZS), w których dawką pilotażową paliwa wysokoreaktywnego jest hydrorafinowany olej roślinny (HVO) będący ekologicznym substytutem oleju napędowego. Temat wykorzystania HVO jako paliwa pilotażowego jest niezbadany empirycznie, jednak informacje dostępne na jego temat wskazują na zasadność takich zastosowań.
Słowa kluczowe: hydrorafinowany olej roślinny (HVO — Hydrotreated Vegetable Oil), CO_2, zasilanie dwupaliwowe, zapłon samoczynny o kontrolowanej reaktywności (RCCI — Reactivity Controlled Compression Ignition), gaz cieplarniany (GHG — Greenhouse Gas)
DOI: 10.5604/01.3001.0015.7011

Wstęp

W spalaniu paliw do silników tłokowych kluczową rolę odgrywają trzy pierwiastki: wodór — H, węgiel — C, tlen — O. Do konwencjonalnego silnika spalinowego nie dostarcza się wodoru ani węgla w czystej postaci — atomy zawarte są w paliwie, dopiero na nim silnik jest w stanie wykonywać pracę. Węgiel w swojej postaci pierwiastkowej jest paliwem stałym, natomiast wodór to gaz o bardzo małej gęstości.

Ze względu na szkodliwe działanie dwutlenku węgla i to, że zalicza się go do grupy gazów cieplarnianych (ang. *Greenhouse Gas – GHG*), jego emisja w procesie spalania paliw jest wysoce niepożądana. Z tego względu inżynierowie i badacze kładą nacisk na minimalizację emisji tego składnika spalin. Aby to zrobić, stosuje się paliwa o najniższym stosunku węgla do wodoru. Ze względu na trend systematycznego zmniejszania dopuszczalnej emisji GHG wodór zawarty w wiązaniach chemicznych paliwa jest najbardziej pożądanym pierwiastkiem jako materiał budulcowy paliwa. Zmniejszenie emisji dwutlenku węgla może zostać osiągnięte poprzez zwiększenie sprawności silnika spalinowego lub zmniejszenie zawartości węgla w spalanym paliwie [4]. Rozsądne wykorzystanie paliw alternatywnych pozwoli na redukcję emisji szkodliwych składników spalin poprzez zastosowanie obu tych czynników oraz usprawni działanie silników spalinowych obecnych już na rynku — dzięki procesom *retrofitingu* [6, 9].

2. Właściwości HVO

Hydrorafinowany olej roślinny (ang. *Hydrotreated Vegetable Oil – HVO*) to forma syntetycznego, alternatywnego, odnawialnego paliwa do silników o zapłonie samoczynnym, został on wyprodukowany z tłuszczów i olejów roślinnych. Jest otrzymywany w wyniku hydrokrakingu lub uwodornienia oleju roślinnego [15]. Hydrokraking rozbija duże cząsteczki na mniejsze przy użyciu wodoru [14], podczas gdy uwodornienie dodaje wodór do cząsteczek [8]. Metody te można wykorzystać do tworzenia substytutów benzyny, oleju napędowego, propanu, nafty i innych surowców chemicznych [7]. W produkcji HVO wodór jest substratem procesu, gdzie analogicznie w produkcji biodiesla FAME wykorzystuje się metanol [7], katalizatorem dla obu tych procesów jest KOH.

Dzięki temu HVO charakteryzuje się dużo większą stabilnością chemiczną w porównaniu do typowych biopaliw [2]. Przeprowadzono badania dla próbek paliwa (rys. 1):

A — olej napędowy bez dodatku estrów metylowych oleju rzepakowego (ON bez RME),

B — hydrorafinowany olej roślinny (HVO),

C — estry metylowe oleju rzepakowego (RME).

Przed przystąpieniem do badania każdą próbkę paliwa przefiltrowano przez filtr membranowy 8×10^{-6} m w celu usunięcia cząstek zanieczyszczeń stałych.

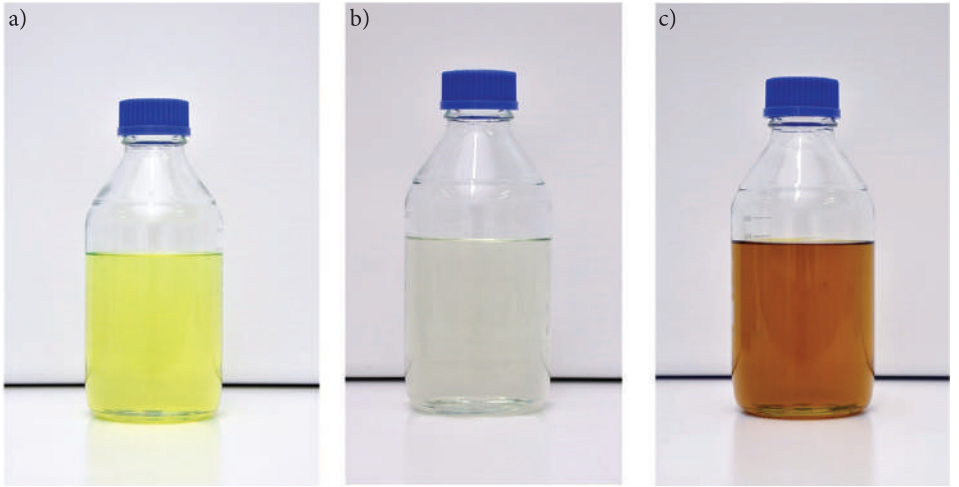
Badano następujące parametry próbek paliw:

— gęstość w temperaturze 15°C (ρ_{15}),

— lepkość kinematyczną w 40°C (n_{40}),

— temperaturę zapłonu (T_Z),

— temperaturę mętnienia (T_M),
— temperaturę zablokowania zimnego filtru (T_{ZZF}),
— temperaturę płynięcia (T_P).



Rys. 1. Wygląd zewnętrzny zbadanych próbek: A — olej napędowy bez dodatku RME (ON bez RME), B —hydrorafinowany olej roślinny (HVO), C — estry metylowe oleju rzepakowego (RME)
Źródło: opracowanie własne

Gęstość badano metodą oscylacyjną z U-rurką na podstawie normy PN-EN ISO 12185:2002. Lepkość kinematyczną oznaczono na podstawie normy PN-EN ISO 3104:2004. Temperaturę zapłonu wyznaczono metodą opisaną w normie PN-EN 2719:2016-08, a temperaturę zablokowania zimnego filtru, posiłkując się normą PN-EN 116:2015-09, temperaturę mętnienia PN-EN ISO 3015:2019-06, natomiast temperaturę płynięcia PN-EN ISO 3016:2019-06.

Wyniki badań gęstości, lepkości, temperatury zapłonu, zablokowania zimnego filtru, mętnienia i płynięcia zestawiono w tabeli 1.

Tabela 1

Wyniki badań gęstości, lepkości kinematycznej i temperatury zapłonu próbek paliwa

	Gęstość w temp. 15°C [kg/m^3]	Lepkość kinematyczna w temp. 40°C [mm^2/s]	Temperatura zapłonu [°C]
ON bez RME	826,0 ±0,18	2,28 ±0,01	72,7 ±1,7
HVO	780,0 ±0,18	2,96 ±0,01	78,5 ±0,9
RME	880,0 ±0,18	2,33 ±0,01	165,0 ±3,0

Źródło: opracowanie własne

Z danych w tabeli wynika, że najmniejszą gęstością charakteryzuje się hydrorafinowany olej roślinny, natomiast największą estry metylowe oleju rzepakowego. Gęstość oleju napędowego bez dodatku RME mieści się w przedziale 820…845 kg/m^3, a więc spełnia wymagania jakościowe, podobnie jak RME. W Polsce nie ma przepisów regulujących wymagania jakościowe dla biopaliw ciekłych innych niż estry metylowe kwasów tłuszczowych.

Lepkość kinematyczna paliw płynnych decyduje o przebiegu rozpylania oraz odparowania ON w komorze spalania [2]. Normatywne wymagania jakościowe dla lepkości kinematycznej w temperaturze 40°C dla olejów napędowych wynoszą 2,0…4,5 mm^2/s. Wynik badanej próbki ON mieści się w tym przedziale. Pozostałe próbki biopaliw wykazały zbliżone wartości do wyniku otrzymanego dla oleju napędowego bez dodatku RME.

Tabela 2

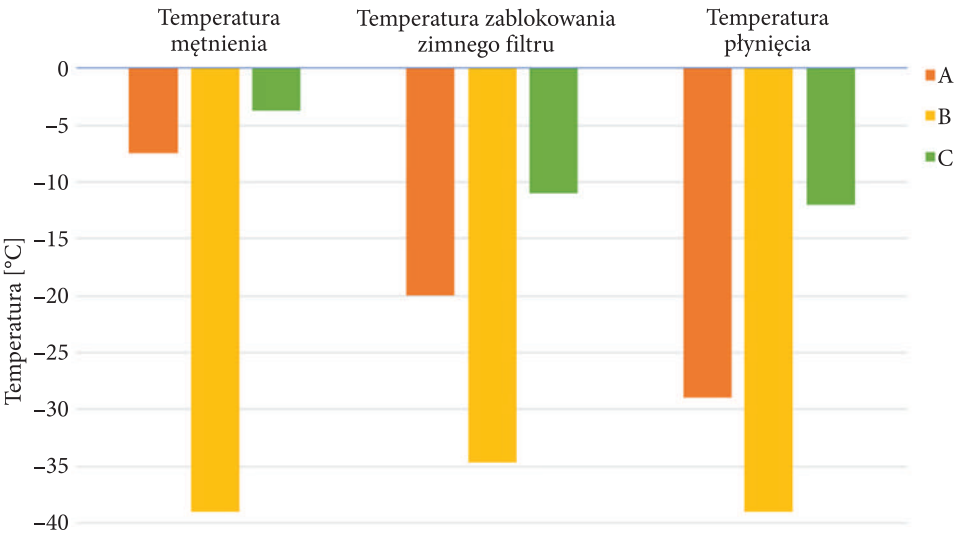
Wyniki badań temperatury mętnienia, zablokowania zimnego filtru i płynięcia próbek paliwa

	Temperatura mętnienia [°C]	Temperatura zablokowania zimnego filtru [°C]	Temperatura płynięcia [°C]
ON bez RME	–7,5 ±0,7	–20,0 ±1,6	–29,0 ±2,9
HVO	< -39,0	–34,7 ±1,8	< –39,0
RME	–3,8 ±0,5	–11,0 ±0,8	–12,0 ±1,5

Źródło: opracowanie własne

Najniższą temperaturę mętnienia, zablokowania zimnego filtru i płynięcia zaobserwowano dla paliwa HVO, co świadczy o bardzo dobrych właściwościach niskotemperaturowych tego biopaliwa (tab. 2). Badana próbka spełnia wymagania jakościowe w tych trzech aspektach określonych dla paliw ciekłych stosowanych w pojazdach wyposażonych w silnik o zapłonie samoczynnym. Próbka ON bez dodatku RME również spełnia wymagania jakościowe dla okresu użytkowania, w którym została pobrana. Nieco gorsze właściwości ON w stosunku do HVO mogą wynikać z faktu, że badana próbka została pobrana bezpośrednio z bazy paliw i w swoim składzie nie zawiera pakietu dodatków producenta.

Właściwości niskotemperaturowe są bardzo istotne z punktu przydatności eksploatacyjnej. Podczas obniżania temperatury oleju napędowego wydzielają się drobne i twarde kryształki węglowodorów parafinowych, które w miarę schładzania próbki mogą się powiększać i aglomerować. Widoczne jest charakterystyczne zmętnienie próbki. Takie zjawisko może prowadzić do osadzania się kryształów parafin na ściankach zbiornika oraz przewodów paliwa [2]. Spośród badanych próbek paliw estry metylowe oleju rzepakowego charakteryzują się najgorszymi właściwościami niskotemperaturowymi (rys. 2). Wynika z tego, że dodanie RME do ON może wpłynąć na pogorszenie właściwości niskotemperaturowych oleju napędowego.



Rys. 2. Zestawianie temperatury mętnienia, zablokowania zimnego filtru i płynięcia dla próbek: A — ON bez RME, B — HVO, C — RME

Źródło: opracowanie własne

Hydrorafinowany olej roślinny charakteryzuje dodatkowo m.in. niska rozpuszczalność w wodzie i wysoka liczba cetanowa. Duże różnice między liczbą cetanową konwencjonalnego oleju napędowego i HVO wymagają pewnych regulacji w sposobie sterowania silnikiem w celu skompensowania samozapłonu paliwa na wcześniejszym etapie cyklu, niemniej nie jest to konieczne do poprawnej pracy silnika. Smarność HVO jest bardzo niska ze względu na brak związków siarki i tlenu w paliwie, z tego powodu wymagany jest dla HVO dodatek smarny. Wartość opałowa hydrorafinowanego oleju roślinnego jest wyższa ze względu na wyższą zawartość wodoru. Gęstość jest niższa ze względu na naturę parafinową i niższą końcową temperaturę wrzenia. Tabela nr 3 przedstawia porównawczy wpływ zastosowania HVO i FAME na pracę typowego silnika o zapłonie samoczynnym. Niemal wszystkie czynniki operacyjne są na korzyść HVO.

Tabela 3

Wpływ rodzaju biopaliwa na pracę i eksploatację konwencjonalnego silnika o ZS [15]

	HVO100*	**Biodiesel B100 (FAME)**
Możliwość wykorzystania w silnikach:	Każdy typ silnika o ZS	Silniki stworzone do pracy tylko na tym paliwie
Właściwości samozapłonowe	Sugerowane skompensowanie różnic wynikających ze zwłoki zapłonu	Podobne do handlowego oleju napędowego
Właściwości smarne	Wymagane jest stosowanie dodatków smarnych	Bardzo dobre

cd. tab. 3

Właściwości niskotemperaturowe	Lepsze w porównaniu do handlowego oleju napędowego	Gorsze w porównaniu do handlowego oleju napędowego, wysoka temperatura mętnienia oraz zablokowania filtra
Atomizacja i odparowanie paliwa w komorze spalania	Na poziomie handlowego oleju napędowego	Pogorszone. Większa gęstość i lepkość oraz niższa lotność
Charakterystyka momentu obrotowego	Znamionowa	Pogorszona
Zużycie paliwa	Na poziomie handlowego oleju napędowego	Większe
Tendencja do tworzenia osadów w układzie paliwowym	Na poziomie handlowego oleju napędowego	Prawdopodobieństwo zwiększonej ilości osadów
Wpływ na zużycie materiałów konstrukcyjnych silnika	Na poziomie handlowego oleju napędowego	Niektóre tworzywa sztuczne oraz lakiery rozpuszczają się w kontakcie z FAME
Wpływ na olej silnikowy	Na poziomie handlowego oleju napędowego	Konieczność częstszej wymiany oleju silnikowego

* 100% zawartości — hydrorafinowany olej roślinny

Źródło: opracowanie własne

3. Produkcja i przechowywanie HVO

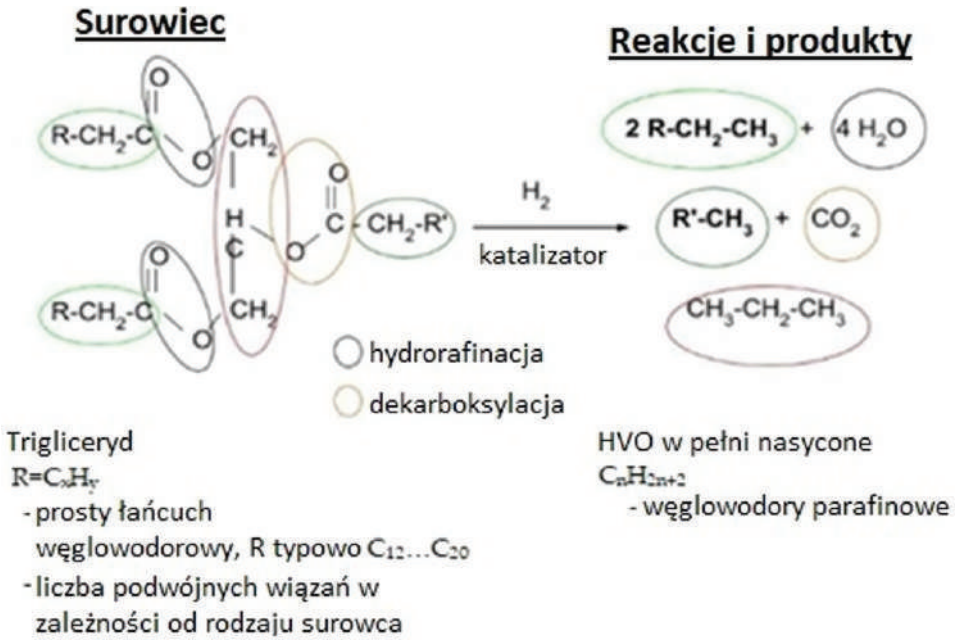
Na rysunku nr 3 przedstawiono schemat procesu hydrokonwersji olejów roślinnych i tłuszczów zwierzęcych, którego produktem końcowym jest HVO.

Biokomponent otrzymywany w procesie hydrokonwersji olejów roślinnych i tłuszczów zwierzęcych, określany jako HVO, to węglowodorowa frakcja parafinowa uzyskana w wyniku katalitycznego procesu hydrokonwersji trójglicerydów kwasów tłuszczowych. Jeżeli stosuje się materiał niespożywczy lub odpadowy, jest on klasyfikowany jako składniki biomasy drugiej generacji. Trójglicerydy można podzielić na proste, zawierające trzy identyczne kwasy tłuszczowe, oraz złożone, składające się z trzech różnych kwasów tłuszczowych. Ze względu na obecność wiązań podwójnych kwasy tłuszczowe można podzielić na nasycone, jednonienasycone i wielonienasycone [8]. Właściwości fizykochemiczne triglicerydu zależą od właściwości kwasów tłuszczowych, a w przypadku złożonych triglicerydów od względnego położenia poszczególnych kwasów w cząsteczce triglicerydu. Triglicerydy nasyconych kwasów tłuszczowych mają wysoką temperaturę krzepnięcia (powyżej 30°C) i lepszą stabilność oksydacyjną niż triglicerydy nienasyconych kwasów tłuszczowych, które są bardziej reaktywne ze względu na obecność wiązań podwójnych [8].

Proces hydrokonwersji triglicerydów kwasów tłuszczowych składa się z trzech etapów [15]:

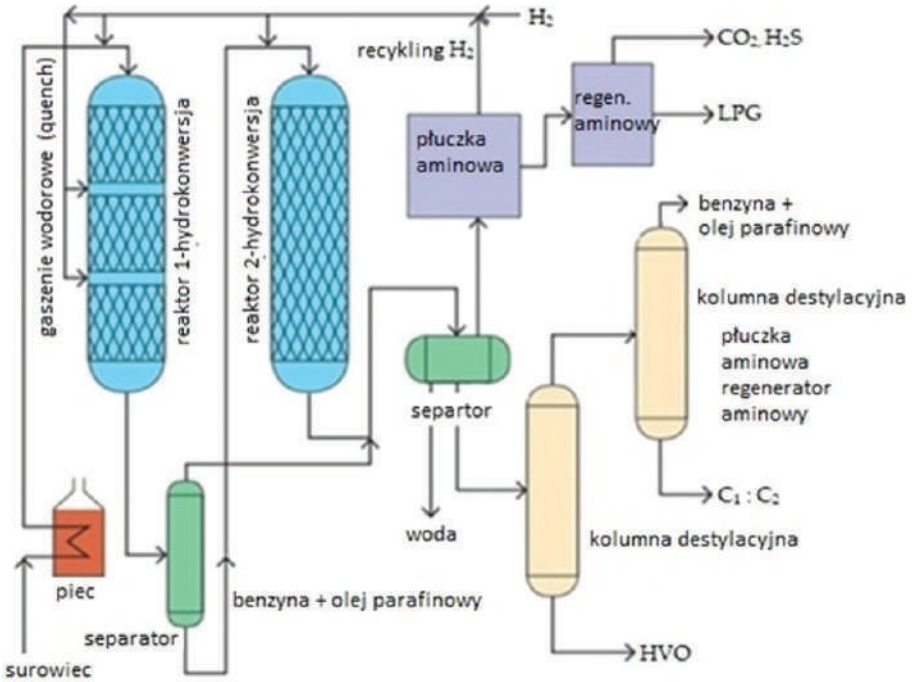
— uwodornienie wiązań podwójnych w łańcuchach węglowodorowych triglicerydów,
— rozpad cząsteczki triglicerydów na propen i kwasy tłuszczowe,
— rozkład kwasów tłuszczowych na węglowodór n-parafinowy za pomocą wody, dwutlenku węgla, tlenku węgla oraz uwodornienia propenu do propanu.

Na rysunku 4 przedstawiono schemat procesu produkcji HVO.



Rys. 3. Tworzenie i skład chemiczny HVO [15]

HVO w porównaniu do biodiesla produkowanego na bazie oleju rzepakowego (RME) lub olejów roślinnych (FAME) zachowuje bardzo dobre właściwości magazynowe (tabela 4). Jest mniej higroskopijny, charakteryzuje się stosunkowo długim rozkładem biologicznym i nie ma tendencji do pienienia i do sedymentacji w instalacjach magazynowych.



Rys. 4. Schemat procesu produkcji HVO [8]

Tabela 4

Wpływ rodzaju biopaliwa do silników o ZS na system dystrybucji i magazynowania [15]

	HVO	RME/FAME
Kontakt z wodą	Niewielka skłonność do tworzenia stabilnych mieszanin	Możliwość rozkładu (hydrolizy), tworzenia trwałych emulsji z wodą, tworzenia związków korozyjnych
Rozpuszczalność wody w paliwie	Mała	Duża higroskopijność
Skłonność do pienienia	Mała	Duża
Podatność na skażenie mikrobiologiczne	Mała	Duża
Stabilność paliwa podczas długotrwałego przechowywania	Bardzo dobra, mała podatność na procesy starzenia	Szybka degradacja paliwa w wyniku procesów starzenia, konieczność stosowania skutecznych dodatków antyoksydacyjnych
Skłonność do tworzenia osadów	Mała	Duża
Biodegradowalność	Mała, wolna dekompozycja	Duża, szybka dekompozycja

Źródło: opracowanie własne

Pionierem w badaniach hydrokonwersji olejów roślinnych i tłuszczów zwierzęcych na składniki paliw węglowodorowych, a następnie komercjalizacji tej technologii na skalę przemysłową był koncern naftowy Neste Oil. W Finlandii w lipcu 2007 roku uruchomiono pierwszą na świecie przemysłową instalację do produkcji wysokowęglowodanowych biokomponentów do oleju napędowego z naturalnych olejów i tłuszczów pod nazwą NExBTL o zdolności produkcyjnej 170 000 ton rocznie [8, 14]. W wyniku rozwoju tej technologii proces HVO oferowany jest przez takie firmy jak:

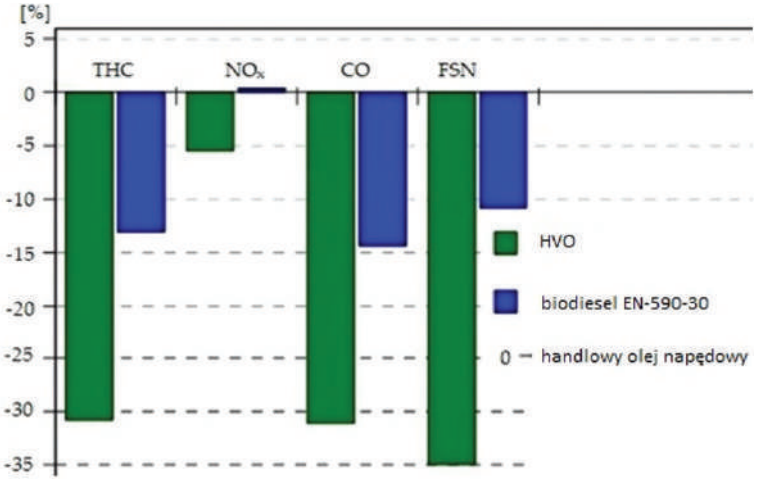
— Axens IFP (Vegan),
— Honeywell UOP (Ecofining - Green Diesel),
— Neste Oil (NExBTL),
— Syntroleum,
— UPM (BioVerno).

Na polskim rynku nowoczesną technologię uwodornienia olejów roślinnych wdroży PKN ORLEN w zakładzie produkcyjnym w Płocku. Jest to działanie zgodne z planem ORLEN 2030, które wzmocni pozycję koncernu na rynku biopaliw. Będzie też odpowiedzią na pakiet Unii Europejskiej o nazwie „Fit for 55", stawiający przed przemysłem rafineryjnym kolejne wyzwania w zakresie ograniczenia emisyjności paliw transportowych. Realizacja inwestycji planowana jest na czerwiec 2024 roku. Łącznie moce produkcyjne istniejących przemysłowych instalacji HVO na świecie przekroczyły 3 mln ton rocznie [16].

4. Emisja szkodliwych składników spalin

Dotychczas przeprowadzone badania wskazują na korzyści środowiskowe i eksploatacyjne HVO jako odnawialnego biopaliwa dla silników o ZS oraz są powtarzalne. Większość dostępnych prac wykazała, że użycie HVO zwykle prowadzi do obniżenia emisji składników spalin. Oficjalna informacja z firmy Neste, dostępna w dokumencie *Renewable Diesel Handbook*, mówi, że przy użyciu HVO zredukowano 90% emisji CO_2 w porównaniu z handlowym olejem napędowym [15].

Wykazano również znaczące redukcje emisji dla HC, NO_x, CO [1]. Istotny wpływ na emisję cząstek stałych, HC i CO zaobserwowano w pracach Murtonena i Aakko-Saksa [11], [12]. Zestawili oni wyniki emisji dla biodiesla EN 590-30, HVO oraz handlowego oleju napędowego. Testy przeprowadzono na stanowiskach-hamowniach z wykorzystaniem silników autobusów miejskich oraz w trakcie rzeczywistych przejazdów pojazdów w ruchu drogowym. W większości przypadków wszystkie regulowane prawnie emisje, takie jak NO_x, PM, CO i HC, zmniejszyły się dla HVO w porównaniu ze zwykłym olejem napędowym [10] — jak na rysunku poniżej (rys. 5).



Rys. 5. Emisja THC, NOx, CO i FSN dla silnika zasilanego HVO, biodieslem EN-590-30 i handlowym olejem napędowym [10]

Według innej pracy [3], w której przeprowadzono badania emisji spalin dla różnych wielkości silników (jednostki napędowe pojazdów ciężarowych i osobowych), zauważono redukcję cząstek stałych, emisji tlenku węgla oraz węglowodorów.

Badania przeprowadzone przez Soo-Young [12] w 2014 roku wykazały, że zastosowanie HVO w silniku bez zmian konstrukcyjnych bądź modyfikacji w sposobie jego sterowania zmniejsza emisje NO_x, cząstek stałych, HC i CO. Zgodnie z badaniem [3], w którym przeprowadzono testy emisji spalin na wysokoobciążonych turbodoładowanych silnikach o ZS, zauważono średnie redukcje wszystkich składników emisyjnych spalin. Najbardziej znaczącą redukcję o około 35% zmierzono w zadymieniu. Dzięki HVO emisja NO_x została zmniejszona o około 5%. Odnotowano również zmniejszenie jednostkowego zużycia paliwa o 5%. W tym zestawieniu zalety stosowania HVO pojawiają się również w kontekście typowego biodiesla [13].

Badania przeprowadzone w Lublinie przez dr. hab. inż. Jacka Hunicza oraz dr. inż. Pawła Kordosa wykazały, że HVO jest dużo bardziej odporne niż ON na duże współczynniki nastaw układu recyrkulacji spalin (ang. *Exhaust gas recirculation — EGR*). Cecha ta umożliwia uzyskanie spalania niskotemperaturowego dzięki dużemu ucząsteczkowieniu (rozdrobnieniu cząstek tru paliwa). HVO umożliwia uzyskanie sprawności cieplnej na poziomie 43%, przy zachowaniu emisji NOx i CO na poziomie normy Euro VI. Jest to wynik o 1,5% lepszy niż zoptymalizowane spalanie ON, przy jednoczesnym zmniejszeniu emisji wszystkich składników spalin, w szczególności aldehydów i węglowodorów aromatycznych [7].

Jak zauważono powyżej, większość badań wskazuje HVO jako paliwo, które może przyczynić się do zmniejszenia emisji gazów cieplarnianych. Inna obserwacja

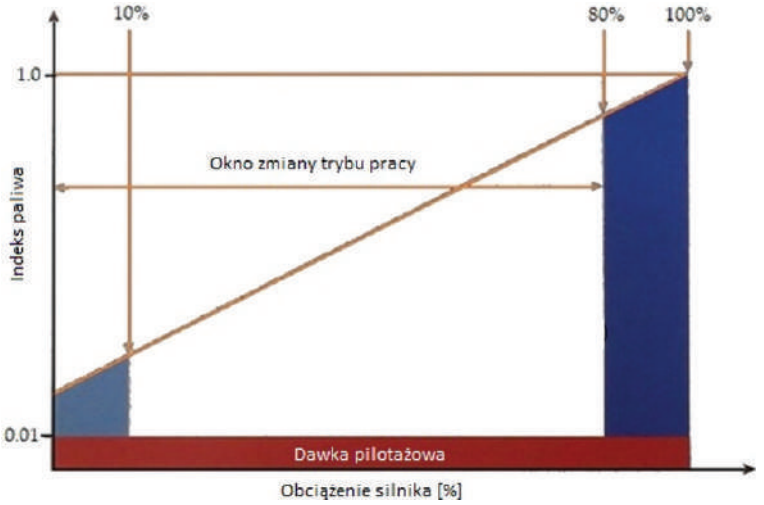
jest taka, że w większości przypadków HVO jest badane w warunkach ustalonej pracy silnika lub pojazdu, co skutkuje brakiem informacji w warunkach nieustalonych, które występują przez większość okresu eksploatacji pojazdu. Brak również informacji o stosowaniu HVO jako dawki pilotażowej w kontekście dwupaliwowych systemów zasilania.

5. HVO jako dawka pilotażowa paliwa

W silnikach o zasilaniu dwupaliwowym paliwo niskoreaktywne (benzyna, alkohol, CNG, LPG, HCNG, biometan lub inne paliwa) o wysokiej liczbie oktanowej dostarczane jest do komory spalania wraz z powietrzem poprzez układ dolotowy. Paliwo o niskiej reaktywności jest sprężane i mieszane z powietrzem do postaci wstępnie zmieszanej palnej mieszanki paliwowo-powietrznej. Ze względu na wysoką odporność na samozapłon i spalanie stukowe mieszanka ta nie zapali się pomimo wysokiej temperatury i ciśnienia w cylindrze. W pobliżu górnego martwego położenia tłoka w cylindrze wtryskiwane jest paliwo wysokoreaktywne (zwane też dawką pilotażową), tj. olej napędowy, RME, GTL, CTL czy HVO. Paliwo to musi charakteryzować się dobrymi właściwościami samozapalnymi, w tym niską temperaturą samozapłonu oraz możliwością dużej atomizacji paliwa w procesie jego wtrysku. Zapłon wysokoreaktywnego paliwa inicjuje zapłon powietrza z niskoreaktywnym wysokooktanowym paliwem. Istotą sterowania momentem zapłonu i przebiegiem spalania paliw stosowanych w silniku jest czas wtrysku (zarówno czas otwarcia wtryskiwacza, jak i moment rozpoczęcia wtrysku) i właściwości paliwa wysokoreaktywnego [5]. W zależności od rozwarstwienia i stopnia wymieszania paliwa wysokooktanowego z powietrzem przebieg jego spalania można scharakteryzować w różny sposób, co często stanowi kluczową kwestię w badaniach spalania typu *Reactivity Controlled Compression Ignition* (RCCI). Ilość wykorzystywanego paliwa danego rodzaju jest uzależniona w największym stopniu od obciążenia silnika oraz możliwości realizacji dozowania przez zastosowane układy wtrysku paliwa. Na rysunku 6 przedstawiono wykres z zależnością obciążenia silnika od możliwości wykorzystania obu paliw w różnych proporcjach.

Na poniższym rysunku kolor czerwony oznacza aktywny wtrysk paliwa pilotażowego, jasnoniebieski, że silnik można uruchomić na gazie, zsynchronizować i obciążać — silnik po pewnym czasie przełączy się na tryb dwupaliwowy, o ile obciążenie nie jest zwiększane poza ten obszar, a kolor ciemnoniebieski oznacza, że ten silnik jest w stanie pracować do 100% obciążenia na dowolnym paliwie, ale transfery z zasilania jednopaliwowego na dwupaliwowe są niedozwolone powyżej 80% obciążenia — transfery odwrotne są dozwolone przy dowolnym obciążeniu [5]. Powyższe informacje wskazują, że nawet nowoczesne układy dwupaliwowe (w szczególności układy typu *retrofit*) część czasu pracy realizują, pracując tylko

na paliwie wysokoreaktywnym. Dlatego też nawet przy dużych współczynnikach wymiany paliwa bazowego (wysokoreaktywnego) na paliwa niskoreaktywne istotne jest, by paliwo ciekłe było możliwie jak najbardziej ekologiczne i charakteryzowało się niską emisją GHG w trakcie spalania.



Rys. 6. Teoretyczne okno trybów pracy zasilania silnika dwupaliwowego [5]

Jeżeli możliwe jest wytworzenie jednorodnej mieszanki paliwa gazowego z powietrzem, paliwo to może również ulec samozapłonowi pod wpływem nacisku wywieranego na nie przez szybko spalające się paliwo o wysokiej reaktywności. Alkohole będące biopaliwami zawierają w swoich związkach oprócz wodoru i węgla również tlen, który zmniejsza gęstość energetyczną paliwa, ale może ułatwiać tworzenie się jednorodnej mieszanki w komorze spalania. Wiele biopaliw (w tym HVO) zawiera w swoim składzie chemicznym alkohole. Wskazuje to na dużą zasadność stosowania tych paliw w silnikach z zapłonem RCCI [2, 4, 6].

Dzięki zastosowaniu niskoreaktywnego paliwa jednorodnie wymieszanego z powietrzem zwiększa się efektywność wykorzystania energii chemicznej zawartej w paliwie. Na wzrost efektywności ma również wpływ równomierny rozkład temperatury w komorze spalania. Zmniejsza to całkowitą ilość ciepła przenikającego przez ściany cylindra i zapobiega tworzeniu się gorących punktów w cylindrze. Dzięki równomiernemu i niskotemperaturowemu przebiegowi procesu spalania zmniejsza się emisja tlenków azotu, cząstek stałych i sadzy. Wysoka temperatura płomienia jest jednym z głównych czynników powodujących powstawanie tlenków azotu podczas procesu spalania. W silnikach RCCI tlenki azotu nie tworzą się tak intensywnie jak w klasycznym silniku wysokoprężnym, również ze względu na spalanie w składzie zbliżonym do składu stechiometrycznego. Ilość tlenu i azotu w komorze spalania

podczas procesu jest zatem zmniejszona w porównaniu do klasycznego silnika wysokoprężnego [5, 6]. W silnikach dwupaliwowych niedziałających w trybie RCCI spalanie nie jest jednorodne w objętości czynnika roboczego ani w funkcji kąta obrotu wału korbowego. Zmniejszenie kinetycznej prędkości spalania (w stosunku do konwencjonalnego silnika ZS) powoduje mniej wytworzonych NOx, natomiast wzrost dyfuzyjnej prędkości spalania zmniejsza emisję PM. Wzrost temperatury w dyfuzyjnej fazie spalania powoduje dopalenie większej części powstałych wcześniej cząstek sadzy i w bilansie ogólnym zmniejsza emisję PM w spalinach dla tego typu silników.

Podsumowując powyższe informacje bazujące na dostępnej wiedzy, należy zauważyć, że istnieje potencjał wykorzystania HVO jako dawki paliwa wysokoreaktywnego (pilotażowego) w kontekście nowoczesnych systemów dwupaliwowych w celu zmniejszenia ogólnej emisji gazów cieplarnianych. Dalsze badania empiryczne na ten temat powinny zostać zrealizowane.

6. Podsumowanie

— Zmniejszenie emisji dwutlenku węgla można osiągnąć poprzez zwiększenie sprawności silnika spalinowego lub zmniejszenie zawartości węgla w spalanym paliwie.
— HVO to forma substytutu oleju napędowego o zmniejszonej zawartości cząstek węgla. Charakteryzuje się lepszymi właściwościami fizykochemicznymi i eksploatacyjnymi w porównaniu do ON i RME/FAME.
— HVO jest paliwem odnawialnym produkowanym w dużych ilościach na świecie i o lepszych niż konwencjonalne biopaliwa właściwościach logistyczno-magazynowych. Od 2024 roku HVO będzie produkowane i dystrybuowane w Polsce, co wskazuje na dobrą dostępność tego paliwa w przyszłości.
— Dotychczasowe badania wskazują na korzyści środowiskowe i eksploatacyjne HVO jako odnawialnego biopaliwa dla silników jednopaliwowych o ZS oraz są powtarzalne. Brakuje badań na temat emisji w wykorzystaniu HVO w układach dwupaliwowych.
— Istnieje potencjał wykorzystania HVO jako dawki paliwa wysokoreaktywnego (pilotażowego) w kontekście nowoczesnych systemów dwupaliwowych w celu zmniejszenia ogólnej emisji gazów cieplarnianych. Należy wykonać badania laboratoryjne w celu potwierdzenia lub obalenia tej tezy.

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J. CHOJNOWSKI, P. NOGAS

The potential of hydrotreated vegetable oil (HVO) as a high reactive biofuel in dual fuel systems

Abstract. In times of increased efforts to reduce greenhouse gas emissions to the atmosphere from the automotive sector, engineers and researchers are finding effective solutions to achieve this goal. The reduction of carbon dioxide emissions can be achieved by increasing the efficiency of the internal combustion engine or by reducing the carbon content in the fuel burned. The combination of both of these assumptions can be the use of modern dual-fuel systems for diesel engines in which the pilot dose of highly reactive fuel is hydrotreated vegetable oil (HVO), which is an ecological substitute for diesel oil. The topic of the use of HVO as a pilot fuel is empirically unexplored, however the available information on it indicates the validity of such applications.

Keywords: HVO, CO_2, dual-fuel, RCCI (reactivity controlled compression ignition), GHG (greenhouse gas)

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Review

A Review of Low-CO_2 Emission Fuels for a Dual-Fuel RCCI Engine

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Abstract: This article discusses the problems of exhaust gas emissions in the context of the possibility of their reduction through the use of fuels with hydrogen as an additive or hydrotreatment. These fuels, thanks to their properties, may be a suitable response to more and more demanding restrictions on exhaust emissions. The use of such fuels in reactivity controlled dual fuel engines (RCCI) is currently the most effective way of using them in internal combustion (IC) engines. Low-temperature combustion in this type of engine allows the use of all modern fuels intended for combustion engines with high thermal efficiency. Thermal efficiency higher than in classic engines allows for additional reduction of CO_2 emissions. In this work, the research on this subject was compiled, and conclusions were drawn as to further possibilities of popularizing the use of these fuels in a wide spectrum of applications and the prospect of using them on a mass scale.

Keywords: alternative fuels for IC engines; combustion control; emission characteristics; biofuels blending; hydrogen enriched fuels; RCCI engine; HVO fuel; HCNG fuel; dual-fuel engine

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1. Introduction

Three elements play a key role in the combustion of fuels for reciprocating internal combustion engines. These include hydrogen—H, carbon—C, and oxygen—O. The reactions leading to the release of chemical energy contained in the fuel in the form of heat used by the engine to perform the work lead to the formation of water molecules.

Hydrogen and carbon are not supplied to the internal combustion engine in pure forms—atoms are contained in the fuel with which engine is able to operate. Carbon in its elemental form is a solid fuel, and hydrogen is a very low-density gas. Both of these elements combine with oxygen after being broken down from the fuel's building blocks.

In view of the harm effects of carbon dioxide, including its being a greenhouse gas (GHG) its release during fuel combustion is highly undesirable. For this reason, engineers and researchers nowadays put emphasis on minimizing the emissions of this exhaust gas component. To reduce emissions, fuels with the lowest carbon to hydrogen ratios are used. Due to the systematic reductions in the permissible emissions of GHG, hydrogen is the most desirable element in fuel.

High carbon dioxide emissions can be reduced by increasing the efficiency of the internal combustion engine or reducing the carbon content of the fuel burned. Reasonable use of alternative fuels will allow one to reduce these emissions through the use of both of these factors. Effective reduction of GHG by reducing the number of carbon atoms in the fuel and improving the efficiency of the engine can be achieved thanks to the fuels presented in this study.

In addition to the three key elements mentioned above other elements present in the combustion chamber during this process also play important roles. Nitrogen is an energy reservoir necessary for engine operation; it accumulates a significant portion of the thermal energy generated in the fuel combustion process. Its participation in the combustion process creates problems, such as the emission of nitric oxides and nitrous

oxides, which are harmful to the health of living organisms. High levels of these compounds in exhaust gas are mainly produced by CI engines. Modern exhaust gas treatment systems allow for their complete elimination from exhaust gases during normal operation of the warmed-up engine and the selective catalytic reduction system. Engines of this type also show the highest efficiency that can be achieved; therefore—with the current state of knowledge—they are the best option for the use of low-emission energy sources [1].

The use of alternative fuels in SI engines that are adapted to burn conventional hydrocarbon fuels such as gasoline shows no measurable increase in overall engine efficiency. The biggest impact that the use of an alternative fuel has on the overall efficiency of the engine is a change to its thermal efficiency. Homogeneous and low-temperature methods of combustion of the fuel–air mixture are the most effective methods to achieve high thermal efficiency of the engine. This translates directly into the overall efficiency of the engine and lowering of carbon dioxide emissions [2].

Fuels intended for internal combustion engines are delivered to them in gaseous or liquid forms. In both cases, it is possible to improve their chemical compositions by adding additives containing high-calorific hydrogen. However, to burn them with high efficiency, it is necessary to ignite the homogeneous mixture with air and a suitably low temperature [3].

The closest thing to this solution is a CI engine. A large portion of the engine biofuels produced in the world is intended for combustion in diesel engines. Oils made from biocomponents have already become popular and have widespread utility as additives to diesel fuels all around the world [4].

In the case of gaseous fuels, their combustion in diesel engines is difficult due to the specificity of the construction and operation of such an engine. Direct gas injection into the engine TDC is not yet a proven solution that allows for independent operation of a diesel engine using gas fuel [5].

A diesel engine can operate effectively using two fuels at the same time. The gaseous fuel burned in this type of engine is ignited by a small dose of diesel fuel (pilot dose), or another liquid fuel that replaces diesel and has similar properties to regular diesel. The current aims when developing alternative engine fuels are continuously increasing engine efficiency and reducing GHG emissions [6]. The most important greenhouse engine combustion gases are colored yellow in all Table 1.

Table 1. Greenhouse gases [7].

Name	Formula	CO_2-Equivalent
Carbone dioxide	CO_2	1
Methane	CH_4	25
Nitrous oxide	N_2O	310
Water vapor	H_2O	Situation dependent
CFC-11, CFC-12 (chlorofluorocarbons)	C_X, F_Y, Cl_Z	5700–11,900
Sulphur hexafluoride	SF_6	22,200
Ozone	O_3	Unknown

The fuel used in the engine has a major influence on GHG emissions. Each of the listed compound has a designated global warming equivalent in relation to carbon dioxide. Figure 1 shows the volume percentage for each of the various greenhouse gases in Earth's atmosphere and a graph of the carbon dioxide concentration in the atmosphere from the XVII century to the XXI century.

The reason why CO_2 emissions have become a significant problem is visible in both the above charts from Figure 1. The largest share of greenhouse gases is CO_2, and the atmospheric concentration of this gas started to increase rapidly during the human industrialization era. As CO_2 absorbs and emits infrared radiation, this and other greenhouse gases trap the sun's warmth over the earth's surface—this is known as global warming [7]. CO_2 is the GHG that is currently the key to stopping the process of global warming, and its emissions can be influenced by using alternative fuels for internal combustion engines.

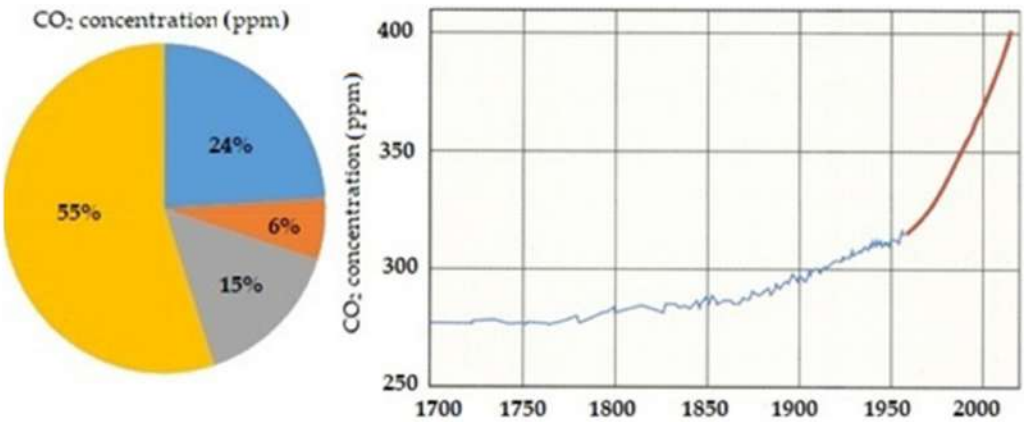

Figure 1. The pie chart shows the volume percentage for each of the various GHG. The graph shows carbon dioxide concentration in the atmosphere over time, from Scripps Institution of Oceanography, 2015. Ice-core data were used before 1958, and Mauna Loa data after 1958 [8].

2. Materials and Methods

This article presents information on the most important achievements of alternative fuels in terms of reducing exhaust emissions in modern and future RCCI dual-fuel engines. Information sources were analyzed to document the possibility of using low-emission fuels in dual-fuel combustion engines and the ways in which these fuels can be most effectively. This paper presents the reasons for limiting the emissions of exhaust gas components, such as carbon dioxide, methane, and nitrous oxide, and their impacts on global warming, which are well-known and widely described. The types of internal combustion engine which allow the use of the mentioned fuels and output the least of the harmful exhaust components have been reported. It is an analytical work based on the state of knowledge available at the moment. The researchers used scientific publications, books, conference materials, university multimedia presentations, scientific search engines (Research Gate and Google Scholar), and reports and studies made for companies. We also paid attention to popular science sources concerning this issue. A detailed list can be found in the bibliography.

3. Results

3.1. Biofuels and Alternative Fuels for Internal Combustion Engines

The fuel resources, fuel conversion technology, fuels themselves, and applications of them are shown in Figure 2, which should be treated as an introduction to the following analysis.

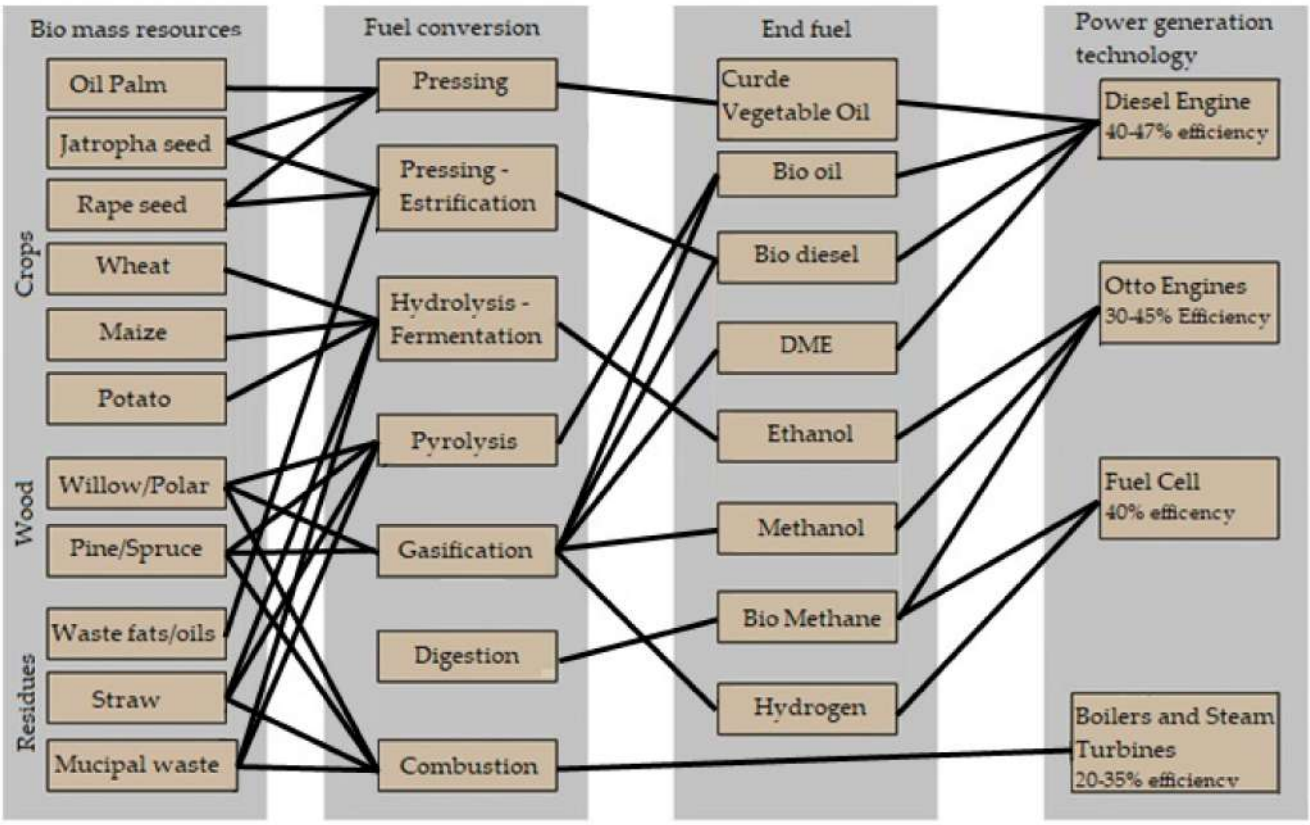

Figure 2. Alternative fuels for internal combustion engines presented by manufacturer Wartsila in [7].

Liquid fuels are used for the primary fueling of classic diesel engines. Popular fuels used to power diesel engines are diesel oil and biodiesel. Oils of vegetable origins are distributed in mixtures with petroleum (hydrocarbon), making diesel fuels. Alternative fuels are created to reduce GHG emissions and exhaust components resulting from the combustion process. The alternative fuel (biofuel) production process may be based on partial recirculation of the carbon that is in the natural cycle. This strategy is based on plant-based products that, as they grow, take carbon dioxide from the atmosphere for their growth. By storing the carbon contained in atmospheric CO_2, they eliminate some of it from the atmosphere. If exactly the same amount of CO_2 is later emitted into the atmosphere as a result of the combustion of the fuel obtained from these plants, it is possible to speak of full CO_2 recirculation. This is one of the main reasons scientists are trying to use biofuels in industry, energy, and transport. The second important factor is the limited amounts of non-renewable energy resources. The emphasis is on making them last by reducing energy consumption and replacing them with alternative energy sources. As a condition for access to the market in the European Union, biofuels must meet statutory requirements regarding sustainability—including cultivation—to receive certification. These requirements are enshrined in the EU directive 2009/28/EC which regulates the "Promotion of the Use of Energy from Renewable Sources." These also include proof of greenhouse gas reduction, production which had to be reduced by 50% by 2018. In Germany, in the meantime, the average reduction of GHG production is about 70%. The transport sector and agriculture are being challenged to contribute to GHG reduction, partly as a result of the decisions of the climate protection agreement in Paris. In addition, rising fossil fuel prices, along with national tax advantages and biofuel quotas, favor a growing global demand for biofuels. Currently biodiesels are mainly used in diesel fuel/biodiesel blends, but also as clean fuels in Europe and many other non-European countries, such as the USA, Mexico, Brazil, Argentina, Malaysia, and Indonesia [9].

Biofuels can be classified on the basis of the quality requirements that are set in a given country for liquid biofuels [10].

Polish domestic biofuels for diesel engines can be classified according to their origins. They can be produced from methanol (FAME), ethanol (FAEE), or RME (rapeseed oil). The biofuels available in Poland for diesel engines are diesel fuels containing 20% biofuel produced from rapeseed oil or methanol. Pure methanol is not a desirable engine fuel on its own, due to its properties and the products of its combustion, but it is a desirable additive for other fuels. Details of the fuels' properties are included in the regulations in Poland [10]. An interesting alternative in the context of bi fuels is hydrotreated vegetable oil (HVO), which is gaining popularity [11]. Biofuels can be manufactured all over the world with many different plants and fruits. Figure 3 shows the global biooil production in millions of tons per year, from the most commonly used plants.

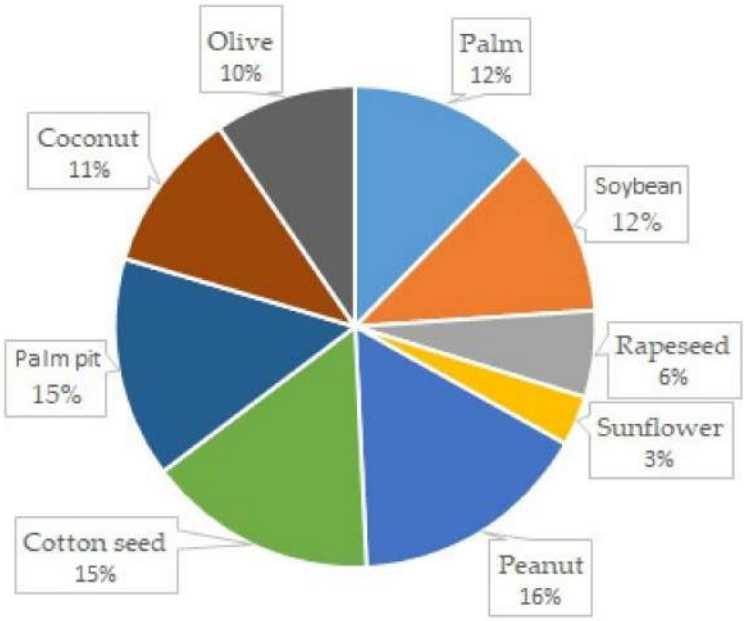

Figure 3. Global biooil production in millions tone per year [12].

Both gaseous and liquid biofuels are commonly produced from many sources, and the examples from the Polish market presented above constitute only a small portion of the global variety of sources from which biofuels are produced today. There is widespread research on the possibility of using oils of biological origins [13]. Such sources of their origins, given by [14], include oils made of pongamia [15], algae [16], jatrophy [17], rice [18], mushrooms [19], coffee grounds [20], palm trees [4], coconuts, peanuts, soybeans, linseeds, and olives [12]; animal fat and sources of waste [21]; plastic [22]; used tires [23] and other rubber products [24]; used oil [25], especially used engine oil [26], transformer oil [27] and vegetable oil and sunflower oils [28]; municipal sewage [14]; and intermediates from various technological processes, such as pyrolytic oil [29]. The plants and fruits most commonly used to produce biofuels are shown in Figure 4. The chart also shows the production efficiencies of individual plants and fruits per hectare. This parameter is very important for potential usage of a given plant as a raw material for the production of diesel biofuel.

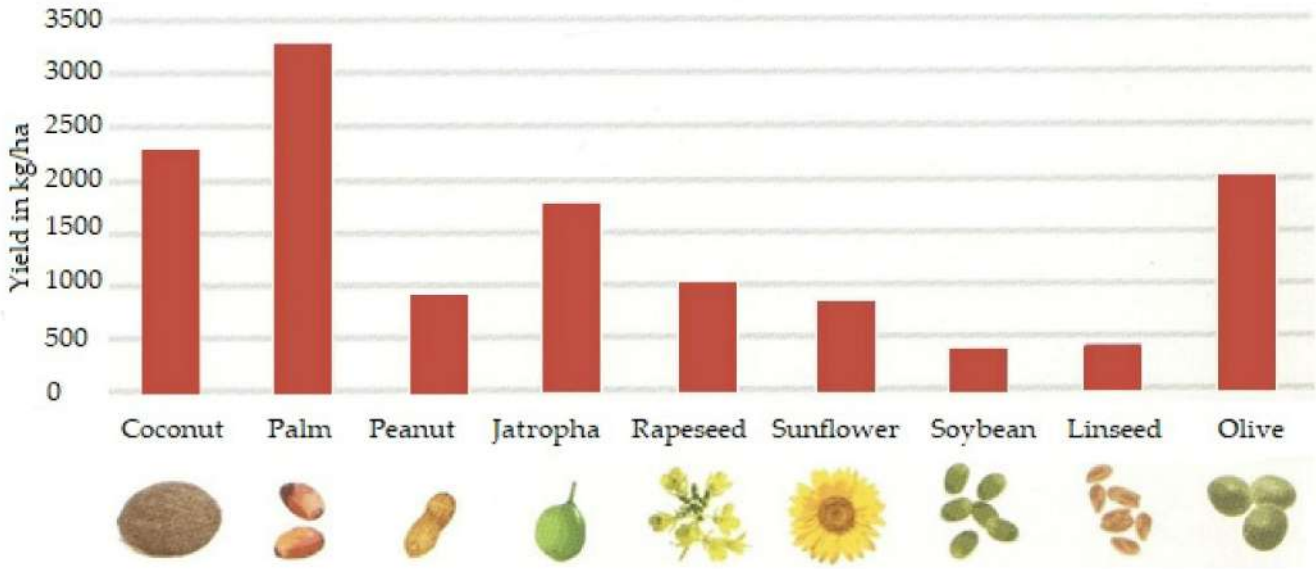

Figure 4. Annual yields of different liquids of biological origins [12].

In addition, alternative diesel fuels are made of ethanol [30] or coal [31]—coal to liquids (CTL)—or synthetically [32,33]. Both natural gases and purified and methane-enriched biogases can be changed into liquid fuels—gases to liquids (GTL)—but changes of this type are associated with increases in the total emissions of well-to-wheels greenhouse gases (WTW GHG) compared to the original compressed natural gases (CNG). However, the emissions of WTW GHG are lower than those of the conventional fuel for diesel engines, that is, diesel, but the methodology for calculating this value in relation to this fuel is still limited due to difficulties in quantifying potential products that could be replaced by GTL. "[The] GTL pathway has been limited until now because of the difficulty in quantifying potential products to be displaced by GTL coproducts" [34]. For these reasons, the analysis in this article covers fuels that are not subject to such transformations. Due to the ease of obtaining them and their ecological value, biogases are desirable fuels. When purified and enriched with methane, a biogas can be directed to gas network installations; compressed and stored in cylinders; or burned in cogeneration installations with simultaneous production of heat and electricity, or in gas boilers. It should be remembered that the main difference between a biogas and a natural gas is the methane content. In order to achieve the caloric parameters in line with the standards of gas injected into the natural gas network, such as the heat of combustion and the Wobbe index, it is necessary to pre-purify agricultural biogas and to enrich it with methane [35]. Biogases, similarly to biooils, can be produced from various types of biological and waste products [36]. The chemical composition of a biogas strongly depends on the source it comes from. It often contains significant amounts of hydrogen, which translates into lower CO_2 emissions in the combustion process, but any crude biogas also contains many ballast components, such as nitrogen and carbon dioxide, which must be removed in the cleaning process in order to use the biogas in an internal combustion engine.

3.2. Description of a Dual-Fuel RCCI Engine

Even when modern and equipped with the best exhaust gas cleaning systems, internal combustion (IC) engines always emit undesirable exhaust components. Figure 5 shows a typical pie chart of the exhaust gases emitted by marine engines.

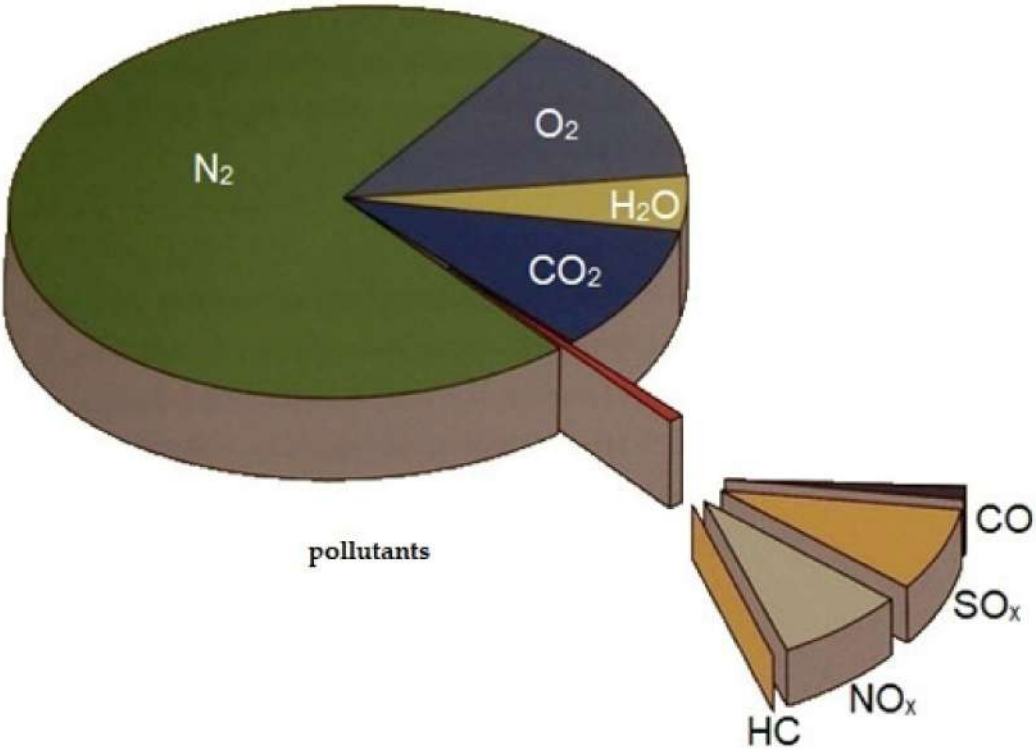

Figure 5. The typical exhaust gases emitted by a modern, marine diesel engine [8].

Intuitively, there is a strong correlation between exhaust gas composition and the type of fuel used in the engine. The basis for multi-fuel combustion is the use of modern diesel engines that allow low-temperature combustion.

Combustion of fuels in internal combustion engines is based on the use of one type of fuel which is ignited either by forced spark ignition (SI) or by compression ignition (CI). The highest efficiency is currently achievable thanks to the ignition of a homogeneous air–fuel mixture in homogeneous charge compression ignition (HCCI) engines, where the ignition of all the fuel in the combustion chamber occurs simultaneously. This combustion takes place at a lower temperature than in a classic diesel engine, which leads to lower heat losses and the formation of less NO_X, which translates into higher thermal efficiency for engines with this type of ignition. To achieve ignition in the combustion chamber of the engine, very specific (and difficult to achieve) conditions are required. Effective control of the moment at which ignition takes place in all areas of the engine operation is hard to achieve. For this reason, one of the variants of engines with this type of ignition method, which has great potential for use today, is an engine powered by two fuels with an ignition method called reactivity controlled compression ignition (RCCI).

RCCI is an ignition method in which highly reactive fuel is injected directly into the cylinder. It mixes with the air and self-ignites, thereby initiating the ignition of poorly reactive fuel, which is delivered via indirect injection to the combustion chamber earlier in the process. The same method of fueling the engine characterizes the classic dual-fuel engine, in which the fuel–air mixture is not homogeneous. Nowadays, every dual-fuel engine that is being developed allows the formation of a homogeneous air–fuel mixture in it. Figure 6 shows a schematic of classic dual-fuel injection system.

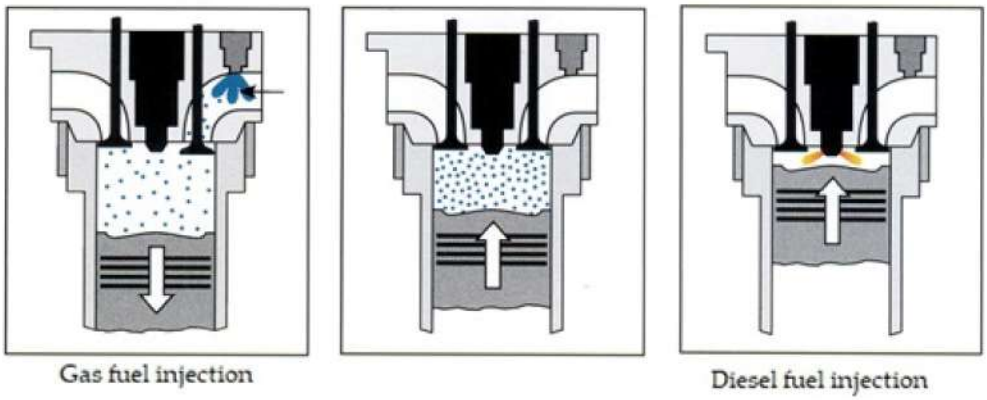

Figure 6. Schematics of a direct dual-fuel injection process (prior injection of the poorly reactive fuel) [8].

A different fueling strategy using these types of fuels is called direct dual-fuel stratification (DDFS)—the difference is the use of direct injection for both fuels used in the engine [37]. Direct injection of two fuels requires the use of several injectors or a special injector, such as the product offered by Westport corporation [38], which has been mentioned in all relevant publications covering dual-fuel engine solutions [39]. A schematic of DDFS is shown in Figure 7.

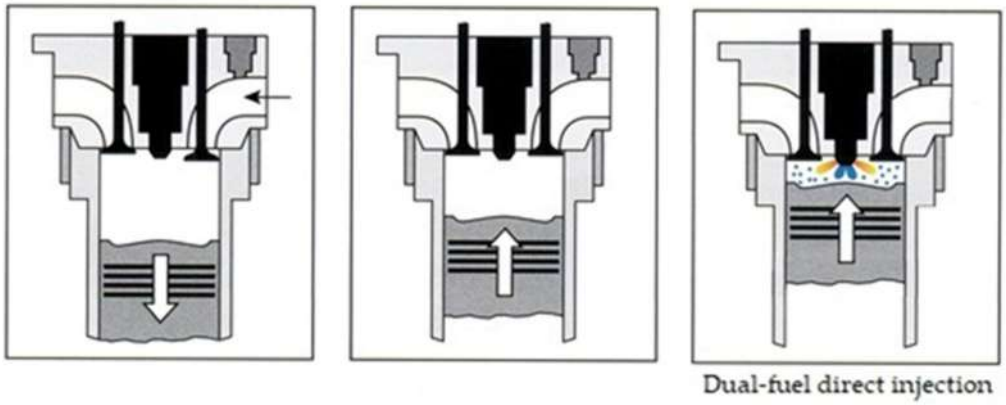

Figure 7. Schematics of direct dual-fuel injection (both fuels injected directly) [8].

An engine powered by dual-fuel CNG/diesel requires only the already mentioned pilot dose of diesel fuel to initiate the ignition. When using a gaseous fuel as an additional poorly reactive fuel, it might be necessary to increase the amount of diesel fuel injected due to the insufficient resistance of the additional gas fuel to knocking combustion. The occurrence of knocking depends on many factors, but mainly the type of fuel and the compression ratio in the engine. Knocking combustion causes the formation of uncontrolled foci of fuel self-ignition, which hinder the proper course of the combustion process and the operation of the internal combustion engine. Its presence determines the impossibility of using certain fuels in an engine and the level of acceptable substitution of the basic fuel. The Figure 8 shows a diagram of the power supply and ignition in an RCCI engine powered by a mixture of CNG and diesel.

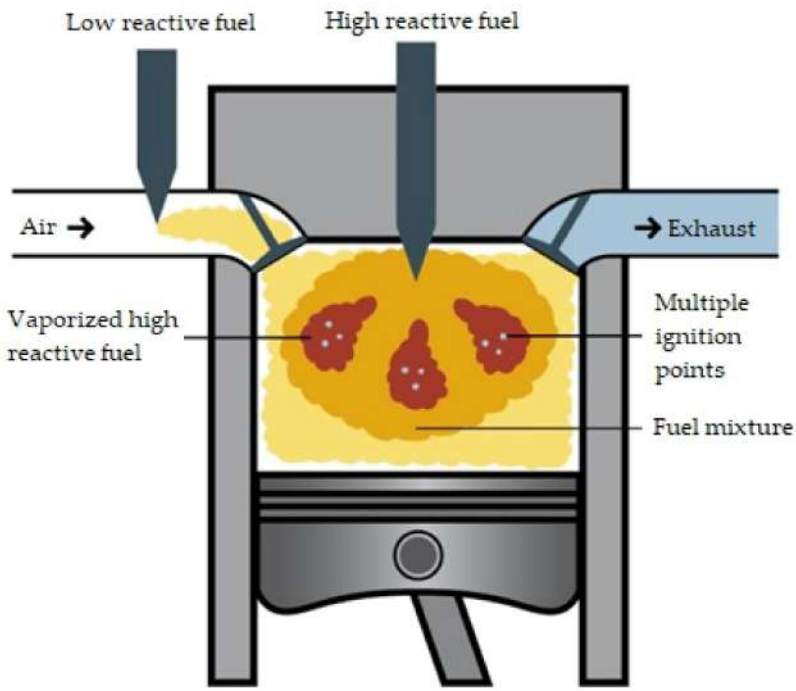

Figure 8. Ignition and combustion in an RCCI engine; custom artwork based on [40].

High octane poorly reactive fuel (gasoline, alcohol, CNG, LPG, HCNG, biomethane, and other fuels) in an RCCI engine is delivered with air by the engine intake system. A premixed form of poorly reactive fuel is compressed and mixed with air to form a combustible air–fuel mixture. Thanks to its high resistance to auto-ignition, this mixture will not ignite despite the high temperature and pressure inside the cylinder. Reactive fuel is injected near the top, dead center with respect to the piston. The fuel must have good self-igniting properties, including a low flash point and the possibility of large atomization of the fuel during the injection process. The ignition of highly reactive fuel initiates ignition of the air housed with poorly reactive high octane fuel. The keys to controlling the moment of ignition and the course of combustion of the fuels used in the engine is the injection time and the properties of the reactive fuel, hence the name of this method of self-ignition. Depending on the stratification and the degree of mixing of the high octane fuel with air, the course of its combustion can be characterized in various ways, which is often a key issue in RCCI combustion studies. The amount of a given type of fuel depends more than anything else on the load on the engine and the fuel injection system. Figure 9 shows a chart with the dependence of the engine load on the possibility of using both fuels in different proportions. These proportions also determine the value of the replacement coefficient, which in the case of a classic compression ignition engine converted into a dual-fuel engine, says that about the amount of basic fuel (diesel) could be replaced by the added fuel (gas). The formula for the mathematical representation of this value in classic fuel supplied dual-fuel engines (the result of this formula is the ratio of energy basis) is presented below:

$$rpe = \frac{m_p h_p}{m_p h_p + m_d h_d} \tag{1}$$

where m_p and m_d indicate the mass flow rates of premixed, low reactivity, port injected fuel (PFI) and directly injected (DI) high reactivity fuel, respectively; h_p and h_d are the lower heating values of the PFI and DI fuels [41].

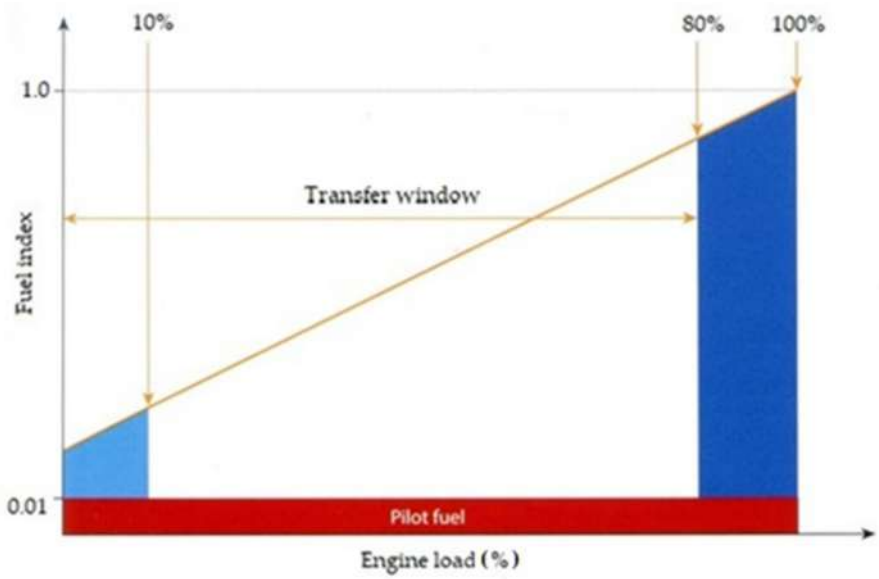

Figure 9. An engine fueling window. Red represents the engine pilot fuel injection is taking place. Light blue shows that the engine can be started in gas mode, synchronized, and loaded—the engine will move to a back-up fuel after a time, if the engine load is not increased enough to get out of this area. Dark blue means that this engine is able to run an up to 100% load on either fuel, but transfers from liquid to gas are not permitted above 80% load—transfers from gas to liquid are permitted at any load. The higher the fuel index, the more liquid fuel required [42].

Once a homogeneous mixture of gaseous fuel is formed with air, the fuel can also self-ignite by a rapidly burning (pressure build-up) highly reactive fuel. Biofuel alcohols contain, apart from hydrogen and carbon, oxygen, which reduces the energy density of the fuel, but may facilitate the formation of a homogeneous mixture in the combustion chamber. Many biofuels (including HVO) contain alcohols in their structures. This points to the legitimacy of using these fuels in RCCI engines [6].

Thanks to the use of a poorly reactive fuel, homogeneously mixed with air, the efficiency of using the chemical energy contained in the fuel is increased. Its growth is also influenced by the uniform temperature distribution in the combustion chamber. This reduces the total amount of heat passing through the cylinder walls and prevents hot spots from forming in the cylinder. This is clearly shown in Figure 10. The maximum combustion temperature in the combustion chamber is lower than in conventional compression ignition engines.

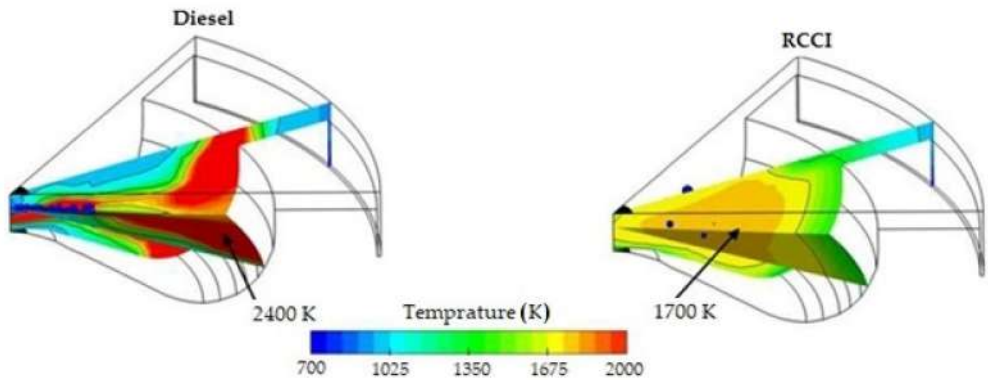

Figure 10. Temperature distribution in the combustion chamber of an RCCI engine powered by CNG and diesel fuel, and in a classic diesel engine powered only by a diesel fuel [6].

Due to low-temperature combustion process, emissions of nitrogen oxides, solid particles, and soot are reduced, just like in the classic HCCI engine. Minimizing the emissions of some harmful exhaust components usually goes along with increasing the emissions of others, such as NOx and the soot, CO, and THC group. The results of the emission levels for RCCI engines depend on the type of engine, types of fuels, fueling strategies, types of engine modifications, and the emissions focus on. For example, researchers in [41], while focusing on measuring the levels of NOx and soot emissions, showed a very strong correlation between the emissions of these two substances. High flame temperature is one of the main factors causing the formation of nitrogen oxides during the combustion process. For this reason, nitrogen oxides are not commonly formed in RCCI engines as intensively as in a classic diesel engine. Another cause is the fuels having the stoichiometric compositions. The amounts of oxygen and nitrogen in the combustion chamber during the process are therefore reduced compared to a classic diesel engine. However, with higher loads and a high degree of fuel exchange—close to the maximum—and when the EGR system is not used—which inevitably results in an increase in the combustion temperature—there will be a significant increase in NOx emissions, but allowing for sufficiently effective THC afterburning noticeably lowers soot emissions, practically to zero [41].

It is extremely important during RCCI engine tests to check what modifications and what ways of fueling the engine have been applied in experiments. After creating an effective dual-fuel supply system, the most important modifications are the geometry of combustion chamber and shape of DI fuel injector. Researchers have shown that piston bowl geometry changes are the most important modifications in the conversion of a classic diesel CI engine to an RCCI engine. Some have created their own piston modifications for existing engines [43,44] using advanced simulation software, such as computational fluid dynamics (CFD), in order to optimize the final shape. Figure 11 shows some changes that can be made to an existing piston bowl.

Figure 11. Cross-sectional views of the stock (left),) and bathtub (right) piston bowl geometries [43].

The form of combustion chamber in an RCCI engine is fundamentally different than in a classical CI diesel engine. The swirl ratio of the air–fuel mixture must be higher and the surface of the combustion chamber should be far smaller. This method of combustion of the air–fuel mixture necessitates flame propagation throughout the entire space above the piston. It also reduces the heat transfer surface, which leads to reduced thermal losses of the engine and an increase in its thermal efficiency. Changing the geometry of the combustion chamber inside the piston also allows one to easily change the compression ratio in the engine. Compression ratios can be 14.9:1, 16.1:1, or 18.7:1 because different piston bowl geometries have different effects on combustion [45]. Each of these compression ratios translates into a different piston shape and requires changes to the reactive fuel injection system. Researchers have created pistons for compression ratios 14.4:1 and 11:1 [43]. The effects of these changes are clearly visible in the emissions of harmful exhaust components and overall engine performance. Two groups of researchers collaborated to propose a piston bowl for an RCCI engine with a 15.3:1 compression ratio in exchange for the 17.5:1 stock engine [42]. This simple change had a significant impact on the engine's operating range, allowing it to be used as an RCCI engine. Figure 12 shows charts of injection-combustion strategies for two different compression ratio options.

