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Optimisation of the combustion process in a low viscosity fuel engine

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ABSTRACT:

The scope of the thesis was to evaluate new engine technologies to enable ultimate fuel flexibility. It was written as part of a Wärtsilä R&D project, where a system that enables the use of Liquid Gases (LGs) as fuel in the diesel principle engine was developed. These fuels are available as side- or waste products of other processes, for example as natural gases condensates. The focus of this work was to maximise engine power output and overall performance, when using ultra-low viscosity fuels which have poor quality characteristics as fuel for engines that are currently available on the market.

To reach these objectives, three experimental areas were carried out:

1. Testing different fuels in the LG range in a Combustion Research Unit (CRU) to evaluate their ignitability and combustion response. Results provide a wide overview about ignitability of different fuels in the low viscosity range and the required amount of pilot fuel to enable the combustion.

2. Fuel injection rig testing to identify the material and geometry validation of fuel injection components for the LG engine. Based on a 500-hours endurance test, one of the three tested materials was selected as candidate material for injector nozzles.

3. Engine testing, which was the major part of the project. This stage validated the previous test stages, the simulations done for the injection system and for engine performance. The outcome provided one single set of parameters (hardware and software) for operation with all LG fuels, based on testing with LPG, LFO and liquid volatile organic compound (LVOC) fuels and two different injector nozzle setups.

The outcome of the tests was an engine able to meet the initial project targets, which consisted of defining a concept that can run freely with all LG fuels and the defined power output of the engine, without any changes in hardware (injector nozzle) or software settings (main fuel pressure and pilot settings), despite a varying chemical composition of the fuel.

KEYWORDS: LG, liquid gas, LPG, liquefied petroleum gas, low viscosity fuels, diesel engine, CI engine

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TIIVISTELMÄ:

Opinnäytetyön tarkoituksena oli arvioida uusia moottoritekniikoita, jotka mahdollistavat laajan polttoainejoustavuuden. Työ kirjoitettiin osana Wärtsilän tutkimus- ja kehityshanketta. Hankkeessa kehitettiin järjestelmä, joka mahdollistaa nestemäisten kaasujen (LG) käytön polttoaineena dieselmoottorissa. Nämä polttoaineet ovat muiden prosessien sivu- tai jätetuotteita, ja niitä syntyy esimerkiksi maakaasujen käsittelyyn liittyvissä proseissa, kuten uuttamisessa. Työn painopisteenä oli selvittää moottorin teho ja -suorituskyky, kun käytetään kaupallisesti saatavilla olevaa erittäin pienen viskositeetin polttoainetta, jonka ominaisuudet eivät ole ideaalisia nykymoottoreihin.

Näiden tavoitteiden saavuttamiseksi toteutettiin kolme kokeellista tehtävää:

1. Tutkittiin erilaisten nestemäisten kaasujen syttymistä ja palamista tähän tarkoitukseen kehitetyssä analysaattorissa (combustion research unit, CRU). Tulokset antoivat yleiskuvan erilaisten pienviskoositeettipolttoaineiden syttyvyydestä ja tarvittavasta sytytyspolttoaineen määrästä palamisen mahdollistamiseksi.

2. Polttoaineen ruiskutuskomponenttien materiaaleja ja geometriaa tutkittiin erilisessä LG-moottorin ruiskutuslaitteiston testipenkissä. Viidensadan tunnin kestävyyskokeen perusteella yksi kolmesta tutkitusta materiaalista valittiin ehdokasmateriaaliksi ruiskutussuuttimiin.

3. Projektin pääosassa olivat moottorimittaukset, jotka validoivat edelliset vaiheet sekä ruiskutusjärjestelmää ja moottorin suorituskykyä varten tehdyt simulaatiot. Tuloksena saatiin parametrijoukko, jota voidaan käyttää kaikkien LG-polttoaineiden kanssa. Laboratoriomoottorilla ajetuissa mittauksissa käytettiin nestekaasua sekä LFO- että LVOC (liquid volatile organic compound) -polttoaineita. Lisäksi mittaukset tehtiin käyttäen kahta eri ruiskutussuutinta.

Tutkimuksen tuloksena moottori kykeni saavuttamaan alkuperäiset tavoitteet. Tässä työssä mainittujen mittauksien avulla määriteltiin uusi moottoriratkaisu, joka voi toimia vapaasti kaikilla LG-polttoaineilla. Moottorilla pystytään saavuttamaan tavoiteltu teho ilman muutoksia moottoriparametreissä ja riippumatta polttoaineen kemiallisesta koostumuksesta.

AVAINSANAT: nestemäiset kaasut, nestekaasu, LG, pienen viskoositeetin polttoaine, dieselmoottori

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Evgenia Okhotnikova

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Abbreviations

BDC	Bottom Dead Centre
BMEP	Break Mean Effective Pressure
BTDC	Before Top Dead Centre
BSFC	Brake-Specific Fuel Consumption
CA°	Crank Angle Degree
CAPEX	Capital Expenditure
CVCC	Constant Volume Combustion Chamber
CHP	Combined Heat and Power
CI	Compression-Ignited (Engine)
CN	Cetane Number
CNG	Compressed Natural Gas
CO	Carbon Monoxide
CO_2	Carbon Dioxide
CRU	Combustion Research Unit
DF	Dual-Fuel
DI	Direct Injection
EGR	Exhaust Gas Recirculation
FEM	Finite Element Method
FSN	Filter Smoke Number
GD	Gas-Diesel (Engine)
GHG	Greenhouse Gases
HC	Hydrocarbon
HFO	Heavy Fuel Oil
HGL	Hydrocarbon Gas Liquids
ICE	Internal Combustion Engine
IMO	International Maritime Organization
LFO	Light Fuel Oil

LG	Liquid Gas (Engine)
LGs	Liquid Gases (Fuels)
LHV	Lower Heating Value
LNG	Liquefied Natural Gas
LPG	Liquefied Petroleum Gas
LVOC	Liquefied Volatile Organic Compound
MEP	Mean Effective Pressure
MFI	Main Fuel Injection
MGV	Main Gas Valve
MN	Methane Number
NO_x	Nitrogen Oxides
OPEX	Operating Expenditure
PDSV	Pressure Drop and Safety Valve
PFI	Pilot Fuel Injection
PID	Proportional-Integral-Derivative (Controller)
PM	Particulate Matter
PV	Pressure-Volume (Diagram)
ROHR	Rate of Heat Release
SCR	Selective Catalytic Reduction
SFC	Specific Fuel Consumption
SG	Spark-Ignited Gas (Engine)
SI	Spark-Ignited (Engine)
SO_x	Sulphur Oxides
TC	Turbocharger
TDC	Top Dead Centre
UNIC	Unified Control and Monitoring System (Wärtsilä Engines)
US EPA	United States Environmental Protection Agency
VIC	Variable Inlet Valve Closing

1 Introduction

Large internal combustion engines (ICE) are used in various power generation applications. Examples include electricity generation, combined heat and power (CHP) plants used to meet the demands of industries, cities and remote areas; and production of mechanical power to propel marine vessels. Engines can be classified according to their working cycle, technical design, speed, power output, application, valve design and location, fuel type and ignition method. ICEs are an attractive choice for power generation where robust operation is needed, as they are capable of fast start-up and loading times. In the tendencies towards renewable energy, engines can play a role as fast backup technology to support power generation that depends on external factors, such as solar and wind power. Currently relevant developments in the field of engine technologies include: increasing the performance and reliability of existing technology; enabling new types of applications or ways of operation; enabling previously unexplored fuels; and decreasing the environmental impact both through the engine operation and life-cycle assessments of whole processes and chains. [1]

The scope of the thesis was to evaluate new engine technologies to enable ultimate fuel flexibility. The focus of this work was to maximise engine power output and overall performance, when using ultra-low viscosity fuels that are available on the market at low price or as process waste. Ultra-low viscosity fuels (also referred to as liquid gases or LGs in this work) are considered to have any composition of mixed hydrocarbons from C3 to C20, which corresponds to the fuel range from liquified petroleum gas (LPG) to kerosene and light fuel oil (LFO). Hydrocarbons in this range are obtained as by-products of natural gas and oil extraction processes. Due to their chemical composition, such fuels are characterised by low methane number (MN) and consequently have low knock resistance in Otto-process engines. This means that engine power output must be drastically reduced to guarantee safe and reliable operation. Alternatively, new technology needs to be developed to supplement classic Otto and diesel. The main focus in this development was LPG fuel, driven by market requirements to have an optimised engine towards power output and consequentially reduce the carbon footprint of the power plant by increasing the power density. These actions will improve the capital expenditure (CAPEX) and operating expenditure (OPEX) figures to the customer and make the business case stronger. In 2015, Wärtsilä pioneered in delivering the first power plant fuelled by LPG, based on Wärtsilä 34 Spark-Ignited Gas (W34SG) technology. This result was achieved by optimising compression ratio, pre-chamber, valves' timing, turbocharger specification and engine control system. Despite the optimisation, due to the poor fuel quality in terms of methane number (around MN 34), engine power output was reduced by 25% of its nominal output.

In order to improve product competitiveness, the project described in this thesis focused on increasing the power output of the developed engine to nominal value. This means that an alternative technology to SG (Otto-process) needed to be developed. Liquid gas (LG) technology was identified as a suitable solution to meet the project target. This technology is based on a high-pressure injection system, which enables using the fuel in liquid mode. Subsequently, this allows to use the complete range of natural gas condensates as LG fuel (and not only LPG), while maintaining emissions at a sufficiently low level and not decreasing the engine's nominal power output, which has not been previously done. To realise this idea into practice, the LG development project was initiated in 2018. This thesis – a part of the project scope - had the following objectives and experimental methods used to achieve them:

1. Evaluate the ignitability and combustion response of different fuels in the LG range. Methods used: fuel testing in a Combustion Research Unit (CRU) (Chapter 4.1)
2. Identify the material and geometry validation of fuel injection components for the LG engine. Methods used: Fuel injection rig testing (Chapter 4.2)
3. Develop the LG concept, based on engine testing. This was the major part of the project. Methods used (Chapter 4.3):
    a. Laboratory engine testing with 6-cylinder in-line configuration (W6L32LG).
    b. Testing the defined concept on 20-cylinder V-form engine (W20V32LG), which is the target for the LG product.

2 Theoretical background

2.1 Basics of engines

Internal combustion engines (ICEs) are used to convert the chemical energy contained in the fuel into mechanical power through combustion (oxidisation), which occurs inside the engine cylinders. The working fluids in this process are air and fuel (before the combustion) and exhaust gases (after the combustion). The percentage of mechanical power which is then converted into electricity depends on the efficiency of the process. In the 4-stroke engine, one complete thermodynamic cycle occurs over two revolutions of the crankshaft. Figure 1 illustrates the different engine strokes.

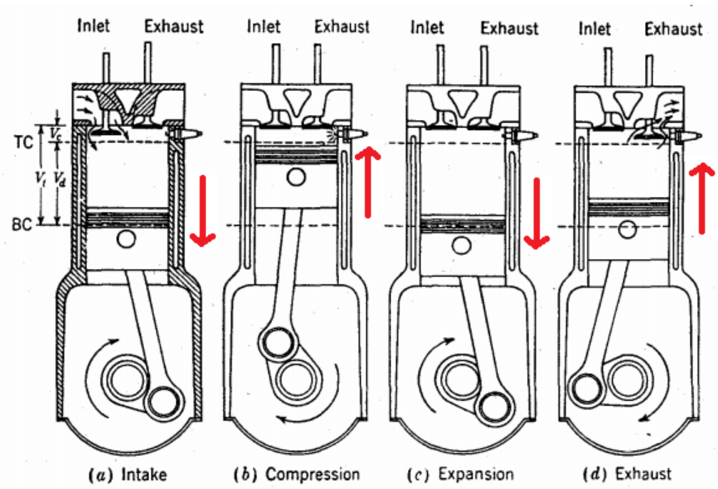


Figure 1. Four strokes of the engine with clockwise rotation. [1]

Theoretical combustion processes that describe engine operation – the Otto and diesel combustion cycles – differ in their combustion mixture formation, ignition method, compression ratio and combustion behaviour. For these reasons and the related chemical properties of the fuel, they operate optimally within different fuel ranges. Theoretical cycles are an idealisation of the real process and do not reflect all the characteristics of an actual combustion, such as thermal and friction losses and real gas properties. [1] [2]

2.1.1 Characteristics of the diesel process

Operation of the LG engine, which is the subject of development in this thesis, is based on the diesel process. Figures 2 – 3 illustrate the ideal and actual pressure-volume (PV) diagrams of this process, which represent the work done. These details demonstrate that the actual diesel process – also commonly referred to as modified diesel process – is a combination of features found in theoretical diesel and Otto cycles.

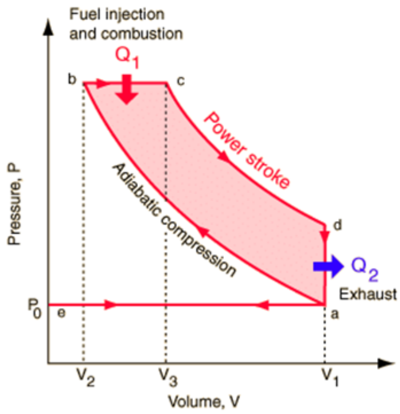


Figure 2. PV- diagram of the ideal diesel process. [3]

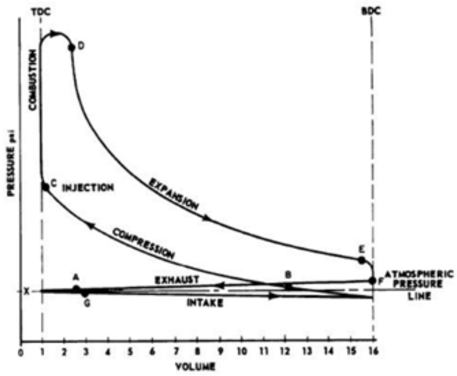


Figure 3. PV- diagram of an actual diesel process. [1]

The differences between the theoretical and actual strokes are summarised in Table 1.

Table 1. Four strokes in theoretical and actual diesel cycles. [1] [4]

Description of the stroke	Theoretical diesel cycle	Actual diesel cycle / Modified diesel process
1) Intake: Fresh air is introduced into the cylinders.	Doesn’t account for losses.	Losses from inertia and friction. Corresponding area is visible in the PV-diagram of the actual process.
2) Compression: Air is compressed into a small fraction of the initial cylinder volume.	Adiabatic compression.	According to a predefined compression ratio of the engine, which is commonly between 15:1 and 20:1. Thermal losses occur and engine cooling is used.
3) Expansion / Power: Liquid fuel is directly injected into the cylinders. Combustion follows, causing a rapid increase in pressure that pushes the piston downwards.	Constant pressure combustion occurs gradually as fuel is injected. This maintains a constant pressure level. The expansion is adiabatic.	Fuel is injected slightly before top dead centre (TDC) via small nozzle holes. Ignition happens after an ignition delay period. Combustion is not even throughout the cylinder. Friction and thermal losses occur.
4) Exhaust: Exhaust gases are released from the cylinders.	The exhaust stroke is isovolumetric.	The gases are rapidly released from cylinders due to large pressure difference before and after the exhaust valves. However, at the same time, the piston is moving. Thus, the volume changes.

2.1.2 Basic engine equations

The basic equations presented in this chapter are used to evaluate the quality of the work done by the engine, as given in the J. B. Heywood book "Internal Combustion Engine Fundamentals". [1] The indicated work per engine cycle is obtained by integrating the area enclosed in the corresponding PV diagram, according to:

$$W_{ci} = \oint p \, dV \qquad (1)$$

This equation relates to the indicated power per each engine cylinder in the following way:

$$P_i = \frac{W_{ci} \, N}{n_R} \qquad (2)$$

where N is the engine speed and n_R is the number of crank revolutions per power stroke per cylinder, which in the case of four-stroke engines is 2. This equation indicates the rate of work transfer from the gases inside the cylinder to the piston and is used to understand the impact of compression, combustion and expansion on the performance of the engine. This value differs from the gross power, which reflects the sum of useful work at the shaft and the work needed to overcome losses. An alternative way to obtain the indicated power is to sum brake power and friction power. [1]

$$P_{ig} = P_b + P_f \qquad (3)$$

The mechanical efficiency of the engine can be defined as the ratio of P_b, which is the useful or brake power delivered by the engine, to P_{ig}, the engine's indicated power.

$$\eta_m = \frac{P_b}{P_{ig}} = 1 - \frac{P_f}{P_{ig}} \qquad (4)$$

The ability of the engine to do work is described by its mean effective pressure (MEP), which is a relative performance measure that does not depend on the size of the engine. It is calculated by dividing the work done per engine cycle by the volume displaced during the cycle.

$$Work\ per\ cycle = \frac{P\ n_R}{N} \tag{5}$$

$$mep = \frac{P\ n_R}{V_d\ N} \tag{6}$$

The specific fuel consumption of the engine is measured as the fuel flow rate per unit power output of the engine.

$$sfc = \frac{\dot{m}_f}{P} \tag{7}$$

The efficiency of the engine, also referred to as fuel conversion efficiency, is measured in the following way:

$$\eta_f = \frac{P}{m_f\ Q_{HV}} = \frac{1}{sfc\ Q_{HV}} \tag{8}$$

where Q_{HV} is the heating value or energy content of a fuel.

