

Article

Comparative Assessment of the Thermal Load of a Marine Engine Operating on Alternative Fuels

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Abstract: The decarbonization of the operational fleet through the implementation of renewable and low-carbon fuels (LCFs) is considered a key factor in achieving the regulatory greenhouse gas (GHG) reduction targets set by the IMO and the EU. In parallel with optimizing engine energy efficiency and emission characteristics during retrofitting for LCF operations, it is equally important to assess and ensure the reliability of engine components under permissible thermal and mechanical loads. This study investigated the key factors influencing thermal and mechanical stresses on the cylinder–piston assembly components as the engine’s operation shifts from diesel to biodiesel, natural gas, methanol, or ammonia. The methodological foundation of this research was an original comparative analysis method that evaluates the impacts of thermal stress and combustion cycle energy efficiency factors. The combustion cycle energy parameters were modeled using a single-zone mathematical model. The thermal load factor was determined based on the ALPHA (α_{gas}) coefficient of heat transfer intensity and the average combustion gas temperature (T_{avg}). The optimization of the combustion cycle during retrofitting was simulated without changes to the engine structure (or without “major” modernization, according to IMO terminology), with modifications limited to the engine’s combustion adjustment parameters. A key characteristic of the transition from diesel to LCFs is a significant increase in the maximum cycle pressure (P_{max}), a factor influencing mechanical stresses: ammonia, +43%; LNG, +28%; methanol, +54–70%; biodiesel, no significant changes. This study confirms the adopted strategy to maintain thermal load factors for engine components equal to D_{max} conditions. It is emphasized that, after ensuring $P_{\text{max-idem}}$ conditions, the thermal load during LCF operation aligns closely with the characteristic diesel level with minimal deviation. The thermal load reduction is associated with an increase in the excess air coefficient (λ) and a controlled reduction in the compression ratio within an allowable variation of ± 1 unit. Based on statistical correlations, a rational increase in λ was identified, reaching up to 2.5 units. Considering the real-world operational load cycle structure of marine engines, further research will focus on analyzing thermal and mechanical stresses according to ISO 81/78, as well as E2 and E3 operational cycles.

Keywords: decarbonization; operational diesel fleet; combustion cycle; mechanical and thermal load factors; parametric analysis



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1. Introduction

The decarbonization of maritime transport has become one of the primary challenges in its development in recent decades. IMO directive documents [1] have outlined a reduction in total CO₂ emissions from ships by 20–30% by 2030, ensuring that at least 5% of the energy

used in international shipping comes from zero- or near-zero GHG technologies, fuels, and/or energy sources. By 2040, CO₂ emissions should be reduced by 40–70%, and by 2050, the goal is to achieve net-zero emissions. One of the key solutions to achieving these targets is replacing petroleum-based marine fuels with renewable and low-carbon fuels (LCFs) [2]. To ensure a smooth transition to the LCFs, a series of new safety regulations were established on 1 January 2017 [3]. Meanwhile, the EU COM (2021) 562-final 2021/0210 (COD) directive [2] aims to reduce GHG emissions (CO₂, CH₄, and N₂O) by 55% by 2030 and achieve GHG neutrality by 2050. According to DNV GL data [4], approximately 51.3% of newly ordered ships are designed for LCF operation, with 40% of all new vessels being LNG-compatible. LNG is an attractive option due to its cost and well-developed infrastructure. During the transition period up to 2030, many existing vessel owners have already chosen this LCF type for the near future, as its gradual replacement with Bio-LNG and E-LNG in the medium term will ensure compliance with decarbonization plans by 2040.

Hydrogen is considered one of the key long-term solutions for decarbonization by 2050. However, the use of renewable energy sources allows for the production of e-diesel, which could achieve the 2050 GHG targets without the need for retrofitting. Nonetheless, the large amount of renewable energy required for e-fuel production means that intermediate LCFs, such as LNG, methanol, ammonia, and biodiesel, will inevitably be used [2].

Although 51.3% of ships designed from 2023 onward are adapted for LCF operation, 93.5% of currently operating vessels (by gross tonnage) still rely on petroleum-based fuels [4]. To meet the IMO and EU targets, these vessels must undergo retrofitting to integrate LCF-adapted technologies [5]. To encourage ship owners to reduce their ships' carbon footprint, the energy efficiency existing ship index (EEXI) went into effect on 1 November 2022 [6]. If ship owners do not comply with these rules, they will have to pay additional fees. A significant portion of research on LCF adaptation primarily focuses on evaluating and optimizing energy efficiency, greenhouse gas (GHG) emissions, and other harmful emissions, without investigating how the proposed changes will influence overall engine reliability. This study contributes to the development of an information platform that allows the simultaneous evaluation of reliability indicators while retrofitting to operate on LCFs, adjusting energy or environmental indicators. Most scientific studies have analyzed specific engine models, individual component designs, or the mechanical properties of materials, as well as structural modifications affecting thermal stresses and their reliability [7–9]. Some research has also focused on engine performance monitoring, digitalization, and early fault detection [10]. Studies have examined changes in engine component wear characteristics when using B100 biodiesel [11], as well as the thermal stress assessment capabilities of single- or multi-zone mathematical models using classical G. Woschni equations or similar ALFA formulas for various engine types and fuels [12–14].

However, from the perspective of a comprehensive approach to decarbonization, it would be rational to develop systematic solutions for LCF adaptation in existing fleets, aligning them with energy efficiency improvements and emission reduction strategies. From a practical standpoint, rational retrofitting solutions should be implemented in stages, starting with parameter optimization without major structural modifications to the engine—known as “minor” modernization, according to [15]—through optimizing only the adjustable engine parameters.

To contribute to solving the decarbonization challenges of the operating fleet, Klaipėda University (KU) has initiated a study aimed at developing a methodological framework for a combined assessment of reliability factors and performance indicators for LCF retrofitting [16]. Given the real conditions of retrofitting, the developed methodology is based on a single-zone combustion cycle engine mathematical model (MM method),

whose accuracy depends on the correct identification of heat release characteristics, particularly when changing fuel types, as required in the case of LCF adaptation [17,18].

In this context, the following literature review aims to identify the key combustion cycle parameters and optimization factors for the most widely used and promising LCF types—LNG, biodiesel, methanol, and ammonia. The analysis focuses on how these factors, including the air excess ratio (λ), compression ratio (ϵ), fuel injection angle (φ_{inj}), compressor pressure ratio (P_k), and manifold air temperature (T_k), among others, influence the adaptation of diesel engines to LCF operation. A significant portion of the LCF adaptation studies related to fleet decarbonization have focused on optimizing the LNG combustion cycle and reducing emissions [19,20]. As previously mentioned [4], LNG is considered one of the most promising fuel types for decarbonization in the short, medium, and long term. Its attractiveness for maritime use is driven by well-developed infrastructure, with over 190 operational and 80 emerging LNG bunkering stations in ports worldwide [21].

A substantial number of mathematical modeling (MM) studies [19,20] and retrofitting experiments have been conducted regarding the application of LNG in diesel engines already in operation. The authors of one study performed MM on the widely used Wärtsilä 9L50DF engine, determining combustion cycle parameters and emissions when operating the engine with both diesel and LNG [19]. The research was conducted under characteristic engine load conditions (50–75–100%), varying the pilot fuel (which has good self-ignition properties) injection angle and optimizing the compressor intake air pressure (P_k) to improve the engine cycle while ensuring the lowest GHG emissions. The study found that, when operating in an optimized LNG mode, the average combustion cycle pressure (P_{avg}) was lower compared to diesel operation, although the engine's effective performance parameters (break mean effective pressure, P_{me}) remained at the same level as in nominal diesel operation. Through optimizing the engine's operating regime, the researchers achieved an 85% reduction in NO_x emissions and a 25% decrease in CO_2 emissions. Additionally, it was determined that the optimal excess air coefficient (λ) in the nominal mode for LNG operation ranged from 2.1 to 2.3, the total heat release duration was 60 crank angle degrees (CAD) (i.e., 5 CAD less than with diesel), and the heat release rate was higher compared to diesel combustion. In another study [20], researchers used MM to investigate the combustion cycle of a two-stroke WinGD 7X82DF engine operating on LNG enriched with hydrogen. They analyzed the effects of different pilot fuel injection pressures (800/1000/1200/1400/1600 bar) and LNG gas injection pressures (10/12/14/16/18 bar) on the combustion process, GHG emissions, and residual methane emissions. The findings showed that increasing the pilot fuel injection pressure helps reduce residual methane levels. The optimal condition was achieved at 1400 bar, which resulted in an 8% reduction in residual methane in exhaust gases, reaching 9.2 g/kWh. The optimal LNG gas injection pressure was 12 bar, which further reduced residual methane levels by 10%.

According to statistical data from [4], methanol is currently the second most popular LCF in new ship orders. Leading companies such as Wärtsilä and MAN are producing a series of engines adapted for methanol operation. Additionally, numerous studies have been conducted on adapting diesel engines to run on methanol, where diesel is used only as a pilot fuel [22–25]. These studies have primarily focused on GHG emissions, combustion cycle analysis, and methanol utilization of up to 30% of its energy value. Researchers have examined the feasibility of methanol injection into the intake manifold versus direct cylinder injection, as well as the impact of injector hole arrangements on heat release and combustion processes. The findings indicate that methanol combustion exhibits a more active heat release than pure diesel. At nominal loads and $P_k = 1.2$ bar, combustion began approximately 5 CAD earlier than for $P_k = 2.4$ bar. The main heat release occurred around 7 CAD for 2.4 bar and approximately 20 CAD for 1.2 bar [22]. Moreover,

the total combustion duration was 5 CAD shorter compared to diesel operation under similar conditions. A series of studies on methanol utilization [26–28] have focused on developing 3D models to simulate the combustion process of methanol blends. In [28], the CFD modeling of the Wärtsilä 9L46C engine was partially compared with experimental results using a 30% methanol blend at a 75% engine load. The research determined that the critical threshold for methanol injection into the intake manifold is 50% of the fuel's energy content—exceeding this amount results in unstable combustion. Additionally, the study confirmed previous researchers' findings that, to reduce the risk of detonation, EGR (exhaust gas recirculation) must be at least 30% when introducing 50% methanol into the intake manifold, as recirculated gases help stabilize the combustion initiation and ensure smoother combustion. When using 5M5D-PI (50% methanol injected into the intake manifold and 50% diesel injected into the cylinder), 90% of heat release (CA90) occurred within 18.1 crank angle degrees (CAD), while 50% of heat release (CA50) happened within 7.5 CAD. In contrast, CA90 for diesel occurred at 30.1 CAD, meaning methanol combustion was approximately 67% faster than diesel under these conditions. When modeling the engine cycle with 9.5M0.5D-DI (95% methanol injected directly into the cylinder alongside diesel, where methanol is injected at -44 CAD and diesel at -7 CAD), the researchers found that for stable combustion, EGR must be around 15%, which is half the amount needed for manifold injection. Under these conditions, heat release was dynamic, with CA90 occurring at ~ 10 CAD and CA50 at 7.5 CAD, while the peak cycle temperature was reached at $+13$ CAD. However, without optimizing the combustion process, CA90 was reached in just 6.2 CAD and CA50 in 1 CAD. The 9.5M0.5D-DI model achieved a maximum NO_x reduction of 85% compared to diesel, along with a 4.2% higher thermal efficiency (TE). However, in the case of manifold injection, the TE was approximately 2% lower than that of diesel.