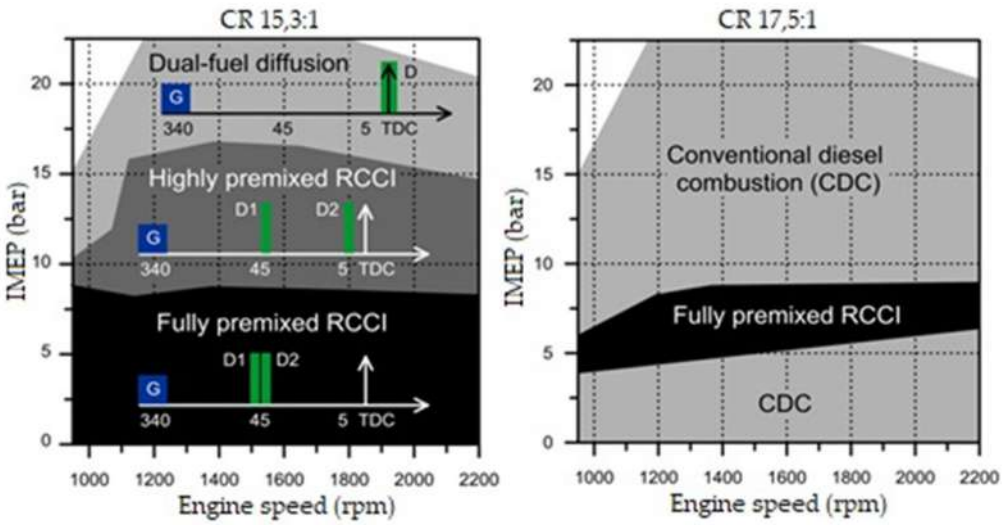

Figure 12. Injection-combustion strategies used for a dual-mode engine: dual-mode RCCI/CDC (left) and a dual-mode dual-fuel (right) [46].

The graphs above show an increase in the range in which it is possible to operate the engine in dual-fuel mode, along with a reduction in the compression ratio. However, in addition to reducing the compression ratio, the researchers significantly changed the shape of the piston, allowing the engine to run on both fuels simultaneously throughout its entire operating range. This change is shown in Figure 13.

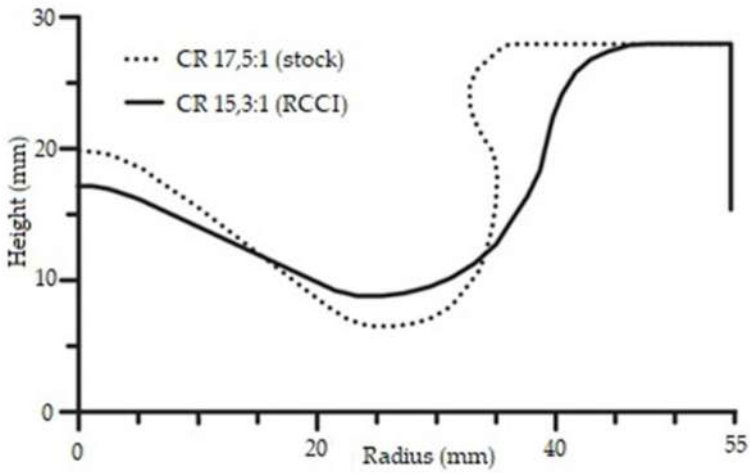

Figure 13. A cross-sectional view of the stock and RCCI piston bowl geometries [46].

Some researchers changed the compression ratio of an engine without significantly modifying the shape of the combustion chamber itself, which is classically located inside the piston crown. The compression ratio values adopted for the designed pistons were 16.5:1, 15.5:1, and 14.5:1 [47]. This made it possible to study the influences of the compression ratio itself on the performance and emissions of dual-fuel engines without any influence from the shape of the combustion chamber itself. The effects of the changes presented by the researchers were visible in the emissions of all harmful exhaust components. It was shown that the emissions of MHC and particulate matter PM_{10} decreased as the compression ratio in the engine decreased. There were slight increases in CO and NOX. However, these changes were not significant. This study also focused on the impact of the compression ratio on the CO_2 emissions, as shown in Table 2 (reproduced).

Table 2. A comparison of NEDC estimations of FC, CO_2, CO_2 equivalents, and global cycle r_p at different compression ratios [47].

	CR = 16.5 D	CR = 16.5 DF	CR = 15.5 DF	CR = 14.5 DF
CO_2 [g/kWh]	154.2	135.4	135.2	135.7
CO_2 saving [%]	-	−12.2	−12.3	−12.0
GHG emissions [$g_{CO2\text{-}eq}$/km]	154.2	191.4	188.2	183.7
FC (diesel+CH_4) [kg/100 km]	4.7	5.6	6.51	6.5
Average r_{pe} on NEDC [%]	-	54.7	58.2	60.4
EC [MJ/100 km]	202.9	261.6	303.2	305.2

The data above clearly show that the engine compression ratio has no influence on greenhouse gas emissions. The authors themselves drew this conclusion: "The DF CO2 saving over the test cycle are completely attributed to the intrinsic H/C characteristics of methane fuel; The GHG potential increase passing from diesel to dual-fuel mode due to the MHC emissions, the DF-CO2 equivalent calculated by means of the NEDC estimation increase of about 22% compared to diesel mode. In order to have the same GHG impact of diesel mode a global reduction of MHC emission (about 65%) is required." The effect of the compression ratio, in the absence significant changes to the shape of the combustion chamber, is not a sufficiently important issue in dual-fuel engines to be considered high-priority among the necessary engine modifications. Nearly a decade ago, researchers optimized the geometry of the combustion chamber in the RCCI engine, along with its compression ratio and the shape of the direct fuel injector [48]. These optimizations are necessary for the RCCI engine to efficiently convert fuel to mechanical energy. It was clearly indicated that the effect of the swirling of the air–fuel mixture in a classic diesel engine does not translate to dual-fuel engines, which is well illustrated by the sharp edges marked in the upper part of the piston shown in Figure 14. Changing the shape of the piston also requires adjustment of the shape of the highly reactive fuel injector, which should be adapted to the other modifications in order to be able to achieve optimal combustion of the fuel–air mixture, leading to minimization of CO_2 emissions in these engines.

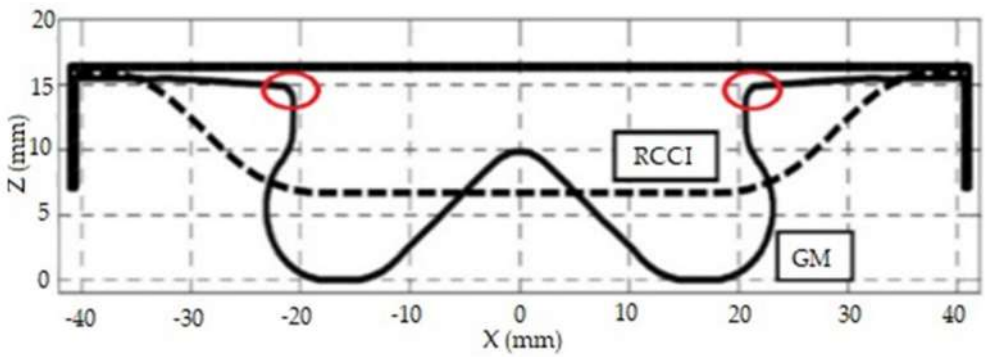

Figure 14. A comparison of RCCI and classic CI engine pistons with a combustion chamber inside each [46].

In the figure above, the edges marked in red circles are the most important parts of the piston, which lead to the formation of turbulence in the fuel–air mixture. In the case of CDC, their use is necessary for proper combustion and ignition. Figure 15 shows examples of the shape of the injector used in a classic compression ignition engine. Clearly, only the shape shown in the left part of the figure could be used in an RCCI engine, in which the ignition of the fuel–air mixture should occur in all of the space above the piston as simultaneously as possible.

Figure 15. A comparison of the combustion process in the CDC for different shapes of piston crown [7].

The shape of piston that is optimal for working in an RCCI engine is shown in Figure 16 [46]. The differences in the shapes of the two pistons show the significance of injection of the fuel directly into the combustion chamber.

Figure 16. A comparison of RCCI and classic CI engines made for dual-fuel research [46].

It should be mentioned here how important in the development of RCCI engines are the tests carried out with changes in the various types of fuels used. Scientists conducting this type of research should be commended. However, despite the significant contributions to the development of this field of science, this type of research has been carried out mainly in dual-fuel engines that do not use the alternative fuels with the greatest potential to reduce carbon dioxide emissions. It will be possible in the future to conduct further research with the use of other types of fuels described in this article.

Shape changes to the fuel injector and other parameters of RCCI engines are potential sources of advancement for modern single-fuel diesel engines, which are powered in a similar way to dual-fuel engines. Part of the fuel is added to the cylinder before the fuel–air compression stroke. Such engines operate on the basis of premixed charge compression ignition (PCCI) [49], where the already pre-stratified mixture is ignited. Developments in these engines are related to the developments in RCCI engines, because designers have to face similar problems and introduce solutions for the same elements. For PCCI engines though, the problems are centered on the use of one type of fuel, as it is difficult to create an engine that works efficiently on this principle. The possibility of using two independent fuels that fulfill different functions in the combustion process gives an advantage to RCCI engines that constitutes their superiority to other engines that use low-temperature fuels. Both of these solutions allow for an effective reduction of CO_2, soot and NOX emissions. However, dual-fuel engines extend the range of fuels that can used to the entire range of fuels currently used in internal combustion engines, because they use the fuels intended for both CI and SI engines.

3.3. Hydrogen-Enriched Alternative Fuels

3.3.1. Pure Hydrogen as Fuel for IC Engines

Using hydrogen to generate energy reduces the level of environmental pollution released (and the degradation of the planet's natural resources) if it is obtained from renewable sources. There are many methods of producing hydrogen, but if hydrogen is treated as a renewable fuel, it must come from a renewable source. It means that its production cannot involve the consumption of other fuels included in the group of non-renewable energy sources [50].

When hydrogen is burned, the energy of hydrogen's chemical bonds is converted into thermal energy, and then into mechanical energy. Currently, there are solutions that allow the electrochemical energy of hydrogen to be produced by the combination of oxygen and hydrogen. Then, for its use, an electric motor is used connected to a fuel cell running on hydrogen or a fuel rich in hydrogen. This solution, known for decades, has not been popularized (despite being an attractive option for cars powered by electrochemical batteries) due to the low availability of fuel cells as a result of the high prices of the raw materials needed for their production, and above all, the problems related to the distribution and storage of hydrogen [51].

To use hydrogen generated by renewable energy sources in internal combustion engines, the most effective strategy is to mix the hydrogen with other fuels and co-combust them.

Hydrogen as the basic fuel for internal combustion engines has not been widely used and does not have a wide distribution network. Unlike natural gas, which is widely available in virtually every country in the world, hydrogen is currently available only in a small number of places. The solution to this problem turned out to be Hythane, in which hydrogen is co-distributed with natural gas. This mixture can use the natural gas distribution infrastructure without any modifications.

3.3.2. Hydrogen and Methane as Fuels for IC Engines

Hydrogen, just like natural gas, is called a cryogenic liquid because its liquid phase is below 200 K (−73 °C). Under atmospheric conditions, hydrogen is present as a gas. For storage, it must be kept under pressure or in cryogenic conditions. However, its liquid phase has a very narrow temperature range (only 6 degrees Kelvin), and it has a low boiling point, which makes it difficult to store in a liquid form. Increasing the pressure under which liquid hydrogen can be stored will not significantly increase the boiling point, because at just 13 MPa it will approach the critical temperature. The density of gaseous hydrogen increases with pressure slower than it would appear from the ideal gas equation.

Many alternative fuels contain hydrogen. Natural gas has the highest ratio of hydrogen to carbon among all hydrocarbon fuels. Natural gas (NG) is distributed for internal combustion engines in several forms. The most popular are compressed natural gas and liquified natural gas (LNG).

NG as a fuel for internal combustion engines is considered an alternative to both CI and SI engines. NG used in conventional SI engines is not problematic in the context of interference with engine structure. On the other hand, a CI engine requires the addition of a spark ignition system, or co-combustion with some liquid fuel dedicated for a diesel engine.

Natural gas contains the lowest amount of carbon among all hydrocarbon fuels. It is a widely available fuel extracted from underground sources, just like oil. A comparison of their basic properties is presented in Table 3.

Table 3. The properties of hydrogen and natural gas. The figures from two sources were used to construct the table [52,53].

	Natural Gas	Hydrogen	Source
Hydrogen to carbon ratio	4:1	-	[52]
Energy density (MJ/kg)	48–50	120	[52]
Energy density (MJ/dm^3)	12.6	3.0	[53]
Auto ignition temperature (K)	813	858	[52]
	858	810	[53]
Minimum ignition Energy (mJ)	0.29	0.02	[52]
Minimum spark ignition Energy (mJ)	0.29	0.02	[53]
Octane rating	120+	130+	[52]
Octane number	RON 120	-	[53]
Flammability limits (%)	5.3–14%	4–70%	[52]
Stoichiometric Air-fuel ratio	9.48	29.53	[53]
Stoichiometric F/A (kg_{fuel}/kg_{air})	0.058	0.029	[52]
Wobbe index (MJ/Nm^3)	47.91–53.28	40.65–48.23	[52]
Flame velocity in air at NTP (cm/s)	37–45	265–325	[52]
Flame temperature (K)	2148	2318	[52]
Density (g/L)	0.7	0.07	[52]
Density (gaseous) (kg/m^3)	0.716	0.09	[53]
Mainly emissions	HC, CO, CO_2, NO_2, CH_4	H_2O, NOx	[52]
Energy Density (MJ/Lts)	25.3	2.9 at 350 bars	[52]
Molecular weight (kg/kmol)	19	2.016	[53]
Lower heating value (MJ/kg)	50	120	[53]
Laminar flame speed (cm/s)	≈42	≈230	[53]

Natural gas may significantly differ in properties depending on its degree of processing after its extraction. Properties that must be met by natural gas sold as CNG or LNG fuel are determined by the law in force in a given distribution area. Sheets of characteristics are available in each country where fuel is distributed. In the case of Poland, CNG and LNG have two independent fuel charters [54].

Liquid fuel based on natural gas, LNG, comes from many different sources and varies significantly in chemical composition. The Wobbe numbers (Wobbe index) among the LNG delivered to Poland from various countries may be in range of 54.22 to 56.86 for the upper Wobbe index and from 48.86 to 51.31 MJ/m^3 for the lower Wobbe index. These depend on low heating value (LHV) and high heating value (HHV) of the fuel, which are 37.15–41.71 MJ/m^3 (LHV) and 41.17–46.06 MJ/m^3 (HHV) for LNG. In delivery systems, the upper index of the Wobbe number should be at minimum level of 45, and the maximum level 56.9 MJ/m^3. Values for various countries are shown in Table 4. Interesting is that for Trinidad LNG fuel, the HHV is lower than the LHV for fuels from Oman, Australia, and Malysia. The Polish standard requires a minimal HHV greater than 34 MJ/m^3, and an LHV not less than 31 MJ/m^3. This shows how important in laboratory research it will be to determine the fuel source and its physicochemical properties [55].

Many sources show different parameters and requirements for CNG fuel. For example, in [7], for LHV the minimal value is 31 MJ/nm^3. That requirement is for a Wartsila ship engine, which can work in dual-fuel mode. HHV and LHV parameters are different for gases from different countries: Groningen NG from the Netherlands has an HHV equal to 35.1 MJ/m^3 and an LHV equal to 31.6 MJ/m^3. Ecofisk NG from Norway has an HHV equal to 44.0 MJ/m^3 and an LHV equal to 39.8 MJ/m^3. Danish Gas from Denmark has HHV equal to 46.2 MJ/m^3 and an LHV equal to 41.7 MJ/m^3. Each country can pride itself on natural gas with unique physicochemical properties.

Table 4. Different sources of LNG fuel in Poland [55].

Extraction of LNG	Ingredients				H_s (HHV)	H_i (LHV)	W_s (Upper)	W_i (Lower)	ρ	d
	Methane	Ethane	Propane	C4+						
	(% mol)	(MJ/m^3)	(MJ/m^3)	(% mol)	(MJ/m^3)	(MJ/m^3)	(MJ/m^3)	(MJ/m^3)	(kg/m^3)	-
Brunei	89.76	45.40	41.10	2.29	45.40	41.10	56.50	51.14	0.835	0.646
Trinidad	96.14	41.17	37.15	0.07	41.17	37.15	54.22	48.86	0.746	0.578
Algeria	88.83	44.15	39.92	0.38	44.15	39.92	55.85	50.5	0.808	0.625
Indonesia	90.18	44.22	39.98	1.03	44.22	39.98	55.88	50.53	0.809	0.626
Nigeria	90.53	44.57	40.31	1.47	44.57	40.31	56.08	50.71	0.817	0.632
Qatar	89.27	44.61	40.36	1.16	44.61	40.36	56.09	50.73	0.818	0.633
Abu Dhabi	85.96	44.61	40.36	0.14	44.61	40.36	56.10	50.77	0.818	0.632
Malaysia	87.64	45.78	41.45	1.50	45.78	41.45	56.71	51.33	0.843	0.652
Australia	86.41	45.69	41.37	0.95	45.69	41.37	56.67	51.31	0.841	0.650
Oman	86.61	46.06	41.71	1.76	46.06	41.71	56.86	51.5	0.848	0.656

Tables 5 and 6 show the chemical compositions of LNG fuel from similar countries as in Table 4. Differences are shown not only among different countries, but among data sources too.

Table 5. Chemical compositions of LNG imported from various countries all over the world [56].

Terminal	Methane	Ethane	Propane	Butane	Nitrogen
Abu Dhabi	87.07	11.41	1.27	0.14	0.11
Alaska	99.8	0.10	NA	NA	NA
Algeria	91.40	7.87	0.44	0.00	0.28
Australia	87.82	8.30	2.98	0.88	0.01
Brunei	89.40	6.30	2.80	1.30	0.00
Indonesia	90.60	6.00	2.48	0.82	0.09
Malaysia	91.15	4.28	2.87	1.36	0.32
Oman	87.66	9.72	2.04	0.69	0.00
Qatar	89.87	6.65	2.30	0.98	0.19
Trinidad	92.26	6.39	0.91	0.43	0.00
Nigeria	91.60	4.60	2.40	1.30	0.10

Table 6. Chemical compositions of LNG imported from various countries all over the world [57]. Data for NG from Schrödinger, in the right column, are from [7].

Composition	Mole (%)	Mole (%)	Mole (%)	Mole (%)	Mole (%)	Mole (%)	Mole (%)	Mole (%)
Country	Nigeria	India	U.E.A	Malaysia	China	Thailand	Norway	Schrodengen
Methane	92.69	82.2	94	94.42	80.00	76.4	99.5	69.97
Nitrogen	2.18	7.7	0	0.44	0	1.8	0	21.52
Carbon dioxide	0.52	0.2	1	0.57	19.05	13.4	0	2.1
Ethane	3.43	6.1	4	2.29	0	6.3	0.3	2.5
Isobutene	0.12	1	0	0	0	0	0	0.47
n-butane	0.15	0.4	0	0.25	0	0	0	
Propane	0.71	2.4	1	0.03	0	2.1	0	0.9
n-propane	0.09	0	0	0	0	0	0	
Hexane	0.11	0	0	0	0	0	0	0.23
Others	0	0	0	2	0	0	0	2.29
Oxygen	0	0	0	0	0.95	0	0	0.02

Natural gas production (extraction) must also be accompanied by numerous processes for purification and enrichment, but the chemical compositions of natural gas from various sources still differ significantly [58].

Noticeable is that at that moment it is difficult to accurately define the chemical composition of natural gas applied as a fuel. That problem appears in every scientific attempt at having natural gas supply an internal combustion engine. Researchers in [57] presented the average CNG composition on the basis of information from many countries and sources, which is shown in Figure 17.

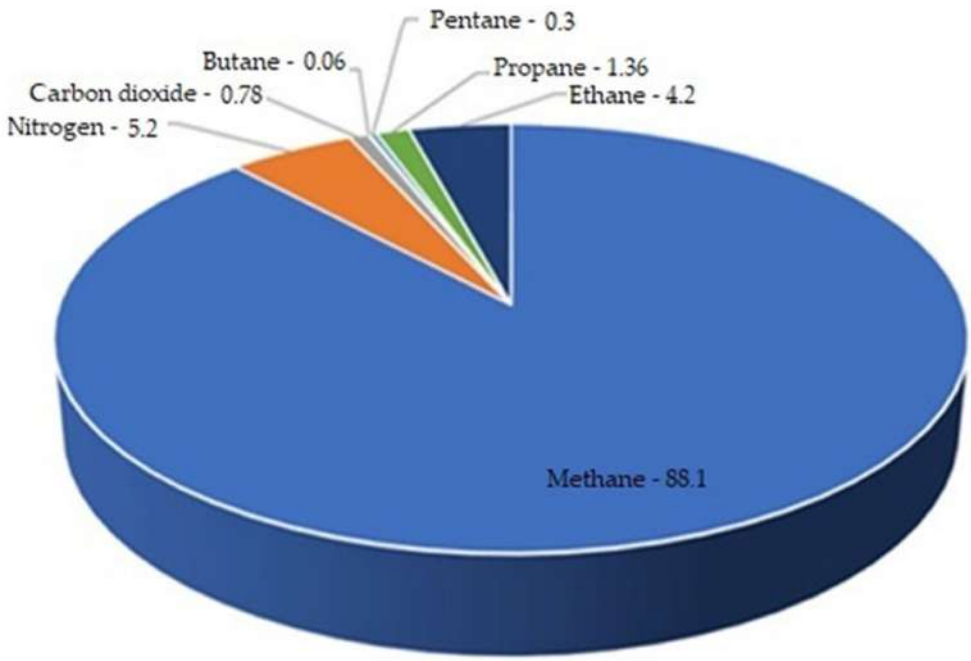

Figure 17. The average chemical composition of CNG fuel [57].

The differences in the chemical compositions of gaseous fuels concern not only the processed natural gas but also the natural biogas itself.

The presented differences in LNG's chemical composition and properties from various sources may facilitate the introduction of biogas (biomethane) into countries where regulations specifying the distribution of natural gas are not too restrictive. Such discrepancies do not normally occur with conventional engine fuels such as diesel and gasoline. Hence, the conclusion can be drawn that there is a greater potential for the production of biogas than biodiesel, due to the lower quality requirements that are currently imposed on it, that is natural gas, compared to diesel fuel.

Production of NG is increasing, and most countries do not allow its production to decline. Stocks of natural gas are greater than those of crude oil, so investing in clean combustion technologies for this fuel is a more far-reaching investment than in the case of classic liquid fuels. Figure 18 shows levels of production for NG. Such a far-reaching strategy for the use of fuel combustion technology also applies to hydrogen fuel and biofuels, which can be treated as renewable fuels for IC engines.

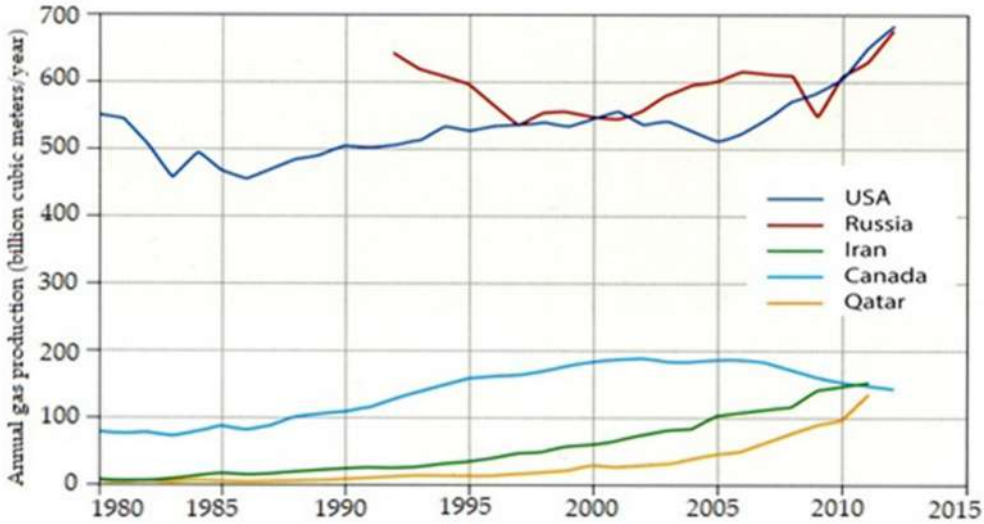

Figure 18. The production of natural gas in different countries [7].

Natural gas mixes well with air, forming homogeneous mixtures. Due to the high activation energy, the laminar flame velocity of the gas–air mixture is lower than it is for other fuels. This is especially true when working with low to medium loads. This extends the combustion period, worsening the efficiency of the circuit. In addition, the extended duration of combustion means that when the exhaust valve is opened, both the pressure and the temperature of the exhaust gas are higher than when burning traditional hydrocarbon fuels. This is counteracted by using different values of the ignition angle, higher values of the compression ratio, and compact combustion chambers, which result in high swirling of the charge [6].

3.3.3. Hydrogen and Methane as One Fuel for IC Engines

Both natural gas and pure hydrogen can be stored in liquefied (LNG, LH2—liquid hydrogen) and compressed forms (CNG, CH2—compressed hydrogen), and in both forms it is possible to store them in similar types of tanks. However, their mixture stored in liquid form is not used. The proper name Hythane should be treated equally to the fuel known as HCNG. This nomenclature is related to the country of distribution. In the USA, this fuel is known as Hythane, and in Europe and Asia, HCNG. Its characteristic features include:

- A high octane number;
- Good ignition properties;
- Wide flammability range and good diffusivity;
- A high auto-ignition temperature at a low boiling point;
- Low density and energy density;
- The possibility of obtaining from processes other than crude oil processing.

In dual-fuel diesel engines, the level of replacement of diesel with natural gas CH_4 is much higher than in the case of LPG, mainly due to the higher resistance of methane to knocking combustion and the higher calorific value of CNG [59]. The methane used as fuel is characterized by the highest octane number, reaching up to 130 octane. In the case of HCNG, this value will depend on the hydrogen content and whether the octane number is determined by the motor octane number (MON) or research octane number (RON). Hydrogen has an MON value of 70 and an RON value of 130; the octane values of methane are between 120 and 130. Generally, for gaseous fuels, it is assumed that the fuel resistance to knocking should be characterized by the value of the methane number, and not the octane number. Two extreme points have been adopted for LPG. Its value is considered to be 100 for pure methane and 0 for hydrogen. The higher the methane number value, the higher the fuel's resistance to knocking combustion. Natural gas belonging to the H group

should have a minimum methane number above 65 (16726: 2018). In [60], calculations were carried out, the results of which allowed them to state that, "Adding hydrogen to natural gas, in an amount that allows to maintain the physicochemical parameters of the gas specified in the relevant standards, reduces the methane number of the resulting natural gas-hydrogen mixture by a maximum of 22.1%. It should be added that in none of the analyzed cases the obtained methane number value was lower than the minimum value of 65. With regard to the optimal methane number value for gaseous fuels, it can be concluded that the addition of hydrogen to natural gas, while maintaining the assumptions made in the scope of energy parameters and gas density, may increase the knocking properties of the resulting mixture and contribute to the fact that it will not be an optimal fuel. The performed calculations and analyses also showed that the change in the value of the methane number of the natural gas-hydrogen mixture is proportional to the amount of hydrogen introduced into natural gas." This means that the hydrogen content in the fuel will proportionally characterize the value of the methane number of the fuel, which in our case is HCNG.

Hydrogen, when co-fired with other fuels, will more completely combust than the other fuels. Its physical ranking gives it a strong possibility of combusting independently in a reciprocating engine. That idea has not been disseminated to this day, despite many attempts and much scientific research [50].

Hythane is far more promising. It is a mixture of CH_4 and H_2, and makes a number of interesting tariff lists for its designation as combustion fuel. Besides the fact that arc fired natural gas by the very nature of crude oil emits more CO_2 than other hydrocarbons, it is possible to enrich natural gas with pure hydrogen. Such fuel is called Hythane, or HCNG, or H2CNG (compressed natural gas enriched with hydrogen). In [61], it was discovered, among other things, that the velocity of the laminar start-up of the methane–hydrogen mixture can be predicted with the square of the hydrogen proportion in the fuel. According to the research of some scientists, the fuel is most economical as a mixture of gas consisting of 11–36% hydrogen. As hydrogen content increases, the moment at which the reinforcement of the mixture is reached and the maximum pressure value in the polymer chamber is reached changes, which results in the need to create a map of the engine's operation in order to test an engine operating with two types of methane fuel, CNG and LNG. Heat release and maximum pressure according to crank angle is shown on Figure 19. Hythane reduces fuel consumption and lowers CO, CO_2, and NO_X emissions, but also provides less power and torque. According to some researchers, the amount of primary HC emitted is unchanged.

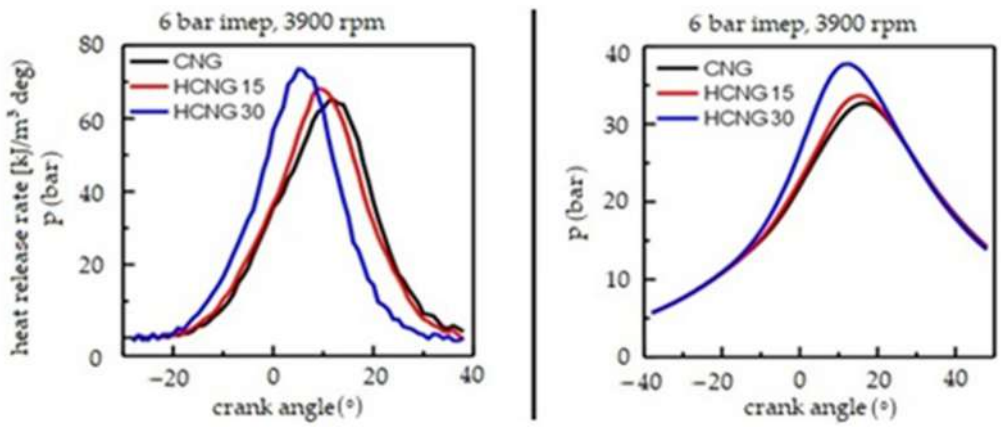

Figure 19. Graphs of temperature and pressure during the combustion of CNG with different amounts of additive hydrogen [61]. HCNG15 fuel had 14 vol.% hydrogen, and HCNG30 29.7 vol.%. The energy content of hydrogen in such fuel is approximately 4.6% for HCNG15 and 11.4% for HCNG30.

The big advantage of HCNG is the possibility of their common storage, both in the vehicle and in petrol stations. Fuel enrichment to an appropriate level of hydrogen content can be carried out in regular CNG petrol stations, and deliveries of hydrogen fuel are not as frequent and expensive for petrol stations as with the sale of pure hydrogen [62].

In order for this fuel to meet certain safety standards, the ratio among the individual gases, although it may be variable (different), should not have above a certain concentration of hydrogen. The maximum ratio that is considered safe for this gas mixture is 1:1 (volume). In most European countries where it has already been legally regulated, HCNG may get up to 9% of its energy content from hydrogen, the same amount as natural gas in a transmission system. In the USA, Hythane contains approximately 20% hydrogen by volume. A higher concentration of hydrogen mixed with natural gas is currently difficult to distribute, due to the intense increase in the impact of hydrogen on steel, which was used to build the existing methane distribution network, and the metal parts of tanks and engines. A mixture of hydrogen and methane does not become highly explosive until it exceeds 50% hydrogen content. Based on the gas distributed in Europe, a value that is even slightly above 50% by volume of hydrogen can be assumed. Not exceeding this value prevents the occurrence of hydrogen disease, in which hydrogen molecules penetrate the walls of the materials they come into contact with, leading to their degradation, which occurs with steel [63].

It should be noted that as the hydrogen content in the mixture increases, the calorific content decreases, and yet the fuel consumption with a hydrogen concentration within a certain range is lower than for a gas containing almost only methane or a large amount of hydrogen. This shows very well how such fuel affects the economics of using the energy contained in it.

In Hythane, there are different numbers of hydrogen atoms per carbon atom, depending on the proportions of the gases. As the concentration of hydrogen in Hythane increases, the amount of carbon dioxide emitted decreases, but the actual emissions of WTW GHG depend on the method of producing mixed hydrogen, as shown in Table 7 [64].

Table 7. WTW CO_2 emissions (g/MJ) for HCNG produced by different means and with different hydrogen contents [64].

H_2 Content	0%	8%	20%	25%	30%
Electrolysis green electricity	66	64	62	60	59
CO_2 change	0%	−2.4%	−6.4%	−8.3%	−10%
Electrolysis EU-mix electricity	66	70	77	80	84
CO_2 change	0%	+6.1%	+17%	+22%	+27%
Reforming natural gas	66	68	72	74	76
CO_2 change	0%	+3.5%	+9.7%	+13%	+16%

The combustion of fuels is always associated with the emission of substances into the atmosphere. In the case of GHG emitted by internal combustion engines, the three most common chemical compounds include carbon dioxide, methane, and nitrous oxide. These all contribute to the greenhouse effect. The amount of carbon dioxide emitted as a result of combustion of a given fuel is standardized for each country and precisely specified in the relevant documents. In Poland reports are prepared by government institutions [65–67]. These data are collected annually due to the Emission Trading System for CO_2 emissions in force in the EU. Carbon dioxide emissions from natural gas combustion for 2017 were found in [65] to total 56.1 kg/GJ. In 2019 [66], the total was about 55.5 kg/GJ. In 2021 [67], the value of emissions is expected to continue to decline. CO_2 emissions from biogas combustion totaled 54.6 kg/GJ; diesel oils, 74.1 kg/GJ; LPG, 63.1 kg/GJ; and petrol, 69.3 kg/GJ. The value for natural gas has changed over the years, in contrast to the other motor fuels mentioned here. This proves the significant influence of natural gas sources on its chemical

composition, and the amounts of natural gas imported from particular sources differ from year to year. The lower emissions value for biogas proves the noticeable increase hydrogen content in this gas. In the case of HCNG—calculating it purely (strictly) proportionally—with a hydrogen content of 15% by volume, Poland could achieve CO_2 emissions at a level close to the level of biogas CO_2 emissions. With a hydrogen content of 30%, it would be possible to fall below this level. At 50% hydrogen by volume, the CO_2 emissions would be reduced to the level of about 51 kg/GJ [68]. The changes in the thermal efficiency of the engine for different hydrogen contents in Hythane should be taken into account, and based on the analyzed test results, it can be assumed that the hydrogen concentration in HCNG should be in the range of a few to a dozen or so percent.

It should be noted that the emissions based on the method of fuel life cycle evaluation (WTW) for gas oil are the highest of conventional motor fuels, and its replacement by fuel such as HVO, CNG, or HCNG should be the highest priority for the internal combustion engine, to meet more ecological standards [69]. Carbon dioxide emissions from an engine operating on HCNG with different hydrogen concentrations, depending on RPM and the air to fuel ratio is shown on Figure 20.

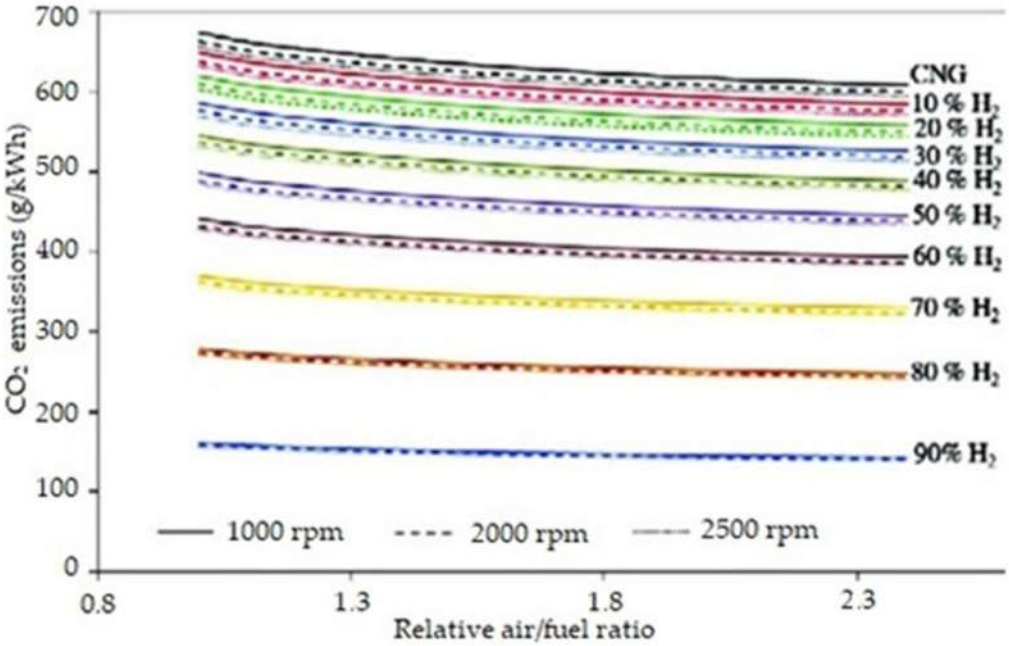

Figure 20. Carbon dioxide emissions from an engine operating on HCNG with different hydrogen concentrations, depending on RPM and the air to fuel ratio [64].

A hydrogen and methane mixture (in addition to reducing GHG) significantly reduces the amounts of toxic exhaust components emitted as well. In many studies, combusting HCNG showed reductions in CO and NO_X emissions. According to some research results, the emissions of HC compounds remained constant. When an engine with constant RPM was supplied with HCNG gas of various concentrations, an increase in NOx emissions was seen as the concentration of hydrogen in the fuel mixture increased [53]. The research in [70] also showed an increase in NOx emissions with an increase in hydrogen concentration in the gas mixture. Such results are obvious from the point of view of the physiochemistry of combustion, because it is very difficult to create a fuel that could reduce both the emissions of chemical compounds containing carbon and those containing nitrogen, because the mechanisms of their formation require opposite physical conditions. The issue of emissions of these compounds by HCNG-powered engines is currently the subject of research in many studies [71–77]. Worth noting is that the experimental values of NO_X emissions are higher than the calculated values resulting from a simulation [72]. Figure 21 shows the NO_X emission values that should theoretically occur in the Hythane combustion process

under various engine operating conditions (A, B, C, D, E, F), modeled by various methods (models I, II, and III), and the experimental values which were recorded (experimental).

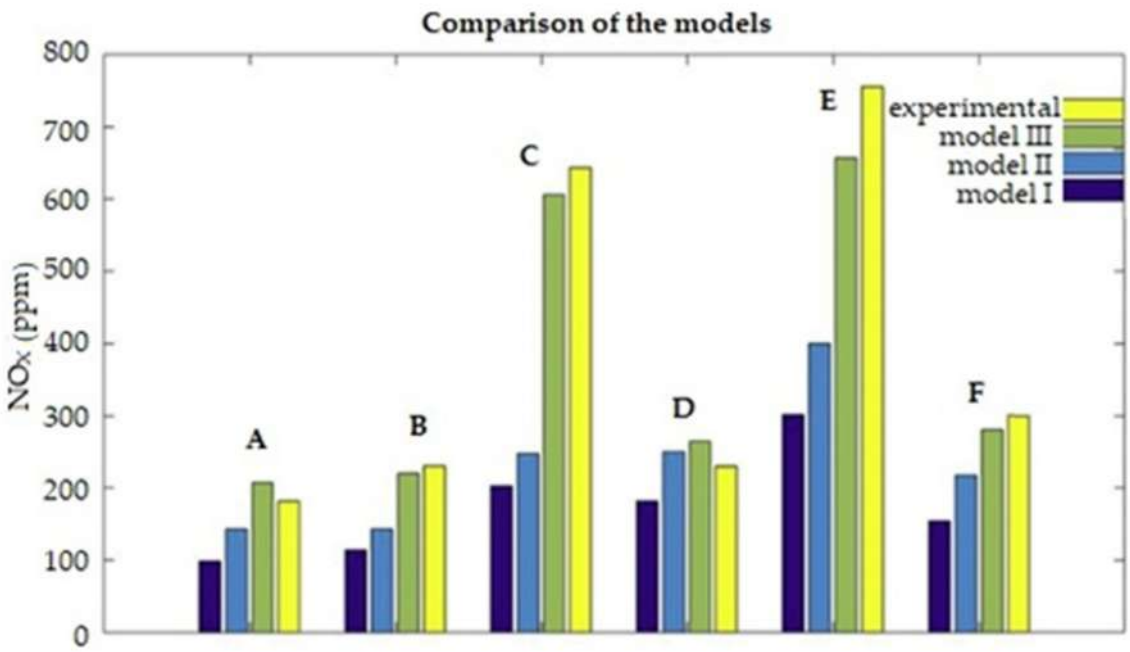

Figure 21. The levels of NOX concentration in the exhaust gas determined on the basis of simulation models—models I, II, and III—and on the basis of the experimental results in various, established engine operating conditions, defined by the letters A, B, C, D, E, and F, signifying the concentrations of H2 in HCNG composition as follows: A—0% H2; B—5% H2; C—7% H2; D—10% H2; E—13% H2 [73].

The emission of other undesirable chemicals, such as HC, CH_4, CO_2, and CO, in most of the studies cited in this article, did not much changed or decrease the overall level of emissions. Emissions always depend on the degree of hydrogen enrichment, and not in a linear way.

During research on the effects of fueling an engine with HCNG gases of different compositions (from 0 to 13% hydrogen content) when using biofuel as the ignition initiating fuel, the lowest emission values of harmful exhaust components and the highest thermal efficiency were achieved with the use of HCNG containing 10% hydrogen [52]. Emission of CO_2 was the lowest for the hydrogen concentration of 10%, which proves the high combustion efficiency of fuel of this composition. By comparison, HCNG with a hydrogen content ranging from 18 to 20% is used to power India's public transport fleet [69]. Figure 22 presents the emissions of toxic exhaust components produced by methane fuel and Hythane, showing some of the benefits of using Hythane.

3.3.4. HCNG Applications

There is a negative side of the popularity of alternative fuels such as CNG and HCNG: they are commonly portrayed as fuels only marked by superlatives. In many countries, such as India [64] and China [78], the implementation of these fuels for common use in automotive transport is very intensive, but knowledge about them is lacking, sometimes promoting false information about them.

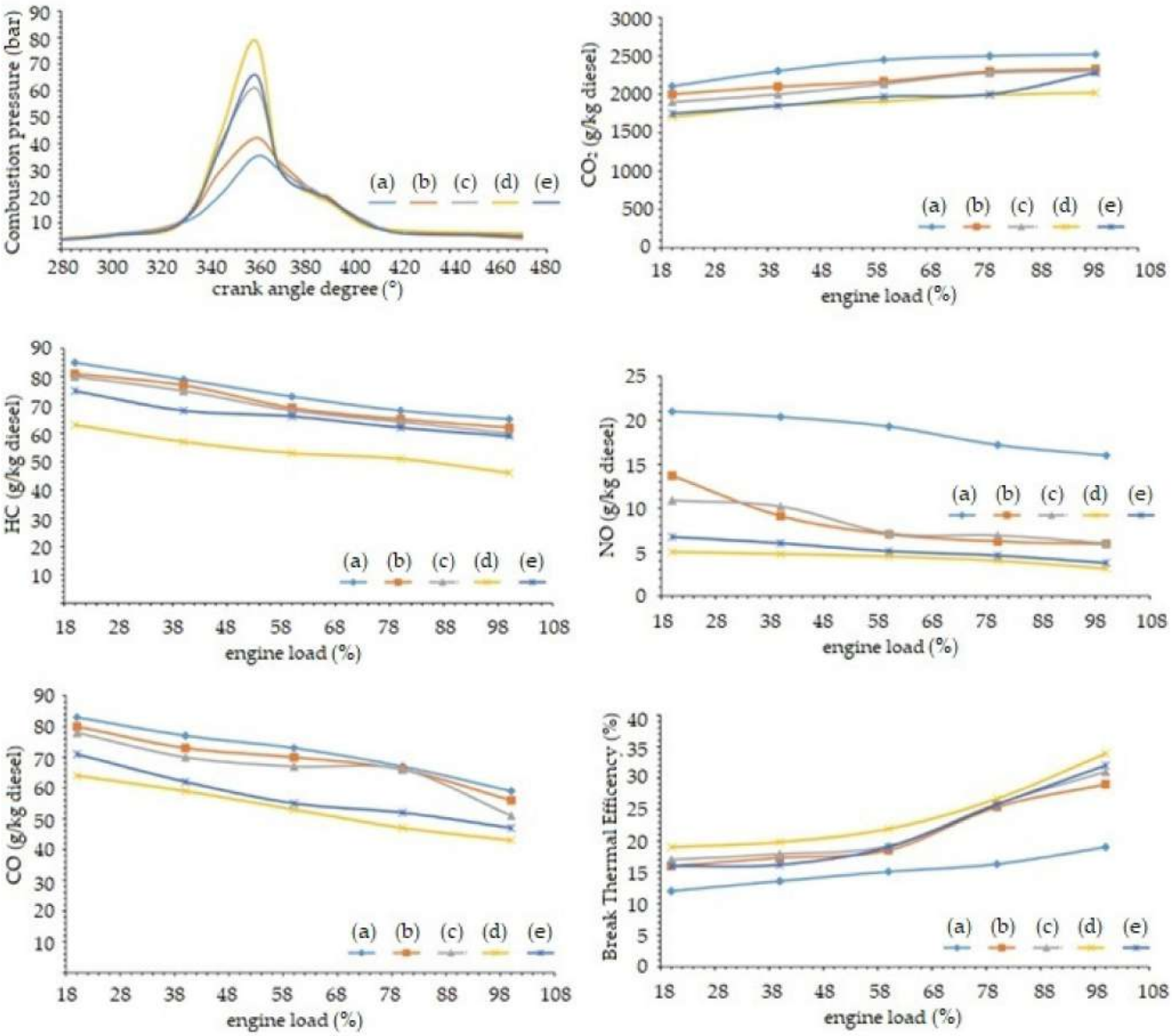

Figure 22. Pressure, emission and efficiency of an engine fueled with HCNG fuel with the following composition: (**a**) —0% H_2, (**b**)—5% H_2, (**c**)—7% H_2, (**d**)—10% H_2, (**e**)—13% H_2 with dual-fuel supply system [72].

Achieving low emissions requires significant interference in the design of the internal combustion engine, which must be preceded by research and optimization of the design for the combustion of a new type of fuel. In both single-fuel diesel engines and dual-fuel diesel engines (RCCI), it is necessary to adapt the combustion chamber to run on a new type of fuel. The shape of the combustion chamber in RCCI engines is very important for the formation of harmful chemical compounds contained in exhaust gases [6,79,80]. The degree of swirling of the air–fuel mixture changes, which makes it possible to completely burn the fuel. After installing the gas installation, engine operation should be optimized with the use of an engine dynamometer and an exhaust pollution measurement system (which is not always the case). Reducing carbon dioxide emissions when running on natural gas or Hythane does not always mean reducing GHG emissions. During the start-up of natural gas engines (CNG/LNG), in the case of unsuccessful ignition, methane is emitted in an amount related to the fuel consumption. Its emission during unsuccessful attempts to start the engine contributes to the overall CH_4 emissions of the engine during its operation. This gas is 20 times more dangerous than carbon dioxide [7]. Much relevant data on HCNG combustion are still being gathered, but information about the differences in emissions

among typical fuels is very popular, especially for dual-fuel engines. Table 8 shows the emissions of a typical, modern, medium-speed marine engine without an exhaust gas treatment system.

Table 8. Emissions produced in the shipping industry: a typical medium-speed engine produced after the year 2000 without a post-treatment system. Emissions depend on fuel quality and engine type [7].

Fuel	SO_X (g/kWh)	NO_X (g/kWh)	PM (g/kWh)	CO_2 (g/kWh)
Heavy Fuel Oil	13	9–12	1.5	580–630
Gasoil	2	8–11	0.25–0.5	580–630
Diesel	0.4	8–11	0.15–0.25	580–630
LNG	0	2	~0	430–480

Researchers try to determine the speed of flame propagation in a combustion chamber (laminar burning speed correlations), which makes it possible to properly design the combustion chamber of an engine [81]. The influences of such obvious matters as the injection pressure of the pilot dose of diesel fuel on the BTE of the thermal efficiency of the engine are still being investigated [82]. This topic has been examined with different fuel injection angles and different combustion chamber shapes [6,79,80]. These parameters have the greatest impacts on the level of swirling of the fuel–air mixture, which for different fuel mixtures has different impacts on efficiency. The operation of an engine with HCNG fuel, compared to the use of a standard engine with a timing system (negative valve overlaps), designed for operation with conventional fuels such as diesel or gasoline, lessens the efficiency of using the alternative fuel, that is, HCNG, as shown in [83]. Only the effects of the use of a double pilot dose in dual-fuel engines have been investigated [84]. All of these factors affect the emissions of an engine powered by these fuels. Taking them into account should be an obligation when publishing any material related to these fuels and potential investments. The cost of such major engine modifications is many times greater than just supplying the engine with gas fuel and enabling its failure-free operation. Alarmingly high emissions of harmful exhaust components have been demonstrated in research [72]. The study [85] showed that the emissions of city buses converted to use CNG significantly exceed the assumed emission standards. The modifications to the engines are very often based on the conversion of a diesel engine to a petrol engine. It has been proven that the use of a laser ignition system in place of spark plugs allows one to increase the engine's operating parameters, which proves the non-homogeneous combustion of the fuel–air mixture in diesel engines powered by such a gas [86]. The use of HCNG is possible in all applications where natural gas is used in compressed form. It is also possible when natural gas is doped with hydrogen at the stage of feeding the engine. In this case, it is possible to use LNG and enrich it with hydrogen. Most often, Hythane is used as a complete fuel, delivered and stored as is.

Adding hydrogen to natural gas at the stage of supplying the engine with both fuels is possible in cases where the hydrogen is stored independently as a second source of power, and when hydrogen is produced or supplied to the engine from another source. This can be achieved in stationary engines powered by natural gas from the municipal network and powered by hydrogen from self-production—for example, from industrial processes that are carried out locally (or produced from renewable energy sources). This would allow companies and industrial plants that use internal combustion engines to produce energy to reduce GHG emissions, which will help any such company's reputation and provide the possibility of receiving subsidies for the development and construction of this type of installation. In the case of Hythane on ships, it would also be possible to transport both fuels independently (or to generate hydrogen onboard the ship), but the use of such installations will be justified when emission regulations begin to urge shipowners to reduce GHG emissions below a level which can be met by using Hythane to power marine engines.

Natural gas transported on large ships is always stored in liquid form, in the form of LNG. This is because, in addition to the different condensation temperatures of methane and hydrogen, and their different densities which make it difficult to mix these fuels in liquid form, the problem is the very simple structure of hydrogen atoms, whose quantum properties have a large impact on their physicochemical properties. Hydrogen (H_2) exists in two forms with slightly different physical properties. Molecules can exist in one of two forms—as orthohydrogen or parahydrogen. The form depends on the orientations of proton spins in the hydrogen atoms. The orthohydrogen molecules have parallel spins, whereas the spins of parahydrogen have antiparallel spins. In other words, in the orthohydrogen, the spins of the protons are in the same direction, and in the parahydrogen, the opposite. The chemical properties of both of these forms of hydrogen are the same, but they differ significantly in thermodynamic and physical properties. The contents of individual forms of hydrogen gas strictly depend on the temperature—the closer it is to absolute zero, the higher the content of parahydrogen. At room temperature, the orthohydrogen to parahydrogen ratio is already three to one. Table 9 clearly shows the differences in the physical and thermodynamic properties of different forms of hydrogen [50]. It differs significantly in density in various forms—pure parahydrogen has a density that is about 22% greater than that of hydrogen in the surrounding air. The specific heat of parahydrogen is also several percent higher, and the entropy of parahydrogen is less by about 9%. The other properties are similar to each other.

Table 9. The physical and thermodynamic properties of different forms of hydrogen [50].

Properties	Hydrogen	
	Para-Hydrogen	**75% of Ortho- + 25% of Para-**
Density in 0 °C, 10^3 mol/cm^3	0.0546	0.0446
Cp in 0 °C, J/(mol × K)	30.35	28.59
Cv in 0 °C, J/(mol × K)	21.87	20.3
Enthalpy in 0 °C, J/mol	7656.6	7749.2
Internal energy in 0 °C, J/mol	5384.5	5477.1
Entropy in 0 °C, J/(mol × K)	127.77	139.59
Thermal conductivity in 0 °C, mW/(cm × K)	1.841	1.740

There are way of changing orthohydrogen into parahydrogen with the help of catalysts—paramagnetic ones—or it can be done by lowering the temperature of the hydrogen [87]. However, these are long-term processes, as a result of which heat is released (at the level of 527 J/g H_2). The "instantaneous" condensation of normal hydrogen (75% orthohydrogen and 25% parahydrogen) would result in liquid hydrogen having the same proportions of both forms of hydrogen as in hydrogen under normal conditions. Only under the influence of long-term storage of hydrogen in liquid form would the ortho–para conversion take place spontaneously, and then the parahydrogen content would reach over 99.8% under the equilibrium conditions. The heat released changes with the temperature of the liquid hydrogen and increases as the temperature of the converting hydrogen decreases. Hence, difficulties in storing hydrogen in liquid form may arise, because the heat of hydrogen vaporization (447 J/g H_2) at low temperatures is lower than the heat of ortho–vapor transformation and causes evaporation of condensed hydrogen and liquid losses, even in the best insulated tanks. Various types of catalysts have been used to convert orthane into parahydrogen before it is stored.

Installations of this type designed for the preparation of hydrogen for the purpose of storing it together with natural gas have not been described in scientific publications. Due to the problem of ortho–para hydrogen transformations described here, currently no technical solutions are used to prepare "HLNG" fuel for distribution.

3.3.5. Hydrotreated Vegetable Oil Properties

Hydrotreated vegetable oil (HVO) is a form of synthetic diesel, an alternative renewable fuel for CI engines that is produced from vegetable fats and oils. Hydrogen in HVO production is used as a catalyst in the creation process instead of methanol in regular FAME biodiesel production. This makes HVO far more stable. In many ways, HVO is actually superior to other forms of fuel. Table 10 compares the different characteristics of regular diesel, FAME, and HVO biodiesel. HVO's impacts on the exhaust emissions of these fuels are helpful. It has low solubility in water and the cetane number is high. Large differences between the cetane numbers of conventional diesel fuel and HVO would require some adjustments in the engine control to compensate for the fuel igniting earlier in the cycle, though this does not always happen. The lubricity is very low due to the absence of sulfur and oxygen compounds in the fuel; therefore, a lubricating additive is required [88]. The heating value per mass of HVO is higher due to the higher hydrogen content. The density is lower due to the paraffinic nature and the lower final boiling point. The cold properties of these fuels can also be controlled to meet the local requirements by adjusting the severity of the process or by additional processing. In any case, HVOs and paraffinic synthetic fuels will require good properties for a viable future, because fuel requirements set by legislation and fuel standards are becoming more stringent, due to new regulations for exhaust emissions, fuel economy, and onboard diagnostics [89].

Table 10. The properties of diesel fuel components obtained from fossil and bio-renewable resources [88].

Properties	Diesel Fuel Made from Crude Oil Processing	FAME Biodiesel	HVO
Oxygen content (%)	0	11	0
Aroma content (% (m/m))	15 ÷ 30	0	0
Density in 15 °C (kg/m^3)	830 ÷ 840	880	770 ÷ 780
Sulfur content (mg/kg)	<10	<5	<5
Heat value (MJ/kg)	43	38	44
Cloud point (°C)	−15 ÷ 0	−5 ÷ +15	−25 ÷ −10
Boiling range (°C)	180 ÷ 360	340 ÷ 355	180 ÷ 320
Cetane numer	50 ÷ 55	50 ÷ 53	70 ÷ 90

Table 11 presents a comparative effect of the use of HVO or FAME on the operation of a typical diesel engine. Virtually all operational factors favor HVO.

Table 11. The influence of the type of biofuel on the operation of a diesel engine [88].

Factor	Hydrocarbon Biofuel from the HVO Process	Biofuel B100 (FAME)
Engine type	Every diesel engine	Engines created for working on this type of fuel only
Ignition properties	At the level of refinery diesel fuel	Similar to diesel fuel
Lubricating properties	Lack. the use of lubricating additives is required	Very good

Table 11. *Cont.*

Factor	Hydrocarbon Biofuel from the HVO Process	Biofuel B100 (FAME)
Low temperature operation	Normally, adequately to the low-temperature properties of the fuel	Difficult compared to normal diesel fuel, inferior low temperature properties
Atomization and evaporation of fuel in the combustion chamber	At the level of refinery diesel fuel. Fuel injector systems are factory set for the typically diesel fuel properties.	Deteriorated. Higher density and viscosity and lower volatility
Engine power	Rated	Slightly smaller
Fuel consumption	At the level of refinery diesel fuel	Slightly bigger
Tendency to form precipitates and lacquers in the supply system	At the level of refinery diesel fuel	Probability of increasing the amount of sediments
Emission of components harmful to health	At the level of refinery diesel fuel	Lower emissions of carbon monoxides and particulate matter. Higher nitrogen oxide emissions are possible
Effects on materials used in the engines fuel system and in the distribution system	At the level of refinery diesel fuel. Used materials are compatible with hydrocarbon fuels	Some plastics and painted surfaces dissolve upon contact with FAME
Influence on engine oil	At the level of refinery diesel fuel. Engine oils are compatible with hydrocarbon fuels	More frequent engine oil changes

Production and storage [11].

The biocomponents obtained in the process of hydroconversion (Figure 23) of vegetable oils and animal fats, known as HVO, are hydrocarbon paraffin fractions obtained as a result of the catalytic process of the hydroconversion of triglycerides of fatty acids present in vegetable oils and animal fats. If non-food or waste materials are used, they are classified as second-generation biomass components. Triglycerides can be divided into simple ones, containing three identical fatty acids, and complex ones, consisting of three different fatty acids. According to the presence of and number of double bonds, fatty acids can be divided into saturated, monounsaturated, and polyunsaturated [11]. The physicochemical properties of a triglyceride depend on the properties of the fatty acids, and in the case of complex triglycerides, on the relative positions of the individual acids in the triglyceride molecules. Saturated fatty acid triglycerides have high freezing points (above 30 °C) and better oxidative stability than unsaturated acid triglycerides, which are more reactive due to the presence of double bonds. The hydroconversion process of fatty acid triglycerides consists of three steps:

- Hydrogenation of double bonds in triglyceride hydrocarbon chains;
- Breakdown of the triglyceride molecule into propene and fatty acids;
- Decomposition of fatty acids into n-paraffinic hydrocarbon. Water, carbon dioxide, and carbon monoxide are produced; and propene is hydrogenated to propane.

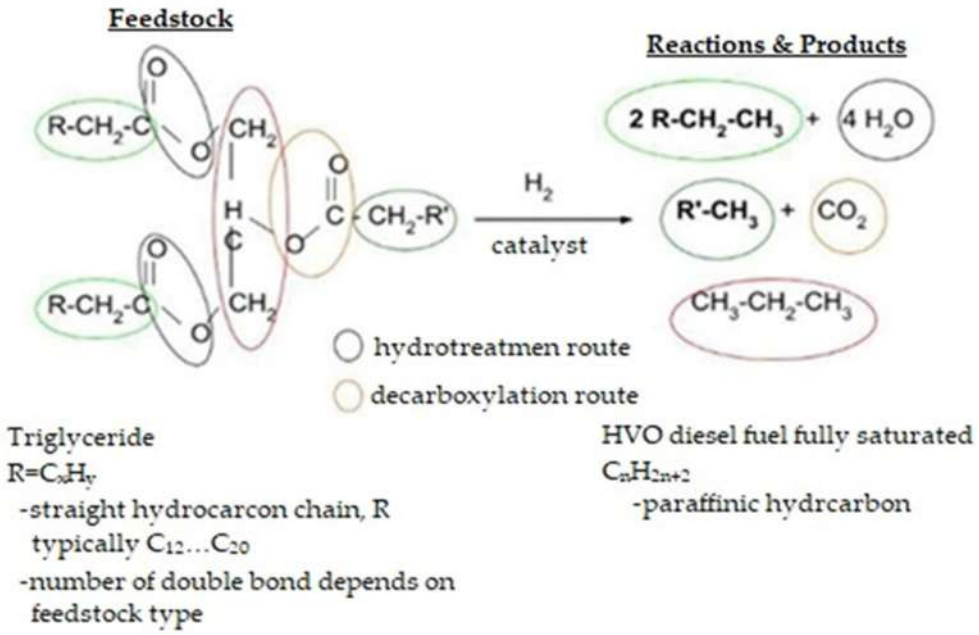

Figure 23. Rapeseed oil's hydroconversion process [90].

Figure 24 shows the chemical molding process and composition of HVO.

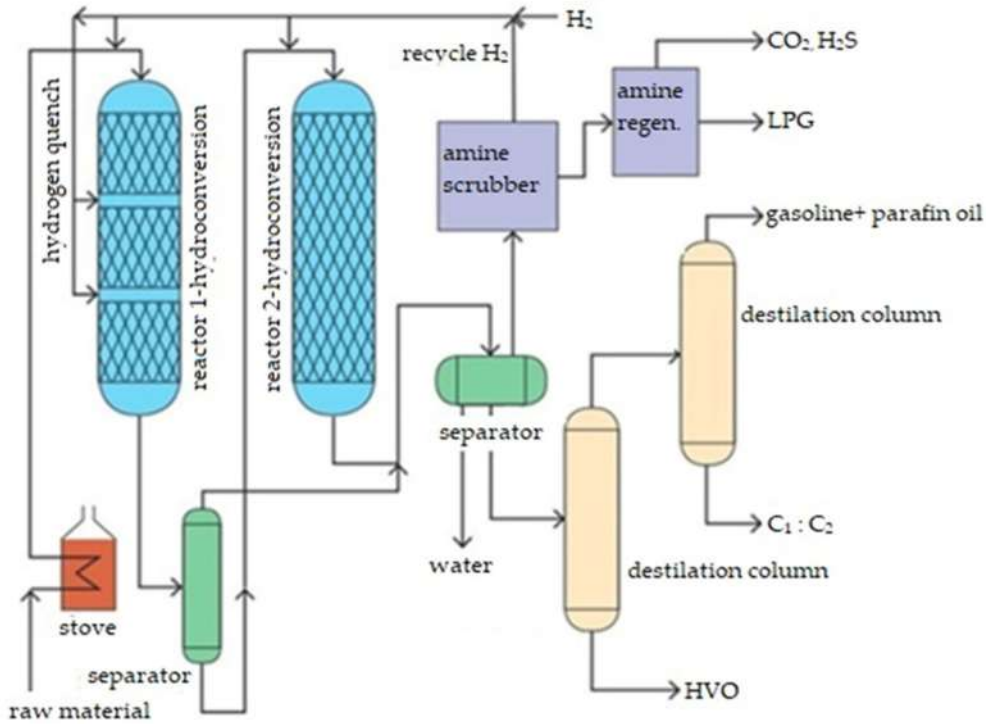

Figure 24. The chemical formation and composition of HVO [11].

The pioneer in the research of the hydroconversion of vegetable oils and animal fats into hydrocarbon fuel components, and then in the commercialization of this technology on an industrial scale, was the oil company Neste Oil. In Finland, July 2007, the world's first industrial site for the production of high-carbohydrate diesel biocomponents from natural oils and fats was commissioned, under the name NExBTL, with a production capacity of 170,000 tons per year. As a result of the development of this technology, the HVO process is offered by companies such as:

- Axens IFP (Vegan);
- Honeywell UOP (Ecofining—Green Diesel);
- Neste Oil (NExBTL);
- Syntroleum;
- UPM (BioVerno).

In total, the production capacity of the existing industrial HVO installations in the world exceeds 3 million tons per year.

Logistics and Storage Benefits

HVO, compared to FAME, retains very good properties in terms of storage capacity (Table 12). It is less hygroscopic, has no tendency to foam, relatively slowly decomposes, and has no tendency to form a sediment in storage installations.

Table 12. Influences of the type of biofuel on the distribution and storage system [89].

Content	HVO	FAME
Contact with free water	Slight tendency to form stable emulsions	Possibility of decomposition (hydrolysis), formation of permanent emulsions with water, formation of corrosive compounds
Solubility of water in fuel	Low	High hygroscopicity
Tendency to foaming	Low	Higher
Susceptibility to microbial contamination	At a different level	High
Fuel stability during long-term storage	High, low aging process	Rapid degradation of fuel due to aging processes; the need to use effective antioxidant additives
Tendency to form deposits	At a different level	High
Biodegradability	Low, long term decomposition	High, faster decomposition

GHG Emissions

Engine tests have shown environmental and performance benefits of hydrotreated vegetable oils as renewable diesel fuels.

Most of the studies available have shown that HVO usually leads to exhaust emissions benefits with normal engine performance. Substantial reductions in THC, NOx, CO, and HC emissions have been reported with the use of HVOs [88]. Official information from the Neste Renewable Diesel Handbook says that 90% of CO_2 emissions are reduced when using HVO as opposed to regular diesel fuel [90]. Significant impacts on PM, HC, and CO engine emissions were observed by [89]. They reported the emission results with biodiesel, HVO, and regular diesel in engines tested in testbeds and city busses. In most cases, all regulated emissions, such as NOx, PM, CO, and HC, decreased with HVO compared to regular diesel, although an increase NOx was also observed [91]. According to another study [92] where exhaust emission tests were performed with different engine sizes (heavy trucks and passenger car engines), reductions of particulate mass, carbon monoxide (CO), and hydrocarbon (HC) emissions were noticed. A study by Soo-Young [93] in 2014 found that the use of HVO reduced NOx, PM, HC, and CO emissions without making any changes to the engine or its control. According to the study reported in [94], where exhaust emission tests were performed on heavy duty turbocharged diesel engines, average reductions in all emissions were clear. The most significant reduction of about 35% was measured in smoke. With HVO, emissions of NOx were reduced by about 5%. A reduction in specific fuel consumption by 5% was also noticed. Therein, the advantages of using HVO also appeared in relation to typical biodiesel. The study showed averages of all speeds and

loads with default injection timing. Emissions of HVO and biodiesel compared with to normal diesel fuel are shown on Figure 25.

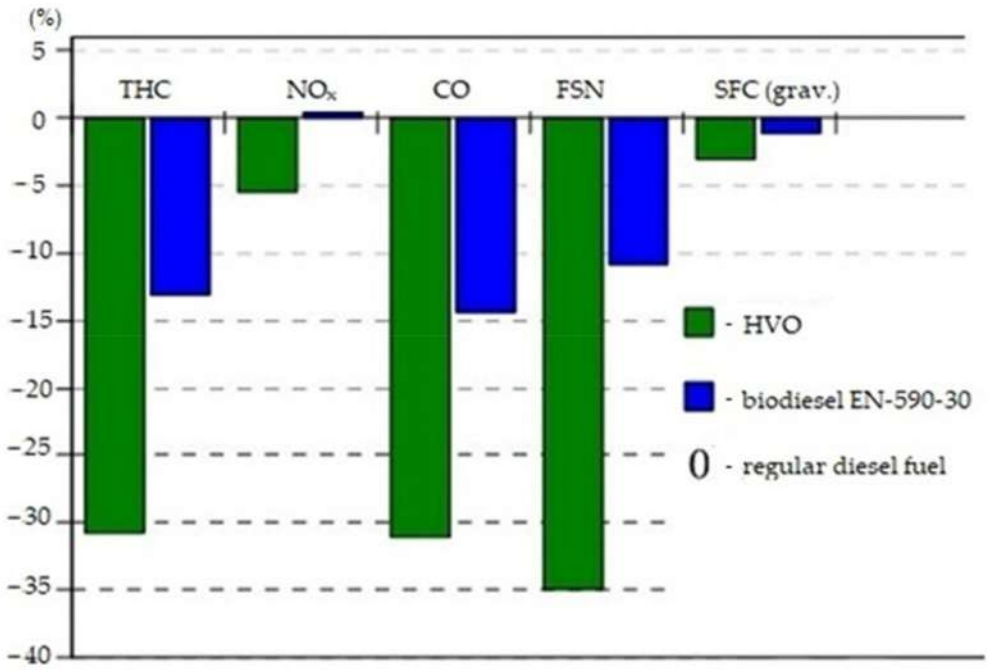

Figure 25. Emissions of HVO and EN 590-30 compared with those of regular diesel fuel [94].

In general, most of the studies that examined HVO found it to be a fuel which can help to reduce GHG emissions. Another observation is that in most cases, HVO i=was examined under steady-state engine operation or in a vehicle, and as a result there is a lack of information from transient conditions, which are experienced for most of the operating life of vehicle. Furthermore, HVO was investigated in existing engines only, by changing the fuel. Default engine settings are not optimal for HVO combustion because of its slightly different properties.

There is also no information about the use of HVO as a pilot fuel in the context of dual-fuel systems. Of course, this should be investigated in the near future. Nevertheless, the available knowledge about HVO indicates that there is potential for the use of this fuel in the context of modern RCCI systems to reduce overall GHG emissions.

3.4. Hydrogen-Enriched Alternative Fuels—Summary

In RCCI engines, a high energy share of a poorly reactive gaseous fuel allows one to reduce exhaust opacity and reduce the emissions of harmful exhaust components, such as solid particles, NO_X, CO, CO_2, and NMHC [95]. As the engine load increases, the share of diesel fuel (HVO) in the proportion of fuels used increases. This leads to slight increases in the emissions of harmful exhaust components, but it also has an impact on the emission of carbon dioxide. The combustion of diesel fuel produces more CO_2 than in the combustion of CNG [96]. The emission of many harmful exhaust components depends on the energy proportions of the fuels used in RCCI engines (it is possible to "control" the emissions). The introduction of appropriate settings for the operation of such an engine may decide whether or not to meet a specific exhaust emission standard, without physical interference in its construction.

In RCCI engines, despite the positive impacts of a high share of CNG or HCNG on CO_2 emissions and some harmful exhaust components, too much of it in the combustion process leads to reductions in performance for the engine's operating parameters. Figure 26 shows the operating parameters of the engine powered by both fuels at its maximum load compared to the engine operating only on diesel fuel [97].

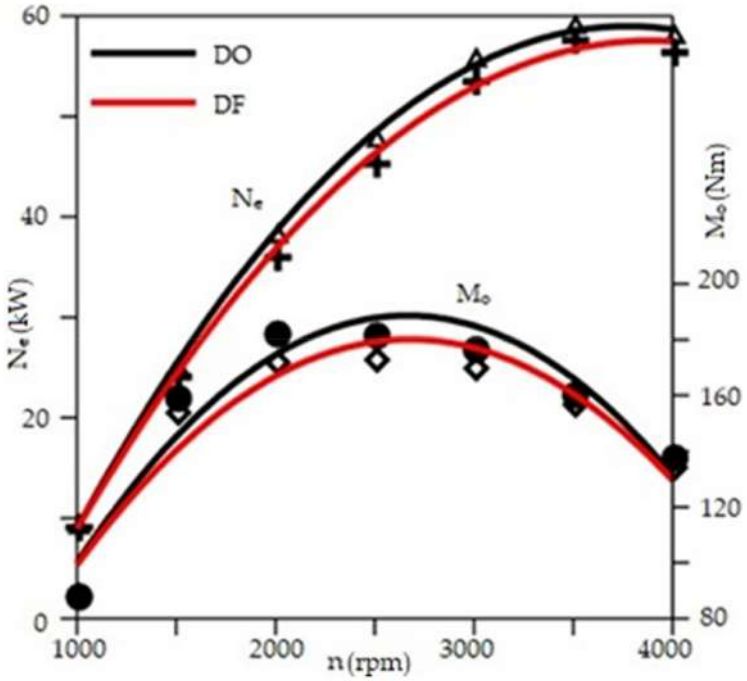

Figure 26. Parameters of a CI engine powered by diesel fuel (black) or CNG and ON (red) with maximum output power and torque [97].

Tests were usually carried out on CNG and diesel fuels. This type of setup in RCCI engines is common in scientific works. Given the trend of reducing GHG emissions, it is advisable to carry out tests with this type of engine with alternative fuels that allow for the maximal reduction of CO2 emissions, and to check these fuels' impacts on the operating parameters of the engine, because there are no studies available in the literature that would allow one to determine the impacts described in this article of fuels on the performance of RCCI engines.

4. Discussion

Certain considerations in the preparation of this article left the authors in doubt about several points, and we address them here.

The work [61] contains quite specific information on the combustion characteristics of hydrogen mixed with natural gas. It was an original work by one author, which is currently not available among the other works by this author. Other publications by this author, where he appears most often with other authors, do not in most cases concern the topics we discussed, and in the case of one topic related to HCNG, they did not present such bold theoretical theses, but numerical and empirical studies and their results. Although the author's publication [61] was a source of very interesting data, we have some doubts as to the validity (2012).

Papers looking at specific results of tests on HCNG with a specific composition are very difficult to compare with each other. The composition of fuels should be determined by the regulations governing its chemical and physical parameters. In many countries, fuel such as HCNG is not registered, so there are no regulations governing its composition. The composition itself may therefore be determined by researchers in a way that differs from other researchers, and the method itself may not be disclosed, because showing all information in scientific works is neither recommended nor common practice. Supposed differences in the composition of HCNG gas, which theoretically can be compared with each other, may in practice turn out to be incorrect. For this reason, one should be very careful when comparing results from different papers.

As already described in detail in the article, natural gas itself can significantly differ in chemical composition depending on the source, and the creation of Hythane is based 100% on this base fuel. The problem with the formation of ortho- and parahydrogen in hydrogen fuel has also been described in detail; it also introduces some inaccuracies in the physicochemical parameters of the final fuel. The use of even chemically pure hydrogen can lead to the use of hydrogen with undesirable physical properties for a given application. Hythane is a fuel that not only widely used yet, and it is very strongly dependent on the "component fuels" that create it; therefore, its parameterization is a noticeable problem for researchers, so we are not convinced that the current information on the precisely parameterized composition of HCNG can be relied upon.

There is also a lack of studies in the scientific literature that describe the use of Hythane in a dual-fuel or RCCI engine. This also applies to the possibility of using HVO as a pilot dose. In the context of the HVO, its use in "dual fueling" was also not tested, and that should be done. In a dual-fuel system, the focus is on replacing liquid fuel with gas fuel as much as possible, so the issue of pilot fuel becomes secondary. Nevertheless, HVO should be examined in terms of its ability to initiate combustion in RCCI and the emissions it puts out in this mode of operation. This leads to the conclusion that despite the analysis carried out, further research is needed in regard to the use of this type of low-emission alternative fuel in high-performance dual-fuel engines. Studies heading in this direction can be found. Take [98] as an example: studies of an engine powered by HCNG and B100 fuel were performed. Additional research is still needed on this subject though. The existing research of this type could not always be linked with the research on emissions of exhaust gas components related to the issues of global warming.

Due to the lack of a sufficient amount of research of this type, we have decided to also support these efforts with significant research on fuels of this type in single-fuel internal combustion engines, and further research will allow us to conduct a far more in-depth analysis on this subject of interest.

Finally, it is necessary to mention the extensive bibliography related to the subject of the dual-fuel engine itself. There is a lot of information about it. Readers may find the references on this subject helpful, but because the subject of the RCCI engine itself is not the focus of this article, the items related to it have been limited to the minimum necessary, which proves the wealth of literature available on this subject [6,79,80].

N_2O emissions are important for global warming, but the N_2O concentrations in IC engines' exhaust gases in comparison to CO_2 concentrations are insignificant, and the number of researchers focusing on that topic is small. The greater problem is CH_4, the emission of which is proportionally more common, and with the advent of new regulations limiting the emission of harmful exhaust components, compliance with them may become problematic when fueling an engine with fuel based on natural gas.

The latest Euro 7 scenarios assume the introduction of a limitation of methane in the exhaust gas, measured as a separate component of the exhaust gas. Its permitted emission level differs in the two main scenarios envisaged for the new Euro 7 standard. Tables 13 and 14 show the emission limits for light and heavy-duty vehicles, taking into account Euro 6, which is already in force, and the two scenarios for Euro 7: A and B.

Table 13. Euro 7 emission limits scenarios—LDV in mg/km, #/km [99].

Euro7 Scenarios	NOx	SPN_{10}	CO	CH_4 (1)	N_2O (1)	NH_3
EURO 6	60/80 (SI/CI)	6×10^{11}	1000/500 (SI/CI)	-	-	-
A	30	1×10^{11}	300	10	10	5
B	10	6×10^{10}	100	5	5	2

(1) Suggested to limit weighted sum of CH_4 and N_2O instead of separate limits.

Table 14. Euro 7 emission limits scenarios—HDV in mg/kWh and #/kWh [99].

Euro7 Scenarios	NOx	SPN_{10}	CO	CH_4 (1)	N_2O (1)	NMHC	NH_3
EURO VI	460	6×10^{11} (SPN_{23})	4000	500 (SI)	-	160 (CI, THC)	10 ppm ~40 mg/kWh
A	120	4×10^{11}	1500	100	50	50	20
B	40	1×10^{11}	400	50	25	25 (2)	10

(1) Suggested to limit weighted sum of CH_4 and N_2O instead of separate limits; (2) Impact of HC-burner for EAT heating on NMHC emissions and durability for CNG tbd.

The predicted emission limit for methane is very strict and may prevent the use of CNG and LNG as standalone fuels, as many engines powered by these fuels may have serious problems with meeting such a stringent standard. Depending on which scenario is finally introduced by the EU, alternative fuels such as HCNG and HVO may have a significant impact on the possibility of meeting them, thanks to the possibility of effectively reducing greenhouse gas emissions caused by internal combustion engines.