2.2 Diesel combustion theory

In the diesel engine, fuel is injected into the cylinders towards the end of the compression stroke, when the temperature and pressure inside of the cylinders are sufficiently high to initiate combustion, which means these conditions are above the fuel's ignition point. The different stages of combustion are illustrated on the typical rate of heat release (ROHR) diagram in Figure 4. These stages are:

Ignition delay

Ignition delay (segment a – b in Figure 4) is the time between the start of fuel injection and the start of combustion. During this period, fuel breaks up into small droplets, vaporises and mixes with the air inside the cylinder until a combustible air-fuel mixture is formed. Turbulence caused by high fuel injection pressures helps to speed up this process. The length of the ignition delay is influenced by the fuel's combustion properties (high cetane number CN reduces the ignition delay) and the fuel injection settings and conditions inside the cylinder when the fuel is injected (temperature and pressure). [1]

Premixed combustion

Premixed combustion (segment b – c in Figure 4) is characterised by a high ROHR and high pressure peak at the start of the combustion of the combustible air-fuel mixture. It lasts a few crank angle degrees (CA°), until all the premixed air and fuel have been burnt. A longer ignition delay increases this pressure peak and ROHR during premixed combustion, which is undesirable as it causes high cylinder temperatures and increased amount of harmful emissions. Once again, this highlights the benefit of using fuels with higher CN in the diesel process. [1]

Mixing-controlled combustion

After the premixed air-fuel has been consumed, the turbulent air inside the cylinder mixes with the fuel spray that is still being injected in the mixing-controlled combustion phase (segment c – d). During this predominant combustion stage in the diesel process, various processes occur (liquid fuel atomisation, vaporisation, mixing of air and fuel, pre-flame chemical reactions), but its rate is mainly controlled by the mixing phase. The combustion rate is high, as more fuel enters the cylinder. Typically, a pressure peak occurs at the end of injection, after which the ROHR slows down. [1]

Late combustion

Late combustion (segment d – e) starts after the end of fuel injection and continues at a lower rate than previously as the piston moves downward during the expansion stroke.

A small amount of fuel might still be present and continue to burn. However, the air and fuel content inside the cylinder steadily decrease. This, combined with the decrease in temperature and pressure due to expansion, slow down and consequently stop the combustion reaction. [1]

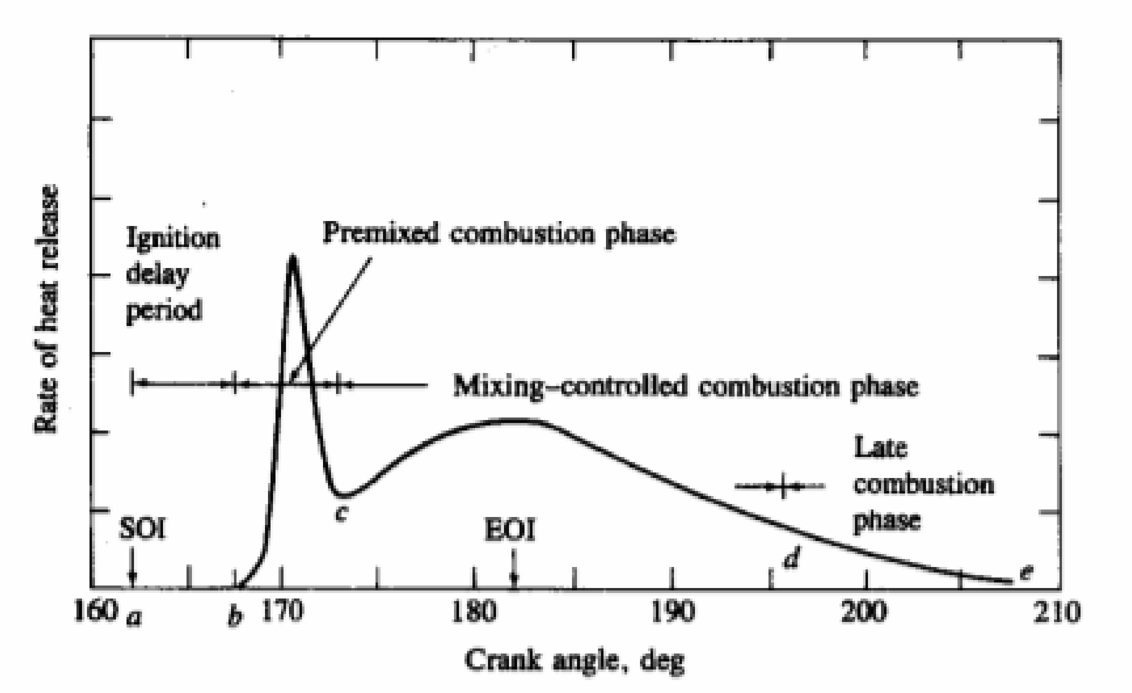


Figure 4. Typical direct-injection engine ROHR diagram identifying different diesel combustion phases. [1]

2.3 Auto-ignition and engine knock

Since the combustion in diesel and Otto processes occur differently, the desirable fuel characteristics are also different. The variation of heavy hydrocarbon content of the fuel is critical in Otto engines, as it has a lower MN, which leads to auto-ignition phenomena. Auto-ignition happens when the fuel-air mixture reaches a temperature over the limit during the compression stroke. This causes early ignition, fast and uncontrolled combustion cycles and leads to engine knock. This phenomenon consists of spontaneous and fast combustion with pressure waves, which can damage engine components such as flame plate, spark plug, valves, pistons and cylinder liners. In addition to using a high MN fuel, knock can be controlled by reducing compression ratio, cylinder cooling and using a leaner air-fuel mixture. Figure 5 illustrates the operating window constraints, based on knocking and misfire. [1]

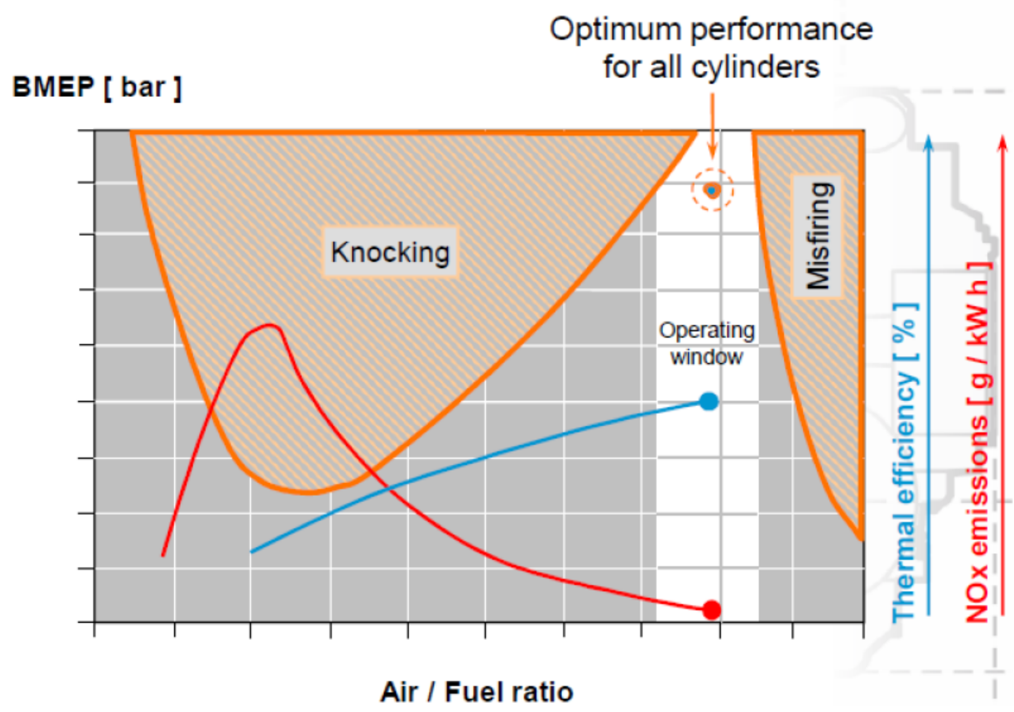


Figure 5. Operating window for Wärtsilä gas engines (knocking and misfiring). [5]

While, due to the use of diesel process in LG technology, these phenomena are not expected to be prominent, they may occur as the characteristics of some LG fuels correspond to low MN fuels used in Otto engines.

3 Liquid Gas (LG) technology

3.1 Reasons for LG development

This chapter provides the motivation for LG technology selection and the benefit of using the diesel cycle process with LG fuels.

Properties of LG fuels

LG fuels consist of a cocktail of aromatics, olefins, naphthenes, paraffins and oxygenates and their composition can vary based on the source and during the ageing of the source. They are obtained as side-stream or waste products of extraction processes, for example as gas condensates. This mix can contain any hydrocarbons that have carbon number from C3 to C20 (from LPG to kerosene and LFO). Appendix 1 demonstrates examples of possible LG fuel composition, where a large variety of different hydrocarbons can be present. This wide fuel range means that properties of the fuel used can change significantly. Therefore, LG technology is required to have a robustness and diminished sensitivity to these variations.

Gas fuels are characterised by a methane number value, based on their composition. This number describes the knock resistance of the fuel and sets the constrain to air-fuel ratio, boost pressure and ultimately limits the engine power output. For example, W34SG nominal power output is 75% when using LPG fuel (MN 34), instead of methane (~MN 100). LPG grade on the market is usually around 96% propane content and the remaining part is different heavier hydrocarbons. Hydrocarbons that are heavier than propane reduce the MN drastically and further reduce the engine power output. Appendix 2 provides an example of composition analysis of LPG available on the market.

Using LG fuels is also linked to the environmental benefit of using waste fuels, which would otherwise be flared. The side-stream of oil and gas production is made up by gas condensates, that as today are flared, recompressed to the ground or utilised in the chemical industry. These processes are illustrated below in Figure 6.

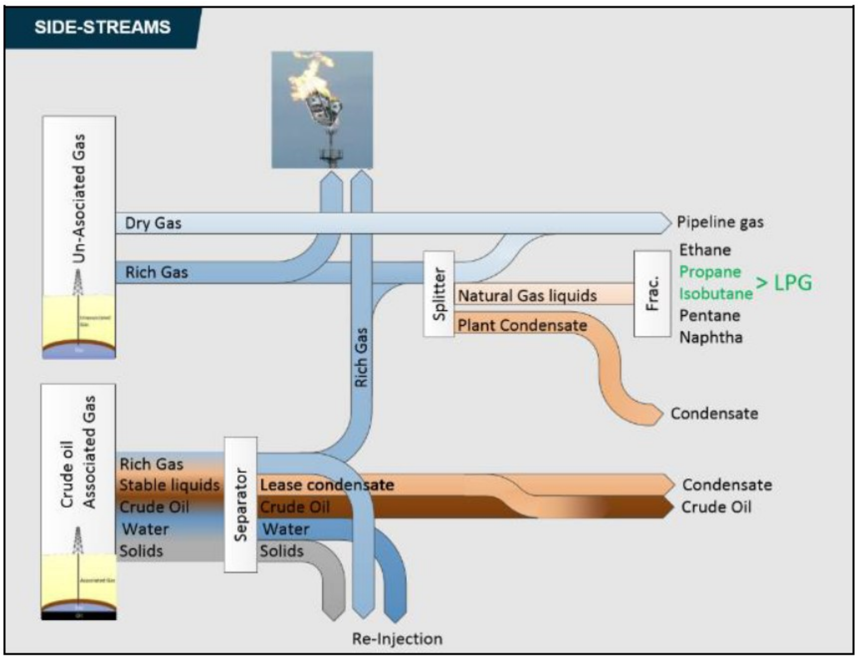


Figure 6. Side streams for associated and non-associated gases. [6]

Gas condensates concentrate an enormous amount of energy. The top ten countries where they are available can be seen in Figure 7.

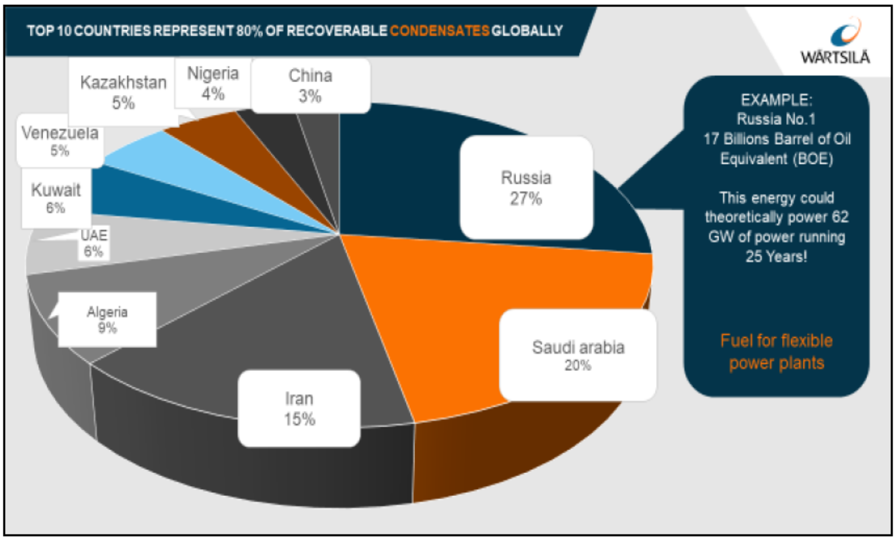


Figure 7. Top 10 countries representing 80% of condensates globally. [6]

Russia has the largest amount of condensate gases in the world, as it has 27% of the total globally available recoverable condensates, which is equivalent to 17 billion BOE (barrel of oil). This amount of energy is theoretically able to create 62 GW of power for 25 years with a LG flexible fuel power plant. This type of installation will be increasingly important, considering the "World Bank's routine flaring by 2030" program that sets the amount of flared gas to zero by 2030, as part of a broader drive to reduce Greenhouse Gases (GHG) emissions and to promote the utilisation of a valuable energy resource in global operation. [7]

In addition to the mentioned benefits, W32LG provides a significant reduction in terms of CAPEX for the customer, considering the higher power density provided by this technology. Direct benefits are identified in reduced number of engines, reduced power plant footprint with consequential saving in civil work and in lower OPEX (due to lower number of engines to maintain, considering the same power output).

Properties of some of the hydrocarbons present in the gas condensate range, and therefore in the LG fuel range, are presented in Table 2. A more detailed table of properties is presented in Appendix 3. These fuels are explored in experimental sections 4.1 (Combustion Research Unit CRU testing) and 4.3 (Engine testing).

LG technology

LG technology was selected in this development project, because it gives fuel flexibility requirements for both LPG and condensates, without limiting the power output of the engine. This technology solution consists of injecting the fuel at high pressure into the combustion chamber towards the end of the compression stroke. Main fuel is ignited by a micro pilot injection of LFO to establish a robust ignition, despite a poor main fuel quality. LG technology is based on the diesel cycle to avoid the risk of knock and pre-ignition phenomena, as the combustion starts as soon as fuel is injected. For this reason, W32LG delivers the full power output, without correlating with the fuel composition.

Table 2. Properties of LG fuels.

Fuel	LHV	LHV % of LFO	Auto-ignition temperature AIT	Flammability limit	Cetane number
	MJ/kg		°C	%	
LFO (reference fuel)	42.95	100			55
Propane	46.4	108.03	510	1.8-9.5	<3
n-butane	45.3	105.47	490	1.5-8.5	
MGO	42.8	99.65	>220	0.6-6.5	67.7
Gasoline	43.4	101.05	257	1.3-7.6	
n-pentane	45.36	105.60	309	1.5-7.8	25.9
n-hexane	44.57	103.78	234	1.2-7.4	46.4
n-heptane	44.57	103.76	223	1.1-6.7	61
Methanol	19.93	46.40	385	6.7-36.0	1.6
Ethanol	26.7	62.17	365	3.3-19.0	-5.1
Propanol	30.68	71.43	371	2.2-...	7.2
Butanol	34.4	80.09	345	1.7-12.0	
Cyclohexane	43.45	101.16	245	1.3-7.8	
Xylene	40.96	95.37	463-528	1.0-7.0	
Toluene	40.59	94.50		1.2-7.1	
Kerosene	43	100.12		0.7-5.0	70-100
Trimethylpentane	44.31	103.17	396		5
Isopentane	45.24	105.33	420	1.4-7.8	10.4

Despite an increased complexity in comparison to a diesel process, LG technology is justified by enabling an unprecedentedly wide fuel range, which is visible in Figure 8. Due to the specific fuel injection design for low-viscosity fuel, this technology cannot use heavy fuel oils, as all the clearances are too tight (as presented in the fuel injection chapter 3.2.2). Figure 8 shows a summary of possible fuels that Wärtsilä 4-stroke engines can manage, by utilising spark-ignited (SG), dual-fuel (DF), liquid gas (LG) and gas-diesel (GD) technology.

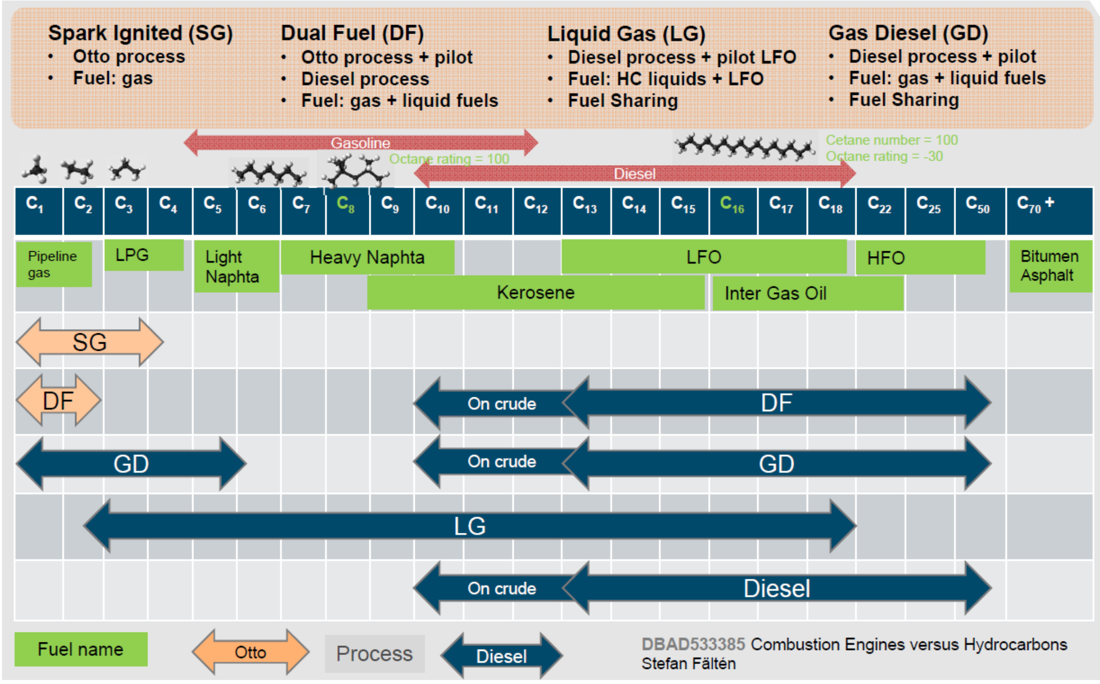


Figure 8. Hydrocarbon variations in Wärtsilä engines. [8]

The use of LPG fuel and LPG-LFO fuel blends have been researched before in different technology types. As background information about the research done with the lowest viscosity fuel in the LG range, Appendix 4 presents a brief overview of previously conducted research to enable LPG fuel application in diesel principle engines.