For medium- and long-term decarbonization goals [1], the transition is expected to move toward carbon-free fuels, one of which is ammonia. Ammonia is currently receiving significant scientific attention, as its combustion process produces minimal CO₂ emissions, aside from those from the pilot fuel. Another major advantage is the widespread infrastructure for ammonia reception and storage terminals, which is larger than that of methanol. Due to this, ammonia is being considered as a promising fuel for the future. However, the use of ammonia in diesel engines presents several challenges. Its combustion characteristics differ significantly from diesel and most LCFs. Ammonia has a low cetane number (5–7), a high autoignition temperature (650 °C), and a laminar combustion speed that is 6 to 12 times slower than diesel. These factors create challenges in ensuring a stable combustion cycle. When using ammonia, it is crucial to balance GHG, NH₃, and NO_x emissions in the exhaust gases.

For this reason, researchers have conducted chemical kinetics experiments [29,30] and developed mathematical models, validating their findings using optical single-cylinder engines. Studies [31,32] investigated the use of EGR in ammonia-fueled diesel engines and found that with 30% ammonia and 20% EGR, the maximum temperature (T_{max}) increased by approximately 15% compared to 0% EGR—rising from 1550 to 1800 K. Additionally, the heat release (CA05–CA90) extended by 5 CAD, from 33 to 38 CAD, while NO_x and NH₃ emissions decreased. However, CO₂ and CO emissions increased [32]. The authors concluded that at high ammonia concentrations, the use of EGR is not recommended, as it increases the risk of autoignition. One of the most common methods of ammonia application in diesel engines is direct ammonia injection in HDDF (high-density dual fuel) systems [33–35], which is associated with optimizing fuel injection phases. In [33], it was found that flame characteristics vary depending on the phase of the pilot diesel injection (SOPI). Early SOPI produces an orange flame due to radiation from pre-mixed combustion

products and suppresses diesel diffusion flames. Late SOPI results in a bright yellow flame due to carbon soot radiation and localizes the flame near the injector. An increasing diesel injection ratio (ROPI) reduces the peak heat release of ammonia and extends the combustion duration. In [34], 3D multi-zone models (MM) were used to achieve optimal thermal efficiency, reaching 47.22% efficiency with 80% of the fuel energy from ammonia injected directly into the cylinder. The lowest greenhouse gas emissions were achieved, with a 21% reduction, when ammonia was introduced via the intake manifold. Based on a detailed analysis of engine characteristics, the preferred mode is direct ammonia injection at an 80% energy ratio. Compared to pure diesel operation, this mode resulted in an increase of 4.10% in mean effective pressure, 2.57% in thermal efficiency, and an 8.31% reduction in greenhouse gas emissions. When injecting up to 80% ammonia into the intake manifold (PI), the CA0–CA90 heat release was similar to that of diesel at 48 CAD. However, when 80% ammonia was injected into the cylinder, CA0–CA90 decreased to 40 CAD. In [35], the authors modeled the Wärtsilä 6L46 engine's combustion cycle using a multi-zone MM AVL FIRE M approach. The engine operated on 95% ammonia and 5% diesel (A95D5), with ammonia directly injected at high pressure into the cylinder. The research was conducted in multiple stages. The effect of ammonia injection pressure on combustion was analyzed, and it was determined that the optimal injection pressure is 1000 bar. As a result, the combustion duration decreased from 90 to 33 CAD (compared to diesel), the thermal efficiency increased by 4.6%, and GHG emissions (expressed in CO₂ equivalent) were reduced by 24%. Additionally, the optimal fuel injection timing was determined to be -10° CAD BTDC for ammonia and -3° CAD BTDC for diesel (A710D717). Under these conditions, the peak pressure (P_{max}) increased by 9%, T_{max} rose by 10.5%, GHG emissions were reduced by 47%, and NO_x emissions decreased by 4.6%.

Comprehensive studies on biodiesel use in marine diesel engines [36–39] have focused on induction period determination, GHG emissions, optimization of combustion cycle parameters, and testing various fuel blends. According to EU regulations [2], next-generation biodiesel is classified as “Part A” and “Part B”:

- “Part A” is derived from lignocellulosic feedstock and is expected to constitute ~77% of biodiesel by 2030, with a projected increase to ~90% by 2050;
- “Part B” is produced from non-agricultural oils and animal fats.

The decarbonization effect follows the “Well-to-Wake” emissions calculation method, which considers GHG impact throughout the entire lifecycle, including fuel production, transportation, distribution, and onboard use. This method requires a minimum 68% reduction in emissions. Due to a lack of data on next-generation biodiesel applications in marine fleets, initial evaluations can be based on RRME studies [36–39]. Given its similar chemical composition to diesel (see Table 1), combustion cycle characteristics and energy efficiency metrics show no significant differences under equivalent power conditions [36–39]. This includes combustion duration (φ_z) and heat release form factor (m). The main advantage of using this LCF is its combustion similarity to diesel under nominal operating conditions, allowing seamless integration into existing fuel systems without major structural modifications [11].

The retrofitting of existing marine fleets for LCF operation is carried out in two main directions: optimizing combustion cycle energy efficiency and minimizing harmful emissions by adjusting combustion parameters (injection pressure and method, excess air ratio, EGR utilization, etc.) and assessing engine reliability factors, particularly mechanical stress and thermal load on components. While individual component failures have been analyzed, there is a lack of systematic solutions that integrate both combustion cycle optimization (efficiency and emissions) and reliability assessment compared to diesel operation.

Table 1. Comparison of the physicochemical properties of fuel types.

Fuel Type	Ammonia	Methanol	Biodiesel (RRME)	LNG	Diesel
Chemical formula	NH ₃	CH ₃ OH	CH ₃ (CH ₂) _n COOCH ₃	CH ₄	-
Density when liquefied, kg/m ³	602.8	792	890	430	832
Lower calorific value, MJ/kg	18.5	19.9	37.5	38.1	42.7
Octane number	110	109	60.6	107	-
Cetane number	5–7	5–8	>51	-	47–55
Ignition temperature, °C	651	385	>150	540	254–285
Flame spread rate, m/s	0.07–0.14	0.50	-	0.38	0.87
Heat of vaporization, kJ/kg	1370	1103	300	510	250–290
C, mass fraction	0.0	37.5	77.0	75.0	86.7
H, mass fraction	17.7	12.6	12.1	25.0	13.3
O, mass fraction	0.0	49.9	10.9	0.0	0.0
N, mass fraction	82.3	0.0	0.0	0.0	0.0

Regarding retrofitting solutions for existing marine fleets, the applied measures primarily focus on engine parameter adjustments that can be optimized under operational conditions. These include the compression ratio (ϵ), air fuel ratio (λ), compressor pressure (P_k), and combustion start angle (φ_{comb}). A broad statistical database has been accumulated on optimal combustion cycle energy efficiency indicators, particularly heat release characteristics when replacing diesel fuel with various types of alternative LCFs. Based on this foundation, combustion cycle optimization should be conducted in an integrated manner, considering the energy efficiency, environmental impact, and the mechanical and thermal load factors affecting engine component reliability.

Using a single-zone multi-model (MM) approach, the Wärtsilä 9L20DF engine's combustion cycle was simulated under diesel operation, determining key cycle parameters such as the maximum combustion pressure (P_{max}), average temperature (T_{avg}), mean indicated pressure (P_{mi}), and average combustion pressure (P_{avg}). These calculations were performed using factory engine settings (λ , ϵ , P_k , G_i , n , φ_z) [16]. During the KU reliability factor study, a dedicated research method was developed to assess the implementation of LCFs in operational fleet engines during retrofitting without requiring major structural modifications. Instead, the focus was on optimizing key adjustable parameters, such as the air fuel ratio (λ), boost pressure (P_k), and compression ratio (ϵ).

This study on mechanical and thermal load variations when transitioning from diesel to LCFs included the following objectives:

- Assess the thermal and mechanical load levels of cylinder–piston assemblies under nominal operating conditions, comparing them to diesel engine operations without altering the factory-set diesel operation parameters;
- Identify correlations between mechanical and thermal load factors and key combustion cycle parameters. The goal is to optimize these parameters for engines operating on a wide range of LCFs, as well as to determine the most effective strategies for reducing thermal stress in components.

2. Methodology

This study employed a strategy based on classical principles of engine component reliability and permissible stress assurance. It evaluated the mechanical and thermal stresses in the most heavily loaded parts of the cylinder–piston assembly (piston and cylinder head), which are influenced by combustion cycle parameters [9,12–14,16]. The

research primarily focused on thermal loading, considering that thermal stresses contribute up to 50–90% of the total stress balance [9]. The study assessed the engine's reliability factor when operating on LCFs through a comparative analysis with diesel operation, without requiring “major” structural modifications (acc MARPOL VI description [15]). Methodologically, when an engine operates on LCFs, mechanical and thermal loading factors are evaluated separately (see Figure 1). As actual stress data for components were unavailable, permissible mechanical and thermal stress levels were assessed based on influencing factors observed under diesel operation. Therefore, the focus was on the comparative analysis and potential reduction in thermal stresses. This study relied on combustion cycle modeling data, ensuring that the mechanical loading conditions remained equivalent to those of diesel operation, with $P_{max} = idem$ maintained. The adopted retrofit concept for low-carbon fuel engine operation solely focused on heat transfer factors, specifically the heat transfer coefficient (α) and the characteristic temperatures of the combustion cycle, as no design modifications were anticipated. Optimization involved fine-tuning the key engine parameters, including the fuel injection timing (φ_{inj}), boost air pressure (P_k), fuel air ratio (λ) (adjusted for boost pressure changes), and compression ratio (ϵ), all within the engine's permissible specification limits.