5. Conclusions

Alternative fuels could be the key to reducing CO_2 emissions and other greenhouse gases in the combustion process.

Researchers regularly write about the advantages of using hydrogen as a fuel for internal combustion engines. For example: "A hydrogen fueled internal combustion engine has great advantages on exhaust emissions including carbon dioxide (CO_2) emission in comparison with a conventional engine fueling fossil fuel" [100]. One must bear in mind the difficulties of using it on its own. However, fuels rich with it enable reductions in CO_2 emissions.

Both HCNG and CNG are commonly researched and used in IC engines. The amount of CNG research by far exceeds the amount of HCNG research, and the same is true of the number of applications in practice. Both of these fuels have shown high potential for reducing CO_2 emissions in tests. HCNG has been more impressive, and CO_2 emissions decreased with the content of hydrogen in the fuel. There is also a lack of widespread research on N_2O emissions, which indicates further uncertainty about the climate impact of this type of fuel.

The studies that were successfully analyzed do not strictly concern the influences of the analyzed fuels on global warming. The works that treated the analyzed fuels as sources of power for dual-fuel or RCCI engines were also rare. In the case of the fuels selected by the authors, the co-combustion of which would have the greatest potential to reduce CO_2 emissions, it was not possible to obtain a research paper in which both these fuels (HVO and HCNG) were used in RCCI engine emission tests. Despite the lack of consistent reference data, the overall picture of the knowledge available indicates the validity of further work in the context of RCCI with a dual-fuel supply of HCNG and HVO (as a pilot dose). All data on the emissions of the combustion of these fuels in mono-fuel systems report measurable benefits through reduced GHG emissions. Further work should be empirical, or measure engine performance and emissions on laboratory engine beds, in dyno testing, and in tests in real cars travelling in traffic.

We noticed a problem with the parameterization of the physicochemical properties of gaseous fuels. The spread between the various fuels is big; the problem is strongest for NG and HCNG fuels. Big differences in parameters are not only allowed by regulations. Large differences in the parameters of some fuels are not only allowed by the regulations, but also often in cases of new fuels, such as HCNG, are not covered by them at all. This is certainly a fact used by their producers. Even in research, there are noticeable differences in the chemical composition of different HCNG, which also did not agree with the theoretical calculations carried out by the author. For this reason, the properties of natural gas taken from one of the cited works were used in the Table 15 below.

Table 15. Summary table with a comparison of important fuel parameters [11,66–68,101–104].

		Petrol	Diesel	B20	B100	HVO	LPG	E100	M100	CNG	HCNG15	HCNG30	HCNG50
C/H ratio	% of weight	9	7.26/6.73	6.63	6.32	5.49	4.65	4	3	3	2.77	2.52	2.15
	molecular	~3:4	~3:5	0.56	0.53	0.46	~2:5	1:3	1:4	1:4	0.23/0.24	0.21/0.22	0.18/0.19
Hydrogen weight content [%] **		10	12 /(12.96)	12.8	12.17	15.4	17.7	13	12.5	25	26.5 (25.75)	28.39 (26.5)	31.75 (27.5)
Carbon weight content [%] **		90	87.2/(86.93)	84.87	76.96	84.6	82.3	52	37.5	75	73.5 (74.25)	71.61 (73.5)	68.25 (72.5)
Approximate CO_2 emission level [g/MJ] [66,67] **		~69 /(71)	~74 /(69)	~73 *	~70	~70 *	~63 /(61)	~62 /(64)	70 [101]	56.1 – 55.35	55 – 54.3	53.56 – 52.85	51.05 – 50.37
CH_4 emission potential		low	low	low	low	low	low	low	low	very high	high	high	medium/high
N_2O emission potential [103]		Very high	medium	medium	high	low	very high	high	high	medium	low	low	low
mainly emissions of another substances:		HC, CO, CO_2, NO_2, NOx.	HC, CO, CO_2, NO_2, NOx.	HC, CO, CO_2, NO_2, NOx.	HC,CO_2, NO_2, NOx.	HC, CO, NO_2, NOx.	HC, NO_2, NOx.	HC, NO_2, NOx,	HC, NO_2, NOx,	NO_2, NOx, CH_4.	NO_2, NOx, CH_4.	NO_2, NOx, CH_4.	NO_2, NOx.
Source [66,67]+		[104]	[101,102]	[102]	[102]	[11]	[101]	[101]	[104]	[68] *	[68] *	[68] *	[68] *

* Due to a different chemical composition of CNG in every country and from every source, in the table, NG-based fuels are pure. The compositions of NG and H_2 are in volumetric proportions from [62], and the items dependent on others, for example, the C/H ratio or the specific content of hydrogen in HCNG15, were calculated by the authors. ** A few values are reported twice, as there are differences among the sources.

6. Final Summary Conclusions

1. The aim is to minimize GHG emissions from internal combustion engines. Alternative fuels are being developed to reduce carbon dioxide emissions by reducing the amounts of carbon therein.
2. RCCI engines, due to the specificity of their functioning, allow for very effective use of the fuels burned, while ensuring low GHG emissions.
3. Gaseous fuels based on natural gas have very diverse chemical compositions, but show the greatest potential to reduce CO2 emissions. These fuels can be enriched with hydrogen, which further improves their emission properties. There are also highly reactive fuels (when used as pilot dose fuels), and hydrogen is used for refining them.
4. There have been a very small number of empirical studies examining the feasibility of using low-emission fuels in RCCI engines.
5. An issue worth discussing is the emissions of other greenhouse gases (N2O and CH4), on which should be focused by conducting further research and reviews in the context of the use of alternative hydrogen-enriched fuels in RCCI.

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Safety in the use of car gas fuel installations

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The safe use of gas fuel installations in vehicles is guaranteed by legal and technical aspects. These topics are included in this study. The regulations ensuring safety in the operation of the mentioned fuel systems serve as a solution for potentially hazardous situations. The components of propane-butane and CNG fueling systems are designed, manufactured and tested to maximize their safety of use. The regulations [6–8] define the guidelines for the arrangement and assembly method of the system components, and additionally, the assembly service itself may be performed only by an authorized workshop with granted permissions. Installations using gaseous fuels are safer than conventional fuels in the event of a collision or fire, as long as the user of the installation does not gross negligence in operation and maintenance. The article also discusses the context of the restriction in access to the car infrastructure for cars powered by gaseous fuels.

Key words: *safety, gas, fuel, LPG, CNG*

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1. Introduction

The safety of use of vehicles with gas fuel supply installations is influenced by at least several factors: meticulously constructed homologation regulations [6, 7] which force constructions and manufacturers to apply solutions ensuring safe use and correct installation of the installation, as well as proper operation and maintenance, the details of which are regulated by applicable regulations [8]. The absence of tragic accidents related to the use of propane butane Liquified Petroleum Gas (LPG) and Compressed Natural Gas (CNG) as engine fuels indicates that the designed system for approving and controlling automotive gas installations is working properly. Provisions in the Regulations 67 and 110 of the United Nations Economic Commission for Europe (UNECE), according to which gas supply systems are approved, minimize the possibility of hazardous situations related to their use and minimize negative effects in emergency situations, such as a collision. A number of self-acting safety devices make the use of a gas fueled car safe and in this respect the same as with conventional fuels.

2. Installation components

A CNG and LPG supply installations are usually alternative fueling systems for spark ignition and compression ignition engines, closely cooperating with the original fuel supply system. Regardless of the type of primary power supply system used, the gas system must meet its level of original solutions, both in terms of technical advancement, as well as safety requirements and exhaust emissions [4]. Gas systems are divided into those installed in the cars at the factory, installed as an additional power supply system, somewhat "next to" the original factory system and single-fuel systems (for example city buses powered by CNG) [1, 3]. There are great structural similarities in the construction of LPG and CNG installations [1]. Different types of gas installations generally use the same set of basic components conceptually: filler, tanks, pipes and fittings, components in the engine compartment.

2.1. Gas fuel tanks

Gaseous fuel tanks as devices operating with overpressure are subject to the obligation of technical inspection, and their use in cars means that they are built on the basis of detailed requirements contained in the type-approval regulations by UNECE. The safety of the exploited tank is guaranteed by the supervision carried out from the design stage to the obligation of technical inspections [6, 7].

Automobile LPG tanks are made of steel compliant with the EN 10120 standard. They are unalloyed steels with a maximum carbon content of up to 0.2%. The materials used for LPG tanks can be successfully processed (also pressed). In the case of CNG, due to the higher operating pressure, the fuel is stored in steel cylinders manufactured by stamping from a block and hot-drawing, or composite cylinders with or without a braid. Compressed natural gas storage tanks undergo a complex test cycle, including [2, 6, 7]:

- strength of construction material,
- hydraulic pressure test,
- fire test,
- tank impact resistance test (composite),
- bullet resistance (composite).

The parent material of the tank is subjected to tensile and bending strength tests. It consists in taking samples of materials from places strictly defined in the regulations. Gas tanks must show certain behavior in a strictly defined manner [3, 6].

The tanks are also burst tested by hydraulic pressure. The pressure during the test must be suitably built up, and its changes must be recorded over time. In practice, the burst pressure of the tank has much higher limits than those imposed by the regulations by UNECE. The working pressure in the tank during normal operation does not exceed 2 MPa for LPG and in steel tanks 22 MPa, and in composite tanks 35 or even 70 MPa for CNG [3, 6]. The tank must be burst after an appropriate increase in its volume, as specified in the regulations (from 8 to 20% depending on the type of the tank). A very important regulatory requirement

is also the fact that when the tank ruptures, its fragmentation cannot occur (no fragments that could injure someone).

Each manufactured tank is subjected to a hydraulic pressure test. The test consists in filling the inside of the tank with liquid (usually water) and increasing its pressure to the value specified in the regulations. During its execution, the tank may not be permanently deformed or unsealed [2, 3, 6]. The purpose of the test is to check whether the tank with the mounted equipment will not burst as a result of the action of strictly defined fire conditions. The test is performed on a copy representing a given type of gas tank. During the test, the tank is equipped with full fittings. The tank is checked on the heat source located under the tank at a specific temperature. During the test, the temperature of the appropriate places in the mantle is measured [3, 6, 9].

The effects of the shock resistance test, consisting in dropping an empty tank at an appropriate angle to the level from a strictly defined height, are presented below in Fig. 1 [4].

Fig. 1. The effects of the impact test of the CNG composite tank [12]

2.2. Method of mounting tanks

As the gas installation tank is its heaviest element, it is subjected to high inertia forces during a collision. During a rapid change of vehicle speed (e.g. during a collision with an obstacle), these forces try to tear the elements fastening the tank out of the place of assembly. During the collision there are very large deceleration, in the order of more than 20 g. The risk of inertia is enormous, therefore the appropriate mounting of the tanks is regulated by the provisions of UNECE. Gas tanks must be installed with the use of appropriate bolts of strength class 8.8. The fastening bolts are counted only for breaking, their shear is neglected due to the moment with which they are tightened and the resulting high friction occurring in the connection between the tank, its frame (Fig. 2) and the place of installation of the tank. If the tanks are fitted with fasteners welded to them, their welds must withstand the external forces of 30 g in all directions. The authors of the approval regulations assumed that the mounting of the tanks must withstand the load resulting from acceleration of 20 g (acting along the longitudinal axis of the car). In terms of transverse forces, the regulations say that the inertia forces resulting from acceleration of 8 g must be withstand by connecting the tank to the body. In the case of a 100 kg tank, this will correspond to a force of 8000 N acting on the tank as if its weight had increased to 800 kg. Gas tanks must not be installed in the front of the vehicle and in the engine working space. It is allowed to mount the tanks on the vehicle roof [4, 11, 12].

Fig. 2. A frame for a cylindrical LPG tank [9]

2.3. Armature securing the tank

„The 80% valve" is responsible for closing the gas supply during refueling at the fuel level corresponding to 80% of the geometric volume of the tank. This allows the space in the tank to be kept, giving the possibility of changing the LPG volume as a function of the ambient temperature. This valve is not used in CNG storage tanks. The operation of the 80% valve should be checked while refueling gaseous fuel. Overfilling the cylinder in the event of high ambient temperature may lead to activation of the safety valve [2, 6, 7].

From April 1, 2002, after introducing amendments to the regulations by the UNECE, a safety valve is installed in all valve assemblies (aka multivalve). The tank is thus protected against excessive pressure build-up. The gaseous fuel is vented through the ventilation ducts connected to the housing. The gas phase of the fuel always escapes through the valves (safety and fire valves). Liquefied gas escaping through the valves, rapidly expanding, could cause the duct diameters to shrink through frost. Approval tests of the valve verify the correctness of its operation during 10,000 opening cycles without changing the operating characteristics [3, 6, 7].

The increase in the temperature of the tank shell reduces the strength of the material from which it is made. With rapid temperature rise, the operation of the safety valve may not be sufficient to bring the pressure down to a safe level. The fire (fusible) valve opens when the tank shell reaches a certain temperature and drops it in order to avoid uncontrolled deformation or unsealing of the tank. The use of a fire valve is required if the safety valve has a capacity below 17.7 m³/min. This parameter results from and is closely related to the surface of the tank [4, 7, 12].

The overflow valve is activated if the gas flow is too intense (exceeding the engine demand of fuel), which may occur, for example, in the case of damage to the mechanical pressure line leading to the engine compartment. This valve is calibrated to operate when the differential pressure is approximately 90 kPa. The flowing gas "carries away" a mechanical element, most often a ball, which limits the gas flow to a level approximately equal to the engine's fuel requirement. In this way, the tank is protected against

a sudden outflow of gas, which poses the risk of its damage and increases the risk of explosion [2].

An indicator placed on the tank (for CNG cylinders, a similar task is performed by a mandatory pressure gauge), cooperating with a float inside the cylinder. Its indications are "approximate" and are very important in emergency situations for emergency services to assess the amount of fuel in the tank. Installed in accordance with the guidelines, the tank must be placed in such a way that its filling level can be easily read [2, 6, 7].

The operation of the working solenoid valve is determined by the operation of the engine. It closes when the engine RPM signal disappears. The necessity to use it results from the protection of gas flow in emergency situations, for example during a collision/road accident, when the engine is stopped without the will of the vehicle driver [2, 6, 7].

The gas supply from the tank to the engine compartment, when it is required to unseal the high pressure system (during service works), is closed on the cylinder with a manual valve.

The gas flow towards the gas tank only when refueling is provided by a non-return valve (Fig. 3). Its construction uses a steel ball supported by a spring. These elements are located in the refueling channel. The pressure from the fuel dispenser moves the ball away from the valve seat and deflects the support spring. After the pressure drops from the distributor side, the ball is pressed against the seat. This way, the tank is protected against fuel leakage in the event of damage to the pipe from the refueling valve [2].

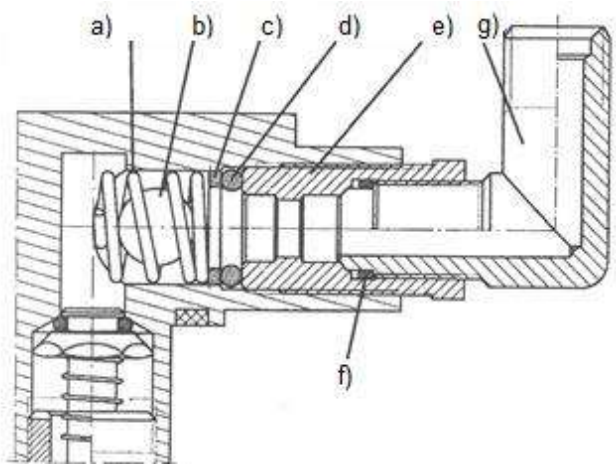

Fig. 3. Multivalve check valve [2]: a) spring, b) ball, c) washer, d) rubber sealing ring, e) reduction sleeve, f) rubber sealing ring, g) elbow fitting

The refueling valve is connected to the multivalve by a line that enables the refueling of gaseous fuel into the tank. Its construction also uses a non-return valve which prevents fuel from flowing out after refueling is finished. The LPG and CNG refueling valves are not compatible with each other (Fig. 4) [2].

All fittings of the LPG tank are placed in a gas-tight casing connected with ventilation ducts. When a leak occurs, the gas escapes through this system outside the vehicle, thus minimizing the risk of gas entering the passenger compartment of the vehicle [2, 6].

Fig. 4. Gas refueling valve: a) CNG, b) LPG – various types [11]

2.4. Gas pipes

Rigid metal or plastic pipes (also flexible) are used to connect automotive components of gas installations in which there is liquid gas. Metal pipes should be seamless, steel or copper (Fig. 5) for LPG and only steel for CNG. Steel pipes should be made of stainless steel or with additional anti-corrosion coating. Rigid pipes made of non-metallic material may be used. The copper wires should be protected with a rubber or plastic cover along their entire length. The internal diameter of the rigid pipe should not exceed 12 mm when used with gaseous fuels, and the wall thickness should be at least 0,8 mm [6, 7].

Fig. 5. Copper pipe for liquid gas fuel [11]

The requirements for flexible hose included in the regulations are very extensive and detailed. In the event of a road collision, when there is a movement of elements in the engine compartment, the priority is to avoid leakage of the gas installation, including its low-pressure part. The rubber gas pipes (Fig. 6) must consist of a smooth-walled inner tube (4) and a reinforced sheath with one or more intermediate layers (no. 1, 2, 3). If the reinforcement is made of anti-corrosion materials, the shield (1) is not required [12].

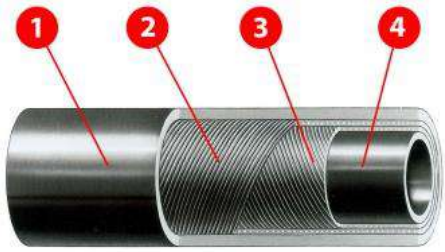

Fig. 6. Flexible fuel gas hose [12]

The inner tube is in contact with the flowing gas. The material from which it is made must be neutral in terms of the action of hydrocarbons. There is a reinforcement in the gas line to maintain the internal bending section and resistance to internal pressure. Synthetic, metal or textile fibers are used in the reinforcement. The outer layer protects the cable against the conditions in the engine compartment and protects against mechanical damage. The inner tube and the sheath must exhibit tensile strength and must be able to achieve a total elongation of at least 250%, according to ISO 37. During approval, the cables are tested for bending resistance. A length of 3500 mm of wire is considered correct when it can withstand a bend test repeated 3000 times without breaking. After the bend test, the hose must withstand the internal pressure without showing any leakage [1, 6].

The provisions of the UNECE regulations define the requirements for low-pressure rubber hoses, which must withstand a maximum working pressure of 0.45 MPa and the operating range at a temperature of –25 to 125°C. The inside of the hose is tested for chemical resistance. It is immersed in n-pentane at a temperature of 23ºC (according to ISO 1817) for 3 days. The change in the volume of the hose sample must not exceed 20%, the change in tensile strength – 25%, and the change in total elongation – 30%. Then, after the cable has been stored in air at 40ºC for 2 days, its weight is checked. The value, in comparison with the initial weight, cannot decrease by more than 5% [4, 11, 12, 18]. The cable sheath (external part) is tested for resistance to n-hexane, in which the sample is kept at 23ºC for 3 days. For sheaths, a greater tolerance of changes in mechanical parameters is allowed than for the internal part of the conduit: the maximum change in volume by 30%, the maximum change in tensile strength by 35%, and the maximum change in the total elongation by 35% [2, 3, 9]. The tightness test in accordance with the method described in ISO 4080 standard consists in connecting a 1000 mm long hose to a tank with liquid propane at a temperature of 23 ±2ºC. The gas loss in the line is monitored for 2 days. A loss greater than 95 cm³ is not allowed [1, 9]. Cables used in gas supply systems are also tested for fire resistance according to DIN 51622, DIN 12642 standards. It is required to maintain the ability to self-extinguish, and during their combustion they should not emit harmful substances [1, 9]. Gas pipes according to ISO 1431/1 must pass the ozone shield resistance test. Samples stretched to 20% elongation are exposed to air with ozone content of 50 ppm and temperature of 40°C for 5 full days. The samples should not show any cracks [2, 4, 6, 12].

The differences between the CNG and LPG lines appear in the context of the lines leading from the tank to the engine compartment. They result from the pressures at which propane-butane and CNG are stored. CNG requires stiff and more durable steel pipes (Fig. 7). The gas moving from the CNG cylinder to the engine still has a pressure several times greater than that of LPG, and any unsealing would be much more violent. Therefore, resistance to external factors as well as internal forces is important.

Fig. 7. High pressure CNG steel pipe

3. Installation assembly

3.1. Arrangement of system components in the vehicle

The provisions in the UNECE regulations, according to which gas supply systems are approved, regulate the position of the system components. In Polish law, annex 9 to the Polish ordinance of the Minister of Infrastructure on "technical conditions of vehicles and the scope of their necessary equipment" regulates the issues related to the installation of gas supply components, the operation of the vehicle with gas supply and its impact on the base fuel supply. The elements of the gas supply system should be arranged in a way that does not hinder the servicing of other car parts. The provisions of the annex concern the protection of gas system components (mechanical damage, corrosion), securing tanks against damage by cargo and that the installation should function properly and safely [2, 12].

In order to maintain the safe operation of the installation, the pipes should be arranged in a way that facilitates their inspection, so that they do not rub against the elements of the vehicle and at a distance of not less than 100 mm from the exhaust system, when a thermal screen is not used. Additionally, the cables must not run in the vicinity of the vehicle lifting points and their fixing must exclude the susceptibility of the cables to vibrations [2].

Gas pipes must not be welded, soldered or joined with snap joints. Standard self-clamping sockets are required to connect the wires, and the number of connectors should be kept to a minimum [2, 12].

Metal pipes connecting elements of the installation structure, which may be subject to a force displacing them during operation, should be shaped into loops with a radius of curvature adapted to the pipe diameter (due to the possibility of breaking) (Fig. 8) [2, 8].

Fig. 8. The method of connecting the evaporator with a rigid copper LPG pipe

The cylindrical tanks can be mounted transversely or longitudinally with respect to the car axis (Fig. 9). The assembly rules apply to all LPG and CNG tanks installed in

cars. Gas fuel tanks cannot be installed in the front of the car or in the engine compartment. They should be installed in such a way that the effects of collisions (mainly at the front and rear of the vehicle) are minimized [2, 8].

There should be no sharp or stiff elements in the vicinity of the installed gas tanks. If the tank is installed under the chassis, its distance from the engine exhaust system (without a heat shield) must not be less than 100 mm. The ground clearance under the tank must not be less than 200 mm.

The guidelines for the installation of gas system components in the engine compartment regulate the minimum distance of the reducer from the exhaust system to 100 mm. The reducer should be installed in a place that is not exposed to temperature fluctuations (drops) while driving. In the vicinity of the reducer, in an easily accessible place, a gas solenoid valve is installed with the coil upwards. The requirement is to install the reducer in such a way that the car's movement does not affect its operation. The controller should be isolated from moisture and mounted away from strong heat sources such as the exhaust manifold or turbocharger. The installation of the gas controller and the wiring harness of the power supply system should be separated from the elements of the ignition system, as this may lead to interference [2, 2]

Fig. 9. Method of mounting CNG tanks: a) composite-aluminum ones on the frame of a truck; b) composite on the roof of a city bus [12]

3.2. Homologation of the method of assembly of the installation

In accordance with Polish law and the provisions of the Regulation of the Polish Minister of Infrastructure of December 24, 2003 on "the approval of the method of assembly of an installation adapting a given type of vehicle to be fed with gaseous fuel", installation services of this type of power supply may be performed by a company with a certificate of approval of the method of installation of gas supply systems it as part of the workshop dealer network. The method of fuel system assembly adapting a given type of vehicle to gas supply, which is to be performed on the territory of the Republic of Poland, should meet certain technical requirements, the fulfillment of which is confirmed in the approval procedure of the installation method adapting a given type of vehicle to gas supply. The entity performing the installation of an installation adapting a given type of vehicle to run on gas may apply for the issue or amendment of a certificate of approval for the method of mounting an installation adapting a given type of vehicle to gas supply. The document is issued by the Polish Transport Technical Inspection Director by way of an administrative decision. The list of companies with approval and cooperating workshops is kept by the Polish Motor Transport Institute [2, 8].

4. Gas installation during a road accident

The discussed construction and legal requirements minimize the negative effects of a road accident in the context of equipping the car with a gas installation. The safety of passengers is ensured by both the functionality of the control electronics and the components made in terms of the materials used. The gas tanks are constructed in a way that ensures their safety in the event of a collision. Figure 10 shows the effects of a rearward collision with an LPG-powered vehicle. Additionally, according to the law, after each collision of a vehicle, they must undergo the legalization process again [2].

Fig. 10. The rear of a vehicle with a toroidal LPG tank after collision [12]

Connectors, nozzles and couplings are made of non-sparking materials (brass), so rubbing is not able to ignite any fuel vapors [11].

The gas ECU response time to switching off the gas installation is 0.3 seconds. The control electronics of the gas systems are designed in such a way that in the event of a sudden pressure drop in the system (e.g. after the fuel supply line to the reducer is broken), the system is immediately turned off and the engine is switched to the base fuel mode. Too low pressure in the system, determined on the basis of data from the gas pressure sensor, which is part of every modern gas installation, causes the gas supply to be disconnected. Therefore, the risk of leakage of the gas system in the engine compartment, and consequently the risk of gas leakage and ignition, is minimal. If, as a result of an accident, the gas flow from the tank to the engine compartment is disturbed (the fuel line is bent or broken), restarting

the engine in gas mode will be impossible. Additionally, a solenoid valve in the engine compartment cuts off the flow of gaseous fuel when the engine (for whatever reason) is not running. The same mechanism occurs also when the fuel in the tank is exhausted – when the gas pressure drops (it stops flowing), the controller perceives it as a signal to turn off the system. Turning off the gas supply system also closes the solenoid valve on the tank, so the only gas that can leak from the vehicle is the one in the lines [4, 7, 12].

In the context of CNG, it is also worth noting its physicochemical property in the form of low density under normal conditions (the density of natural gas is lower than that of air), thanks to which all leakages caused by the collision are released into the atmosphere, where they are quickly spread in the air without reaching the lower flammability limit. This further improves safety in the event of a collision.

Thanks to the security of the gas systems, damage to the gas supply system does not adversely affect the safety of the vehicle's operation.

5. Danger during a fire in a car with a gas fuel system

The fire test of an LPG fueled vehicle [13] carried out at the Polish Military Institute of Armament Technology in Zielonka (Poland) simulated a situation where, as a result of unsealing the gasoline tank and spilling fuel, it may ignite, which is tantamount to a car fire. The fire started at the rear of the vehicle due to the location of the gasoline tank. The gas tank was also installed in the rear part. During the fire test, there were about 25 liters of fuel in the gas tank, which accounted for 73% of its geometric capacity [11]. The fire was initiated with the use of gasoline in a container placed under the vehicle. The fire engulfed the entire rear of the car. After 5 minutes and 44 seconds, the gas tank fittings were activated. The cyclical operation of the safety valve lasted up to 9 minutes of the fire. The frequency of its operation (opening and closing) was initially increasing, and then the period began to lengthen to approx. 9 seconds, while the acoustic effect decreased. After the last closing of the safety valve in 9 minutes, the fire valve opened, as evidenced by the constant stoking of the flames and the characteristic, gradually decreasing in intensity, hiss of flowing gas, which ended 11 minutes and 30 seconds after the initiation of the test [3, 13].

The test showed that the LPG tank in a car poses no greater threat than a gasoline or diesel tank, and in some respects it is superior to them because it is a high-strength component. Gas tanks are more durable than conventional fuel tanks, which are most often made of plastic. The possibility of mechanical damage to the gas tank in the event of a fire is minimal. In the extremely unfavorable case of deformation of the tank shell, it is flexible enough (required by the production technology used) that it does not become unsealed. The high durability of the tank results from its operating conditions. Nominal pressure for LPG, depending on the degree of filling and the ambient temperature, ranges from 0.6 to 1.7 MPa. During the test, the pressure in the tank reached a value that allowed the safety valve to be opened (in LPG tanks – 2.7 MPa). The pressure increase takes place at a very high temperature, which at the same time weakens the material of the tank shell and its fittings. However, even such extremely unfavorable conditions did not cause the tank to become unsealed during the test. Releasing excess gas through the safety valve eliminated the increase in internal pressure in the tank. The fire gas escaping through the safety valves and the fire was removed through the vent under the car, which minimized stoking the flame [12, 13].

6. Legal restrictions on a gas-powered vehicle

Signs visible on public roads in Europe prohibiting the entry or parking of vehicles equipped with a gas installation (Fig. 11) are not included in the state's road traffic codes or in EU regulations. Therefore, the regulations do not regulate the liability for breaking it by the driver, there is no legal basis for issuing a fine for non-compliance with this and similar signs.

Fig. 11. A sign prohibiting entry for a vehicle powered by LPG

In Polish law, the document regulating the underground parking of a vehicle equipped with gas is the regulation of the Polish Minister of Infrastructure on the ventilation of parking lots. In those where cars powered by LPG and underground gas are parked, mechanical ventilation must be used, controlled by detectors of the unacceptable level of propane-butane gas concentration (Ministry of Infrastructure Regulation of 12 April 2002 on technical conditions that should be met by buildings and their location and its amendment of March 12, 2009). The LPG entry ban sign is not a grassroots initiative by the owners of underground garages. They are obliged to do so by § 4 ust. 2 pkt. 5 of the Regulation of the Polish Minister of Interior and Administration of 7.6.2010 on fire protection of buildings, other construction facilities and areas, which says that: *"owners, managers or users of buildings, storage yards and shelters, with the exception of single-family residential buildings: place, at the entrances to closed garages with the floor below the ground level, legible information on whether or not cars powered by LPG, propane-butane, referred to in technical and construction regulations, are allowed or not to park in these garages*" The above-mentioned regulations refer to the information obligation and ventilation. The ban on entry may therefore only result from the regulations of the car park established by its owner.

Due to the lower density of natural gas than air, entry bans do not apply to vehicles powered by this type of gas fuel. CNG does not pose a fire hazard as it lies low next to the floor and in sewers.

7. Conclusion

Both legal, legalization and technical aspects guarantee the trouble-free and safe use of gaseous fuel in motor vehi-

cles, as long as the user of the installation does not lead to gross negligence in operation and service. Gaseous fuels are safer than conventional fuels, and the ongoing development of the latest generation installations with direct gas fuel injection should, in the near future, support a stable growth in their use in the automotive industry, which will translate into a further increase in the safety of vehicle operation.

Nomenclature

CNG compressed natural gas
DIN Deutsches Institut für Normung (German Institute for Standardization)
EN European Norm Standards
ISO International Organization for Standardization
LPG liquefied petroleum gas
UNECE United Nations Economic Commission for Europe

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Analysis of the market structure of long-distance transport vehicles in the context of retrofitting diesel engines with modern dual-fuel systems

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This work contains considerations on the structure of the road tractor market in Poland in the context of their types of fuel supply systems and the possibility of adapting modern dual-fuel solutions in them. The current injection solutions in most vehicles available on the market indicate the possibility of highly effective use of modern dual-fuel or RCCI fuel supply solutions.

Key words: *truck market, fuel supply systems, dual-fuel, RCCI*

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1. Introduction

The history of dual fuel systems is long, but it has never been popularized. Adapting a diesel engine to a dual-fuel system with additional gaseous fuel is not an easy task. There were many reasons for this: from infrastructure problems to technical problems such as inadequate advancement of original diesel fuel supply systems or problems with valve opening timing overlap and difficulties with methane oxidation by the exhaust gas aftertreatment system, resulting in increased total hydrocarbons (THC) emissions [2]. These are the main reasons why retrofit diesel engines are not popular. The development of electronics led to the popularization of high-pressure injections systems such as Common Rail (CR) and pump injection unit (PD) systems. The technological advances of these systems bring the modern dual fuel solutions (controlled by diesel injection signals) and idea of Reactivity Controlled Compression Ignition (RCCI) power supply back to life. RCCI is an ignition method in which highly reactive fuel is injected directly into the cylinder [1]. It mixes with the air and self-ignites, thereby initiating the ignition of poorly reactive fuel, which is delivered via indirect injection to the combustion chamber earlier in the process as shown on Fig. 1.

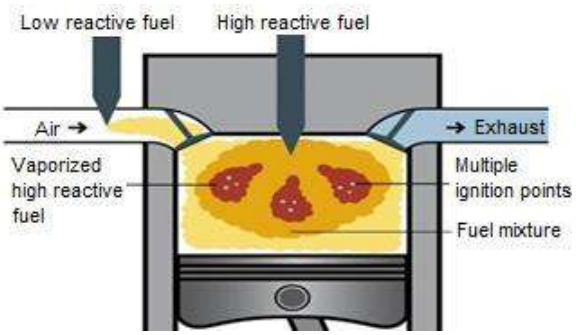

Fig. 1. Ignition and combustion in RCCI engine (based on [18])

The same method of fueling the engine is characteristic of a classic dual-fuel engine, where the difference is that the fuel-air mixture is not homogeneous. The possibility of implementing precise pilot doses and emulating the operation of CR and PD injectors opens up new prospects for high rates of replacement (50–70% or modern type dual-fuel system or 90–99% for RCCI) of the basic fuel with gaseous fuel in such a system [2, 19]. High replacement coefficient ratios directly affect the benefits of such a solution of fueling a diesel engine, through lower emission of harmful exhaust components [1, 2]. This is significantly different from the systems currently available on the market using simplified versions of dual-fuel systems, which most often use a simplified method of "cutting" diesel doses by emulating the acceleration pedal or manifold absolute pressure (MAP) sensors, as shown in Fig. 2 [20].

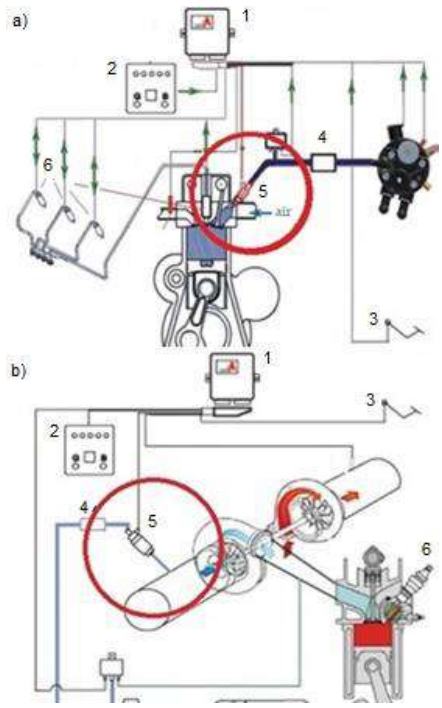

Fig. 2. Comparison of dual fuel systems: a) sequential gas injection pre-intake valve (controlled by diesel injection signals), b) pre-turbocharger gas injection: 1 – gas ECU, 2 – operation mode switch, 3 – acceleration pedal, 4 – gas filter, 5 – gas injector, 6 – diesel injector (based on [10, 20])

Manufacturers of such vehicles offer similar solutions in brand new vehicles, the market of used cars with factory installed systems is quite narrow. Therefore, this work focuses on the analysis of the polish structure of long-distance road transport in the context of the means of transport used by these companies (tractor units) and the possibility of using modern dual-fuel systems in them with the use of the original diesel injection system as a highly reactive fuel dispachers.

2. The structure and the use of road tractors in the Polish transport companies

The last report on the long-distance road transport market issued by the Polish Automotive Industry Association indicated that in Poland in 2019, 372.7 thousand road tractors were registered in Poland (data adjusted for vehicles excluded from use in Polish Central Register of Vehicles and Drivers database). The share of the oldest, over 20 years old, amounted to 5%. The youngest – up to four years old reached 30%. Similarly, tractors from five to ten years of age accounted for 30% of the machine park at the end of 2019. Thus, cars up to 10 years old constitute 60% of all vehicles. Those from eleven to twenty years remained the most numerous and accounted for 35% of the fleet. The average age of truck tractors was 8.9 years, with a median of 8 years [3]. At the end of 2019, in the updated part of the park, the number of road tractors registered in Poland was by 88 thousand. greater than the number of registered trucks (complete chassis, not including tractors and special vehicles) with a maximum permissible weight of more than 3.5 tons. For the first time, truck tractors outnumbered trucks in 2015 [3, 5]. This reflects the scale of investments of carriers specializing in international road transport. The year 2020 brought a decrease in sales, in all segments of commercial vehicles weighing more than 3.5 tons. The piling up of a handful of industry problems, led by the pandemic theme, has resulted in one of the worst results in the past few years [17]. Table 1 shows the registrations by manufacturers of new tractor units in 2020, 2019 and 2018 in Poland [6, 17].

Table 1. Registration of new road tractor units in Poland [6, 17]

Brand	2020	2019	2018
DAF	3 812	5 597	6 358
Scania	2 644	5 269	4 901
MAN	2 221	4 519	5 295
Volvo	2 644	4 235	4 439
Mercedes-Benz	2 226	3 961	3 916
Renault	581	1 405	1 734
Iveco	534	382	731
Ford Trucks	146	23	-
Total	14 808	25 391	27 375

Apart from new trucks, the market structure may also be illustrated by imported used tractor units. In 2020, 29 503 used commercial vehicles were imported to Poland, which, however, is a decrease by 6.2% compared to 2019 (–1948 units), of which 27 174 are road tractors – this segment also recorded a decrease by –3.8% (–1,070 units y/y) [6, 17]. To illustrate the structure of the demand for used vehicles by their manufacturers (it does not have to be a measure of the real demand for used trucks), data from the search engine were used. The data comes from the portal that sells used trucks in Poland. The data provided is presented in the Fig. 3.

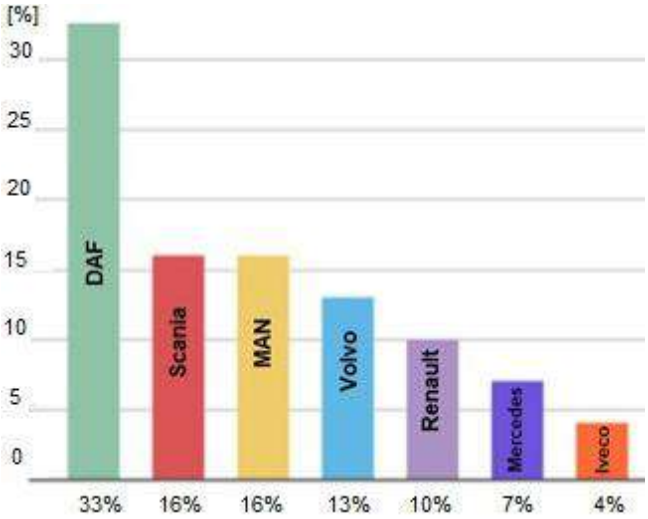

Fig. 3. The popularity of searches by brand for used tractor units [21]

Reliability is an important factor in using truck. Therefore, among the used vehicles, the most popular are road trucks not older than 7 years [4, 9]. The search data for used tractor units by their year of manufacture may also indicate when selling a vehicle makes the best economic sense. Figure 4 shows the percentage of searches for used tractor units by age.

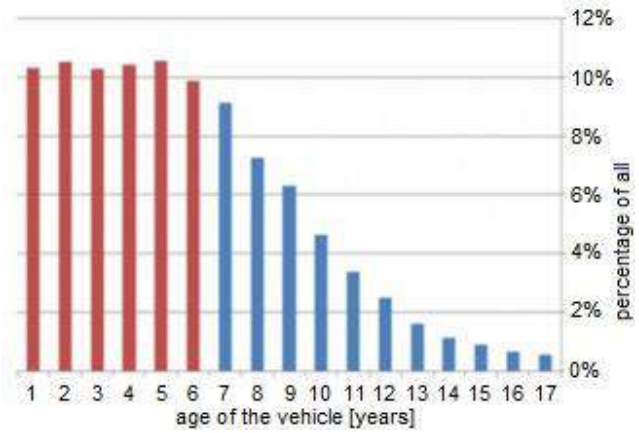

Fig. 4. Searching for used tractor units by their age [21]

Mentioned above data concerning the search preferences for used vehicles according to their year of production are indirectly reflected in the average time of use of a tractor unit in road transport companies. Of course, it is difficult to clearly define the economically justified service life of a tractor unit, as it depends on factors such as the intensity of use, annual mileage, type of work, etc. However, it is assumed that in the case of international long-distance transport it is about 5–7 years, national and local 10–15 years. Vehicles with specialized bodywork are in service the longest. including city ones, incl. garbage

trucks, sweepers, cranes, concrete pumps, etc. In Poland, there were 44% of specialist vehicles over 20 years old [3, 6].

Summing up, a typical Polish long-distance road transport company uses tractor units of the following brands: DAF, Scania, MAN, Volvo, Mercedes-Benz, Renault, Iveco (Ford Truck due to the beginning of entering the market will be omitted in further considerations) not older than 7 years for transport international or 15 years for domestic transport.

The analysis of the types of fuel supply systems in long-distance vehicle engines, presented in the following paragraph, focuses on the group of trucks mentioned above.

3. Engines used in road tractors and types of fuel supply systems used in them

3.1. DAF [7, 8, 15]

The Paccar MX engines were introduced in 2006 with XF105 road truck model. They have a displacement 12.9 liters and maximum powers of 408, 462 and 510 horsepower (hp). Thanks to the use of selective catalytic reduction (SCR), they meet the Euro IV or Euro V standards. They have 4 valves per cylinder, turbocharging and charge air cooling. They use single-section PD injection pumps with the pressure increased from 1500 to 2000 bar. As a result, there is no need for a diesel particulate filter in the exhaust system. Successor of the model XF105 in 2014 introduced upgraded Paccar MX engines in two displacement variants. They are 10.8 liters with 440 hp and 12.0 liters with 410, 460 and 510 hp respectively. Thanks to the use of SCR, they comply with the Euro VI standards. Like the previous engines, they have 4 valves per cylinder, turbo charging and charge air cooling. They use single-section Common Rail injection pumps with a pressure increased from 2,000 to 2,500 bar. As a result, there is no need for a diesel particulate filter in the exhaust system.

3.2. Scania [7, 8, 14]

Scania use engines equipped with XPI (short for "extra high-pressure") common-rail fuel system developed in cooperation with Cummins since 2003 in their models of road tractors from series R and S. These vehicles have a wide range of engines depending on the year of production and the needs of use. These are, respectively, units:

- In-line five-cylinders 9.0 liters 250/280/320/340/360 hp,
- In-line six-cylinders 13.0 liters 370/410/450/480/490 hp,
- V8 16.0 liters 500/520/580/730 hp,

3.3. MAN [7, 8, 13]

Models from series TGS and TGX since 2007 are powered by engines of the D20, D26, D36 series with the following power ratings and exhaust standards:

- Euro V with 360 hp, 400 hp, 440 hp, 480 hp, 540 hp and 680 hp.
- EURO V EEV (Enhanced Environmentally Friendly Vehicles) with 360 hp, 400 hp, 440 hp, 480 hp and 680 hp.
- Euro VI with 440 hp, 480 hp, 520 hp, 560 hp and 640 hp (the last 3 variants are the new inline 6-cylinder 15.2 liters engine D38).

In the fuel supply system of these units, injection pump systems are used for some units that meet Euro V and Common Rail Euro V and Euro VI standards.

The TGX series (the most popular long-distance MAN tractor units on the Polish market) uses six-cylinder in-line engines from the MAN D20 (10.5 liters) and D26 (12.4 liters), Euro VI series, in four power ranges: 360 and 400 hp (D20) and 440 and 480 hp (D26) obtained in the range of 1600–1800 rpm. In most cases, these trucks have a Common Rail power system.

3.4. VOLVO Trucks [7, 8, 12]

In September 2012, Volvo Trucks launched the production of a new generation FH model with many technical improvements. These tractors use 3 engine variants: D13C (unit injection, and Common Rail after july 2015), D13K (Common Rail), D13TC (Common Rail) in power variants from 360 to 540 hp. A variant is also available with the D16 engine with 550, 650 or 750 hp and a torque of up to 3550 Nm. All variants powered by CR.

3.5. Renault Trucks [7, 8, 12]

The T Series tractor units were presented in 2013 as the first of the new line of vehicles for the company and offers two six-cylinder Euro VI engines, 11.0 liters – DTI11 (380, 430 and 460 hp) and 13.0 liters DTI13 (440, 480 and 520 hp). All engines are variants taken from previous Premium and Magnum trucks. These are modified Volvo's D series engines. The fuel supply systems used in them are unit injection and Common Rail.

3.6. Mercedes-Benz Truck [7, 8, 16]

The Mercedes-Benz Actros MP4 made its debut on 30 September 2011. The fourth version of the Actros model offers several engine options in the Euro V or Euro VI variant. Euro VI uses a six-cylinder in-line engine, both OM471 series with 12.8 liters engines with output power: 422 hp to 530 hp and the OM473 from 15.6 liters engine with output power: 517 hp to 626 hp. Both engines use the X-Pulse injection system, which uses the Common Rail system equipped with a system that increases the injection pressure and the function of easy injection modification. The maximum fuel rail pressure is 1160 bar, which makes the maximum injection pressure 2700 bar.

3.7. IVECO Trucks [7, 8, 11]

The Stralis model is equipped with 6-cylinder engines with four valves per cylinder. They come in three variants of capacity, achieving different power parameters:

- Cursor 8, 7.8 liters: 310–360 hp
- Cursor 10, 10.3 liters: 420–450 hp
- Cursor 13, 12.9 liters: 500–560 hp

Changes in this engines took place in 2013 with the introduction of new engines meeting the EURO VI exhaust gas standard. All cursor engines after this date use the common rail system.

4. Discussion

The demands placed on commercial vehicles are constantly increasing. European legislation has forced commercial vehicle manufacturers to develop ever more powerful and dynamic engines with low fuel consumption. With

the appearance of exhaust gas purity standards, truck manufacturers realized that it was necessary to improve the fuel supply system so that the combustion process was more efficient. To achieve this, the fuel injected into the cylinders had to be finer in order to mix more easily with air. High-pressure unit injection systems have proved to be a good and reliable solution. They are also relatively cheap to produce and less prone to fuel contamination. Many years and millions of failure-free kilometers traveled on unit injectors effectively distracted some users and producers from the Common Rail system. Exhaust gas standards and increasing consumer expectations forced manufacturers to take another step in their development, i.e. the need for more precise fuel injection control. The development of electronics enabled the creation of PD and CR fuel supply systems, which were designed to reduce emissions and improve the operation of the diesel engine. These systems enable the feeding of the fuel dose in several portions – including the amount of the afterburner – which makes this type of fuel system an advantage in meeting the high requirements of exhaust gas purity for the EURO VI standard. Since all types of diesel fuel are characterized by the so-called delay of ignition – the use of a small pilot dose before top dead center (TDC) allows to apply the correct dose to the ignition area of the pilot dose – which makes the engine work softer and quiet – while maintaining high thermodynamic efficiency of the engine. The achievements of this technology indicate the potential of its use in a highly effective way, also in modern dual-fuel and RCCI power systems. Adequate control of pilot doses of highly reactive fuel (diesel fuel) to low reactive fuel (gaseous fuel) may be the key to using the original engine fuel supply system for retrofit modifications. Figure 5 shows the theoretical method of implementing such an injection in dual-fuel operation mode.

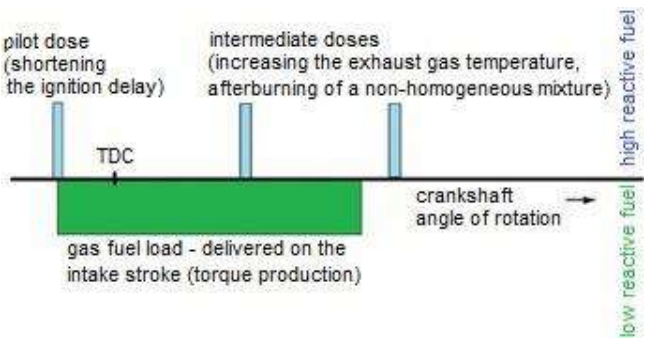

Fig. 5 Diagram of the co-combustion strategy with the use of multiple fuel injection (based on [1, 2, 18])

All mentioned above drastically reduce the cost of such a modification, which in turn gives a greater economic justification for the use of such solutions also in used road tractors, exactly like nowadays with the modification of passenger cars for LPG fueling, which is popular in Poland for many years.

The strict emission limits also leave no choice to the producers. The Common Rail system, although more expensive to produce and operate due to very precise cooperating elements, seems to be a solution that will eventually replace other fuel injection solutions in compression ignition engines. There are also electric and hydrogen solutions on the horizon, but at the moment the high energy consumption of road transport does not indicate a quick displacement of self-ignition engines. The future of road tractors' powertrains systems is uncertain. Strict emission limits leave no choice to manufacturers. The Common Rail system, although more expensive in production and operation (due to the very precise cooperating elements), has in principle already replaced and will eventually replace all other conventional fuel injection solutions in self-ignition engines. At least for a while. The overriding goal is to completely postpone the combustion of conventional fossil fuels. Temporarily, it is possible to use dual-fuel and RCCI solutions as less pollutant emitting solutions. At the moment, there is a lot of work on the use of modern dual-fuel or Reactivity Controlled Compression Ignition instead of regular diesel fuel injection only. The conversion to retrofit was successfully made, for example, by Arena Red in a vessel [19] and in more others. There are also electric and hydrogen solutions for truck tractors on the horizon, but at the moment the high energy consumption of road transport does not indicate a quick replacement of compression ignition units with alternative power sources [2].

5. Conclusion

- Dual fuel solutions for diesel engines are known, but due to their problems, they never gained popularity.
- New types of dual-fuel installations (originally used in factory vehicles) and Reactivity Controlled Compression Ignition systems can be used as retrofit installations in used vehicles such as Pump injection unit and Common Rail thanks to the use of modern diesel injection systems.
- The structure of the market for long-distance road transport vehicles indicates that the use of vehicles is not older than 7 years for transport international or 15 years for domestic transport. Which indicates the market's readiness (its absorptive capacity) to use modern dual-fuel solutions.
- All road tractors from the group of vehicles used in road transport have injection systems that enable the use of new dual-fuel and Reactivity Controlled Compression Ignition systems.
- Due to the pressure of the legislator and certification, the use of propulsion sources in vehicles is changing. Modern dual-fuel and Reactivity Controlled Compression Ignition releases may, in the factory edition as well as retrofit, prove to be an intermediate link before the full displacement of fossil fuels and their replacement with hydrogen electric powertrains or their combination.

Acknowledgements

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Nomenclature

CR common rail
ECU Electronic Control Unit
EURO IV, V, VI European emission standards
EN European Norm Standards
MAP manifold absolute pressure
PD pump injection unit
RCCI Reactivity Controlled Compression Ignition
TDC top dead center
THC total hydrocarbons

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THE PHENOMENON OF KNOCKING COMBUSTION AND THE IMPACT ON THE FUEL EXCHANGE AND THE OUTPUT PARAMETERS OF THE DIESEL ENGINE OPERATING IN THE DUAL-FUEL MODE (DIESEL-CNG)[1]

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Summary

The performance of dual fuel engines may be limited by the phenomenon of knocking combustion. Avoiding the phenomenon of knock in a diesel engine using its positive attributes, such as high compression ratio, may be difficult to achieve in engines converted to dual-fuel operation. This study presents research of handling with this problem and the result on the obtained percentage of fuel gas exchange as well as the impact on the engine performance in the inlet of a turbocharged six-cylinder inline unit.

Keywords: dual-fuel, fuel exchange, fuel replacement, CNG, knock

1. INTRODUCTION

The phenomenon of uncontrolled combustion, common in spark ignition engines, knocking, is linked with a high rate of energy release, a rapid build-up of pressure and, excessive heat transfer. In order to obtain correct and effective engine operation, this phenomenon must be avoided. In order to avoid knocking, there are many different solutions related to the control of the engine, also included in the process of its design. In the case of the dual-fuel system, the degree of replacement of diesel with gas fuel may increase the risk of knocking. The presented studies show responses to tune in the context of knocking in an experimental sequential gas injection system implemented just pre-inlet valve, which are characterized by a much greater replacement factor than the old injection systems available on the market before the turbine or injection into the intake manifold, but without emulating diesel injectors (Fig. 1).

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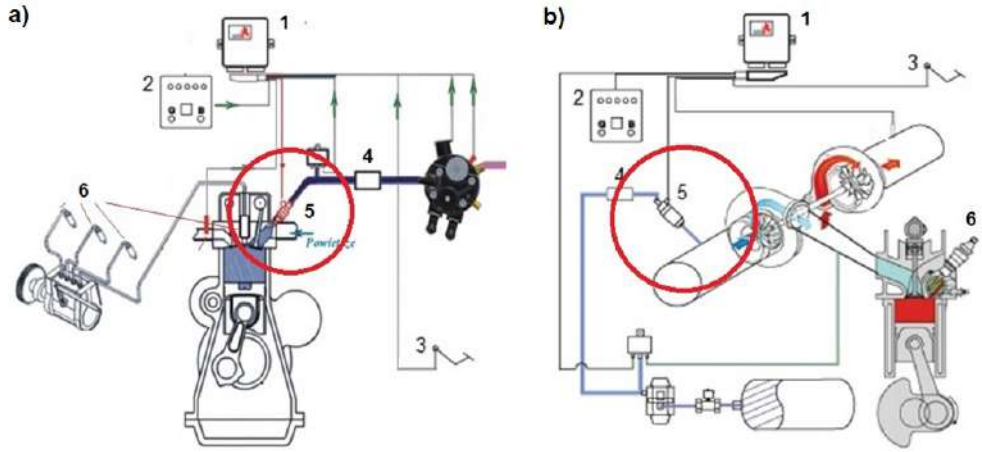

Figure 1 Comparison of dual fuel systems: a) sequential gas injection pre- intake valve, b) pre-turbocharger gas injection: 1 – ECU, 2 – switch, 3 – acceleration pedal, 4 – gas filter, 5 – gas injector, 6 – diesel injector

Source: [16, 17]

1.1. The knock phenomenon

The knock phenomenon is caused by uncontrolled rapid combustion energy release. This is due to the uneasy flame propagation due to the self-ignition of part of the unburned air-fuel mixture as its shown in figure 2 [1]. This phenomenon is the result of a complex interaction between turbulent flame propagation processes and pre-ignition-oxidation reactions of part of the gas at the end of the flame in the mixture that is still consumed by the flame [3].

Knocks affect the life of the engine by impeding lubrication and increased susceptibility to overheating and mechanical damage. In order to avoid knocking, motors are usually designed conservatively, without the motor reaching the best possible performance [10].

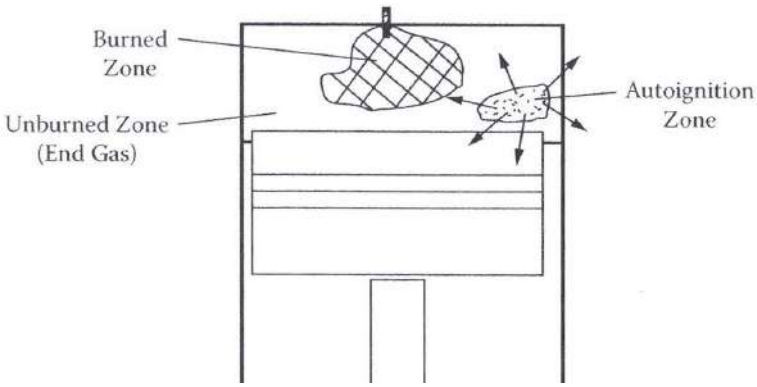

Figure 2 Knocking phenomenon in a SI engine with indirect gas injection

Source: [4]

The knocking process can be detected by:

- monitoring specific engine vibrations,
- monitoring of the specific noise emitted by the engine,
- unusual increases in cylinder pressure (pressure pulsation effect),
- drop in performance,
- negative change in exhaust gas composition and temperature.

The knock intensity is directly related to the rate of pressure increase in the cylinder. Then this is a function of the net energy released by self-ignition, which mainly controls the intensity of the cylinder pressure change over time. Proven solutions of knock sensors using strain gauges or pressure sensors placed in the cylinder are available on the market, but they have their limitations. [1, 3, 15].

It can be seen from figure below (Fig. 3) that knocking occurs with CH_4 for high compression ratios [2]. The figure shows a typical representation of the area of the knocking mixture relative to the normal limits of the working mixture with variations in the compression ratio for a SI engine running on CH_4.

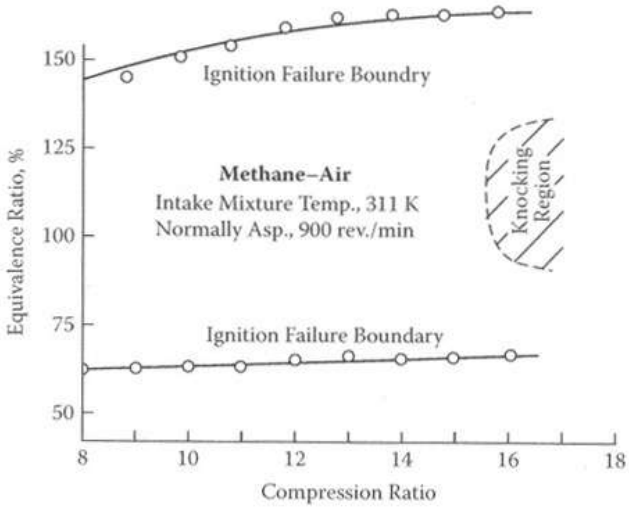

Figure 3 The quality of the methane-air mixture on its ignition ability (SI engine)

Source: [2]

Self-ignition of a homogeneous mixture of methane (without an external ignition source) is also possible if a sufficiently high inlet temperature or a high compression ratio is applied (Fig. 4).

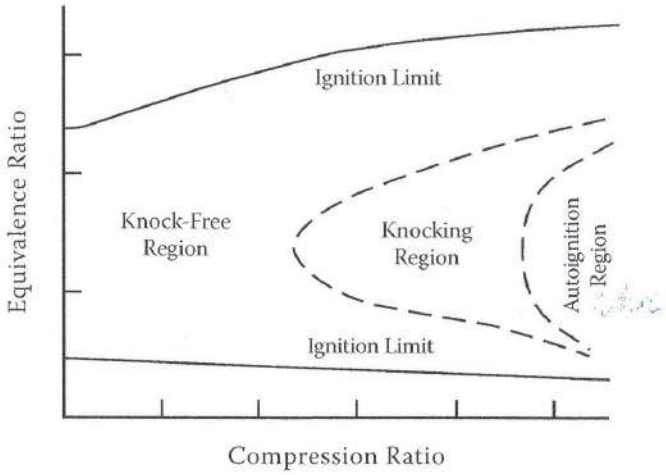

Figure 4 Representation of knock and compression-ignition an ignition limits with compression ratio for methane-powered engine

Source: [3]

1.2. Diesel engine knock

The characteristic knocking noise can also be found in a compression-ignition engine, depending on the intensity of the pressure build-up after self-ignition. This effect appears noticeably after long ignition delays or at low inlet temperatures as shown in Figure 5. Operating in this condition results in increased shocks being transferred to further engine operating parts. This effect can be called diesel knock [4, 14].

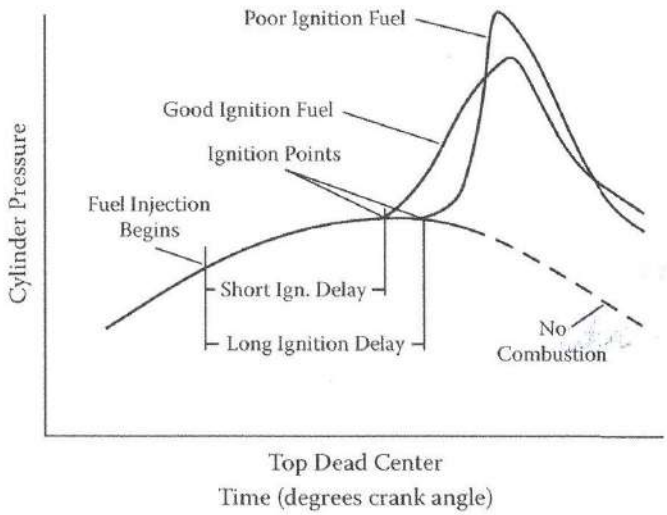

Figure 5 Pressure build-up diagram for two fuels with different ignition delay in CI engines

Source: [4]

Unlike spark-ignition or dual-fuel engines, where knocking combustion is associated with the final stages of combustion, in a diesel engine it is associated with the initial combustion phase. The factors determining the knocking in a CI engine are [1].:

- engine design,
- injection type (direct or indirect),
- ignition method,
- ignition delay.
- compression ratio,
- engine size,
- engine speed,
- internal combustion chamber temperature,
- engine wear level

Reducing the knock intensity in a CI engine consists in changing (reducing) ignition delay. It is also effective to reduce the amount of fuel stored in the combustion chamber during ignition delay, but modern common rail systems solve this problem well [3, 5].

1.3. Dual-fuel engine knock

The release of combustion energy in a dual-fuel engine is related to the sum of the energy of the pilot fuel (diesel) and gas fuel (CNG in our tested engine). A pilot dose after ignition spreads the flame and initiates turbulent combustion of most of the remaining gas charge. This combustion mode is responsible for the ability of dual fuel engines to benefit from the combustion of air-fuel mixtures that are much leaner than those normally possible in SI gas engines. This contributes to the high efficiency of dual-fuel engines. [1, 3, 12].

In a dual-fuel engine, immediately after the injection of the combustion initiation fuel and a short delay, ignition takes place and combustion begins very quickly and practically simultaneously at many points of the gas charge. The turbulent flames then begin to spread from these ignition centers to local areas of the gas/air mixture. Such an ignition mode, despite the high compression ratios used in dual-fuel engines, reduces the likelihood of self-ignition of the mixture compared to SI [13]. Under certain circumstances, when large pilot doses are used and, at the same time, during their ignition, the areas of the gas-air mixture ignite together, an effect similar to diesel knocking can be obtained. [3].

The gas-air mixture should have adequate resistance to spontaneous combustion and should be introduced into the cylinder at such a time that it does not have enough time for self-ignition before, during or after the rapid release of energy from the pilot fuel into its immediate vicinity. Failure to do so may lead to intense knocking, which results in too rapid pressure increase, overheating of the engine (which negatively affects its performance) and ultimately to mechanical damage, as shown in fig. 6. [13].

Figure 6 Damaged engine liner and piston due to dual fuel knocking

Source: [13]

Figure 7 shows a schematic record of the cylinder pressure in a dual-fuel engine running on CH_4 with and without knocking. Figure 8 shows the size of the pilot dose during the operation of a dual-fuel methane diesel engine in the context of a rapid increase in pressure leading to a knock

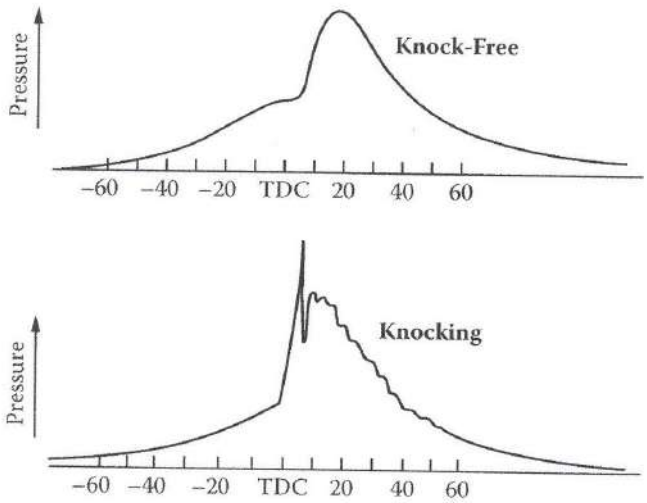

Figure 7 Cylinder pressure in a dual-fuel engine running on CH_4 with and without knocking combustion

Source: [3, 5]

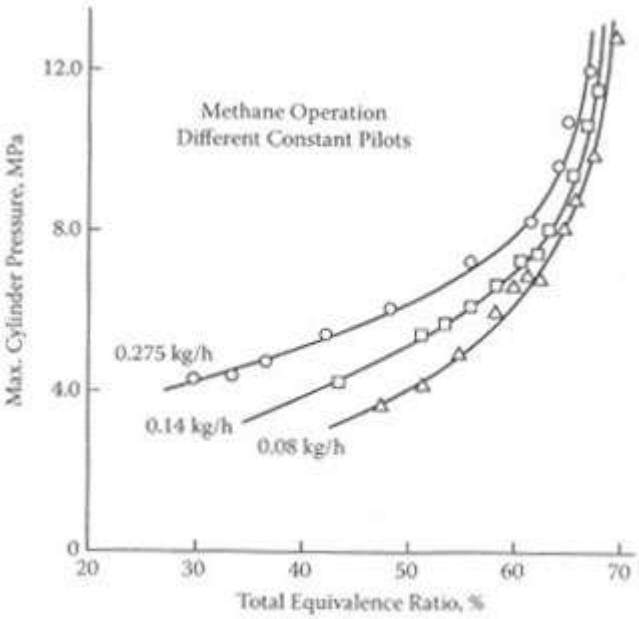

Figure 8 Three different amounts of pilot doses in the context of the variability of the maximum pressure value and total equivalence ratio (diesel-CNG engine)

Source: [3, 7]

The knocks found in engines operating on dual-fuel mode can be caused by both types: diesel or gas knockings. A large amount of lean premixed fuel-air which can be considered as an end gas relation to the regions of pilot self-ignition, may be subject to rapid autoignition reactions assisted with high temperature afeter pilot ignition, leading to end fuel type knocking [3]. Too small pilot doses may lead to turbulent combustion of the gas-air mixture, which may, under specific conditions (e.g. high temperature), lead to knocking combustion similar to that of SI.

When it comes to knocking, the temperature of the mix has a paramount importance. The effects of changes in the fuel-air ratio are relatively less severe and primarily regulate the resulting combustion temperature as shown in Figure 9, it shows changes in the graph of the heat release rate from a dual-fuel engine with increasing fuel gas concentration at the inlet, causing knocking at a constant pilot size.

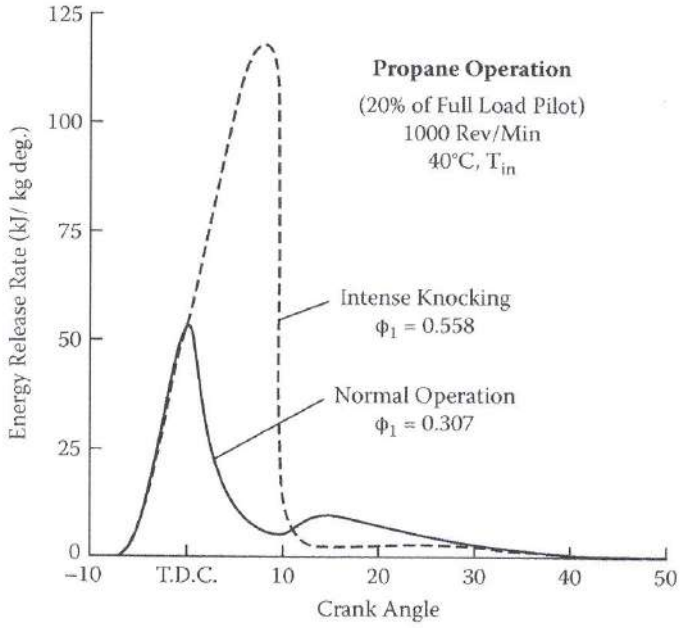

Figure 9 The course of energy release in a function of crankshaft rotation for two different gas-air mixtures at a constant pilot dose.

Source: [3, 6]

With the reciprocal of the absolute intake manifold temperature for a dual fuel type engine, the output power for different fuels and pilot settings appears to decrease logarithmically (limited by knocking) as shown in Figure 10 [3].

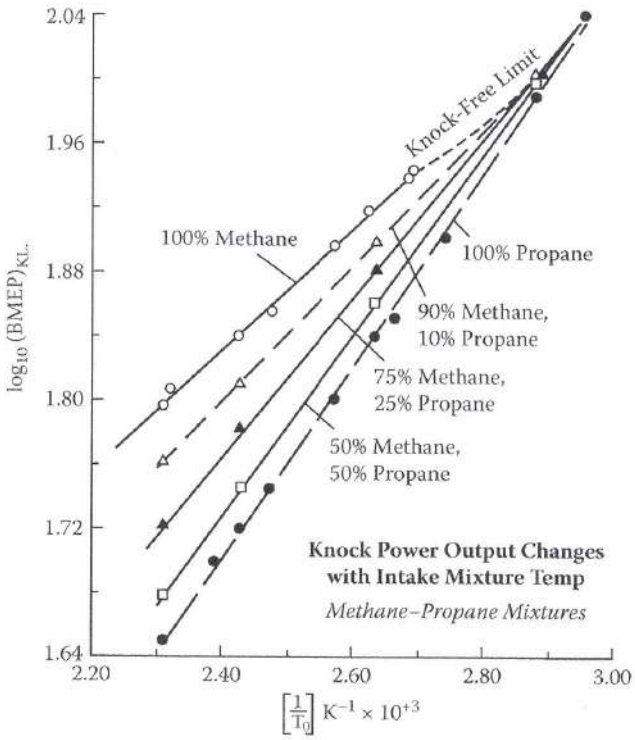

Figure 10 Log BMEP with the reciprocal of the absolute temperature of the inlet mixture for various methane-propane mixtures (constant pilot, constant RPM, reduction of knocking combustion)

Source: [3, 5]

Methane is more tolerant to temperature increase in inlet mixture than other gas fuels. The action on LPG also shows that the log-limited output of power is linearly related to the reciprocal of the absolute temperature of the inlet mixture. Increasing a pilot size causes the knocking to start faster with a corresponding reduction in knockless power [1, 3, 11, 14]

2. METHODOLOGY

The aim of the research was to tune the Volvo D13C engine configuration to work on dual-fuel diesel-CNG mode. The goal was to maintain similar (or better) torque values in the dual-fuel cycle compared to the nominal values on diesel fuel operating mode with the focus of possible the greatest replacement of the base fuel (diesel) to gaseous fuel (CNG) till the points reaching the knock limit. In the tested case, the engine was converted to dual-fuel operation by adding a prototype sequential indirect gas injection system. The tests were carried out on an engine dynamometer. Figure 11 shows the operation scheme of the tested engine and the principles of the research setup.

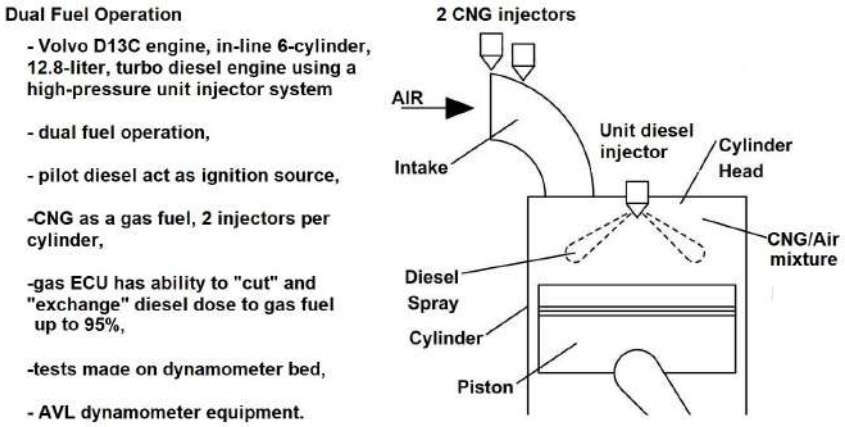

Figure 11 Schematic of the tested engine operation and principles of the research setup.

Source: Authors' archive

3. RESULTS

3.1. Fuel replacement

Figure 12 shows a screenshot from the gas controller software showing the percentage of replacement of the diesel fuel to CNG with the area of knocking combustion marked. Below the red line, the replacement percentage is set as close to knock as possible.

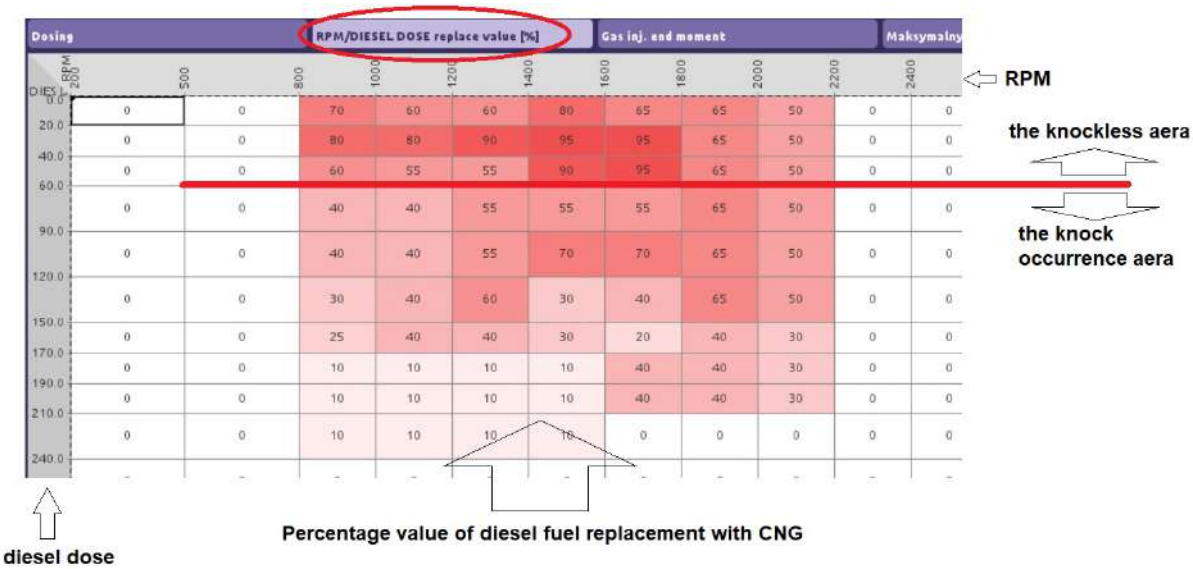

Figure 12 Screenshot from the gas controller software

Source: Authors' archive

3.2. Performance

Figures below (no 13 and 14) shows the values of the torque in a function of hourly fuel consumption for single- and dual-fuel operation for 2 constant rotational speeds.

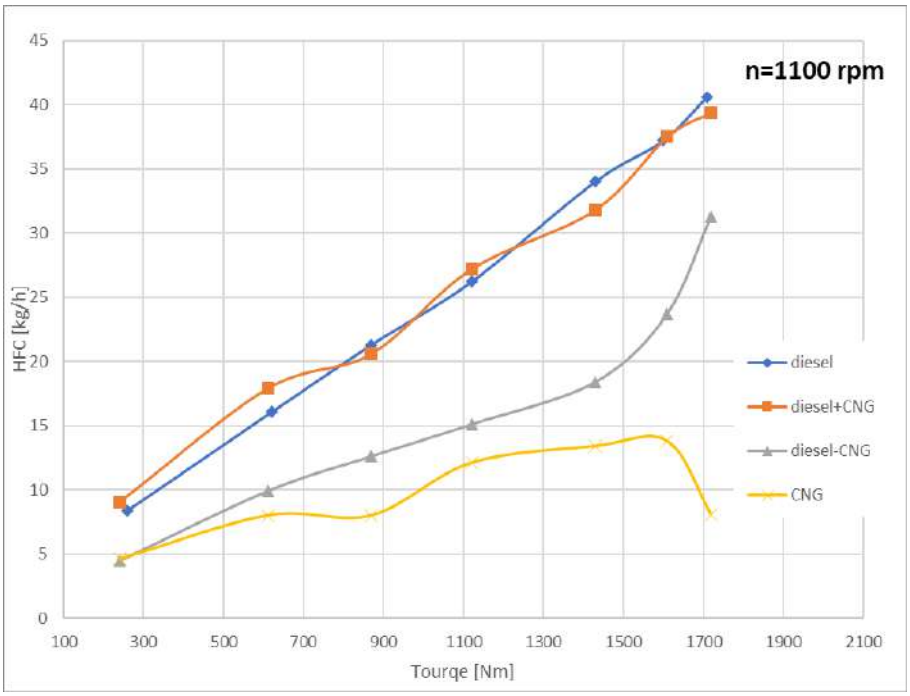

Figure 13 HFC and torque chart for operation on the base fuel and in the dual-fuel mode at 1100 RPM

Source: Authors' archive

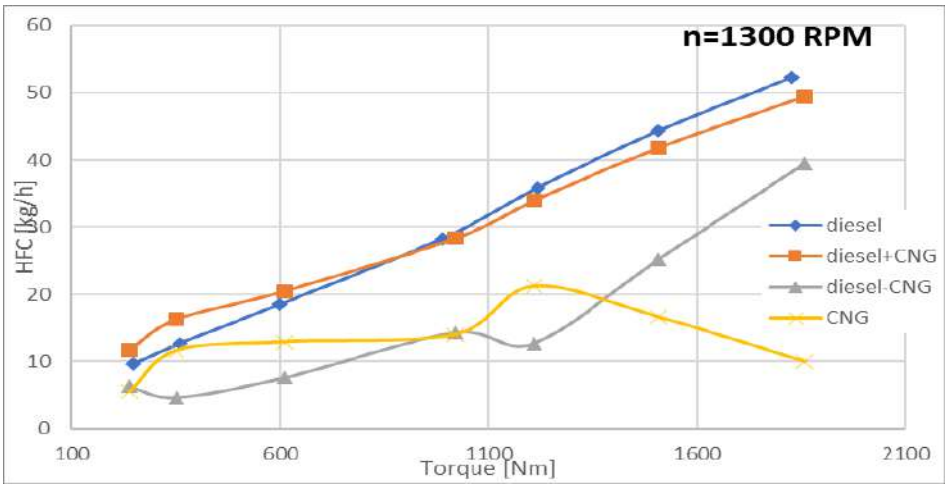

Figure 14 HFC and torque chart for operation on the base fuel and in the dual-fuel mode at 1300 RPM

Source: Authors' archive

3.3. Discussion

Research has shown that the distinct point at which knocking occurs is related to the fuel exchange rate and engine load. With increasing boost pressure, the percentage of fuel replacement decreases due to the appearance of knocking combustion. Figure 12 it can be seen in the representation of the calculated diesel dose, which corresponds to the possibility of generating upper torque values by the tested engine. Figures 13 and 14 also indicate a reduction in the possibility of fuel exchange near the maximum possible to generate torque. There is a clear discrepancy in the applicability of high fuel replacement rates. This situation is most probably caused by the fact that by indirectly introducing gas fuel into the intake system of the engine, it fills the space that was originally distorted in the diesel engine for air load. Thereby, as it were, the engine becomes overloaded with the gas-air mixture, which ultimately causes the resulting end gas knocking combustion.