The following parameters have been set as target in the LG development project:

- Engine power output: 480/500 kW per cylinder at 720/750 rpm
- NO_x emissions within Emission World bank (710 ppm)
- Fuel consumption below 200g/kWh
- SCR optimised to achieve at least 320 °C exhaust gas temperatures at the stack

3.1.1 Environmental aspects

Environmental goals include limiting the amount of harmful exhaust gas emissions released by the combustion process (sulphur oxides (SO_x), nitrogen oxides (NO_x), carbon dioxide (CO_2), unburnt hydrocarbons and particulate matter (PM)). Emissions are regulated by organisations such as IMO, Marpol, US EPA, and World Bank. Some methods to reduce the amount of harmful emissions are:

- introducing cleaner burning fuels, for example:
  - low-sulphur LFO in CI engines; and
  - natural gas, LPG or alcohols, which are more commonly used in SI engines due to their properties, such as low CN;
- in-cylinder methods, which reduce emissions by optimising the combustion, for example by:
  - adjusting fuel injection timing and amount;
  - optimised geometry and design of the piston top and fuel injection equipment;
- exhaust gas recirculation (EGR) to reduce NO_x emissions;
- exhaust aftertreatment methods, also referred to as secondary emission control, include for example selective catalytic reduction (SCR), filters and scrubbers.

Increasing an engine's sustainability can also be done by utilising waste or side products of another process as fuel. Combining industrial processes helps to optimise local networks of distributed energy. Potential to develop this approach can be seen in oil and natural gas wells. During extraction processes at crude oil and condensate wells, a side-stream of hydrocarbons is often produced. These fuel fractions are known as liquid gases (LGs), hydrocarbon gas liquids (HGL), or natural gas condensates. They are often disposed of through flaring on-site, which creates CO_2 and other emissions without adding energetic or economic value to the process. The LGs cannot be used in the typical SI engine, due to properties such as low MN and possibly high amount of impurities, nor in the typical CI engine, due to an extremely low CN. Additionally, common CI engines are not equipped for operation with ultra-low viscosity fuels and extremely high pressures in the fuel supply system, which is needed to maintain LGs in liquid state. Thus, the LGs fall into an intermediate range of currently applicable fuels in internal combustion engines. There is a clear potential: developing appropriate technology for the use of LGs for power and/or heat production, for example at natural gas extraction sites.

3.2 LG engine

This chapter presents the technology details of the LG concept and the working principle of the W32 engine. As introduced in chapter 3.1 (Reasons for LG development), the development of LG technology is driven by the goals of increasing fuel flexibility in engine technology, maximising the power output and overall performance of the product by using process waste as fuel, which also goes hand in hand with environmental aspects. These goals are achieved by implementing an unprecedented fuel range in an engine that combines typical diesel-principle operation with additional features, such as twin-needle injectors to enable the use of pilot fuel to ignite the main fuel, and a common-rail system supported by a high-pressure fuel pump.

The Wärtsilä Liquid Gas engine (W32LG) is classified as medium-speed, compression-ignition internal combustion engine. It has been developed based on the mechanical design of the most recent design stage of the Wärtsilä 32 diesel-principle engine, which is the design stage W32E3. When testing the LG concept, the minimum performance requirement is to reach the same performance values as this base engine.

Figure 9 below demonstrates the W6L32LG laboratory engine. Development of the LG concept has been done by testing different fuels, setups with different external fuel systems and different fuel system components to optimise the concept.

Figure 9. W6L32LG laboratory engine.

Figure 10 illustrates the product engine W20V32LG, which is the target configuration of the development.

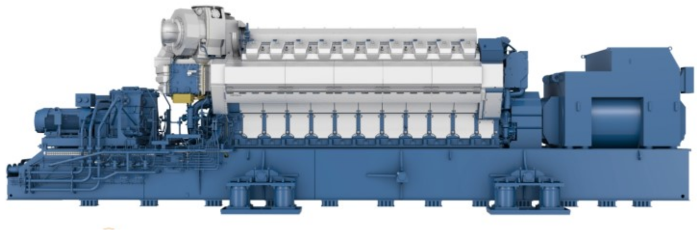


Figure 10. W20V32LG product engine.

3.2.1 Wärtsilä 32 Liquid Gas (W32LG) basic parameters

Table 3 presents the basic parameters of the W32LG engine.

Table 3. Main parameters of the W32LG engine. **[9]**

Parameter	**Unit**	**W32LG**
Working process		CI, direct injection, use of pilot fuel, main fuel in liquid phase
Cylinder configurations		6L laboratory engine and 20V product engine
Bore	mm	320
Stroke	mm	400
Engine speed	r/min	720 / 750
Theoretical compression ratio		16:1
Mean piston speed	m/s	10
BMEP	bar	24.9
Cylinder output	kW/cyl	480/500
Heat rate	kJ/kWh	8117
Max. firing pressure	MPa	23
Turbocharger		Napier
Max. injection pressure	bar	2000

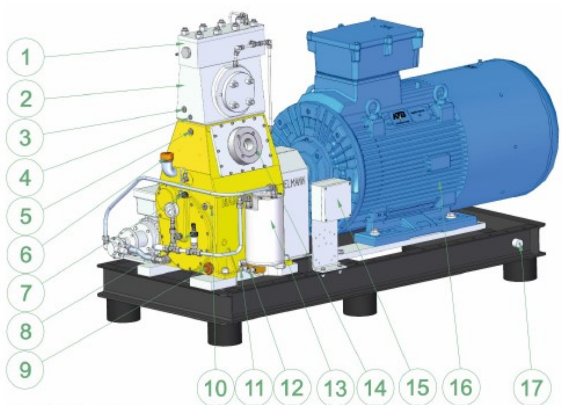
3.2.2 Fuel system

The core of the LG development project is the main fuel injection system that consists in a common rail system, operating up to 2000 bar. This system guarantees the pressure build up, the fuel delivery and the fuel injection in the combustion chamber. The following main components are identified in the fuel injection system:

- Common rail system for main fuel
- PDSV pressure drop and safety valve
- Twin-needle injectors
- Leaks during engine operation

Common rail system for main fuel

Common rail system is based on a common high-pressure line that delivers the fuel to each cylinder. The common high-pressure line is supported by the Hammelmann high-pressure fuel pump electrical driven, which is a product already used in other Wärtsilä applications (for methanol fuel).



Item	Description	Item	Description
1	Pump head	9	Oil drain plug
2	Suction chamber	10	Dip stick
3	Suction chamber drain	11	Cooling water in
4	Plunger cooling	12	Cooling water out
5	Fushing connection	13	Oil cooler/Filter combination
6	Oil filler cap	14	Exhaust intermediate chamber
7	Oil pressure regulator	15	Terminal box
8	Oil pressure sensor	16	E-motor (main drive)
		17	Earthing connection

Figure 11. Construction and layout of the Hammelmann high-pressure fuel pump. [22]

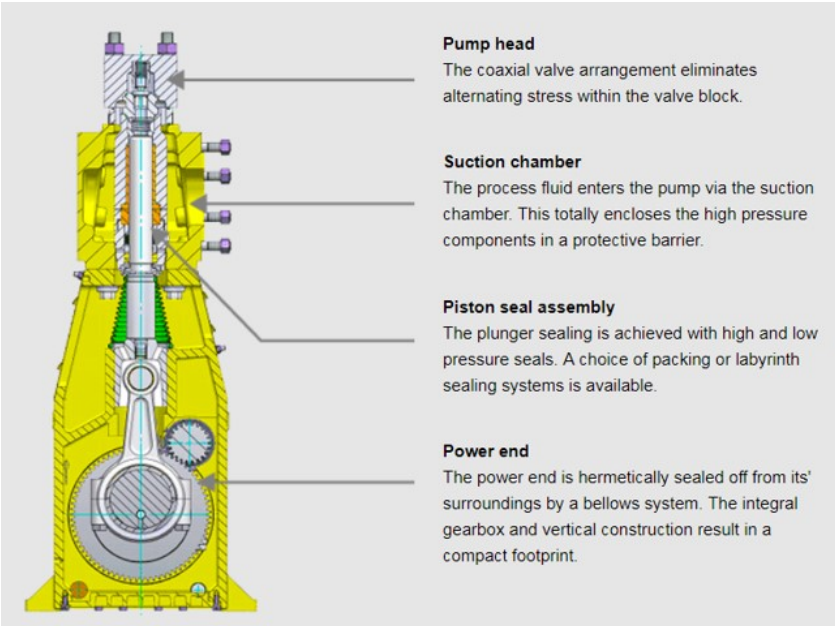


Figure 12. Details about the Hammelmann high-pressure fuel pump. [22]

This pump did not require a full development, but only an adaptation to the low viscosity fuel application. These activities were required to be compliant with full LG fuel scale. The pump supplier, after performing different tests, developed a new set of sealings which has a wider fuel compatibility and thus enables a wider fuel flexibility for the engine. This means the same pump, without any change, can handle light fuel oil and all LG fuels and provides a backup fuel option to the power plant, if needed. Pump characteristics are visible in Table 4. The chosen pump can provide up to 2000 bar fuel pressure, by operating in a speed range 100 and 1500 rounds per minute.

Table 4. Hammelmann pump specification. [10]

Pump number	HDP 204	Unit
Flow rate	27.7	l/min
Max. operating pressure	2000	bar
Flow medium	Propane, diesel	
Crankshaft speed	570	1/min
Motor speed	1448	1/min
Plunger (piston) diameter	20	mm
Number of plungers	3	
Rod force	88 000	N
Stroke	75	mm
Motor rating	142	kW
Frequency	50	Hz
Gear ratio	1 / 2.54	

Pressure Drop and Safety Valve

Pressure drop and safety valve (PDSV) is designed to operate the fuel injection system safely from engine and operator's point of view. Its characteristic allows to protect the system from unwanted overpressures and from pressure drop. Overpressures can be generated in case of engine control system failure, pump control system failure or from any unplanned situations. If pressure exceeds the limit, dangerous situations can occur, such as pipe break down, injector failure, pump failure, as well risks for the personnel or hazards (fuel spray can create risk of fire). In order to protect from such events, the opening pressure is adjusted at 2400 bar. Pressure drop safety occurs when pressure is below the requested value within a certain pressure window (approx. 50 bar). Pressure drop can occur in case of injector failure (over fuelling situation), excessive leakage from high-pressure pipe connections or any other failure that results in a loss of pressure. PDSV, in case of 6-cylinder engine is one and in 20-V configuration is one per bank. Figure 13 illustrates the PDSV.

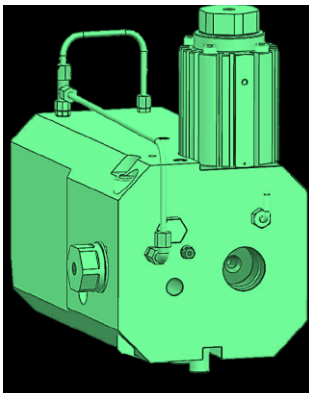


Figure 13. Pressure Drop and Safety Valve. [6]

Twin-Needle Injectors

The core of the LG fuel injection system is the twin-needle injector. This injector is characterised by one needle for the main fuel and one small needle for the pilot injection. Wärtsilä has extensively used this technology over 20 years in dual-fuel applications (diesel and gas engines), where the main needle provides the capability to run the engine in diesel mode and the pilot needle provides the capability of injecting a small amount of light fuel oil to ignite the gas fuel, previously admitted into the combustion chamber through a dedicated main gas valve (MGV). LG fuel injection system uses the same technology, but the injector needs some changes to handle low viscosity fuel and as well light fuel oil. This is a big challenge, because a compromise is needed to make the system work.

The fuel properties of LG are characterised by low viscosity (lower compared to light fuel oil). Based on the fuel analysis for LG fuels, a dynamic hydraulic analysis of the injector was performed with GT-SUITE simulation tool. The scope of this job was to identify the proper injector drillings, volumes and clearance optimisation. Drilling optimisation was needed due to the lower LHV compared to LFO, which is a typical characteristic of LG fuels. By changing the diameter of the holes, the injector was able to deliver increased fuel flow, while maintaining similar injection duration. Clearance was reduced to optimise and reduce the pressure losses. One difficulty was defining good injection parameters with LFO and LPG, respectively, without compromising the engine performance with either one. Figure 14 below shows part of the simulation, regarding the effect of the needle clearance. As visible in the picture, even slightly wrong parameters in the clearances can lead to no injection condition. Tolerances are kept in the tightest possible range, while enabling a reliable manufacturing.

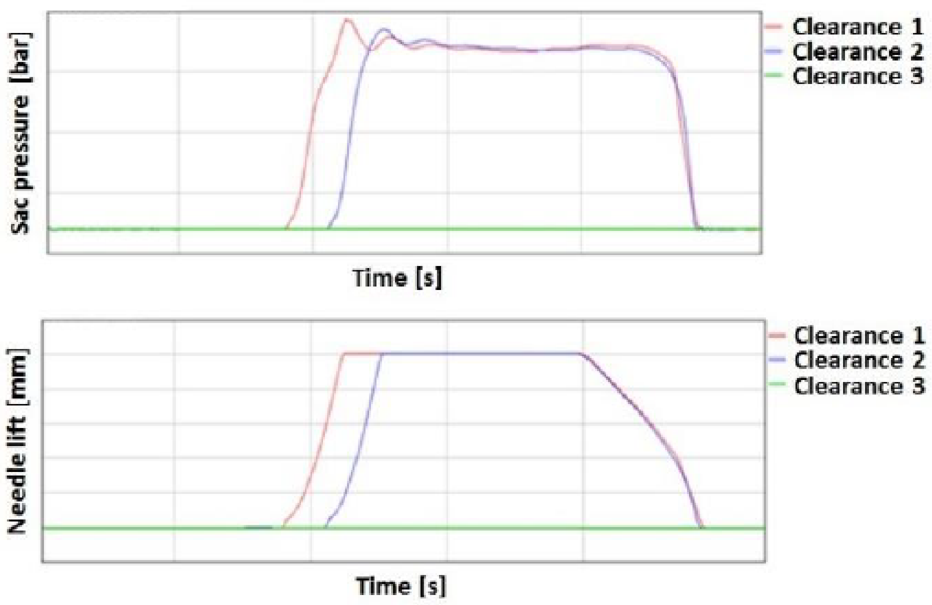


Figure 14. Hydraulic simulation of main fuel needle. [6]

Additionally, injection equipment components are subject to high pulsating stresses due to injection pressure up to 2000 bar on both main and pilot fuel lines, localised stresses due to the impact of valves and needle, wear and erosion phenomena due to high fluid velocities and dirty particles present in the fuel and high cavitation risk.

Extensive Finite Element Method (FEM) calculations were also done to optimise the lifetime of the components, considering the size limit of the injector that could be fitted inside the cylinder head and able to manage pressures up to 2000 bar, on the way down to combustion chamber. Material selection was done based on best available material and heat treatment knowledge.

Figure 15. LG fuel injector.

Based on the mentioned simulations (hydraulic and FEM), a detailed fuel system specification document was created to give the specification and boundaries to the supplier. The most significant parameters for the scope of this thesis are collected in Figure 16.

In general, performance simulations provided a valuable starting point for the spray pattern of the injector. Based on this, additional injector configurations were ordered for engine testing purpose, in order to identify the optimum spray pattern. This consisted of two different injection spray angles (+/- 5 degrees) and in two or more different hole diameters (usually, in a range +/- 0.03 mm). Before proceeding with engine testing, rig testing (described in Chapter 4.2) was needed to prove the reliability and the right material selection for the production version of the injector.

LG & Diesel main side		
Calculations are based on following **LG** properties	Fuel compression not considered	Density: 507 kg/m³ @ 15 °C (Propane) Lower heating value LHV: 42.0 MJ/kg (used as minimum, 46.3 MJ/kg for Propane) Viscosity min: 0.1 cSt @ 20 °C, 10 bar
SFOC in calculations		200 g/kWh @ 100/120% load
Injection volume at nominal load: 450 kW/cyl	8272 – 8313 mm3/stroke	LG, 100% load, 720 RPM - 0.5 – 1.0% pilot
Injection volume at nominal load: 460 kW/cyl	8117 – 8158 mm3/stroke	LG, 100% load, 750 RPM - 0.5 – 1.0% pilot
450 kW/cyl	9926 – 9976 mm3/stroke	LG, 120% load, 720 RPM - 0.5 – 1.0% pilot

Calculations based on following **MDO** properties:	Fuel compression not taken into account	Density 890 kg/m³ @ 15 °C Lower heating value: 42.7 MJ/kg
Injection volume at nominal load: 450 kW/cyl	4682 mm3/stroke	MDO, 100% load, 720 RPM
Injection volume at nominal load: 460 kW/cyl	4594 mm3/stroke	MDO, 100% load, 750 RPM

LG & Diesel main side		
Nozzle spray configuration		10*0.58*160°
Nozzle q100		21700 ml/min
Nozzle q100 value accuracy		±2%
Nozzle flow coefficient		0.87
LG & diesel main side o-rings		All LG & Diesel main side o-rings must be resistant to LG, Diesel, lube oil and mix of these mediums.
Maximum injection duration		20° CA with LG (simulated at 100%) 16° CA with MDO (simulated at 100%)
LG & diesel main side volume location	In injector	LG & diesel main side volume should locate as close as possible to nozzle. Drillings specified according to hydraulic simulation.
LG & diesel main side volume size		40-60*100% load injection (~332000 – 499000 mm³) only as a target

Diesel pilot side		
Nozzle spray configuration		4*0.2*155° (with 2000 bar)
Nozzle q100		1028 ml/min
Nozzle q100 value accuracy		±2%
Nozzle flow coefficient		0.87
Nominal injection duration		Target ~8° Crank
Maximum injection duration		20° CA (simulated at 5%)
Diesel pilot volume location	Before injector	Diesel pilot volume should locate as close as possible to injector. Drillings specified according to hydraulic simulation.
Diesel pilot volume size		40-60*5% load injection quantity (~14000 – 21000 mm³) only as a target

Nozzle orientation		Main nozzle: First spray hole at the opposite side of the main quill pipe Pilot nozzle: First spray hole at 45° of the opposite side of the main quill pipe Specified also in nozzle body drawing DAAF081640

Figure 16. LG fuel system specification.