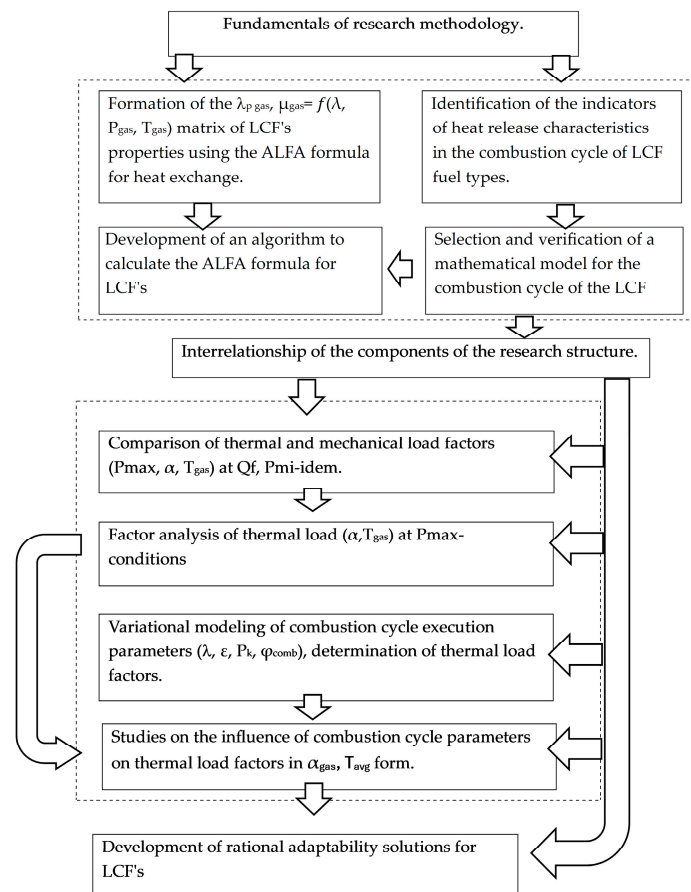


Figure 1. Research plan for the thermal and mechanical load factors of the components of a marine diesel engine while operating on LCFs.

2.1. Properties of the Investigated Fuel Types

This study aligns with the fleet decarbonization strategies of the IMO, EU, and classification society DNV GL [1,2,4], which outline technological adaptations for LCFs. In the near term (2030–2035), LNG will remain widely used but will gradually be replaced by Bio-LNG, with a long-term shift toward synthetic LNG. According to EU decarbonization principles [2], biodiesel is considered a promising alternative under the “Well-to-Wake” ap-

proach. Meanwhile, methanol adoption is accelerating in the maritime sector [4], whereas ammonia remains in the research phase, with potential long-term implementation [2,4]. The physical and chemical properties of the investigated LCF types are summarized in Table 1 [40].

The mathematical model of the combustion cycle accounted for the poor self-ignition properties of alternative low-carbon fuels such as ammonia, methanol, and LNG, which have a cetane number of 5–8, compared to 40–55 for diesel. Consequently, these fuels are used in diesel cycles with a pilot fuel—typically diesel or biodiesel—which provides the necessary ignition characteristics. Moreover, ammonia and methanol have less than half the lower heating value (LHV) of diesel, requiring a significant increase in injection volumes when implementing a high-pressure injection strategy for these LCFs.

Given the limited availability of key operational data for in-service engines, applying a multi-zone model based on the classical theoretical foundations of extended gas dynamics to the physical processes in the internal combustion engine (ICE) cylinder during the combustion cycle is challenging due to its reliance on numerous differential equation variables [41,42]. Consequently, it requires extensive research under controlled laboratory conditions and detailed graphical representations of the design geometry to ensure accurate results. Instead, this study utilized a well-established single-zone MM, commonly used for ICE analysis [43,44]. The model is structurally similar to AVL BOOST [45,46] and is based on fundamental conservation laws of energy, mass, and thermodynamic state variables (see Equations (1)–(3)).

$$\frac{dU}{d\tau} = \frac{dQ_{re}}{d\tau} - \frac{dQ_{ex}}{d\tau} - p \frac{dV}{d\tau} + h_{inj} \frac{dm_{inj}}{d\tau} - h_{ex} \frac{dm_{ex}}{d\tau}, \quad (1)$$

$$\frac{dm}{d\tau} = \frac{dm_{inj}}{d\tau} + \frac{dm_{ex}}{d\tau} - \frac{dm_{ex}}{d\tau}, \quad (2)$$

$$\frac{dp}{d\tau} = \frac{mR}{V} \frac{dT}{d\tau} + \frac{mT}{V} \frac{dR}{d\tau} + \frac{RT}{V} \frac{dm}{d\tau} - \frac{p}{V} \frac{dV}{d\tau}, \quad (3)$$

where U is internal energy (J); Q_{re} and Q_{ex} are heat release and heat exchange energy (J), respectively; p is the pressure (Pa); V is the volume (m^3); h is enthalpy (J/kg); m is the total mass (kg); m_s is supply air mass (kg); m_{inj} is the mass of fuel sprayed (kg); m_{ex} is the mass of exhaust gas (kg); τ is the time (s); R is the gas constant (J/kg K); T is the temperature (K).

The combustion cycle modeling was based on the I. Wiebe model (4) [47,48], with additions by G. Woschni ((5) and (6)) [49,50], which are widely used for modeling and studying the operating processes of internal combustion engines.

$$X = 1 - \exp \left[-6908 \left(\frac{\varphi}{\varphi_z} \right)^{m+1} \right], \quad (4)$$

where φ is the current angle of the cycle calculated from the beginning of the heat release; m is the combustion form factor; φ_z is the conditional burning time.

$$m = m_0 \left(\frac{\varphi_{i0}}{\varphi_i} \right)^{a_2} \left(\frac{p_{air}}{P_{air0}} \right) \left(\frac{T_{air0}}{T_{air}} \right) \left(\frac{n_0}{n} \right)^{a_1}, \quad (5)$$

$$\varphi_z = \left(\frac{\lambda_0}{\lambda} \right)^{a_3} \left(\frac{n_0}{n} \right)^{a_4}, \quad (6)$$

where a_1 , a_2 , a_3 , and a_4 are constants (in the diesel engine cycle, 0.8;0.5;0.6; and 0.5, respectively); φ_i is the induction period; P_{air} and T_{air} are the cylinder air portion pressure and temperature, respectively; n is the engine speed; λ is the excess air factor.

The corrections for the parameters m and φ_z , in comparison to diesel, were based on the results of a literature review focused on combustion cycle optimization for LCF operation.

2.2. Thermal Loading Factors of the Components

In alignment with the single-zone combustion cycle model and the widely validated analytical solutions of Molodtsov [51] at the Central Diesel Engine Research Institute (CDERI), the key thermal loading indicators for the components—specifically, the heat transfer coefficients—were defined using the ALFA Formula (7):

$$\alpha_{\text{gas av}} = \left(2.75 + 58.6 \frac{D}{C_m} \right) \frac{\lambda_{p \text{ gas}}}{\mu_{\text{gas}}^{0.5}} \left(\frac{C_m}{D} \right)^{0.5} \left(\frac{P_{\text{gas}}}{R_{\text{gas}} T_{\text{gas}}} \right)^{0.5}, \text{ W/(m}^2\text{K)}. \quad (7)$$

where $\alpha_{\text{gas av}}$ is the heat load coefficient during the cycle; D is the piston diameter (m); c_m is the piston mean velocity (m/s); P_{gas} and T_{gas} are the gas pressure (N/m²) and temperature (K), respectively; R_{gas} is the universal gas constant (J/(kg·K)); $\lambda_{p \text{ gas}}$ is the gas thermal conductivity coefficient (W/(m·K)); and μ_{gas} is the gas dynamic viscosity coefficient (kg/(m·s)).

The formula is based on classical similarity theory criteria. The model proposed by G. Woschni [52] is commonly used to solve practical problems and has been extensively validated on different types of ICEs operating over a wide load range and on different types of fuels. These types of formulae describe the intensity of heat release (Nu), the movement of the working fluid around the component (Re), and the physical and chemical properties of the working fluid (Pr) number. Due to the varying chemical compositions of LCFs (see Table 1), the $\frac{\lambda_{p \text{ gas}}}{\mu_{\text{gas}}^{0.5}}$ factor indicates the impact of the working fluid composition in the cylinder on the engine's thermal load of components, while $\left(\frac{P_{\text{gas}}}{R_{\text{gas}} T_{\text{gas}}} \right)^{0.5}$ reflects the influence of thermodynamic parameters during the combustion cycle. In the α formula, the parameters $\lambda_{p \text{ gas}}$ and μ_{gas} are calculated additively, based on the properties of the combustion product components, corresponding to the thermodynamic values of P_{avg} and T_{avg} during the combustion cycle:

$$\lambda_{p \text{ gas}} = r_{\text{CO}_2} \times \lambda_{\text{CO}_2} + r_{\text{H}_2\text{O}} \times \lambda_{\text{H}_2\text{O}} + r_{\text{O}_2} \times \lambda_{\text{O}_2} + r_{\text{N}_2} \times \lambda_{\text{N}_2}; \quad (8)$$

$$\mu_{\text{gas}} = r_{\text{CO}_2} \times \mu_{\text{CO}_2} + r_{\text{H}_2\text{O}} \times \mu_{\text{H}_2\text{O}} + r_{\text{O}_2} \times \mu_{\text{O}_2} + r_{\text{N}_2} \times \mu_{\text{N}_2}. \quad (9)$$

where r_i is the relative mass fraction; λ_i is the coefficient of thermal conductivity of the gas; μ_i is the dynamic viscosity of the gas.

The component r_i values are determined based on the chemical oxidation equations of the fuel types. The structural element of the ALFA formula, $\frac{\lambda_{p \text{ gas}}}{\mu_{\text{gas}}^{0.5}}$, formed from the physical parameters of the combustion material, $\lambda_{p \text{ gas}}$ and μ_{gas} , reveals the impact of the working fluid's composition (due to the different chemical compositions of LCF types) on the intensity of heat. The ALFA (α) formula applied in this study can be used in two parallel methods for practical implementation: calculating its instantaneous values during the cycle or using the average thermodynamic parameters of the working body, P_{avg} and T_{avg} . The decarbonization research was based on the second approach.