3.4. Knock reduction methods

The incidence of knock can be avoided by a number of possible design and operational measures. For example, by: reducing the temperature of the mixture and the water jacket, delaying pilot injection, decreasing the compression ratio - they could undermine pilot ignition and the CI operation. The most appropriate way to avoid knocks (legitimate economic law and without excluding technological interference) is controls of the composition of the gas fuel mixtures, characteristics of the pilot injection or using EGR. As a general rule, an effective procedure to reduce the frequency of knocking, if this possible, is optimal distribution of gas fuel-air ratio with good stratification of gas components in the cylinder, with the reactive, leaner gas-air mixtures being in the regions of the pilot's ignition means where there are richer mixtures. This would require control of the gas distribution so as not to obstruct the ignition of the pilot fuel or to encourage the spontaneous combustion of areas adjacent to the pilot, while reducing the tendency of the charge away from these centres to self-ignite despite the high rate of increase in cylinder pressure and temperature [3, 5, 11].

Engine knocking combustion cannot be permanently eliminated, but there are many solutions to minimize it. In the case of converted engines, a second injection system is often extra added and minor changes are made to the equipment of the engine and the control system, which was also the case in the engine discussed in this paper. Taking this into account, only a few solutions are economically viable in dual-fuel converted engines. In addition to parameters that are difficult to improve, such as the material of the

engine-head or the combustion chamber shape, for these changes that are economically acceptable, most cases the following can be used:

- fuel replacement ratio,
- gas injection advance,
- boost pressure,

In this discussed case, the most economically viable solution chosen by the authors of this study was to reduce the boost pressure through the use of a bypass engine intake circuit (figure 15). Research on its influence on the possibility of increasing fuel replacement ratio is underway.

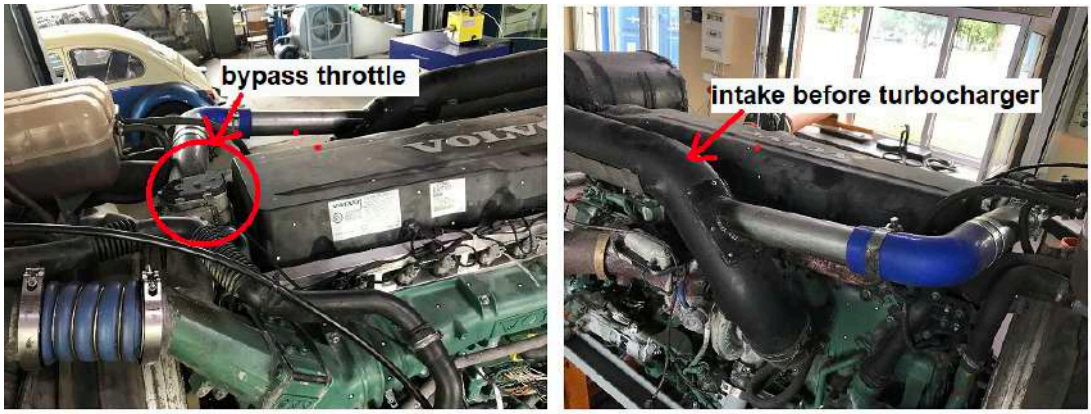

Figure 15 Construction of the bypass lowering engine's boost pressure

Source: Authors' archive

4. CONCLUSION

It is difficult to compare the results of the exchange coefficient and its impact on knocking combustion in the presented research with external results of other researchers, due to the experimental type of installation and the injection system, which is not commonly used on the market. This short research paper shows that sequential pre-intake valve CNG injection has great potential for use in a dual-fuel engine system where the pilot diesel acts as an ignition source in a compression-ignition engine. It is possible to recreate almost similar engine performance in such a system. It also has a positive effect on exhaust gas emissions, but to do so the fuel replacement in the operation should be as high as possible. The literature deals with knocking problems in a diesel-CNG dual-fuel engine. Empirical studies also confirmed the problem of dual-fuel knocking, in particular when trying to obtain high values of torque and with high load. More work needs to be done to reduce or eliminate knocking in an effective manner that maintains good engine performance and minimizes exhaust emissions. The authors' next step is to investigate the effect on the replacement ratio by lowering the boost pressure.

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SOLUTIONS OF CNG AND LNG SUPPLY SYSTEMS IN MODERN LAND AND MARINE CI ENGINES WORKING IN DUAL-FUEL (NG – DIESEL) MODE[1]

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UDK 629.5:621.43
665.6

Summary

In modern world, due to exhaust emissions restrictions in all economy sectors, alternative fuels might be the fundamental element of further development of internal combustion engines. Natural gas as a fuel for internal combustion engines allows to meet expectation of incoming requirements of exhaust gas emission standards. These standards would highly limit usage of internal combustion engines working on conventional fuels. The method of supplying gaseous fuel to the engine combustion chamber determines the effectiveness of the applied solution and the efficiency of fuels energy conversion. A compression ignition internal combustion engine is a type of reciprocating piston engine that results in the highest efficiency in converting the chemical energy contained in the fuel into mechanical work. The use of Compressed Natural Gas (CNG) and Liquefied Natural Gas (LNG) as the only engine fuel does not allow the Compression Ignition (CI) of the air-fuel mixture. It becomes necessary to use other fuel, which initiates the autoignition of the air-fuel mixture. The aim of this study is to indicate modern solutions for supplying dual-fuel engines with natural gas (in the form of LNG or CNG), which enable compliance with the latest exhaust emission standards for piston engines. Fuels like LNG and CNG are useful in many modern engines applications such as: ships, cars, trucks, trains, factories, generators, power plants and working machines. This study will outline the solutions used in all of these applications.

Keywords: dual-fuel, RCCI, CNG, LNG, engine supply systems.

1. INTRODUCTION

Solutions shown in this paper are used in combustion engines fuel supply systems working in dual-fuel mode. Most of these types of systems are based on already existing solutions, but their common application requires changes and reflections on their assembly and purpose. All the solutions presented in the article are

[1] The publication was created as a result of work under: Poland Republic, National Centre for Research and Development, Military University Technology; Name of the competition: Ścieżka dla Mazowsza; Agreement No. MAZOWSZE / 0123 / 19-00; Subject: "Innovative ecological CNG installation for diesel engines limiting the emission of harmful exhaust components together with a mobile diagnostic platform."

used in Internal Combustion (IC) engines of various sizes, speed and applications. These solutions vary depending on the application of the engine and its size. All cars solutions are much more simply than those which are used in marine engines, applied on land and at sea, where power generated by the engine is bigger than in a land vehicle, and access to natural gas is better. Such engines are used in ships but also in big manufactories and power plants, generating electricity and/or useful heat. The scope of their application depends only on the human level of energy demand in the chosen place or region and the availability of two compliant fuels, which can be used in that kind of engine.

1.1. The basics of dual-fuel engine working mode

Engines working on two fuels simultaneously can be named in a few acronyms, but the most promising for future use is the Reactivity Controlled Compression Ignition (RCCI) engine. RCCI is an engine type where ignition method based on small amount of high reactive fuel use in combustion chamber full-filled with low-reactive fuel. The high reactivity fuel can be a diesel fuel, and low reactivity fuel (high octane fuel) is usually the gaseous fuel [1]. In research studies there was a lot of fuels used in this type of engine - basic requirement is a high octane number. Basically in the process of developing this type of ignition fuels such as gasoline, Natural Gas (NG), Liquefied Petroleum Gas (LPG) or alcohol and high octane biofuels. The injection of diesel fuel fundamentally based on proven technologies used in classical Compression Ignition (CI) engines. Apart from diesel fuel, in RCCI engines could be used every fuel whose properties are similar to those of diesel fuel and ensure its immediate self-ignition. Commonly it can be a diesel type biofuel like Fatty Acid Methyl Esters (FAME) or Fatty Acid Ethyl Esters (FAEE), Rapeseed Methyl Ester (RME), soyabean methyl ester (SME), Jatropha Oil or even hydrotreated vegetable oil (HVO) which is a highly converted biofuel. The choice of fuels used depends primarily on the engine concept, so the problem of a given fuel supply system is a very important point in RCCI engines solutions. RCCI engines are able to use any fuel known for engine applications, whose reactivity level (octane number) is enough high or enough low to accept their usage. This paper will present all the dual-fuel systems used in engines of various sizes and uses of which the highest known form are RCCI engines which in a very promising way increase thermal engine efficiency and reduce the emission of those harmful exhaust components whose are difficult to eliminate in classical CI diesel engines, like NO_X, CO_2, PM and SO_X - in the maritime applications. Using these types of engines together with well-chosen fuel supply systems can take piston engine greenhouse gas emissions to a whole new level.

2. FUEL SUPPLY SYSTEMS FOR DUAL-FUEL ENGINES

Fuel supply systems in dual-fuel engines are based on proven solutions from other piston engine types. The most common used in dual-fuel fuel supply systems are based on diesel engine injection system and gas fuel supply system commonly used in Spark Ignition (SI) engines. Fuel supply systems for dual-fuel engines can be divided into several chapters, starting with the oldest and simplest basic systems, and ending with those that seem to be the most advanced and efficient today.

2.1. Basic fuel supplying in dual-fuel engines

For small, medium speed engines used in cars and small power generators dual-fuel supply systems are quite popular and well known. The most common used supply system is indirect injection of high octane fuel and direct injection of high reactivity fuel. In unscientific use, the high octane fuel is always a gaseous fuel, like CNG, evaporated LPG or evaporated LNG. The most popular point for gaseous fuel injection is the intake manifold, or intake system pipe in the older technologies - where the gas mixer is placed. A sample look of gas mixers or its throats - used in marine engines on ships or energy plants, and small engines applied in cars or boats are shown in figure 1.

Figure 1 The examples of throat constructions from gas mixers for different engines.

Source: [2]

Gas mixers from figure 1 shows the throats and all gas mixers. Gaseous fuel is supplied via bores in the housing and via bores in the tube in the middle, which is known as a bar [2]. This way of supplying fuel to the engine through its intake system is essentially the oldest type and bars shown in the figure 1 represent the most primal way of fuel supplying. In present IC engine technology only gaseous fuels are supplying in this way. Those kind of technology is commonly use in gas engines in all sizes. Their widespread use is due to their simplicity and low maintenance costs. The principle of operation of such a device and its cross-section are shown in the next figure, figure number 2.

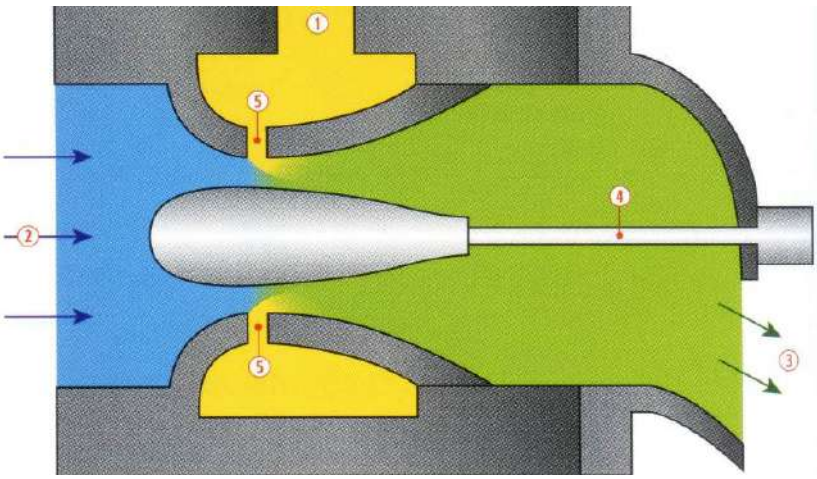

Figure 2 A cross-sectional view of a gas mixer with an adjustable venturi insert,

Source: [3]

Figure legend:
1 – the gaseous fuel supply,
2 – air supply,
3 – the air-fuel mixture to the intake manifold of the gas- or dual-fuel engine,
4 – the adjustable venturi insert shaft connected to stepper motor,
5 – bores for the supply of fuel-gas in the throat of the gas mixer.

Operation of the simple gas mixer is similar to the basic carburetor, where a venturi is used to suck fuel by the air flowing through it. This type of dual-fuel injection is the first and the simplest type of fuel

supply. A development of this fuel delivery method is the introduction of fuel injection in the intake manifold. The point of fuel delivery should be as close to the intake valves as it is possible, because of the time required for the fuel to flow from the injection point to the combustion chamber and the inertia of the entire system. In this system the pressure of the fuel is not so important as in another, because of Venturi effect. In large-scale engines this way of fuel supplying is not the simplest way to deliver the gaseous fuel into the intake manifold.

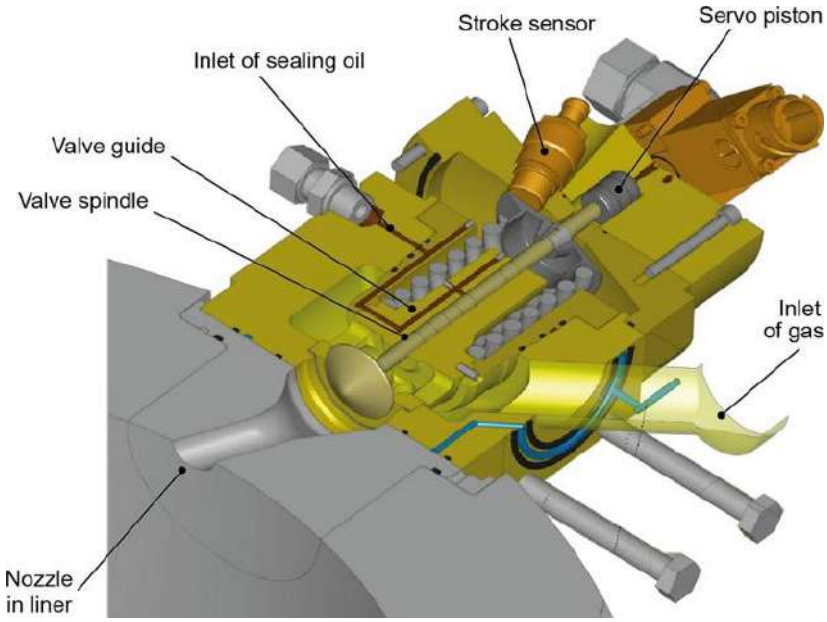

Figure 3 CAD drawing of the valve-controlled gaseous fuel direct supply.

Source: [4]

The simplest way of gaseous fuel delivery is through the valve. Valve can be placed in the intake manifold or in the structure attached to the engine cylinder head. Pressure of the gaseous fuel is sufficient for creating the enough amount of fuel flow through the valve, when it is opened. The most common kind of this solution supplies fuel into the intake manifold, but in the figure 3 more advanced application of that is showed.

2.2. Injection of both fuels in dual-fuel engines

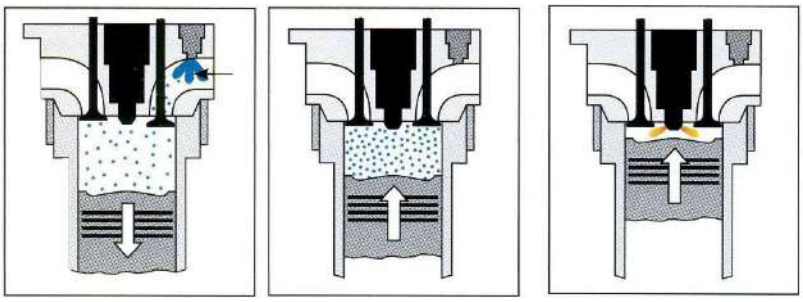

Figure 4 Schematic drawing of simply dual-fuel injection engine work.

Source: [3]

Schematic show of classic described in this paper dual-fuel injection system of both used fuels is shown on figure 4. The way of low reactivity fuel supply can be improved by using a fuel injector in a place of gas mixer. When there is a necessity of using liquid low reactivity fuel, the fuel injection may be

indispensable. The indirect injection of fuel can be placed anywhere in the intake manifold, but the best quality of fuel supply can be achieved by placing the injector as close to the inlet valves as possible. The long way to go for fuel with air gives the opportunity to create a well-mixed air-fuel mixture, but to long way creates a mismatch in the injection time to the fuel demand of the engine. There could be also a problem of the flow of already injected fuel by the turbo charger, what is undesirable and potentially dangerous. So the best solution is to locate the injector directly in front of the intake valve. This type of dual-fuel supply system is a classic way of fuel delivery too, but more technically advanced due to the use of more expensive than carburettors fuel injectors.

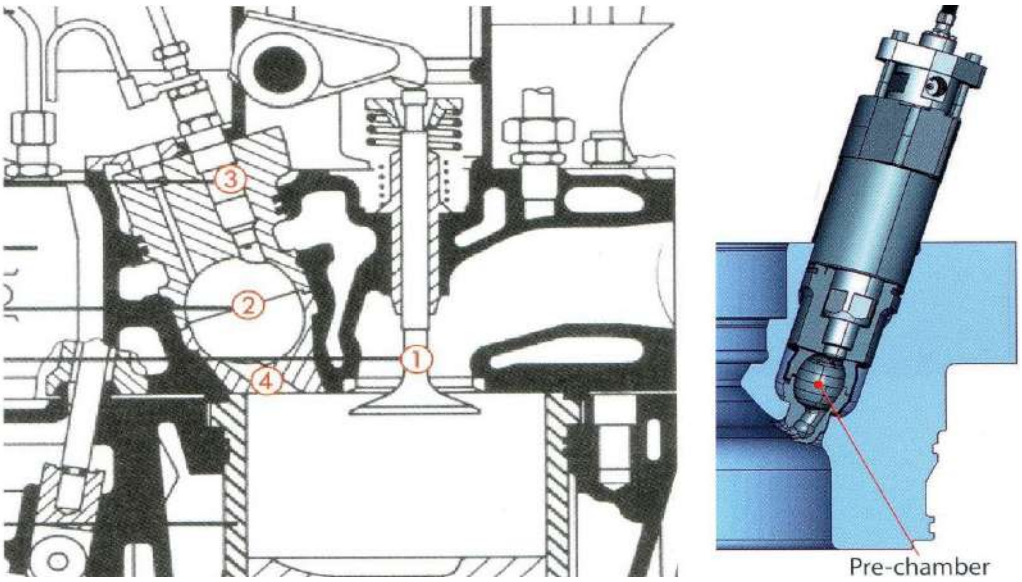

Figure 5 A Ricardo pre-chamber in a conventional four-stroke medium-speed diesel engine on the left and the full pre-chamber draw with the indication of main pre-chamber on the right.

Source: [2], [3]

Figure legend:
1 – exhaust valve,
2 – pre-chamber,
3 – pintle- type injector,
4 – tangential channel.

In bigger gas engines, used in maritime, where bores are larger than 200mm, a pre-chamber is used, so solutions describe above are not enough to make a correctly fuel supply [3]. The view of the modern pre-chamber drawing is shown in the figure 5 on the right, in comparison to primary kind of pre-chamber shown in the same figure on the left.

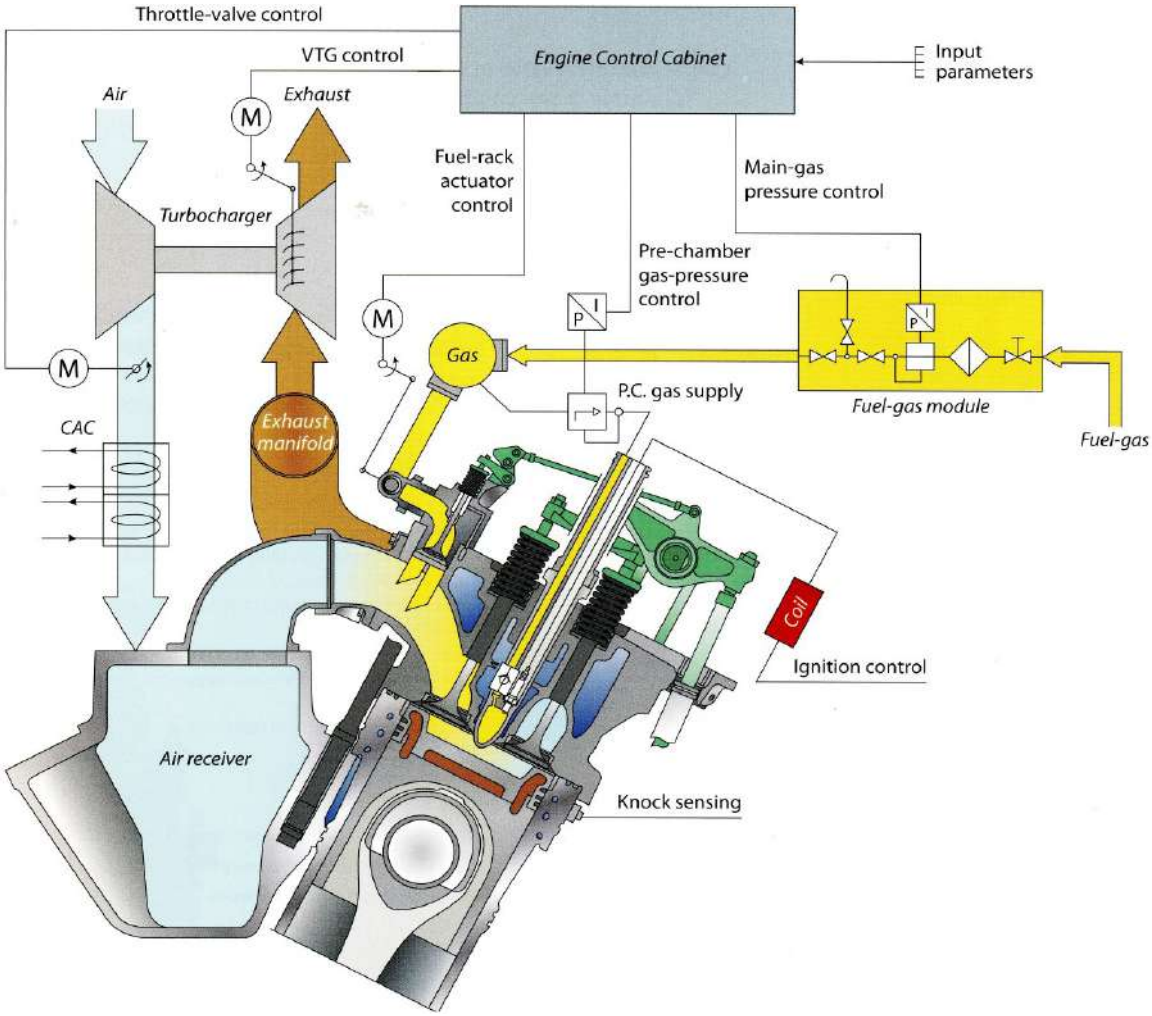

Figure 6 An overview of the gas supply for Rolls-Royce turbocharger gas engine - the gas supply principle for the Rolls-Royce gas engine showed on the cross-section drawing.

Source: [3]

Those kind of pre-chamber is still used in a conventional CI engine. The very similar kind of pre-chamber was used in CI engines from the beginning of the existence of CI engines. Pre-chamber can be easily fuelled by diesel fuel, but the high octane fuel supply into pre-chamber is not so easy to performance. To supply in this pre-chamber two types of fuels in marine engines there are used two points of fuel supply (usually both of them are being released by injection). In the figure 6 it is possible to see all schematic mode of action for this hybrid type of fuel supply in the Royce-Rolls engine.

Figure 6 shows the direct injection of gaseous fuel into the pre-chamber and the gas supply valve which is responsible for fuel release into the intake manifold, as it was described in the previous subchapter. The way of gaseous fuel supply injector or valve steering to replace with operation could be electric or fully mechanic. In maritime engines, mechanically operated valves are common, because the factor causing the gas to enter the intake manifold is its pressure. Every gaseous fuel is stored under pressure. Rolls-Royce engine from figure 6 is called Lean-Burn Gas Engine, where engine operates according to the Otto Cycle [5].

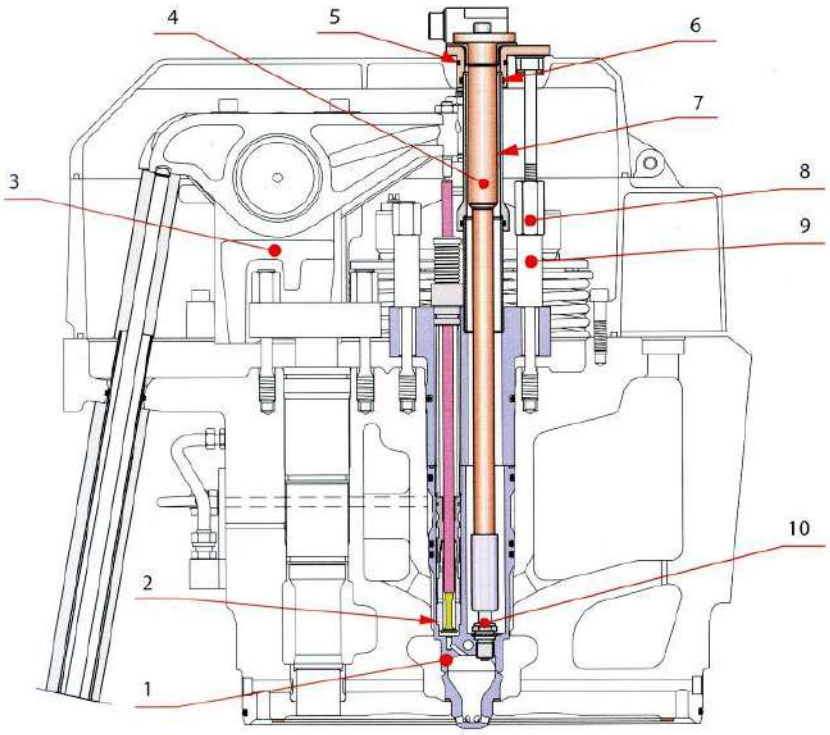

Figure 7 The pre-chamber, installed in the cylinder head of a Wärtsilä gas engine, type W34SG.

Source: [3]

Figure legend:
1 – pre-chamber,
2 – pre-chamber valve,
3 – rocker arm bracket for the pre-chamber fuel-gas valve,
4 – ignition coil,
5 – O-ring,
6 – O-ring,
7 – sealing sleeve,
8 – a nut,
9 – sleeve,
10 – spark plug.

In the figure 7, the fuel supply system for dual-fuel engine with pre-chamber is shown in the greater zoom. On those two picture we can see the spark plug which is responsible to start the ignition of air-fuel mixture. This types of engines are not a dual-fuel engine, but the idea of using the pre-chamber, and direct fuel injection in dual-fuel engines are used in the next type of fuel supply for them, described in next part of article. The location of the injector in the place of the spark plug is not a technical problem - as was described – the first pre-chambers in CI engines were equipped with a diesel injection system.

Such a type of gaseous fuel injection and its system of work is shown in the next figure number 8, presenting a schematic operation processes of single-fuel gaseous engines. This kind of engine fuel supply is described because of its importance in the further development of engines using gaseous fuel in dual-fuel mode. In the most maritime engines, their manufactures would not develop dual-fuel engines without using described here gaseous engines with the pre-chamber.

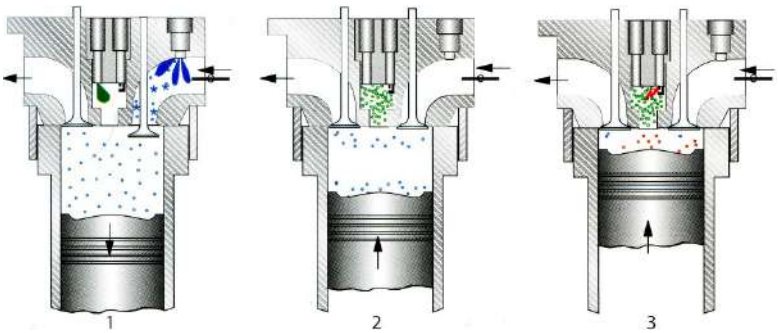

Figure 8 Operations of gaseous fuel maritime of Wartsila 20V34SG, gas engine principle.

Source: [3]

There is possibility to use two different high octane fuels where blue colour is responsible for gaseous fuel, and green colour is responsible for liquid high octane fuel like gasoline. Numbers shows the sequence of engine operations like in the figure legend.

Figure legend:
1 – fuel injection,
2 – air-fuel mixture forming,
3 – spark ignition of rich air-fuel mixture in pre-chamber.

Figure 8 present the mere use of two different types of injectors (or direct injector and indirect supply of gaseous fuel) makes possibility for use more than one fuel for combustion. Those type of dual-fuel engine, where another than gaseous fuel with high octane ratio is used is shown in the figure 8, where for example gasoline fuel may be used in direct injection into the pre-chamber. This is possible thanks to the use of more than one fuel injector, but the essence of this method - with pre-chamber usage - is the possibility to use two high-octane fuels like CNG, LPG or gasoline. In that conception, even NG in different physical properties could be used, for example to supply pre-chamber. This solution introduced the possibility of using two fuels in dual-fuel engine to a new level, but changing the technical solutions allowing the use of two fuels remained a technical conundrum for designers. One of the key problems was trying to fit two injectors into the combustion chamber.

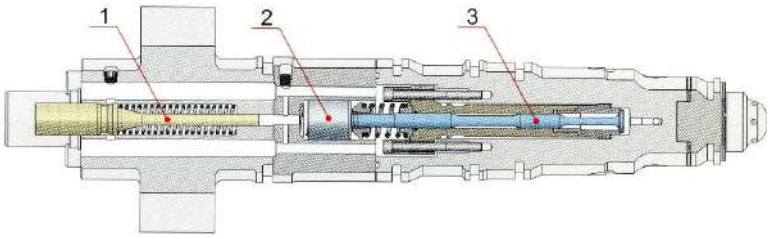

Figure 9 The mechanical-driven valve for the fuel gas to the engine pre-chamber.

Source: [5]

Figure legend:
1 – hydraulic plunger for oil pressure,
2 – hydraulic piston opens gas valve needle 3,
3 – guide for gas valve needle.

Production of a gas fuel injectors are problematic for reasons of physicochemical properties of gaseous fuel. Its dimensions are bigger than diesel injectors, materials from which it is made are different and the method of control, and thus the technology of its implementation, is also a bit different. Mechanical-driven injectors are a bit bigger than modern electronically controlled injectors. The structure of such an

exemplary mechanical injector is shown in Figure 9. Figure 9 shows the mechanical-driven valve in the gas fuel injector. Important things are engine and injectors – what is related to it – dimensions of the pre-chamber. Fit both of these devices in one cylinder is important, and impossible in smaller units. Even in maritime such a combination is not common and the difficulty of applying them together forced further development of direct fuel supply systems for dual-fuel engines.

Figure 10 A complete pre-chamber that will be installed in the centre of cylinder head.

Source: [2]

In small engines, pre-chamber is not big enough to fit two injectors into it. Even only the gaseous fuel injector could be too large for that application. In bigger engines pre-chamber and secondary fuel injector is not enough tight to apply both of them in the engine cylinder. In large marine engines pre-chamber could be big enough for that – in the figure 10 the photo of the pre-chamber mounted in the engine cylinder head is showed.

Gaseous fuel injectors are bigger than injectors intended for liquid fuels. Gas injectors intended for injecting fuel to pre-chamber could be possible only in large scale combustion engines.

Next chapter shows the techniques used for direct injection of gaseous fuel into the combustion chamber.

2.3. Direct injection of both fuels in dual-fuel engines

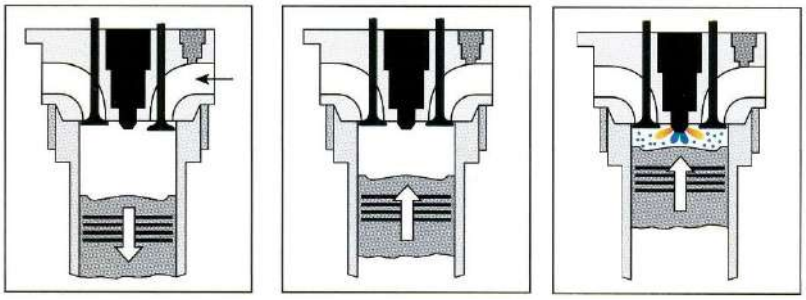

Figure 11 Schematic drawing of direct dual-fuel injection engine work.

Source: [3]

The last type of fuel supply for dual-fuel engine is direct injection of both fuels into the engine combustion chamber. That strategy could be realized by a few technologies, based on the both fuel injection process.

First is realized by use two or more independent injectors. Second idea is to use one injector for two or more types of fuel or substances – which was successfully realized by a few manufacturers. Those strategies are schematically shown in the figure 11.

This kind of fuel supplying in RCCI engine have got another name which is Direct dual-fuel Stratification (DDFS). Switching to this type of fuel system enables multiple injection of both or one of the fuels directly into the combustion chamber, depending on the needs of the engine developer or his researcher, and the appropriate adjustment of the fuel injection timing is a matter of engine fuelling strategy. Basically those dual-fuel engines have got simply fuel supply strategy, where the gas and a pilot fuel is injecting in a similar moment in time. It can be more than one pilot injection of diesel fuel, and different time of injection of fuels, but in standard dual-fuel or RCCI engines with direct injections system, both fuels are injecting near the end of the compression stroke. In all types of RCCI engines problematic is the angle of injection and its shape.

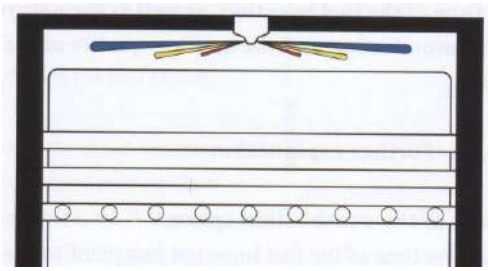

Figure 12 Shown of the fuel injection process with various fuel-injection possibilities in example of different angles of injection.

Source: [3]

In RCCI engine the injector angle must be different than in classic CI engine. Normal diesel fuel injector is placed near the centre of the top of combustion chamber and its spray angles are limited by its position and construction. Figure 12 shows the possible spraying angles for classical fuel injector and options to change them. Geometry of combustion chamber in RCCI engine is not similar to that known from classic CI diesel engine and it requires completely different shape of injection. Even the amount of high reactive fuel injection is smaller than in single fuel engine, the points where this fuel must be delivered is fundamentally different.

Figure 13 Correct injection spray shapes in the injection flow of a four-stroke trunk-piston engine (left), in a large two-stroke crosshead engine with three injectors (centre), and a two-stroke crosshead engine (right).

Source: [3]

Low reactive fuel mixed homogenously with air in RCCI engine must be located everywhere in the space of combustion chamber, so injected high reactive fuel must initiate ignition in all of those homogenous structure at the same time. Classical in CI diesel engines combustion chamber is located inside the piston, where all fuel is injected and burned. Figure 13 shows the correct positioning of the air-fuel mixture in the combustion chamber. From injectors construction depends the correctness of arising the air-fuel mixture and its arrangement in the combustion chamber. Combustion gases leave the combustion chamber space after burning and they spread over the entire space of the cylinder. In dual-fuel engine, those

burning process takes place in the entire over the piston cylinder head space. The shape of piston in good designed RCCI engine is closer to the known one in such an SI engine. For this reasons, the high reactive fuel injector must be different in those type of engines.

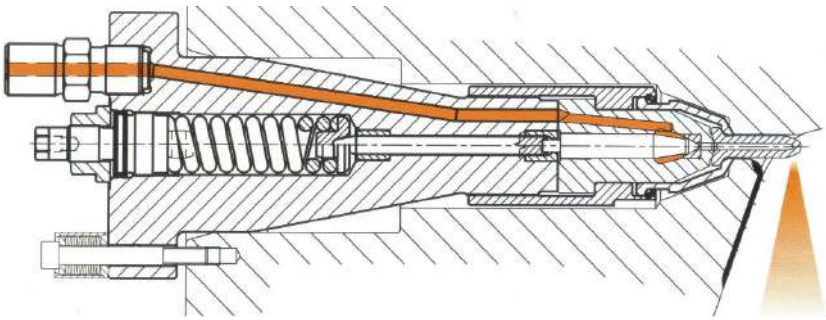

Figure 14 The position of the injector at the edge of the combustion chamber in a two-stroke crosshead diesel engine

Source: [3]

In addition to using an injector located centrally in the cylinder head, in many maritime engines injectors are located in different places of the cylinder head or such a cylinder. In marine two-stroke engines problem of different injection shape is solved by cylinder head injectors with a spray injecting at right angle or injectors placed in the middle of the cylinder – not in the cylinder head. In the figure 14 the drawn of the special right-angle spray injector is showed.

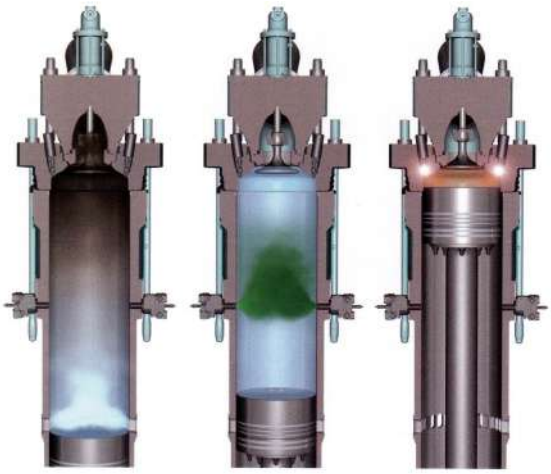

Figure 15 Fuel injection process in two-stroke maritime engines, where the principle for the injection of low pressure gas through the cylinder liner during the upwards movement of the piston is clearly visible.

Source: [3]

Figure 15 shows the injection system in two stroke marine engine, where gaseous fuel is injecting direct into the cylinder when the piston starts to move upwards to its top position. Injected gas has got much time to create air-fuel mixture, and at the top piston position secondary high reactivity fuel is injecting, starting the combustion process. The same effect can be achieved by spark plugs, but it would not be a dual-fuel engine, as defined in that article. Those solution is known only from a large-scale two-stroke maritime engines.

In many ways this solution is great, but prevents the implementation of more advanced fuel injection strategies based on multiple injections of low reactive fuel at the end of the compression stroke,

when the piston is already above the fuel injection line. Of course, apart from that, the use of this technique is possible only in two-stroke dual-fuel engines and their occurrence is limited only to the large-size engines, applied in stationary use or in maritime use.

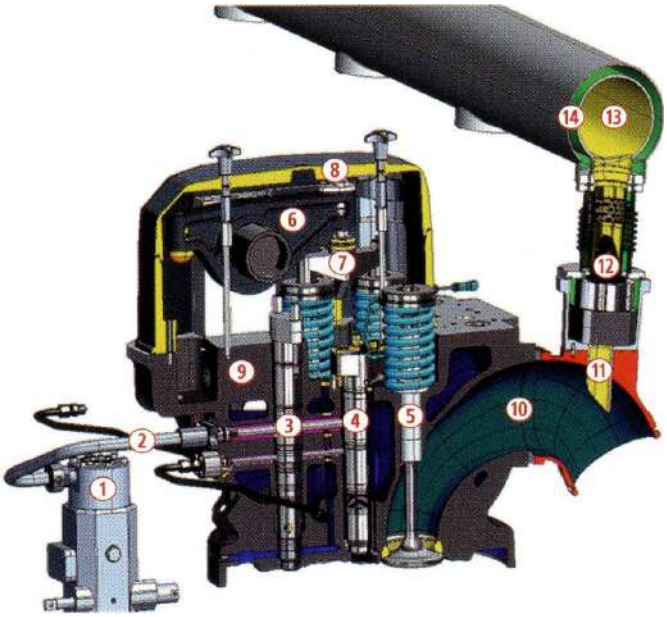

Figure 16 The fuel-gas-air mixture supply in Wartsila dual-fuel engines, where:

Source: [3]

Figure legend:
1 – high-pressure fuel pump,
2 – high pressure fuel supply pipe, double wall design,
3 – pilot liquid-fuel-injection nozzle,
4 – main-liquid fuel fuel-injection nozzle,
5 – one of two inlet valves,
6 – rocker arm of both the intake- and inlet-valves drive,
7 – yoke for the drive of both intake valves,
8 – valve cover,
9 – cylinder head,
10 – air-intake manifold,
11 – fuel-gas supply in intake manifold,
12 – solenoid valve for the fuel-gas supply,
13 – main-fuel gas supply-pipe,
14 – space between inner- and outer wall. In this space, where a lower pressure relative to the engine space is found in order to prevent fuel-gas leakages to surrounding areas.

One of the more technically advanced options for delivering both fuels to the engine's combustion chamber is usage of two independent injectors located in the engine cylinder head. In four stroke engines such a solution is much more popular than use one dual-fuel injector because of its construction complexity. Figure 16 shows the cross-section of cylinder head of Wartsila engine, where different numbers represent the different parts of the fuel supply system. On the left side of this figure we can see liquid-fuel pilot injection in the cylinder head for the ignition of the compressed fuel gas/air mixture. On right side there is the intake duct or manifold with the fuel-gas supply via the electronically-actuated and operated solenoid valves [3]. That is the development of the simply valve-supply concept of gaseous fuel supply. In that engine cylinder head sufficient space for two injectors has been found. But the functionality of those injectors corresponds to two working modes for this engine – dual-fuel working mode, and single fuel work, where only a diesel fuel is injecting. Those bigger injector, marked in the figure 16 by number 4 is responsible for the possibility of engine work only with a diesel fuel. Usage of two high reactive fuel injectors per one cylinder also response

for a very high demand for high reactive fuel in a bigger engine loads. The occurrence of such a phenomenon results from the fact, that high reactivity fuel is usually much more caloric, because its energy value is higher than those of low reactivity fuel. The necessity to use such a solution shows how many requirements must be met by injectors used in dual-fuel engines of this type, and the use of both fuels direct injection carried out by one injector would require it to perform tasks that would normally be performed by three, independently operating fuel injectors. In this matter, we are already stopping at the current stage of development of these technologies, because such solutions are currently under research, and even single fuel direct injectors in RCCI engines are revolutionary exceptions for today.

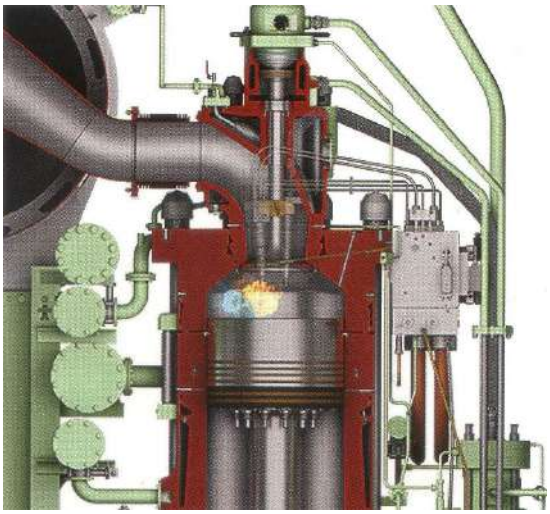

Figure 17 The gas injection or GI-version from MAN B&W for two-stroke crosshead engines in category IV engine.

Source: [3]

Example of such a unique fuel injector is the LNG injector used in MAN B&W for two-stroke crosshead engines in category IV (category of the largest engine size types). Injector applied there is the only existing possibility to inject liquid fuel-gas and diesel fuel into the one combustion chamber in dual-fuel engine. That kind of fuel supply in marine engines are commonly use in two-stroke engines too, but, direct injection of LNG is realized only by MAN B&W for now [3].

In the figure 17 there is the cross-section of that engine with two injectors of different fuels. Two injectors used in other engines practically always include the injection of two fuels in liquid form, if both of these fuels show such a state of aggregation under normal physical conditions, or direct injection of gaseous fuel when such fuel is used as a low reactive fuel. In large scale engines those solutions are becoming common, but in smaller scale, in engines used in cars and trucks the usage of two independent injectors is rare in RCCI engines.

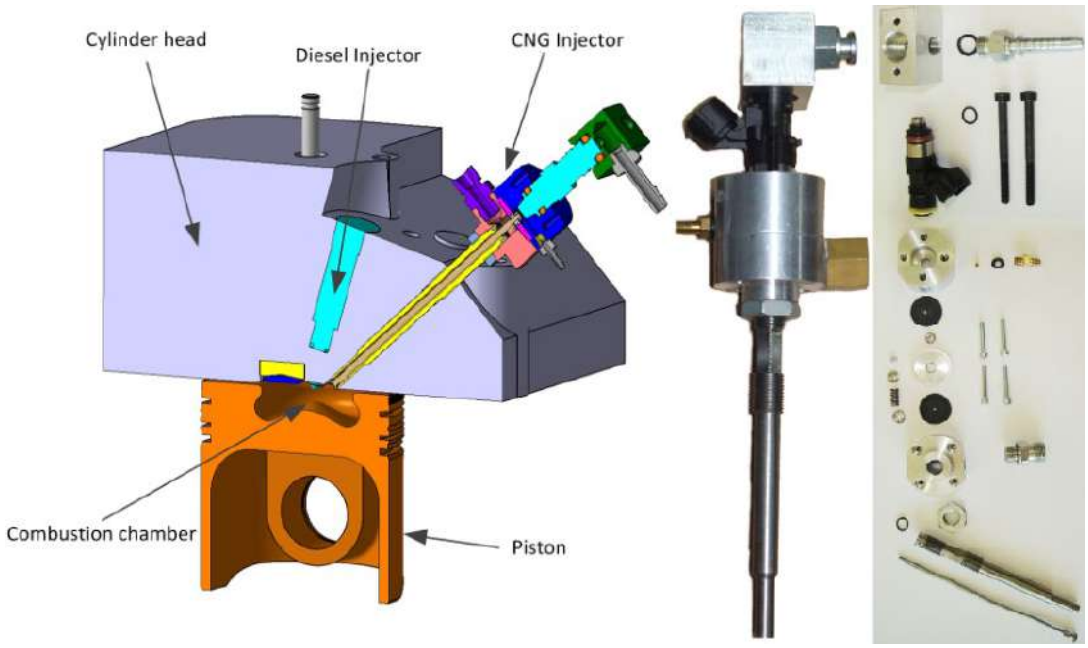

Figure 18 The localisation of the injector in CAD drawing of the cylinder head on the left and a custom gaseous-fuel injector with a parts of it on the right.

Source: [6]

Conception from figure 18 is created at the university research [6]. The dual-fuel engines with direct injection of both fuels for passenger car engines exist only in researcher concept. There is the crafted research injector created for direct injection of gaseous fuel which is shown on the right part of figure 18. Those type of injectors are not commonly used in marine engines. Fuel supply systems where is enough place to install two independent fuel injectors are the most popular in large engines, . Direct injector technology for gaseous fuel is still in developing in every engine size types. In [6] researchers make their own injector for engine research. In the free market it is hard to buy gaseous fuel injector, and basically impossible to buy dual-fuel injector – designed to inject two types of fuels simultaneously. In the figure 18, on the left side of the picture is showed the cross-section of the cylinder head with gaseous fuel injector and connection block, which is used for gaseous fuel supply for the injector. On the right side there is the research injector with parts used for its build. The idea was to create direct injection gaseous fuel injector which can be used in dual-fuel engine, because there are no such products on the market. The problem could be still the shape of injection spray, when in a lot of RCCI engines so simply option of injection quite straight into the combustion chamber is not desirable.

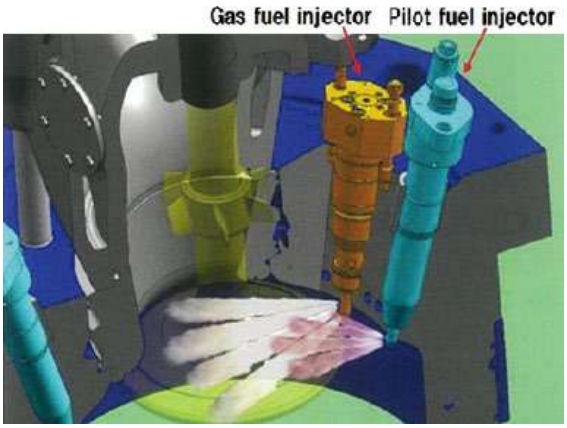

Figure 19 Cylinder cover design for two-stroke dual-fuel engine.

Source: [7]

In the figure 19 the visualisation of two independent injectors shows the possibility of injecting high and low reactivity fuels directly into the combustion chamber at the right angle, enabling coverage of the entire space in the combustion chamber by both fuels used in the dual-fuel engine.

Last undescribed here parameter is the pressure of the injection. This parameter is extremely important for the formation of the air-fuel mixture. For diesel fuel injection that parameter was maximized by introducing the common-rain injection system, where a single fuel line tank supplies diesel fuel under high pressure, ready for injection it to all of the cylinders in the CI engine. The same idea is being used in the direct injection of gaseous fuels.

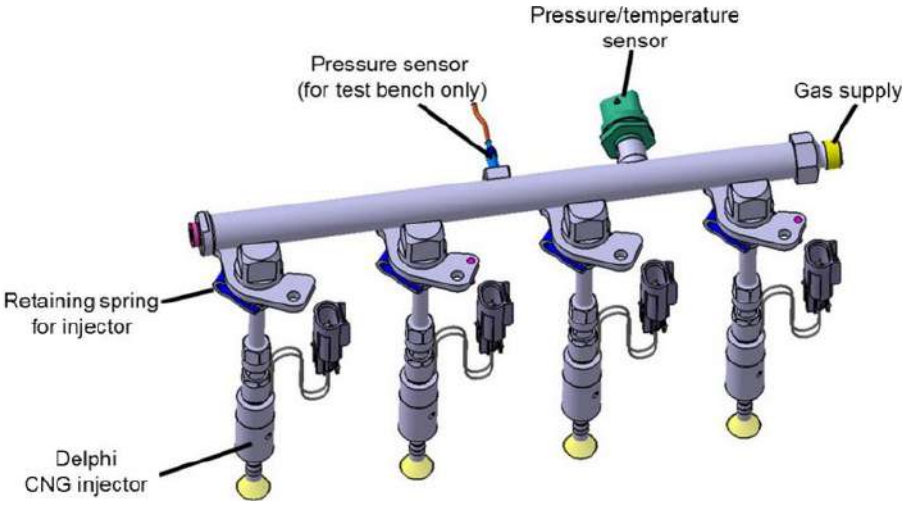

Figure 20 Common rail system for gaseous fuel – CNG – from Delphi company.

Source: [8]

The same type of injection system is available for gaseous fuel – an example can be common rail system for CNG fuel showed in the figure 20. The common rail system for CNG fuel is a promising technology that could help in the future to introduce a dual-fuel direct injection system into common use in the piston engines. It has proven itself well in supplying diesel engines, so probably the future of gas supply to internal combustion engines will also belong to it.

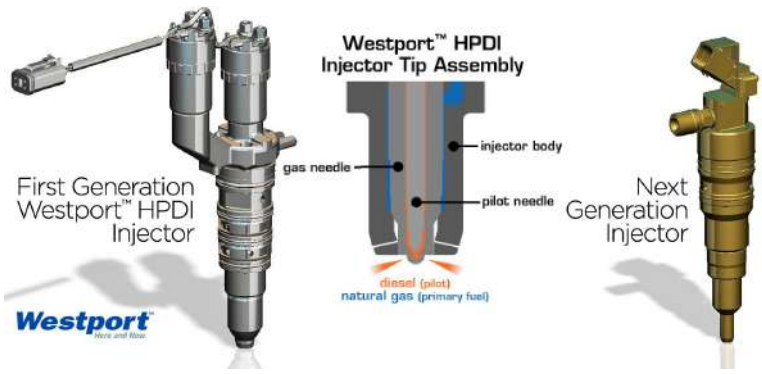

Figure 21 View of two Westport HPDI injectors generations – on the left the older version of injector, and on the right the newest version. In the centre there is a drawing showing a principle of injector operations.

Source: [9], [10]

That possibility to install two injectors for one cylinder in the cylinder head is possible only in larger engines. In small engines to use dual-fuel direct injection fuel supply, often only one fuel injector can be

used. In the passenger cars, those type of fuel supply is still not applied yet. In trucks, there is only one manufacturer who supplied dual-fuel engines by one injector. That injector – in two generation options – is showed in the figure 21. HPDI (High Pressure Direct Injection) injector is developing only for few engines dedicated for it [9]. That product is proposed by Westport corporation [11], and described in all relevant publications covering dual-fuel engine solutions for cars and trucks [12]. This is only one factory-made injector available in the global market. Schematic show of direct dual-fuel injection of it, is shown in the middle of the figure 21. Its use offers enormous possibilities in terms of implementing various engine fuelling strategies, but it is only available in a small range of engines from one manufacturer and limited by the and to a limited capacity to a certain extent only available for a specific group of trucks .

3. CONCLUSION

Dual-fuel engines have got diverse types of fuel supply systems. The systems ary for different sizes and uses of engines in which they were applied. Many systems are still under testes and development, especially in smaller engines for passenger cars. Contrary to applications in small engines, dual-fuel supply system are commonly use in marine engines working on ships and in power plant, or manufactories with their own energy generation system based on internal combustion engines.

The very promising technology, which are RCCI engines, requires the use of very sophisticated types of injectors. Injectors of that types are in use only in engines bigger than those used in passenger cars, and access to that solutions is limited.

Supply systems for dual-fuel maritime engines stands at a noticeably higher technical level and allows the use of supply strategies not available for smaller engines. The only upgrade device unavailable in this engine size category is a single multi-fuel injector, although such a product is available for engines used in a certain group of delivery vehicles.

In small size engine categories, the technique of supplying the engine by two independent fuels requires refinement and implementation of technologies already known and working in larger engines. For all engines used in passenger cars in which the dual-fuel technology can be used, intensive development of devices supplying the engine with alternative fuels is required, the use of which allows to reduce CO_2, which emission is strongly limited in passenger cars.

A very important issue is also the development of RCCI engines, which as they develop will constantly require changes and corrections in the method of fuel supply to the engine. The amount of work on the development of fuel delivery systems in RCCI engines will be needed as long as the engines themselves are still in the development stage.

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A review of technical solutions for RCCI engines

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Engines working in dual-fuel mode need special conditions to ignite air-fuel mixture without spark plug in a good moment with high combustion efficiency. To create homogenous air-fuel mixture the conditions in the cylinder are even more demanding. Many concepts of ignition was developed, but the most effective needs perfect mixing of fuel and air, which is a serious technical challenge. Technical solutions for dual-fuel engines cover the complexity of these problems thus leading to the further development of ignition systems in internal combustion engines. Fuel supply systems, the operation strategy of them, the shape of the combustion chamber are the most important elements to change and develop for correct operation of dual-fuel engines. The literature analysis showed a small amount of research carried out to optimize the operation of dual-fuel engines The variety of engines in which a dual-fuel system can be used requires much more research about them, and solutions necessary for their correctly operation.

Key words: *HCCI, RCCI, Dual-Fuel, DDFS*

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1. Introduction

Internal combustion engine (IC) and the combustion process itself are constantly developing from XIX century until today. The amount of energy released in the combustion chamber and the composition of raw exhaust gases both depends of combustion. The improvement of these processes in first century of IC engines existence was mainly based by improving the shape of combustion chamber. The next century opened an opportunity for researchers to develop new solutions of the air-fuel ignition methods. These fresh solutions introduced new directions in the development of internal combustion engines.

1.1. Explication

IC engines always emit undesirable exhaust components, even if they are the most efficient, modern and equipped with, even the best exhaust gas cleaning systems. Usually systems of this type may be large and complex [9]. The typical composition of exhaust gases from marine engines are shown on Fig. 1.

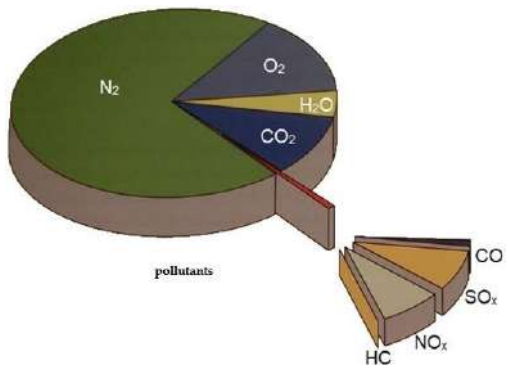

Fig. 1. The typical composition of exhaust gases from the modern, marine diesel engine (reproduced from [9])

Composition of exhaust gases is strongly dependent on what kind of fuels are used in the engine. In order to be able to use many types of fuels while maintaining a modern and well-developed combustion technology, engines have been modernized, allowing for the combustion of many fuels at the same time. This type of solution is Dual-Fuel (DF) system. DF may turn out in the future as a key point in the development of IC engines and reducing the emission of harmful exhaust components.

This work was created in order to distinguish specific solutions enabling the implementation of the most effective DF combustion systems in classic piston CI engines. A lot of works focusing on the literature review have already been done. The researchers selected engines used in various research studies, with a method of air-fuel mixture combustion similar to that one was focused for this study [22]. A very detailed review of carried out tests in analyzed studies engines performance are presented in detailed tables and summarized in the description without going into the deep technical details regarding the design of the engine and the technical solutions used in them. More detailed data on technical solutions are presented in the different studies [24, 25]. Authors decided to refine and expand with additional information from further literature analysis. Works such as [29] and [23] present a detailed description of the research carried out, and the influence of various types of fuels on engine performance, that which are not main focus in this study. The above-mentioned works also take into account the simulation results, which in this paper is limited to presenting the concept of technical solutions that may bring beneficial effects. Model simulations for the marine engine presented in the study [23] are a very important part of research on this topic due to the advantages that slow- and medium-speed engines working in the DF mode have. This study is also largely based on technical solutions used in marine engines, the development of which largely determines the future of DF engines, which is supported by many publications [18–20, 34], including technical solutions in marine engines.

The studies quoted here may constitute the core of considerations, developed with technical issues enabling the

implementation of engine operation in DF systems. Important issue which is also highlighted in this article is the flow of technology from large industrial engines to small drive units. The technical solutions presented in this work show the current state of technology and define the possibilities that arise from their widespread use and the combination of independently used technical solutions.

2. IC engines ignition systems

A simplified view of engine development through the centuries of existence of IC engines reflects the mainstream of their development. Figure 2 shows the important types of air-fuel mixture ignition systems regarding to the date of their construction.

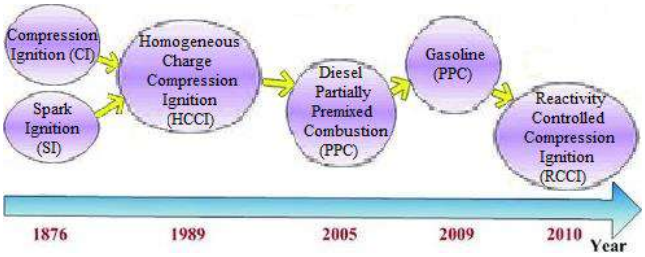

Fig. 2. Chosen ignition methods in ICE by year of construction (own graphic based on data and graphics from [25])

Ignition systems in IC engines are being developed one dependent on the other and the new design is replacing the old one. Relations between different ignition systems allow for further exploration of various IC engines concepts. Different ignition types are shown on the graph showing addiction of their use on fuel type (Fig. 3).

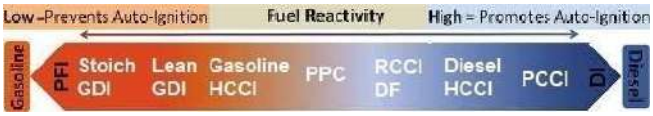

Fig. 3. The dependence of the methods of ignition of the air-fuel mixture in IC piston engines on the method of fueling the engine and the type of fuel used (own graphic based on data and graphics from [25])

Different engine ignition types are designed for use with certain fuels. The combination of some solutions may result in a significant increase in the efficiency of IC engines. Their development is possible only thanks to research on single types of ignition. DF technology is now paramount in further improving engine combustion.

2.1. IC engines ignition systems

The typical way of using fuels in IC engines is based on the use of one type of fuel that ignites either as a result of forced spark ignition (SI) or as a result of compression ignition (CI). It allows for better use of the chemical energy contained in the fuel due to the higher efficiency of this type of engines. However, the highest efficiency is currently achieved thanks to the ignition of a homogeneous air-fuel mixture, which can be ignite by spark or by compression. In Homogeneous Charge Compression Ignition (HCCI) the ignition of all the fuel in the combustion chamber occurs simultaneously. This combustion takes place at a lower temperature than in a classic diesel engine, which leads to lower heat losses and the less NO_x formation. It translates into higher thermal efficiency of engines with this type of ignition. However, in order to achieve ignition in the combustion chamber of the engine, very specific conditions are required, which are difficult to achieve [5]. Effective control of the moment at which ignition takes place in all areas of the engine operation is currently beyond the possibilities of designers. Fuel used in that type of engine must have a high octane ratio, because of its early injection. For this reason, one of the variants of engines with this type of ignition method, which has the full potential of use today, is an engine powered by two fuels by Reactivity Controlled Compression Ignition (RCCI).

RCCI is an ignition method in which high reactive fuel such as diesel fuel injected directly into the cylinder mixing with the air and self-ignites a of low reactive fuel, which is delivered to the combustion chamber earlier via indirect injection. The same way of fueling the engine characterizes a classic dual-fuel engine in which the fuel-air mixture is not homogeneous. Every modern dual-fuel engine is currently developed in the direction that allows the formation of a homogeneous air-fuel mixture [32]. Schematic of classic dual-fuel injection system is shown on Fig. 4.

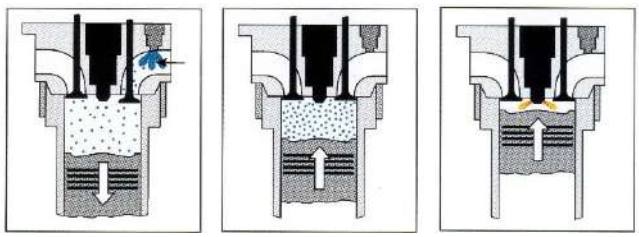

Fig. 4. Schematic drawing of direct dual-fuel injection engine work (reproduced from [9])

Before RCCI, there was a few conception which could generate better IC engines parameters. When the first engines in which a homogeneous fuel-air mixture was burned were developed, partially pre-mixed combustion (PCI) and partially premixed compression ignition (PPCI) was known. This type of ignition allows a mixture that has already been pre-mixed with air to be burned. Only when the last dose of fuel is injected into the cylinder, a fuel-air mixture is formed. In contrast to PCCI or HCCI, the name emphasizes that only a part of the fuel was mixed with air before ignition. Thanks to this method of fuel combustion, a large part of it is burned earlier than in conventional CI [12]. Gasoline Direct-injection Compression Ignition (GDCI) engine is operating basically on the same principle as PPCI or even PCCI. Difference is that the fuel used in this case is gasoline. The operation of such an engine is difficult to maintain, but since 2010 the concept of such an engine has been constantly developed [15]. HCCI ignition not possible at the current level of knowledge for use in the entire range of operation in modern piston engines. This is the basic difficulty with the widespread introduction of this type of ignition in IC engines, and it consists a number of technical problems still unsolved. Considering that these problems must be faced by HCCI and PCCI engine designers, a new concept has been introduced for a specific type of ignition: Spark (Plug) Controlled Compression Ignition (SPCCI) or Spark Assist Compression Ignition (SACI). It is self-

ignition controlled by spark plug. It is a specific method of self-ignition of a homogeneous air-fuel mixture located in the combustion chamber [30]. This type of ignition is currently in use, but mastering the ignition of a homogeneous air-fuel mixture without spark plug assist is now possible only in RCCI engines, and these kinds of ignition are the keys to introduce diesel HCCI or PCCI engine.

2.2. Dual-Fuel ignition systems

RCCI is a special type of HCCI engine that allows to use many types of fuels with high combustion efficiency and low emission of harmful exhaust components.

The low-temperature method of burning fuels in HCCI engines allows to improve the efficiency of the engine, thanks to the reduction of energy lost during combustion process. For the further development of auto-ignition engines, with increasing the range of fuels used in it, it is necessary to control the air-fuel mixture auto-ignition by using the injection of second fuel injected into the combustion chamber. The development of modern compression ignition engines is strongly dependent on engines with this type of ignition. Using two fuels with different physico-chemical properties makes it possible to control the time of compression ignition and allows to use engine in full range of operation [2].

The engine working in dual-fuel mode with diesel fuel as the reactive fuel requires only the already mentioned, pilot dose of diesel fuel to initiate the ignition. The Fig. 5 below shows a schematic view of the power supply and ignition in an RCCI engine powered by a mixture of gas fuel as the low reactive fuel and diesel as the high reactive fuel [29].

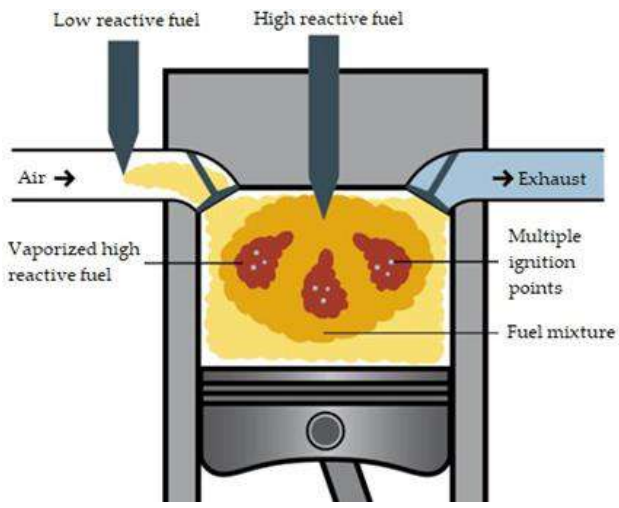

Fig. 5. Ignition and combustion in the RCCI engine (custom artwork based on [37])

In RCCI engines, low-reactive fuel, with a high octane number, is supplied to the engine along with the air through the engine's intake system. Low-reactive fuel is compressed and mixed with air to form of pre-mixed, combustible fuel-air mixture. It can be classic gasoline, alcohol, LPG, CNG or any liquid fuel with high octane ratio. Due to the high resistance to auto-ignition and knocking combustion, this mixture will not ignite despite the high temperature and pressure already in the cylinder. In the vicinity of TDC, high reactive fuel (diesel, bio-oil, GTL, CTL or HVO) is injected. This kind of fuel must be characterized by good self-ignition properties, including a low ignition temperature and the ability to large atomization of the fuel in the process of its injection. Ignition of high-reactive fuel initiates ignition of the air housing with low-reactive high-octane fuel. The essence of controlling the moment of ignition and the course of combustion of fuels used in the engine is the injection time and properties of the reactive fuel.

When using additional, low reactive fuel in DF CI engine, it's necessary to increase the amount of injected diesel fuel due to the insufficient resistance of the additional fuel to knocking combustion. Its occurrence depends on many factors, but mainly the type of fuel and the compression ratio in the engine. Knocking combustion causes the formation of uncontrolled foci of fuel self-ignition, which hinder the proper course of the combustion process and the operation of the IC engine. Its presence determines the possibility of using fuel in the engine and the degree of substitution of the basic fuel.

There is also a different fueling strategy in this type of fuel called Direct Dual Fuel Stratification (DDFS) – the difference is the use of direct injection for both fuels used in the engine [26]. Direct injection of two fuels requires the use of several injectors, or a special design injector, such as the offered product proposed by Westport corporation [34], and described in all relevant publications covering dual-fuel engine solutions [16]. Schematic show of direct dual-fuel injection work is shown on Fig. 6.

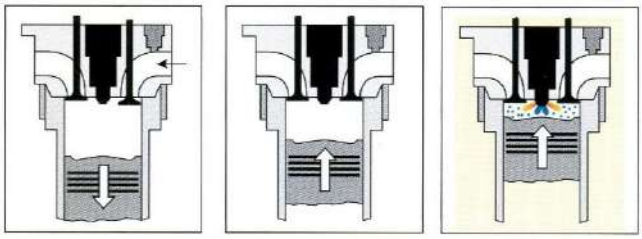

Fig. 6 Schematic drawing of direct dual-fuel injection engine work (reproduced from [9])

DDFS is a specific kind of RCCI engine, where both fuels are direct injected into the combustion chamber, where part of the low-reactive fuel is injected early enough to mix with the air, while the remaining part of the fuel can be injected in the piston's upper dead center – depending on the engine's needs [13].

There are many strategies for fueling RCCI engines and they are constantly being developed. Many solutions depend on fuels which are used in a particular RCCI engine. In the case of gaseous fuels used as low-reactive fuels, the basic parameter that is being attempted to maximize is the percentage of replacing diesel fuel with gaseous fuel. The value of this parameter strictly depends on the level of adaptation of a given engine to dual-fuel operation. Researchers want to achieve a high value of this parameter due to fuel prices and the level of emission of harmful exhaust components that can be achieved with the use of certain fuels. In most DF engines, the value of this parameter is not constant and it is not possible to maintain its high

value under high engine load. Depending on the stratification and the degree of mixing of high-octane fuel with air, the course of its combustion can be characterized in various ways, and this is often a key issue in RCCI combustion studies. The amount of fuel of a given types depends to the greatest extent on the engine load and the fuel injecting system. Figure 7 shows the chart with the dependence of the engine powered by natural gas (NG) and diesel fuel on the possibility of using both fuels in different proportions.

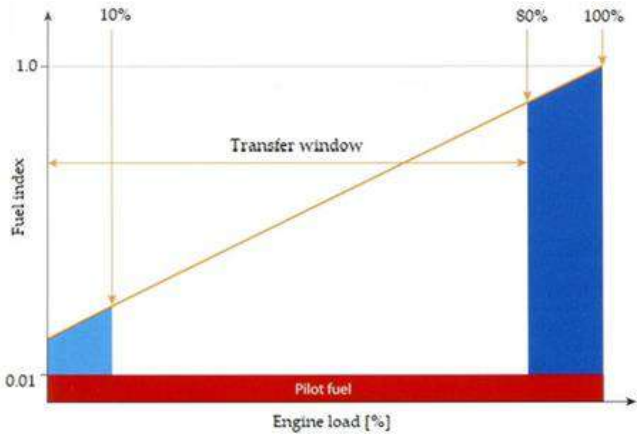

Fig. 7. The various operating conditions known as "modes" for dual-fuel engines, where horizontal is the engine load between 0–100%, and vertical is the fuel-index from 0 to 1 or 0 to 100% (reproduced from [11])

Figure above shows the DF engine fueling strategy: where the red color is pilot-fuel injection active, light blue shows that engine can be started, operated and loaded in gas mode– here engine will trip to back-up fuel after a present time, if engine load is not increased out of this area, and dark blue color means that this engine is able to run up to 100% load on either fuel, but transfers from liquid to gas are not permitted above 80% load – transfers from gas to liquid are permitted at any load. As high the fuel index is, as much liquid fuel must be used [11].

Fuels used in engines significantly influence the shape of this type of graph. Essentially all liquid and gaseous fuels of relatively high or relatively low octane numbers can be used in RCCI engines. Only those fuels that cannot be clearly classified as highly reactive or low reactive fuels would be difficult to apply in RCCI engines. If it is possible to create a homogeneous mixture of low reactivity fuel with air, this fuel may also self-ignite under the influence of pressure exerted on it by a rapidly burning highly reactive fuel, and in those option engine can be called RCCI engine. If the self-ignition occurs without forming homogeneous air-fuel mixture, the engine can be called only a DF engine. From the fuels used in those type of engines alcohols that are bio-based fuels contain in their compounds may facilitate the formation of a homogeneous mixture in the combustion chamber, so sometimes it depends on the fuel used in a given engine.

Thanks to the use of a low-reactive fuel homogeneously mixed with air, the efficiency of using the chemical energy contained in the fuel increases. Its increase is also influenced by the even temperature distribution in the combustion chamber. This reduces the total amount of heat penetrating the cylinder walls and prevents the formation of hot spots within the cylinder what is clearly visible in Fig. 8. The maximum combustion temperature in the combustion chamber is lower than in conventional compression ignition engines.

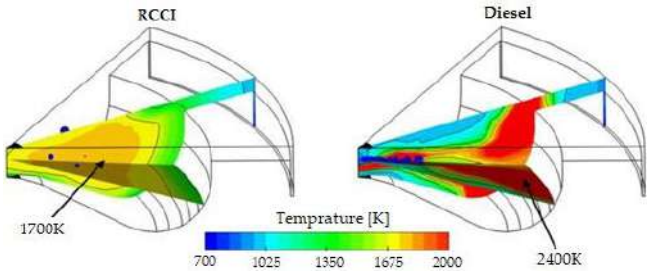

Fig. 8. Temperature distribution in the combustion chamber for a RCCI engine powered by CNG and diesel and a classic diesel engine powered only by diesel (modified graphic based on [7])

Due to the even and low-temperature course of the combustion process, the emission of nitrogen oxides, solid particles and soot is reduced, just like in the classic HCCI engine. High flame temperature is one of the main factors causing the formation of nitrogen oxides during the combustion process. Nitrogen oxides are not formed in RCCI engines as intensively as in a classic diesel engine. Due to the combustion close to the stoichiometric composition, the amount of oxygen and nitrogen in the combustion chamber during the process is therefore reduced compared to a classic diesel engine [36].

The real benefits of lowering the combustion temperature appear in large, low-speed marine engines, where the problem of cooling the combustion chamber is solved in more ambitious ways than in a small piston engines. Piston and combustion chamber cooling is more advanced, because it is possible in engines of such dimensions. Figure 9 shows the drilled holes in the large engine piston.

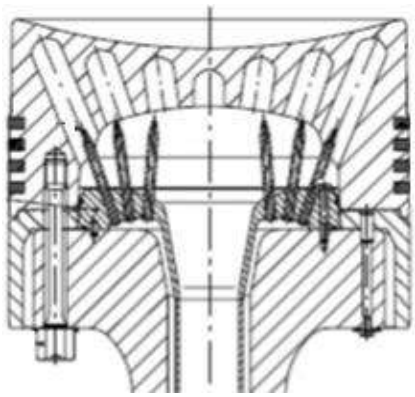

Fig. 9. Example of piston cooling drills in the Sulzer IC engine (reproduced from Pounder's Marine Diesel Engines [35] with permission from Elsevier)

Drills made in the piston enable its effective cooling with the use of engine oil. The differences in combustion temperature between conventional CI engine and HCCI or RCCI engine can potentially eliminate the need for such a solution. The lower temperature reduces the occurrence of the NO_x formation effect, which obviously allows limiting

the participation of SCR in changing the composition of exhaust gases. Low emissions that can be achieved in research engines requires significant interference in the design of the IC engine, preceded by its research and optimization of the design for the combustion of a new type of fuel. In both single-fuel diesel engines and dual-fuel diesel engines (RCCI), it is required to adapt the combustion chamber to run on a new type of fuel [7]. In the present CI engines the combustion chamber is widely profiled inside the piston. Fuel injection shape depends on to the combustion chamber shape too. Figure 10 shows the potential of injection angles for typical DI engine.

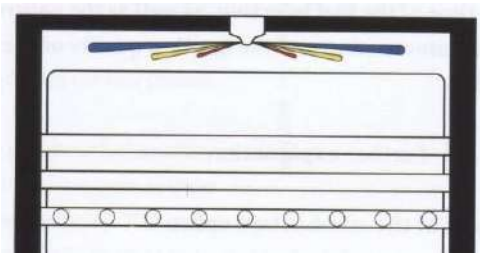

Fig. 10. Shown of the fuel injection process with various fuel-injection possibilities in example of different angles of injection (reproduced from [9])

This ability to inject fuel in different angles is greatly needed in RCCI engine. Some studies [4, 8, 29] has repeatedly shown that the shape of the combustion chamber in RCCI engines is very important for the formation of harmful chemical compounds contained in exhaust gases. Standard injection shapes in CI engines are showed in the Fig. 11. Those shapes and angles of injection in standard CI engines are not good enough for RCCI.

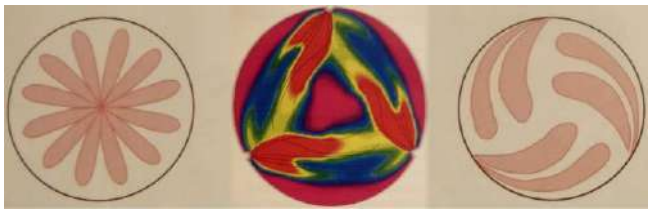

Fig. 11. Shown of the correct injection spray shapes in the injection flow of a four-stroke trunk-piston engine (left), in a large two-stroke crosshead engine with three injectors (centre), and a two-stroke crosshead engine (right) (reproduced from [9])

Those shapes and angles of injection in standard CI engines are not good enough for RCCI. While DF engines are often derived from standard CI diesel engine conversion, the limitations that result from this should be borne in mind. Defining the needs for RCCI engines reliance on such limitations, and on fact, that the most that the biggest changes involve the engine combustion chamber. Next chapter will focus on these shifts, where they can be divided into technical solutions including the construction of the piston – combustion chamber and the fuel injector.

2.3. Important RCCI technical solutions

The basic technical modification that should be carried out on an existing engine adapted to work in RCCI mode is the change of its combustion chamber and modification of the power supply system.

Important in that point is that combustion chamber space that can enable proper work of the single fuel CI engine will have a swirl ratio of low reactivity fuel problems and can have a hot spots on its surface, what which disqualifies the possibility of using low reactive fuel with an octane number that does not provide resistance to spontaneous combustion in contact with such a hot point on the surface of the combustion chamber. When dealing with this problem, designers often use additional cooling of some elements (e.g. injectors), which heat up excessively in the combustion processes [28].