Leaks

High-pressure fuel injection systems for classification and safety point of view require a double-wall pipe to collect leakages and protect from fuel spray, in case of pipe breakage. Fuel leakages are a consequence of small fuel leakages coming from each and every high-pressure connection around the engine. These leakages are collected from a common leakage line and addressed to a dedicated tank, in case of LFO engine. In the LG engine, due to the higher complexity of the fuel injection system, three different leakage lines are identified:

- Main clean leakage. Based on the fuel operation (LPG or LFO), fuel leakages are addressed to different collecting points, due to the nature of the fuel (gas or liquid state at atmospheric conditions). Liquid leakages are coming from the LFO operation and are collected in order to be re-utilised (this fuel is pumped back to the fuel tank). LPG leakages in gas form at atmospheric conditions are following a different path, compared to the liquid leakages and this is automatically selected by the fuel operation. Leakages are collected in a tank with 10 bar design pressure, that is connected to a burner to eliminate the fuel.
- Pilot clean leakage. As for the main clean leakage, this system is common and fuel is re-used by sending it back to the fuel tank.
- Mixed leakage (fuel and oil). This leakage comes from fuel mixed with oil used for the injector cooling. Leakages are addressed to a fuel separator that removes the oil from the fuel. Lube oil and LFO mixed with oil is sent to the sludge tank. In LPG mode, this leakage is sent to the tank that leads to the burner to be eliminated.

Figures 17-19 illustrate the cylinder head fluid lines, leak lines and engine hot box overview.

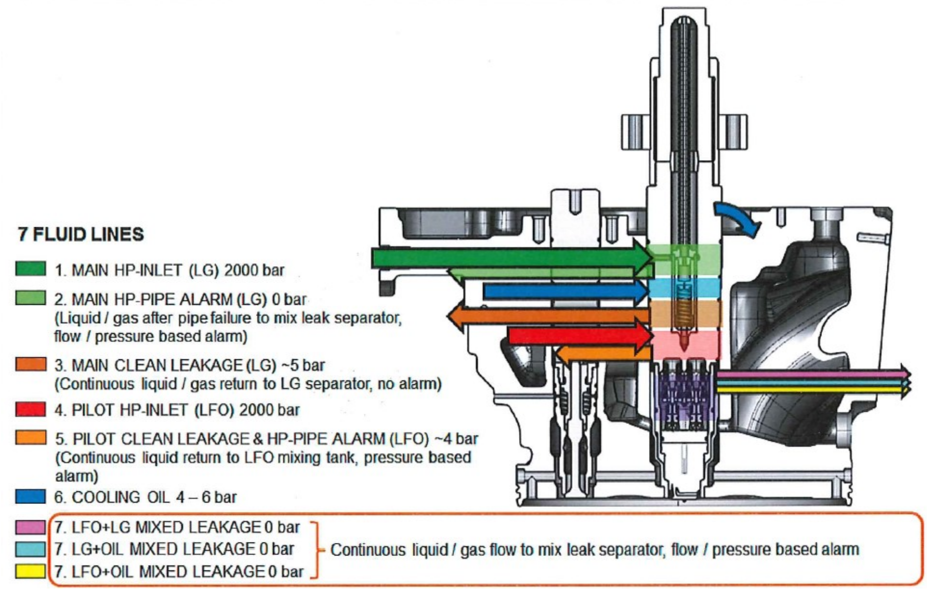


Figure 17. LG cylinder head fluid lines.

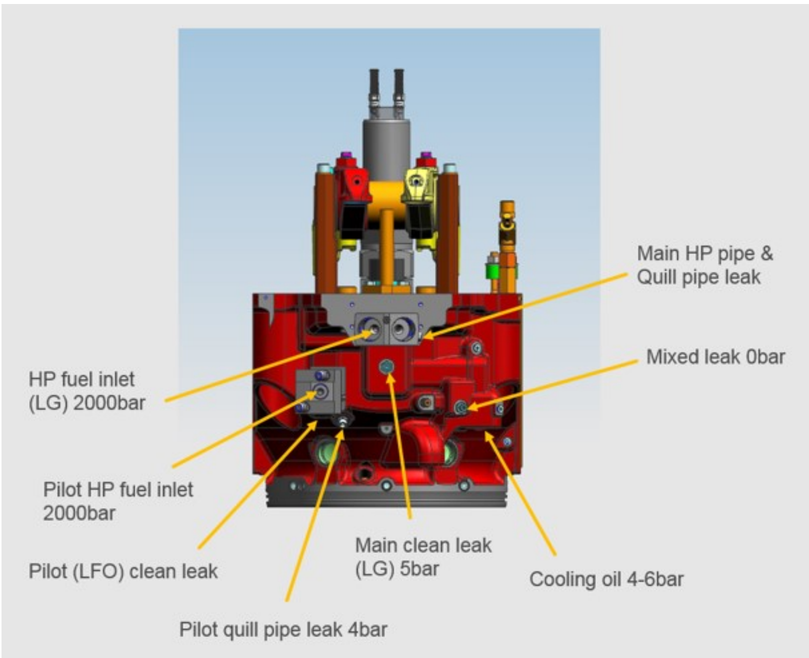


Figure 18. LG cylinder head leak lines.

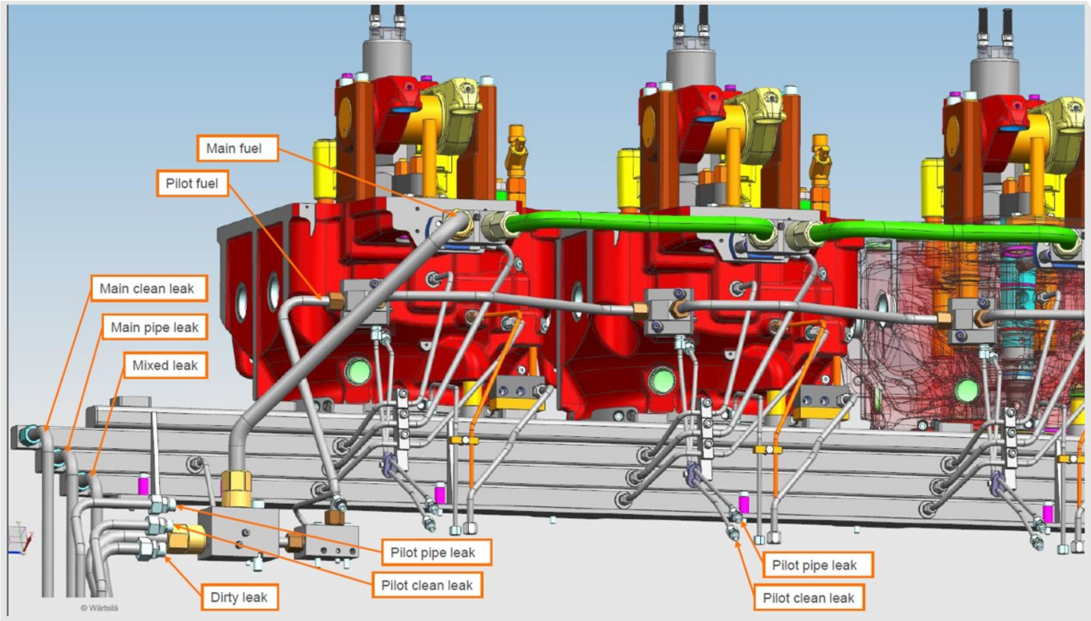


Figure 19. LG engine hotbox overview.

3.2.3 Control requirements

The automation and control system of the LG engine is an embedded system developed by Wärtsilä for 4-stroke engines, which includes a combination of hardware and software (UNIC 2 Series 5 and UNITool) especially developed to enable the functionalities needed in Wärtsilä engines. While this automation system is already applied in other engines, some additional or modified functionalities will be needed to operate this new engine. The added LG functionalities include:

<u>Pilot fuel related control:</u>

Pressure control

Regardless of the main fuel type used in the LG engine, pilot fuel injection is always implemented. LFO pilot fuel will be supplied through an electrically driven pilot fuel pump. Engine automation will include a pilot fuel pressure control. An average of two pressure measurements is taken and used in a closed PID controller loop to control a pilot fuel flow-control valve. The reference input for the PID, which ensures that the engine receives the adequate amount of pilot fuel for the different running settings, is a control map based on engine speed for starting mode and engine load for running mode.

Injection control

Injection control is control map based, depending on engine speed and engine load (BMEP). Settings allow to change the injection timing/duration to enable the proper start of injection. For this control, there are two sets of maps, based on the fuel used. Below the LPG maps (PFI control)

Combustion check

This safety is built to verify the pilot injector functionality during engine start up. It consists in checking that all exhaust gas temperatures after each cylinder are increasing and staying within a certain window. This check guarantees that all cylinders have the pilot injection working and protects the engine from misfire or late combustion, due to missing or insufficient pilot injection. This safety test is performed during each and every start and it takes approximately 5 – 30 seconds.

<u>High-pressure main fuel pump related control:</u>

The high-pressure fuel pump regulates pressure in the common-rail. In the V-form engine configuration, one pump per bank will be used. However, when starting the engine, only one pump is used. Control for this pump follows a similar principle as the pilot fuel pump: the average of two fuel pressure measurements is taken and used in a closed loop PID controller. The reference input for the common-rail pressure in this PID is a control map based on engine load (BMEP) and speed.

A condition for the main fuel pressure pump is that the combustion check sequence during the starting mode of the engine has been passed. If the common-rail pressure drops below a pre-defined value, the engine will shut down. If the common-rail pressure rises above a pre-defined value, the PDSV opens to release rail pressure to a safe level.

Stop and standby control:

Start/stop and standby engine mode was developed for the LG engine. This consists to integrate the mentioned engine mode with the main fuel high-pressure pump. Pump control unit is connected to the automation system of the engine (Unified Control and Monitoring System, UNIC) to provide its status. If there are fault codes or the pump is not ready to start or operate, then the same applies to the engine. In that case, it will remain in stop/standby mode. In the V form engine, start can be achieved when only one pump is operated up to a certain maximum load.

Start sequence control:

The start sequence consists of the following steps, aimed to ensure that combustion begins:

- The start solenoid valve is opened to start intaking fuel, and circulation valve and PDSV are closed.
- Pilot fuel injection starts at a pre-defined engine speed.
- The pilot fuel pump is activated to ensure there is enough level of pressure to supply the pilot fuel to the engine, according to the pilot fuel map (pressure, duration and timing).
- Before reaching nominal engine speed, a combustion check sequence is performed. If this check is not passed, the engine shuts down.
- After the combustion check, the first high-pressure fuel pump is started and, after a certain pressure level is reached, main fuel injection begins.
- Then the engine continues to speed up to reach nominal speed, using a normal speed/load control.

Engine running mode control:

A circulation valve located close to the high-pressure fuel pump is closed after the engine reaches a pre-defined low load level. The rail pressure then keeps the valve mechanically closed during normal engine operation.

Main fuel pressure is enabled with one fuel pump in the beginning, until the PID control reaches a high threshold. At this point, the second pump is started and ramped up to the same PID control level as the first pump. The PID control is made so that both pumps operate when needed, and only one pump operates if the engine load is reduced enough.

<u>Engine stopping control:</u>

After unloading the engine, rail pressure is reduced to a predefined shutdown level, which protects the PDSV by keeping it closed at this stage. The engine then enters the shutdown mode, where high-pressure pumps and main fuel injections are stopped and PDSV is opened.

<u>Machinery protection / emergency shutdown control:</u>

If an engine shutdown due to machinery protection occurs, the engine enters in shut-down mode. High-pressure fuel pumps and main fuel injections are stopped and PDSV is opened.

<u>SCR control:</u>

This functionality consists of actively controlling the exhaust gas temperature after the engine (turbocharger outlet to exhaust gas stack). It consists of adjusting the charge air pressure, by controlling the exhaust gas wastegate or charge air wastegate. For this control, there is a dedicated map where the target temperature can be set along the load range. In general, on 4-stroke medium speed engines, this temperature is kept between 300 and 420 °C to guarantee an efficient reaction between exhaust gas and the reagent (urea or ammonia).

<u>Wärtsilä 32E3</u>

The W32LG has been developed based on the W32(E3) engine. This engine has been a successful product in the marine and power sectors since the 1980s and has a vast experience in these fields. It exists in 6, 7, 8 and 9 cylinder in-line configurations and 12, 16, 18 and 20 V-form configurations. It operates at a speed of at 750 r/min for 50 Hz electric power grids, which is used in a large part of the world, and at 720 r/min for 60

Hz electric power grids, for use in USA, some parts of Asia and in marine vessels. The engine's rated power output ranges between 3 MW and 9.3 MW, depending on the number of cylinders. W32 is often used as the main engine in various vessel types, such as tankers, container vessels, cruise and ferry. It is also used as an auxiliary engine for electricity production in vessels that require high auxiliary load and in power plant applications. For emission control, this engine can be equipped with an SCR catalyst, which significantly reduces NO_x emissions. Additionally, it utilises Variable Inlet Valve Closure timing (VIC), which regulates the amount of intake air. This allows to close the inlet valves earlier when operating on higher load, which helps to reduce both NO_x emissions and fuel consumption. A delayed closing of the inlet valves, on the other hand, improves performance and helps to reduce smoke levels at lower engine loads and during transient mode. The control system of this engine includes both automatic monitoring and adjustable control to optimise engine efficiency at different operation modes. The original W32 burns diesel fuel of different categories: light and heavy fuel oil (LFO and HFO).

4 Experimental Methods

This chapter describes the tests done in scope of this development project, together with their results and main findings. As mentioned in the introduction, the tests consisted of:

- Fuel testing in a CRU. Some fuels in the LG range where selected and tested to evaluate their ignitability and combustion response, before introducing them into the engine.
- Fuel injection rig testing. The goal was to identify the material and geometry validation of fuel injection components.
- Engine testing on W6L32LG to develop LG technology, define preliminary performance results and identify possible limitations of this technology. Based on these results, the work of testing and optimising the 20-cylinder V-form W20V32LG, which is the target for the LG product, will be carried on.

4.1 Fuel tests in a Combustion Research Unit (CRU)

The CRU is a constant-volume chamber that can be used to simulate the combustion process in a CI engine. It consists of the constant volume chamber, fuel injector with one nozzle for pilot and another for main fuel, and sensors to measure pressure and temperature inside of the chamber. The parameters that can be controlled are chamber's temperature and pressure, fuel pressure and injection duration and injection timing of the main and pilot fuels relative to each other. This simulation setup cannot fully reflect a real engine's working cycle, since there is no piston or valve movement and it lacks conditions such as turbulence and expansion work. However, it is useful to view the differences in the combustion that arise from differences in fuel properties. It can provide valuable insights about combustion behaviour such as ignition delay, combustion speed and propagation and heat release rate.

4.1.1 Testing setup

The settings for the CRU have been selected to reproduce the most critical conditions for poor quality fuels in terms of ignitability. The diesel engine can suffer from misfire and unstable combustion when using fuels with low cetane number, especially during start and low load operation. The scope of the CRU was to replicate the idle and low-load operation, where both compression pressure and temperature at the end of the compression stroke are low compared to the optimum value. While a diesel engine usually operates with fuels that have a cetane number above CN 45, LPG has CN below 3. This means that LPG will not ignite spontaneously in a diesel process, but a pilot fuel injection is needed to start the combustion. The scope of the testing was to evaluate the performance of low CN fuels in terms of ignitibility. For this test, the following fuels have been used to evaluate the combustion quality:

- straight-chain hydrocarbons: n-pentane, n-hexane, n-heptane.
- alcohols: ethanol, methanol, propanol, butanol.
- cyclic hydrocarbons: cyclohexane, xylene, toluene.
- other fuels: kerosene, isopentane, trimethylpentane and two different naphtha samples.

Physical and chemical properties of these fuels are presented in Table 2 (Properties of LG fuels).

The chosen settings for the CRU tests are presented in Table 5. The scope of these tests was to identify:

- pilot fuel requirement: this consists to define if pilot fuel is needed to start the combustion and its amount expressed in pilot duration.
- heat release rate: to understand flame propagation, based on HRR.
- fuel comparison with LFO and HFO from HRR point of view. This compares the HRR of high CN fuels versus the low CN fuels under testing, supported by the pilot fuels.

Table 5. Settings for fuel testing in the CRU.

CRU setup	Combustion chamber pressure [bar]	Combustion chamber temperature [C]	Main fuel injection duration [µs]	Main fuel injection pressure [bar]	Pilot fuel injection duration [µs]	Pilot fuel injection pressure [bar]
Setup 1: Engine idle	55	550	1500	1000	No injection / 250 µs / 350 µs	800
Setup 2: Engine low-load	70	590	1500	1000	No injection / 250 µs / 350 µs	800

4.1.2 Test results

The test results in this chapter present firstly the graphs for idling conditions and then for low-load conditions. In the figures presented, the red line represents LFO and black line is HFO. They have been added to each graph to be used as a reference. The first graph shows chamber pressure the second graph shows the rate of heat release (RORH), both as a function of time and starting from the moment when fuel is injected.

Straight-chain hydrocarbons: n-pentane, n-hexane, n-heptane

The tested straight-chain hydrocarbons ignited without the use of pilot fuel, both at idling and low-load conditions. The performance of these fuels resembled the results obtained with LFO and HFO. N-heptane had similar combustion characteristics as LFO: ignition timing (delay), heat release rate and pressure curve of these two fuels were almost overlapping in both tested scenarios. The combustion characteristics of n-hexane and n-pentane were very similar to each other and closely resembled those of HFO. However, in comparison to HFO, their ignition delay was slightly longer, with a difference of approximately 0.6 µs. In practice, this can be optimised by changing the injection timing of these fuels.