2.3. Research Object

The Wärtsilä 9L20DF diesel engine, widely used in the maritime industry, was selected as the research object [19,20]. Manufactured by the Finnish company Wärtsilä, it operates on both diesel and LNG fuels at medium speeds (see Table 2, [53]). Under nominal conditions, the engine generates 1665 kW of power at 1200 RPM for both fuels. This engine can serve as the main engine ("ME" refers to a scenario in which the engine is directly connected

to a constant-pitch propeller; “DE” denotes a configuration where the propeller’s pitch is adjustable, while the engine speed remains constant) or as an auxiliary generator (AE). The parameters of the main research object are provided in Table 2.

Table 2. Wärtsilä 9L20DF technical data.

Wärtsilä 9L20DF		AE/DE		ME	
		Gas mode	Diesel mode	Gas mode	Diesel mode
Cylinder output	kW		180		
Engine speed	rpm		1200		
Speed mode		Constant		Variable	
Engine output	kW		1665		
Cylinder bore	mm		200		
Stroke	mm		280		
Compression ratio			13.4		
Piston displacement	l/cyl		8.8		

According to the reliability indicators, the anticipated optimization of engines involves changes in the intake air pressure (P_k), air excess coefficient (λ), compression ratio (ϵ), and injection angle (φ_{inj}) for both the pilot fuel and LCFs.

2.4. Variational Research Plan

Improving the energy efficiency and reducing emissions in diesel engines is primarily achieved through combustion cycle optimization [19,20,28,31–35]. Methods to increase the indicated thermal efficiency (η_i) of diesel engines are directly associated with changes in the heat release characteristics in the cylinder. In [54–56], it was shown that under the conditions of $P_{max} = idem$ and $\lambda = idem$, the form of the heat release law (according to the I.Wibe model, factor m) [48] has a negligible impact on the thermal efficiency of a diesel engine. The main factor influencing η_i is the duration of the combustion cycle or the heat release characteristics (φ_z), which is largely determined by the excess air coefficient. This principle served as the foundation of the research strategy.

The research strategy included two consecutive stages:

- In the first stage, the thermal load on the components was evaluated based on the multiplication of $\alpha_{gas\ av} \times T_{avg}$, and the mechanical load was assessed using P_{max} as an indicator. No changes were made to the adjustment parameters compared to diesel operation. The aim was to assess, through a comparative approach, the thermal and mechanical load factors of components when the engine operates on LCFs versus diesel while identifying effective directions for combustion cycle optimization;
- In the second stage of the research, a detailed evaluation and optimization of the thermal load were conducted by analyzing the impact of the compression ratio (ϵ) and the air excess coefficient (λ) in variable models (with P_{max} held constant and adjusted by modifying the combustion start angle φ_{comb}). The objective was to validate the effectiveness of these parameters in influencing the thermal load on engine components and to generalize their application using the comparative thermal load evaluation methodology developed in [16]. The comparative assessment was graphically represented in the coordinate system of α_{gas} and T , considering various combinations of (λ, ϵ)—var (P_{mi} -idem, P_{max} -idem).

3. Results

This study examined two phases of engine conversion to LCF: first, replacing diesel with LCFs without modifying the primary engine adjustment parameters, and second, optimizing the interrelated parameters ($\epsilon, \lambda, \varphi_{comb}$, and P_k).

3.1. Impact of Diesel Replacement with LCFs on Performance Parameters

The studies simulate an initial conversion phase where the engine is converted to LCFs (at P_{mi} -idem) to evaluate the changes in energy efficiency (η_i) and mechanical and thermal load factors compared to the parameters specified for diesel operation. The replacement of diesel with LCFs was conducted under the Q_f -idem condition, ensuring that the introduced fuel maintained the same energy potential and released an equivalent amount of heat during combustion. Across all evaluated scenarios, the propulsion system's power output remained constant when operating on LCFs ($n = \text{const.}$, based on P_{mi} , the mean indicated pressure). Compared to the specified load regime, P_{mi} deviations remained within -1% to -5% (see Table 3 and Figure 2). These findings provide a foundation for identifying effective optimization strategies for adapting operational marine diesel engines to LCFs. The changes in the energy efficiency parameter (η_i) do not exceed ± 0.012 (or 2.5%). However, the mechanical and thermal load factors have significantly increased compared to diesel operation.

Table 3. Diesel replacement with LCFs simulation results on performance parameters (ϵ and λ equal).

Test	Parameters					
	P_k , bar	P_{mi} , bar	P_{max} , bar	T_{max} , K	η_i	$\alpha_{gas} \times T_{avg}$, %
Diesel	4.35	23.8	176.7	1586	0.476	100.0
Biodiesel	4.42	23.5	177.9	1558	0.476	99.0
Ammonia Gi = QD	4.13	23.5	253.2	1759	0.490	102.3
Methanol Gi = QD	4.14	24.9	301.0	1940	0.499	110.6
Methanol (m) Gi = QD	4.14	25.0	273.1	1860	0.502	111.6
LNG Gi = QD	4.39	24.9	226.9	1666	0.493	104.5

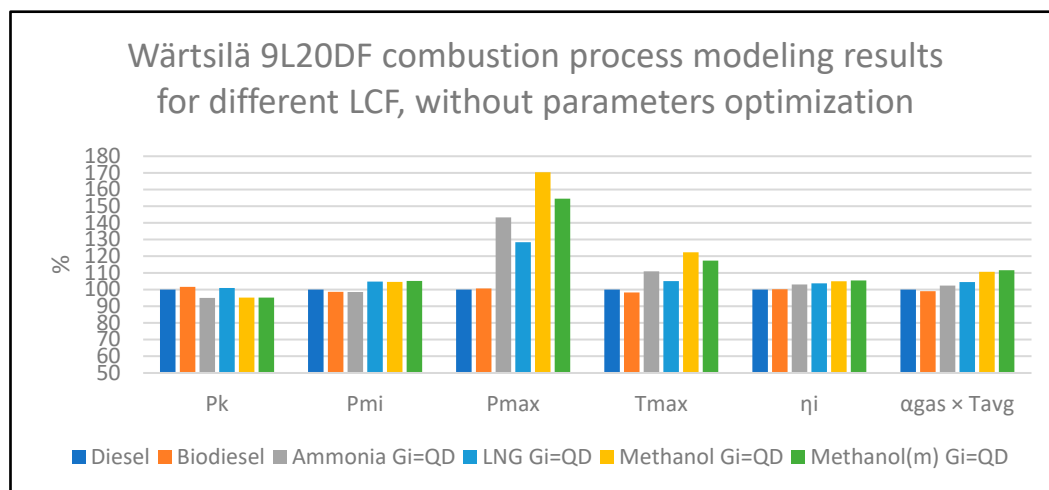


Figure 2. Impact of switching from diesel to LCFs on the Wärtsilä 9L20DF engine performance with ϵ and λ set to diesel values.

The obtained results, excluding biodiesel, indicate that reducing P_{max} should be a key objective in combustion cycle optimization.

Methanol. Under equal conditions (ϵ , λ , and φ_{comb}), the most significant P_{max} increase was observed with the use of methanol. Compared to diesel operation, where P_{max} is 176 bar, switching to methanol raised P_{max} to 301 bar—an increase of 70% (considering direct methanol injection into the cylinder [27,28]). A major factor contributing to the increase in P_{max} was the fuel injection method. Studies have shown that direct methanol injection into the cylinder significantly increases P_{max} . However, alternative injection strategies—such as port fuel injection (PFI) into the intake manifold or exhaust gas recircula-

tion (EGR)—can help moderate this effect. The use of an EGR system enables a single-phase heat release, which can limit the Pmax increase to 54% (methanol (m) Gi = QD) [28]. When replacing diesel with LCFs at the same λ , the indicator efficiency (η_i) showed the highest increase with methanol under single-phase heat release (5.5%), followed by operation without EGR (4.8%), ammonia (3%), and LNG (−3.3%). For biodiesel, η_i remained comparable to that of diesel. The increase in thermal load on the components (evaluated using the conditional criteria $\alpha_{\text{gas}} \times T_{\text{avg}}$) compared to diesel usage was 10.6% for methanol without EGR and 11.6% with EGR.

Ammonia. This study evaluated only the high-pressure direct injection of ammonia into the cylinder, aimed at improving decarbonization performance and reducing harmful emissions [34,35]. As with methanol, a significant increase in Pmax of 43% was observed, reaching 253 bar (compared to 176.7 bar with diesel), while η_i increased to 0.490 (2.5% within the margins of error), and the thermal load on the components increased by 2.3%.

Natural Gas. The Wärtsilä 9L20DF engine is designed to operate on both diesel and LNG, with some data provided in the engine's technical specifications. Although it is noted in [19] that the engine operates with a reduced λ of 2.1–2.3 in LNG mode (compared to $\lambda = 2.5$ for diesel), the modeling for the first phase of this study was performed with $\lambda = 2.5$. An increase in λ to 2.5 from the optimal level was associated with increases in all evaluated parameters: Pmax by 28%, and thermal load on components by 4% (from a practical standpoint, changes in P_{mi} and η_i are not significant). This significant increase in Pmax requires reducing Pmax and also managing thermal stresses.

Evaluation of the changes in the structural components of the thermal load indicators. The goal of the analysis is to evaluate the importance of mutual changes in structural components of the thermal load ALFA factor due to changes in physical parameters $\frac{\lambda p_{\text{gas}}}{\mu_{\text{gas}}^{0.5}}$ and combustion cycle parameters $\left(\frac{P_{\text{gas}}}{R_{\text{gas}} T_{\text{gas}}}\right)^{0.5}$. The data on the changes in the components of the thermal load factor ($\alpha_{\text{gas}} \times T_{\text{avg}}$) for the engine operating on LCFs are shown in Table 4.

Table 4. Changes in thermal load factor components ($\alpha_{\text{gas}} \times T_{\text{avg}}$).