The shape of the RCCI combustion chambers, with different compression ratio options, compared to the combustion chamber of a modern CI engine is shown in the Fig. 12.

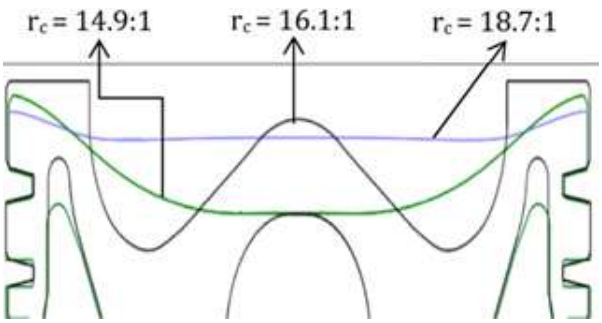

Fig. 12. Comparison of RCCI and classic CI engine piston with combustion chamber inside of it (reproduced from [31])

Form of combustion chamber in RCCI engine is fundamentally different than in classical CI diesel engine. The swirl ratio of air-fuel mixture must be higher, surface of combustion chamber can be much smaller, and the shape itself will be softer. In the Fig. 13 is shown the cross section of the piston where the sharpest edges of the combustion chamber are colored by the red circles.

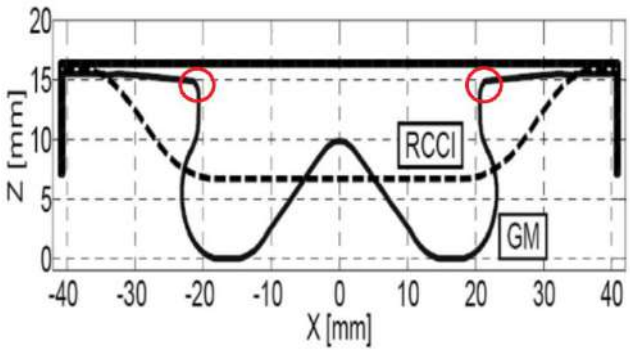

Fig. 13. Cross-section drawing of GM stock piston (solid line) and modified RCCI piston (dotted line) (reproduced from [6])

Injection of high reactivity fuel in RCCI engine takes place differently than in a classic diesel engine. Direct injection of high reactivity fuel with a centrally located injector in a classic CI engine requires the use of a combustion chamber located in the piston, the shape of which is optimal only for single-fuel engine operation. DF operation needs different combustion chamber shape. The amount of this fuel itself is many times lower in most of the engine operation area, so the space required for uniform mixing of the

injected fuel is not so great, but the places where the fuel must be delivered are extended by the area above the piston and the combustion chamber hollowed in it. Because there is already low-reactive fuel in the entire in-cylinder space over the piston, high reactive fuel must be delivered to an area already filled with this fuel. It forces a change of the place where the fuel will be injected. The changes in the injector design alone will not be enough and the researchers decided to change the factory piston into custom piston. More important is the issue of correct swirling of the air-fuel mixture [4, 8, 29]. These parameters have the greatest impact on the degree of swirling of the fuel-air mixture, which for different proportions of the fuel mixture has a different impact on the efficiency of their use. The red edges of the combustion chamber in the RCCI engine constitute a serious barrier to the proper distribution of the injected highly reactive fuel inside the combustion chamber. It is necessary in existing engines to modify them in order to improve the distribution and swirling of the air-fuel mixture. Researchers in [1] prepared project from the original GM piston to modify it into investigatory RCCI piston. Effects are shown in the Fig. 14.

Fig. 14. Photo of modified RCCI piston and stock GM 1.9 L piston (reproduced from [1])

Project of RCCI piston has been meticulously prepared and, after its construction, installed and used for further research. Researchers indicate an increase in the thermal efficiency of the engine thanks to the use of such a piston. Such scientific efforts are very much needed because there is not enough research into RCCI engines to interfere with the essential design of existing engines. In the Fig. 15 plans comparison both pistons are shown.

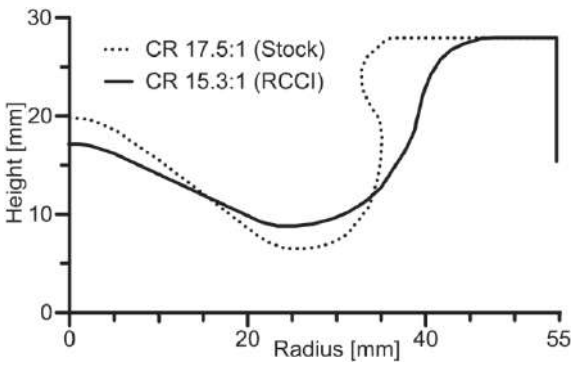

Fig. 15. Differences between piston bowl geometries in different compression ratios for RCCI engine, and classical diesel engine (reproduced from Applied Energy [22] with permission from Elsevier)

In order to adjust the combustion chamber in the piston for DF combustion, the researchers proposed an alternative shape of the combustion chamber [22], which is shown in comparative Fig. 16, containing two numerical models of combustion chambers.

Fig. 16. Potential of piston combustion chamber geometries (reproduced from [22])

Conceptions of bowl piston geometry for RCCI engines are new and different. Most of them are computer simulations concepts, but researchers [1, 7] create real piston which was showed in the Fig. 14, and prepared technical documentation for different piston shapes for testing RCCI engines, showed in the Fig. 17.

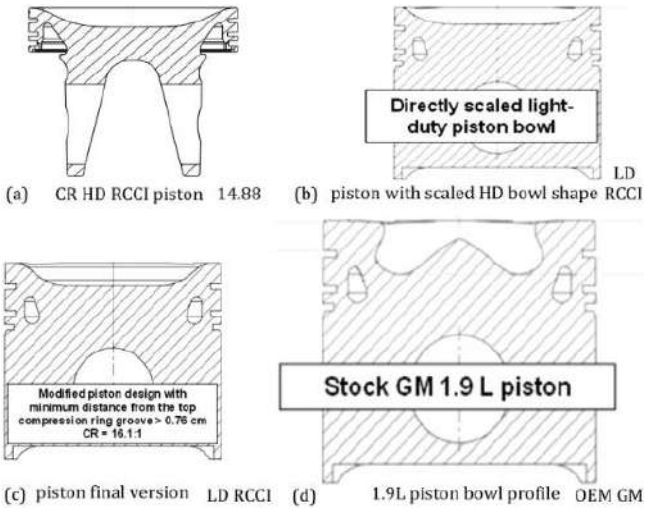

Fig. 17. Comparison of piston bowl geometries (graphic based on [6])

The geometry of RCCI piston is strongly dependent on the injector position and spray shade. There is multiple option for further research about it. The good example can be quoted in work where researchers focus on negative valve overlap (NVO) process as a promising variable valve actuation (VVA) measure to improve low-load efficiency in RCCI engines and (external) fuel reforming techniques to improve low temperature combustion (LTC) performance in those engines [19]. Was shown that low-load RCCI operation can be improved with a combination of NVO and direct diesel injection into recompressed residuals. Validating this hypothesis was the goal of their research in which they create computer simulation of the present research. In this work it's possible to find that even the fuel supply system effects on correct injection shape and piston bowl geometry what is showed in the Fig. 18.

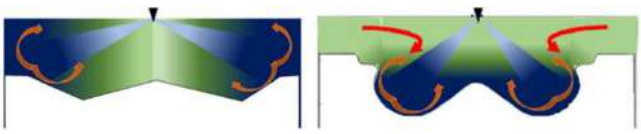

Fig. 18. Piston bowl geometries and injection configurations in a different fuel supply systems in RCCI engines (reproduced from Applied Energy [19] with permission from Elsevier)

The way the fuel is supplied to the RCCI engine affects the geometry of the combustion chamber and the shape of injection of highly reactive fuel. On the left part on Fig. 18 the Single Point Injection (SPI) with mixer after turbocharger is showed, and on the right site is Port Fuel Injection (PFI) to individual cylinder ports. It is important to say at this stage that the generally specified methods of supplying the engine can be correlated with other engine supplying techniques. In the case of RCCI engines it can give measurable effects, as in the case of work where NVO allowed to achieve heavy exhaust gas recirculation RCCI case shows up to 75% methane emission reduction is attainable at low load through combined effect of increased mixture homogenization and elevated temperature at intake valve closing [19].

When it comes to fuel supplying RCCI engines, much depends on the size of the engine, and its construction. In smaller engines, which are commonly use in land vehicles like cars and trucks – space for using more than one fuel injector in cylinder head per cylinder is limited. Basically engines like that are equipped with fuel supply installations (Fig. 4 and 5), where low reactive fuel is supplied by PFI, and only high reactive fuel can be injected directly into cylinder. That technology is commonly used in all engines capacity, the idea of fuel delivery may be slightly different, but usage of carburetor or the gaseous fuel inlet valve in bigger engines are not fundamentally different from each other and their technical construction are well known. Figure 19 shows a scheme of a gas mixer.

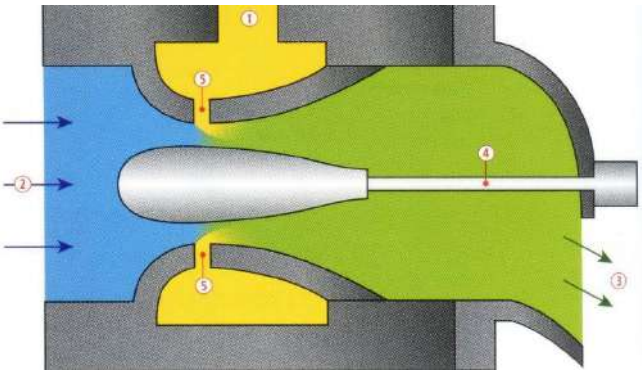

Fig. 19. A cross-sectional view of a gas mixer with an adjustable venturi insert, where 1 is the gaseous fuel supply, 2 is the air supply, 3 is the air-fuel mixture to the intake manifold of the gas- or dual-fuel engine, 4 is the adjustable venturi insert shaft connected to stepper motor, and 5 are bores for the supply of fuel-gas in the throat of the gas mixer (reproduced from [9])

The differences in the construction of such devices may differ in details but also in size, because they are used in all types of IC engines, even in the largest maritime engines. Figure 20 shows samples photos of throats intended for gas mixers mounted in the intake manifold.

When it is necessary, gas mixers working similar to carburetor are replaced by fuel injectors, localized as close to the intake valves as it is possible. It depends on the possibility of adjusting the fuel injection moment to the current demand of the engine. Those fuel injectors are not technically novelties, so their availability and knowledge of their construction is common.

Fig. 20.The examples of throat constructions from gas mixers for different engines (reproduced from [10])

The problem in further development of RCCI engines and in general of engines operating in DF mode are direct injection systems covering injection of both used fuels.

The use of two independent injectors in the cylinder head is possible, but only in sufficiently large engines. This solution allows for direct injection of two fuels, but is technically more demanding and expensive. In maritime engines exist the system of two independent injectors of diesel fuel and liquefied natural gas (LNG) in pure form [11].

The only existing mass-produced solution in small-sized engines that allows the injection of two different fuels with one injector is the dual-fuel injector technology developed by Westport. Those injectors are already available in two generations. Injector that works in high pressure direct injection (HPDI) system includes injection of both fuels at big pressure at similar time in the same engine operation moment. The scheme of that kind of injector work is shown in the Fig. 21.

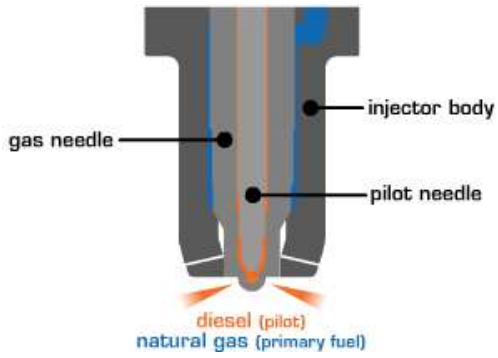

Fig. 21. Scheme of Westport company HPDI Injector Tip Assembly (reproduced from [33])

Product proposed by Westport corporation [27], and described in all relevant publications covering dual-fuel engine solutions for car and trucks [16] is only one factory-made injector available in the global market. It intended use is related to IC engines installed in heavy truck vehicles

Apart from the invention of Westport Company – and marine engines technologies with two independent injectors – DI of two fuels in small engines for passenger cars can be realized only in laboratory conditions for research purposes. In one study research team made their own handcrafted injector created for DI of CNG fuel in DF engine. Figure 19 shows the cross-section of the cylinder head with gaseous fuel injector and connection block, which is used for gaseous fuel supply for that concept DI gas injector.

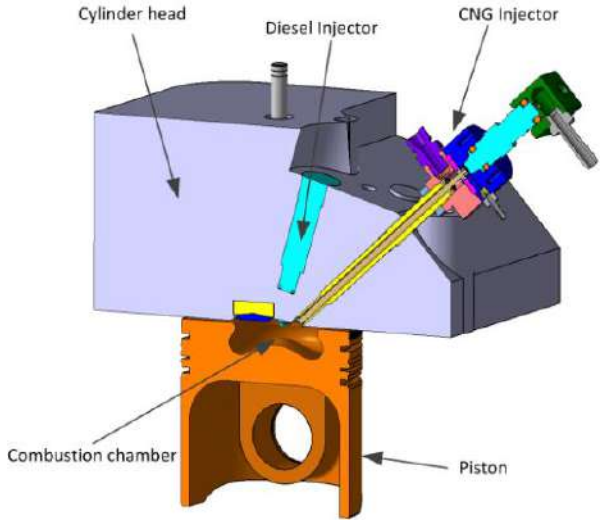

Fig. 22. The localization of the injector in CAD drawing of the cylinder head on the left and a custom gaseous-fuel injector with a parts of it on the right (reproduced from [14])

In the free market for car engines, it is hard to buy gaseous fuel direct injector, and basically impossible to buy Dual-Fuel injector – designed to inject two types of fuels simultaneously. This illustrates the early stage of development of this type of technique and its relatively low prevalence at the market. It should be borne in mind, that only DI of both fuels allows for realizing advanced fuel delivery strategies and paves the way for the development of advanced ignition techniques in IC engines, for which RCCI technique is a stop on the way to their further development.

Different group of researchers make the objective of their study to assess the potential of RCCI with direct injected low reactivity fuel in terms of thermal efficiency and methane emissions introducing gaseous fuel stratification by direct injection of it is considered to be beneficial for combustion efficiency increase [18]. Their simulation results, explanations for the observed trends are provided, and important phenomena are identified that are associated with increased low reactivity fuel stratification, which contributed to the reduction of methane emissions and an increase in NO_x emissions. Authors noted the lack of a significant amount of research on the RCCI direct injection engine. Authors of [18] noted the lack of a significant amount of research on the RCCI direct injection engine: "the improvement potential of such natural gas stratification in RCCI engines is unknown" which authors of present article can also confirm to some extent, with a few exceptions, like [21, 36] where authors introduced term DDFS, but their research is so far focused on a limited number of engines and fuels used, so that topic is still under development for the researchers

A different approach for fuel delivery can be found in the case of two-stroke engines, where fuel can be inject into the cylinder without using the injector located in the engine's head. Figure 23 shows the injection system in two stroke marine engine, where gaseous fuel is injecting direct into the cylinder when the piston starts to move upwards, to its top position. Injected gas have got much time to create air-fuel mixture, and at the top piston position secondary high reactivity fuel is injecting, starting the combustion process. This type of RCCI engine fuel supply system is only available in marine two-stroke engines.

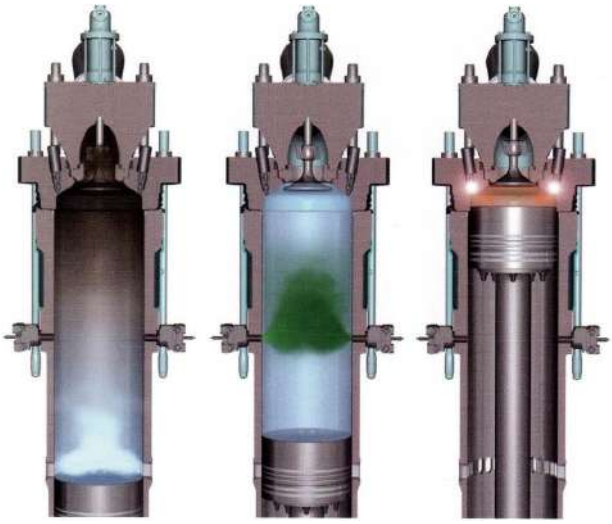

Fig. 23. Fuel injection process in two-stroke maritime engines, where the principle for the injection of low pressure gas through the cylinder liner during the upwards movement of the piston is clearly visible (reproduced from [9])

In that kind of marine engines, VVA is not a perspective technology because of two stroke engine specific construction, but researchers strongly recommend to increase knowledge and development of that technology [20]. Their experience with that technic shows that VVA makes it possible to improve combustion efficiency and thermal management (better after-treatment efficiency) and extending high load engine range. Maintaining a homogeneous mode of operation in the engine requires maintaining specific conditions. Researchers in their work create simulation concept for VVA in RCCI engine with a full load range and for two types of engine fueling: CNG-diesel and gasoline-diesel.

RCCI engines are commonly used in powerplants and marine applications, so their technology is more elaborate than in small engines. Researchers also found such potential applications for these engines, and their work show the variety of research that should be carried out on engines of this type, laying the foundations for their further work based on the stratification of low reactivity fuel and VVA [17]. The development of DF engines can be traced on the basis of gaseous engines in Wärtsilä company is shown in the Fig. 24.

It is clearly visible increase in the share of engines based on dual-fuel technology. Not all of those engines listed in that figure can work in dual-fuel mode – because a few of them was only gaseous fuel – but each of them was another important step in the development of RCCI engines.

However, development issues of small-size RCCI engines also includes external equipment, such as the engine control unit, which are currently available from only one manufacturer [3]. The difference between RCCI engine development in those different engine sizes are clearly visible and still require much more research work, especially for smaller combustion engines

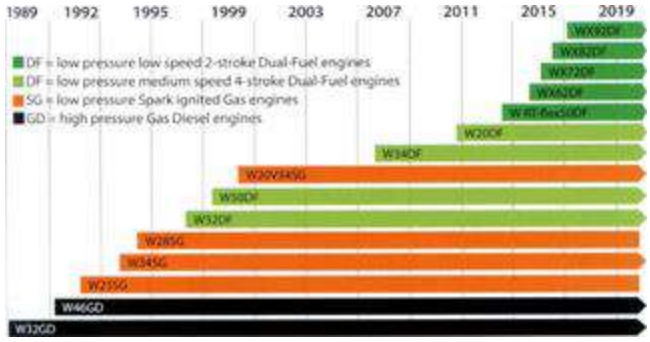

Fig. 24. Example of gaseous fuel engines development in Wärtsilä company between 1989 to 2015+ is the extensive portfolio of Wärtsilä gas- and dual-fuel engines for both two-stroke and four-stroke engine (reproduced from [9])

The last but extremely important technical challenge facing the researchers is to control the entire engine systems so it can work in the RCCI mode throughout the entire range of engine performance. No solution in this matter can be considered ready if it has not been tested in a working engine, which has so far been addressed by a small number of researchers and the publication [28]. There are not many such publications at that moment, but they are the most desirable type of publications summarizing the achievements in the field of all techniques used in RCCI engines. The correct electronic control of the engine operation will be necessary for the RCCI mode to be effectively used in the future of IC engines.

3. Conclusions

The intensive development of injection technology for both independent fuels is still needed in the context of current solutions of RCCI engines. Low reactivity fuel stratification could be the main point for future development of these engines, unfortunately there are few publications regarding DDFS topic.

There are also problems with the correct air-fuel swirl ratio and the injection angle of highly reactive fuel. It is necessary to adapt the injection shape by adapting it to the new geometry of the combustion chamber. There is some research on the subject, but much remains to be done in this subject (most studies are only computer simulations or theoretical considerations).

The combustion chamber geometry is not yet optimized for all engines where there is a need for the DF RCCI operating mode.

The most important challenge with low reactive fuel injection is the need to create a homogeneous mixture under all engine operating conditions.

In the case of highly reactive fuel injectors, the new challenge is to spray the fuel over the entire space of the combustion chamber. The RCCI combustion chamber geometry may be substantially the entire space above the piston in the cylinder space. Injection angles and injector efficiency differ significantly from that of a classic diesel injector. Basic design assumptions should be changed.

It is worth mentioning that in the future the engine industry will have to make an effort to put these types of engines into production. All the innovative technical solutions described here have only a conceptual character or were made in a single made test engine unit. The problems of mass production of new constructions are still ahead of us.

New technical solutions, especially those that are only computer simulations, definitely require further research and tests covering the use of various types of fuels, materials and test conditions.

Researchers are also faced with the challenge of combining all the technical solutions mentioned in the article in one engine and controlling its operation in a way that enables continuous operation in the RCCI mode. This challenge may be the most demanding, but its implementation requires even more detailed research into the technical solutions themselves. Especially when we are talking about small motors that are not used in industry.

Acknowledgements

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Nomenclature

CI compression ignition
CNG compressed natural gas
DDFS direct dual fuel stratification
DI direct injection
GDI gasoline direct injection
GDCI gasoline direct-injection compression ignition
GM general motors
IC internal combustion
HCCI homogeneous charge compression ignition
HPDI high pressure direct injection
LNG liquefied natural gas
LPG liquefied petroleum gas
LTC low temperature combustion
NVO negative valve overlap
PCCI Premixed Chargé Controlled Ignition
PFI port fuel injection
PPC Partially premixed combustion
PPCI partially premixed compression ignition
RCCI reactivity-controlled compression ignition
SACI spark assist compression ignition
SCR selective catalytic reduction
SI spark ignition
SPCCI spark plug controlled compression ignition
SPI single point injection
TDC top dead center
VVA variable valve actuation

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energies

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Article

Experimental Assessment of the Impact of Replacing Diesel Fuel with CNG on the Concentration of Harmful Substances in Exhaust Gases in a Dual Fuel Diesel Engine

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Abstract: The problem of global warming and related climate change, as well as rising oil prices, is driving the implementation of ideas that not only reduce the consumption of liquid fuels, but also reduce greenhouse gas emissions. One of them is the use of natural gas as an energy source. It is a hydrocarbon fuel with properties allowing the reduction of CO_2 emissions during its combustion. Therefore, solutions are being implemented that allow natural gas to be supplied to means of transport, which are trucks of various categories and purposes. This article presents the results of tests of an engine from a used semi-truck, to which an innovative compressed natural gas (CNG) supply system was installed. This installation (both hardware and software), depending on the engine operating conditions, enables mass replacement by natural gas (up to 90%) of the basic fuel—diesel oil. During the tests, on the basis of the obtained results, the influence of the diesel fuel/CNG exchange ratio under various engine operating conditions on the concentration of toxic CO_2, CO, NO, NO_2, CH_4, C_2H_6, NMHC, NH_3 and exhaust smoke was assessed. The test results confirm that, compared to conventional fueling, the diesel/CNG-fueled engine allows for a significant reduction in CO_2 concentration even in a car operated for several years with diesel fuel and with high mileage. The use of a non-factory installation significantly increased the concentration of methane CH_4, nitrogen dioxide NO_2 and carbon monoxide CO in the exhaust gas. It was found that the smoke content and the temperature of exhaust gases did not decrease with increasing ratio of fuel replacement. The concentration of CO, NO_X, CH_4 and NMHC was increased, while the concentration of CO_2, C_2H_6, NH_3 and the consumption of diesel fuel by the engine, decreased significantly. The innovation of the research is based on the use of a modern and unique engine gas fuel system control system where the original fuel supply system with unit pumps is able to reduce diesel oil consumption by up to 90%.

Keywords: dual fuel; diesel-CNG; CNG

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1. Introduction

In a dual fuel engine, fuel with a high octane number, which in the case of our tests was natural gas (NG), is fed into the combustion chamber of the engine first, along with the air supplied to the cylinder. After reaching a sufficiently high pressure and temperature, resulting from the compression of the gas-air mixture, a second fuel is injected into the combustion chamber which has good self-ignition properties allowing it to initiate self-ignition of the injected fuel. In the presented research, the high cetane number autoignition initiating fuel was classic diesel fuel commonly used in single-fuel diesel engines. Such a combination is the most popular way of powering "dual-fuel" compression ignition (CI) engines due to the possibility of achieving a very high degree of replacement of diesel fuel by natural gas, the use of which is supported by the cost and ecological aspects and the uncomplicated modification of the engine installation [1].

The degree of substitution in dual-fuel engines is defined as the ratio of the amount of high-octane fuel consumed to the total amount of fuel consumed by the engine. The diesel/CNG compressed natural gas power supply system used in the research determined the degree of mass replacement in this way. The ratio is expressed as the ratio of the amount of energy supplied to the engine in the form of high-octane fuel, in our case, CNG, to the total amount of energy contained in the fuel supplied to the single-fueled engine. In simple terms, it is also assumed to be the ratio of the amount of energy supplied to the engine in the form of high-octane fuel to the total amount of energy that is supplied at a given point of operation in the form of two fuels—the results in the case of both methods of calculation will be identical when the efficiency of the engine in both cases (mono-fuel and dual fuel power) does not change. This relationship can easily be calculated by relying only on the hourly fuel consumption based on the change in the consumption of diesel (or other high-cetane fuel) alone at the operating point where the substitution is measured and calculated. This can be done on the basis of Formula 1 below, without knowing the calorific value of the fuels used [2]:

$$\frac{diesel}{CNG} = \frac{Ge_{diesel} - Ge_{diesel_CNG}}{Ge_{diesel}} * 100\% = 1 - \frac{Ge_{diesel_CNG}}{Ge_{diesel}} * 100\% \quad (1)$$

Ge_{diesel}—hourly consumption of diesel fuel when running only on diesel oil,
$Ge_{diesel\ on\ CNG}$—hourly diesel fuel consumption when running on CNG/diesel.

In the case of the controller that was used in our research, the degree of replacement was calculated from a relation based on the mass fuel consumption of the engine, in which the calorific ratio of one fuel to another was expressed by the coefficient set by the operator at the beginning of the tests. The value of the adopted ratio was diesel/CNG 1:1.1 [2].

The degree of replacement of diesel fuel by natural gas (CNG) in a dual fuel engine depends on a number of factors, but the most important limitation that does not allow a high degree of replacement of diesel fuel by substitute fuel is the appearance of knock combustion and the expenditure of the CNG power supply installation. On the basis of research by many authors [3–5], an approximate diagram based on [6] has been developed, and is shown in Figure 1 schematically showing the possibilities of substitution of diesel by natural gas in dual fuel engines according to [7].

As part of the dual fuel supply, various fuels can be used, but the greatest number of advantages is characterized by compressed or liquefied natural gas. It is possible to co-burn diesel fuel in diesel autoignition engines together with a mixture of propane and butane gases, high-octane biofuels, or with gasoline. However, at the current level of technology in widespread use, such installations only achieve a partial, relatively small replacement of diesel fuel by these fuels. Diesel oil in classic installations of this type constitutes not less than half of the energy value of the fuel supplied for combustion. Depending on the source of information, it is possible to achieve a different replacement ratio with liquefied petroleum gas (LPG) fuel, but most often it does not exceed 30% [8]. However, in the case of using natural gas combined with diesel oil, which is the main research goal of this work, it already allows for a significant reduction in the operating costs of the engine and ecological improvement. The exchange ratio of diesel with natural gas is usually much higher than in the case of gasoline, biofuels, LPG or alcohols, mainly due to the much higher resistance of methane to knock combustion and its higher calorific value [9]. The same applies to the use of natural gas in liquid form, which has some additional advantages over its use in compressed form. These advantages relate to a large extent to the operation of the vehicle in which the installation of this type was placed, but also the possibility of direct injection of this fuel straight into the combustion chamber under high pressure. In the case of the gas combustion process, it may be more efficient if an appropriate fuel installation is used in which high gas pressure [10] or the physical state in which it is located [11] is used. The first installations of this type already exist in the world and are used both in motor vehicles [12] and in marine engines [13]. Although they are not yet widely available solutions, intensive work is being carried out on their dissemination and further development. The biggest

advantage of dual fuel engines is the combination of many advantages of CI and SI engines. There are a few ways to supply fuel to those engines and the two most important differ in the place of delivery of gaseous fuel [14]. The specificity of dual fuel power supply with direct injection of both fuels by one injector is shown in the schematic drawing of the injector in Figure 2.

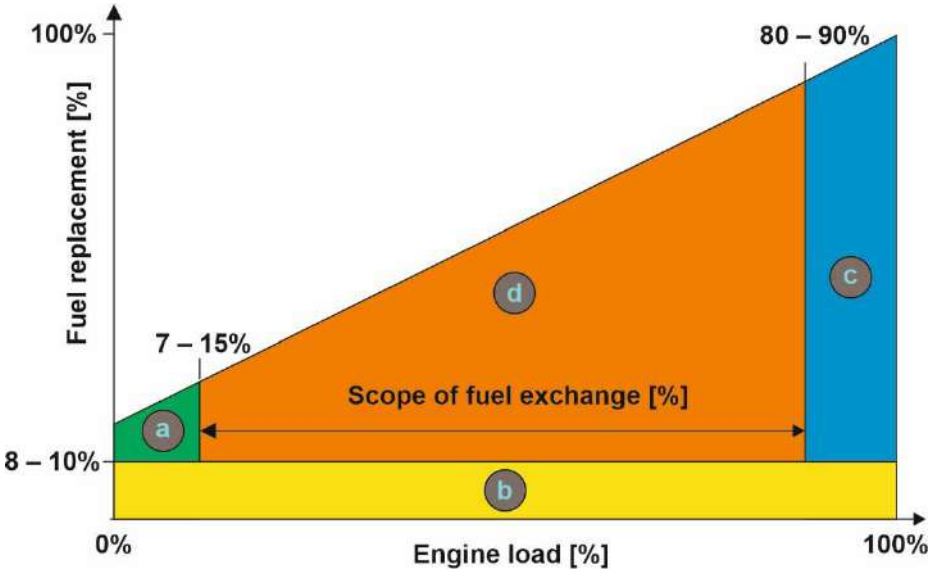

Figure 1. The various operating conditions known as "modes" for dual fuel engines, where horizontal is the engine load between 0–100%, and vertical is the rate of fuel replacement from 0% to 100%. The individual areas above result from the characteristics of the engine, the diesel fuel system and the CNG fueling system. Area (**a**) results from the properties of the CNG supply system—e.g., minimum CNG injector opening time, (**b**) minimum fuel dose necessary for proper ignition of the air-fuel mixture (its minimum value is also limited by the minimum opening time of the unit injector), (**c**) maximum load engine (**d**) range of proper dual fuel engine operation on diesel/CNG fuels. Authors source based on the chart from [6].

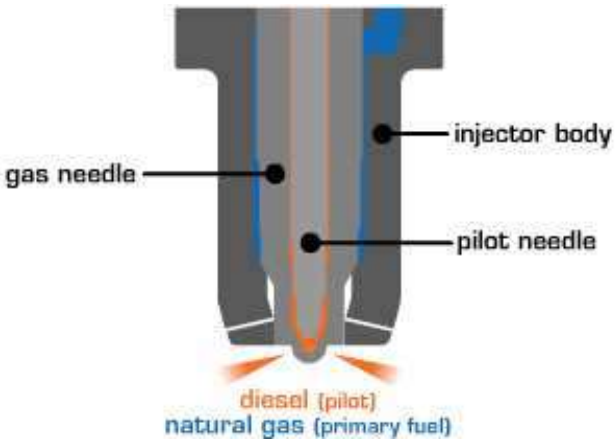

Figure 2. Schematic drawing of a direct dual fuel injector in use at the moment of diesel fuel injection [15].

This type of injector is developed especially for dual fuel engines working in diesel/CNG mode. This type of fuel supply is still very rare. A much more popular type of fuel supply is indirect injection of CNG and direct injection of diesel fuel. That was the type of supply researchers used in their project. A schematic drawing of an engine at work with this type of fueling is shown in Figure 3.

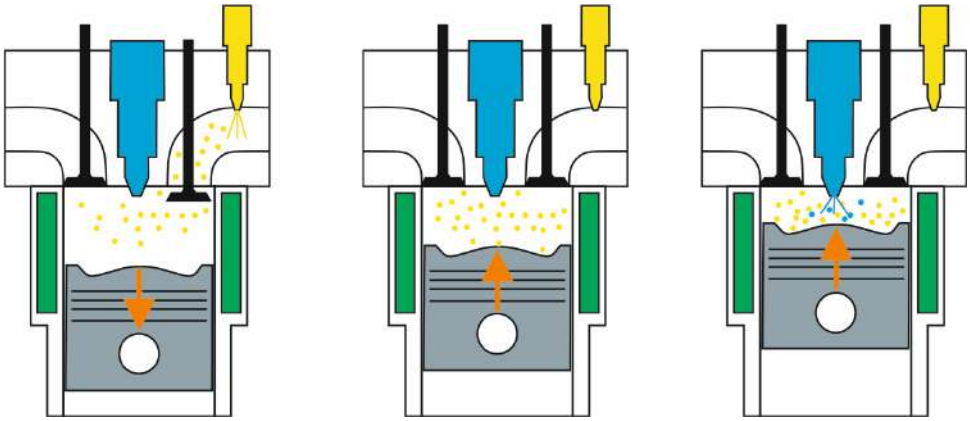

Figure 3. Schematic drawing of fueling of a working dual fuel engine with indirect injection of CNG fuel (yellow drops) and direct injection of diesel fuel (blue drops) [Authors' source].

The type of fueling shown in the Figure 3 is much more common, simpler and well-researched, and is a much more popular way to make a dual fuel engine from a diesel CI engine. In dual fuel engines, during operation the injected liquid fuel is dosed into the combustion chamber in the form of a pilot dose, initiating the ignition of the fuel already supplied to the combustion chamber under the effect of a steady increase in pressure resulting from self-ignition of the injected high-cetane fuel. Figure 4 below shows a schematic view of the method of supply and ignition in a dual fuel engine powered by a mixture of CNG and diesel oil.

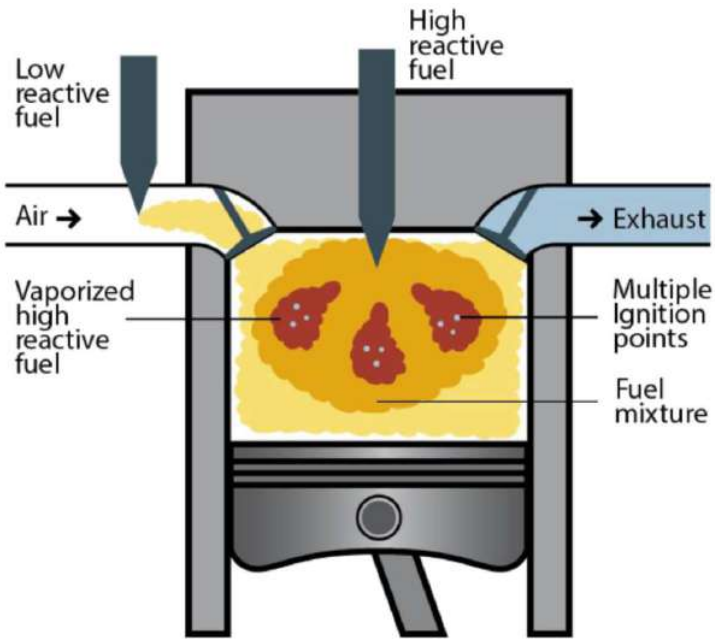

Figure 4. Autoignition and combustion in a Dual Fuel CI engine, own artwork based on [16].

An engine fueled by dual fuel CNG and diesel oil requires, for standard work mode, only the already mentioned, pilot dose of diesel fuel to initiate the ignition. When using LPG as an additional fuel, it is necessary to use a greater amount of injected diesel fuel than in the case of natural gas, due to the insufficient resistance of the additional gaseous fuel to knocking combustion, which is dangerous due to the possibility of engine damage. Its presence determines the possibility of using fuel in the engine and the degree of replacement of the basic fuel, which is diesel fuel in a dual fuel CI engine. Highly loaded dual fuel engines operate under conditions that make it impossible to replace as much chemical energy contained in high-cetane number fuel, as is possible at low loads There is a lower

ratio of replacement at such operating points due to the need to use more high-cetane number fuel with a higher energy density than gaseous fuels.

The sense of using natural gas in CI engines results not only from the physicochemical properties allowing its effective use in such engines, but also the ecological and economic impact that this fuel could have on the industry based on the traditional piston internal combustion (IC) engine. Along with the development of civilization and the increase in consumerism, the demand for the transport of goods using wheeled vehicles will increase, especially in increasingly crowded cities where people have to deal with the negative health effects caused by pollution from the growing number of trucks [17,18]. It is estimated that in Poland alone, about 50,000 people die every year due to poor air quality, or even twice as many taking into account the percentage of deaths in Europe caused by exposure to fine particulate matter (PM 1.0 and PM 2.5) resulting from the combustion of fossil fuels for populations aged >14 years [19,20]. The use of a low-emission alternative fuel instead of the coal and diesel fuel commonly used in Poland could contribute to avoiding these deaths, which is why research is currently placing so much emphasis on the use of natural gas to power thermal machines.

Public transport can offer a more rational use of energy [21] as it reduces fuel consumption, exhaust emissions and vehicle noise [22–24]. The concept of sustainable transport combines high-quality transport with care for the natural environment [25]. Therefore, these vehicles should have a power source other than diesel or gasoline and reducing carbon dioxide (CO_2) emissions can help limit the development of global warming in the world.

An example of such activities may be the increasing restrictiveness of requirements for products created during the combustion of a fuel mixture. Their implementation means that, for example, even relatively new semi-trucks (5 years old) are classified as polluting and their use is charged with additional fees. This forces vehicle manufacturers to look for technical solutions that reduce the emission of harmful exhaust components. The obvious path of development seems to be the construction of trucks with hybrid or only electric drive. However, in this case, the range remains a limitation, which for an electric car of average payload ("N2") is currently about 200 km, while for a tractor unit with a semi-trailer ("N3") it is, according to available data, only 100 km [26]. In addition, the price of an electric tractor unit in 2021 was five times the price of the same conventionally powered tractor unit (based on the example of DAF and Volvo cars). In addition, increasing the weight of the semi-truck as a result of the usage of batteries that are still very heavy, reduces the weight of the load that can be transported, and the charging process is time-consuming. Both of these factors have a very negative impact on the logistical costs of using such a vehicle [27]. Therefore, a good direction of change is still the introduction of more environmentally friendly solutions into transport, mainly through the use of alternative and ecological fuels in relation to diesel, including natural gas in various forms: compressed natural gas and Liquid Natural Gas (LNG) and its ecologically produced varieties Renewable Natural Gas—RNG, BioCNG, or eCNG and natural gas enriched with hydrogen fuel—HCNG, or H2CNG. Many fuels can be used as high-cetane fuel, the properties of which do not differ significantly from those of diesel oil. Very promising are the possibilities of using, together with ecological gaseous fuels, also ecological, alternative fuels replacing diesel oil in this case, such as oil made of algae [28], rice [29], pongamia, palm, peanuts, mushrooms, jatrophy, coffee grounds, soybean, coconut, linseed, olives or even animal fat, which the authors described in detail in [30]. Also, a gas used as a fuel corresponding to the properties of natural gas can be produced in many ways and has great potential as a clean alternative fuel for additional reduction of GHG emissions in the process of its combustion or production [29]. Such a gas can be produced under the same conditions as bio-oil from many plant and animal products [30] or waste [31] and individual final products will differ mainly in the concentration of methane and other components of this fuel.

Combustion of natural gas, due to its different composition compared to diesel oil, with a lower proportion of carbon and a greater proportion of hydrogen, results in lower

emissions of carbon dioxide, which is one of the main greenhouse gases emitted by human activity. In the short and medium term, natural gas may become the most important alternative fuel [32]. Its large global resources increase the potential of this fuel as a green fuel used in transport. Many European countries have a well-developed infrastructure adapted for the reception and distribution of CNG and LNG. Natural gas can also be a relatively cheap fuel compared to diesel oil. In 2020, the average price of natural gas accounted for approx. 50% of the diesel oil price. The average cost of using natural gas to transport heavy goods vehicles is approx. 40% lower than that of diesel fuel [33].

The properties of natural gas (in particular in the form of compressed CNG) in terms of the use of IC piston engines in transport or in utility vehicles, as well as the growing number of supply sources in individual European countries justify undertaking scientific, research and development works on the use of CNG to supply the IC engines of heavy-duty vehicles. This is in line with the energy policy of the European Union—Directive 2009/33/EC of 23 April 2009 on the promotion of ecologically clean and energy-efficient road transport vehicles. The presented work is an example of such activities and describes the effects of the application of an innovative concept of effective adaptation of an engine of a utility vehicle to a CNG supply.

2. Materials and Methods

The aim of the study was to experimentally determine the impact of the exchange ratio (diesel/CNG), defined by Equation (1), of the replacement fuel, which was CNG, to the base fuel, which was diesel oil, for an innovative non-factory two-fuel CNG/diesel fuel supply installation in a used tractor unit engine on the emissions of individual exhaust components: carbon dioxide (CO_2), carbon monoxide (CO), nitric oxide (NO), nitrogen dioxide (NO_2), methane ($CH_{4)}$, ethane ($C_2H_{6)}$, non-methane hydrocarbons(NMHC), ammonia ($NH_{3)}$ and smoke in fixed operating states.

"The tests and calculations were carried out for the D13C460 EU5EEV (338 kW) engine, used in Volvo FH13 series tractor units. The mileage of the car, the unit of which was tested on an engine dynamometer and on which a non-factory diesel/CNG installation was adapted, was 790,500 km. The total operating time of the engine is 11,800 h, and the engine has used 229,100 L of diesel fuel since the beginning of its operation. The tested engine, according to the approval documents, met the requirements of the EURO V standard when it left the factory" wrote the authors in [33]. As a part of our research, the engine operation points were determined on the basis of good engineering judgment and as equivalents of the operating points in which ISO 8178 [34] tests are carried out. In many cases found in the literature, the research program is prepared with the use of certain algorithms [28,31,35] however, the authors' experience shows that this type of methodology is not fully effective in the areas in which the authors of this work operate. The research presented in the paper is an investigation of an innovative, high-pressure dual fFuel installation. The planned tests were divided into the following stages:

- Determination of external and load characteristics of the D13C460 engine when supplying the basic fuel, which was diesel fuel, in order to assess its technical condition.
- Installing a dual fuel supply system on the engine.
- Carrying out research into the impact of the diesel/CNG replacement ratio in the range from 10% to the knocking limit at selected engine operating points—the condition for "tuning the CNG supply system" was to obtain the same useful power during supply (diesel/CNG) as during supply of the basic fuel, which was diesel oil.
- On the basis of the obtained test results, the impact of the replacement factor (diesel/CNG) on the concentration of toxic exhaust gas components at selected test points was assessed.

The tests were carried out on a standard dynamometric stand. The engine was loaded with a Zöllner PS1-3812/AE water brake with a maximum braking power of 1250 kW. The torque generated by the motor was measured using a strain gauge. The rotational speed was measured by an impulse transducer in cooperation with a gear which was

located on the collar of the dynamometer. Measurement of liquid fuel consumption was carried out with the use of a fuel scale by AVL, measuring at a frequency of 5 s, and the temperature of the cooling liquid was maintained in the range of 87–92 °C using an external heat exchanger. The smoke opacity was tested with an AVL 439 OPACIMETER, which operated on the principle of light absorption—measuring the extinction coefficient of the absorbed radiation. The composition of the exhaust gases was measured with the use of an AtmosFIRt analyzer, working on a hot sample of 180 °C, using the method of infrared spectroscopy using the Fourier transform (FTIR) method with an oxygen analyzer installed in the measuring chamber using the zirconium cell method. In the configuration used, the device allowed the measurement of the following components: CO_2, CO, NO, NO_2, CH_4, C_2H_6, NMHC and NH_3. The measurement results were automatically converted to normal conditions (101,325 Pa, 273.15 K = 0 °C). The spectral range of the analyzer was 485–8500 cm^{-1}. The ambient conditions were determined using the ABB SensyMaster FMT430 air flow meter, which was also used to determine the amount of air sucked in by the engine.

The temperature of the exhaust gases was measured using a NiCr-NiAl thermocouple installed in the exhaust manifold with a measuring range from −200 °C to 1200 °C. Selected engine parameters, measured by standard sensors mounted on the engine, were read from the engine controller using a Texa Navigator diagnoscope. These were defined as:

- boost pressure,
- coolant temperature,
- engine oil temperature,
- air temperature in the intake manifold.

The tests were carried out while keeping the thermal state of the engine approximately constant, controlled with the use of an air heat exchanger (cooling tower) and a cooling liquid circulation system. During the tests, the temperature of the cooling liquid and engine oil was within 81–96 °C.

Measurements of individual quantities were carried out under the specified engine operating conditions when the force on the brake and the engine rotational speed were constant for at least 1 min. During the measurements, a constant engine speed was automatically maintained (range n = constant water brake operation).

Measurements of fuel consumption (from which the value of hourly fuel consumption was obtained) were carried out for each measuring point during approx. 1 min. Measurement of other quantities (brake force, ambient temperature, ambient pressure, ambient relative humidity, exhaust temperature and smoke (light absorption coefficient), air consumption by the engine and exhaust gas analysis was carried out simultaneously after establishing the test conditions.

The following parameters of the engine operation and the engine's intake system were measured directly:

- engine torque, Mo [N·m]
- engine speed, n [rpm],
- hourly fuel consumption diesel oil, GeON [kg/h],
- hourly fuel consumption CNG, GeCNG [kg/h],
- hourly air consumption, Gp [kg/h],
- temperature in individual engine systems T [°C],
- smoke D [m^{-1}],
- Concentration of gaseous components in exhaust gases: CO_2, CO, NO, NO_2, CH_4, C_2H_6, NMHC and NH_3,

On the basis of the quantities measured in a direct way, the quantities measured indirectly were determined:

- Knock combustion signal, [–],

On the basis of the directly measured quantities, the values measured indirectly were determined:

- actual diesel/CNG exchange rate

The value of the useful power of the engine was determined on the basis of the value of the loading torque realized by the brake and was determined from the dependencies:

$$N_e = \frac{nM_o}{9550} K_{d1} \text{ [kW]} \tag{2}$$

where:

n is rotational speed in rpm,

Ne is engine useful power in kW,

Mo is engine loading torque (rotational) read from the brake controller in Nm,

K_{d1} is correction factor for normal conditions determined in accordance with the PN-ISO 15550: 2009 standard.

Determination of the engine power value and knowledge of the concentration of harmful exhaust components allowed us to determine the emissions of a given exhaust component at a specific point of engine operation. For all results, the uncertainties of the measurement results were calculated, taking into account the possibility of both systematic and random errors. Systematic errors of direct measurements were determined on the basis of the accuracy class of the measuring instruments or on the basis of an elementary division on a scale. Errors of the value determined indirectly (from mathematical dependencies) were calculated after determining the size of systematic error Δx_i (where i = 1, 2, 3, . . . , k) values of measurements of quantities measured directly as the maximum error Δy from the general dependence in the form [36]:

$$y = \sum_{i=1}^{k} \left[\frac{f(x_1, x_2, \ldots, x_3)}{x_1} \right] x_i \tag{1}$$

The values of systematic errors of the measured quantities are presented in Table 1.

Table 1. List of investigation equipment used during investigation.

No.	Name of Device/Measured Quantity	Type	Range	Accuracy
1.	Water dynamometer Torque—Mo, rotated speed—n	Customs officer PS1-3812/AE	Mo = (0 ÷ 7000) N·m, n = (0 ÷ 3000) rpm Ne = 0÷1250 kW	±1 N·m, ±1 rpm, ±1 kW,
2.	Fuel weight-meter (diesel)	AVL 733S Fuel Balance	(0 ÷ 200) kg/h	±0.005 kg/h
3.	Fuel weight-meter (CNG) 1.2–104.6 kg/h ± 0.6% measured quantity	SwirlMaster FSS450 Intelligent Swirl Flowmeter	1.2–104.6 kg/h	±0.6% measured quantity
4.	Exhaust analyser—measuring of toxic elements concentration in exhaust gases - carbon dioxide (CO_2), - carbon monoxide (CO), - nitrogen oxides (NO), - nitrogen diooxide (NO_2), - methane (CH_4) - ethane (C_2H_6) - ammonia (NH_3)	Atmosphere FIR, emissions monitoring FTIR systems	CO_2 (0.01 ÷ 23)% CO (1.0 ÷ 11,000) ppm, NO (1.0 ÷ 6000) ppm NO_2 (1.0 ÷ 6000) CH_4 (1 ÷ 5000) ppm C_2H_6 (1 ÷ 1000) ppm NH_3 (1 ÷ 5000) ppm)	±0.1% measured quantity
5.	Smoke concentration—extinction coefficient of light radiation—k.	AVL Opacimeter 4390	(0.001 ÷ 10.0) m^{-1}	±0.002 m^{-1}
6.	Thermocouple—measuring of exhaust temperature—T	NiCr—NiAl (K)	(−50 ÷ 1100) °C	±1 °C
7	Mass air consumption	SensyMaster FMT430 Thermal Mass Flowmeter	120 ÷ 7000 kg/h	±1.2 kg/h

Control of the engine load, as well as its rotational speed, was carried out from the control and measurement cabin. In this cabin there is a cabinet on which instruments are mounted to control the brake load and to control the servo. In this cabinet there are also engine speed indicators, as well as an indicator for measuring the temperature of individual systems.

During the tests, the operation of the engine was constantly controlled by the use of the NAVIGATOR TXTs diagnostic interface with the IDC 5 TRUCK software. A diagram of the test bench is shown on a Figure 5.

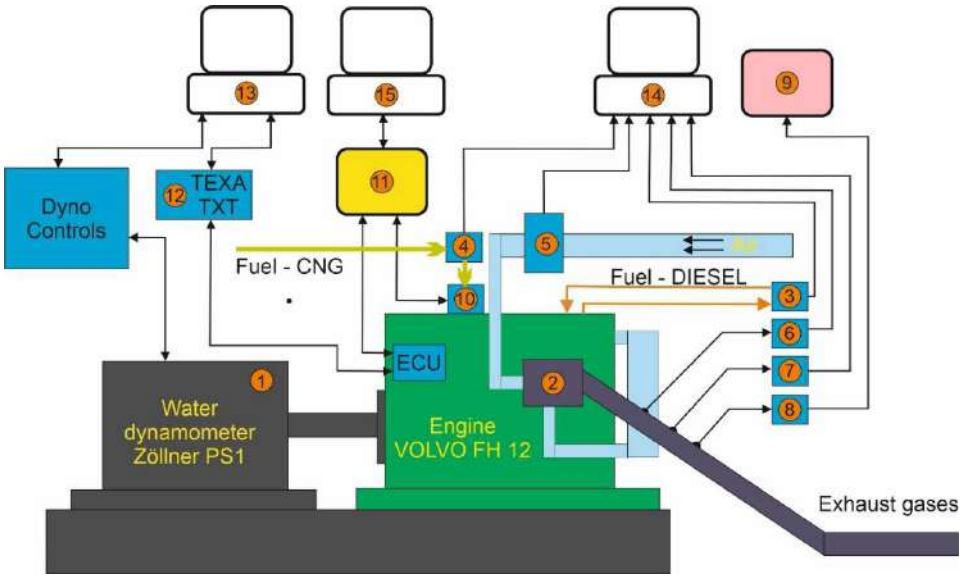

Figure 5. Diagram of the dynamometer with Volvo D13C460 EURO V engine: 1—water brake, 2—turbocharger, 3—basic diesel fuel consumption measurement system, 4—CNG gas consumption measurement system, 5—air consumption measurement system, 6—exhaust temperature measurement system, 7—smoke measurement system, 8—gas component consumption system for analysis, 9—FTIR analyzer, 10—CNG injection system for the engine intake manifold, 11—CNG power supply system with software, 12—TEXA TXT diagnoscope, 13—computer controlling the operation of the torque brake and recording engine operating parameters, 14—computer controlling the operation of measuring systems, 15—computer for programming the CNG power supply system controller.

Tests and results of useful measurements of the engine have been reduced to normal conditions in accordance with PN-ISO 15550:2009—Reciprocating IC engines—Determination and method of measuring engine power—General requirements [36], which recommends correcting the engine power according to ambient conditions by multiplying the value determined directly by the calculated correction factors.

Methodology for Determining the Effect of the CNG/ON Substitution Factor on Engine Performance

The beginning of the test was to determine the external characteristics and load characteristics of the D13C460 engine. The tests were carried out on a dynamometer station located in the Department of Engines and Operation Engineering of the Military University of Technology in Warsaw. For supplying the engine commercial diesel fuel was used. This diesel fuel was from one production batch and was used during all tests. As a result of these activities, measuring points were designated to carry out further work.

Then, an original gas supply system was installed on the engine, allowing us to bring two types of fuel to the combustion chambers—diesel and CNG. The installation consists of a gas supply rail mounted between the engine head and the intake manifold. Gas injectors were installed in the rail, 2 per cylinder and the working pressure of the CNG gas used was 0.8 MPa. The installation works on the principle of shortening the time of injection of the

basic fuel, which was diesel oil in proportion to the set coefficient of replacement of diesel fuel by natural gas.

In the course of adaptation activities, the installation was subjected to an optimization process, in which the main criterion was to determine the optimal angle of advancement of CNG gas fuel injection for different rotational speeds and engine loads relative to the position of the crankshaft at which the piston TDC (top dead center) occurs.

Subsequently, for such an optimized installation, studies were carried out on the impact of the coefficient of replacement of mass diesel fuel by natural gas on smoke and exhaust gas composition. The test was carried out in selected engine operating states corresponding to the most commonly used engine operating conditions when performing a road transport task. In the individual operating states selected by our team, the ratio of diesel to natural gas exchange was changed in leaps and bounds, with a jump of 10%, starting from 10% of replacement, up to the knock combustion. At each point of the engine's operation, the amount of gas supplied was corrected using the CNG controller in such a way that the power developed by the engine was the same as the engine power obtained when running on the basic fuel, which was diesel oil. The assumed conversion rate of diesel to natural gas was set using a computer controlling the operation of the CNG installation, and the obtained exchange rate was determined on the basis of changes in the hourly consumption of the basic fuel. The research was carried out until the exchange ratio was achieved at which point the signal level from the knock combustion measurement system exceeded the level of 2.

During the tests, commercial diesel from one production batch in accordance with en-590 was used, due to the fact that the normative document [36] allows the use of a fuel similar to the reference fuel in a situation where the appropriate reference fuel is not available. The advantage of choosing this method of proceeding is to determine the actual values of the engine's operating characteristics with fuel of a random composition (random petrol station). It was remembered to reduce the impact of fuel properties on the result of comparative tests as much as possible, so for all tests, diesel fuel from one delivery was used.

The measurements were made using the prepared research version of the CNG dosing controller in which the optimal individual control functions were determined and control maps were developed.

The results of the research were illustrated by comparing them in subsequent drawings to changes in the concentration of the components of individual gaseous components, i.e., smoke as a coefficient of extinction of absorbed radiation (Figure 10), CO and CO_2 (Figures 11 and 12), NO and NO_2 (Figures 13 and 14), and CH_4 (Figure 15). To facilitate the analysis process and to quantify the changes taking place, the results are also included in a relative way, comparing those obtained for diesel/CNG power supply with only On power supply (right side of drawings). The concentration of individual gaseous components and smoke opacity were determined directly at the exit of the engine, before the exhaust after-treatment systems, so as to eliminate the result of the exhaust gas neutralization systems. As part of the verification of the technical condition of the engine, the external characteristics of the Volvo D13C460 engine used in the Volvo FH13 car were measured, determined when it was powered by diesel, and presented and compared with the original characteristics as shown in Figure 6.

Then, on the basis of the analysis of the course of useful (effective) power and torque of this engine, research engine operating states were developed in which the impact of the degree of diesel/CNG replacement on the concentration of exhaust gas components was assessed. The designated external characteristics indicate that the engine develops a maximum torque of about 2300 Nm in the speed range of 1000–1500 rpm, and a maximum value of useful power of about 338 kW (460 hp) in the speed range of 1400–1900 rpm. Above an engine speed of 1500 rpm, the torque decreases sharply and reaches a value of about 1700 Nm at a speed of about 1900 rpm, above which the useful power of the engine decreases. Analysis of the course of useful power and torque and the operating conditions

of the vehicle from which the engine came showed that the range of economical, operational engine speed is approx. 1100–1500 rpm. In this range, the engine develops a constant (approximately) maximum torque, and at the same time increases (as the rotational speed increases) its useful power.

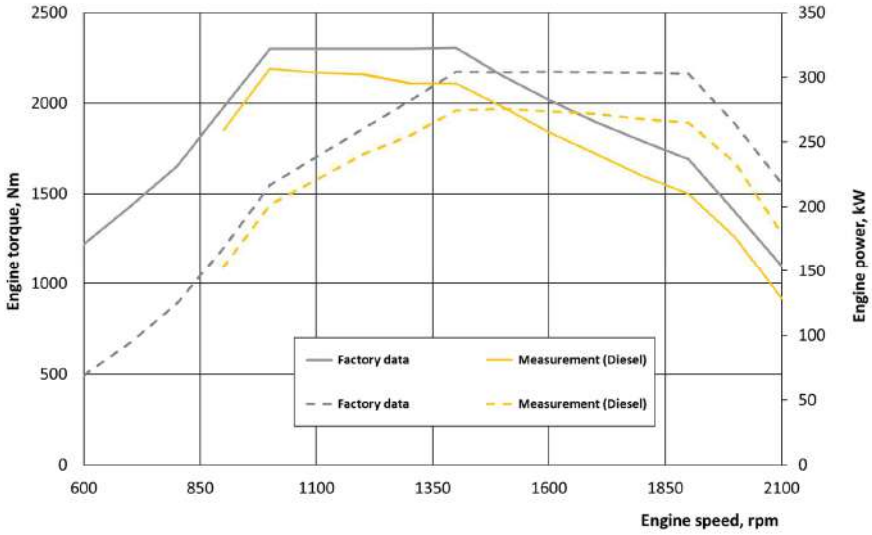

Figure 6. External characteristics of the VolvoD13C46 engine.

In order to influence the supply of the engine with different proportions of diesel/CNG to determine the optimal proportion of fuel supply with the CNG/diesel mixture (percentage replacement of diesel oil with compressed natural gas), a test program was developed taking into account the above-mentioned results of the analysis of the external characteristics of the engine powered by diesel fuel. The scope of research was limited to determining the effective parameters of the engine in the most economical operating speed range, 1100–1500 rpm. In this respect, the basic engine operating parameters were measured at the dynamometer for three speed values (1100 rpm, 1300 rpm, 1500 rpm) at a constant value of loading torque.

The tests were carried out:

- at a constant engine speed of 1100 rpm for two load torque values—approx. 520 Nm (61 kW) and 1150 Nm (139 kW);
- at a constant engine speed of 1300 rpm for two load torque values—approx. 480 Nm (67 kW) and 980 Nm (134 kW);
- at a constant approximately variable engine speed of 1500 rpm for two load torque values—approx. 500 Nm (79 kW) and 1040 Nm (163 kW).

For each engine speed and the value of (loading) torque, tests were performed for different proportions of the diesel/CNG fuel supply—from diesel only (replacement rate 0%) to the size of the replacement, at which, under the given measurement conditions, the occurrence of detonation combustion was recorded by means of a sensor attached to the engine block.

3. Results

The graphs showing the results of the research were divided according to the measured parameter. Each of the graphs presents the results for all six operating states in which the engine was tested. In each of the engine operating states, the engine speed and load were strictly determined, while the degree of replacement of diesel fuel with natural gas was variable. The test results are presented in graphs. The value of the degree of replacement of diesel by CNG is expressed as the real value of replacement calculated on the basis of fuel consumption, which is visible in the diagrams in which the marked points are usually near the assumed value of the replacement ratio.

Figures 7–18 show the effect of the diesel/CNG replacement factor for six engine operating states, differing in speed and load, on the engine operating parameters or the concentration of individual exhaust gas components.

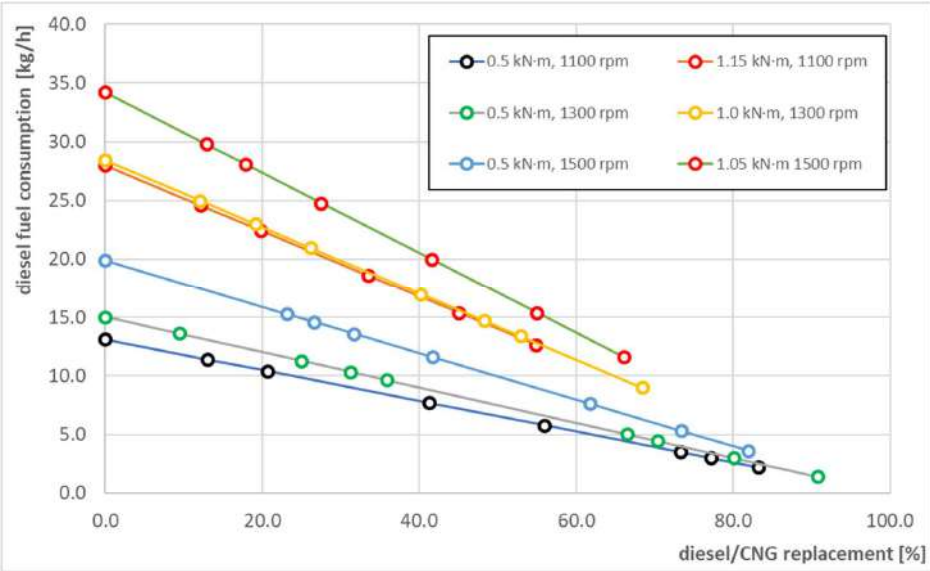

Figure 7. Hourly diesel fuel consumption for individual engine operating points (rotation speed, torque) in the diesel/CNG replacement function.

Figure 7 shows the effect of the diesel/CNG replacement factor for the six engine operating states, differing in speed and load, on the hourly consumption of the diesel base fuel. A linear decrease in hourly diesel fuel consumption is visible as a function of the replacement coefficient with Ge diesel = f(diesel/CNG) for the tested CNG power supply system of the Volvo D13C460 EURO V engine. The nature of the changes in fuel consumption are illustrated by the linear relationship between the consumption of the basic fuel with the designated replacement coefficient, which due to the compliance of the results with the theoretical assumptions indicates the high accuracy of the measurements made. A similar correlation can be observed for changes in hourly CNG consumption GeCNG = f(diesel/CNG) as shown in Figure 8.

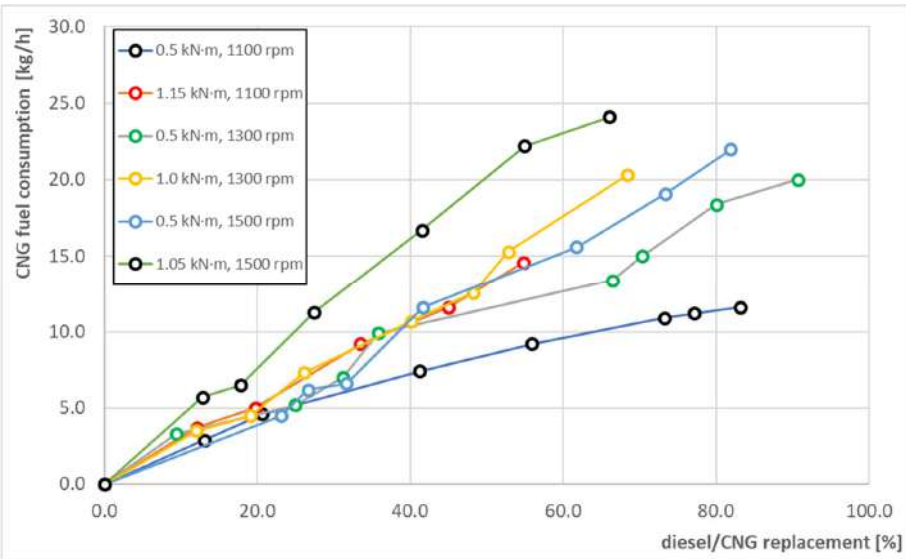

Figure 8. Hourly CNG fuel consumption for individual engine operating points (rotation speed, torque) in the diesel/CNG replacement function.

Differences in hourly CNG consumption for individual engine operating points are, of course, due to different values of power generated by the engine, but deviations from the linear nature of the graphs may have different backgrounds. A shape more parabolic than the linear graph showing the hourly fuel consumption may indicate, in the case of the tested engine loaded at 1100 rpm with a torque of ~0.5 kN·m, a systematic increase in the efficiency of the tested engine. The change in the efficiency with which the energy supplied to the engine was used also potentially had an impact on the results of fuel consumption in other measurements in which some nonlinear changes are clearly visible at individual stages of fuel exchange. The smallest increases in gaseous fuel consumption can be seen in the lower load points, where higher degrees of replacement have also been achieved than with a higher engine load. This is also correlated with the lower achieved temperature of the exhaust gases, which can be seen in the graphs shown in Figure 9.

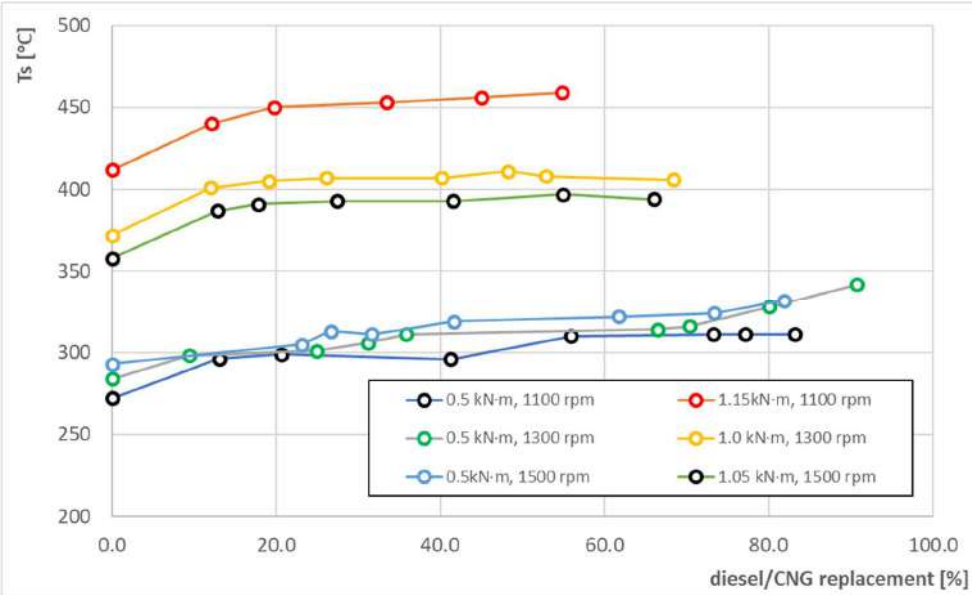

Figure 9. Exhaust gas temperature for individual engine operating points (rotation speed, torque) as a diesel/CNG replacement function.

Figure 9 shows the effect of the diesel/CNG replacement factor for six engine operating states, differing in speed and load, on the exhaust gas temperature at the turbocharger output Tsp = f(diesel/CNG). A clear increase in exhaust gas temperature is visible with an increase in the diesel/CNG replacement coefficient of about 50–60 °C. This phenomenon is caused by a change in the way the fuel is burned and with a decrease in the stoichiometric lambda. Natural gas, due to its density, occupies a much larger space in the combustion chamber than diesel fuel, which leads to a decrease in the amount of energy ballast that is mainly nitrogen, which leads to a noticeable increase in the temperature of exhaust gases.

In the case of our application, in which the geometry of the combustion chamber was not modified for a dual fuel power supply, and the mixture of natural gas and air was not homogeneous, some of the unburned natural gas molecules could burn only in the exhaust system, which in some cases may result in a significant increase in the temperature of the exhaust gases. This is a very unfavorable phenomenon due to the possibility of increasing the emission of nitrogen oxides and increasing the heat load of the engine. It should be emphasized that these changes are the strongest in the low range, where differences in higher loaded engine working states between the 1100 rpm and 1500 rpm of speed causing an increase of about 60 °C between this curves. For the exchange rates between 20% 70% and higher engine loads at medium speeds, an increase in engine speed above 1300 rpm causes a decrease in exhaust gas temperature to 300 °C and a maximum temperature in this engine operation state increase of about 50 °C. A higher exhaust gas temperature at

higher loads and a slightly lower one at higher speeds is natural due to the engine using more fuel at high loads and significantly higher flow through the engine at high speed, which is clearly visible in the diagram.

Seeing the values of the exhaust gas temperature achieved at higher engine loads, one can immediately see the association of these values with the achieved smoke in those engine operating states where these values (visible in Figure 10) are clearly higher than in states in which the engine was loaded with less braking torque. This figure shows the dependence of smoke on the degree of replacement in six defined operating states.

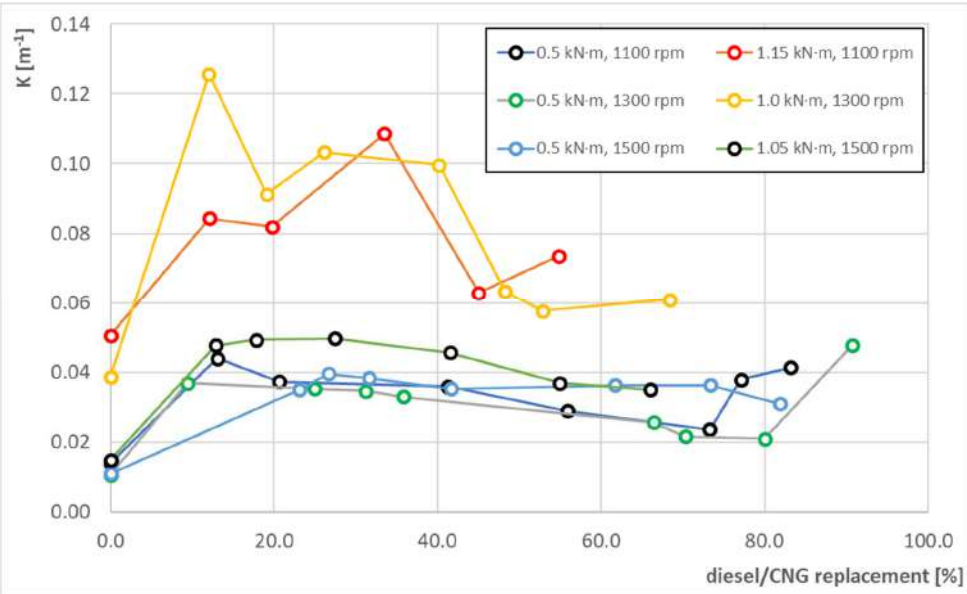

Figure 10. Smoke opacity for individual engine operating points (rotation speed, torque) in the diesel/CNG replacement function.

Figure 10 shows a clear increase in smoke with an increase in the diesel/CNG replacement ratio of about 200–300% at different points, depending on the operating conditions. Smoke opacity at low engine load regardless of replacement is slightly higher than for smoke obtained for an engine sown exclusively with diesel fuel. In the case of engine operation at a higher load, the smoke increases many times and at some degrees of replacement significantly deviates from the trend, which is a critical case, and which will be additionally verified during subsequent tests of this engine. Significantly higher concentrations recorded for two engine operation states may result from high engine load at lower rpm, but comparing the absolute values of concentrations for individual engine operation states is not advisable without checking the emissions at these operating points. For low loads, only at small and high replacement values, did smoke increase, which will require more in-depth diagnostics in the future.

Unlike smoke and exhaust gas temperature, the dependence of carbon dioxide concentration as a replacement function is laid. Theoretically, carbon dioxide emissions should decrease as the rate of replacement of diesel by natural gas increases. The diagram shown in Figure 11 shows a decrease in the value of carbon dioxide concentrations in the exhaust gas is visible and in most cases it takes on a close to linear character.

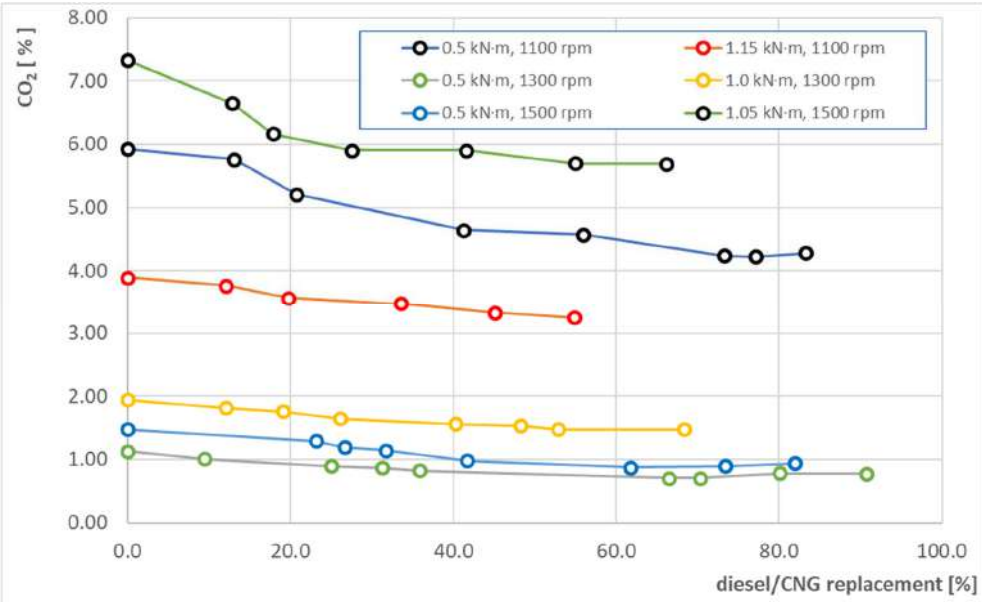

Figure 11. Concentration of carbon dioxide in the exhaust gas for individual engine operating points (rotation speed, torque) as a function of replacing diesel/CNG.

The strongest concentration drops occur at lower replacement values, then the level of carbon dioxide concentration in the exhaust gases begins to stabilize. There is a small visible decrease in carbon dioxide concentration with an increase in the diesel/CNG ratio, which reaches a maximum decrease of about 20–30% depending on the operating conditions. The reduction in the concentration of carbon dioxide is associated with a change in the elementary H/C ratio (hydrogen/carbon) as a result of replacing diesel fuel with a high carbon content, with a gaseous fuel, methane, which has a high hydrogen content. This phenomenon is accompanied by an increase in the water content of the exhaust gas [37].

The linear relationship is no longer visible in the diagrams in Figure 12 which shows the effect of the diesel/CNG replacement factor on the concentration of carbon monoxide in the exhaust gases.

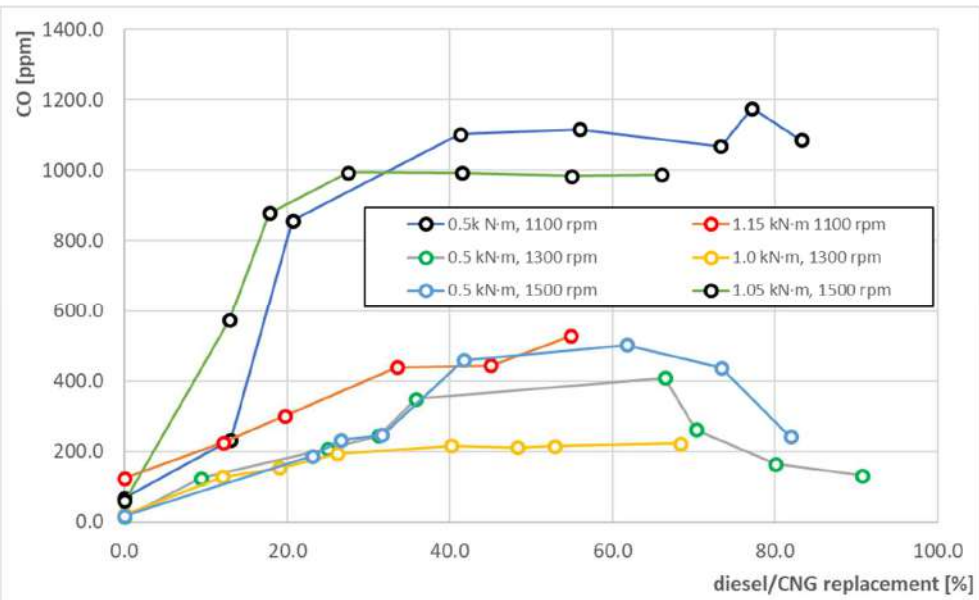

Figure 12. Concentration of carbon monoxide in the exhaust gas for individual engine operating points (rotation speed, torque) as a diesel/CNG replacement function.

Figure 12 shows a clear increase in the concentration of carbon monoxide with an increase in the degree of diesel/CNG replacement. The course of curves is difficult to associate between individual engine operating states, the mileage of the graphs clearly differ from each other. The CO content of diesel/CNG was up to 20 times higher in some dual fuel engine operating states than when powered solely by diesel and generally increases with an increase in the degree of diesel-CNG replacement. The most intense increase in CO concentration can be observed with small (up to 30%) replacement values. The CO content at an engine load of ~0.5 kN·m and at 1300 crankshaft revolutions per minute increases with an increase in CNG conversion to 60%. Above this level, it decreases to a value of about 150 ppm, but it is still 10 times higher than when the engine is powered solely by diesel. Possible decreases in the concentration of CO in the exhaust gases occur at the replacement level of 40–70%, but they are many times higher than when powered only by diesel.

The increasing share of CO in the exhaust gases, which increases with the increase in the degree of replacement, is correlated inversely with the NO content in the exhaust gases, which is consistent with theoretical assumptions [38]. The concentration of nitric oxide in the exhaust gas is shown in Figure 13.