Figure 5 illustrates the results obtained about the combustion characteristics of straight-chain hydrocarbons at a) idling condition and b) low-load condition with only main fuel injection events, since pilot fuel was not required.

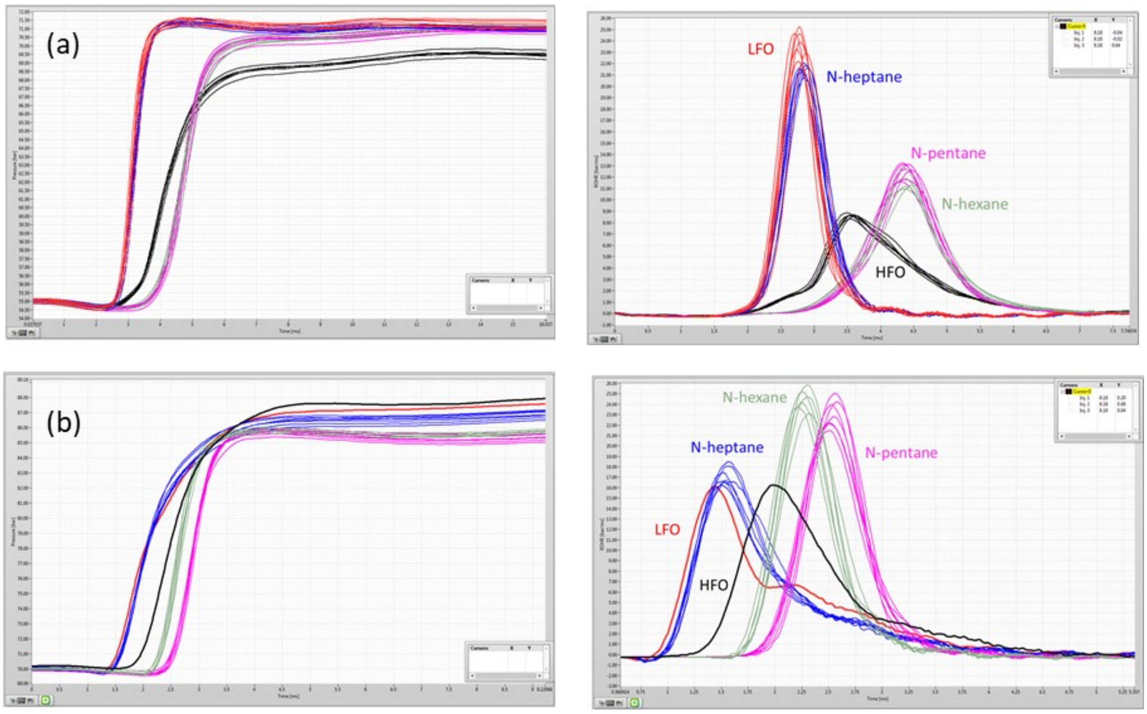


Figure 20. N-pentane, n-hexane and n-heptane pressure and ROHR at a) idling condition and b) low-load condition. [23]

Alcohols: methanol, butanol, propanol

Alcohol fuels did not ignite without pilot fuel injection. However, with the addition of LFO pilot fuel, combustion occurred in a stable and replicable manner. Figure 21 illustrates results achieved at idling conditions, where 350 µs pilot injection duration was necessary to stabilise the combustion. At low-load conditions (higher temperature and pressure than idle) a smaller amount of pilot fuel (250 µs) was sufficient to stabilise the combustion, so 350 µs pilot injection duration was not tested. When a stable combustion of alcohol fuels was achieved, it resembled a ROHR which is comparable to LFO.

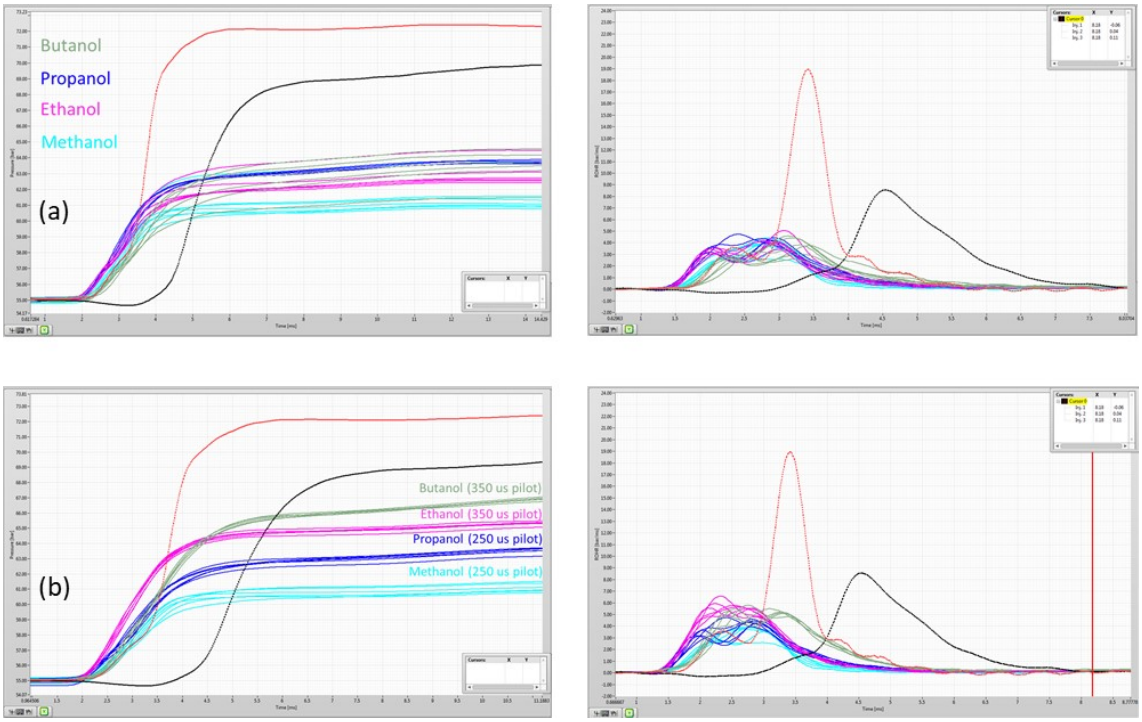


Figure 21. Methanol, ethanol, butanol and propanol combustion at idling condition with a) 250 µs and b) 350 µs pilot injection durations. [23]

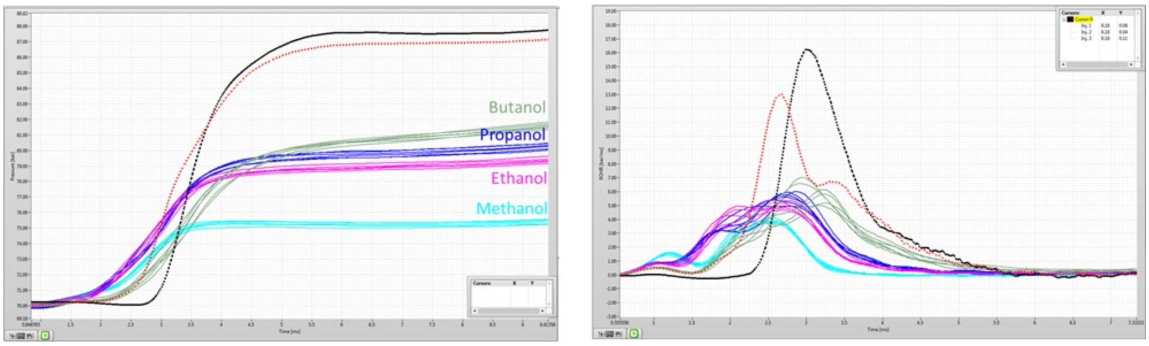


Figure 22. Methanol, ethanol, butanol and propanol combustion at low-load condition with 250 µs pilot injection duration. [23]

Cyclic hydrocarbons: cyclohexane, xylene, toluene

Figure 23 illustrates the combustion of cyclic hydrocarbons at idling condition. In 23 a), it is visible that cyclohexane was the only fuel in this category which ignited without pilot fuel injection. Its combustion was comparable to HFO, but the ignition delay was longer by approximately 2.5 µs. In 23 b), pilot fuel was injected for 250 µs and tested with these three fuels. In the case of cyclohexane, ignition delay was reduced, but the pressure increase was deteriorated, and repeating pressure peaks were visible (multiple combustion). With this pilot fuel injection duration, xylene and toluene were also able to ignite, but their combustion was not stable. A longer pilot fuel injection, visible in 23 c), helped to improve the combustion for all three cyclic hydrocarbons, which were then able to combust in a more stable way with acceptable combustion speed and propagation. However, repeating pressure peaks were still present to a smaller extent.

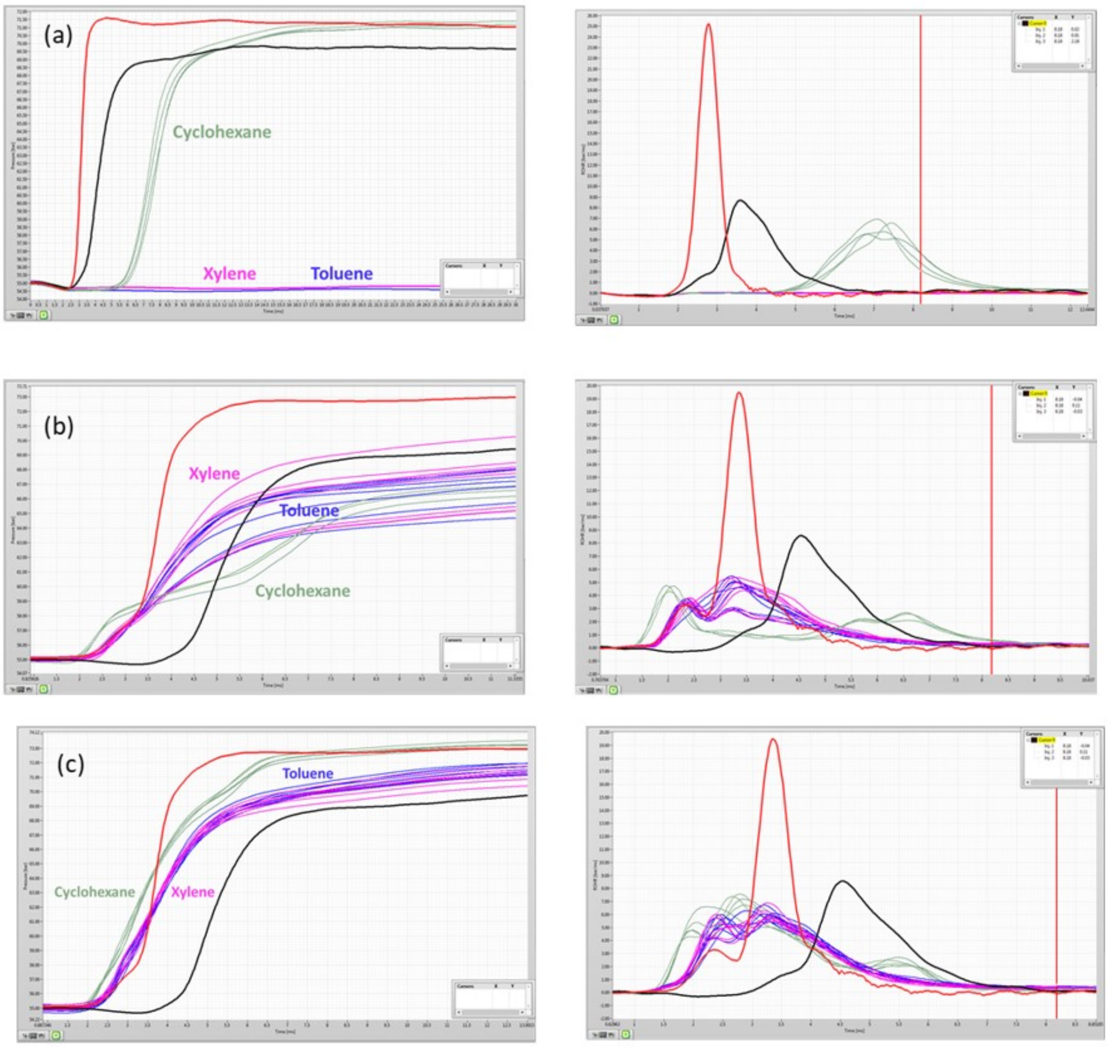


Figure 23. Cyclic hydrocarbon (cyclohexane, xylene and toluene) combustion at engine idling conditions with a) only main fuel injection, and pilot fuel injection duration of b) 250 µs and c) 350 µs. [23]

Low-load condition of cyclic hydrocarbons is presented in Figure 24 a-c, and its results confirmed observations from idling conditions for all tested events. In the case of cyclohexane, using a longer pilot fuel injection (24 c) resulted in more defined repeating pressure peaks, which is not a desirable outcome.

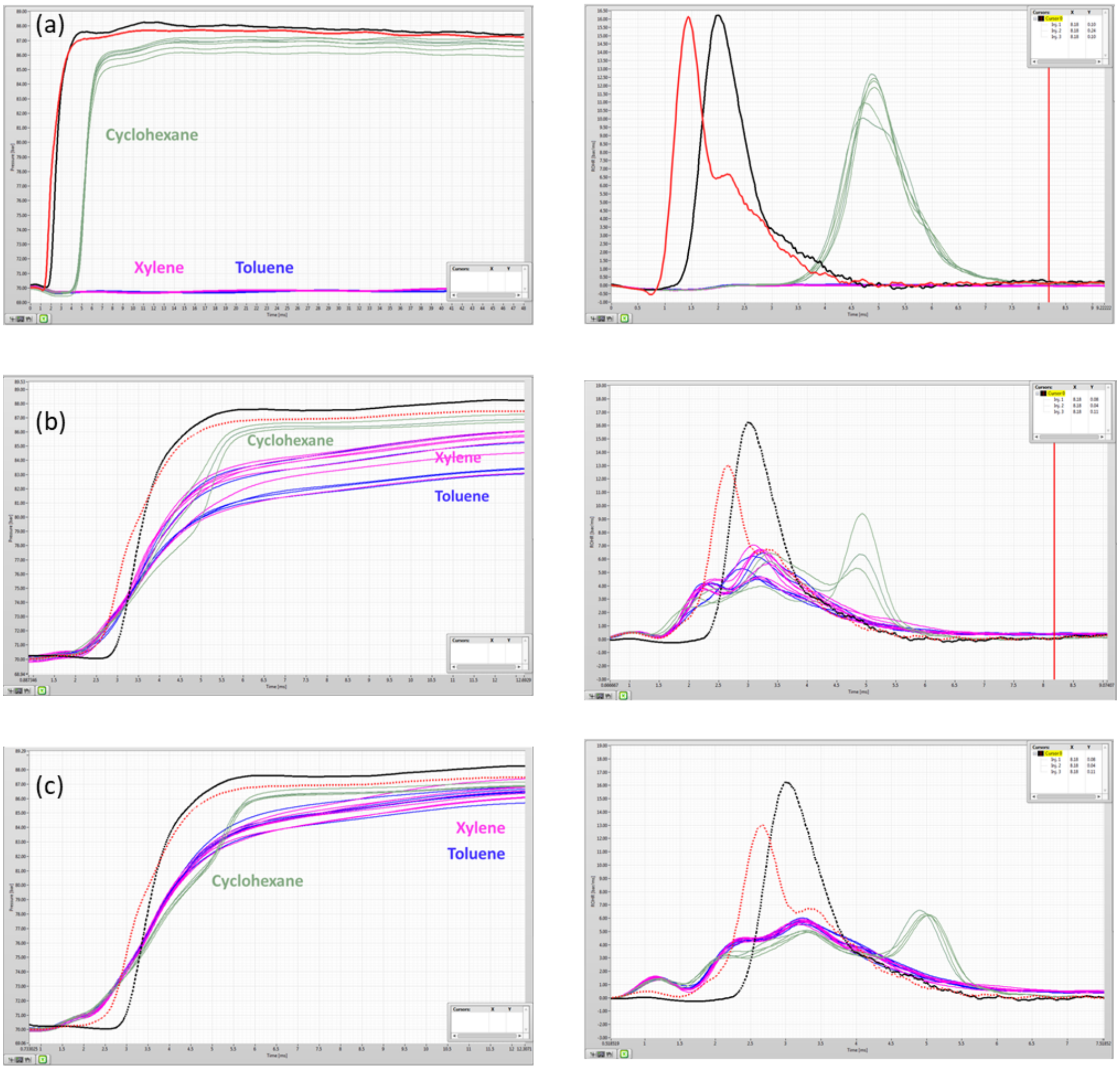


Figure 24. Cyclic hydrocarbon (cyclohexane, xylene and toluene) combustion at engine low-load conditions with a) only main fuel injection, b) pilot fuel injection 250 µs and c) pilot fuel injection 350 µs. [23]

Other fuels: Kerosene, isopentane, 2 different naphtha samples

Kerosene demonstrated to be a promising fuel candidate, which ignited without pilot fuel injection. While ignition delay was slightly long, this can be adjusted through engine calibration.