Test	P_{avg}	T_{avg}	$(\lambda_{\text{gas}}/\mu_{\text{gas}})^{0.5}$	$(P_{\text{gas}}/(R_{\text{gas}} \times T_{\text{gas}}))^{0.5}$	α_{gas}	$\alpha_{\text{gas}} \times T_{\text{avg}}, \%$
Diesel	65.7	1087	1.11	14.48	457	100.0
Biodiesel	66.2	1072	1.10	14.64	458	99.0
Ammonia Gi = QD	69.7	1077	1.14	14.51	472	102.3
Methanol Gi = QD	72.7	1114	1.16	15.09	497	110.6
Methanol (m) Gi = QD	75.5	1128	1.15	14.89	487	111.6
LNG Gi = QD	70.9	1087	1.13	14.89	477	104.5

When directly converting a diesel engine to operate on LCFs, only biodiesel met the comparative criteria affecting the engine's mechanical reliability, Pmax, and the relative thermal stress criteria ($\alpha_{\text{gas}} \times T_{\text{avg}}$), with deviations of both parameters not exceeding 1%. However, the direct use of other fuels without optimization significantly exceeded the established reliability standards (see Table 4 and Figure 2). Based on the adopted strategy (where the results are evaluated comparably), the optimization principle should ensure that the thermal load factor values do not exceed those of diesel operation. The increase in thermal load is caused by changes in both structural parts of the ALFA factor related to higher Pmax and T_{avg} . Based on this, the hypothesis is proposed that ensuring Pmax-idem equal to diesel operation will have a positive impact on reducing the thermal load (see Section 3.2).

3.2. Optimization of Component Thermal Load Factors

To test the hypothesis, the combustion cycle was optimized by modifying the adjustment parameter (φ_{comb}), ensuring $P_{\text{max-idem}}$ at the specified level while operating on diesel. Simultaneously, variational simulations of the main combustion cycle parameters (λ , ϵ , P_k) were conducted to identify the mutual influence of these parameters on factors that affect reliability. This contributes to the development of an informational platform, which serves as one of the components for performing the engine retrofitting to run on LCFs in a comprehensive manner, based on energy, ecological, and reliability indicators. The optimization of combustion cycle parameters involved varying the combinations of λ (adjusting P_k) and ϵ , ensuring that P_{mi} and P_{max} remain consistent. All the adjustments were made within the engine's technical specification range (typically ± 1 unit). In all test modes, the aim was to maintain the engine's rated power with an average indicated pressure of $P_{\text{mi}} = 23.8$ bar. The change in the thermal load factor of the engine components was assessed in two ways: using the conditional heat exchange criteria ($\alpha_{\text{gas}} \times T_{\text{avg}}$) and separately considering the heat exchange parameters, α_{gas} and T_{avg} , by implementing the methodological format created by the authors [16].

A summary of the investigation results is presented in Appendix A Table A1, and Figure 3 illustrates the influence of factors affecting the thermal load on the ($\alpha_{\text{gas}} \times T_{\text{avg}}$) criteria in graphical form. The data in Appendix A Table A1 allow for predicting the most efficient retrofitting approach, if necessary, by reducing the thermal load on the components. It also enables the evaluation of the impact of optimizing energy and environmental performance on the thermal load, compared to the diesel option, in the form of $(\alpha_{\text{gas}} \times T_{\text{avg}}) = f(\lambda, \epsilon)$, and identifies the rational ranges for changes in variational parameters.

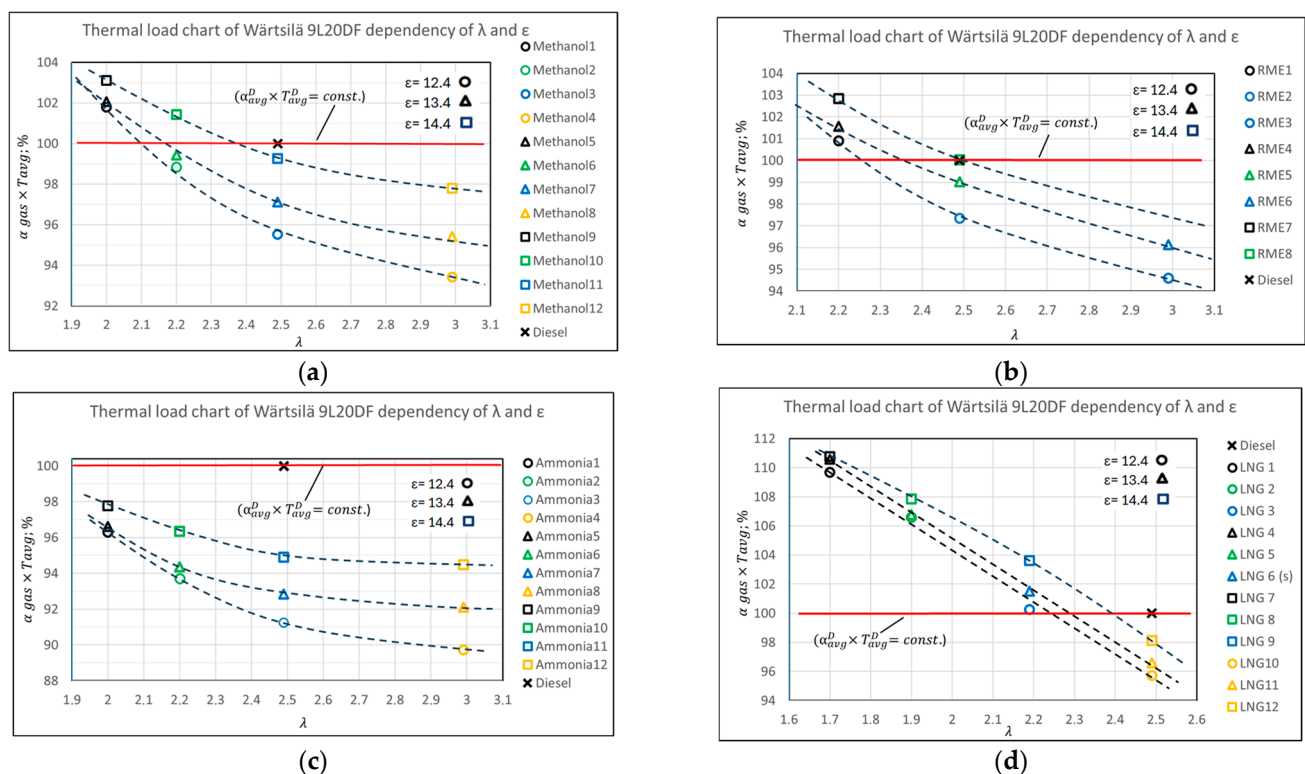


Figure 3. Dependence $(\alpha_{\text{gas}} \times T_{\text{avg}}) = f(\lambda, \epsilon)$ for different LCFs ($P_{\text{max}} = \text{cons}$): (a) methanol; (b) biodiesel; (c) ammonia; (d) LNG.

A clear relationship was established between $(\alpha_{\text{gas}} \times T_{\text{avg}})$ and the parameters λ and ϵ , with consistent characteristics observed across all evaluated LCF types (see Figure 3):

- Increasing the excess air coefficient (λ) led to a continuous decrease in the $(\alpha_{\text{gas}} \times T_{\text{avg}})$ criterion for biodiesel and LNG across the entire λ variation range (1.7–2.5). Meanwhile, as (ϵ) increased, $(\alpha_{\text{gas}} \times T_{\text{avg}})$ also increased, indicating a rise in the thermal load on the components;
- It is important to note that the influence of (λ) on $(\alpha_{\text{gas}} \times T_{\text{avg}})$ for methanol and ammonia was not linear: it remained roughly linear up to $\lambda \approx 2.5$ but stabilized asymptotically for $\lambda > 2.5$. The main reason for this behavior lies in the difference in the heat release duration (φ_z) between ammonia and methanol, on the one hand, and biodiesel and LNG, on the other. The combustion duration φ_z for methanol and ammonia (40–33 crankshaft degrees) was significantly shorter than that of LNG and biodiesel (60–65 crankshaft degrees). As a result, the absolute change in φ_z at $\lambda \geq 2.5$ had a lesser impact on the indicator diagram and, consequently, on $(\alpha_{\text{gas}} \times T_{\text{avg}})$.

Based on the results obtained, an important conclusion regarding the LCF retrofit procedure can be drawn: increasing the λ value of the engine beyond 2.5 is not an effective method for reducing the thermal loads on the components. Moreover, the effectiveness of increasing λ decreased as ϵ increased, which is typical for modern diesel engine types. For instance, in the conducted studies, at $\epsilon = 12.4$, increasing λ (up to 2.5) resulted in a ~5% reduction in the $(\alpha_{\text{gas}} \times T_{\text{avg}})$ criteria for ammonia, while at $\epsilon = 14.4$, this reduction decreased to ~2.9%. This suggests that increasing λ for engine conversion to LCF operation is reasonable when applied with acceptable ϵ values, in line with the criteria for limiting changes in energy and environmental performance. Below are the detailed results of applying the evaluated LCF types instead of diesel fuel based on the heat exchange factor $(\alpha_{\text{gas}} \times T_{\text{avg}})$.

Natural Gas. In the studied λ range (1.7–2.2), identified based on the dual-fuel engine manufacturer's specifications and energy performance improvement studies [19,20], the thermal load criteria exceeded the values observed in the diesel operation mode (see Figure 3). The maximum $(\alpha_{\text{gas}} \times T_{\text{avg}})$ increase compared to diesel operation was up to 10.7%. Increasing λ to the diesel level of 2.5 (with $\epsilon = 13.4$) resulted in a ~3% reduction in $(\alpha_{\text{gas}} \times T_{\text{avg}})$ compared to diesel operation. Therefore, for natural gas operation, considering reliability, it is recommended to maintain λ within the upper range of 2.0–2.2, and this should be implemented alongside a reduction in ϵ .

Biodiesel. In the primary λ range of 2.5–3.0, the thermal load was up to 5% lower than diesel operation with the specified variant. However, at the lower end of the λ range (2.2), $(\alpha_{\text{gas}} \times T_{\text{avg}})$ increased by 2–3% compared to diesel operation, with λ and ϵ reaching the specified values—essentially the same as diesel operation (see Figure 3b).