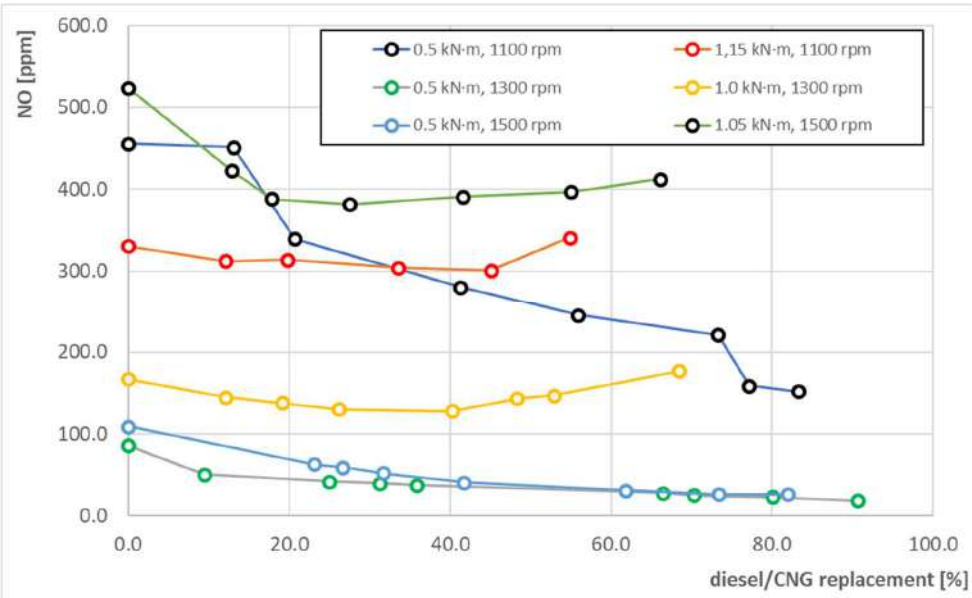

Figure 13. Concentration of nitric oxide in the exhaust gas for individual engine operating points (rotation speed, torque) as a diesel/CNG replacement function.

The graph shows a decrease in the concentration of NO in the exhaust gases with an increase in the degree of replacement to a value of about 30% of the degree of replacement, while with a further increase in the degree of replacement, from a level of about 40%, the concentration of NO in the exhaust gas in some operating states increased. There is no clear correlation with the temperature of the exhaust gases, so the increases may result from the formation of local combustion areas with elevated temperatures. This is due to the failure to adapt the geometry of the combustion chamber to the needs resulting from the supply of natural gas to this engine. Decreases in NO concentration, on the other hand, most likely result from reduced availability of methane used by the methane supplied to the engine, because along with an increase in the degree of exchange, the value of the excess air coefficient decreases.

The reverse situation can be seen in the diagram shown in Figure 14, showing the effect of the diesel/CNG replacement factor on the concentration of nitrogen dioxide in the exhaust gas.

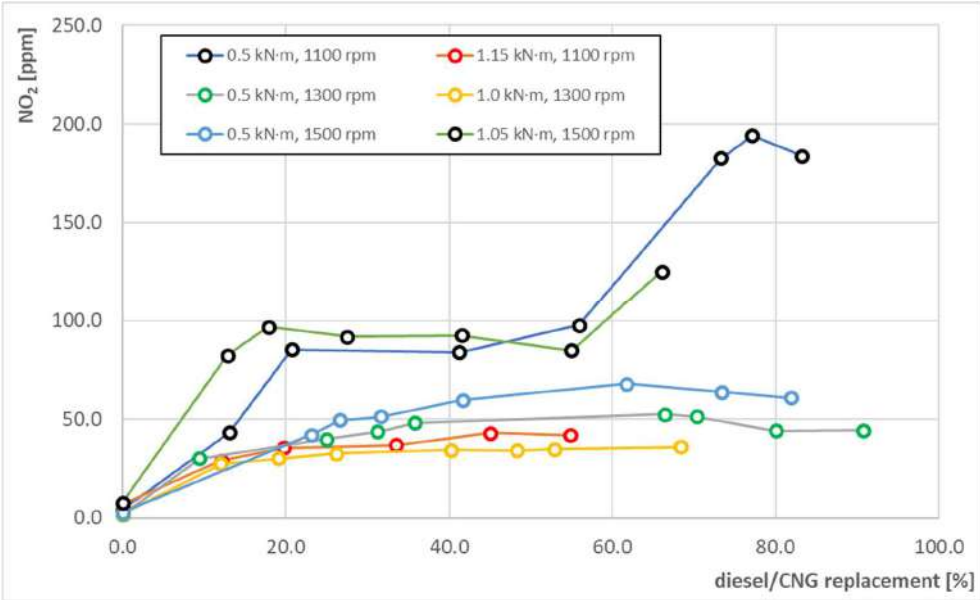

Figure 14. Concentration of nitrogen dioxide in the exhaust gases for individual engine operating points (rotation speed, torque) as a function of replacing diesel/CNG.

Intensive growth with an initial small degree of replacement may indicate the formation of a larger frontal surface of the flame, which results in an intensification of the phenomenon of frontal (fast) NO_X formation, including NO_2. There is no correlation of such a clear increase in NO_2 concentration with the increased temperature of the exhaust gases, which, given the lack of this correlation also in the case of NO concentration, may indicate the widespread occurrence of a reduction reaction (2) in flame areas with low temperatures, the OH component is stable and can react with NO formed in areas with a higher temperature [39].

$$NO_2 + HNO + OH \quad (4)$$

It should be remembered, however, that the combustion temperatures that could occur locally are uneven, and at low temperatures a small concentration of hydrocarbons can accelerate the process of transition of nitric oxide to nitrogen dioxide. When the exhaust gas components forming the so-called NO_x are in chemical equilibrium with each other, then the amount of NO_2 reaches very low concentrations compared to NO [40]. However, there is a clear increase in the concentration of NO_2 in the exhaust gas in the case of two engine operating states—at a load of ~0.5 kN·m and 1100 crankshaft revolutions per minute, and at a load of ~1.05 kN·m and 1500 revolutions per minute. The concentration increases most strongly from the initial low replacement to a value of about 20%, then for subsequent replacement values it stabilizes so that at values exceeding 60% of the replacement, it increases again. This trend does not occur in other engine operating states, in which the NO_2 concentration values after the initial increase stabilize and remain approximately constant in the remaining replacement values exceeding 15–20%. The linear course of these graphs indicates the stability of combustion for different degrees of replacement in these engine operating states, while the states in which a clear increase in NO_2 concentration is observed are also correlated with the increase in CO concentration visible in the previously presented Figure 12 and with the increase in CH_4 shown in Figure 15, illustrating the influence of the diesel/CNG on the methane concentration in the exhaust gases as shown below.

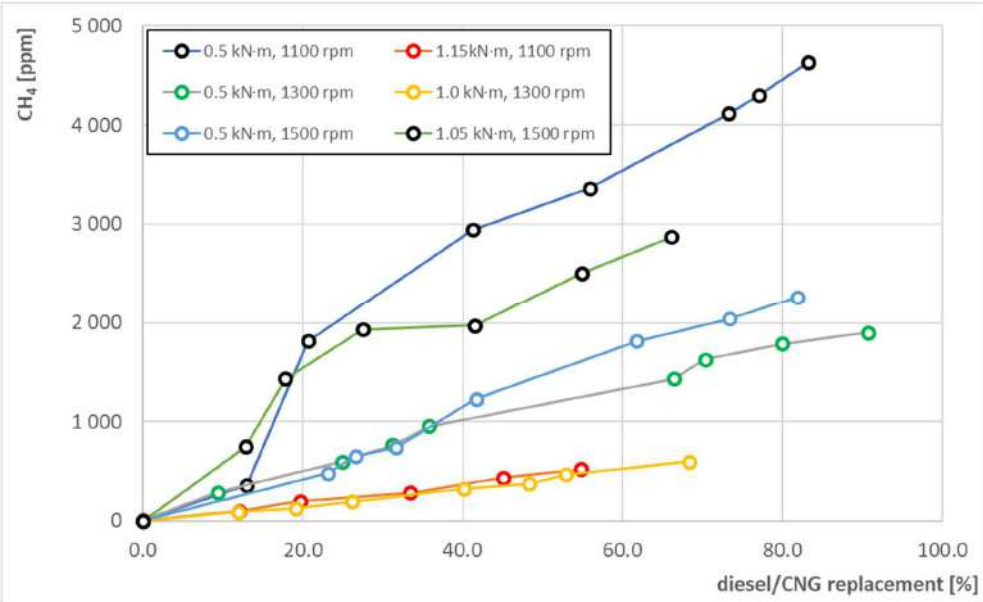

Figure 15. Methane concentration in the exhaust gas for individual engine operating points (rotation speed, torque) as a function of replacing diesel/CNG.

The graph shows a clearly higher concentration of CH_4 for those engine operating states for which higher concentrations of CO and NO_2 have also been recorded. This clearly indicates an increase in the phenomenon of incomplete combustion of hydrocarbons supplied to the engine, which is most likely due to the previously described incompatibility of the combustion chamber geometry to supply the gaseous fuel supplied to the engine. Incomplete combustion can occur in various engine operating states, because it depends to a large extent on the composition of the natural gas-air mixture in the combustion chamber. The formation of poorer and richer fuel zones results in part, from the degree of turbulence of the [41] mixture, which, with a combustion chamber not adapted to such a mode of operation, may differ significantly in different engine operating conditions. Hence, in individual engine operating states, there may be intense increases in emissions of unburnt CH_4 which is difficult to explain by other reasons. The increasing emission of CH_4 results directly from the increase in the exchange ratio, i.e., the appearance in the combustion chamber of more methane that will not be fully burned [42]. Its appearance in a high concentration has an impact on the formation of NO_2, which, according to the Fenimore mechanism, converts the nitrogen contained in the mixture into its oxides as a result of hydrocarbons present in the fuel [43], which in the case of the correlation of their high concentration with high concentration of unburnt methane suggests an intense occurrence of this phenomenon. Another reason for the presence of methane in the exhaust gases is the failure to adjust the geometry of the camshaft to the change of method of fuel supply. A common rule is the use of "cylinder flushing" in turbocharged engines, i.e., part of the supercharged air in the final phase of the exhaust process gets, along with the exhaust gases, into the engine exhaust system. In this case, the inlet air to the engine contains methane, which causes an increase in its concentration in the exhaust gases.

In the case of a working state in which the engine was loaded with a braking torque of ~1.05 kN·m at a crankshaft speed of 1500 rpm, there was also a high concentration of ethane, as can be seen in Figure 16, showing the influence of the diesel/CNG replacement coefficient on ethane concentration in the exhaust gases.

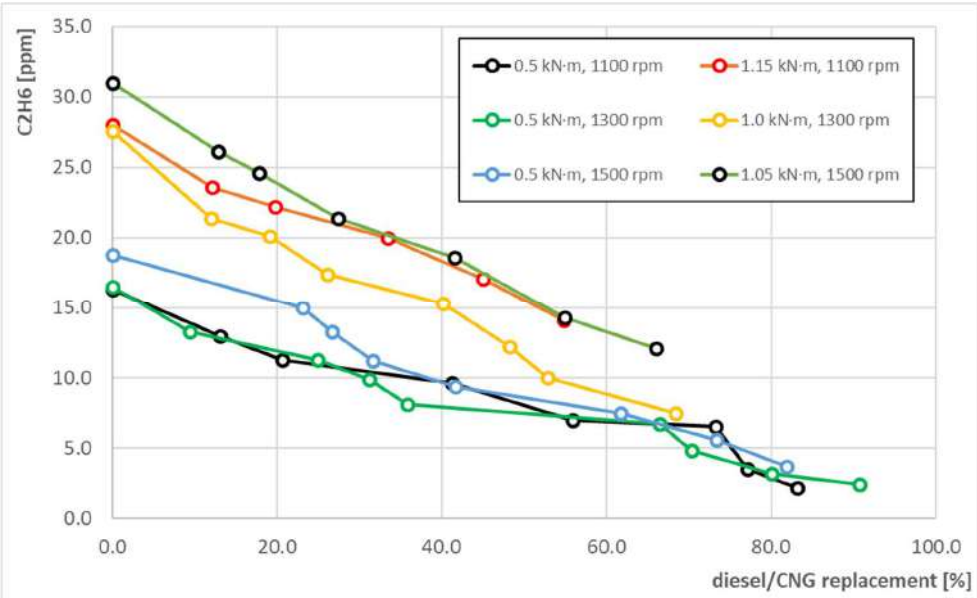

Figure 16. Ethane concentration in the exhaust gases for individual engine operating points (rotation speed, torque) as a function of diesel/CNG replacement.

A clear decrease in ethane concentration is visible with an increase in the diesel/CNG replacement coefficient in all engine operating states. The values of decreases exceed 75%, which is mainly due to the reduction in the share of hydrocarbons from injected diesel fuel, whose energy share with the increase in fuel replacement is systematically decreasing.

However, the upward trend can be seen in the diagrams in Figure 17, which shows the concentration of all hydrocarbon components excluding methane—NMHC (Non Methane Hydro Carbons)—depending on the degree of replacement, in which the points which are extremely different from the adjacent operating points clearly indicate incomplete combustion resulting from insufficient turbulence of the mixture in a given load state of the IC engine, which, as previously demonstrated, results from the inadequacy of the design of this engines combustion chamber for combustion of natural gas.

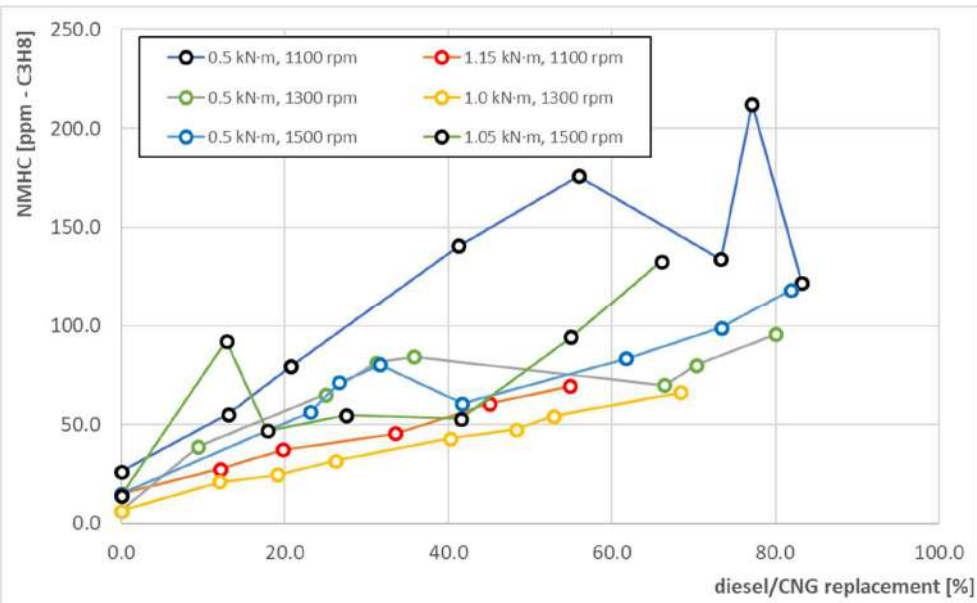

Figure 17. NMHC concentration in the exhaust gases for individual engine operating points (rotation speed, torque) as a function of diesel/CNG replacement.

In general, the charts show an upward trend of NMHC concentrations with an increase in the degree of replacement of diesel by CNG. Despite the decrease in the amount of injected diesel fuel, the number of unburned hydrocarbons heavier than CH_4 is increasing, which indicates an increase in the share of unburnt fuel in exhaust gases. This phenomenon suggests that the combustion chamber of the tested engine should be rebuilt in order to improve the possibility of using natural gas as an additional fuel in this engine.

On the other hand, a clear decrease in all concentration values with an increase in the degree of replacement is also visible for NH_3, which is visible in Figure 18 showing the influence of the diesel/CNG replacement coefficient on the concentration of NH_3 in the exhaust gas at the output from the engine.

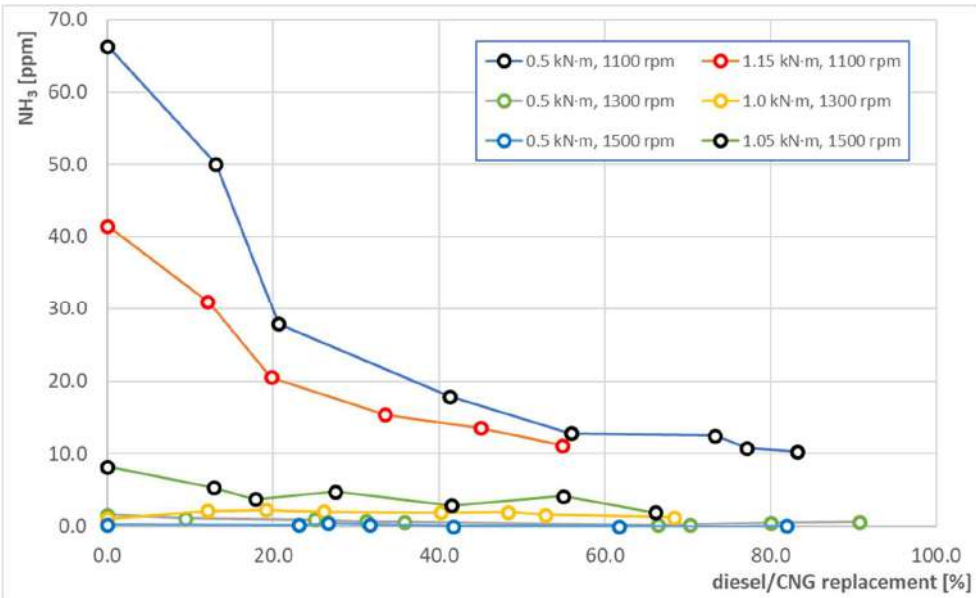

Figure 18. Detonation status for individual engine operating points (rotation speed, torque) in the diesel/CNG replacement function.

Clear decreases in the concentration of ammonia in the exhaust gases are an interesting phenomenon, because they are associated with the emissions of other harmful exhaust components already presented, and the phenomenon of direct NH_3 emissions from the IC engine most strongly affects engines powered by LPG or CNG. In the case of a dual fuel engine, it is therefore interesting to look at the phenomenon of the formation of this harmful substance in the process of co-combustion of diesel fuel and natural gas.

NH_3 can be emitted from a piston engine every time it is operating on stoichiometric or close to stochiometric parameters of combustion. NH_3 emissions are mainly created by the engine NO emissions by reactions (5) and (6).

$$2\,NO + 4\,CO + 2\,H_2O + H_2 \rightarrow 2\,NH_3 + 4\,CO_2 \quad (5)$$

$$2\,NO + 5\,H_2 \rightarrow 2\,NH_3 + 2\,H_2O \quad (6)$$

As the concentration of CO, NO and H_2 in the exhaust gas increases, the amount of NH_3 increases simultaneously. The mentioned substances are natural components of the exhaust gases, which determine the amount of NH3 produced. NO is formed when nitrogen-containing substances are involved in the combustion process (it can come from both air and fuel). CO is most strongly produced in rich fuel-air mixtures, where there is a small amount of oxygen for the fuel to be burned. H2 can be produced when fuel is burned under high pressure and in high temperatures, because under these conditions the chemical

bonds that hold the hydrogen contained in the fuel break down. Other possibilities of obtaining hydrogen are water gas shift (7) or steam reforming (8) [44].

$$CO + H_2O \rightarrow CO_2 + H_2 \quad (7)$$

$$CnHm + n\ H_2O \rightarrow n\ CO + (m/2 + n)\ H_2 \quad (8)$$

Higher engine NO, CO, HC and H_2 emissions could bring about higher NH_3 emissions, but not all of these substances were measured in our research. It can be seen that the type of fuel used may have an influence on the amount of H_2 produced in the combustion chamber according to reaction (8), so the level of replacement can influence it too.

Summing up, along with the enrichment of the air-fuel mixture, the conditions for the formation of NH_3 improve in general, but in our case, the high content of methane led to a reduction in its concentrations due to the reduction of the concentrations of substances generated mainly in the process of combustion of fuels with longer hydrocarbon chains [45].

The authors of the article says that the main way to reduce NH_3 emissions in SI engines is to reduce CO and NO emissions in the exhaust gas. A similar situation can be found in dual fuel engines using a large amount of gaseous fuel. The authors state that "in particular, this may be a significant problem in gaseous engines". Variability in gaseous SI or CI dual fuel engines should be lower than that for gasoline engines.

In the case of our research, the concentration of NH_3 in the case of four engine operating states did not change significantly, while in the case of two others it decreased significantly. No concentrations in the exhaust gases were fairly stable, while CO increases were more intense, but in operating states where an intense decrease in NH_3 concentration was observed, it is difficult to find a correlation between their concentrations. On the basis of the theoretical assumptions cited, for a more thorough analysis it will be necessary to know the concentration of water and hydrogen in future tests. Reactions (9) and (10) between ammonia and nitrogen oxides inside the (inactive) SCR reactor may have affected the final emission of both of these substances, which also makes it impossible to clearly indicate the cause of such intense changes in NH3 concentration in some engine operating states [39].

$$4\ NO + 4\ NH_3\ th\ O_2 \rightarrow 4\ N_2 + 6\ H_2O \quad (9)$$

$$NO + 2\ NO_2 + 4\ NH_3 \rightarrow 4\ N_2 + 6\ H_2O \quad (10)$$

Summary of the Results

Summarizing the obtained test results and dividing them into individual engine operating states, the following analysis of the obtained results can be carried out:

- Test results obtained for an engine loaded with approximately 520 Nm at 1100 rpm

Smoke opacity, regardless of the CNG content, is similar to the smoke obtained for an engine sewn exclusively with diesel fuel. Only with a 20% replacement, the smoke is approximately 3027 times greater than the average smoke value for the remaining CNG substitution factor, which will require more in-depth diagnostics in the future. The CO content of diesel/CNG is up to 20 times higher than that of diesel-only and generally increases as the degree of replacement of diesel oil by CNG increases. The content of NO, NOx and ammonia NH_3 decreases with increasing degree of substitution, despite the increase in exhaust gas temperature. When increasing the CNG content, the share of methane CH_4 and ethane C_2H_6 in the exhaust gases increases.

- Test results obtained for an engine loaded with a torque of approx. 1150 Nm at a speed of 1100 rpm.

For these test conditions, it was not possible to achieve stability, without detonating engine operation with diesel replacement by CNG above 50%. Smoke opacity, regardless of the CNG content, is similar to the smoke obtained for an engine supplied exclusively with diesel fuel. Only with a replacement of 15% is the smoke opacity 10 times greater than the average smoke value for the remaining CNG diesel replacement values, which is a

critical case, and which will be further verified in subsequent tests of this engine. As the CNG content increases, the share of CO increases. The content of NO decreases and NO_2 increases as the conversion rate increases. With a replacement rate of 50%, the NO_2 content increases significantly compared to the nitrogen oxide content when powered solely by diesel. This is despite the decrease in exhaust gas temperature in this range of engine operation. When increasing the CNG content, the share of methane CH_4 (up to five times higher than when powered only by diesel) and ethane C_2H_6 (three times compared to diesel-only) increases in the exhaust gases.

- Test results obtained for an engine loaded with approximately 480 Nm at 1300 rpm

For these test conditions, it was possible to obtain stable, stemless operation of the motor with the replacement of diesel by CNG even at the level of 85%. Smoke opacity regardless of the CNG content is very small and similar. It ranges from 0.01 to 0.05 m^{-1}. The CO content increases as CNG conversion increases up to 60%. Above this level, it decreases (to a value of approx. 150 ppm), but it is still 10 times higher than when the engine is powered only by diesel. An increase in the degree of conversion causes a decrease in both the amount of NO and NOx. The proportion of ammonia NH_3 decreases with increasing degree of conversion (up to 60%), and then increases slightly. With an increase in the share of CNG, the content of CH_4 is systematically increasing. A similar relationship was observed for ethane (C_2H_6) to 50%, and then above 80%. The share of CH_4 increases from 10 ppm (when powered only by diesel) to 130 ppm for 85% CNG conversions.

- Test results obtained for an engine loaded with a torque of approximately 980 Nm at a speed of 1300 rpm

For these test conditions, it was possible to obtain stable, stemless operation of the motor with the replacement of diesel by CNG up to 70%. The smoke with diesel is 0.04 m^{-1}. With a 15% share of CNG, it increases to a value of about 0.13 m^{-1}, and then generally decreases. The CO content increases as CNG conversion increases. At 40–70% share, CNG is 9 times higher than when powered solely by diesel. An increase in the conversion rate to 40% causes a decrease in the amount of NO. At higher conversion rates, NO increases. With a conversion of 70%, the proportion of nitrogen oxides is greater than when powered solely by diesel. With an increase in the share of CNG, the content of both methane CH_4 and ethane C_2H_6 is systematically increasing. A similar relationship was observed for C_2H_6.

- Test results obtained for an engine loaded with approximately 500 Nm at 1500 rpm

For these test conditions, it was possible to obtain stable, smooth operation of the motor with the replacement of diesel by CNG up to 80%. The smoke is very small and is 0.01 to 0.04 m^{-1}. As the CNG content increases, the CO_2 content decreases, while the share of CO increases. The share of both NO and NOx decreases with increasing CNG conversion rate. When increasing the share of CNG, the share of methane CH_4 and propane C_2H_6 increases.

- Test results obtained for an engine loaded with approximately 1040 Nm at 1500 rpm

For these test conditions, it was possible to obtain stable, stemless operation of the motor with the replacement of diesel by CNG only up to 60%. Smoke opacity increases in the conversion rate range of 0–30% from 0.015 m^{-1} (for diesel only) to 0.05 m^{-1} for a conversion rate of 15–30%. With a further increase in the share of CNG, the smoke decreases to 0.035 m^{-1} with the largest share of CNG. The share of CO increases with the increase in the degree of CNG conversion (up to 30%). Above this value, the share of CO is approximately constant. In this respect, it is 20 times larger than when the engine is powered solely by diesel. As the CNG content increases, there is a tendency to reduce the proportion of both NO, NO_X and NH_3. At the same time, for the degree of CNG conversion above 50%, the share of nitrogen oxides NO_X increases rapidly).

The aggregate graphs clearly show the twin trends occurring for the same degrees of diesel fuel replacement by CNG at individual engine operating points. On the XY diagrams,

a clear deviation from the trend of the NO_2 value can be seen at an engine load of 0.5 at 1500 rpm. This is also visible for CH_4 and CO emissions, where also for a rotational speed of 1300 rpm the CO concentration value was higher than at other operating points. At this engine speed, regardless of the load, significantly higher smoke was recorded than at other operating points.

Based on the analysis of the obtained test results, the conversion values (share) of CNG in the fuel dose were determined, which allowed the minimum lowest smoke k min, emissions CO, NO, NO_2, CH_4, NH_3 and C_2H_6 for the considered operating parameters of the Volvo FH13 engine to be obtained—these values are summarized in Table 2.

Table 2. The replacement of diesel/CNG setting with the lowest concentration of harmful components.

Rotational Speed [RPM]	Load [Nm]	CNG Replacement Setting [%] When There Is a Minimum Share of:						
		CO	NO	NO_X	NH_3	C_2H_6	CH_4	k_{min}
1100	520	20	70	50	80	30	20	60
	1150	15	40	30	50	15	15	40
1300	480	20	85	85	60	20	20	80
	980	15	40	40	70	15	15	60
1500	500	20	70	80	70	50	20	80
	1040	15	30	30	60	20	15	60

4. Conclusions

1. It was noticed that the smoke density and exhaust gas temperature did not decrease with increasing degree of diesel oil replacement with natural gas. The concentration of CO, NO_X, CH_4 and NMHC increased, while the concentration of CO_2, C_2H_6, NH_3 and the consumption of diesel fuel by the engine decreased significantly. The conducted research has shown the possibility of obtaining high degrees of diesel/CNG exchange. An increase in the methane content in the air-fuel mixture will match the increase in the proportion of toxic exhaust gas components (before the exhaust after-treatment system).
2. The biggest amount of fuel replacement that could be carried out without knocking the combustion effect was about 90% of CNG fuel. This is a good result in comparison to other researchers' test results.
3. Unambiguous indication of the optimal settings for adjusting the diesel/CNG ratio due to the different impact of the same CNG proportions on the parameters under consideration is difficult. The performed research allowed us to quantify the impact of the CNG share on the parameters under consideration, and the obtained dependencies, shown in the figures, are often not unambiguous.
4. The tests showed that the use of CNG fuel causes an increase in the share of toxic compounds in the exhaust gases, to the greatest extent CO and HC hydrocarbons. The content of these components at many measuring points is several times to several dozen times higher than for an engine powered solely by diesel. The dependencies shown in the drawings are difficult to explain unequivocally, comparing them with the results available in other studies [46–49] it is possible to clearly state the inadequacy of the engine to supply a dual fuel using natural gas and diesel, and in order to use the full potential of this type of power supply thorough structural changes to the engine are required. The increase in hydrocarbon emissions may result from the course of fuel combustion (however, it can be said with certainty that unchanged design solutions of an engine originally adapted to burn only diesel fuel contribute to a large extent to the emission of unburnt methane, which is the largest share of hydrocarbon compounds emitted by the engine). In turn, the increase in the share of carbon monoxide is difficult to explain; reducing the coefficient of excess air leads to

an increase in the concentration of CO in the exhaust gases. This is due to the much lower availability of oxygen in the combustion chamber than in the case of mono-fuel diesel fuel.

5. A decrease in CO_2 content was found with the diesel/CNG mixture, which is associated with a change in the elementary H/C ratio in the outgoing exhaust gases. This is beneficial from the point of view of toxic and greenhouse gas emissions. Changes in CO_2 content are proportional to changes in the diesel/CNG substitution factor.
6. A dozen or so percent increase in the content of nitrogen oxides—mainly nitrogen dioxide in the exhaust gases—was found. This is related to the change in the organization of the process—from qualitative to quantitative control. The introduction of CNG gas into the intake manifold reduces the excess air coefficient and increases the combustion temperature of the fuel-air mixture. Increasing the combustion temperature results in an increase in the NO_2 content, which affects the total NOx emissions. The vehicle from which the engine comes is equipped with an SCR system, which should reduce NOx emissions to the required level.
7. A several-fold increase in the methane content of the exhaust gases was found. This is related to the contraction of the intake and exhaust system—flushing the combustion chamber. This phenomenon may also cause an increase in carbon monoxide emissions during type-approval tests and, as a result, exceeding the limits resulting from the requirements of EURO standards. This conclusion is confirmed by a several-fold increase in the content of carbon monoxide in the exhaust gases. This is due to the introduced changes in the organization of the combustion process, i.e., the transition from qualitative to quantitative control, as in the case of NOx concentration. The introduction of CNG gas into the intake collect causes the fuel to be mixed with air beforehand, resulting in a mixture close to homogeneous, which is then supplied to the cylinder. After compression, diesel is injected into this gas mixture in the liquid phase. This causes local oxygen deficiencies which promotes incomplete combustion and the formation of carbon monoxide and an increase in smoke when feeding diesel/CNG in the range of 50–120% depending on the operating conditions. This is a very unfavorable phenomenon from the point of view of environmental protection due to the increase in the emission of particulate matter into the atmosphere.
8. The reasons for the increase in smoke can be several:
   - reduction of the excess air coefficient caused by the supply of CNG fuel in gaseous form,
   - change in the method of combustion of the fuel-air mixture, resulting from the shift of the flammability limit and homogenization of the mixture,
   - local oxygen deficiencies, resulting from co-combustion of CNG and diesel, which leads to a local reduction in the coefficient of excess air, especially in the area of diesel combustion.
9. In order to optimize the engine due to the emissions of individual exhaust components, tests must be carried out to select the timing phases and to optimize the CNG fuel injection process.
10. In the opinion of the authors, it is necessary to further evaluate the energy state of the tested engine and to evaluate the emissions of individual exhaust gas components. According to the researchers, some issues, such as drops in NH_3 concentration, increased NMHC concentrations or the lack of noticeable smoke reduction, require further research. Moreover, the concentrations of other exhaust gas components and the achieved overall efficiency of the engine are also interesting and require further analysis.

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Influence of the Working Parameters of the Chassis Dynamometer on the Assessment of Tuning of Dual-Fuel Systems

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Abstract: The article presents the justification for the necessity to use chassis dynamometers in the tuning process of dual-fuel trucks. The research system used and the research methodology are presented. The research results present the approach to solving problems related to setting the technical (physical) data of the tested vehicle on the dynamometer, selection of the vehicle engine operation range, the impact of the value of the forced load on the vehicle drive axle, selection of the dyno operation mode for the expected tasks and the impact of the correctness of the selection of the scope of the analysis of data on losses in the drive system. The article shows the above-mentioned influence on the test results on the dynamometer and on the tuning results. The article closes with a conclusion detailing prospects for developing the presented results.

Keywords: chassis dynamometer; semi-truck; dual-fuel; tuning

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1. Introduction

Modern digitally controlled gas fueling systems require appropriate adjustment called calibration, i.e., proper selection of gaseous fuel injection times, depending on the rotational speed and engine load [1]. Most of the commercially offered gas installation controllers are equipped with the auto-calibration function, which allows the appropriate settings to be selected automatically [2]. It is most often performed at engine idling, which does not take into account the variable load, the operation of additional engine-driven systems and the related need to increase or decrease the dose of gaseous fuel, and a number of other factors that have a significant impact on the engine performance [3]. In most popular vehicles, the auto-calibration procedure is sufficient for the proper operation of the gas supply system; however, there are cars in which this is insufficient, and it is required to perform a more accurate calibration of the engine running on gas fuel under load [4]. In such cases, the auto-calibration serves only to prepare the engine for operation and start it in the "gas" mode [2,3,5,6]. Subsequently, a road test and continuing tuning under road conditions is suggested.

Semi-tractor engines using dual-fuel installations (pilot diesel dose starting the combustion process of a compressed gas dose in the engine cylinder) do not always have the possibility of auto-calibration, because they result from the inability (or incomplete ability) to work in the dual-fuel mode at engine idle speed (with no injection a suitably small pilot dose, e.g., in engines using unit injectors) [6]. Turbocharged diesel engines used in trucks are not able to generate the full boost pressure, and thus the maximum torque values, without an appropriate load [7]. In addition, semi-trucks, due to the specificity of their work, generate very high torque values on the wheels, which is necessary to enable the movement of heavy loads while maintaining appropriate dynamics and in changing road conditions (e.g., changing the elevation of the ground or wind direction). It is impossible or very difficult to force the engine to work in extreme conditions during road tests without a load. A reasonable solution is therefore to use a chassis dynamometer, which allows you

to enforce appropriate operating conditions of the drive system and maintain them for a specific time [5,8].

A chassis dynamometer is a device that is most often used to measure the power and torque of a vehicle engine [1]. The measurement is performed indirectly by registering the torque on the rollers of the chassis dynamometer [1]. Relating the value of this parameter to the rotational speed of the rollers, and then via the drive system to the drive speed of the engine crankshaft, allows the power of the vehicle to be calculated [1]. However, these are not the only possibilities of a chassis dynamometer [9]. It also allows you to simulate the actual load conditions that occur on the wheels of the vehicle while driving [1,6]. This allows, for example, fuel consumption to be measured, exhaust emissions to be checked while driving, or the behavior of various engine systems to be tested under load [5,8]. It is also possible to measure the flexibility of the car's engine within a certain speed range, to determine the correctness of the speedometer and tachograph indications, and to check the hill-climbing ability. The ability to simulate various load conditions and maintain certain conditions over a longer period of time allows for accurate observation of the gas supply system settings or their changes, and allows for accurate calibration of the gas supply system and detection of faults occurring under strictly defined conditions, e.g., at very high loads or boost pressure [7]. The installer can comfortably observe the indications of measuring instruments using different variants of the dynamometer operation (maintaining a constant load or engine speed), depending on the needs [3,6,10].

Constant engine speed dynamometer mode is very useful in calibrating gas supply systems. Thanks to this, you can adjust the gas injection time and the angle of its injection by moving the controller map so that it reflects the work on the original fuel supply system as much as possible. Under the conditions of the road test, it is impossible to maintain a constant engine speed with increasing load, even when using a hill or a large load for this purpose. That is why the chassis dynamometer is a device ideally suited for the calibration of gas installations for trucks [3,11]

2. Materials and Methods

2.1. Research Setup—Materials

This work deals with the aspects and correctness of the use of a chassis dynamometer in the process of tuning dual-fuel CNG/diesel installations in a 2014 Volvo FH 13 semi-truck and the impact on the tuning results and their correctness. The vehicle's engine was converted to dual-fuel operation by adding a prototype sequential indirect gas injection system. The tests were carried out on a mobile dynamometer device built on the structure of a 6 ft container which is shown in Figure 1 below.

(**a**) (**b**)

Figure 1. A mobile dynamometer device: (**a**) front view, (**b**) view of the interior of the dynamometer (authors' source).

Table 1 presents the technical data of the mobile dynamometer device declared by the manufacturer. Table 2 presents the technical data of the tested tractor unit.

Table 1. Selected declared parameters of the mobile dynamometer device according to the design assumptions and the manufacturer (manufacturer data).

Parameter	Value/Name/Type
Mode of Operation	Load Type
Brakes type	Frenelsa FF16—eddy current 2 × 3300 (Nm)
Number of measuring axle	1
Max/min wheel track	2700/1000 (mm)
Rollers diameter	320 (mm)
Minimum wheel diameter	700 (mm)
Maximum load on the measuring axle	15,000 (kg)
V-max	200 (km/h)
Maximum measurement error	1 (%)
Permissible ambient temperature (operation)	od −10 do +30 (°C)
Permissible ambient temperature (storage)	od −40 do +50 (°C)
Mechanical synchronization of rotational speed of dynamometer rollers	Yes

Table 2. Selected factory technical data of the tested vehicle [12].

Parameter	Value/Name/Type
Vehicle Brand/Model/Year of Production	Volvo/FH13/2014
Engine type	Turbodiesel, D13C460, EU5SCR-M
Engine displacement/Number of cylinders/ Cylinder diameter/Piston stroke/ Compression ratio	12.8 L/6 cylinders/131 mm/158 mm/17.8:1
Max power	460 hp (338 kW) at 1400–1900 rpm
Max engine speed	2100 RPM
Economic speed	1000–1500 RPM
Optimal speed range	1150–1400 RPM
Gearbox type	VTO2214B, Automatic with manual selection
Number of gears	14 gears

Due to the nature of the research, the measured parameters should be divided into the following groups of parameters characterizing:

- The operation of the dual fuel (CNG/diesel) system controller;
- The work of the chassis dynamometer;
- The operation of the semi-tractor engine control system.

The parameters for each group that should be measured, and the necessary measuring equipment are presented below.

In the tested case, the engine was converted to dual-fuel operation by adding a prototype sequential indirect gas injection system. In this system, pilot diesel fuel acts as the ignition source, the system has an ability to "cut" and "exchange" the diesel oil dose to a gas fuel up to 95%, and it uses 2 gas injectors per cylinder [5,13]. Diagram of operation is presented in Figure 2 below.

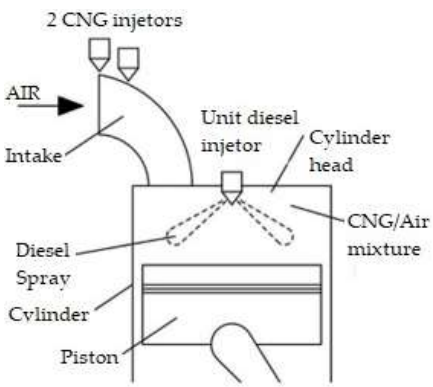


Figure 2. Diagram of operation of a dual-fuel supply system (authors' source based on [14]).

Parameters characterizing the operation of the dual-fuel supply system, recorded with the use of the gas fuel supply system controller are shown in Table 3 below.

Table 3. Selected parameters characterizing the operation of the gas fuel supply system (manufacturer's data).

Parameter	Value/Name/Type
Time [s]	Time from the Beginning of the Mileage Recording, the Basis of the Recorded Mileage
Engine speed [RPM]	Engine's crank speed registered by gas supply controller
Instantaneous CNG consumption [kg/h]	Gas mass fed to the inlet manifold,
Instantaneous diesel oil consumption [kg/h]	Instant consumption of diesel oil—diesel mass fed to the cylinders
Diesel actual [%]	Actual instant share of diesel oil in the dose of fuel delivered to the engine
Diesel averaged [%]	The average periodic share of diesel oil in the fuel dose delivered to the engine
Diesel dose [mg]	Instantaneous dose of diesel fuel delivered to one cylinder for one work cycle
Gas time [ms]	Instantaneous gas injector opening time for one/particular cylinder for one work cycle
Calculated gas time [ms]	Instantaneous gas injector opening time for one/particular cylinder for one work cycle after taking into account the correction parameters
Temp. reducer [°C]	Temperature of the gas pressure reducer
MAP pressure [bar]	Air pressure in the intake manifold
Gas pressure [bar]	Gas pressure on the gas pressure reducer
Pressure on gas rail [bar]	Gas pressure on the injection rail
Lambda	Signal from the oxygen concentration sensor in the exhaust gas—lambda probe
System faults	Monitor of faults generated by the gas supply system

Parameters characterizing the work of the chassis dynamometer measured with the use of the dynamometer controller—software are shown in Table 4 below.

Table 4. Selected parameters characterizing the chassis dynamometer measured with the use of the dynamometer controller (manufacturer's data).

Parameter	Value/Name/Type
Selected Time [s]	Time from the Beginning of the Test
Engine/roller speed ratio	Speed transmission ratio—motor/rollers
Engine speed [RPM]	Calculated engine speed
Engine speed rotational acceleration [RPM/s]	The determined speed of the engine rotational speed change during acceleration
Road speed [km/h]	Calculated linear driving speed of vehicle
Rollers received power [hp]	The power received on each of the roller
Rollers inertial torque [Nm]	The moment of inertia on each of the roller
Rollers brakes torque [Nm]	The braking torque for each roller
Power on wheels [hp]	Calculated power on vehicle wheels
Loss power [hp]	Total power lost resulting from speed changes and load changes
Drivetrain inertial power [hp]	Power of inertia of the drive system
Engine inertial torque [Nm]	Motor moment of inertia
Engine inertial power [hp]	Motor inertia power
Engine power [hp]	Calculated engine power
Engine corr. power [hp]	Corrected power of the engine
Engine torque [Nm]	Engine torque
Engine corr. torque [Nm]	Corrected motor torque
Exhaust gas temperature [°C]	Exhaust gas temperature measured by external thermocouple
Ambient temperature [°C]	Ambient temperature
Ambient pressure [hPa]	Measured by the system of the built-in weather station of the dynamometer
Ambient humidity [%]	Measured by the system of the built-in weather station of the dynamometer

Parameters characterizing the operation of the tractor engine control system. Measurement using a TEXA TXT diagnoscope with IDC5—Truck software:

- Engine rotational speed [RPM];
- Vehicle travel speed [km/h];

- Boost pressure [bar];
- Torque [Nm];
- Hourly basic fuel consumption—diesel fuel [kg/h];
- Engine load [%];
- Engine management system errors.

The elements listed above have been compiled in a research arrangement. Figure 3 below shows a block diagram of the test stand. Figure 4 shows the vehicle undergoing tests.

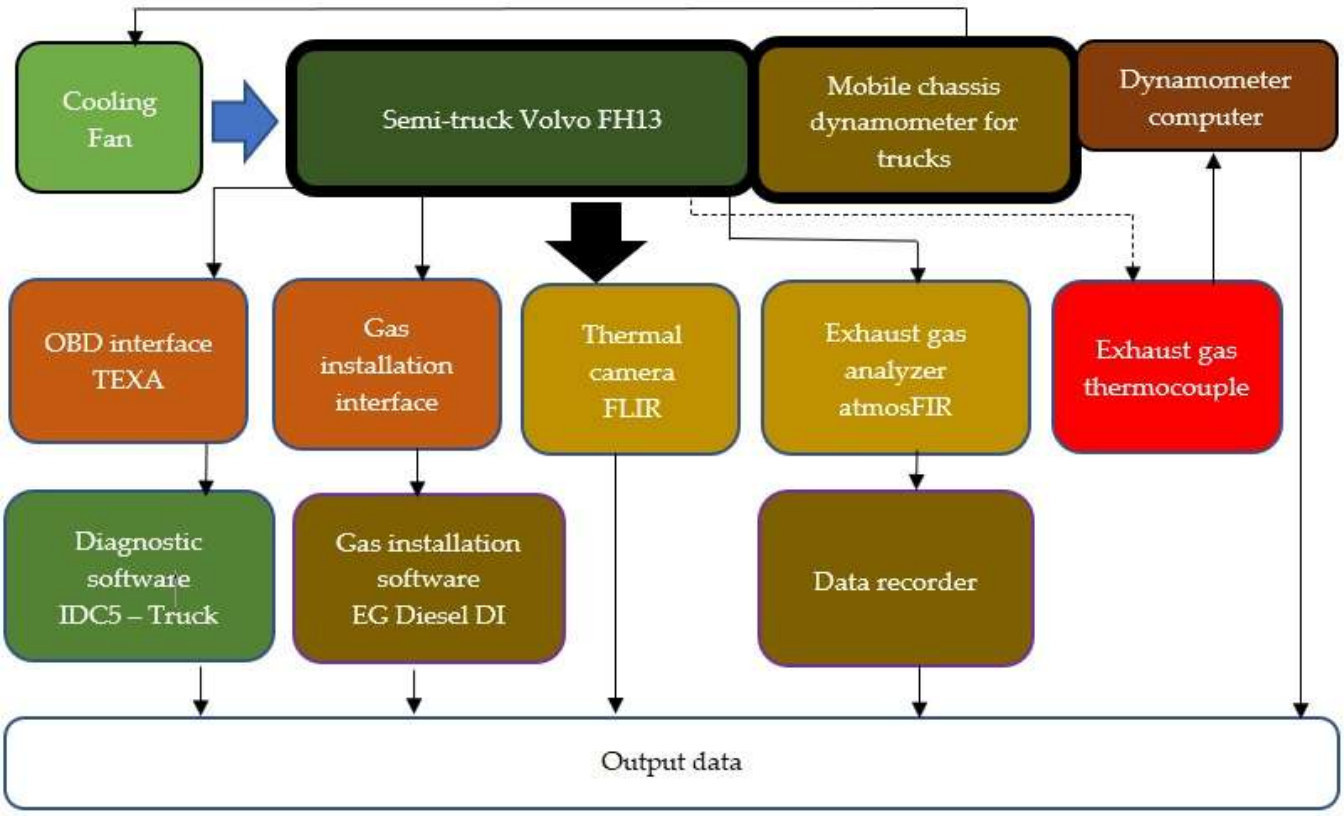


Figure 3. Block diagram of the test stand (authors' source).

(**a**) (**b**)

Figure 4. Volvo FH13 on mobile dynamometer: (**a**) front view, (**b**) view of the vehicle drive axle on dyno rollers (authors' source).

2.2. Methodology

The main goals of the research were to tune the Volvo FH13 semi-truck equipped with D13C engine to work on dual-fuel CNG-diesel mode and also to test the mobile semi-truck dynamometer device. The titular research on the influence of the aspects of the dynamometer operation on the tuning of dual-fuel installations is a derivative of the main research. The research methodology dealing with the above subject matter has been developed empirically and includes an approach to solving problems related to:

- Preliminary data settings for the technical (physical) data of the test vehicle on the dynamometer;
- Selection of the operating range of the vehicle's engine (gear selection);
- The influence of the value of the forced load of the vehicle driving axle;
- Selection of the dyno working mode appropriate to the expected tasks;
- The impact of the correctness of the selection of the scope of the analysis of data on losses in the drive system.

Before the tests, the vehicle was always secured in accordance with the manufacturer's instructions and safety guidelines. The car's operating fluids were heated to normal operating temperature, and the tires at the beginning of each test had a temperature of 50 °C. In order to ensure repeatability of the tests, a number of parameters such as temperature, pressure and speed in the vehicle were monitored in real time.

3. Results

3.1. Settings for the Technical (Physical) Data of the Test Vehicle on the Dynamometer

The main task before starting the test and tuning on the dynamometer is the need to configure the vehicle in the dynamometer software in a way that best suits the actual technical conditions of the vehicle. In the software of the dynamometer on which the tests were carried out, in order to validate the tests, when starting a "new project", the vehicle data such as losses in the drive system, engine and drivetrain system inertia should be correctly configured. At this point, it should be noted that the tool for estimating the inertia of the drive system built into the dynamometer software, calculating this value from the geometric, physical data used on the drive wheels on the vehicle (shown in Table 5), differed from the actual value calculated empirically. When determining the mass moment of inertia of its wheel in relation to its axis of rotation, the method presented in [15] was used. The tested wheel is suspended (Figure 5) on 3 vertical steel ropes l = 6.664 m long, evenly spaced along the circumference with a radius of r = 0.1405 m (the radius value results from the diameter of the hole in the wheel rim).

Table 5. Data table for calculating the inertia of the vehicle propulsion system tested on the dynamometer (authors' source).

Tire size *	315/70/R22,5
Used wheels count *	4
Calculated wheel inertia **	16.72 kg·m^2
Used wheel inertia **	66.87 kg·m^2
Inertia transferred to roller **	0.66 kg·m^2

* Data entered manually. ** Data calculated by the dynamometer software on the basis of *.

Figure 5. Wheel mounted on the stand for determining the mass moment of inertia.

The vibrating system constructed in this way is set in motion by the rotation of the wheel in the vertical plane by a certain angle alpha (its value cannot exceed 15°). Then, the vibration period T is measured, which is determined by measuring the execution time of 10 complete deflection cycles. The following relationship is used to calculate the mass moment of inertia IY:

$$I_Y = \frac{T^2 \cdot r^2 \cdot m \cdot g}{4\pi^2 \cdot l}, \text{ kg}\cdot\text{m}^2 \quad (1)$$

After substituting the value of the measured period T = 15.6 s, the mass moment of inertia is:

$$I_Y = 19{,}315 \text{ kg}\cdot\text{m}^2 \quad (2)$$

This deviates by 15.5%, which may ultimately bend the measurement. The engine inertia is configured on a similar principle, although during research it was unable to verify the correctness of the engine inertia estimation. The range of losses in the drivetrain, entered manually and suggested by the dynamometer software (4–8%), also differs from the value calculated on the basis formulas for the overall efficiency of the drivetrain, according to which the losses are approximately 8–10% depending on the gear ratio used. While during tuning, the final result is not the value of the comparator of performance on diesel only supply and in dual-fuel mode, but the value that approaches the measurement as close to reality as possible, it is worth bearing in mind that the basic settings suggested by the software can really affect the measurement results.

3.2. Selection of the Operating Range of the Vehicle's Engine (Gear Selection)

When tuning the installation, determine the gear ratio that covers the widest range of operating/usable speeds of the engine while maintaining the highest possible speed of the vehicle's drive wheels should be determined (the aim is to reduce the torque on the wheels, the high values torque quickly degrade the tires and increase the measurement error, more on this subject in Section 3.3 of this work [1]). Alternatively, the engine rotational speeds that are of greatest interest to us in the context of the vehicle's operational properties resulting from the specificity of the vehicle's operation or the installation used in it should be determined. For this purpose, it is worth analyzing the factory diagrams of torque and power waveforms, as well as the range of optimized (economic) shaft revolutions. A graph of this type is presented in Figure 6.

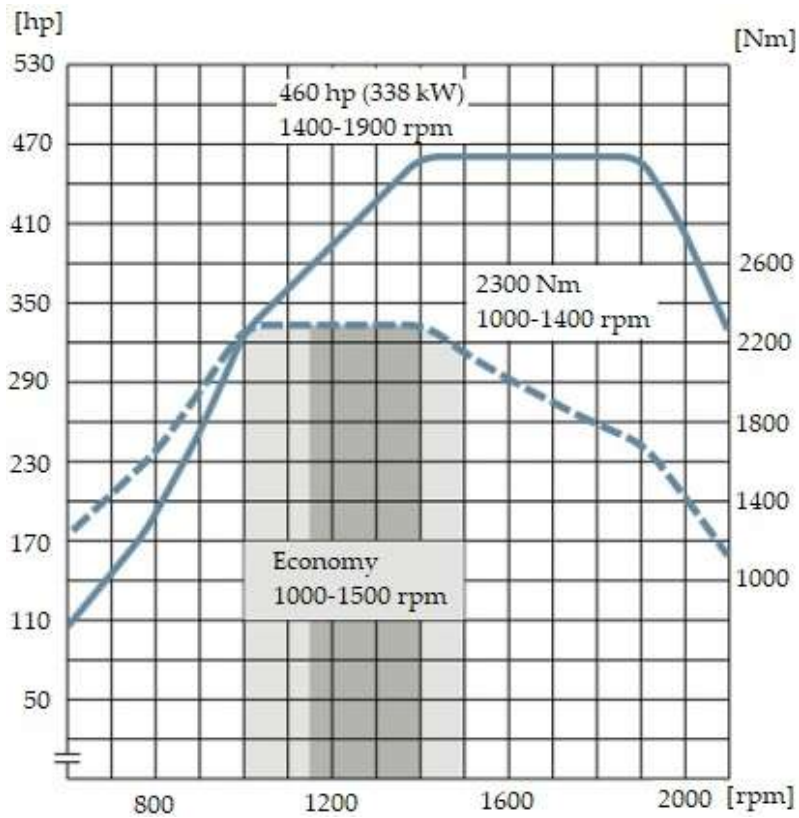


Figure 6. External characteristics of the tested engine [12].

Bearing in mind the manufacturer's chart, it was compared with the results obtained from covering the RPM range for the gears in the test vehicle, which was presented in Figure 7.

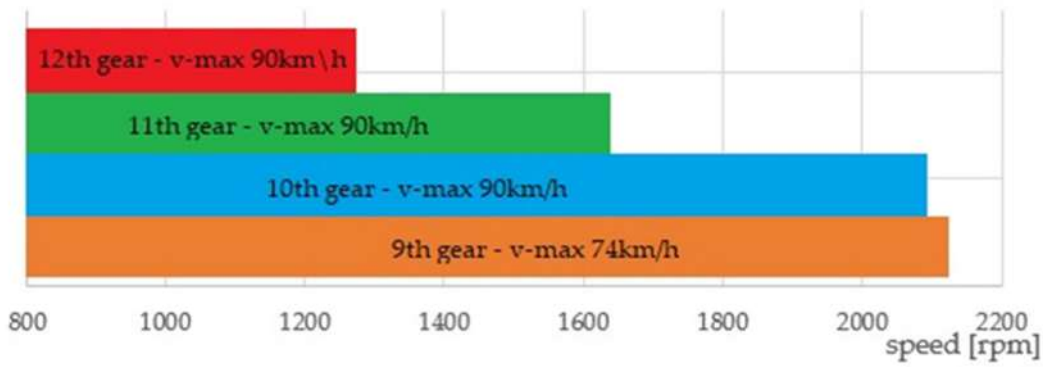


Figure 7. Graph of the coverage of the RPM engine operating range for selected gears (authors' source).

The above data show that the 10th gear is the most appropriate gear for the study of external characteristics and this gear was used to carry out further research presented in this article. In the case of tuning the installation at the engine speed declared by the manufacturer as the most optimal and economical, it should be the 11th gear. The speed range from idle to 800 rpm is irrelevant in the context of these tests, because the gas installation is not able to switch to the dual-fuel mode for technical reasons—the unit injector is not able to generate a correspondingly low pilot dose in this range [11,16].

3.3. The Influence of the Slip of Tires on Rollers

During the research, the influence of slipping of the driving wheels on the rollers of the dynamometer was noticed. This creates a multidimensional problem that can be broken down into several components:

- Dependence of the slip of the driving wheels on the roller due to the load of the driving axle on the rollers;
- Measurement error resulting from the above and from the influence of the engine/roller ratio settings in the dynamometer settings;
- Impact of the tires (heating up due to work on rollers, change in their dynamic radius).

Table 6 summarizes the data from the research on the sliping of the wheels on the rollers for different values of the engine load and different values of the thrust of the driving axle of the tested vehicle.

Table 6. Table of the results of tests of slip of tires on rollers.

Const. Revs. *	1000 RPM (~Max Torque)				1300 RPM (~Eco Revs)				1500 RPM (~Max Power)				1700 RPM (~High Revs)				Axle Load ****
Engine load [%] **	25	50	75	100	25	50	75	100	25	50	75	100	25	50	75	100	
RPM—TEXA **	1001	1022	1039	1066	1299	1323	1349	1380	1489	1519	1545	1574	1679	1710	1737	1768	20,000 N
Dyno (Nm) ***	192	585	935	1310	180	540	886	1275	122	473	799	1110	78	400	700	970	
RPM—TEXA **	992	1009	1028	1045	1288	1309	1331	1353	1483	1503	1524	1549	1673	1697	1717	1741	30,000 N
Dyno (Nm) ***	192	560	917	1293	190	549	870	1250	140	467	790	1117	63	370	688	972	
RPM—TEXA **	993	1005	1019	1041	1289	1306	1324	1345	1481	1500	1520	1542	1670	1692	1714	1733	40,000 N
Dyno (Nm) ***	195	565	920	1280	208	545	880	1250	163	480	790	1125	74	389	700	965	
RPM—TEXA **	996	1015	1036	1061	1292	1320	1344	1375	1490	1514	1541	1570	1676	1707	1733	1760	20,000 N (warm tires) *****
Dyno (Nm) ***	192	583	918	1315	214	559	908	1279	160	480	810	1130	80	405	710	970	

* The test was carried out using the constant revs mode of the dyno. ** Reading made using OBD TEXA TXT diagnoscope with IDC5—Truck software. The correctness of measurements using TEXA was validated on an engine dynamometer. *** Braking force value on a single brake, reading made using dyno software. **** The load on the driving axle of the vehicle was generated by the load adding system with the use of an electric winch, connected with a strain sensor, and the system of mounting to the driving axle of the tested truck—the axle pressure system is shown in Figure 8. The axle load was read using the dyno software. ***** Test started at 70 °C.

(**a**) (**b**)

Figure 8. Vehicle axle load mechanism: (**a**) general view, drive axle slings, (**b**) view of rope linked with an electric winch (authors' source).

Figure 9 summarizes the above data from Table 6 in the values of sliping wheels on rollers for different engine speeds in the context of the engine load and driving axle load (pressure).

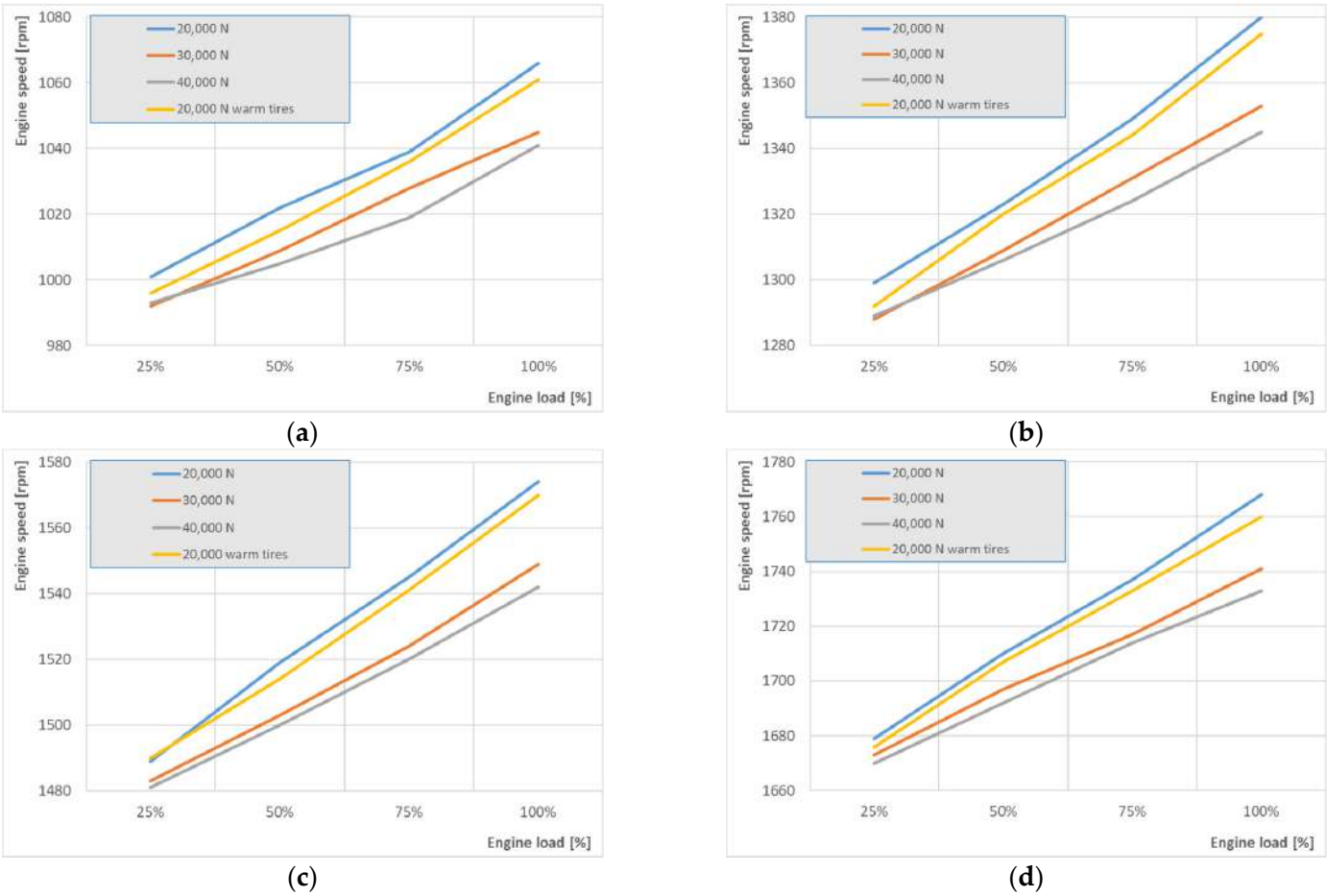


Figure 9. Graphs of differences in engine speed for rollers where n is constant, resulting from wheel slip on rollers for different engine speeds in the context of engine load and drive axle load: (**a**) 1000 rpm, (**b**) 1300 rpm, (**c**) 1500 rpm, (**d**) 1700 rpm (authors' source).

The above charts show that the increase in slip correlates with the increase in engine load (which results from increasing the torque on the wheels of the vehicle [10]). As the engine load increases, the measurement error resulting from the slipping of the wheels on the roller increases, which is also shown in Figure 10. Due to the fact that there is a dynamic difference in the geometric value of the tire diameter [17], all the parameters that affect the tire slip also have an impact on it (axle load, wheel speed, tire temperature and so on). The change in the tire diameter also leads to a negative error at low engine loads [15]. According to [17], measurements on the tire imply that for a high load and low inflation pressure, the deformation in the shoulder area will contribute more to rolling resistance. In an opposite manner, for a low load and high inflation pressure, the deformation in the crown area will contribute more. Meanwhile, the deformation in the shoulder area always contributes more to the rolling resistance for the worn tire. This will in turn accelerate the wear process on the shoulder area. It should also be noted that the change in the dynamic diameter of the tire resulting from the centrifugal force acting on the turning wheel in the context of the thrust of the drive axle generated on the dynamometer is marginal and oscillates around the value of 0.5–1 mm at 90 km/h [18]. The influence of temperature can be observed by analyzing the blue and yellow lines for a load of 20,000 N. The influence of the axle load is more important than the temperature. Considering the necessity of maintaining low tire temperatures (optimally less than 80 °C and no more than 120 °C [19]) while tuning the gas system, in order to prevent their increased wear, it seems reasonable to increase the axle load to 30,000 N while monitoring the temperature. Tire temperature values at the end of each test from all measurement setups are summarized in Figure 10. The graph can

be shifted in the vertical axis by modifying the engine/roller ratio in the dynamometer settings. In the context of the data presented above, it can be concluded that it seems reasonable to modify the engine/roller ratio depending on the range of engine loads and thus work in different ranges for the fuel exchange maps in the gas ECU [3]. Setting a constant gear ratio will generate measurement errors and may lead to discrepancies in the readings of rpm values in the dyno and the gas controller.

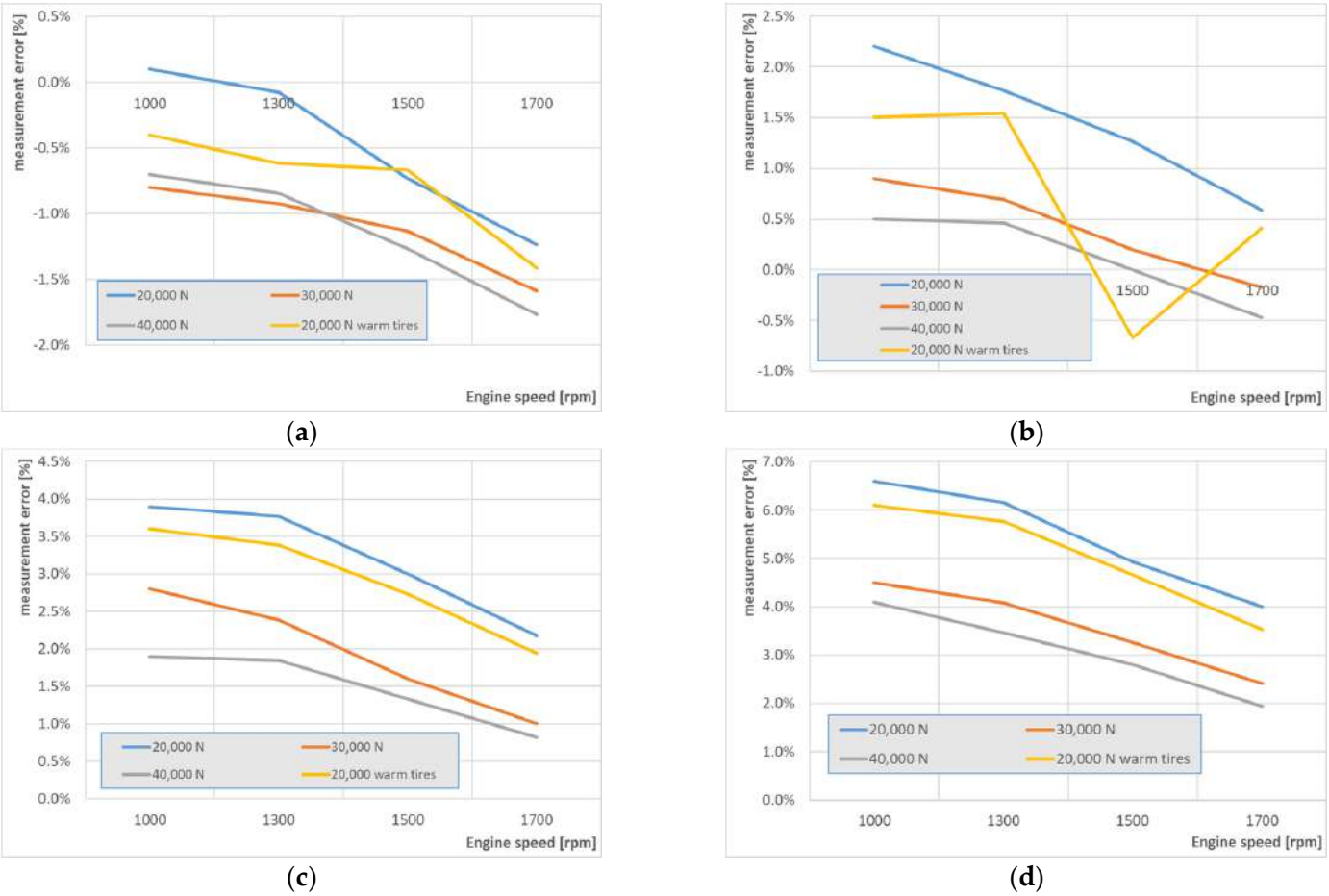


Figure 10. Graphs of errors resulting from the tire slipping on the roll for different rpm depend on engine load: (**a**) 25%, (**b**) 50%, (**c**) 75%, (**d**) 100% (authors' source).

In Figure 11, in the upper right corner of each photo on parameter described as "maks", the highest temperature recorded by the thermal camera during test is shown. It can be noticed that it was always the temperature of the inner wheel near the inside of the tire tread. Figure 11 above shows that it is worth keeping low tire temperature with higher axle pressure during tuning in order to extend the service life of the vehicle tires. Axle load 30,000 N in the case of the tested vehicle seems to be the most appropriate. A possible solution to this problem is adding a ventilator for extra tire cooling.

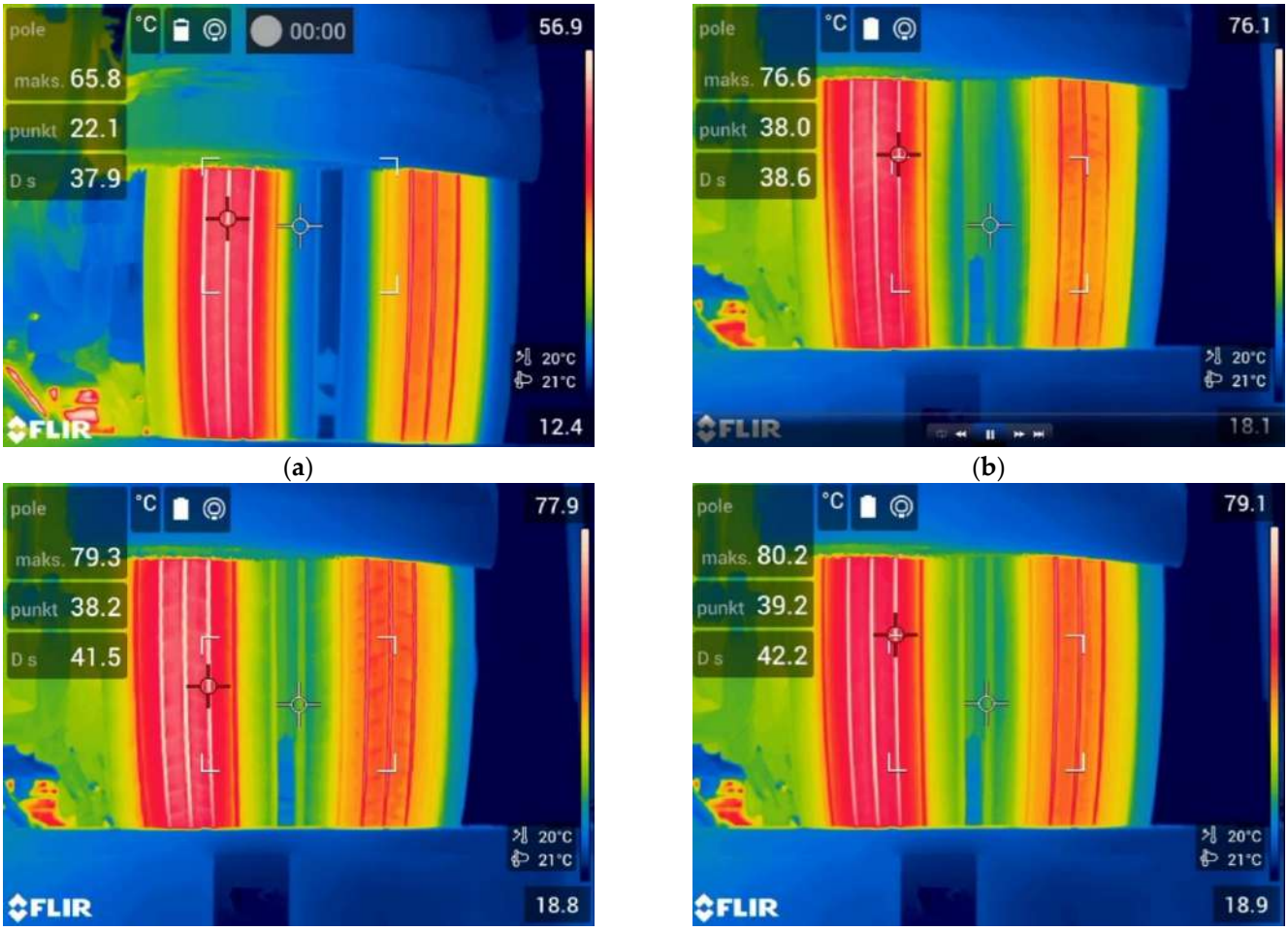


Figure 11. Tire temperature at the end of each test for axle load: (**a**) 20,000 N (**b**) 30,000 N (**c**) 40,000 N (**d**) 20,000 (warm tires) (authors' source).

3.4. Selection of the Optimal Operating Mode of the Chassis Dynamometer

The tuning of dual-fuel installation with the use of a chassis dynamometer in places with partial engine load can be performed in fuel exchange map spots, using the function of maintaining constant revolutions of the dynamometer [20,21]. However, in the case of tuning the external characteristics of the engine (maximum dynamic torque values), it is necessary to use the dyno ramp mode, which allows the vehicle to be braked in the entire range of rotational speeds with an appropriately defined acceleration. The selection of the appropriate acceleration of the vehicle engine on the dynamometer device is of decisive importance in the creation of the boost pressure, and thus the possibility of achieving the maximum value of the torque [7]. In the case of semi- truck, it seems important to generate a sufficiently high braking load on the rollers on the dynamometer, because the semi-truck has a diesel engine with a turbocharger that has a high inertia, which requires a high load to achieve full spool [7,21].

During this test, the highest value of dn/dt as rpm/sec at which it was possible to correctly determine the external characteristics of the engine (creating highest boost pressure) was experimentally determined. Results are shown in Figure 12 below.

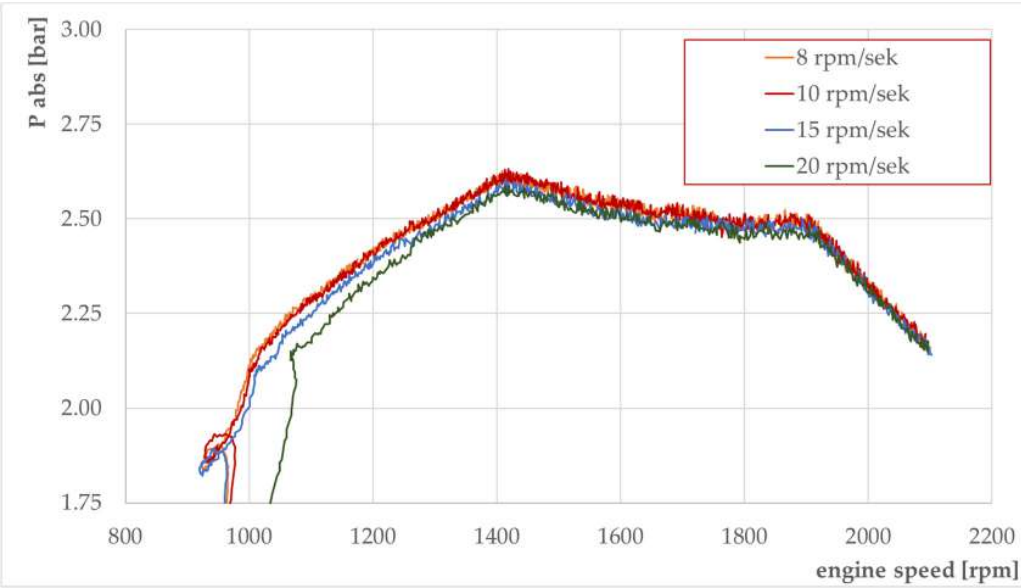


Figure 12. Influence of the coasting-down speed for the test of the external characteristics (maximum torque values) on the boost pressure (authors' source).

The chart above shows that 10 rpm/s meets our boost pressure requirement. Braking at lower acceleration values unnecessarily lengthens the test, which in turn stresses the tires thermally and may lead to more intensive wear. The effects of overheated tires in the form of tearing off parts of the tire tread are shown in Figure 13.

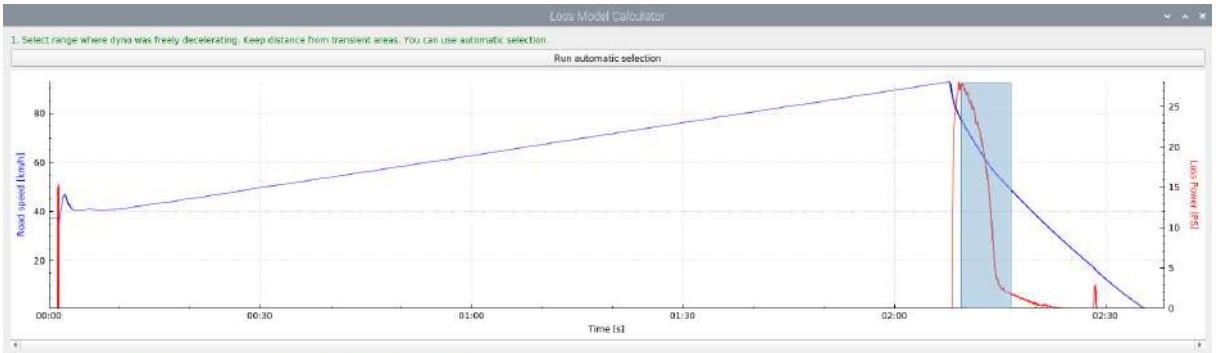
(**a**) (**b**)

Figure 13. Damaged tire due to overheating: (**a**) view of the tire, (**b**) view of the broken part of the tread (authors' source).