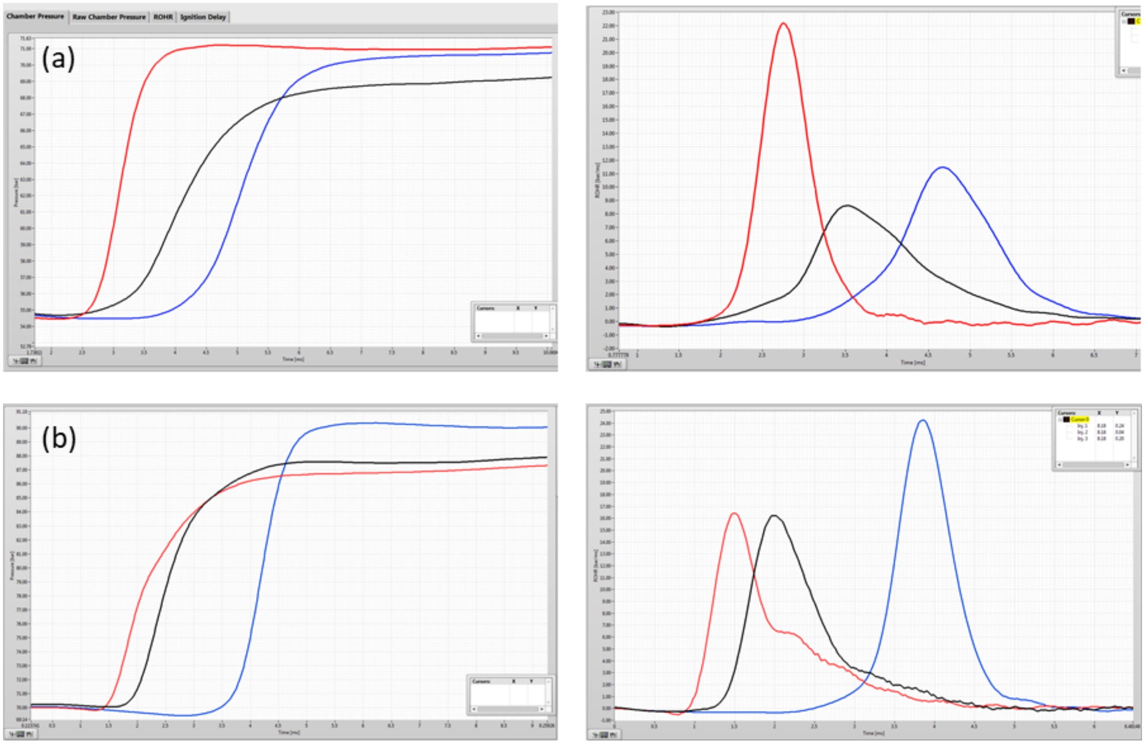


Figure 25. Kerosene combustion with only main fuel injection at a) idling and b) low-load conditions. [23]

Isopentane demonstrated similar behaviour to cyclohexane, with an overall steady combustion without pilot fuel injection but with long ignition delay. The addition of pilot fuel resulted (250 µs) in less stable combustion (repeated pressure peaks). These observations could be seen in both idle (Figure 26) and low-load (Figure 27) conditions.

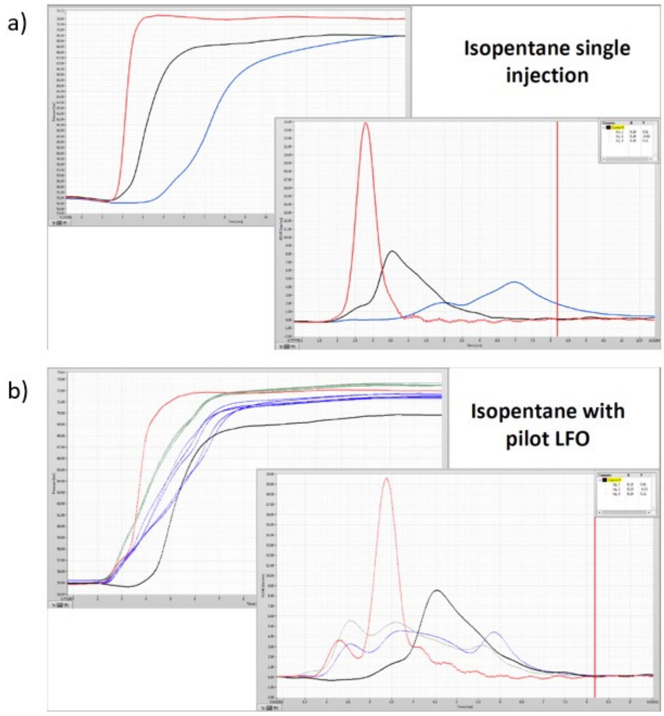


Figure 26. Isopentane combustion at idle condition with a) only main fuel injection and b) pilot fuel injection 250 µs. [23]

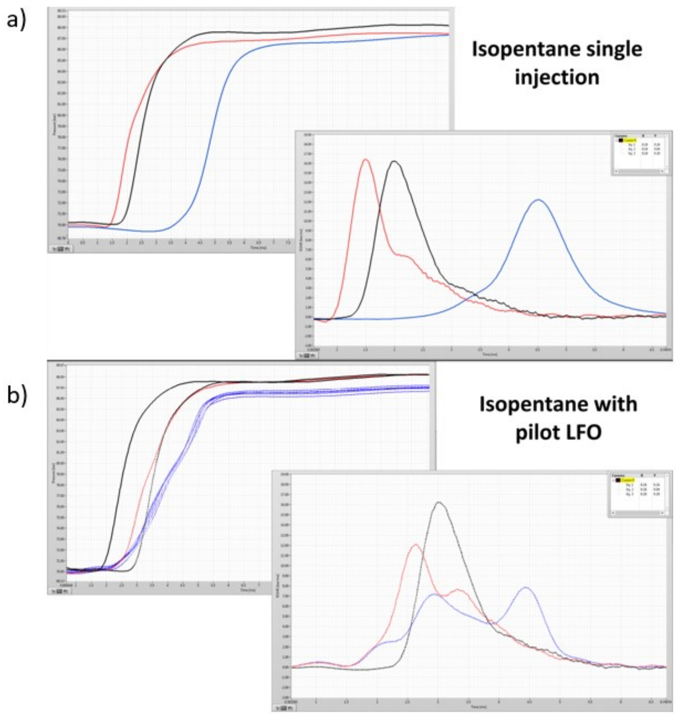


Figure 27. Isopentane combustion at low-load condition with a) only main fuel injection and b) pilot fuel injection 250 µs. [23]

Two naphtha samples with different octane number were tested in conditions simulating idle operation of the engine. The use of pilot fuel decreased ignition delay, but multiple pressure peaks were noticed in both samples.

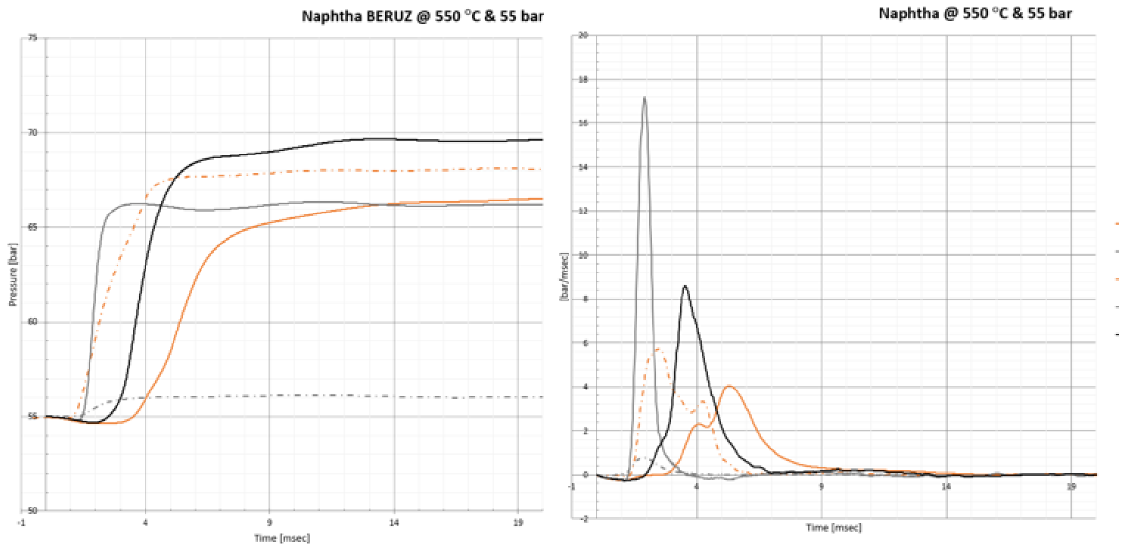


Figure 28. Naphtha sample 1 (RON 60) combustion at idle condition. [23]

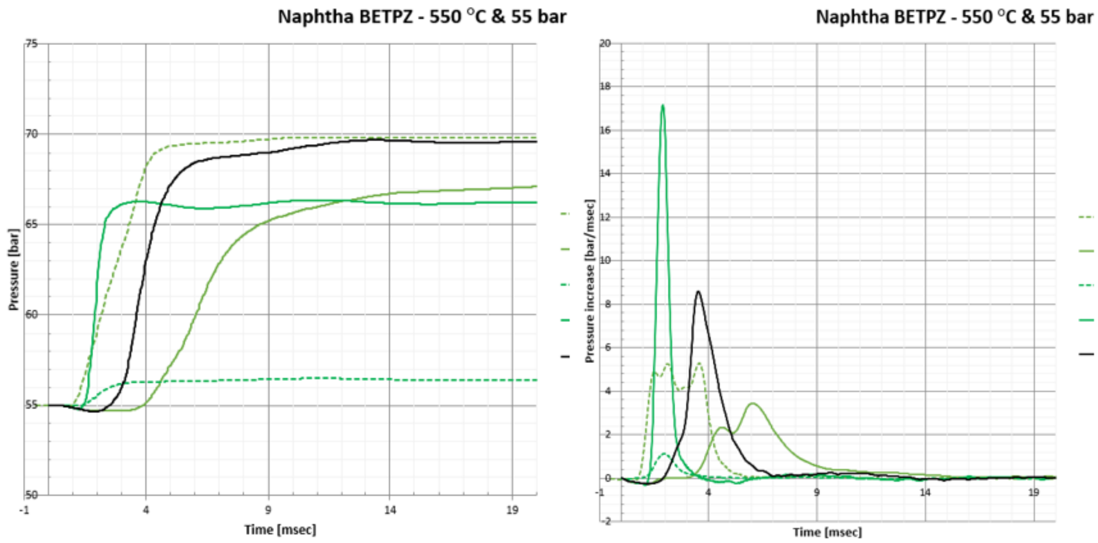


Figure 29. Naphtha sample 2 (RON 70) combustion at idle condition [24].

Tables 6 and 7 summarise the results of the fuel tests. Straight chain hydrocarbons demonstrated promising results, as the combustion stability, ignition delay and ROHR were comparable to LFO. Alcohols are also a promising fuel type. However, they require pilot fuel injection. Additionally, their lower energy content (approximately half compared to LFO) needs to be considered by changing injection duration.

Cyclohexane was the most promising fuel candidate from the category of cyclic hydrocarbons. However, its ignition delay was long. Isopentane demonstrated similar combustion behaviour. Kerosene demonstrated a stable combustion without pilot fuel injection, with a slightly longer ignition delay than LFO. Isopentane combustion without pilot fuel injection had a long ignition delay and instability when pilot fuel was added. Naphtha of both tested grades demonstrated a long ignition delay without pilot fuel injection and unstable combustion with the use of pilot fuel.

Table 6. Summary of the CRU fuel test results for LFO, n-pentane, n-hexane, n-heptane, methanol, ethanol, butanol and propanol. [11]

Fuel type	CN	Idle (1) or low-load (2)	Only main fuel injection	Main fuel ignition delay	PFI duration 250 µs	PFI duration 350 µs	Combustion speed	Comments
Reference fuel: LFO		1	OK	Good	-	-	Fast	
		2	OK	Good	-	-	Fast	
N-pentane	26	1	OK	Slightly longer than HFO	Not needed	Not needed	Quite fast	Slightly worse than HFO
		2	OK	Slightly longer than HFO	Not needed	Not needed	Quite fast	Slightly worse than HFO
N-hexane	46	1	OK	Slightly longer than HFO	Not needed	Not needed	Quite fast	Slightly worse than HFO
		2	OK	Slightly longer than HFO	Not needed	Not needed	Quite fast	Slightly worse than HFO
N-heptane	61	1	OK	Good, same as LFO	Not needed	Not needed	Fast	Similar to LFO
		2	OK	Good, same as LFO	Not needed	Not needed	Fast	Similar to LFO
Methanol	1.6	1	No ignition	Shorter than LFO	OK	Not needed	OK	Longer injection needed compared to LFO, based on the fuel's energy content. This makes direct comparison more difficult. Energy equivalence test can be done for alcohol fuels.
		2	No ignition	Shorter than LFO	OK	Not needed	OK	
Ethanol	- 5.1	1	No ignition	Shorter than LFO	Not sufficient	OK	OK	
		2	No ignition	Shorter than LFO	Not sufficient	OK	OK	
Butanol		1	No ignition	Shorter than LFO	Not sufficient	OK	OK	
		2	No ignition	Shorter than LFO	Not sufficient	OK	OK	
Propanol	7.2	1	No ignition	Shorter than LFO	OK	Not needed	OK	
		2	No ignition	Shorter than LFO	OK	Not needed	OK	

Table 7. Summary of the CRU fuel test results for cyclohexane, xylene, toluene, kerosene, isopentane and naphtha. [11]

Fuel type	CN	Idle (1) or low-load (2)	Only main fuel injection	Main fuel ignition delay	Pilot fuel injection duration 250 µs	Pilot fuel injection duration 350 µs	Combustion speed	Comments
Cyclohexane		1	OK	Quite long (2.5ms longer than HFO without pilot injection)	Not critical, but helped to reduce ignition delay, but led to multiple pressure peaks.	Helped to further stabilise combustion.	OK	More stable combustion without pilot fuel.
		2	OK	Quite long without pilot injection	Multiple pressure peaks.	Multiple pressure peaks.	OK	
Xylene		1	No ignition	OK with longer pilot injection	Unstable combustion	OK	OK with longer pilot injection	Good candidate with sufficiently long pilot injection duration.
		2	No ignition	OK with longer pilot injection	Slightly unstable combustion	OK	OK with longer pilot injection	
Toluene		1	No ignition	OK with longer pilot injection	Unstable combustion	OK	OK with longer pilot injection	Good candidate with sufficiently long pilot injection duration.
		2	No ignition	OK with longer pilot injection	Slightly unstable combustion	OK	OK with longer pilot injection	
Kerosene	70-100	1	OK	Quite long	Not needed	Not needed	OK	Good candidate with engine calibration to manage long ignition delay.
		2	OK	Quite long	Not needed	Not needed	OK	
Isopentane	10	1	OK	Quite long	Multiple pressure peaks	Multiple pressure peaks.	OK	Similar behaviour as cyclohexane.
		2	OK	Quite long	Multiple pressure peaks	Multiple pressure peaks.	OK	
Naphtha		1	Late ignition	Improved with pilot injection	Two pressure peaks	-	OK	Pilot injection decreased ignition delay, but pressure peaks remained.
		2	Late ignition	Improved with pilot injection	Multiple pressure peaks	-	OK	Pilot injection decreased ignition delay, but pressure peaks remained

4.2 Rig testing

Rig testing is an established method used by engine manufacturers to test and validate critical components, for example those included in a new fuel injection system. In this case, it consists of creating a test bench that reproduces the engine boundaries to test fuel injection components. The rig represents a faster and simplified way of testing fuel pumps, fuel injectors, high-pressure pipes and other new components, without needing to run an engine. It is time and cost-efficient. As there is no combustion process and hence no fuel consumption, only a small amount of fuel is involved.

In the LG development, rig activities are divided into two parts:

- Fuel injection system performance: this consists of evaluating characteristics of the main components.
- Endurance test: this consists of accumulating a certain number of hours, usually in the order of 100s or 1000s running hours, to validate the fuel injection design and identify the best material.

4.2.1 Fuel injection system performance

The tests procedure in this chapter provides an overview about the standard measurements performed by fuel injection and/or by engine manufacturers. As presented in the Chapter 3.2.2 (Fuel system), the fuel injection specification needed to be verified in order to reach the LG targets. Some typical measurements performed to test and develop injectors are:

- Opening pressure
- Opening/closing time
- Needle lift
- Pressure drop
- Sac pressure measurement
- Injection stability (cycle to cycle quantity variation)

- Leakage measurement
- Flow measurement

4.2.2 Endurance test

The scope of the LG injector endurance test was to evaluate different material composition and coating for LG injector nozzles. This test consisted of accumulating a total of 500 running hours by running three LG injectors simultaneously on a fuel injection rig. Each of the injectors was identical but was equipped with a different nozzle variant. The purpose was to evaluate the performance of the three different nozzle materials to define a suitable candidate for LG application. The endurance test was split in two series of 250 running hours, in order to have an intermediate inspection of the components after half of the test had been run.

The injector nozzles had identical whole size (0.58 mm), amount and geometry configurations. Therefore, the only variable in the scope of the endurance test was the nozzle material tested:

- Material sample 1
- Material sample 2
- Material sample 3

The mentioned parts were tested with LPG fuel, which was selected for low-viscosity fuel testing purpose. In the picture below, the test rig setup is visible and as well the main components, such as: high-pressure fuel pump (1), its electric motor (2), and on top it the injector testing setup (3).

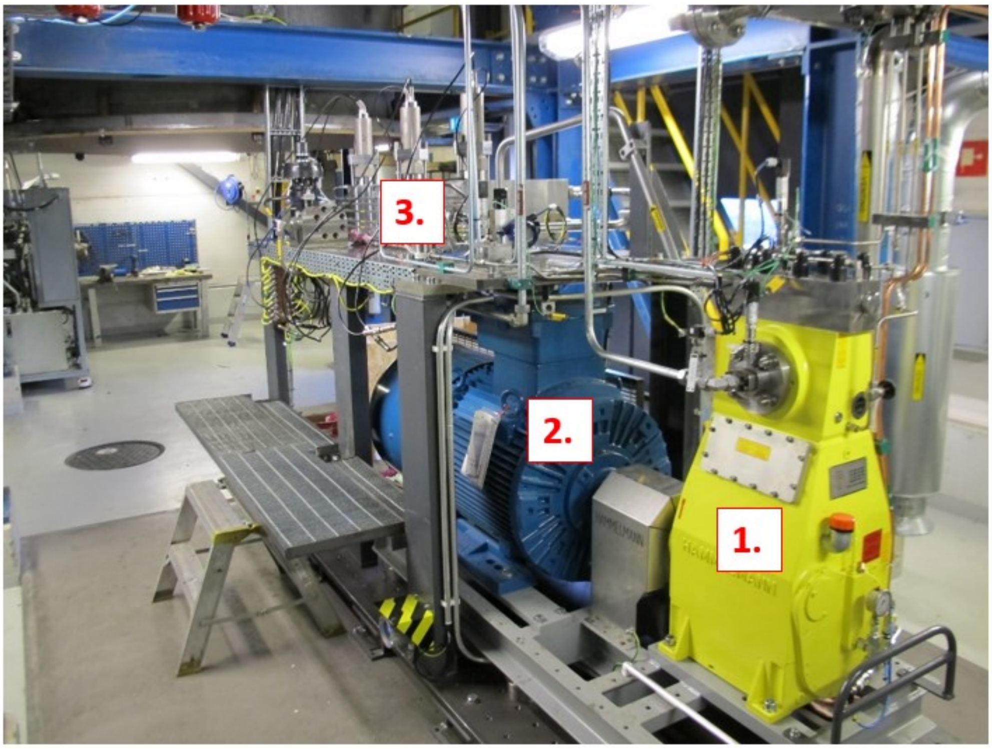


Figure 30. Test rig setup.