Ammonia. In the studied λ range of 2.0–3.0, for all tested λ and ϵ combinations, the $(\alpha_{\text{gas}} \times T_{\text{avg}})$ level was 2–10% lower compared to diesel operation (see Figure 3c), creating favorable conditions for retrofitting the engine by focusing the main optimization efforts on improving energy and environmental performance. It should be noted that, in contrast to the combustion cycle modeling for the engine operating on natural gas and biodiesel, the heat release characteristics, as indicated by the I. Wiebe model parameters m and φ_z did not change with variations in λ . This is because, with a fixed injection phase difference between liquid ammonia and diesel (~7 crankshaft degrees), the phase shift from the top dead center (TDC) is characterized by auto-modeling of fuel combustion, meaning that m and φ_z remain unchanged [35]. A similar solution with fixed m and φ_z is characteristic of the evaluation case for methanol use [28]. Changes in the heat release characteristic parameters (from $m = 0.5$ to 0.8 and $\varphi_z = 33$ to 50 crankshaft degrees, see Table A1) when modeling a lower pressure ammonia injection strategy into the cylinder affected $(\alpha_{\text{gas}} \times T_{\text{avg}})$ by no more than ~3%.

Methanol. The distribution of $(\alpha_{\text{gas}} \times T_{\text{avg}})$ values in the evaluated range of $\lambda = 2\text{--}3$ and $\varepsilon = 12.4\text{--}14.4$ was symmetrically half of the diesel limit distribution. For a combination of λ and ε similar to diesel, the $(\alpha_{\text{gas}} \times T_{\text{avg}})$ criteria value aligned with the specification variant criteria values, showing a $\sim 3\%$ similarity (see Figure 3a). Under the same λ and ε boundary conditions, the engine's thermal load when operating on methanol was 2.9% lower compared to diesel.

Thus, optimization in the examined λ and ε range, potentially based on the engine's energy and environmental parameters, is not limited by reliability indicators. The comparative analysis of $(\alpha_{\text{gas}} \times T_{\text{avg}}) = f(\lambda, \varepsilon)$ provides a foundation for identifying optimal combustion cycle conditions for LCF operation. This includes determining the combinations of excess air coefficient and compression ratio that maintain acceptable thermal and mechanical loads on the components. It would be beneficial to extend the analysis by separately examining the influence of the heat exchange parameters, α_{gas} and T_{avg} , in comparison to the diesel variant, on the components' thermal load. This approach will help us better understand the key parameters influencing thermal conditions, specifically the intensity of heat exchange factors, α_{gas} , and the temperature gradient of T_{avg} (see Figure 4).

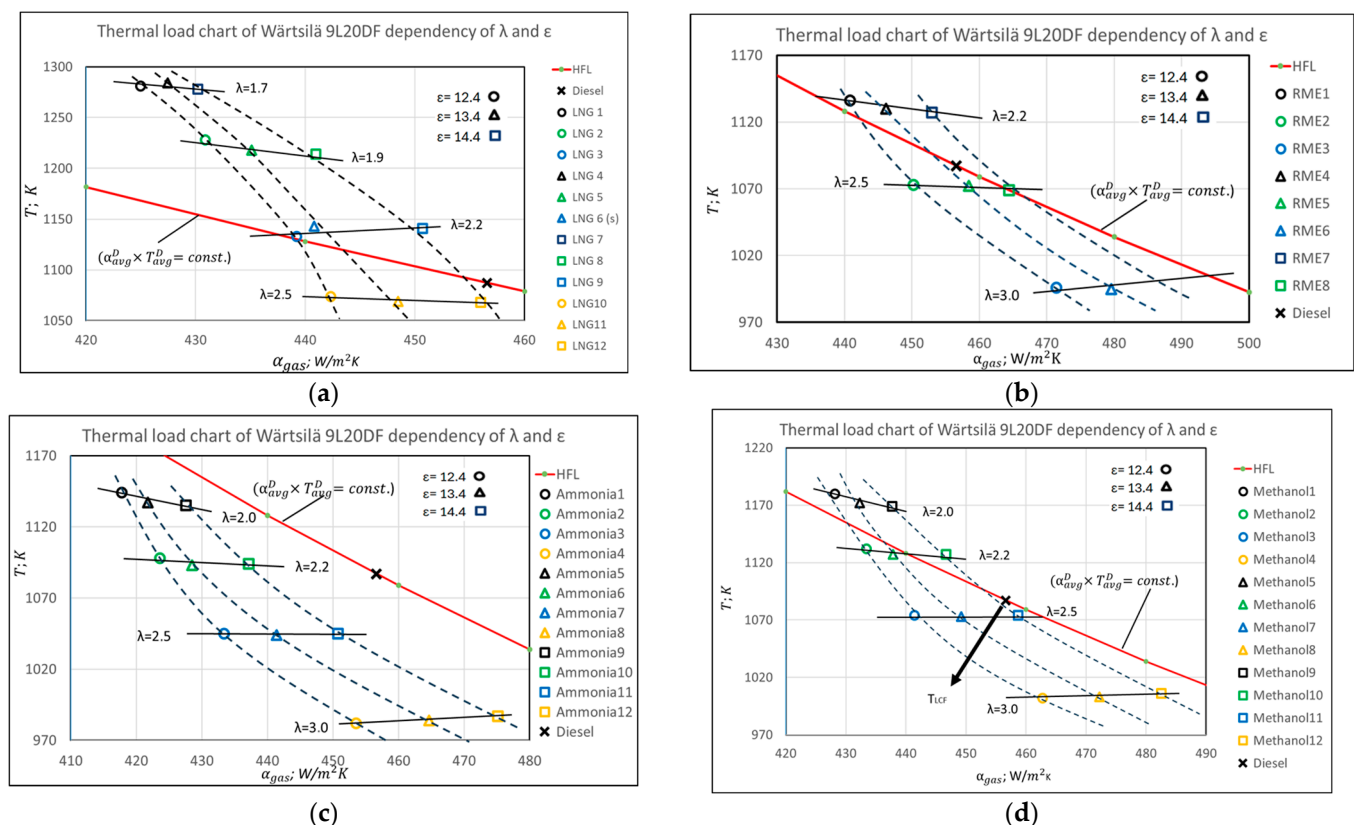


Figure 4. Reducing the thermal load factor potential of the Wärtsilä 9L20DF engine by optimizing the λ - ε parameters: (a) LNG; (b) biodiesel; (c) ammonia; and (d) methanol.

In general, the characteristics of all evaluated LCF types showed that the excess air coefficient, as the main performance parameter of the combustion cycle, has an opposite effect on the heat transfer parameters: an increase in λ reduced T_{avg} , while conversely, it led to an increase in α_{gas} . When λ was kept constant (related to the P_k adjustment), a change in ε , without altering T_{avg} , directly increased α_{gas} , with this effect becoming more pronounced as λ increased (see Figure 4). The result is determined by the deformation of the combustion cycle's indicator diagram shape, which is linked to the decrease in P_{avg} , which in turn affects the ALFA formula $(P_{\text{gas}}/(T_{\text{gas}}R_{\text{gas}}))^{0.5}$ component. From a thermodynamic perspective,

at $P_{\max}$ -idem, a decrease in ε reduced the pressure during the compression cycle phase and throughout the combustion cycle, while the $\lambda_{p\text{ gas}}/(\mu_{\text{gas}})^{0.5}$ ALFA formula structural component remained virtually unchanged. The identified relationship between the factors influencing thermal load and the control parameters forms the basis for optimizing the engine for LCF operation based on energy and environmental indicators, as well as for optimizing reliability factors.

To verify the obtained research results for the real combustion cycle of LCFs using a single-zone model, it is important to emphasize the positive correlation of these results with the idealized thermodynamic engine cycle, specifically the variational relationships between α_{gas} , T_{avg} , and the control parameters (ε , λ , P_k , and T_k) [43,57]. The physical justification of the revealed characteristic is determined by the heat release coefficient structure used in the studies, which includes the physical parameters of $\lambda_{p\text{ gas}}$ and μ_{gas} , as well as the thermodynamic parameters of α_{gas} and T_{avg} , of the working gas in the engine cylinder during heat exchange (see Equation (7)). The structure of Equation (7), with minor ICE research corrections, replicates the classically known analytical expressions for convective heat exchange processes involving the Reynolds (Re), Prandtl (Pr), and Peclet (Pe) numbers, where $\text{Re} = \text{Re}^n \text{Pe}^m$ [49,50,58]. The common trend observed in all LCF types studied was the change in the structural components of the α -formula $\lambda_{p\text{ gas}}/(\mu_{\text{gas}})^{0.5}$ and $(P_{\text{gas}}/(T_{\text{gas}}R_{\text{gas}}))^{0.5}$ when ε -idem: increasing λ led to a decrease in $\lambda_{p\text{ gas}}/(\mu_{\text{gas}})^{0.5}$ and an increase in the structural component of the thermodynamic indicators.

Since the variations in T_{avg} and α_{gas} exhibited qualitatively identical graphical trends across all evaluated LCF types (see Figure 4), a unified approach to reducing thermal load on components can be established. Based on the gradient principle, the most effective parameter optimization method for minimizing thermal stress involves the interdependent adjustment of λ and ε along the gradient direction (T_{LCF}) from the limiting ($\alpha_{\text{gas}} \times T_{\text{avg}}$) diesel reference line (see Figure 4d).

Figure 5 graphically represents the thermal load parameters of the engine after its conversion to LCF operation at $P_{\max} = \text{idem}$. For biodiesel and methanol, the optimal working cycle parameters in terms of P_{mi} and η_i were achieved without exceeding ($\alpha_{\text{gas}} \times T_{\text{avg}}$), maintaining the same λ and ε values as in diesel operation. Compared to diesel, the changes in P_{mi} were -1.4% and $+4.5\%$, while η_i remained unchanged for biodiesel and decreased by 1.2% for methanol. For LNG, the most efficient operating cycle was identified as LNG6 (S), which, according to [19], is the cycle recommended by the engine manufacturer. In the case of ammonia, the best results were obtained with the Ammonia10 cycle at $\varepsilon = 14.4$ and $\lambda = 2.2$, where P_{mi} decreased by 5.5% and η_i by 1.2% .

Figure 6 compares different m/φ_z values for ammonia, confirming that increasing combustion intensity (i.e., reducing combustion duration) not only enhances energy efficiency but also lowers the thermal load on engine components. As shown in the figure, a similar effect can be achieved by increasing the fuel injection pressure, which accelerates the combustion cycle [20,22,35,59,60]. In modern engine designs, rapid and intense fuel combustion is a standard requirement for efficient combustion cycle execution. This principle was notably applied in the development of MTU's fourth-generation diesel models [58,59]. Although preliminary and requiring further experimental validation, the findings suggest that this approach can further contribute to reducing the thermal load on engine components.