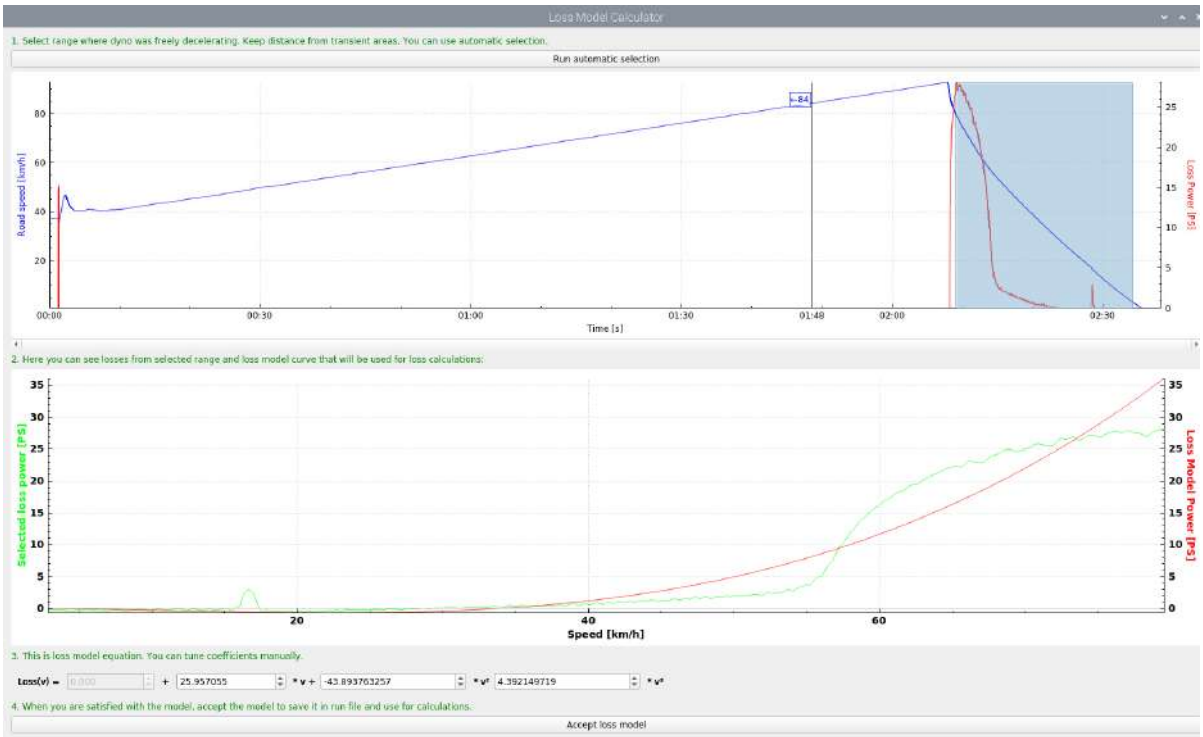
3.5. The Impact of the Correctness of the Selection of the Scope of the Analysis of Data on Losses in the Drive System and Repeatability of Tests

As in the case of the tested vehicle, in which an automated gearbox was used, also in other semi-truck, there may be a lack of repeatability in the calculation of the model of losses in the drive system of the vehicle implemented through the free coast of the vehicle drive on the rollers of the dynamometer [1,14]. During the tuning process, in the case of the chart of the maximum performance or the one set for the optimal revolutions, the incorrectly calculated loss model may generate errors in reading the power and torque in dual fuel operation [11]. After the measurement is completed, the automatic gearbox may

generate fluctuations in the recorded resistances of the vehicle drive. A similar situation does not take place when the brake is pressed in a car with a manual gearbox. At this point, it is worth considering the repeatability of the selection of the measuring range for the calculation of losses. Whether the operation of the gearbox is repeatable should be noted, as well as whether the scope of the analysis of data for the loss model is appropriately wide and whether it does not include places evidently bending the measurement results. Figure 14 shows a screen from the dyno software, and calculation of the loss model in the drive system.



(**a**)



(**b**)

Figure 14. Screen from the dyno software, calculation of the loss model in the drive system: (**a**) nominal settings, (**b**) manually selected range of the loss model, and comparison of the repeatability of the automatic vehicle gearbox operation (authors' source).

It is worth remembering about the repeatability of the tests in the case of compiling data before and after conversion to dual fuel power. The key in this aspect is the possible reproduction of the test conditions before and after this conversion. Figure 15 shows the result of the repeatability of the tests for the graph of the boost pressure related to the operating conditions of the fully warmed up and not fully warmed up engine. These types

of errors affect a number of elements of the installation (e.g., the temperature of the gas pressure reducer, the possibility of creating the boost pressure).

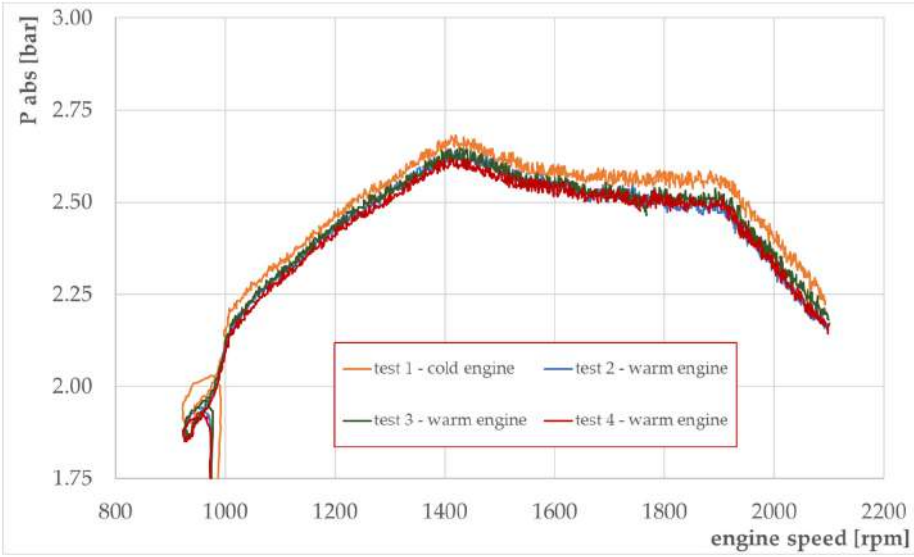


Figure 15. Graph of the repeatability of the boost pressure build-up due to the measurement conditions (engine warm-up) (authors' source).

4. Discussion

All the operating parameters of the chassis dynamometer mentioned in this document affect the assessment and tuning effects of dual-fuel systems in semi-trucks. As part of this study, the aim was to present in general terms the key parameters of the chassis dynamometer that affect the final effect of measurements and tuning for a dual-fuel vehicle. As a result of negligence resulting from the lack of awareness of the above-mentioned critical elements of work with the chassis dynamometer during the tuning of the installation and comparing the effects of this tuning with the single-fuel engine supply, it is possible to present the results of power measurements of the operating parameters for the external characteristics of the engine in combination with a correctly measured and adjusted installation. There is such a statement in Figure 16.

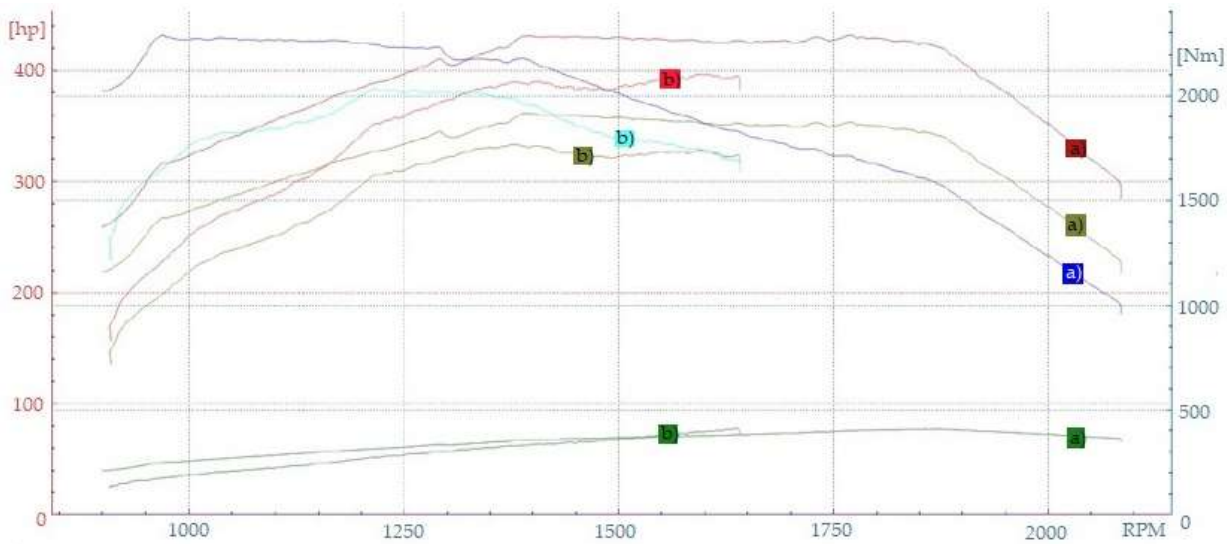


Figure 16. *Cont.*

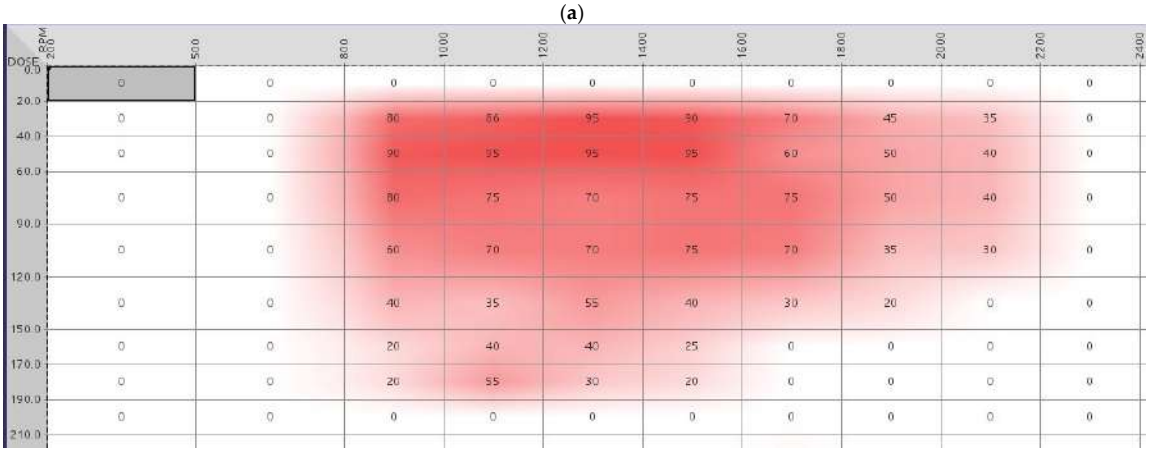
	Loss power	Wheel power	Engine power	Engine torque	Length	Speed source	Engine/roller ratio	Corr. Value	Amb. Press.	Amb. Hum.	Peak. Eng. speed
a)	77 hp 1874 rpm	361 hp 1389 rpm	432 hp 1770 rpm	2290 Nm 970 rpm	160s	calculated	1.36	0.6%	1014 hPa	23%	2085 rpm
b)	77 hp 1636 rpm	333 hp 1376 rpm	397 hp 1593 rpm	2034 Nm 1216 rpm	113s	calculated	1.08	-1.5%	1024 hPa	16%	1641 rpm

Figure 16. A graph showing the results of external characteristics for the research where: (**a**) attempts were made to reduce the impact of errors, and (**b**) the errors mentioned in this work are combined (authors' source).

The above chart shows the wrongly selected measurement course and the influence of other factors mentioned in this paper, which resulted in the torque and power values lower by 8.2% and 11.2%.

The scale of problems resulting in inaccurate tuning of the dual-fuel system is presented in the map of diesel fuel replacement with gaseous fuel. The higher its value in a given field and the higher the total value of the exchange (while maintaining torque and power waveforms similar to single-fuel operation), the better. Figure 17 below shows the comparison of the results of tuning the map of conversion of diesel fuel to gas fuel. In the example below you can clearly see much worse fuel exchange values, and the overall exchange value decreased by 9.2%.

(**a**)



(**b**)

Figure 17. Comparison of the results of tuning the map of conversion of diesel fuel to gas fuel in terms of: (**a**) attempts were made to reduce the impact of errors, and (**b**) where the errors listed in the paper are combined (authors' source).

5. Conclusions

This document presents the influence of the operating parameters of the chassis dynamometer on the evaluation of the tuning of dual-fuel systems. The resulting map of the exchange and the tuning itself are of course influenced by many other elements, such as the condition of the car (e.g., state of filters [22]) or the correct calibration of the dynamometer itself. It is also worth mentioning that the dynamometer presented in the test cannot be used for standardized tests and the effect of the work related to it should always be a comparative effect in which we have an input and final test, then compare both. However, it should be remembered that some of the errors can be effectively eliminated, and this article was also supposed to present this. It is also worth mentioning that the dynamometer presented in the test cannot be used for standardized tests. The effect of work on it should always be a comparative effect in which the received first input and final output are compared with each other. In the context of tuning, this can already give correspondingly good results. Nevertheless, in order to be closer to reality, it seems important to eliminate or minimize the errors described in this work. In the opinion of the authors, further tests in the research system presented in this paper should focus on a more in-depth analysis of the problems presented in this paper. It also seems justified to test in the context of exhaust emissions on the test system presented in this paper, compared to work under standardized engine dynamometer conditions [23–25]. The authors of this work are considering taking up such a topic in the future.

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The concept of a mobile automated diagnostic and dynamometer station for heavy trucks

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Trucks are a key element that performs transport functions in many sectors of human activity. This makes ensuring their proper maintenance and performance critical. One of the solutions to this problem is the concept of a mobile automated testing and testing station. A mobile diagnostic and dynometric station based on a 6-foot container was built at the Military University of Technology in Warsaw. This bench is a self-contained unit that can be transported to various locations and carry out on-site testing and diagnostics of heavy trucks. The station is equipped with a fully open system and software that can accurately measure the performance and efficiency of the truck's engine and driveline and enforce specific automated diagnostic processes analogous to quality control tests during the production process. This information can then be used to make informed maintenance and repair decisions, helping to minimize downtime and increase overall vehicle life. The mobile diagnostic and dynamic station also provides a convenient and cost-effective alternative to traditional off-site testing methods, making it a valuable tool for transport companies and fleet managers.

Key words: *mobile dynamometer, automated tests, trucks, dyno*

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1. Introduction

A chassis dynamometer is a device most often used to measure the power and torque of a vehicle engine [19]. The measurement is carried out indirectly by recording the torque on the rollers of the chassis dynamometer. Referring the value of this parameter to the rotational speed of the rollers, and then, via the drive system, to the engine crankshaft drive speed, allows the vehicle power to be calculated (**F**ig. 1). Due to the specificity of their work, trucks generate very high values of torque on the wheels, which is necessary to move heavy loads while maintaining appropriate dynamics and in changing road conditions (e.g. changes in ground elevation or wind speed and direction). Forcing the engine to run under extreme conditions during no-load road tests is impossible or very difficult [16]. Therefore, a reasonable solution is to use a chassis dynamometer, which allows forcing appropriate working conditions of the drive system and maintaining them for a certain period of time.

Fig. 1. View of a truck on a chassis dynamometer [16]

Thus, the work of the vehicle on the dynamometer can perform basic tasks such as [19]:

- measurement of power and torque
- determination of the actual resistance to motion of the vehicle drive system.

However, as additional tasks, we can distinguish the ones that allow to simulate real load conditions that occur on the wheels of the vehicle while driving and the ability to simulate various load conditions and maintain certain conditions for a longer time (constant rotation, constant load) extending the diagnostic capabilities of the tested vehicle. This opens doors to non-standard applications [24].

Small mobile chassis dynamometers and mobile dynamometers loaded from the power take-off shaft, used in the heavy industry and agriculture, have gained popularity on the market mainly among tuning companies that offer modifications and tuning of vehicles [13, 26]. Thanks to the measurements on the dynamometer, it is possible to accurately determine the actual performance of the vehicle before and after making the changes, which allows the tuning to be adjusted to the individual needs of the customer and is extremely helpful in diagnostics. Dynamometers are also used in scientific research where they are used to test new materials, components and drive systems, and to study the impact of various factors on the efficiency and fuel consumption of vehicles, and their mobility allows to achieve vehicle operating conditions as close as possible to those in use In the case of heavy trucks, apart from hub solutions, and one wheeled application, the market of mobile dynamometers is very narrow. This was the motivation to tackle this topic.

2. Characteristics and functionality

2.1. Diagnostic possibilities

Taking into account the diagnostic possibilities, possible applications for the discussed device/dynamometer can be selected, i.e.:

- checking the vehicle before shutting down or putting it into operation (after an accident, damage or repair) [9]

- evaluation of the vehicle's drivetrain wear level (e.g. torque and power analysis) [9]
- tuning (increasing power, modifications to the exhaust gas treatment system, mechanical modifications, eco tuning) [12]
- calibration of non-factory fuel supply systems (LPG, CNG, LNG, hydrogen) [15]
- measurements of emissions of exhaust gas components, fuel consumption (using an exhaust gas analyzer) [19, 20]
- the possibility of using automated procedures for assessing the technical condition of the vehicle using stations and dedicated OBD2 interfaces and tests implemented in them.

2.2. The mobility of the dynamometer

Convenience and Cost-effective are the factors where mobility of the station allows for on-site testing, reducing the need for trucks to be taken off the road and transported to a separate location for operation. The mobile station eliminates the costs associated with transporting the truck to an off-site location for testing, saving time and money. Servicing a large fleet of vehicles directly at the customer's machine park - in the case of modification of the fuel supply system or tuning it's a big advantage [2, 11]:

- Convenience and Cost-effective: The mobility of the station allows for on-site testing, reducing the need for trucks to be taken off the road and transported to a separate location for operation. The mobile station eliminates the costs associated with transporting the truck to an off-site location for testing, saving time and money. Servicing a large fleet of vehicles directly at the customer's machine park – in the case of modification of the fuel supply system or tuning it's a big advantage;
- Increased efficiency: By conducting diagnostics and testing on-site, the time required to diagnose and repair a truck is reduced, leading to increased efficiency and decreased downtime. This is of key importance when operating on the battlefield or, for example, during motorsport competitions;
- Improved accuracy: The advanced sensors and software used in the mobile station provide highly accurate measurements of the truck's performance, helping to identify potential problems before they become major issues;
- Increased accessibility: The ability to bring the station to various locations increases accessibility for fleet managers, allowing them to conduct testing and diagnostics even in remote areas;
- Increased safety: By conducting testing and diagnostics on-site, the risks associated with transporting the truck to an off-site location are reduced, improving the overall safety of the vehicles and the people operating them.

3. Construction of a mobile chassis dynamometer for trucks

3.1. Benchmarking of technological solutions

As part of the research, learning and creative use of the best market models at a given moment, only one company that implemented this idea was found. The company Tylor Dyno, from Milwaukee, USA, was the only one in the world to deal with the subject. The company developed a mobile dynamometer for trucks, in a comprehensive style. Company used an 18-meter semi-trailer, a crane on board and a lot of spectacular automation as shown on Fig. 2 [17].

Fig. 2. Render of the mobile chassis dynamometer solution by Taylor Dyno [19]

The design of the mobile truck dynamometer form Tylor Dyno has a structure built on a semi-trailer, which makes it impossible to use this idea in non-road applications with an emphasis on military use. This solution also takes up a lot of space and requires a truck tractor to change the place of operation each time. Unfortunately, the solution found was far from meeting the needs of the designed device, which prevented deeper inspiration from market solutions.

3.2. Design, construction and testing

The design and construction process of a mobile automated diagnostic and dynamometer station for heavy trucks involved several stages [5, 17].

- Requirements gathering: The assumptions of various types of use (the original need to build a dynamometer was related to the project for mass modifications of trucks with dual-fuel CNG/diesel) and the dedicated possibility of military use in bad or unknows terrain meant that the chassis dynamometer designed at MUT had to be redefined compering to Tylor Dyno solution. Due to this fact, the need to build a dynamometer station with the following design assumptions was put forward:
  - loading/unloading using a hydraulic lift which is an integral part of the transporting vehicle
  - provide the possibility of temporarily setting up a mobile diagnostic and dynamometer station at the customer's site for the duration of the assembly
  - the station is to be able to be folded into a standardized container for transport
- Conceptual design: Based on the requirements, a conceptual design of the station is created. This includes the overall layout of the station, the type of equipment to be used, and the overall functionality of the system as shown on the Fig. 3 below.
- Detailed design: In this stage, the conceptual design was refined into a detailed design. This included the design of the mechanical (as shown on Fig. 4 and 5) , electrical, and software systems that make up the station. All model tests were also performed at this stage which is shown on Fig. 5.

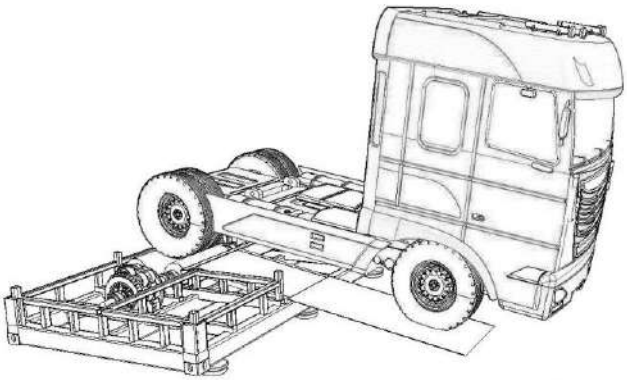

Fig. 3. Concept drawing of a mobile dynamometer for heavy trucks with the vehicle placed

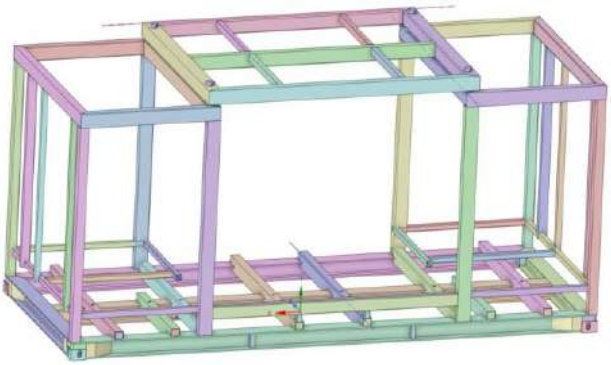

Fig. 4. Technical drawing of the mobile dynamometer support structure

The figure above shows the base of the dynamometer structure, for which a lattice structure with dimensions of a 6-foot sea container was used. In the upper part of the drawing it's notacible a removable part of the roof of the container, enabling diagnosis on the rollers of the dynamometer of higher trucks with bodies.

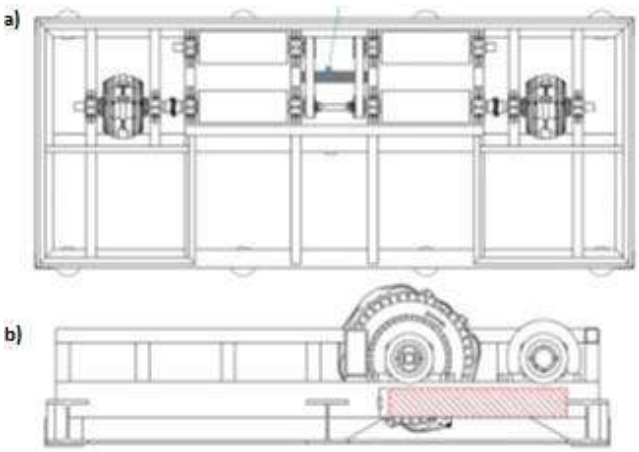

Fig. 5. Technical drawing of the low part of mobile dynamometer with the location of brakes and rollers: a) top view; b) side view

The Figure 5 above shows the low mounting of the roller assembly and eddy current brakes allowed to lower the center of gravity of the machine. The off-center offset of this assembly helped to reduce the angle of attack necessary for the vehicle to climb onto the rollers. This allows vehicles with a small ground clearance to interact with the device and is of great importance, for example, in the context of road tractors or motorsport vehicles. The use of a removable roof and asymmetry in the location of the weight of the device required in-depth research on the stiffness of the load-bearing structure and its loads, which is shown in Fig. 6. This was crucial for maintaining the axiality of the roller-brakes system and the possibility of multiple manipulation (moving) of the device without damage.

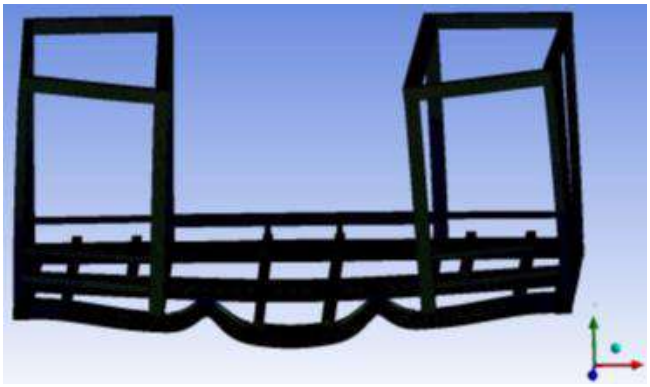

Fig. 6. Visualization of the mechanical load model of the mobile dynamometer structure

In Figure 6, there is a tendency to deformation on the axis perpendicular to the axis of the rollers and breaking deformations due to the removable roof. Engineers and designers had to strengthen the base structure and develop a roof structure that would transfer loads in the closed state (only in this state can the station be moved with a crane).

- Manufacturing and assembly: Once the design was finalized, the station has been manufactured and assembled. This included the fabrication of the mechanical components, the assembly of the electrical components, and the integration of the software. The project was implemented by members of a consortium consisting of the following companies: Dyno Revolt, Bart-Bud with the substantive and conceptual support of MUT. The author of this article was the designer and coordinator of this project.
- Testing and commissioning: The completed station has been thoroughly tested to ensure it meets all requirements. Problems that needed to be solved were also identified and a concept for further development of the station was developed (more on this in point 4 of this article). Validation included performance tests, reliability tests and security tests. The figures below show a general view of the station and the dynamometer station being tested before deployment.

During the tests starting operation of the station as shown on Fig. 7c and Fig. 7d, apart from typical things such as the correct operation of the electrical installation, software or correct operation of mechanical systems, they were also checked vibration levels, long-term operation under load and heat distribution were checked, the repeatability of measurements as well as the measurement systems themselves, i.e. strain gauges and rotational speed sensors.

More on this topic the author of this article wrote as part of the work [8].

Fig. 7. General view of the mobile dynamometer station: a) front view with ramps; b) back view; c) view with the truck on the dyno; d) view of the wheels of the vehicle on the dyno

It should be emphasized that this article is a general aid to the recipient. More details on the construction can be obtained by contacting the author of this publication directly.

3.3. The final results

The station has been built on the construction of a 6-foot container and has a fastening system compliant with ISO 1161, i.e. the same as a standard sea container. This allows the dynamometer to be transported using standardized mountings [18]. The roof of the station is removable, which allows for testing vehicles with built-ups. On the Fig. 8 below, a view of the stations while being transported on a truck chassis.

Fig. 8. The diag- dyno station transported on a truck chassis

The dyno structure can be manipulated using the hydraulic crane provided with the vehicles (Fig. 9). The dynamometer can be transported through difficult terrain and set up and armed for operation in several minutes.

One of the goals of the researchers and designers was to leave the dynamometer architecture open in terms of hardware and software, as well as the possibility of designing proprietary operating modes. Therefore, the dynamometer controller has several connectors for digital and analog signals, using USB Bluetooth or PWM controller support. In addition, the software has a fully configurable and compatible CAN protocol. What's more, measurements can be easily corrected according to DIN, EC, ISO, JIS and SAE standards thanks to the built-in weather station

Fig. 9. The diag- dyno. station during unloading using a hydraulic truck crane

In the Table 1 below, some selected parameters of the dynamometer contoured by MUT are highlighted.

Table 1. Selected parameters of the mobile diagnostic and dynamometer station

Parameter	Value/Name/Type
Operating mode	Load, quasi-dynamic
Type of Brakes	2 × Frenelsa FF16 eddy current 3300 Nm
Max/Min track width	5700/1000 mm
Roller diameter, axle pressure system	320 mm, electric winch with tension reading
Dimensions, weight	6058 mm; 2545 mm; 2870 mm; 5574 kg
Transport system	Standardized 6ft container, ISO 1161 lashing system, handling with HDS
V-max on rollers	200 km/h
Modes of operation	Constant torque, constant speed, dynamic, load, ramp, combined
Temperature tolerance (operation) [storage]	–10 to +30 [–40 to +50] °C
Hardware driver	OEM DynoRevolt on Raspberry Pi 4, OBD2 compatible, HDMI connector, 4 × USB 3.0, 4 × HP analog input, 4 × analog input, 3 × low side PWM
Software	OS Debian Linux 5.4.14, OEM dyno software DynoRevolt "Dyno2 RPI 2.17.0" fully configurable, support for CAN, analog, digital PWM, bluetooth, USB and OBD signals, Measurements with corrections according to DIN 70020, EC 95-1, ISO 1585, JIS D1001, SAE J1349

The diagnostic and dynamometer station measures typical dynamometer values such as power, torque and all derivatives resulting from the possibility of programming your own test runs. It is worth noting that the software allows to read up to 166 real values of engine accessories using the vehicle's OBD connector (using a chip, e.g. ELM 327, STN1110). Leaving the system open allows more complex tests to be performed. Table 2 below summarizes selected values can be measured and those that may be

involved in the work of the dynamometer and the implementation of special tests on it.

Table 2. Selected measured, represented and involved in testing values used by discussed station

Parameter	Value/Name/Type
Engine speed [rpm]	Calculated engine speed
RPM acceleration motor speed [rpm/s]	Engine speed delta over time
Road speed [km/h]	Calculated linear "running" speed of the vehicle
Power loss [HP]	total power lost due to speed changes and load changes
Inertia power of the drive system [HP]	Inertia force of the drive system
Power on wheels [HP]	Calculated power at the wheels of the vehicle
Motor moment of inertia [Nm]	Calculated engine moment of inertia
Engine power [HP]	calculated engine power
Motor torque [Nm]	measured and calculated engine torque
Cor. engine power [HP]	corrected engine power
Ambient pressure [hPa], Ambient temperature [°C], Ambient humidity [%]	measured by the dynamometer's built-in weather station system
Cooperation with the vehicle's CAN line	Support for up to 166 actual values of engine accessories using the OBD connector of the vehicle (chip ELM 327, STN1110)

The device in this form has been put into operation, but solutions are being developed at the moment to extend its capabilities by carrying out automated diagnostic tests under load.

4. Automated tests CAN-dynamometer

To explain the possibilities offered by the open architecture of the dynamometer let's use an analogy to the possibilities integrated into the universal electronic external diagnostic system Bosch ESI [tronic] 2.0 KTS Truck [4]. Similar possibilities can also be found in other commercial solutions such as Texa or Delphi diagnostics solution [25]. At ESI [tronic] it's possible to perform many tests that can directly or indirectly tell about the condition of individual components of the engine and drive system. These tests enforce executive functions in the engine controller for the appropriate engine components in order to check their correct functionality [26]. For example, in the software KTS can "provoke" a test regarding:

- examination of the engine using a high-pressure test
- cylinder deactivation test
- compression test
- Adblue dosing test.

Many more are implemented

Let's take an example of a compression test for wallpaper. The current values read from the starter during its operation (without fuel doing which is forced by test algorithm) may indicate discrepancies as to the compression pressure in individual cylinders, this gives us a picture of what can happen to the engine without time-consuming disassembly or classic tests with a manometer.

In a similar way, we can carry out tests under load using the dynamometer's capabilities [10, 14]. The dynamometer can maintain the operating parameters of the drive system in a given state while performing a test defined by us or sewn into its code. What's more, these tests can be blocked into entire test suites. Ultimately, it gives (similar to production control tests) comprehensive, quick knowledge about the actual condition of the vehicle and its drivetrain components. Diagnostics and non-destructive testing play a key role in keeping vehicles operational. They allow you to identify potential technical problems before they become serious problems, which can lead to vehicle failure and danger. Non-destructive testing allows you to detect damage and defects such as cracks, deformation or wear without having to dismantle or destroy vehicle components. Such tests include visual, measurement, ultrasonic, magnetic, penetrant and radiographic tests. In the context of using a dynamometer, testing of wheeled vehicles on them also fits into this definition.

The use of automated CAN-dynamometer tests using advanced technologies such as artificial intelligence, machine learning, data analysis and vision systems translates into more precise and effective diagnostics, which in turn leads to cost reduction or determines the success of a military mission or winning a race. Such investigations can detect potential technical problems and make repairs in time before they become serious safety hazards. In this way, damaged vehicles can also be deployed after repair with great efficiency.

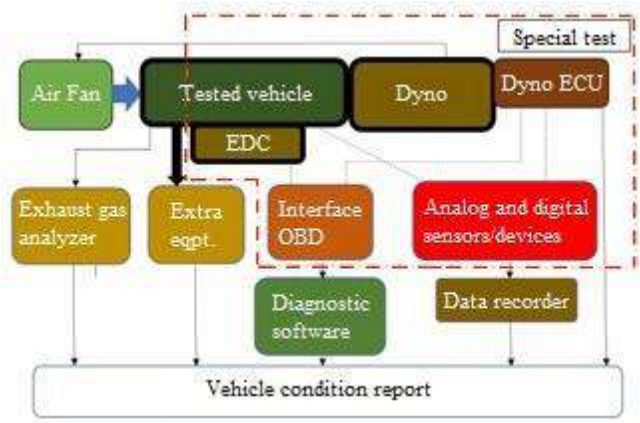

Fig. 11. A block set of elements necessary to perform forced special tests on a mobile chassis dynamometer

The picture above shows the test blocks algorithm. Real-world testing for heavy-duty vehicles may include checking the correct operation under load of components such as:

- the state of wear of individual sets of cylinder-piston rings
- the state of wear of the turbocharger
- the correctness of increasing the boost pressure
- proper operation of the wastegate valve
- correct operation of the engine brake and retarder
- testing the inertia of the drive system
- different tests including exhaust gases
- NVH test and durability can be selected depending on the application.

And much more in pre-programmed dedicated research systems. Tests can be fully automated and end with a ready

report on the status of individual elements. This allows for a quick and automated assessment of vehicle wear or damage, as well as a dynamic assessment of the vehicle's ability to return to service after a breakdown or collision.

5. Discussion

The concept of a mobile automated diagnostic and dynamometer station for heavy trucks built in MUT has great potential for revolutionizing the way in which heavy trucks are tested and calibrated.

One of the limitations of the device is the requirement for an open ECU architecture in order to achieve its full research capabilities. This means that the device and the special tests algorithm may not be suitable for use with all heavy trucks, as some with factory ECU closed setup may not be suitable. However, this limitation is not insurmountable, as open architecture is a typical configuration of vehicles modified in the context of e.g. motorsport [7].

In the case of military cars, the specification of the car delivered to the recipient is a matter of the needs of the army and related tenders [6], in which the exact parameters and functionalities of the vehicle are specified. If the military notices the potential of the dynamometer, adapting the vehicles and getting a new engine controller with an open architecture seems possible [7, 21].

However this device is capable of providing a wide range of special vehicle-dynamometer tests, and can be used for full-scale auto-calibration of external power supply systems, such as LPG or CNG. The use of the mobile automated diagnostic and dynamometer station has the potential to significantly reduce the costs related to modifying vehicles with auxiliary fuel supply. In today's applications, the auto-calibration process is able to adjust the system settings only for low loads at engine idling speed [22]. Full-scale auto-calibration, which is necessary to modify the vehicle with auxiliary power, is a time-consuming process that requires people with appropriate knowledge. However, the use of the station and the algorithms embedded in it may result in a reduction of costs related to modification by several percent.

6. Conclusion

The mobile automated diagnostic and dynamometer station for heavy trucks built in MUT is a highly innovative device that has the potential to revolutionize the way in which heavy trucks are tested and calibrated. Although there are limitations to its use, the device can still be used for a wide range of special vehicle-dynamometer tests and full-scale auto-calibration of external power supply systems. The potential cost savings associated with its use make it an attractive option for those involved in modifying heavy trucks with auxiliary fuel supply [1, 25]. Overall, the device represents a significant step forward in the field of heavy truck testing and calibration.the diagnostic and braking station allows for handling external signals, including data from CAN, and has the ability to program special tests; The above makes it possible to carry out specific automated diagnostic processes analogous to quality control tests during the production process. Research on the topic is ongoing.

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Nomenclature

CAN	controller area network
CNG	compressed natural gas
DIN	Deutsches Institut für Normung e.V. (DIN; in English, the German Institute for Standardisation Registered Association)
EC	Directive 95/1/EC of the European Parliament and of the Council
EDC	electronic diesel control
HDS	hydraulic truck mounted crane
ISO	International Organization for Standardization
JIS	Japanese Industrial Standards
LPG	liquified petrolum gas
MUT	Military University of Technology
NVH	noise, vibration, and harshness
OBD	on-board diagnostics
PWM	pulse-width modulation
SAE	Society of Automotive Engineers
UGB	university research grant
USB	universal serial bus

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Dual-fuel engines using hydrogen-enriched fuels as an ecological source of energy for transport, industry and power engineering

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Displacing internal combustion engines (ICE) from the passenger car sector does not mean displacing it from all industries and specific applications. Thanks to the analysis of data on compression ignition (CI) engines used in the world, it is possible to prepare ready-made solutions for the most common engines in selected industries or for those whose greenhouse gas emissions will be the largest and most expensive for their owners in the coming years. The basic solution presented in this article gives the possibility of powering the engines with the most ecological currently known alternative engine fuels and using the already existing methane transmission infrastructure around the world. Their greatest advantage is their availability and low carbon content, which allows to minimize carbon dioxide emissions, both by burning hydrogen-enriched fuels and by increasing the efficiency of the engines modified by dual fuel supply system. Properly made external dual-fuel installation allows to improve the thermal efficiency of the CI engine. Work on this issue may help in the development of, for example, high-efficiency flex fuel power generators, which, as the current situation in Ukraine shows, are worthy. Thanks to the diversification of power sources for power generators, the countries is able to increase the reliability and security of energy supplies even in difficult conditions, such as armed conflict or natural disasters.

Key words: *dual-fuel, alternative fuels, hydrogen-enriched fuels, industrial engine*

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1. Introduction

1.1. Global CO_2 emissions and energy sources in the world

Improving the protection of the natural environment, reducing the emission of harmful exhaust components and greenhouse gases is a path of development that is of interest to the general public. The idea of using the current technological achievements in the form of Dual Fuel (DF) and Reactivity controlled compression ignition (RCCI) engines proposed in this article, together with the use of ecological fuels blended with hydrogen, is part of the trend of reducing CO_2 emissions into the atmosphere, because it covers all the most emission-intensive economic sectors, which have been marked in the Fig. 1.

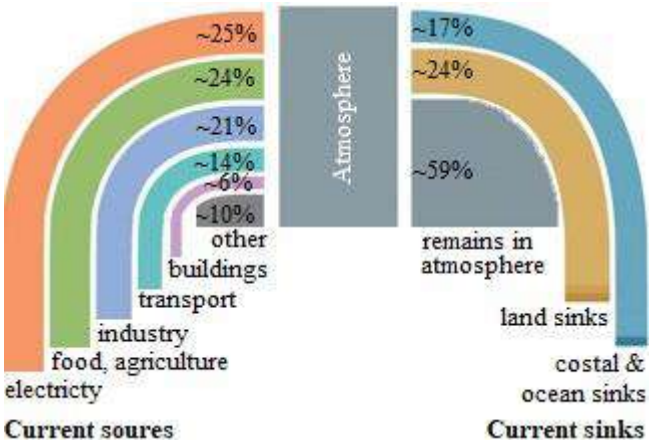

Fig. 1. Sources of CO_2 emissions to the atmosphere in the world [52]

The largest CO_2 emitters use mainly natural gas (NG), coal and oil to generate energy (Fig. 2).

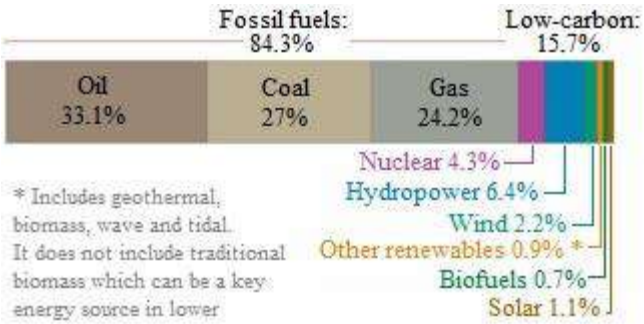

Fig. 2. Total world energy mix (electricity, transport & heat) [3, 52]

The idea presented in this article, which will make it possible to replace them with hydrogen in internal combustion engines, can reduce the consumption of two of these fuels.

1.2. Worldwide use of ICE in industrial applications

The use of combustion engines in power systems plays a key role in maintaining system continuity in various emergency situations, including armed conflicts, natural disasters and weather anomalies. These engines serve as backup power sources and ensure reliable electricity generation in the event of failure of the basic power infrastructure and will important in stabilizing electricity grids with a large share of uncontrollable renewable energy sources [9, 19].

In times of armed conflict or war, the electrical network may become the target of disruption or destruction. In such scenarios, internal combustion engines provide power con-

tinuity. These engines can be integrated with stand-alone power generators or connected to the grid through synchronized operation with existing power plants [9, 19].

In the event of natural disasters such as earthquakes, hurricanes or floods, the electrical infrastructure is exposed to severe damage, leading to power outages. Combustion engines, equipped with suitable generators, can be quickly deployed to affected areas to restore electricity. Their mobility and flexibility allow them to be stationed near critical facilities such as hospitals, emergency response centers or communication networks, ensuring that essential services continue to function [9, 19].

Weather anomalies such as extreme heat waves, periods of frost or storms can also stress the power grid and potentially cause blackouts. During such events, internal combustion engines may be run as peak power plants to meet the increased demand. These engines can quickly increase energy production and stabilize the grid, compensating for fluctuating renewable energy sources or reduced efficiency of conventional power plants affected by extreme conditions. Figure 3 shows the process of stabilizing the power system in New England USA at the beginning of 2023 [10, 51, 53].

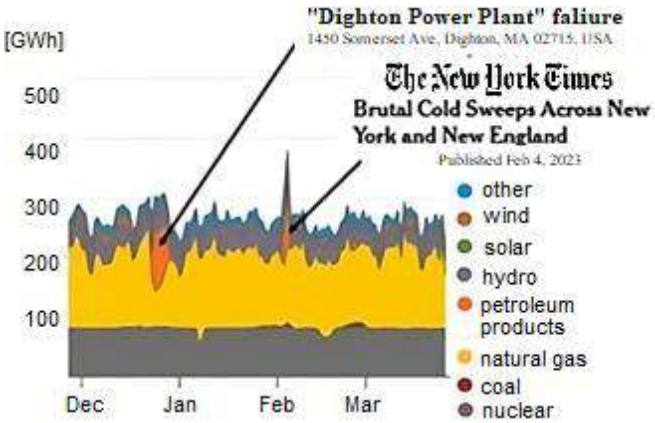

Fig. 3. Graph of the use of electrical energy sources for New England, USA in the period from 12.2022 to 4.2023, with marked anomalies where it was necessary to use emergency generators using ICE [10, 51]

The chart above showing the daily production of electricity by energy source shows two crises in which internal combustion engines were used as an intervention, which in the event of a failure of the Dighton natural gas power plant, and later during severe frosts in New England, over 5 GW of power installed in power generators was launched [51, 53].

In addition, the availability of fuel storage facilities for internal combustion engines allows them to operate independently of external power sources for extended periods of time. This self-sufficiency becomes invaluable in situations where the primary power supply is cut off for various reasons, such as damaged transmission lines or disruptions in fuel supplies. To increase the resilience of power systems in emergency situations, it is essential to have a well-maintained fleet of ICE with regular fuel supply and maintenance procedures. Proper training and preparedness are critical to the successful deployment and operation of these engines in emergency situations. However, it should be noted that while internal combustion engines provide a reliable backup power solution, they come with emissions and environmental concerns. To mitigate these problems, efforts must be made to minimize their use and shift to cleaner and more sustainable alternatives such as renewables, energy storage systems and microgrids. Nevertheless, in the context of maintaining the continuity of the system in emergency situations, internal combustion engines continue to play an important role until more sustainable solutions become commonplace and available.

In total, there are 34,937 electricity generation installations installed in the world. Diesel oil and derived fuels are used in 3626 power plants with a total installed capacity of approximately 250 GW, including approximately 800 of these installations also running on natural gas, and additionally 3697 power plants with natural gas turbine engines with a capacity of max. 1.2 TW. For comparison, 127 GW of capacity is installed in pumped storage power plants around the world [11, 12, 56]. This shows how important the internal combustion engines play in the power system.

As far as industry and heavy industry are concerned, the authors of this article are currently unable to find data on the global use of internal combustion engines, however, legislative changes in the United States (US) and European Union (EU) will lead to a reduction in CO_2 emissions from these engines in the near future and will force entities using them to disclosure.

1.3. Worldwide use of ICE in transport

Combustion engines also continue to play a key role in transport. There are currently around 350 million [16, 55] diesel engines in goods transport vehicles. There are a total of 450 million cars with such engines on land, and about 50,000 heavy transport ships (over 1000 tons of load capacity) and a total of over 2 million various types of ships and ships powered by diesel engines [32]. In 2018, 8×10^{18} Joules of energy contained in the fuel were needed to power marine transport with diesel fuels and its derivatives, and in total it is already 10×10^{18} joules of energy when adding this value with the demand for natural gas as shown in Fig. 4 [21, 32].

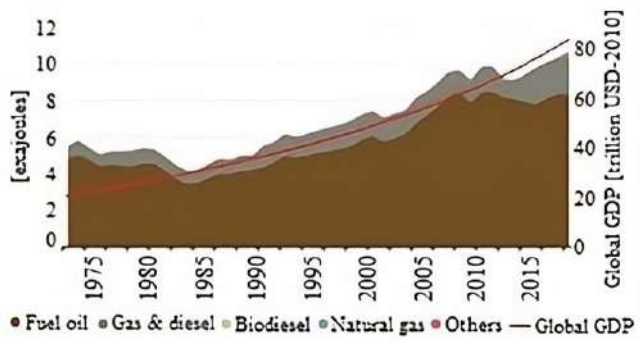

Fig. 4. Estimated energy demand contained in water transport fuel in the world [21]

In the case of road vehicles, the possibility of reducing carbon dioxide emissions is even higher than in the case of energy, industry or water transport, due to the possibility of additionally increasing the efficiency of the engine operating on two fuels and reducing the emission of harmful ex-

haust components. Carbon dioxide emissions in road transport are more than 2.5 times higher than in sea transport and account for nearly 30% of all carbon dioxide emissions from transport [40]. Despite legislative efforts and the hard work of constructors and engineers building motor vehicles, due to, among others, consumption, they did not ultimately contribute to drastic decreases in emissions, as shown in the Fig. 5. Consider here is the best-executed scenario of actions, i.e. the effect of reforms in the EU. In the European Union, the trend of reducing carbon dioxide emissions is the strongest compared to other countries in the world. As you can see in the graph in Fig. 5, emissions in the last decade began to decrease (very slightly, and the decrease in 2020 was the result of the pandemic), which, however, is not yet the effect that could be achieved by using alternative fuels proposed by our team, which, as mentioned , can significantly exceed the 35% reduction compared to diesel emissions, thanks also to the increase in efficiency of the engine operating in dual-fuel mode [8, 40].

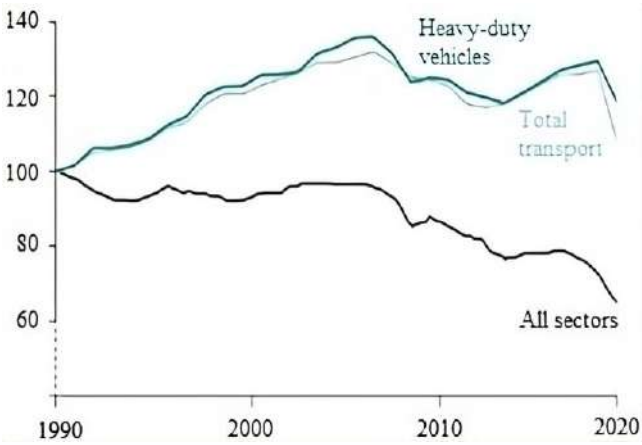

Fig. 5. Relative CO_2 emissions from transport since 1990 in EU [8]

2. Material and methods

2.1. Hydrogen enriched fuels

Hydrogen-enriched fuels may be the key to reducing carbon dioxide emissions generated in the combustion process in internal combustion engines. Their current development has been very intensive for several years, and after the EU decision to limit the sale of combustion cars by 2035 [7, 46], work on low- and zero-emission fuels has accelerated even further. Their production may allow the operation of new cars with ICE powered only by those fuels which total greenhouse gas (GHG) emissions over the entire use cycle will be zero. Many car manufacturers believe that they can determine the future of automotive development, the aviation and maritime industries already treat them as the only source of energy to maintain their transport fleet in the future in the form of zero-emission means of transport [18]. The lack of real alternatives means that industries that need large energy resources invest in this technology in a way that is not only declarative – like the automotive industry – but also fully real, already implementing a huge number of investments related to their use [15, 23, 57]. The existing research methodology still mainly takes into account how many substances the engine actually emits in the place where it is used. In addition to carbon dioxide and water, each of the other substances that affect the natural environment can be neutralized in the exhaust gas catalytic reactor, which is why carbon dioxide, which is the basic greenhouse gas produced in the combustion process, is the focus of reducing emissions in internal combustion engines. The currently used fuels are able to reduce the level of carbon dioxide emissions. Its complete elimination can be done using pure hydrogen or substances binding it with elements other than carbon dioxide, such as ammonia. The use of pure hydrogen or ammonia is currently used, but in a very limited form [47], and it is easier to introduce fuels that can directly replace fuels distributed in the current infrastructure, such as Hydrogen enriched Compressed Natural Gas (HCNG) also in configuration with fully renewable biomethane – replace fossil Compressed Natural Gas (CNG), or Hydro-treated Vegetable Oil (HVO) – replacing diesel, and methanol – replacing gasoline. The possibility of their hydrogenation translates directly into the level of carbon dioxide emissions during their combustion. In addition, both of these fuels come from renewable energy sources such as biogas for HCNG and vegetable oil for HVO. Their total carbon footprint is very low, it can be practically zero when hydrogen is obtained from green energy sources. The methods of obtaining hydrogen are varied and, depending on the energy source used for its production, it may be characterized by a different degree of GHG emissions.

For the purposes of our considerations, green hydrogen should be taken into account, because the EU directs its industry and energy sector to the production of such hydrogen. Green hydrogen is hydrogen whose total carbon footprint does not exceed 2 kg of CO_2 equivalent per kg of hydrogen produced. Hydrogen produced from renewable energy sources has a zero-carbon footprint, and combining it with biogas we get a fuel whose carbon footprint will be practically zero – HCNG is a mixture of natural gas and hydrogen with proportions set by the manufacturer of this fuel, with limitations resulting primarily from the technical capabilities of the system distribution of this fuel. The distribution of natural gas is easier than the distribution of hydrogen and its high content in this fuel causes a number of changes in its properties, making it impossible to distribute it through the natural gas distribution system [42]. However, the hydrogen content can reach up to 50% of the mixture volume, which is a noticeable share of hydrogen in this fuel and allows to reduce carbon dioxide emissions to the atmosphere during its combustion compared to pure natural gas or methane [38].

HVO, on the other hand, reduces carbon dioxide emissions both by reducing the chemical share of carbon in this fuel in favor of hydrogen and by using plant substances in the process of its production, thanks to which practically all carbon dioxide emitted to the atmosphere was previously absorbed by the plants from which this fuel was created [4].

The situation is similar with methanol, which, although it is not an additionally hydrogenated fuel, its production is based primarily on agricultural production from plants that quantitatively absorb most of the carbon dioxide that is emitted in the process of burning this fuel [35].

In another article by the authors of this publication entitled "A review of low-CO_2 emission fuels for a dual-fuel RCCI engine" [25] a list was developed, which selected fuels with the lowest potential for carbon dioxide emissions into the atmosphere, and on its basis, those with the greatest potential to reduce CO_2 emissions from internal combustion engines in the future were selected. It clearly shows that the emission for HCNG is the lowest of all fuels, and its decrease is proportional to the hydrogen content in this fuel. Further opportunities to increase the concentration of hydrogen in the mixture will increase along with the development of distribution technology and the development of hydrogen networks, which will allow for a gradual transition to pure hydrogen in such installations in the future, which will be related to the EU "FIT for 55" assumptions [50]. Table 1 below shows part of the developed table.

However, only two hydrogen-enriched fuels, HCNG and HVO, were selected for the purpose of this work. This is due to the analysis carried out by the authors in a number of their articles [25, 48, 49] in which we presented the concept of using dual-fuel engines in the future, whose overall efficiency exceeds both conventional CI and Spark Ignition (SI) engines, and the flexibility of fuel use is practically unlimited. These engines require both high-octane and high-cetane fuels at the same time, which means that two fuels with diametrically opposed properties must be used. The selection of two fuels with the lowest carbon dioxide emissions allows you to determine how low carbon dioxide emissions will be possible in the future for ICE, the use of which in the power industry and heavy industry will not be phased out in Europe by 2050 [25, 48].

2.2. Principle of operation and advantages of a dual-fuel engine

To ignite the mixture of air and high-octane fuel, a dual-fuel engine uses high-cetane fuel injected directly into the combustion chamber just before its self-ignition, causing further combustion of the entire mixture located in the space above the piston. In a variant of the dual-fuel engine, called the RCCI engine (Reactivity Controlled Compression Ignition), the self-ignition of the high-cetane fuel, in this case the high-reactive fuel, causes the auto-ignition of a homogeneous mixture of the high-octane fuel, called the low-reactive fuel, with air. The RCCI engine is a DF engine variant of the Homogeneous Charge Compression Ignition (HCCI) engine, which is characterized by higher thermal efficiency than a classic dual-fuel engine or a standard CI diesel engine. Figure 6 below shows the view of the combustion process in the chamber operation of the Gasoline, Diesel, HCCI and RCCI dual fuel engine.

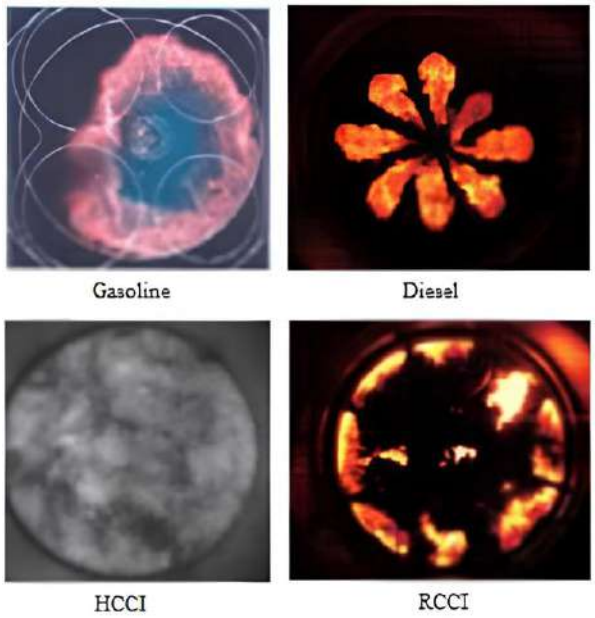

Fig. 6. A view of the combustion process in the chamber during operation of the Gasoline, Diesel, HCCI and RCCI dual fuel engine [22]

In industry, maritime transport and energy, DF engines have been the main source of propulsion for years. This is due to the possibility of using cheaper and less emission-intensive fuels such as CNG, Liquefied Natural Gas (LNG) or bio-CNG/LNG. Their use in SI engines is less economical due to the lower efficiency of these engines compared to dual-fuel engines. Engines on an industrial scale usually work only with necessary breaks, so the higher efficiency of a given solution translates directly into the economy of operation of a given engine and this indicator is crucial in this type of application. DF engines, especially RCCI engines, achieve higher efficiency thanks to a very even temperature distribution in the combustion chamber and a lower maximum combustion temperature. The following Fig. 7 shows an example of the temperature distribution in an RCCI engine and a classic CI engine.

Table 1. Selected fuels properties (full table available at [25])

		Petrol	Diesel	HVO	M100	CNG	HCNG15	HCNG30	HCNG50
C/H ratio	% of weight	9	7.26/6.73	5.49	3	3	2.77	2.52	2.15
	molecular	~3:4	~3:5	0.46	1:4	1:4	0.23/0.24	0.21/0.22	0.18/0.19
Hydrogen weight content [%]		10	12/(12.96)	15.4	12.5	25	26.5 (25.75)	28.39 (26.5)	31.75 (27.5)
Carbon weight content [%]		69–70	69–74	~70	~70	56.1–55.35	55–54.3	53.56–52.85	51.05–50.37
CH_4 emission potential		Low	Low	Low	Low	Very high	High	High	Medium/ high
N_2O emission potential		Very high	Medium	Low	High	Medium	Low	Low	low
Mainly emission potential		HC, CO, CO_2, NO_2, NO_x	HC, CO, CO_2, NO_2, NO_x	HC, CO, NO_2, NO_x	HC, NO_2, NO_x	CH_4, NO_2, NO_x	CH_4, NO_2, NO_x	CH_4, NO_2, NO_x	NO_2, NO_x

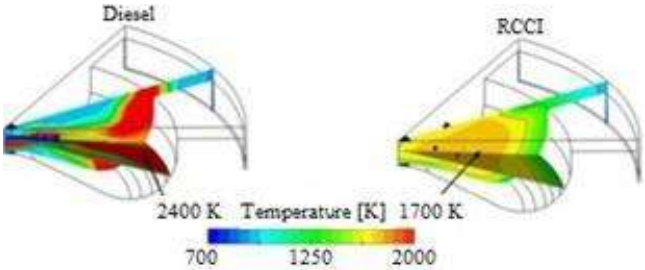

Fig. 7. Temperature distribution in the combustion chamber for diesel only and RCCI [45]

The high efficiency of these engines also results from the possibility of using the entire space above the piston as a combustion chamber, which in a classic CI engine is limited to a small combustion chamber located in the piston. The time needed for a good mixing of low-reactive fuel with air is provided by indirectly supplying this fuel to the intake manifold, or a series of high-pressure injections directly into the combustion chamber in various phases of engine operation called Direct Dual Fuel Stratification (DDFS) [5, 17].

This strategy is used in more advanced solutions, and both methods of powering the dual-fuel engine can also be mixed to improve the results.

Dual-fuel engines are also characterized by extraordinary fuel flexibility – they can be used with all fuels that can be used in classic SI and CI engines. This allows for an emergency change of the fuel used in the engines in the event of difficulties with the supply of the standard power source, and it is also easy to adapt the engine to burn more ecological fuels when they become widely available. among others for this reason, HCNG is such a good example of a fuel evolving towards an ecological fuel of the future, from which it will be easy to switch to clean, green hydrogen, and the highly reactive fuel, which in our considerations is HVO [4, 14], in the future may be a zero-emission e-fuel that can be used in these engines with only minimal or zero emission fuels and use them with higher efficiency than SI engines and in some cases also fuel cells [16].

3. Potential and reasons for the use of dual-fuel engines using hydrogen-enriched fuels

It should be remembered that the engines used in the power industry, maritime transport or industry have a much higher efficiency than high-speed SI and CI engines. The most efficient engines already achieve an efficiency of 54.6% [30, 31, 44, 54]. Overall efficiency of up to 55% in low-speed engines is no longer uncommon, and designers predict that thanks to RCCI technology in engines of this type it will be possible to achieve values of 60% [44]. Such performance makes engines of this type an attractive alternative to turbine engines, fuel cells or batteries that allow the development of electromobility. Hydrogen, which is not only a fuel, but also a storage of energy generated from electricity sources, combusted in internal combustion engines, allows for the recovery of this energy when it is needed. The graph below in Fig. 8 below shows the electricity production in the power system, which shows the scenarios for different capacities of installed PV installations.

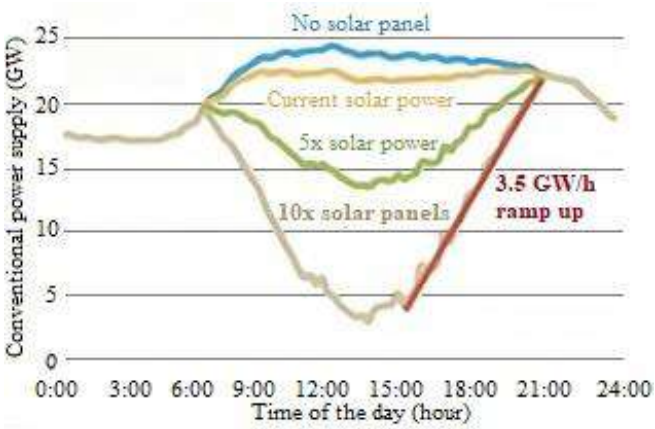

Fig. 8. Scenarios of electricity production using PV in the energy mix (based on [26])

The figure above shows how the power of the installed photovoltaic installations in the power grid affects production during the day. The visible decrease in production from conventional sources around noon is problematic for the system operators and for this reason the need to store the energy produced during this period will increase. A network constructed in this way will require energy to be delivered in the period when Renewable Energy Source (RES) does not operate with high power and its storage in the period when it is operating. One of the methods of energy storage is the production of hydrogen and its subsequent use, e.g. in internal combustion engines. Thanks to the possibility of using the existing natural gas distribution network, it is possible to distribute hydrogen to the target customer in the form of HCNG fuel. In combustion engines, the use of liquid fuels also allows the combustion of green hydrogen, because fuels enriched with it, such as HVO, can use for this process green hydrogen produced during the surplus of electricity production from RES. Due to the possibilities offered by DF ICE in the use of green energy sources, simulations have been developed showing the level of carbon dioxide emissions depending on the hydrogen content in HCNG, at various degrees of replacement [5] of high-reactivity fuel with low-reactivity fuel. The graph below in Fig. 9 shows the effect of the hydrogen content on the carbon dioxide emissions during complete combustion of HCNG.

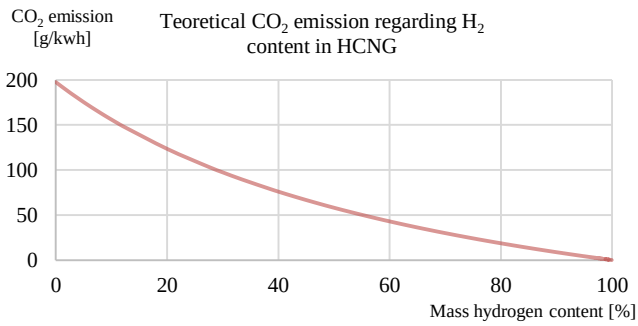

Fig. 9. Impact of hydrogen content on CO_2 emissions from HCNG combustion

The influence of hydrogen content on carbon dioxide emissions visible in the graph in Fig. 9 clearly shows that with the initial increase in hydrogen content at low concentrations, carbon dioxide emission decreases more intensively than with high hydrogen content in the mixture. This shows how important it may be to add methane, even with a small amount of hydrogen, to improve the ecological parameters of this fuel.

The chart on the Fig. 10 below shows the level of carbon dioxide emissions for various types of ICE and the fuels used in them, with the assumed values of overall efficiency of SI engines fueled with gasoline at the level of 36.4% [54], SI fueled with CNG at the level of 39% [27, 37], CI fueled with oil or HVO of 44% [37], DF of 45% [37], and RCCI of 46% [43].

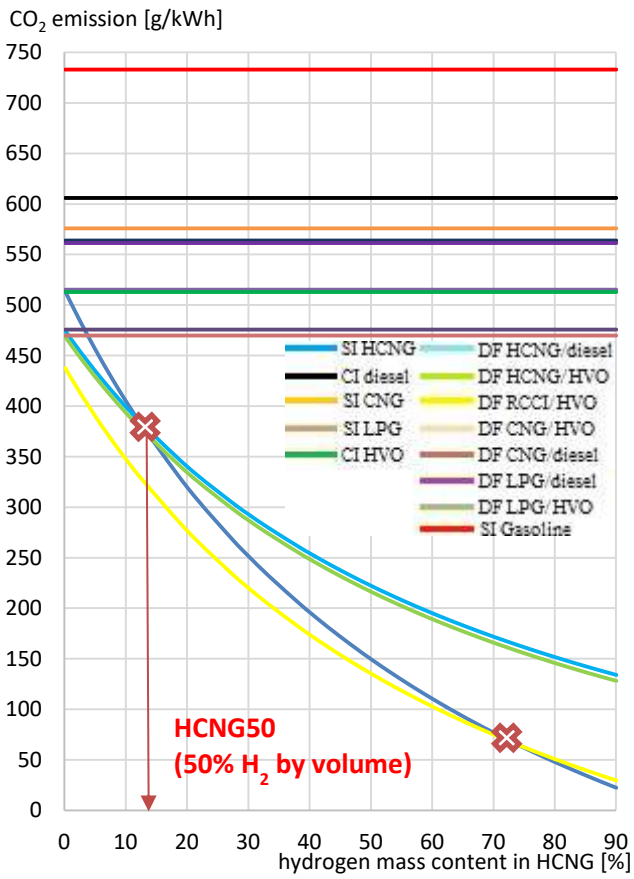

Fig. 10. Carbon dioxide emission intensity for different types of engines and different types of fuels used as a function of hydrogen content in HCNG fuel with maximal fuel replacement in DF mode at the 80% level for CNG & HCNG fuel and 35% for LPG

The graph above clearly shows the impact of the hydrogen content in HCNG on carbon dioxide emissions and shows the advantage of DF engines in the range where the hydrogen concentration in the HCNG mixture does not exceed 50% of the volume content in the mixture (~12% of the mass content). This is the limit value at which hydrogen can be mixed with NG without serious consequences related to the safety and impact of this fuel on the corrosiveness of steel [34] and other serious problems with combustion and distribution [1, 2, 13, 14, 20, 24, 29, 33, 36, 39, 41]. These hydrogen contents in HCNG constitute a mixture that can be effectively used in combustion engines, as determined in many tests [14, 20, 24]. To better understand the advantages of using two different types of engines in specific operating conditions, three additional simulations, presented in the charts below were presented and analyzed. Figure 11 and 12 represent engine used in transport, Fig. 13 represent the same type of engines but used in power engineering or industry.

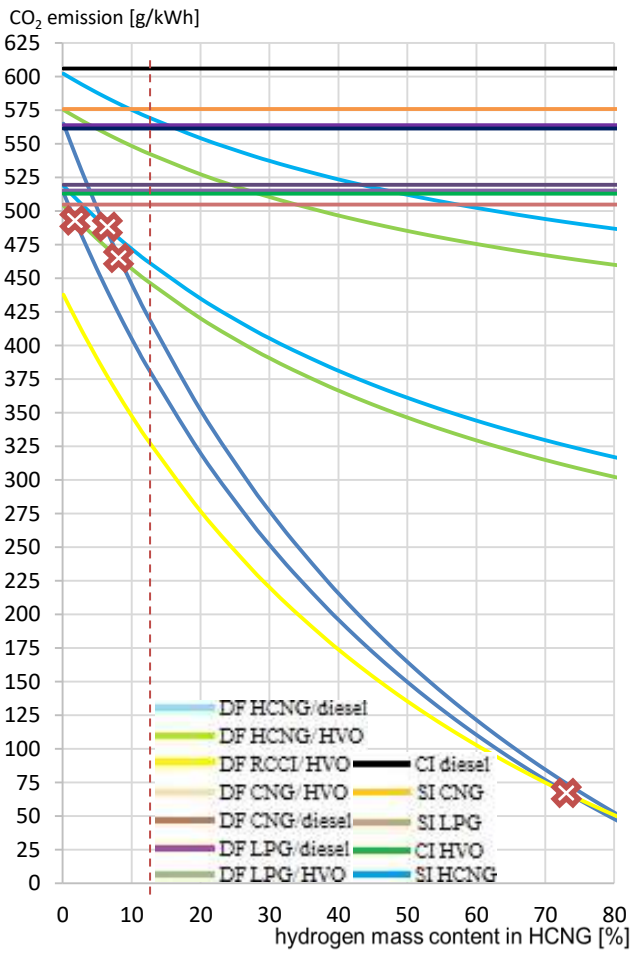

Fig. 11. Carbon dioxide emission intensity for different types of engines and different types of fuels used as a function of hydrogen content in HCNG fuel with maximal fuel replacement in DF mode at the 80% level and 50% level for CNG & HCNG fuel and 35% for LPG and different efficiency for engines using HCNG fuel

Lines of the same color symbolize the range in which an engine of a given type can operate when powered by a specific fuel. The line with the lowest achieved CO_2 emissions symbolizes the optimal operating point at which the engine operates, at which the assumed replacement rate is 80% and the maximum efficiency, as written earlier, is 45%. The line of the same color with the higher emission value shows the engine operating condition in which the maximum efficiency is reduced by 10%, which in this case results from the reduction of the engine speed, while the degree of replacement decreases in the square of its maximum value, i.e. for the maximum replacement of 50% it drops to 25%, and for 80% it drops to 64%. These values correspond to the values achieved on two types of gas in-

stallations that are most often used in DF engines. Lower replacement values allow the engine to operate at low load without the risk of knocking combustion. Both of these factors are taken into account in the emissions calculation for the line representing the maximum emissions during normal operation of the described engine. Dotted line shows the HCNG50 level.

Figure 11 shows that only when operating optimally (with high efficiency and degree of replacement), a DF engine is able to produce lower CO_2 emissions than an SI engine using HCNG fuel with a low H_2 content (< 50% V/V). The low replacement degree, which is characteristic of low-advanced and inexpensive gas installations, means that the use of SI engines actually becomes a more ecological solution, despite their lower overall efficiency. The graphs below (Fig. 12) show the CO_2 emissions with maximum replacement of a DF engine of 50% and 80%. The emission crossover points for the SI & DF engines show the H_2 content of HCNG at which CO_2 emissions are equal.

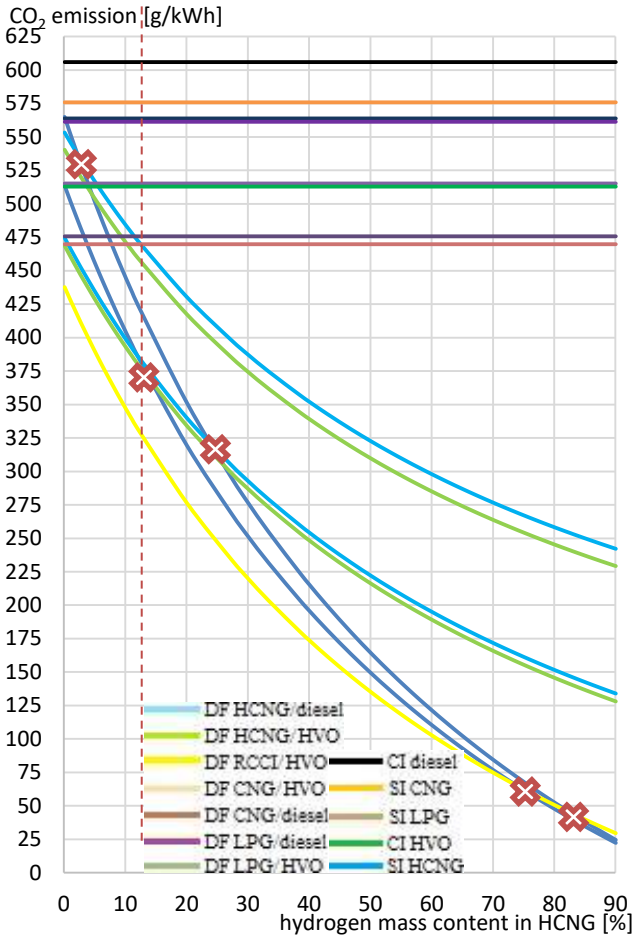

Fig. 12. Carbon dioxide emission intensity for different types of engines and different types of fuels used as a function of hydrogen content in HCNG fuel with maximal fuel replacement in DF mode at the 80% level for CNG & HCNG fuel and 35% for LPG

This clearly shows what type of engine is worth to use when HCNG fuel with a bit higher H_2 content is available. We can clearly see the contents, which SI engines prepared for gas fuel are a less emission-intensive option than DF engines. Even the use of HVO does not significantly change this situation and only when operating at the highest efficiency similar CO_2 emission values for both engine types are achieved. The level at which the RCCI engine is more emissive than the SI engine is a value very close to using pure H_2 with an admixture of NG, not NG with an admixture of H_2, and is beyond the reach of the currently used transmission infrastructure.

For the HCNG fuel at the level of hydrogen mass content around 5% (~20% V/V) the CO_2 emission is similar in DF HCNG fueled engines and SI HCNG engines. HCNG with a low H_2 content between 5–20% V/V is the popular option for the nearest future, what stands DF engines as the more ecological than SI engines, fueled by that kind of HCNG fuel mix.

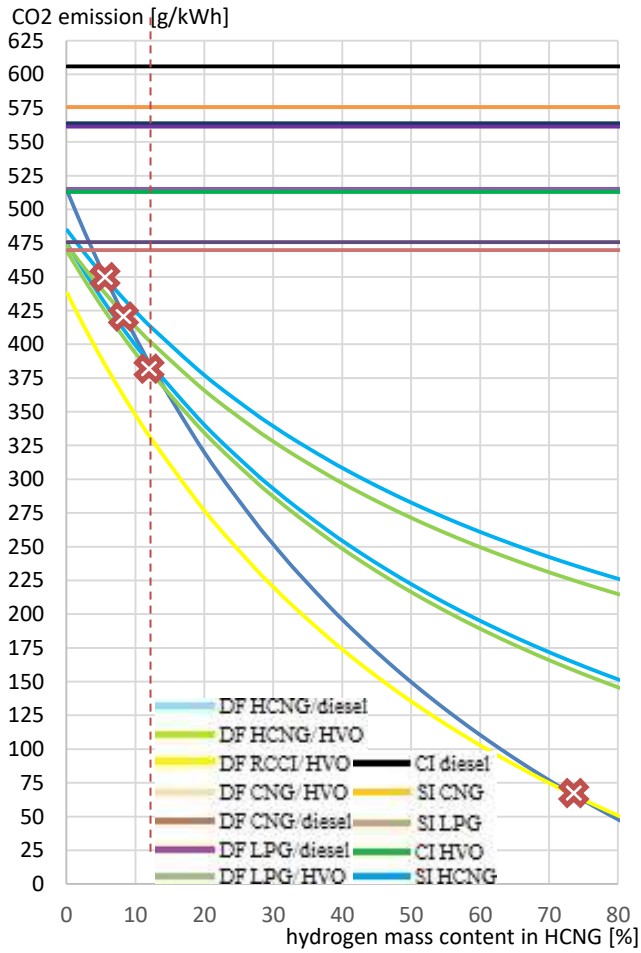

Fig. 13. Carbon dioxide emission intensity for different types of engines and different types of fuels used as a function of hydrogen content in HCNG fuel with maximal fuel replacement in DF mode at the 80% level and 50% level for CNG & HCNG fuel and 35% for LPG

The third graph shows a stationary engine operating at constant speed but with different loads. In the scope of operation of this engine, only the impact of changing the degree of fuel replacement was taken into account, and it was assumed that a higher generation gas installation was used, allowing for achieving high degrees of replacement, which in continuous operation as a power generator has its economic justification. A smaller range of CO_2 emissions in which the engine operates is visible. This allows to see

more clearly when such an engine is emitting a similar amount of CO_2 as an equivalent SI engine where their emissions levels are crossing. For HCNG with H_2 mass content below 10%, it can be seen that even in the least optimal operating range, DF engines have a similar emission level to SI engines. The impact of the use of HVO is also more clearly visible here, which, with a reduced degree of replacement, significantly reduces the level of CO_2 emissions.

This proves that until the infrastructure for supplying hydrogen to end users is in place, dual-fuel engines will have an advantage over SI engines. When pure hydrogen can be distributed and burned in engines as the only fuel, or with highly reactive fuel, which is completely zero-emission.

4. Conclusions

In industrial engines, low-emission fuels will be of great importance due to their common occurrence in the power plants, industry and maritime transport, and the amount of fuel they consume. The cost-critical nature of these industries adds to the importance of the efficiency of the engines used in these industries. Therefore, our simulations allow us to determine the limits according to which we can be guided in the application of given types of engines, where dual-fuel engines should be used when using HCNG 0–50 fuel, while in cases where ammonia or pure hydrogen would be used, it would be more favorable for environmental reasons would be the use of CI engines.

Summing up, the presented solution can provide:

- reduction of CO_2 emissions in operating installations using internal combustion engines by over 35%
- lowering the operating costs of enterprises; flexibility in the use of different fuels
- increasing the operational reliability of the power system
- improving the structure of the energy mix
- increasing the potential for further development of green energy sources in the existing energy infrastructure
- extending the lifetime of internal combustion engines already produced and the existing power generation structure
- the use of a gas installation in DF engines, which allows for low levels of fuel substitution, leads to performance that is uncompetitive compared to SI engines in terms of CO_2 emissions
- stationary engines whose speed is constant and the load is always high enough to ensure that these engines operate with relatively high overall efficiency make their use justified at any available HCNG concentration
- the RCCI engines allows a reduction in CO_2 emissions practically over the entire range of H_2 content in HCNG compared to SI engines. When RCCI engines become widely available, the use of hydrogen-enriched fuels in fueling should become the standard.

Acknowledgements

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Nomenclature

CI	compression ignition
CNG	compressed natural gas
CO_2	carbon dioxide
DDFS	direct dual fuel stratification
DF	dual fuel
EU	European Union
ICE	internal combusiton engines
GHG	greenhouse gas
HCCI	homogeneous charge compression ignition
HCNG	hydrogen enriched compressed natural gas
HVO	hydro-treated vegetable oil
H_2	hydrogen
LNG	liquefied natural gas
NG	natural gas
PV	photovoltaics
RES	renewable energy source
RCCI	reactivity controlled compression ignition
SI	spark ignition
UGB	university research grant
US	United States

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Multi-Criteria Analysis of Semi-Trucks with Conventional and Eco-Drives on the EU Market

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Abstract: The research provides a comparative theoretical investigation of the operational characteristics of an electric semi-truck and vehicles powered by conventional combustion engines using diesel fuel, hydrotreated vegetable oil (HVO), and methane (including biomethane) in the dual fuel configuration. The Volvo tractor units that are offered for retail in 2024, namely the Volvo FH Electric, Volvo FH500 in dual fuel configuration, and Volvo FH500TC Diesel Euro VI, were chosen for comparison. The considerations encompassed include the road tractor's mass, energy usage, power-to-weight ratio, dynamics, ability to recharge or refuel, payload restrictions, impact on logistics expenses, compliance with regulations on drivers' working hours, and a report on carbon dioxide emissions. The study concludes by discussing and drawing conclusions on the competitiveness of different drive types in truck tractors, specifically in relation to identifying the most suitable areas of application. Synthetic conclusions demonstrate the high effectiveness of the electric drive in urban and suburban conditions. However, vehicles equipped with internal combustion engines using renewable fuels fill the gap in energy-intensive drives in long-distance transport.