After 250 running hours, all three injector nozzles were disassembled and inspected. This procedure included a visual inspection, microscope inspection, needle lift measurement and examination of moulds of the nozzle seat surfaces.

The outcome of the first 250 running hours is summarised in the table below. Sample 1 and sample 2 were showing some excessive wear on the needle tip. Sample 1 provides a clear understanding that the used material is not suitable for this application. While Sample 2 was showing some wear, it was not as severe as Sample 1. Further investigation was taken to understand if the wear was caused by material failure or assembly mistake.

Table 8. Results of the intermediate inspection of injection nozzles.

Sample	Condition after 250 running hours	Decision to proceed
Number 1	Poor condition, significant amount of wear	Replacement of nozzle
Number 2	Acceptable condition, with small amount wear and material bedding in	Proceed with same nozzle
Number 3	Excellent condition	Proceed with same nozzle

Based on these results, to finalise the remaining 250 running hours, the following parts have been mounted on the rig:

- Sample 1 replaced with new one (same model)
- Sample 2 as found to evaluate if any further deterioration appears
- Sample 3 as found

After additional 250 running hours, the total targeted 500 running hours were reached. All parts were inspected in the same manner as was done previously. The outcome of the final inspection is summarised in Table 8 below.

Table 9. Results of the final inspection of injection nozzles.

Sample	Condition after 500 running hours	General conclusion
Number 1	Average	Noticeable amount of wear
Number 2	Average	Noticeable amount of wear
Number 3	Excellent condition	Chosen material candidate

Sample 3 confirms the good results of the previous inspection. This material is the candidate to be the selected one for the production version. Other two samples confirm the previous outcome.

4.3 LG engine testing

4.3.1 Engine testing setup

This chapter describes the testing setup of the W6L32LG laboratory engine. This description consists of:

- Main measurements taken on the engine
- External fuel systems

In general, this setup can be replicated for use with product LG engines. However, when translating this system for use with the V-form engine W20V32LG or with yet untested fuels, matters such as dimensioning of the system and compatibility with the fuel's characteristics need to be considered. Dimensioning of the system is related to a much larger fuel flow in the 20-cylinder, 10 MW product engine compared to the 6-cylinder, 3 MW laboratory engine. Considerations for use with different fuels includes the phase of the fuel at storing conditions, different requirements of fuel flow rate due to the fuel's heat value (lower heating value requires increased flow to maintain the same engine power output, which is achieved by increasing pressure in the fuel feed), and compatibility of the system's materials with the fuel.

The composition of LG fuels can vary greatly. Due to this, a combination of separate fuel handling systems (off the engine) will be applied. In the case of the laboratory engine, the external fuel system consists of two separate systems: one for LFO fuel, which has been tested to obtain reference values and another one for LPG fuel, which is the lowest viscosity LG fuel. This case is an example where two LG fuels cannot share the same external system, as LPG needs to be stored in a pressurised tank. Alternatively, if the fuel range for a specific LG engine is narrower, then a single system can be applied, optimised to operate with the fuel range available at the engine's site. Additionally, a module for pilot fuel will be required in all LG engine applications. A small quantity of LFO (below 2%) is always used as pilot fuel, because it is needed to ignite certain LG fuel types. Pilot fuel injection will be used even if the main fuel ignites independently to avoid blocking the pilot injection nozzles.

Measurements taken on the engine

The figure below illustrates the main measurement points on the engine, where temperature and pressure are monitored. Additionally, turbocharger (TC) speed is monitored. [25]

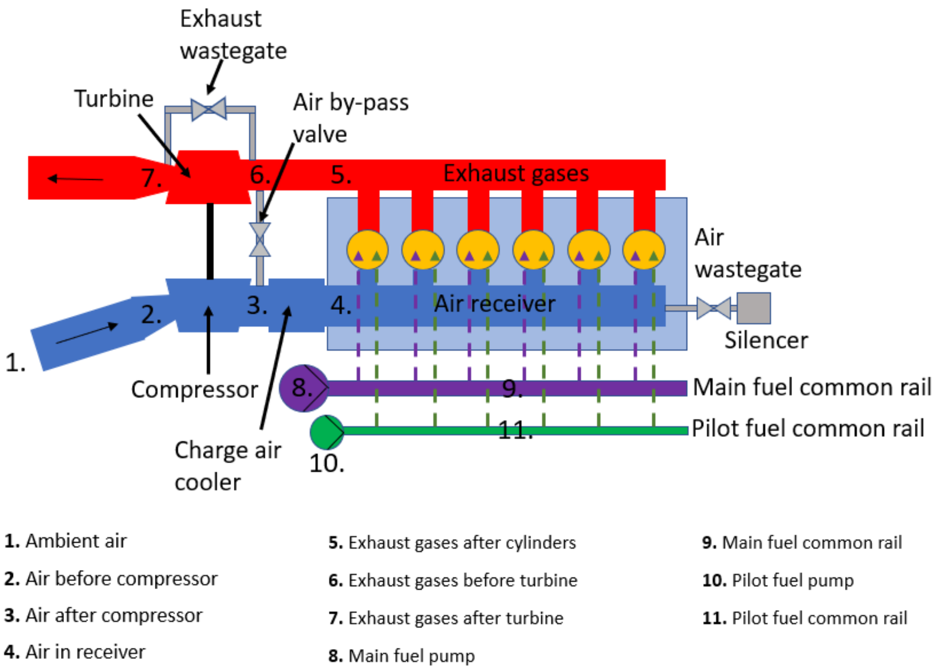


Figure 31. Measurements taken on the engine. [25]

External fuel system

The pictures below illustrate the simplified external fuel system diagram of W32LG with liquid fuel and with LG fuel.

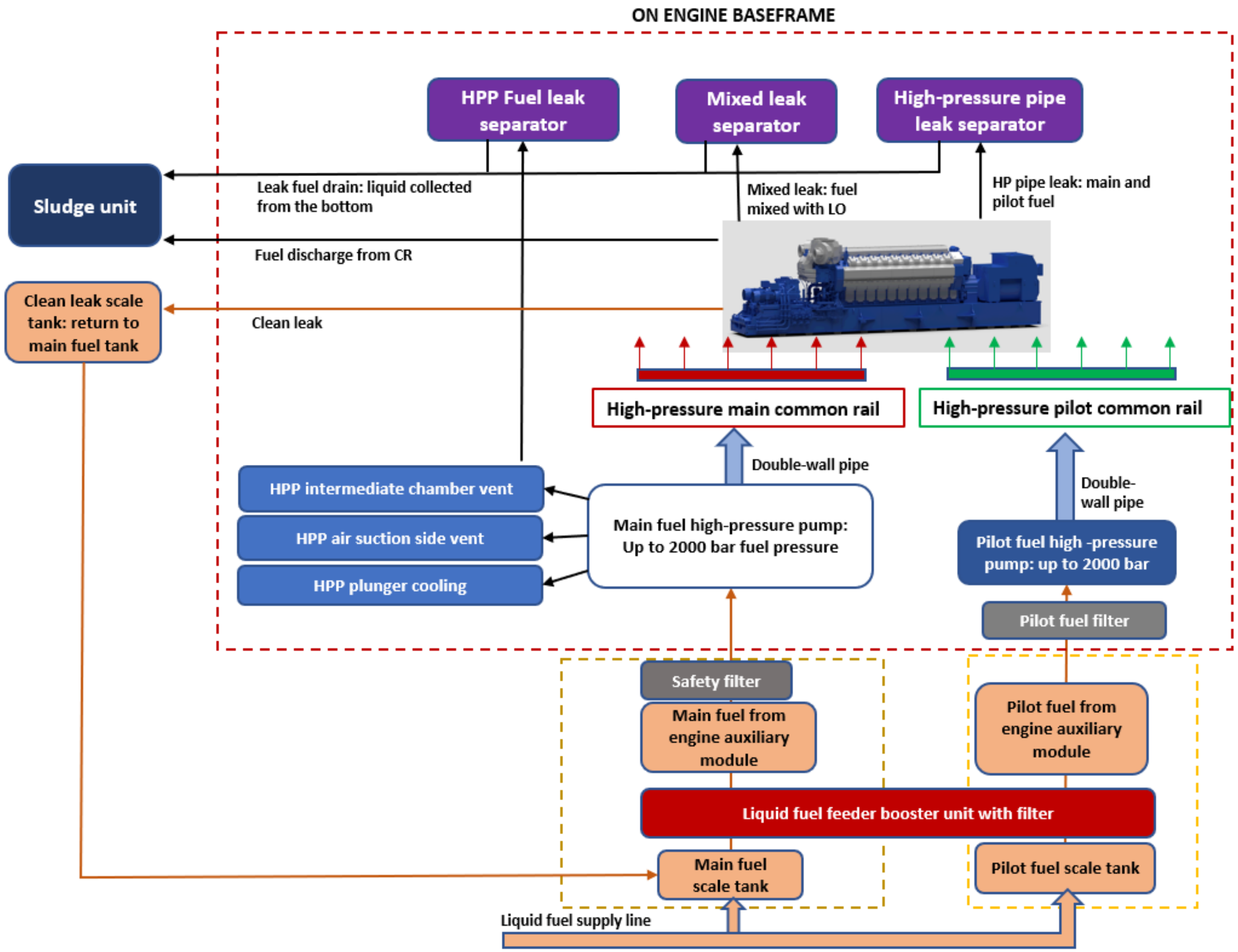


Figure 32. Simplified external fuel system diagram, liquid fuel mode.

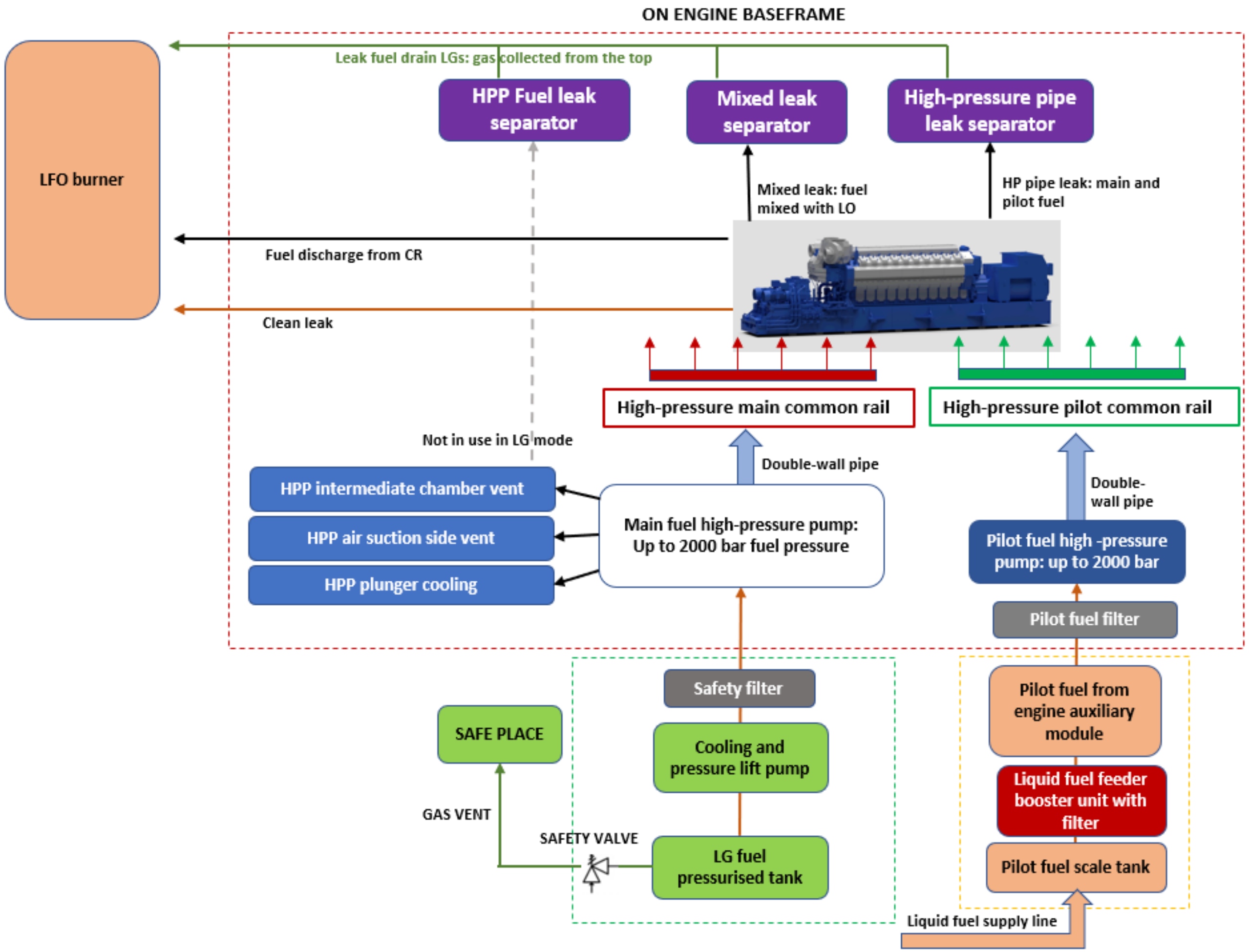


Figure 33. Simplified external fuel system diagram, LG fuel mode.

Two different fuel systems, based on the operating pressure, can be identified as high-pressure and low-pressure systems.

Low-pressure system

Low-pressure fuel system consists of all the equipment operating at low pressure (below 20 bar) and is characterised as auxiliary system, which is mounted off the engine. Such a system includes the fuel storage tanks, fuel transportation and feeding pipes to the engine, feeder and booster pumps and fuel filtering units. From the schemes above, we can identify two different fuels involved in the low-pressure system as Liquid fuels (Figure 10) and LGs (Figure 11).

LG system

The core of the LG low-pressure system is the fuel tank management. As mentioned in Chapter 2.5 (Fuels), fuel characteristics can change. For this reason, a continuous composition analysis is required to maintain the right pressure in the tank to guarantee the liquid phase. This is important, because if the liquid phase is not guaranteed, unwanted phenomena such as cavitation on feeder pump can occur. Fuel feeder pump delivers the fuel from tank to the high-pressure system with an operating pressure around 15 bar.

Liquid fuel system

LFO fuel tank is common to all engines in laboratory and a dedicated line feeds the W6L32LG. This line can be separated into two further systems: the LFO main fuel line and LFO pilot fuel line. This separation happens when the initial LFO line is split into two scale tanks, one for LFO main fuel consumption (900 kg) and another one for LFO pilot fuel consumption (23 kg). Feeder pumps are entitled to supply fuel from these scale tanks to the booster unit with a pressure around 5 bar and the booster unit is builds up the pressure to 15 bar towards the high-pressure system.

Fuel filtering

On the low-pressure side, we can identify two different types of filtering. One consists of coarser filtration. Along the LG and LFO piping, there are filters to eliminate bigger impurities coming mainly from the tank and to protect the fuel feeding lines. In the laboratory setup, there is a 6 µm automatic filter before the LFO weight tanks. In the LFO booster unit, there are main fuel filters with 32 µm and 15 µm ratings and a 10 µm filter for pilot fuel. The most important filtration is the filter located on the connection between the low-pressure system and the high-pressure pump, with a high filtration grade, because of extremely tight clearances in the high-pressure system, for example in the fuel injectors. In the laboratory setup, a 5 µm filter is located before the high-pressure fuel pump, which is the connection point between the low- and high-pressure systems.

High-pressure system

The high-pressure system consists of two main parts. The core of the LG engine is the high-pressure skid connected to the common base frame of the engine in the free end side. The other part refers to the fuel delivery system from the pump to the combustion chamber (high-pressure pipes, quill pipes and fuel injectors).

The high-pressure skid consists of the main fuel pump (Hammelmann) that is fed from the low-pressure system and builds up the pressure towards the injectors with a pressure up to 2000 bar. During the engineering process, the main challenge was to create this module to be engine mounted, in order to reduce the activities at site. In addition to the main fuel pump, the fuel filters and the fuel leakage system were also located in the same module, which is illustrated in the chapter 3.2 (LG engine). In addition to this, the vibration level of the whole system required many considerations to evaluate the feasibility of having it engine mounted. In the beginning, the design team was working closely with vibration experts to simulate the system behaviour and the expected vibration level. When this data was available, a vibration campaign was organised on Stena Germanica vessel (which has a similar Hammelmann pump running on methanol, but not engine mounted) to obtain reference values to compare with the system under development.

Simulation and field vibration data provided additional details for the final production of W20V32LG high-pressure skid. To confirm the robustness of the system, a vibration measurement campaign was carried out on the W6L32LG laboratory engine during September 2019. The outcome of the measurements was within Wärtsilä design guidelines and the system has been certified and approved. Additionally, measurements were taken in 2018 on the first W20V32LG product engine. There, a high-pressure non-conformity was identified, based on a high-pressure pipe connection to the main fuel pump. This required design modifications to be updated for production engines.

The fuel delivery system includes all the high-pressure connections and pipes which deliver the fuel from the high-pressure pump to the injector that is entitled to deliver the fuel at the right time into the combustion chamber. The main challenge related to this system was the connection between the quill pipe and the injector, due to the high-pressure involved in the system (up to 2000 bar) and the narrow location. Different designs were considered and simulated, which led to the final choice of the design. This design is shown in the picture below.