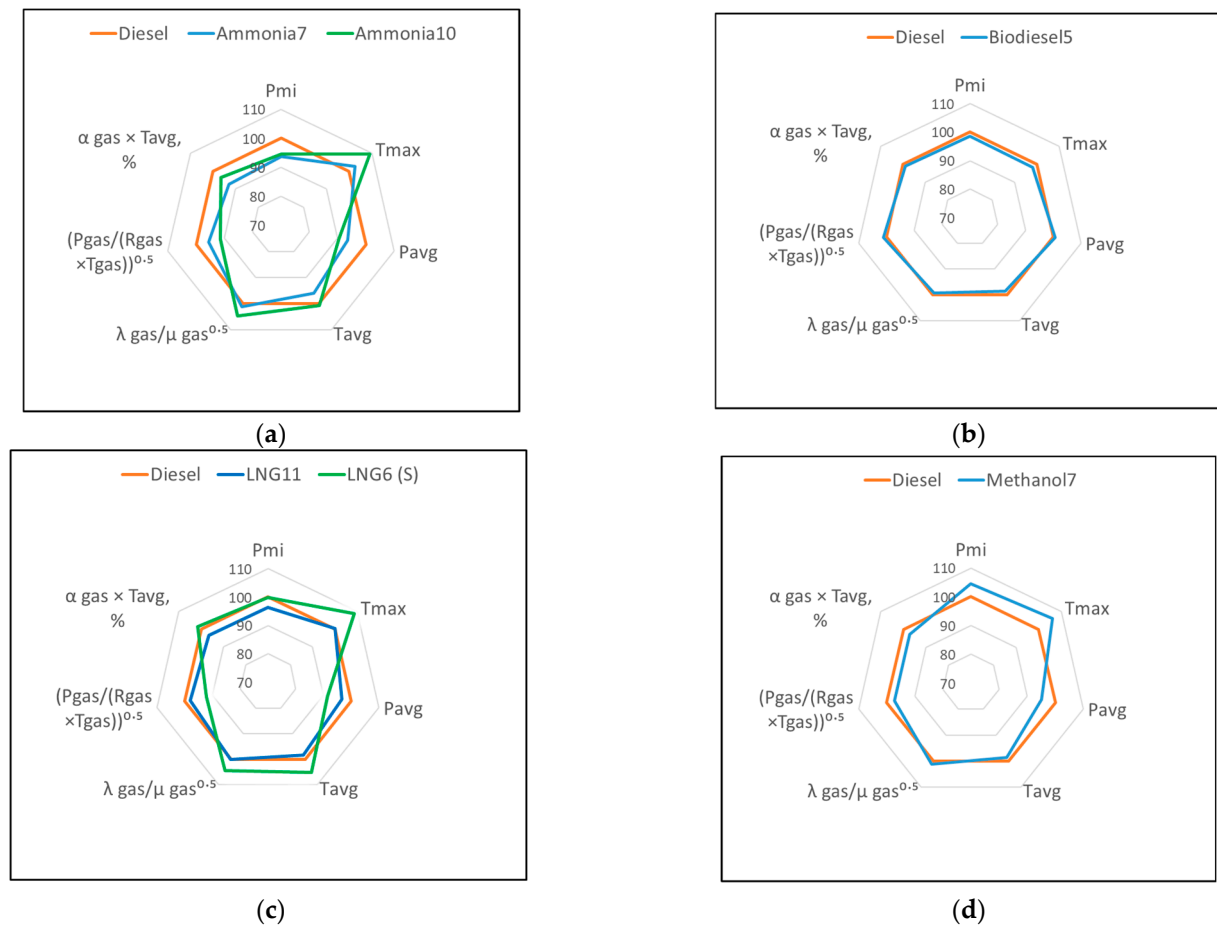


Figure 5. Comparison of thermal load parameter rates on the components of the Wärtsilä 9L20DF engine operating on LCFs ($P_{max} = idem$) versus diesel: (a) ammonia, (b) biodiesel, (c) LNG, and (d) methanol.

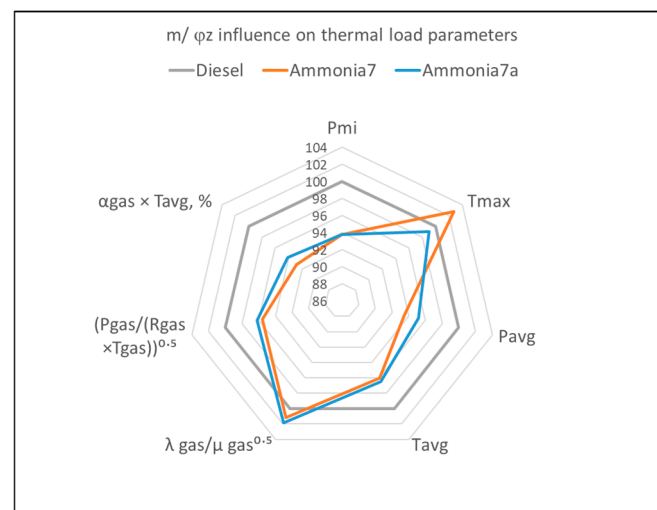


Figure 6. Thermal load dependence on the combustion cycle parameters (m and ϕ_z) for ammonia.

Among other operational measures to reduce thermal load without extensive engine modernization, increasing the cooling of the intake air temperature (T_k) in the intercooler is considered beneficial (typically implemented at 45–60 °C in practice) [53]. By evaluating the physical mechanism of T_k 's influence on the $\lambda_{p\ gas}/(\mu_{gas})^{0.5}$ and $(P_{gas}/(T_{gas}R_{gas}))^{0.5}$ parameters, it is predicted that the position of the $(\alpha_{gas} \times T_{avg}) = f(\lambda, \epsilon)$ combination mesh

in Figure 3 will shift below the $(\alpha_{\text{gas}} \times T_{\text{avg}})$ diesel limit line, thereby increasing the thermal load margin for engine components.

4. Discussion of Research Results

The proposed methodology and the results obtained from its application can be used as a tool for evaluating the simultaneous optimization of the combustion cycle based on energy and environmental indicators, as well as changes in thermal and mechanical loads on components during the retrofitting of operating ship fleets for LCF use. This ensures the reliable operation of engines for LCF use. For the practical application of the retrofit, characteristic features for optimizing the combustion cycle for LCF use were revealed:

- By converting the engine to operate on all of the LCFs evaluated, after ensuring $P_{\text{max}} = \text{idem}$ with diesel, the thermal stress indicator $(\alpha_{\text{gas}} \times T_{\text{avg}})$ also approached the permissible limits, similar to when the engine operates on diesel. The temperature factors were approximately 3% lower for methanol, ~7% for ammonia, ~1% for biodiesel, and ~3.5% for LNG (see Table A1 and Figure 3). Increasing the excess air coefficient λ beyond 2.5 units to reduce thermal stresses on components is not rational, as the positive effect of λ on thermal stress diminishes.
- The influence of the combustion cycle duration on $(\alpha_{\text{gas}} \times T_{\text{avg}})$ was revealed: for fuels with relatively short combustion durations, such as ammonia and methanol, the effect on $(\alpha_{\text{gas}} \times T_{\text{avg}})$ diminished in the $\lambda \geq 2.5$ range, while for biodiesel and LNG, only a decreasing trend was observed.

To ensure the practical application of the results, it is rational to continue research in two main directions:

- Alongside the α_{gas} -based structure (widely validated across different diesel engine types), it is important to adapt the α_{gas} formulas from other developers, primarily G. Woschni's analytically derived expressions, which are approved for various engine fuel types [49,50]. Methodological aspects should be expanded by determining the ALFA formulas for the combustion cycle's current parameters, T_{gas} and P_{gas} .
- Additionally, it is important to consider that operational indicators are shaped when the engine operates in characteristic cycle modes (e.g., the E3 or E2 test cycle, according to ISO 8178 for ship engines). In this context, it would be beneficial to expand the research to include partial load engine modes, where engines typically operate for extended periods. Thermal and mechanical loads on components should be assessed based on the average values of the operational cycle. When modeling the engine's combustion cycle characteristics under partial load conditions, it will be necessary to reassess the applicability of G. Woschni's m and φ_z recalibration model, taking into account the author's observations on the inverse relationship between these parameters and λ [61], especially when compared to diesel operation.

5. Conclusions

Assessing the impacts of thermal and mechanical loads when converting existing marine diesel engines to operate with LCFs—particularly those on the cylinder–piston group components—is considered a rational part of the decarbonization challenge in the maritime sector. This approach, alongside energy and environmental optimization, allows for evaluation of the optimal scope and rational directions for necessary retrofitting.

Based on the study of indicators determined for the medium-speed marine engine Wärtsilä 9L20DF and the application of the developed original methodology, the impact of the thermal load on the engine's cylinder–piston group components was identified when switching from diesel fuel to prospective LCF types: LNG, biodiesel, methanol, and ammonia. Without altering the main engine control parameters (φ_{comb} , P_k , λ , and

ϵ) and operating at rated power, the thermal load on components increased by 2–11%, with the highest increase observed for methanol. Additionally, the peak combustion cycle pressure, which determines the maximum mechanical load level necessary for reliable engine operation, rose significantly: by 23% for LNG, 45% for ammonia (with direct injection), and 72% for methanol (with direct injection). Therefore, the retrofit strategy should initially focus on reducing mechanical loads (based on the $P_{\max}$ factor), followed by optimizing the factors that influence thermal loads, all while maintaining constant $P_{\max}$ conditions.

The indicator of the energy efficiency of the combustion cycle—the useful work coefficient (η_i)—improved by 3–5% in correspondence with the increase in the cycle dynamics (P_{mi}), thus creating a reserve for further optimization of the control parameters while maintaining the specific fuel consumption.

The structural analysis of the thermal load factors showed that, except for biodiesel, the thermal load on the components increased as both the characteristic temperature T_{avg} and the heat transfer coefficient α_{gas} increased. In addition, the influence of the two structural components of α_{gas} increased: the components characterized the thermodynamic parameters of the working body $(P_{gas}/(T_{gas}R_{gas}))^{0.5}$ and the components of the physical properties $\lambda_{p\ gas}/(\mu_{gas})^{0.5}$ in a manner dependent on the thermodynamic parameters (T_{gas} , P_{gas}). Based on this, a hypothesis is proposed regarding the priority of reducing thermal loads on components while ensuring the set $P_{\max}$ value, which gains additional confirmation.