Keywords: semi truck; EV; dual-fuel; diesel; HVO; LNG; operational properties; logistics; ecology

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1. Introduction

1.1. General Context and Market Situation

Road freight transport is a significant contributor to the economy and a crucial component of both internal and international trade in Europe and beyond [1]. Poland is the top performer in the road transport sector within the European Union (EU), accounting for 19.8% of the total ton-kilometers (TKM) in the EU, surpassing Germany (16.0%) and Spain (14.1%) [2], as illustrated in Figure 1 below.

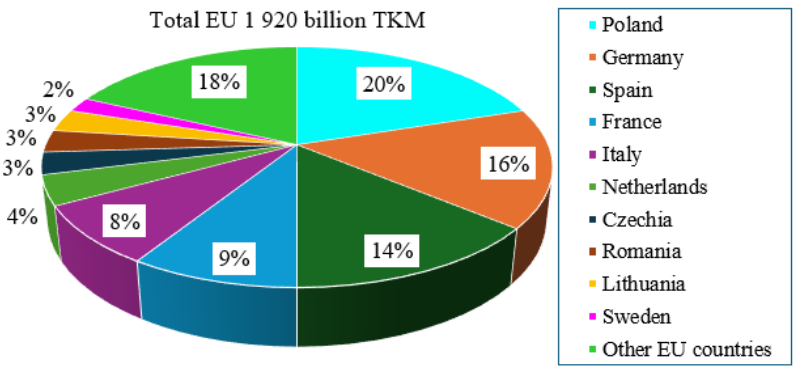


Figure 1. EU market share for top 10 countries of total road transport in 2022. Drawing made by the authors based on available data found in the paper [2].

In 2022, the European Union had a total of 6.2 million transport trucks in operation [3]. Considering the size and importance of this industry in the economy, it is crucial to make appropriate arrangements for the upcoming changes.

The transport industry in the EU is currently experiencing a dynamic process of transitioning vehicle propulsion sources. The adoption of hybridization and electrification in the passenger car industry is becoming a reality, as indicated by the consistent growth in the percentage of vehicles with these types of propulsion in new car sales, as depicted in Figure 2 below.

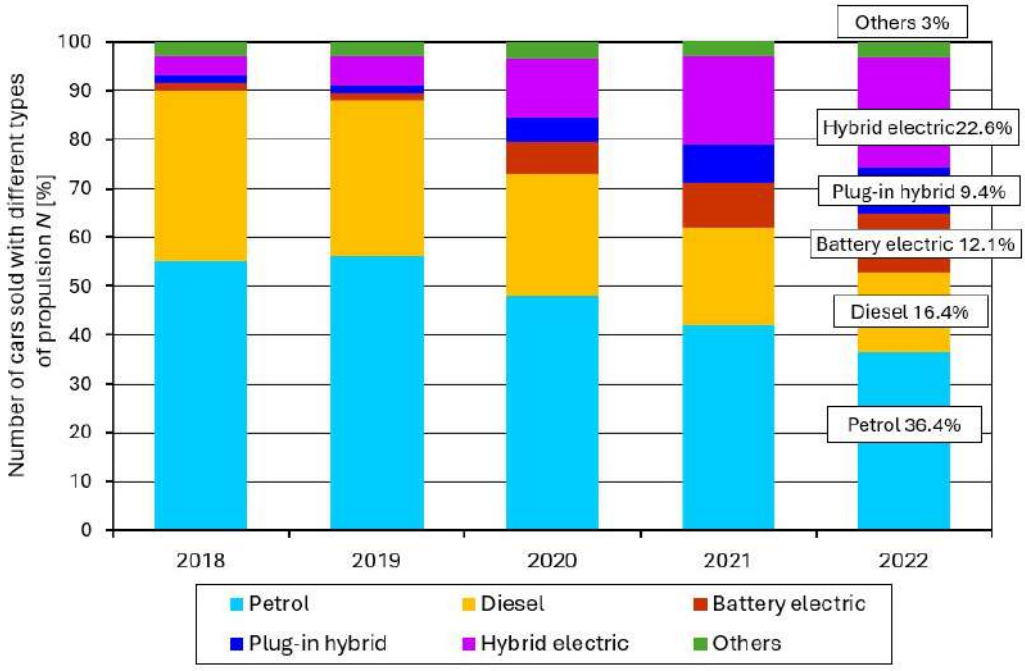


Figure 2. New EU passenger cars sold by power source-market share 2018–2022. Drawing made by the authors based on available data found in the paper [3].

1.2. Motivation and Research Gap

As depicted in Figure 2, there is a significant push to decrease CO_2 emissions and replace traditional internal combustion engines (ICE) with more environmentally friendly alternatives such as hybridization and electrification. However, in the context of long-distance heavy road transport (as well as in the sea and air transport sectors), there are still issues with energy consumption and the limited potential to replace hydrocarbon fuels [4]. This situation entails a clash between the necessity to protect the environment and the practical constraints imposed by economic considerations necessary for preparations for the forthcoming changes.

The heavy-duty transport market has adopted electric vehicles (EV) and vehicles using alternative fuels, such as biomethane or hydrotreated vegetable oil (HVO), in order to align with sustainable development and mitigate climate change by reducing CO_2 emissions. The European Parliament has officially supported a position on the European Union's proposed legislation for reducing carbon dioxide (CO_2) emissions from Heavy-Duty Vehicles. Parliament emphasized the importance of CO_2-neutral fuels in the overall climate plan. Enacted on 21 October 2023, this stance mandates the European Commission to compile a comprehensive inventory of lorries, buses, and trailers that solely run on CO_2-neutral fuels. While the details are not explicitly stated, it is suggested that these vehicles will help manufacturers achieve their obligatory targets for reducing carbon emissions: a 45% decrease by 2030, 65% by 2035, and 90% by 2040 [5,6].

Modern semi-trucks, equipped with different propulsion systems like electric, dual-fuel, or those running on ecological fuels, have operating differences compared to typical internal combustion engine (ICE) vehicles that use diesel fuel. This differentiation is notable in road transportation, as efficiency incorporates frequently conflicting characteristics. It is essential to comprehend the significance of selecting the appropriate drive source throughout this shift in vehicle propulsion technologies, which is a critical concern for numerous entrepreneurs in the EU. Opting for the appropriate vehicle can provide a strategic advantage in a sector that accounts for over 6% of Poland's GDP and employs approximately 1 million individuals [7].

The significance of this subject is emphasized by the requirement for extensive industry knowledge regarding the efficiency of logistics and the preservation of the environment in the development, production, and upkeep of facilities and systems, particularly modern logistics systems that are essential for road transportation [8]. The article provides a comprehensive overview of the current technologies utilized in road transport and their effects on the environment, functioning, and economy. The discussion will focus on the contrasting operational characteristics of contemporary electric and renewable fuel vehicles versus conventional vehicles powered by fossil fuels. The objective of the article is to demonstrate the fundamental changes in the power source of semi-trucks and the consequent effects on their potential applications. Additionally, it aims to identify the most operationally advantageous areas for specific types of power sources. This study offers a novel approach to comparing the available options for purchasing real semi-trucks in retail, specifically focusing on the different drives available. It goes beyond previous research by considering a broader range of factors related to the suitability of Heavy-Duty semi-trucks for commercial applications. This encompasses the overall expenses (acquisition, maintenance, operation, energy consumption), distance coverage, capacity and payload restrictions, influence on transportation costs and logistics, environmental consequences, and other relevant factors. Previously, scholars in this field have examined certain elements [9–16]. This publication provides a comprehensive and expansive view of this issue. Until now, there has been no complete comparison of all the major elements of genuine vehicles available for purchase on the market. The material offered in this work addresses this gap. The essence of the work is to use a theoretical approach and collect all the necessary aspects relevant to the actual attractiveness of the use and selection of a means of transport in the form of a semi-trailer.

The subsequent sections below provide a presentation of the data and methodology utilized for the calculations.

2. Materials and Methods

Volvo FH tractor models offered for open retail sale in 2024, with different drive sources, were selected for comparison. Volvo's FH model significantly influenced the development of safety and environmental laws in Europe. Volvo consistently surpassed legislative demands by pioneering innovative technologies that eventually became industry norms. The manufacturer provides semi-trucks equipped with electric propulsion, dual fuel capability, and a conventional EURO VI diesel engine. The technical parameters presented in Table 1 are based on the information provided by the manufacturers of the vehicles being compared.

Table 1. Selected technical data of the compared semi-trucks. Table made by the authors based on available data found in [17–22].

Vehicle	Volvo FH Electric [17]	Volvo FH Gas (Dual Fuel—DF) [18,19]	Volvo FH500 Euro VI [19]
Drive type: engine, gearbox	3 electric motors, 12 gears I-Shift drive gearbox	G13C500, 12 gears I-Shift drive gearbox	D13K500TC (turbo compound), 12 gears I-Shift drive gearbox
Fuel type	electric	Diesel/HVO and LNG/BioLNG	Diesel/HVO
The maximum power/torque	330 kW (eco mode) 490 kW (power mode) continuous power 2400 Nm full range	368 kW through 1404–1700 RPM 2500 Nm through 980–1404 RPM	370 kW through 1250–1600 RPM 2800 Nm through 900–1250 RPM
Energy consumption at full load	1.1 kWh/km (eco mode) [20]	0.246 kg/km LNG and 0.024 L/km diesel [21]	0.27 L/km [22]
The largest energy storage offered	Battery pack 540 kWh (useful capacity 80%)	LNG gas tanks 225 kg (545 L) Liquid fuel tank 150 L	Fuel tanks 1480 L
Range *	~392 km	~915 km	~5480 km

* theoretical range calculated from the capacity of the energy storage tank and average consumption.

The electric version of the vehicle offers various driving modes that enhance the electricity economy. When comparing energy consumption, the most efficient results were considered for further calculations and comparisons, applicable to both electric and ICE vehicles. These figures were gathered from actual tests and operational driving, adhering to the provided references. It should be noted that Volvo is the sole manufacturer offering dual fuel (DF) heavy trucks powered by CH_4 derivatives (LNG in this case) in a non-SI (Spark Ignition) version. This allows for a higher compression ratio, which enhances the energy efficiency overall. The vehicle in question can utilize both LNG and BioLNG as primary fuels, and diesel or its alternatives such as fatty acid methyl esters (FAME), biodiesel, or HVO for pilot dose ignition [23]. These diesel fuel alternatives are also compatible with the D13K500TC EURO VI engine. The turbo compound technology is employed to optimize the engine's torque curve, consequently reducing fuel consumption, which is advantageous for long-haul operations [19]. Irrespective of the source of propulsion, the manufacturer has selected a 12-speed gearbox due to its ability to choose gear ratios that are suitable for different terrains, therefore affecting the vehicle's energy efficiency. The drivetrain systems provided by the manufacturer were also taken into account to provide a comparison of vehicle dynamics. Table 2 summarizes a brief overview of the technical data of the vehicles under consideration, which is essential for comparing their traction characteristics.

In the configuration with both ICE (diesel, DF), the manufacturer offers a combination of the same AT2612G gearbox and rear axle. In the case of the electric drive, an integrated drive system containing 3 electric motors and a gearbox assembly with the same gear set as the gearbox used in the ICE configuration is considered. For trucks, which typically use a range of gears to handle heavy loads and varying speeds, the mechanical efficiency at different gear levels will be different. Higher gears are generally more efficient than lower gears as they provide a greater speed increase relative to the increase in energy input, while lower gears are designed to increase torque. The final drive ratio in the rear axle can be configured in each case (RSS1344D and RSS1344E), but the gear ratios were selected as in Table 2, which was justified in point 3.2 of this work.

All the above data were taken into account for further calculations. For the sake of transparency of content, the rest of the necessary data and methods to perform the calculation will be entered and presented directly in the appropriate subsections.

The next part of the article (Section 3) compares the operational aspects of the considered vehicles in terms of the following:

- Payload and the associated semi-truck curb weight;
- Vehicle dynamics in terms of power to weight and driving force at the wheels;
- Logistics costs with the distinction of purchase and general maintenance costs, energy and TKM costs and others;
- CO_2 emissions over the entire life cycle of the vehicles.

The article is finished with a discussion section and synthesized conclusions.

Table 2. Selected transmission data of the compared semi-trucks. Table made by the authors based on available data found in [24–28].

Engine Type	**Diesel**	**Dual Fuel**	**Electric**
Gearbox Type	**Volvo I-Shift 12 AT2612G [24]**		**Volvo I-Shift 12 EPT2412 NEM3 [25] ***
Gear No.	**Gear Ratio**		**η—Gear Efficiency [26]**
1st	14.94:1		η—88%
2nd	11.73:1		η—90%
3rd	9.04:1		η—91%
4th	7.09:1		η—92%
5th	5.54:1		η—93%
6th	4.35:1		η—94%
7th	3.44:1		η—95%
8th	2.70:1		η—96%
9th	2.08:1		η—97%
10th	1.63:1		η—98%
11th	1.27:1		η—99%
12th	1:1		η—100%
Rear axle	Volvo RSS1344D [27]		Volvo RSS1344E [28]
Final gearing	2.85:1		3.36:1
Gearing efficiency [26]	η—98%		η—97%
Drive wheels size	315/70R22.5 = Tire radius ~500 mm		

* Volvo Electric Drive Unit equipped with NEM3 (Number of electrical motors—3 units) total electric traction at 2400 Nm and EPT (Electric Propulsion Transmission) 2412 in-house I-Shift 12-speed gearbox [8].

3. Results

3.1. Payload

Semi-trucks are vehicles that provide users with a full configuration in terms of equipment, type and size of the cabin and the number of drive axles. Whatever drive type you prefer, Volvo tractor units are available with one or two driven rear axles. Each element of the semi-trailer can be adapted to the individual needs of the user. Depending on their preferences, customers can choose between a long or a short cabin, as well as between different cabin sizes. In addition, the tractor units are fully configurable in terms of equipment. Users can choose from a variety of options and extras such as safety systems, lighting, navigation systems, seats and seat mattresses to enhance comfort on long journeys. Although aspects such as equipment configuration, cab type and size, and number of driving axles are factors in terms of total vehicle weight, they will not be considered in further assessment. It was assumed that the compared vehicles have the same configuration of additional (selectable) equipment and differ only in the type of drive. There is no exact information on the weight of the Volvo FH electric powertrain. According to [5], the battery energy density in an EV is around 272–296 Wh/kg. The mass of the car battery under

consideration, adding 35% to the net cell mass of the battery, can be calculated according to the following equation:

$$\frac{540{,}000\,\frac{\mathrm{Wh}}{\mathrm{kg}}}{\sim 284\ \mathrm{Wh}} * 1.35 \sim 2570\ \mathrm{kg} \quad (1)$$

Contrary to generally established beliefs, a production EV semi-truck, despite the very low energy density of chemical batteries, does not have to differ significantly in weight from their combustion versions. According to the information from [29], a fully prepared ICE engine with operating fluids and an exhaust gas treatment system can weigh approx. 1500 kg which, to some extent, compensates for the weight of the battery. The weight of the FH tractor unit may vary depending on the configuration; according to the source, the "light version" weight starts at approx. 6600 kg [30]. Differences in weight between the diesel and dual-fuel designs are almost the same [31], so they should not have a major impact on the overall energy intensity of the vehicle. The rest of the components of the drivetrain systems necessary to implement torque to the drive wheel of the vehicles can be considered equivalent mass for all types of propulsion. In further considerations, the typical weight of a universal curtain semi-trailer was assumed for 7200 kg [32]. Therefore, Table 3 below shows the curb weight of the vehicles concerned and their payloads. These data also form the basis for further calculations in this article.

Table 3. Curb weight and according to the payload for selected semi-trucks.

	Electric	DF	Diesel
Semi weight [kg]	8100	7000	7000
Payload [kg] *	24,700	25,800	25,800

* The maximum total weight of a typical heavy goods road transport set in Europe, i.e., a 2-axle semi-truck with a 3-axle semi-trailer, cannot exceed 40,000 kg [33].

In the case of the discussed vehicles, the ratio of the weight of the transported goods to the weight of the vehicle with the semi-trailer is, respectively, 1613 for EV and 1.81 for ICE. The difference in the vehicle load capacity is only 4.45% or in absolute numbers 1100 kg.

3.2. Dynamics of the Vehicle

The power-to-weight ratio, torque and power curves have a significant impact on the dynamics of a heavy vehicle. The power-to-weight ratio is the ratio of the vehicle's weight to its maximum power output, while the torque curve describes how torque changes with engine speed. In general, a higher power-to-weight ratio means that a vehicle will have better acceleration. Additionally, a flatter torque curve can also lead to better dynamics, as it allows the engine to maintain a consistent power output at different engine speeds. Table 4 below summarizes the power-to-weight ratio for several selected load configurations.

Table 4. Power-to-weight ratio of selected vehicle sets and various loads.

Power-to-Weight Ratio (kW/ton)				
Vehicle	Electric		DF	Diesel
	Eco Mode	Power Mode		
Semi only	40.74	60.49	52.49	53.57
Semi with trailer only	21.56	32.02	25.8	26.4
25% of the total weight of the possible freight load *	15.36	22.81	17.77	18.16
50% of the total weight of the possible freight load *	11.93	17.72	13.54	13.87
75% of the total weight of the possible freight load *	9.50	14.48	10.93	11.18
Gross combination weight: 40 tons	8.25	12.25	9.17	9.37

* Calculated from the formula: maximum load weight = semi-truck weight + semi-trailer weight − 40,000 kg [33].

The above data show a similar degree of vehicle operability due to their power-to-weight ratio. The specificity of the electric drive allows you to choose between performance

exceeding the considered conventional drives, while sacrificing range, but we have more dynamics at our disposal (~13–30% more than ICE). Differences in the dynamics of DF and diesel are in the range of 2% in favor of the diesel-powered drivetrain. The traction characteristics of an EV differ from the traction characteristics of a vehicle with ICE, which has a significant impact on the dynamics and acceleration of the vehicle. Figure 3 shows the power and torque characteristics of an EV propulsion system. Figure 4 shows the power and torque curve for an FH engine equipped with an ICE: (a) diesel, (b) DF.

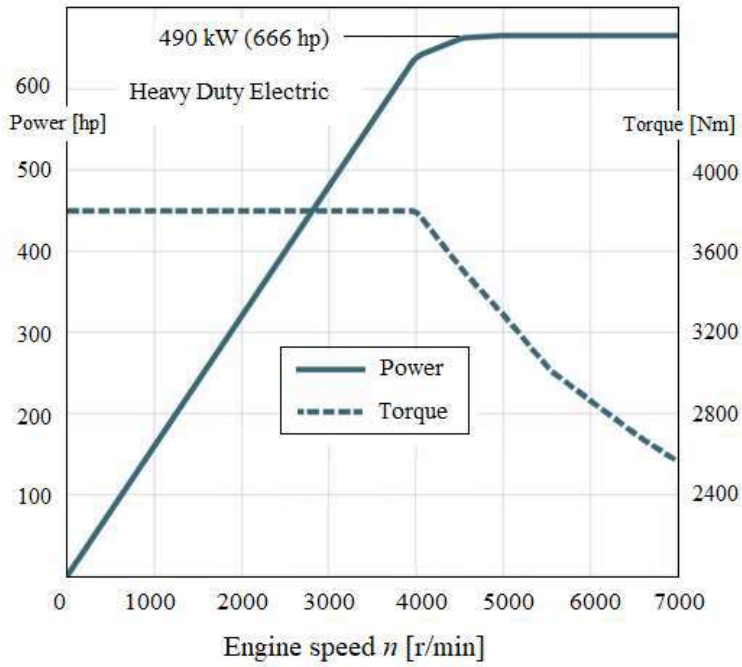
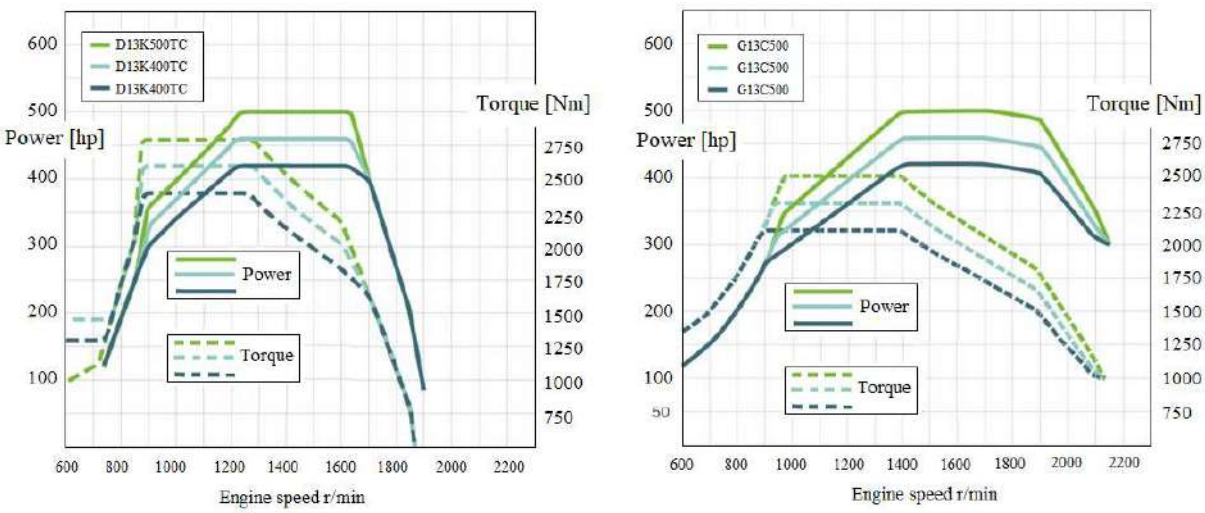


Figure 3. Power and torque characteristics for an EV propulsion system. Drawing made by the authors based on available data found on the vehicle's producer website [34].



(**a**) (**b**)

Figure 4. Power and torque characteristics: (**a**) diesel; (**b**) DF. Drawing made by the authors based on available data found on the vehicles' producer website [34].

The figures presented above show a clear advantage in the torque curve (and therefore power) for the electric drive, in which the maximum torque is available from 0 rpm up to 4000 rpm. However, inaccuracies in the data presented by the manufacturer are clearly

visible, both in Table 1 and in the chart in Figure 3. According to the formula for engine power [35].

$$N_e = \frac{M_o \cdot n}{9549.3} \text{ [kW]} \tag{2}$$

$$N_e = 1.36 \frac{M_o \cdot n}{9549.3} \text{ [HP]} \tag{3}$$

where M_o—motor torque [Nm], n—motor speed [rpm].

For the exemplary value of 3000 rpm and 3800 Nm from the graph in Figure 3 the result power is ~1623.5 HP. This is approx. 3.35 times more power than actually shown on the chart. This indicates that the presented torque data use values multiplied by the vehicle's final drive (Table 2). This may have purely marketing justification. The manufacturer Volvo did not respond to questions about discrepancies in this aspect asked by the authors in an e-mail. In further considerations, the ratio between the maximum power curve and the rotational speed from Figure 5, as well as the actual equivalent calculated torque, were taken into account. No such inaccuracies were found for ICEs.

Tractive effort curves serve as a comparison tool for the performance of vehicles with varying designs. These curves are depicted through a diagram that illustrates the relationship between the traction force (F_T) exerted on the vehicle's wheels and the driving speed (v) [36,37].

$$F_T = \frac{M_o \cdot i_N \cdot \eta_N}{r_k} \text{ [N]} \tag{4}$$

where i_N—gear ratio, η—drivetrain efficiency, and r_k—drive wheel radius [m].

$$v = 3.6 \frac{2\pi n \cdot r_k}{60 \cdot i_N} \text{ [km/h]} \tag{5}$$

In Figure 5, the example of tractive effort curve–driving force as a function of speed for a car equipped with a four-speed gearbox is presented below.

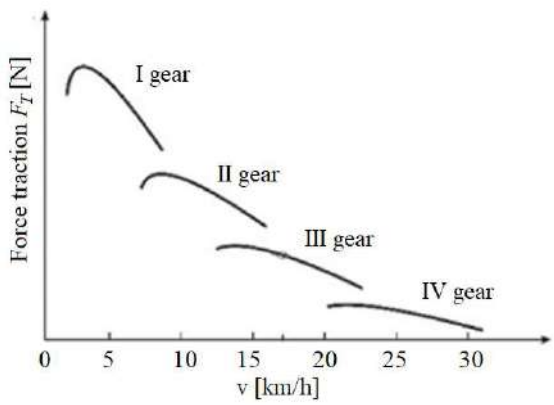


Figure 5. The example of tractive effort curve for a car equipped with a four-speed gearbox. Drawing made by the authors based on available data found in the paper [37].

The above diagram can also be superimposed with maximum power hyperbola, which in the F-V coordinate system is calculated as [37]:

$$F_T * v = P_{TMAX} \tag{6}$$

where P_{TMAX} = max. power on drive wheels.

In Figures 6–8, the tractive effort curve with maximum power hyperbola is presented for electric, DF and diesel drives, respectively, and Figure 9 shows the combined hyperbolas and first gear curve which represents the maximum avalible wheel force.

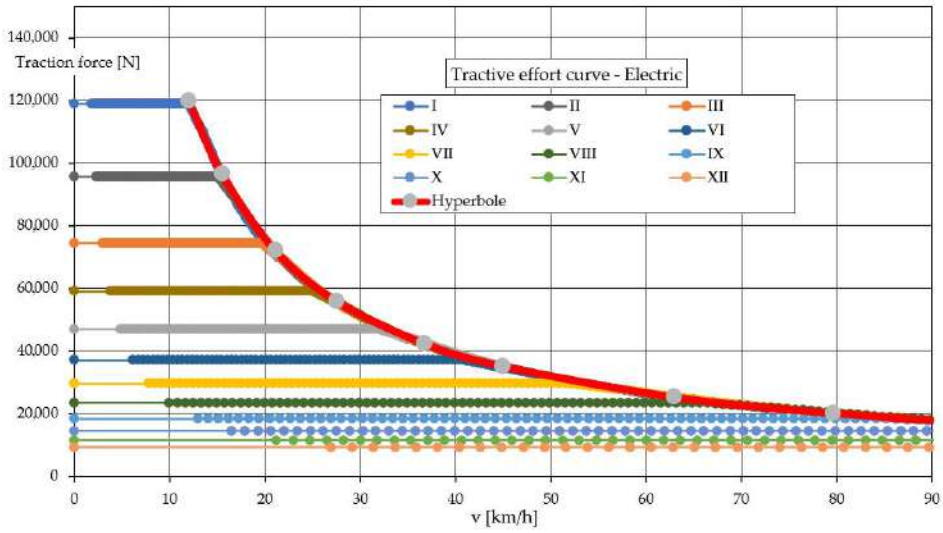


Figure 6. Tractive effort curves determined for electric powertrain.

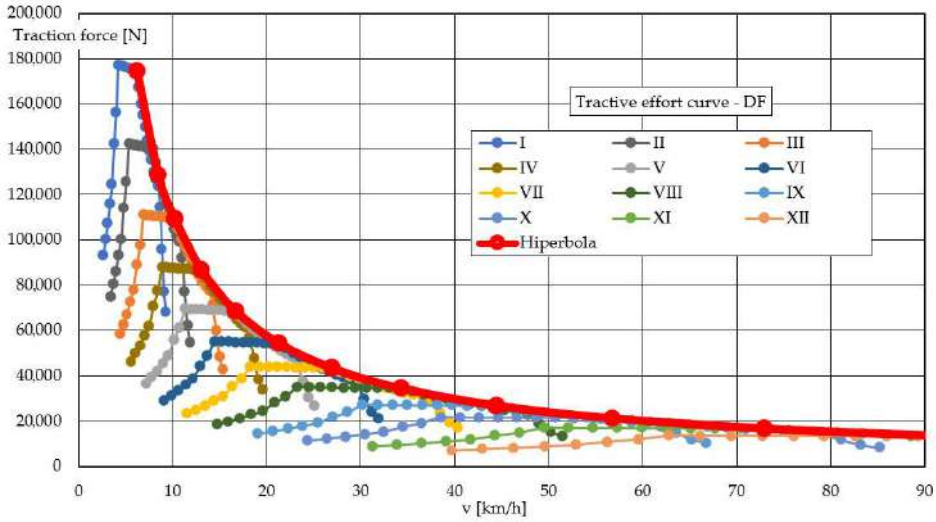


Figure 7. Tractive effort curves determined for DF powertrain.

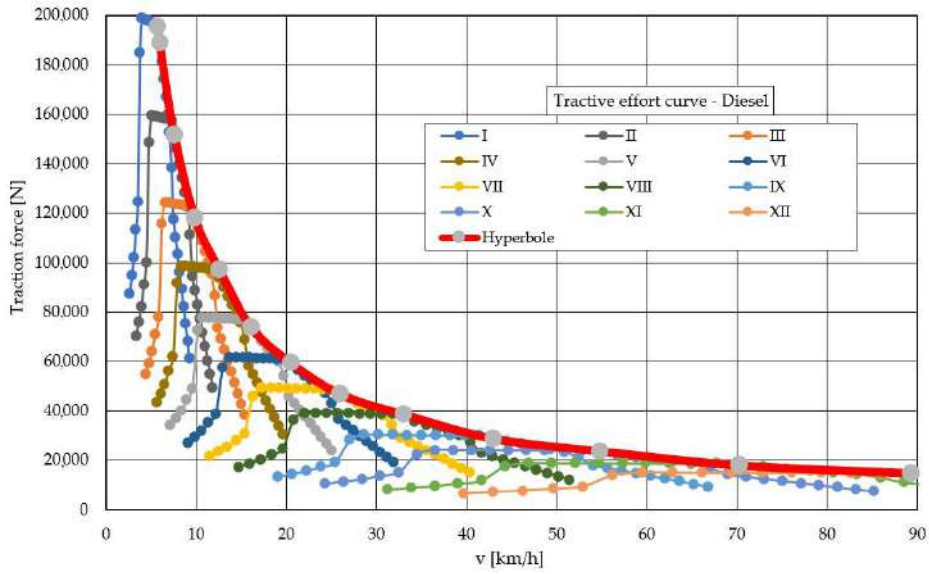


Figure 8. Tractive effort curves determined for diesel powertrain.

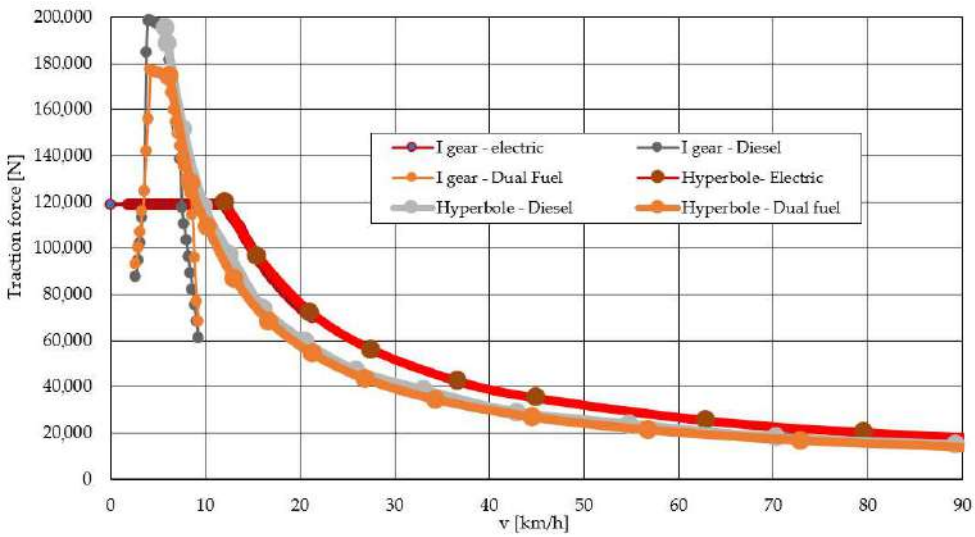


Figure 9. Hyperboles and 1st gear combined for all powertrains.

The peak wheel force is greater in ICEs with the two highest gear ratios. However, the frequent gear shifts required—due to the ICE's limited range of high torque—slow down the process and adversely impact the vehicle's dynamics. In contrast, the electric drivetrain displays a vast range of speeds over which their torque remains consistent, offering significant flexibility and positively influencing their dynamics, especially in the context of the frequent need to start and stop (urban and suburban conditions). Similarly, it is positively reflected in situations of variable demand for partial power, which reflects most of the operating time of a typical tractor unit. The use of ICE indicated that peak propulsion power is practical only in rare, extreme circumstances that seldom occur in everyday road transport on paved infrastructure with a known maximum load. High wheel peak power can be effectively used for, e.g., transporting oversized loads.

As mentioned earlier, for the highest gear ratios, differences in favor of ICEs are clearly visible. The electric drive system, thanks to constant high torque values from 0 rpm and the highest maximum power, has the most pro-dynamically improved maximum power hyperbole for the other 10 gears. The dynamics converge to one point as the maximum speed is approached.

To sum up, the area under the graph in Figure 9 above for the EV is only 2% higher than the ICE, but this does not reflect the actual subjective dynamics of such a vehicle, as explained in the previous paragraphs. The electric drive shows its advantage in frequent start–stop conditions and in situations of large load fluctuations. As we approach the maximum speeds, a convergence in the maximum performance of the electric drive and the ICE is observed. The differences between DF and diesel apart from the first gear ratio are almost identical.

3.3. Logistics Costs

Amidst the current era of instability characterized by unpredictable energy prices, fluctuating demand and supply, and disrupted supply chains, transport companies operating fleets of semi-trucks have significant issues in managing logistical expenses. These costs encompass various elements, including fuel expenses, vehicle maintenance costs, expenses associated with transport management, and the costs of foregone benefits resulting from operational limits and forced vehicle stops. Therefore, for semi-truck fleet owners, controlling logistics costs is crucial to achieving profitability. This chapter will discuss, in detail, the logistics costs associated with the use of selected truck tractors.

3.3.1. Purchase and Maintenance Costs

The purchase costs of semi-trucks obtained directly from a Volvo dealer on the Polish market. The starting price for selected models is as follows: Volvo FH Electric—EUR ~400,000; Volvo FH Gas (DF)—EUR ~150,000; and Volvo FH500 Euro VI—EUR ~115,000 [38].

To compare the costs of vehicle maintenance due to the differences in the drive, elements that distinguish specific drives in the context of their service were selected. Electric vehicles, due to recuperation, do not wear brake discs and pads in such an intensive way, and there is no need to change oil and filters, as is the case with ICE. According to the information obtained about the authorized Volvo repair service [38], the cost of the oil and filter service is PLN 2999 (EUR ~640). In the case of service costs for DF, due to the linkage of the maintenance of LNG tanks and additional fuel filters, as well as the higher MOT cost, the total cost of servicing was increased by 30% in comparison to the FH500 Euro VI. Oil and filter servicing should be performed every 120,000 km. Replacing a set of brake discs and pads for one axle is PLN 800 (EUR ~170). It is worth remembering that the use of an electric vehicle also reduces the wear of the brakes on the towed semi-trailer. In the comparison for semi and trailer combination, it was established that brakes are replaced after 150,000 km, and in the case of EV, it is four times less often, i.e., 600,000 km. The considered distance is the lifetime of the truck 1.5 million kilometers. A one-time replacement of the clutch in ICEs (EUR 2.500) was also assumed. Figure 10 below shows the costs associated with the purchase and maintenance of selected vehicles:

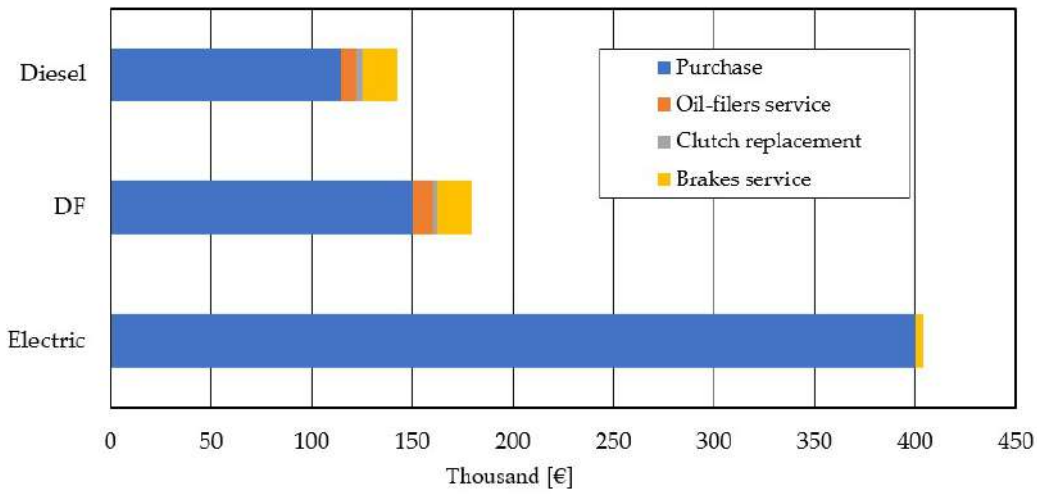


Figure 10. Comparison of service and purchase costs over a lifetime distance of typed vehicles (1.5 million kilometers).

3.3.2. Energy Costs

Due to geopolitical and economic turmoil, wholesale electricity prices in Europe have changed drastically, as shown in Figure 11 below.

It is difficult to give an accurate prediction of future electricity costs. For the purpose of calculations for this article, the average value of electricity is at the level of 0.1 EUR/kWh. It is worth noting that the costs of electricity from fast DC chargers on highways or in logistics centers are approximately three times higher [40].

As in the case of electricity, the prices of all types of fuels in Europe have been very volatile recently. Figure 12 below shows changes in diesel fuel prices in selected European countries.

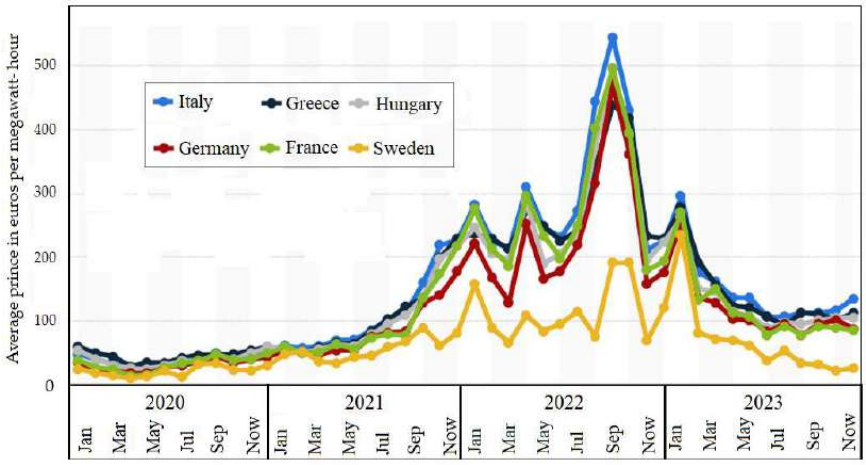


Figure 11. Average wholesale electricity costs in European countries since January 2020. Drawing made by the authors based on available data found in the paper [39]. Average price in euros per megawatt-hour.

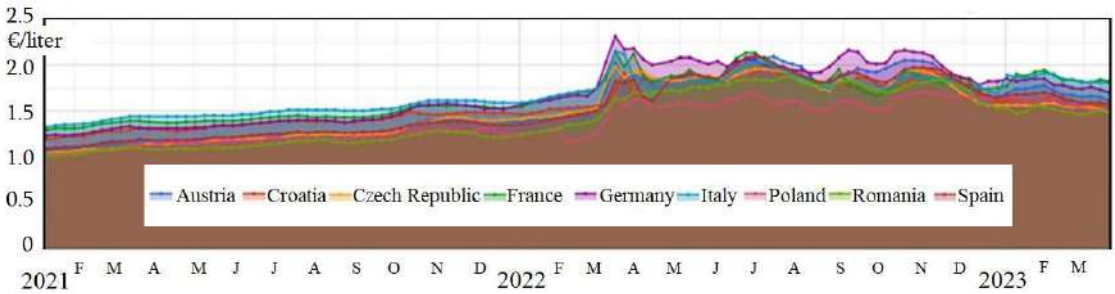


Figure 12. Average diesel fuel prices in European countries since January 2021. Drawing made by the authors based on available data found in the paper [41].

For the purpose of calculations for this article, the average net price for fuels was taken: diesel—1.75 EUR/L [39]; HVO—2.2 EUR/L [42]; LNG—2 EUR/kg [43]; BioLNG—1.6 EUR/kg [44]. AdBlue consumption was assumed at 0.2 L/100 km and cost 0.6 EUR/L. Fuel/electricity consumption for ICEs and EVs was selected in accordance with Table 1. Figure 13 below shows the costs related to fuel over a lifetime of 1.5 million kilometers for individual selected configurations.

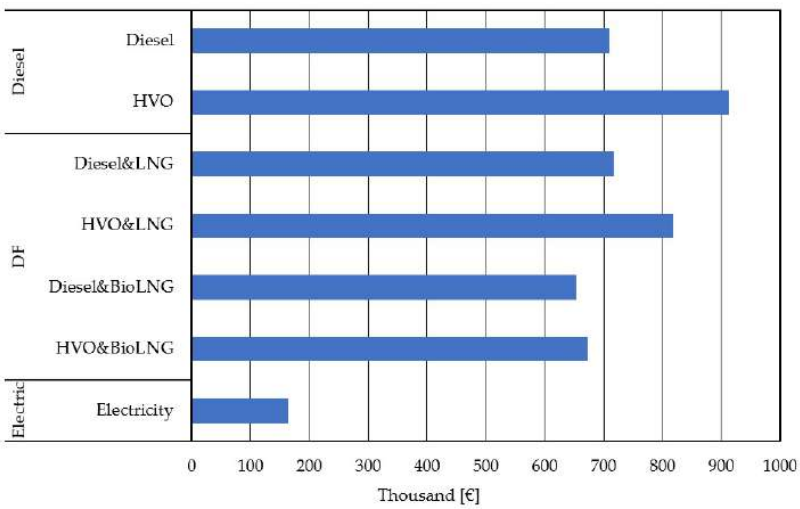


Figure 13. Energy cost for selected vehicle configurations and drives/fuel type in a lifetime of 1.5 million kilometers.

3.3.3. Ton-Kilometre Costs

The impact of the TKM on logistics and transport is related to the optimization of logistics and transport processes. Thanks to this tool, companies can conduct analyses and make decisions regarding the selection of the best transport solutions, such as the choice of the appropriate means of transport, transport route or the date of delivery of goods [45]. The TKM is an important indicator used in logistics and transport because it allows us to accurately determine the cost of transport and the efficiency of transport activities. The greater the TKM, the greater the volume of goods transported in a shorter time, which means a higher efficiency of logistics activities. To calculate the cost of one TKM, the data on the full payload value from Table 2 and the unit cost per kilometer are used in accordance with the data from Section 3.3.2 of this paper, summing a total vehicle distance of 1.5 million kilometers. The cost of one *TKM* was calculated according to the following formula [45]:

$$\frac{\frac{energy\ cost\ [€]}{distance\ [\mathrm{km}]} * total\ distance\ [\mathrm{km}]}{payload\ [tonne] * total\ distance\ [\mathrm{km}]} = \frac{cost\ [€]}{TKM} \tag{7}$$

The costs of one ton-kilometer over a period of operation of 1.5 million kilometers for individually selected vehicle configurations are presented below in Figure 14.

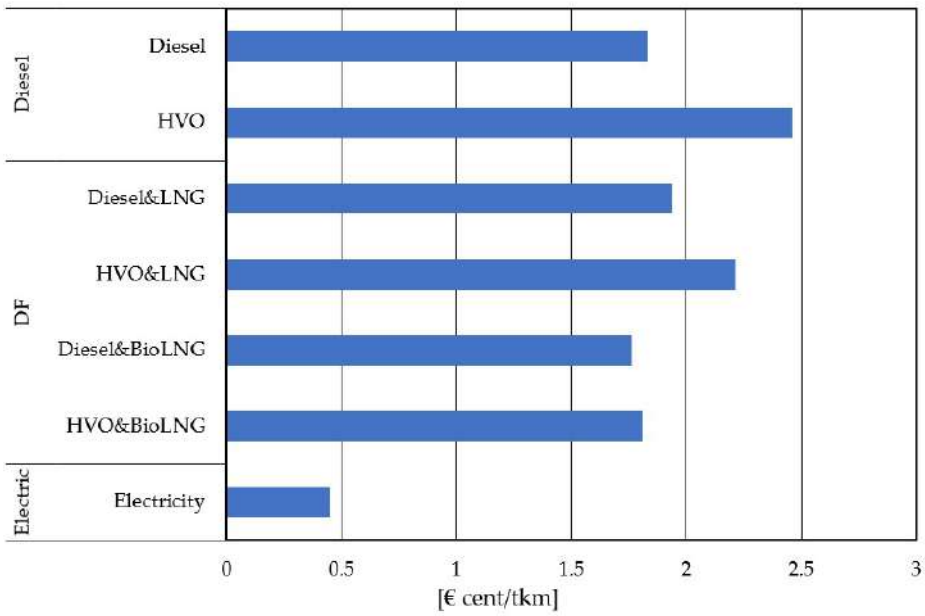


Figure 14. Ton-kilometer cost for selected vehicle configurations and drives/fuel type in a lifetime.

3.3.4. Non-Physical Costs and Other

Refueling/Replenishing Energy

The battery charging rate declared by the manufacturer is about 43 kW using alternating current (AC) and 250 kW using direct current (DC) [46]. This gives the full battery charging time of 9.5 h for AC and 2.5 h with DC, respectively. It takes less than 13 min to fill up the Volvo FH500 with diesel fuel [18]. LNG refueling takes 3–5 min [18]. The need for extended EV charging time eliminates the use of the vehicle over long distances. The profitability of this type of solution in today's reality is far from being competitive with liquid or gaseous fuels. This problem may not be relevant in the context of short-distance permanent types of work.

Refueling Infrastructure

There are currently around 130,000 filling stations in Europe, and most of them offer the possibility of refueling with diesel. In addition, most petrol stations also offer various

services and amenities, such as car washes, grocery stores, and service points. However, in recent years, diesel has started to lose its popularity due to issues related to exhaust emissions and environmental impact. Consequently, an increasing number of European nations are endeavoring to augment the proportion of alternative fuels, such as LNG, CNG, or electricity, potentially resulting in modifications to the accessibility of refueling infrastructure. According to data from the European Environment Agency (EEA), there are currently around 600 LNG and BioLNG filling stations in Europe, with most of them located in countries such as Spain, France, Germany and Italy. However, in some countries, such as Poland, the Czech Republic and Slovakia, it is still rare. Nevertheless, over the next few years, it is planned to expand the LNG refueling infrastructure in Europe, especially in the heavy transport sector. Many European countries are striving to increase the share of alternative fuels in transport, and LNG is considered one of the most promising solutions. As part of such a process, more and more companies plan to build new LNG refueling stations, as well as expand the existing infrastructure. Most of these projects focus on creating so-called LNG and BioLNG corridors, i.e., a network of LNG refueling stations on selected routes in Europe, which are particularly important for heavy transport. The charging infrastructure for trucks in Europe is developing along with the growing interest in electric vehicles. There are currently around 365,000 charging stations, both public and private (Figure 15) [47].

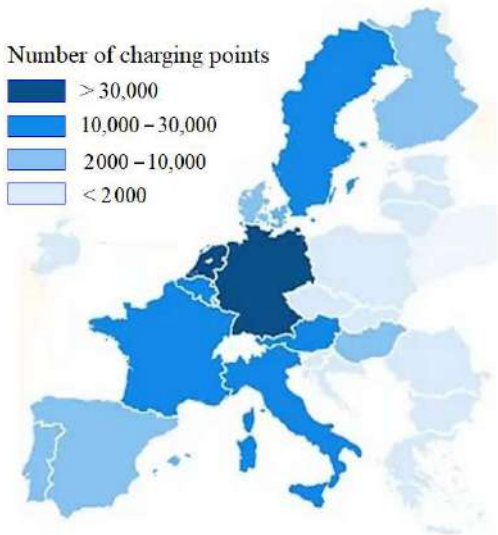


Figure 15. The distribution of EV charging stations in Europe. Drawing made by the authors based on available data found in the website [47].

Unfortunately, about 90% of charging stations, due to their location and/or installed capacity, are not able to handle heavy goods vehicles [48]. Volvo also offers a mobile charger for applications with three-phase power connections [20]. In certain European months, it allows an increase in the operability of this type of vehicle. Charging on your own, i.e., on the premises of companies, logistics hubs, and parking lots, is one of the solutions for truck fleet operators. At this point, the company's use of its own charging station, preferably in combination with renewable energy sources, gives the best results to save time and costs and establish market competition at certain distances. A similar situation applies to HVO and other ecological liquid fuels. At the moment, there are already developed technologies and power systems that make it possible to charge trucks with high power consumption. Along with technological progress, it can be expected that these technologies will become more and more advanced and available, which will allow for further development of electric motoring in Europe, and this is a clear trend. Unfortunately, e-limitations related to the energy consumption of long-distance heavy transport combined

with the low energy density of batteries indicate the lack of alternatives in this sector for chemical fuels, regardless of whether they come from renewable sources.

Toll Relief

For EV trucks, many countries offer various forms of toll relief, such as freeway toll exemptions or preferential toll rates compared to combustion vehicles. In Poland, electric vehicles are exempt from tolls on roads managed by the General Directorate for National Roads and Motorways [49]. Also a good example of favoring electric trucks is the German exemption from Toll Collect fees starting in December 2023 [50]. Similar discounts are also available for trucks using alternative fuels, such as natural gas, LPG, biofuels or hydrogen. Some countries offer toll exemptions or preferential rates, as well as tax credits. Toll credits for EVs and alternative fuel vehicles are good for the environment because they encourage the replacement of traditional internal combustion vehicles with greener alternatives [5]. At the same time, they help companies switch to greener solutions without incurring additional road toll costs [51].

Ecological Subsidies

In Europe, there are also programs to subsidize biofuel-powered vehicles and EVs. These programs aim to accelerate the transition to more sustainable and greener power sources in the transport sector. In Poland, the Clean Air program enables co-financing the purchase and installation of photovoltaic installations and other renewable energy sources that can be used to charge electric vehicles [52]. The Polish government also subsidizes the purchase of electric and hybrid cars, with the amount of subsidy depending on the vehicle category and battery capacity [53]. In other European countries, there are also programs to subsidize electric and biofuel vehicles. In Germany, there is the "Umweltbonus" [54] program that provides subsidies for the purchase of electric or hybrid cars, and in Sweden, the government offers scholarships for the purchase of electric and hybrid cars. With regard to vehicles powered by biofuels, in Sweden there is a program "Förnybart drivmedel" [55] that offers subsidies for the purchase of biofuels, and in Germany the government supports producers of biofuels through tax rates [56]. In the UK, there is also the "Renewable Transport Fuel Obligation" program, which obliges fuel distributors to supply increasing amounts of biofuels [57]. In other European countries, there are also programs to subsidize ecological trucks. In Belgium, the government subsidizes the purchase of natural gas trucks, and in the UK, there is the Clean Vehicle Retrofit Accreditation Scheme, which supports the modernization of existing trucks to meet exhaust emission standards [58].

3.4. Emission of CO_2

The average CO_2 emission per energy unit varies significantly depending on where it is produced and the energy mix used. Table 5 below shows the different emission values for different countries taking into account the energy mix:

Table 5. Structure of electricity production and intensity of carbon dioxide emissions for the listed countries. Table made by the authors based on available data found in the works [59,60].

	Electricity Production Mix [%]						Carbon Intensity, g CO_2-eq/kWh
	Renewables	Nuclear	Gas	Oil	Coal	Other	
Poland	21	0	3	0	69	7	657
France	20	72	5	0	2	1	48
EU average	33	25	19	2	20	1	269
China	25	4	3	0	65	3	555
India	19	2	5	2	71	2	708
USA	18	19	33	1	28	1	401
Canada	64	15	9	1	8	2	140

In the EU, the average electricity production is 0.269 kg CO_2-eq/kWh [60]. Due to the locality of the operation of electric trucks, data for Poland, France and the average for the EU will be shown in the results. Electricity transmission lines efficiency according to [61,62] is 88%. The electric efficiency of EV chargers is typically measured as a percentage of how much energy is transferred from the grid to the vehicle's battery during the charging process. The efficiency of a charger can vary depending on the charging rate, power output, and technology used. For the purposes of the calculation, we will take the efficiency of the charger at the average level of 90%. Therefore, the emissions resulting from the electricity supplied to the battery, well to wheels (WTW) emissions, are 0.829 kg in Poland, 0.061 kg in France and 0.340 kg for the average EU CO_2-eq/kWh.

Well-to-tank (WTT) and tank-to-wheel (TTW) emissions (which gives WTW emissions) consist of many variables. For the purposes of this study, the following data in Table 6 were selected.

Table 6. Well-to-tank, tank-to-wheels and well-to-wheels CO_2-eq emissions for diesel, HVO, LNG and BioLNG.

	Diesel [63]	HVO	LNG	BioLNG
WTT [CO_2-eq/kg]	0.76	−2.71	1.34	−2.48
TTW * [CO_2/kg]	3.18	3.10	2.75	2.75
WTW ** [CO_2-eq/kg]	3.94	0.39 [64]	3.09 [65]	0.27 [66]

* The design value resulting from the chemical reaction burns the given fuel according to [67]. ** For HVO, LNG, and BioLNG, the value is directly related to the change in WTW emissions compared to diesel fuel as indicated in sources.

Emissions should also include the environmental costs incurred to produce the vehicle. According to the study [68], the article assumes that the average emission from battery production is 150 kg CO_2-eq/kWh. Vehicle production (without batteries) produces about 5 kg CO_2-eq/every vehicle's kg [69], the same value as set for ICE vehicles.

Fuel/electricity consumption for ICEs and EVs was selected in accordance with Table 1 (page 2). Below are the calculated lifetime emission results, including the production and use of the car in its full lifetime (1.5 million kilometers) for individual drives in Figure 16.

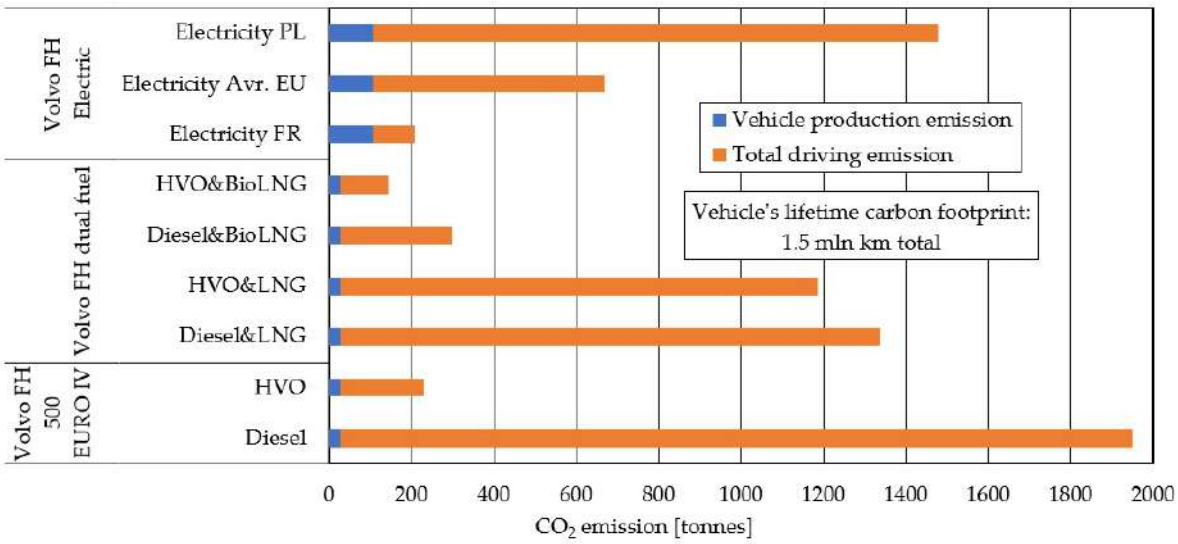


Figure 16. CO_2-eq footprint of the life cycle of selected vehicles for different fuels and electricity markets.

4. Discussion

The semi-trucks featured in the comparison are the currently available products on the open market. This demonstrates the manufacturer's recognition of the economic viability

and sales potential of each of these cars, and each of them adheres to relevant legislative restrictions.

The weight of the EV truck tractor, due to the very heavy and low-capacity electrochemical batteries, is greater than that of a typical tractor unit. In the comparison shown, however, this vehicle was not significantly heavier than the vehicles with ICE. Calculations showed that it was about 15% heavier. This turned into a difference of approx. 4.5% less payload. This is due to the battery's weight being compensated for by the weight of the large ICE and the power suitable for a heavy vehicle. In this comparison, the vehicles show a similar value in use and are characterized by a similar usability.

The power-to-weight ratio for an EV covers, to a large extent, the ratio for ICE vehicles, which is related to the possibility of using different operating modes of the drive system. This is dictated by the desire to extend the EV range due to the use of a lower power mode. In maximum power mode, it is definitely higher by (13–30% or more, depending on the load). The dynamics of the vehicle are also improved by the torque and power characteristics of the electric motors (flat maximum torque from 0 RPM), which, combined with the 12-speed i-shift gearbox, ensure high dynamics. This means that the electric vehicle can deliver maximum torque instantly, providing quick acceleration and excellent performance in stop-and-go traffic, such as in cities and mountainous roads. In addition, regenerative braking in electric vehicles recovers energy that would otherwise be lost during braking, contributing to a more efficient and smoother ride in choppy driving conditions. The graph under the dynamic characteristics curve for EVs has only a slightly larger area, just 2% larger than that of ICE; however, this does not fully capture the true driving dynamics of the vehicles as detailed earlier. Apart from the initial gear ratio, the differences between diesel (DF) and diesel engines are remarkably similar. At higher speeds, similar peak performance values of the electric drive and the ICE engine are noticeable. The higher maximum output torque in an ICE is only available within a certain narrow speed range. In the case of long-haul transport, the benefits resulting from the energy density of chemical fuels extending the range of the vehicle and the low potential of kinetic energy recovery indicate the use of ICE vehicles.

A brand new electric semi-truck is almost four times more expensive than a semi-truck powered by a diesel engine and over 2.5 times more expensive than a DF one. A small fraction of the costs at the moment are compensated by the simplified operation of the electric drive and the extended life of the braking system resulting from the implementation of the EV recuperation system. After taking into account the energy costs of operating the vehicle during its lifetime of 1.5 million kilometers, the EV takes the lead. DF vehicles outperform single-fuel configurations. Values of the total lifetime costs of vehicles are summarized in Figure 17 below.

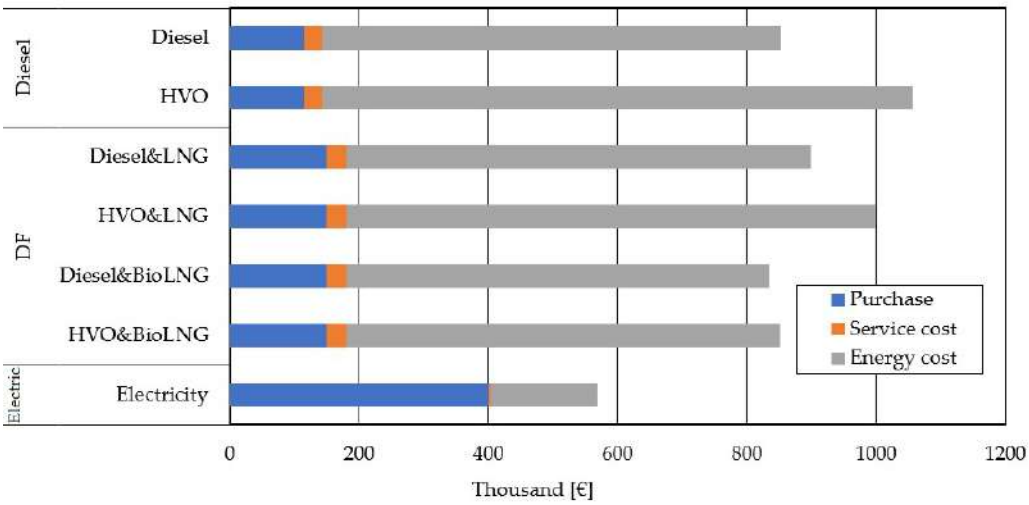


Figure 17. Total lifetime operation cost of compared vehicles.

The TKM costs for an EV are more than two times lower than for semi-trucks with ICE. DF vehicles have a TKM cost of an ICE diesel and an 11 to 39% lower cost than HVO-only.

A big problem in the case of a semi-truck EV is the time-consuming process of recharging energy, which, in the context of long-distance road transport, makes it impossible to use such a vehicle in an economically rational framework. This does not change the fact that these vehicles can be used on certain short routes in a pendulum pattern; in the market of local deliveries, it is perfectly justified. Due to the pressure and the need to eliminate fossil fuels, eco-driven vehicles are rewarded with purchase subsidies and exclusions from typical tolls. This puts DF solutions using alternative fuels as a potential candidate to fill the gap in long-distance heavy road transport. Unfortunately, this requires further investment in refueling infrastructure, which applies in particular to LNG and BioLNG fuels.

The use of fossil fuels to generate energy each time significantly increases the emission of carbon dioxide [70,71]. We are talking here about the variant of diesel combustion, using it in dual fuel in the form of natural gas or in the form of electricity generated from non-renewable sources. The lowest life cycle emission in the comparison was shown by the DF vehicle powered by HVO and BioLNG. It was better compared to an EV powered by the "cleanest" electricity due to the high greenhouse gas emissions correlated with the production of the vehicle's battery. The WTW GHG emissions of EVs depend on the electricity carbon intensity, decarbonizing is easier in areas with low-carbon electricity. Still, a zero-emission tailpipe indicates a social benefit in places of high population density.

The transition towards alternative power sources for semi-trucks, in relation to environmental concerns, is filled with difficulties. An essential concern is the urgent need to replace fossil fuels in the goods transport industry, given their harmful environmental consequences, which necessitate the adoption of alternative energy sources, such as vehicles that run on electricity or biofuels, which are the focus of this analysis. The issue of hydrogen, due to its low energy density and problems resulting from low efficiency in piston engines and big problems in maintaining an appropriate narrow ΔT on fuel cells [72], happens to currently not be a favored option among truck manufacturers, a fact mirrored by the absence of such vehicles in retail sales as well as ammonia powered option [73]. However, work on this type of drive is ongoing [74]. A similar situation applies to purely synthetic fuels. The low energy density of batteries, coupled with their considerable weight and prolonged charging times, presents seemingly insurmountable hurdles for long-haul heavy-duty transport. Regrettably, EVs may not be cost-effective for long-distance logistics. However, for urban and intercity travel within their maximum range (approximately ~400 km in our consideration), and everywhere else, regenerative braking is feasible (urban, suburban areas, mountainous terrain), and the low cost of electricity and minimal environmental impact render them unbeatable. EVs are already making inroads into the market for short-range distribution and local transport. For tractor units, the DF configuration offers a viable middle ground between range and ecological concerns. The operational costs of such vehicles are comparable to those of long-established diesel vehicles. While infrastructure access remains an obstacle, there are plans and ongoing efforts to improve this. The impact of ICE vehicles burning green fuels under the conditions of today's electricity generation infrastructure in Europe in most cases outweighs the environmental benefits of "zero emission" EVs. This is best seen in the total emissions CO_2 of a truck used, for example, in Poland, where, as of 2023, its full-lifetime emissions are only slightly better than the latest semi-truck diesel meeting the Euro VI standard. Despite this, the lack of emissions from EVs in urban settings is a distinct advantage from the standpoint of public health. In each case, the types of heavy vehicle drives considered in the article can be used today and have their own market, as shown by the structure of freight transport in the EU presented in Figure 18 below.

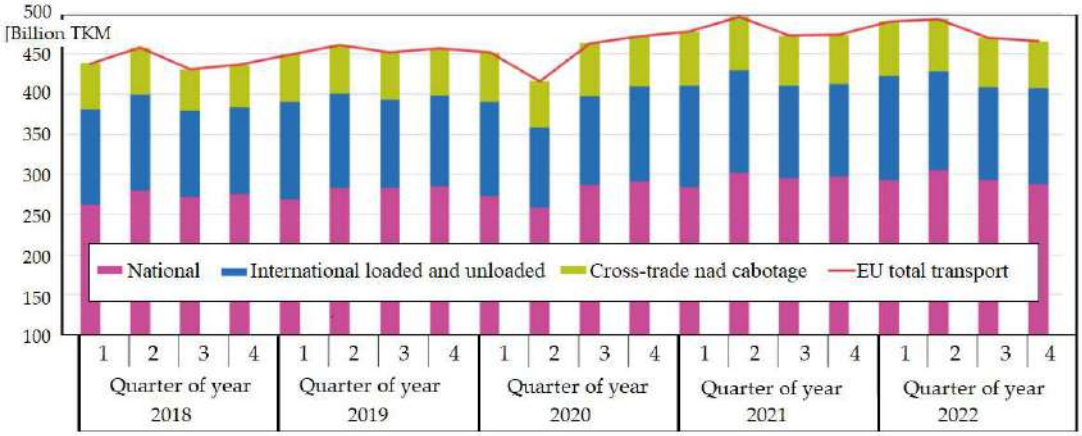


Figure 18. Quarterly road freight transport by type of operationm, EU 2018–2022 (bilion TKM). Drawing made by the authors based on available data found in the paper [56].

5. Conclusions

The article provides a comprehensive evaluation of the appeal of traditional and eco-friendly semi-trucks within the European road freight sector. The condensed conclusion summarized in the bullet points below discusses the balancing act between economic feasibility, functional effectiveness, and ecological consequences of different semi-truck drive options available in the retail market. These points illuminate a wider view of the transition toward greener transportation solutions, focusing on the interplay among regulatory compliance, environmental considerations, and the logistics of cost-effectiveness:

- EV truck tractors are about 15% heavier than typical ICE semi-trucks, translating to a ~1.5-ton lower payload, which is offset by the absence of a heavy ICE engine.
- The power-to-weight ratio of EVs is comparable to ICE vehicles, with electric semi-trucks providing better dynamics due to instant torque delivery and efficient performance in varied traffic conditions The highly dynamic EV loses its importance at higher cruising speeds.
- Despite being costlier upfront, EVs lead in energy cost savings over a lifetime of 1.5 million kilometers when compared to diesel and DF vehicles.
- Total lifetime operation costs for EVs are lower than ICE vehicles, with DF vehicles being a cost-effective middle ground.
- Long and often recharging for EVs is unacceptable for long-haul transport but is viable for short-haul and local deliveries, benefiting from subsidies and toll exemptions.
- The lowest lifecycle emissions were from DF vehicles using HVO and BioLNG, even when compared to EVs with "clean" electricity, due to battery production emissions.
- In regions with low-carbon electricity, EVs have a social benefit in densely populated areas due to zero tailpipe emissions.
- Despite the big challenges of alternative energy adoption for long-haul transport, EVs excel in urban and suburban transport within a range of ~400 km.
- The environmental impact of ICE vehicles using green fuels may outweigh the benefits of EVs in regions with carbon-intensive electricity infrastructure.
- ICE vehicles, thanks to their longer range due to high energy density and high ecological potential of ecological fuels, the lack of difficulties resulting from energy replenishment, and good bases for the rapid transformation of infrastructure, indicate further unrivaledness on long-distance routes. The potential of BioLNG and HVO could be applied to the EU's international road transport industry in the coming years, provided that enforcement does not change the trajectory of engineers and real market applications. Different types of heavy-duty vehicle propulsion systems already have distinct market niches, as evidenced by EU freight transport data.

Following this study, future work could take a broader perspective on the sustainability of automotive fuels and energy carriers such as synthetic fuels. Taking into account the limitations of raw materials and other resources. It also seems reasonable to conduct a broader comparison using more market information, i.e., vehicles from more manufacturers and more markets such as the USA, China, India, etc.

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Abbreviations

CNG	Compressed Natural Gas
DF	Dual Fuel
EEA	European Environment Agency
FH	model of Volvo semi-truck
FT	Traction force
EPT	Electric Propulsion Transmission
EU	European Union
EV	Electric Vehicle
ICE	Internal Combustion Engine
HP	Horse Power
HVO	Hydrotreated vegetable oil
LNG	Liquefied Natural Gas
LPG	Liquefied Petroleum Gas
MOT	an annual test of vehicle safety and roadworthiness aspects. The name derives from the Ministry of Transport
RPM	Revolutions Per Minute
TC	Turbo compound
TKM	Ton-kilometre
UK	United Kingdom

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III. WYKAZ DOROBKU NAUKOWEGO DOKTORANTA

Publikacje:

Liczba publikacji	pkt MEiN	h-index	Impact Factor
13	1000	3	12,8

- **Chojnowski J,** Nogas P. The Potential of HVO as a Highly Reactive Biofuel in Dual Fuel Systems. *Bulletin of the Military University of Technology.* 70, nr 2 (2021): 65-76. https://doi.org/10.5604/01.3001.0015.7011

- Karczewski M, **Chojnowski J,** Szamrej G. A Review of Low-CO2 Emission Fuels for a Dual-Fuel RCCI Engine. *Energies.* 2021; 14(16):5067. https://doi.org/10.3390/en14165067

- **Chojnowski J.** Safety in the use of car gas fuel installations. *Combustion Engines*. 2022;189(2):3-9. https://doi.org/10.19206/CE-142172

- **Chojnowski J**, Karczewski M. Analysis of the market structure of long-distance transport vehicles in the context of retrofitting diesel engines with modern dual-fuel systems. *Combustion Engines.* 2022;188(1):13-17. https://doi.org/10.19206/CE-142167

- **Chojnowski J,** Karczewski M, Szamrej G. The phenomenon of knocking combustion and the impact on the fuel exchange and the output parameters of the diesel engine operating in the dual-fuel mode (Diesel-CNG) *NAŠE MORE 2021 -CONFERENCE PROCEEDINGS* https://www.nasemore.com/wp-content/uploads/2024/04/NASE%20MORE%202021_Conference%20proceedings.pdf

- Szamrej G, **Chojnowski J,** Karczewski M. Solution of CNG and LNG supply systems in modern land and marine CI engines working in dual-fuel (NG-Diesel Mode) *NAŠE MORE 2021 - CONFERENCE PROCEEDINGS* https://www.nasemore.com/wp-content/uploads/2024/04/NASE%20MORE%202021_Conference%20proceedings.pdf
- Szamrej G, Karczewski M, **Chojnowski J**. A review of technical solutions for RCCI engines. *Combustion Engines.* 2022;189(2):36-46. https://doi.org/10.19206/CE-142551

- Karczewski M, Szamrej G, **Chojnowski J.** Experimental Assessment of the Impact of Replacing Diesel Fuel with CNG on the Concentration of Harmful Substances in Exhaust Gases in a Dual Fuel Diesel Engine. *Energies.* 2022; 15(13):4563. https://doi.org/10.3390/en15134563

- **Chojnowski J**, Karczewski M. Influence of the Working Parameters of the Chassis Dynamometer on the Assessment of Tuning of Dual-Fuel Systems. *Energies.* 2022; 15(13):4869. https://doi.org/10.3390/en15134869

- **Chojnowski J.** The concept of a mobile automated diagnostic and dynamometer station for heavy trucks. *Combustion Engines*. 2023;193(2):122-128. https://doi.org/10.19206/CE-163320

- **Chojnowski J.** Analysis of the influence of substitute fuels on properties operating conditions of military hybrid drive systems. *Combustion Engines*. 2024. https://doi.org/10.19206/CE-174553

- **Chojnowski J,** Karczewski M, Szamrej G. Dual-fuel engines using hydrogen-enriched fuels as an ecological source of energy for transport, industry and power engineering. *Combustion Engines*. 2024. https://doi.org/10.19206/CE-176800

- **Chojnowski J**, Dziubak T. Multi-Criteria Analysis of Semi-Trucks with Conventional and Eco-Drives on the EU Market. *Energies.* 2024; 17(5):1018. https://doi.org/10.3390/en17051018

Nagrody:

- 2-miejsce w konkursie BASF Poland – „Drive Innovation" (nagroda 10 000 zł) - 19.10.2022

- Nagroda Rektora WAT 2023: za osiągnięcia w zakresie działalności naukowej będące wynikiem badań obszarze możliwości zmniejszenia emisji szkodliwych składników spalin oraz gazów cieplarnianych przez współczesne silniki o zapłonie samoczynnym - 13.10.2023

Patenty:

Zgłoszenia patentowe w wyniku działań w Projekcie Mazowsze/0123/19-00,

- Stacja diagnostyczno-dynamometryczna do montażu i diagnostyki układów zasilania silników pojazdów ciężarowych, Nr zgłoszenia: P.445699, Autorzy: dr inż. Mirosław Karczewski – WAT, dr inż. Filip Polak – WAT, **mgr inż. Janusz Chojnowski** – WAT, mgr inż. Jarosław Kubajek – GASLUX, Tomasz Bogusz – GASLUX;

- Mobilne stanowisko magazynowo narzędziowe, Nr zgłoszenia P.445806, Autorzy: dr inż. Mirosław Karczewski – WAT, dr inż. Filip Polak – WAT, **mgr inż. Janusz Chojnowski** – WAT.

Konferencje:

- XXXVIII Międzynarodowe Seminarium Kół Naukowych Wydziału Mechanicznego 29-31.05.2019 – Wojskowa Akademia Techniczna;

- II Międzynarodowa Konferencja Nauki i Technologii Morskiej, Dubrownik, 17-18.09.2021 – Uniwersytet w Dubrowniku, Chorwacja;

- IX Międzynarodowy Kongres Silników Spalinowych PTNSS 27-28.09.2021 – Politechnika Lubelska;
- XVIII Konferencja automatyzacji i eksploatacji systemów sterowania i łączności ASMOR 2022 – 12-14.10.2022 – Władysławowo;
- X Międzynarodowy Kongres Silników Spalinowych PTNSS 19-21.06.2023 – Politechnika Wrocławska.

Praktyki/staże krótkoterminowe:

- IBiRM BOSMAL Sp. z o.o. Sarni Stok 93, 43-300 Bielsko-Biała – 08-19.08.2022;
- EuropeGAS Sp. z o.o. ul. Hurtowa 13 15-399 Białystok – 14-16.09.2022;
- AC S.A.,42 Pułku Piechoty 50, 15-181 Białystok – 26-28.10.2022;
- Uniwersytet w Vaasa,ul. Wolffintie 32, 65200 Vaasa, Filnlandia – 28.08-10.09.2023;
- Horus Energia, ul. Drobiarska 43, 05-070 Sulejówek – 15-28.09.2023.

Szkolenia:

- Obsługa impaktorowego licznika cząstek stałych PM w splinach DEKATI ELP+ oraz DEKATI e-Diluter Pro – 09.10.2020;
- Obsługa analizatora gazów spalinowych AtmosFIR AFS-B2T-C-2008 – 03.11.2020;
- Wymagania bezpieczeństwa przy budowie i modernizacji maszyn – Dyrektywa Maszynowa 2006/42/WE – 29-30.04.2021;
- Szkolenie z obsługi hamowni podwoziowych organizowane przez firmę Dyno Revolt – 10-12.01.2022;
- Obsługa programu „Fluent" organizowane przez Koło Naukowe „Silny" – 23-24.02.2022;
- Szkolenia dla zespołów pracujących nad innowacjami, Warmińsko-Mazurski Inkubator Przedsiębiorczości (on-line) – 25-29.04.2022;
- Diagnostyka pojazdów z wykorzystaniem diagnoskopu KTS Truck – 13.06.2022;
- Szkolenie Bosch i Intercars - Podstaw diagnostyki samochodów z napędem hybrydowym – 30.05.2023;
- Programowanie sterowników silników spalinowych MS Group – poziom podstawowy – 07.12.2023.

Inne:

- Uczestnictwo w Targach Poznań Motor Show – 7-10.04.2022 ;
- Studium przypadku koncernu Toyota jako globalnego przedsiębiorstwa funkcjonującego w różnych obszarach przemysłu i gospodarki organizowanym przez

Toyota Central Europe na Wydziale Zarządzania Uniwersytetu Warszawskiego – 10.05.2022;

- Targi GAS Show w Warszawie – udział w panelu dyskusyjnym – 20-21.10.2022;
- Przeprowadzone szkolenie: „Granty naukowe wyszukiwanie, sporządzanie wniosków" dla SDR WAT w ramach współpracy z Krajową Reprezentacją Doktorantów – 29.11.2022;
- Uczestnictwo w serii wykładów Koła naukowego Energetyków WAT: "LNG i jego ukryte problemy rynkowe„; „Gaz SF6"; „Sieci Elektroenergetyczne"; „SMR w Polsce" – 2022 i 2023 rok;
- Udział w targach: Solar Energy Expo - Targi odnawialnych źródeł energii – 17.01.2023;
- Udział w konkursie o Nagrodę Siemensa i Politechniki Warszawskiej „Dane dla środowiska" – 31.03.2023;
- Uczestnictwo w szczycie klimatycznym ToGetAir 2023 Warszawa – 20-21.04.2023;
- Uczestnictwo w Konkursie Jamesa Dysona – 08.05.2023;
- Uczestnictwo w sesji pitchingowej dla programu Horyzont Europa – technologie wodorowe – 12.12.2023.