The proposed strategy suggests that, in order to maintain component loads similar to those when the engine operates on diesel and reduce $P_{\max}$ to the specified values, it is rational to keep the values of λ and ϵ constant (while optimizing P_k and ϕ_{comb}). This approach ensures that the thermal load on the components remains close to that during diesel operation ($\alpha_{gas} \times T_{avg}$).

In the proposed retrofitting stage, without deep structural modernization, the greatest regulatory effect on component thermal loads (while ensuring $P_{\max} = idem$) was achieved by improving the air supply (λ) up to a set rational limit of 2.5 units and reducing the compression ratio, taking into account fuel consumption and ensuring a “cold” state engine start.

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Abbreviations

The following abbreviations are used in this manuscript:

LCF	low carbon fuels;
GHG	greenhouse gas;
IMO	International Maritime Organization;
LNG	liquified natural gas;
MM	mathematical model;
ICE	internal combustion engine;
TDC	top dead center;
PFI	port fuel injection;
LHV	lower heating value;
CAD	crank angle degrees;
CPG	cylinder–piston group;
EGR	exhaust gas recirculation;
TE	thermal efficiency;
HDDF	high-density dual fuel;
SOPI	start of pre-injection;
ROPI	ratio of pre-injection;
BTDC	before top dead center;
CA50	crank angle at which 50% of the heat from combustion is released;
CA90	crank angle at which 90% of the heat from combustion is released;
CO ₂	carbon dioxide;
CH ₄	methane;
N ₂ O	dinitrogen oxide;
NO _x	nitrous oxides;
T _{max}	maximum combustion cycle temperature (K);
T _k	manifold air temperature (K);
T _{avg}	combustion average temperature (K);
P _{max}	maximum combustion cycle pressure (bar);
P _{avg}	combustion average pressure (bar);
P _{mi}	mean indicated pressure (bar);
P _{me}	break mean effective pressure (bar);
P _k	manifold air pressure (bar);
G _i	fuel injection portion;
η_i	useful work coefficient;
φ_z	combustion duration (CA°);
φ_{inj}	fuel injection crank angle
φ_{comb}	start of combustion (CA°);
m	heat release form factor;
λ	air fuel ratio;
$\alpha_{gas\ av}$	heat load coefficient during the cycle;
ε	engine compression ratio;

Appendix A

Table A1. Experiment results.

Test	Adjustable Parameters					Mathematical Modeling Results						Calculation		
	ε	φ_{comb}	λ	m/ φ_z	P _k	P _{mi}	P _{max}	T _{max}	P _{avg}	T _{avg}	η_i	$\lambda_{p\ gas}/(\mu_{gas})^{0.5}$	α_{gas}	$\frac{\alpha_{gas}}{T_{avg}} \times 100$
Diesel	13.4	−6	2.5	1.1/65	4.4	23.8	177	1586	65.7	1087	0.48	1.11	457	100
LNG 1	12.4	−8	1.7	0.5/60	3.0	23.5	177	2013	54.0	1281	0.47	1.25	425	109.7
LNG 2	12.4	−6	1.9	0.5/60	3.4	23.7	178	1723	56.2	1228	0.47	1.22	431	106.6

Table A1. Cont.

Test	Adjustable Parameters					Mathematical Modeling Results						Calculation		
	ϵ	φ_{comb}	λ	m/φ_z	P_k	P_{mi}	P_{max}	T_{max}	P_{avg}	T_{avg}	η_i	$\lambda_{p\text{ gas}}/(\mu_{\text{gas}})^{0.5}$	α_{gas}	$\frac{\alpha_{\text{gas}}}{T_{avg}} \times \%$
LNG 3	12.4	−3	2.2	0.5/60	3.9	23.9	179	1726	59.8	1133	0.47	1.15	439	100.3
LNG 4	13.4	−5	1.7	0.5/60	3.0	23.7	177	2006	54.6	1284	0.47	1.26	427	110.6
LNG 5	13.4	−3	1.9	0.5/60	3.4	23.8	178	1876	57.3	1218	0.47	1.21	435	106.8
LNG 6 (s)	13.4	0	2.2	0.5/60	3.9	23.8	176	1723	60.1	1143	0.47	1.16	441	101.5
LNG 7	14.4	−2	1.7	0.5/60	3.0	23.7	174	1974	55.3	1278	0.47	1.25	430	110.8
LNG 8	14.4	−1	1.9	0.5/60	3.4	23.9	179	1859	58.8	1214	0.47	1.21	441	107.9
LNG 9	14.4	2	2.2	0.5/60	3.9	23.8	176	1708	62.8	1141	0.47	1.16	451	103.6
LNG10	12.4	−2	2.5	0.5/60	4.2	23.0	180	1612	61.7	1074	0.48	1.11	442	95.7
LNG11	13.4	1	2.5	0.5/60	4.2	23.0	177	1590	63.4	1069	0.48	1.11	448	96.6
LNG12	14.4	3	2.5	0.5/60	4.2	22.9	177	1576	65.5	1068	0.47	1.11	456	98.1
RME1	12.4	−11	2.2	1.1/65	3.9	23.5	175	1701	60.3	1136	0.48	1.14	441	100.9
RME2	12.4	−8	2.5	1.1/65	4.4	23.4	174	1572	63.9	1073	0.47	1.10	450	97.3
RME3	12.4	−4	3.0	1.1/65	5.3	23.1	177	1411	70.6	996	0.47	1.06	471	94.6
RME4	13.4	−8	2.2	1.1/65	3.9	23.5	174	1673	61.8	1130	0.48	1.14	446	101.6
RME5	13.4	−6	2.5	1.1/65	4.4	23.5	178	1558	66.2	1072	0.48	1.10	458	99.0
RME6	13.4	−1	3.0	1.1/65	5.3	22.8	178	1392	73.0	995	0.46	1.06	480	96.1
RME7	14.4	−6	2.2	1.1/65	3.9	23.5	176	1657	63.7	1127	0.48	1.14	453	102.8
RME8	14.4	−3	2.5	1.1/65	4.4	23.2	175	1535	67.9	1069	0.47	1.10	464	100.0
RME9	14.4	0	3.0	1.1/60	5.3	23.2	194	1399	76.6	1004	0.46	1.06	491	99.3
Ammonia1	12.4	0	2.0	0.5/33	3.3	22.2	179	1847	53.6	1144	0.46	1.20	418	96.3
Ammonia2	12.4	2	2.2	0.5/33	3.7	22.3	179	1753	56.0	1098	0.47	1.16	424	93.7
Ammonia3	12.4	5	2.5	0.5/33	4.1	22.3	175	1638	59.3	1045	0.46	1.12	433	91.3
Ammonia4	12.4	8	3.0	0.5/33	4.9	22.1	175	1501	65.5	982	0.46	1.08	454	89.7
Ammonia5	13.4	3	2.0	0.5/33	3.3	22.3	177	1827	54.6	1137	0.47	1.20	422	96.6
Ammonia6	13.4	5	2.2	0.5/33	3.7	22.3	175	1734	57.3	1093	0.47	1.16	428	94.4
Ammonia7	13.4	7	2.5	0.5/33	4.1	22.3	175	1629	61.4	1044	0.47	1.12	441	92.8
Ammonia7a	13.4	−1	2.5	0.8/50	4.1	22.3	177	1571	62.5	1049	0.47	1.13	445	94.1
Ammonia8	13.4	9	3.0	0.5/33	4.9	22.3	181	1501	68.7	984	0.47	1.08	465	92.1
Ammonia9	14.4	5	2.0	0.5/33	3.3	22.4	176	1816	56.1	1135	0.47	1.19	428	97.8
Ammonia10	14.4	6	2.2	0.5/33	3.7	22.5	180	1733	59.5	1094	0.47	1.16	437	96.4
Ammonia11	14.4	8	2.5	0.5/33	4.1	22.4	179	1628	64.0	1045	0.47	1.12	451	94.9
Ammonia12	14.4	10	3.0	0.5/33	4.9	22.4	184	1500	71.7	987	0.47	1.08	475	94.5
Methanol1	12.4	−1	2.0	0.5/40	3.3	24.7	177	1911	54.9	1180	0.47	1.20	428	101.8
Methanol2	12.4	1	2.2	0.5/40	3.7	24.8	175	1811	57.2	1132	0.47	1.17	433	98.9
Methanol3	12.4	4	2.5	0.5/40	4.1	24.8	177	1694	60.4	1074	0.47	1.12	441	95.5
Methanol4	12.4	7	3.0	0.5/40	5.0	24.7	178	1537	67.2	1002	0.47	1.08	463	93.4
Methanol5	13.4	2	2.0	0.5/40	3.3	24.9	179	1898	56.0	1172	0.47	1.20	432	102.1
Methanol6	13.4	4	2.2	0.5/40	3.7	24.9	177	1798	58.4	1127	0.47	1.16	438	99.4
Methanol7	13.4	6	2.5	0.5/40	4.1	24.9	177	1683	62.5	1073	0.47	1.12	449	97.1
Methanol8	13.4	9	3.0	0.5/40	5.0	24.7	178	1528	69.9	1003	0.47	1.08	472	95.4
Methanol9	14.4	4	2.0	0.5/40	3.3	25.0	179	1883	57.4	1169	0.47	1.19	438	103.1
Methanol10	14.4	5	2.2	0.5/40	3.7	24.9	175	1785	60.7	1127	0.47	1.16	447	101.4
Methanol11	14.4	7	2.5	0.5/40	4.1	24.8	175	1671	65.1	1074	0.47	1.12	459	99.3
Methanol12	14.4	10	3.0	0.5/40	5.0	24.8	182	1525	72.9	1006	0.47	1.08	483	97.8
Ammonia Gi = QD	13.4	−6	2.5	0.8/50	4.1	22.9	206	1622	65.5	1060	0.48	1.13	456	97.5
Ammonia Gi = QD	13.4	−6	2.5	0.5/33	4.1	23.5	253	1759	69.7	1077	0.49	1.14	472	102.3
Methanol(m) Gi = QD	13.4	−6	2.5	0.4/30	4.1	25.0	273	1860	72.7	1114	0.50	1.16	497	111.6
Methanol Gi = QD	13.4	−6	2.5	0.2/17	4.1	24.9	301	1940	75.5	1128	0.50	1.15	487	110.6
LNG Gi = QD	13.4	−6	2.5	0.5/60	4.4	24.9	227	1666	70.9	1087	0.49	1.13	477	104.5

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