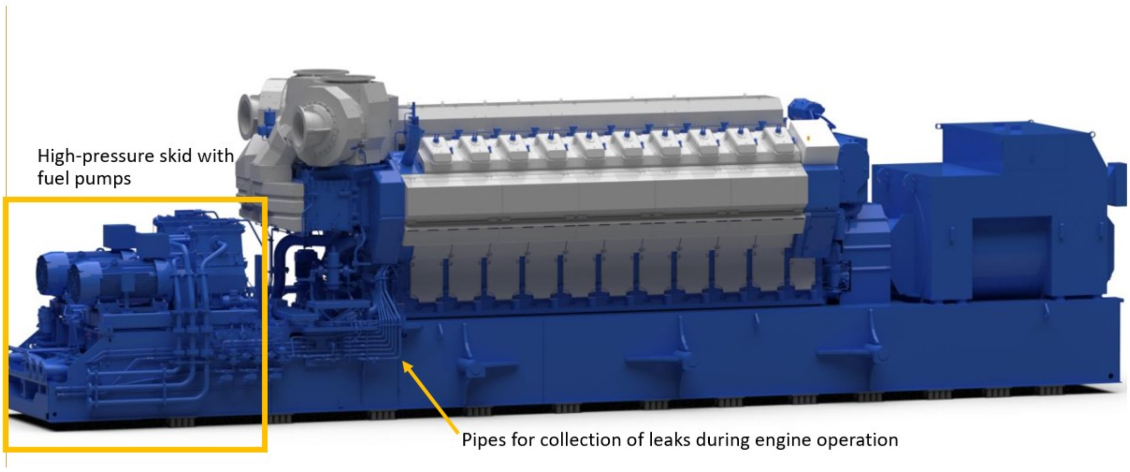


Figure 34. High-pressure fuel pump and leak line locations on W20V32LG engine. [26]

4.3.2 Engine testing activities

The following topics comprised the scope of LG engine testing activities:

- Engine performance evaluation
- Engine hardware components' validation

Engine performance evaluation

During the conceptual phase, different simulations were performed to identify the engine setup. This consisted of defining the compression ratio, valve timing, turbocharger and fuel injector specifications (needed mass flow and nozzle configuration). Engine testing consisted of evaluating the engine performance with the selected hardware. In our case, the focus was to evaluate two different injector nozzles:

- 10 *0.58 mm nozzle orifices
- 10 *0.52 mm nozzle orifices

The injector nozzle comparison was performed by maintaining all fuel injection system parameters constant to observe the differences introduced by the different setup. Before running this test, engine calibration was performed to develop an initial set of engine control settings. Engine calibration consisted of finding the right settings for the parameters which affect the start and duration of the combustion and, consequently, the emissions. The following parameters were mapped for the LG engine:

- Main fuel injection pressure
- Main fuel injection timing
- Pilot fuel injection timing

This activity consisted of running a sequence of hundreds of performance tests with different settings for the parameters mentioned above.

Engine calibration results

Engine calibration was performed with 0.58 mm injector nozzle. Pressure - timing swing was run for both main and pilot fuel injection systems, according to the test program below. Based on the results, engine performance and sensitivity to these parameters was evaluated.

Test series	Test point	Fuel	Load			BMEP	Engine speed	MFI timing	MFI pressure	Pilot inj timing	Pilot inj pressure	Pilot inj duration
			%	kW	kW/cyl	bar	rpm	°CA	bar	°CA	bar	µs
Standard settings	1	LPG	100	2880	480	24.9	720	9	1500	11	1000	1000
MFI pressure swing	2							9	1300	11	1000	1000
	3							9	1400	11	1000	1000
	4							9	1600	11	1000	1000
	5							9	1700	11	1000	1000
MFI timing swing	6							10	1500	10	1000	1000
	7							8	1500	10	1000	1000
	8							7	1500	10	1000	1000
PFI timing swing	9							9	1500	12	1000	1000
	10							9	1500	10	1000	1000

Figure 35. Testing program for engine calibration.

Main fuel injection pressure

This test consisted of evaluating the effect of the main fuel injection pressure on the overall engine performance. The investigation was done from 1300 bar to 1700 bar main fuel rail pressure with 100 bar steps. The graphs below illustrate the engine response to the change in rail pressure in terms of NO_x emissions, engine efficiency and heat release rate (5% and 90%), which was very linear in the tested pressure range.

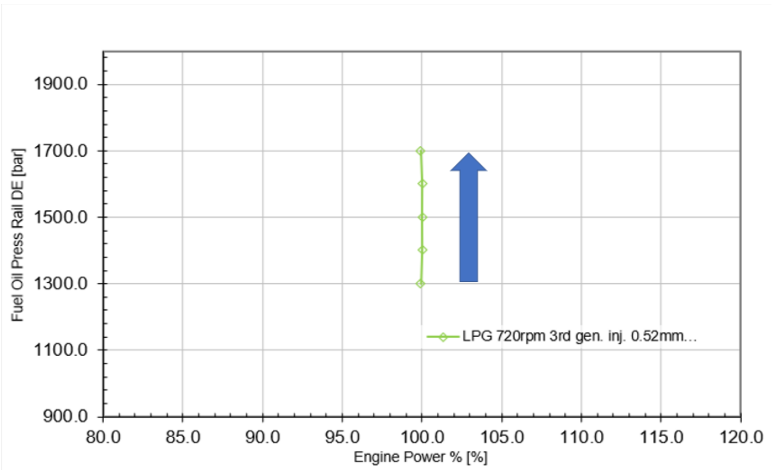


Figure 36. Main fuel rail pressure swing.

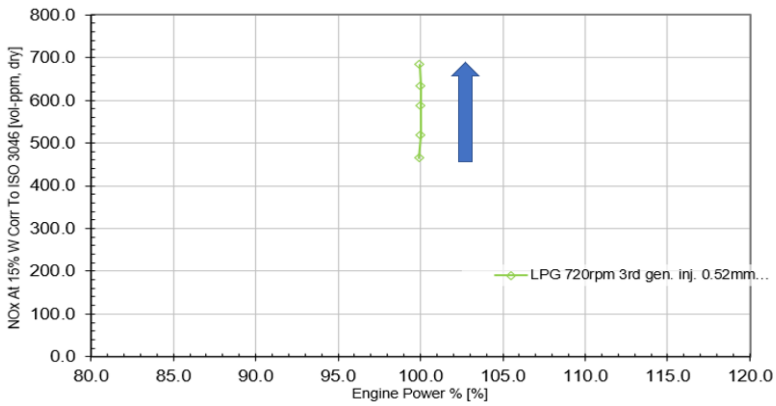


Figure 37. NO$_x$ emissions increased with higher rail pressure.

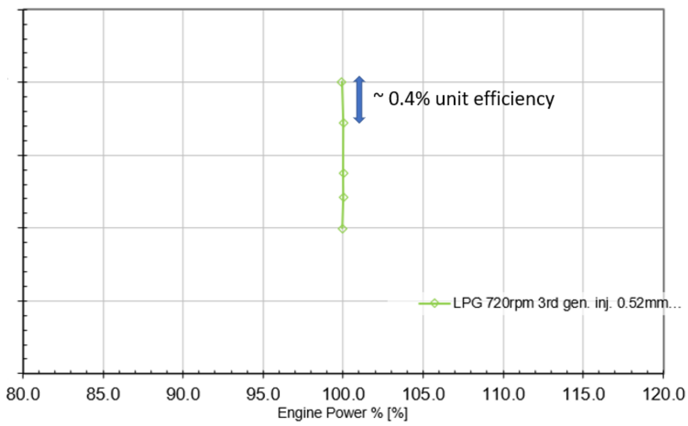


Figure 38. Engine efficiency increased with higher rail pressure.

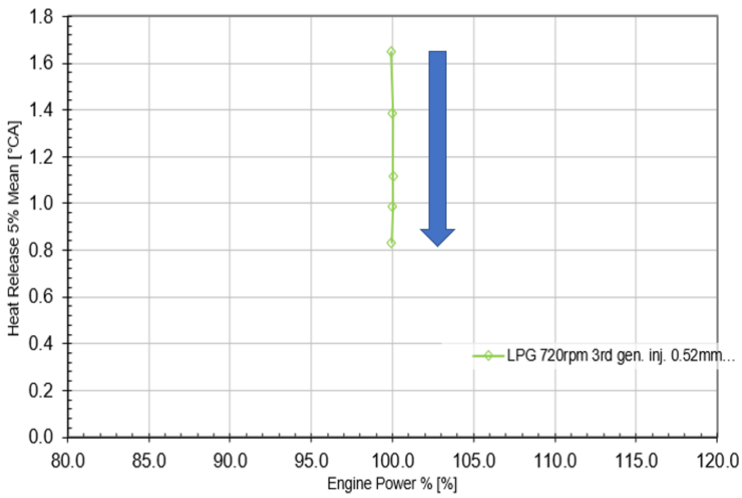


Figure 39. Heat release 5% during rail pressure swing.

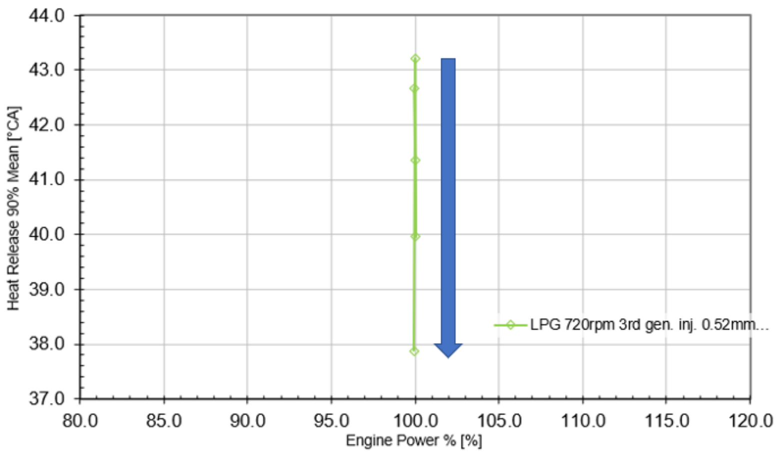


Figure 40. Heat release 90% during rail pressure swing.

As can be seen from the graphs above, an increase in 100 bar rail pressure corresponded to a consequent increase of approximately 50 ppm of NO_x emissions and 0.4%-unit increase engine efficiency. These phenomena are supported by the heat release data (5% and 90%), where it is visible that higher rail pressure enables earlier start of combustion (1 degree CA for every 100 bar of MFI) and overall shorter combustion duration (2 degree CA every 100 bar of MFI). In general, faster combustion generates higher firing pressure that must be within the engine design limit and higher mechanical and thermal stress on the hot components.

Main fuel injection timing

This test consists of exploring the performance behaviour, when changing the point of the start of injection. For this test, the investigated area for main fuel injection timing was between 7 and 10 degrees.

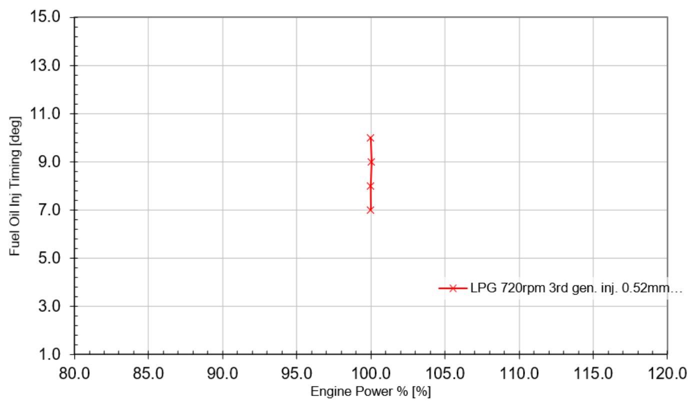


Figure 41. MFI timing swing.

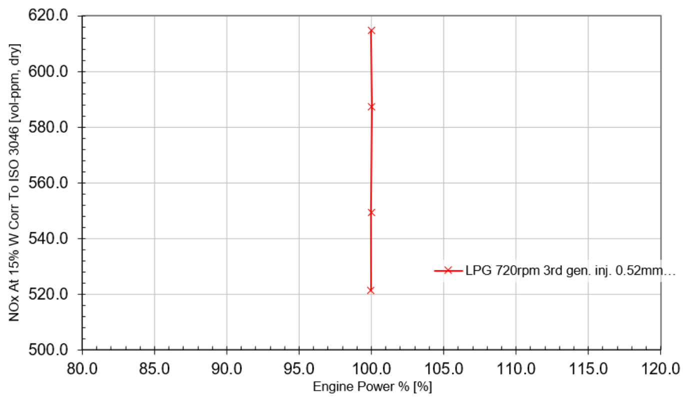


Figure 42. NO_x emissions during the MFI timing swing.

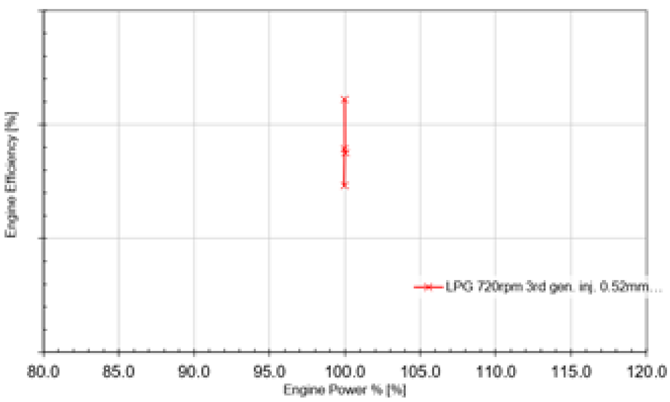


Figure 43. Engine efficiency during the MFI timing swing.

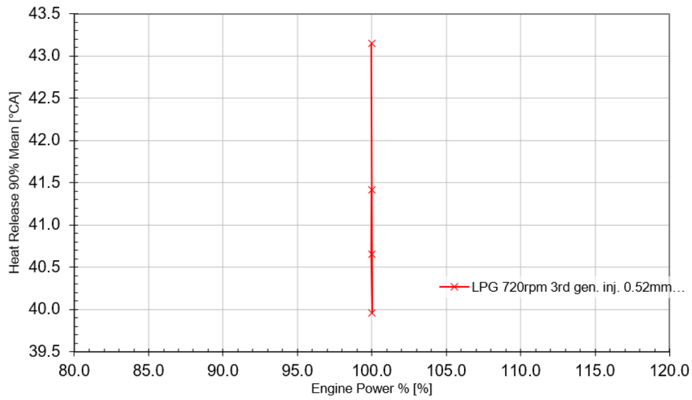


Figure 44. Heat release 90% during the MFI swing.

From the graphs above, a linear correlation between MFI timing and NO_x emissions is visible. For each crank angle advanced, there was an increase of approximately 30 ppm NO_x and an improvement in engine efficiency. This trend is not linear, efficiency gain is consistent with advanced MFI timing and approximately in the range of 0.2% efficiency unit per 1 advanced CA degree. In the graphs, the point with MFI timing 8 and 9 show similar values, which may be because the efficiency gain is in the measurement tolerance area. All other parameters supported the efficiency increase, as heat release (5% and 90%) are respectively earlier and shorter.

Pilot injection timing

Pilot injection timing defines the start of combustion. This parameter was investigated to find an operational area to reduce the risk of misfire cycles. If pilot injection timing is retarded (too close to the TDC), combustion starts when the piston is moving down from TDC, which generates low efficiency cycles. On the other hand, if pilot timing is advanced (too early compared to TDC), fuel doesn't find the right temperature to be ignited, which usually leads to a late combustion cycle.

In this test, pilot timing investigation was done between 10- and 12-degrees CA before TDC. In this timing window, the temperature of the air-fuel mixture in the combustion chamber is at a level where pilot fuel ignites easily (over 620 °C). Pilot timing had a linear effect on emissions (NOx) and heat release (5% and 90%). Efficiency figures showed a minor improvement. The three measured points were in a small range (almost within the measurement tolerance) for the firing pressure. The overall minor changes in the engine performance define that this operating area is safe (from start of combustion reliability point of view). The selected timing was 11 degrees CA.

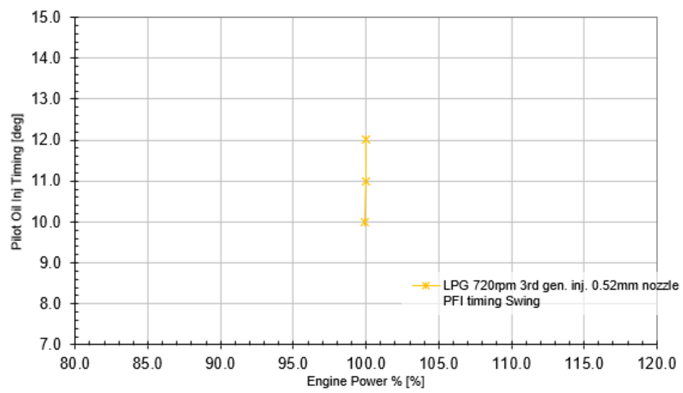


Figure 45. Pilot injection (PFI) timing swing.

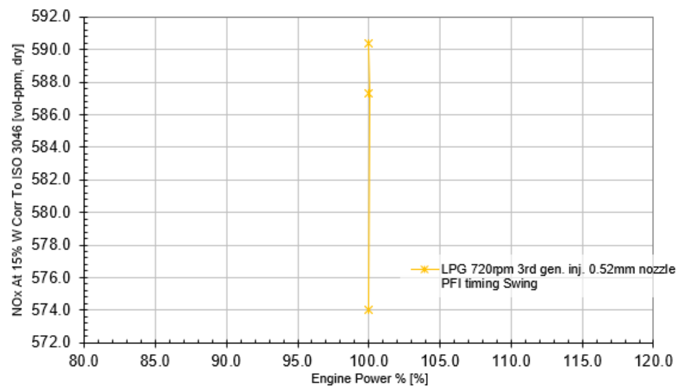


Figure 46. NO_x emissions during PFI timing swing.

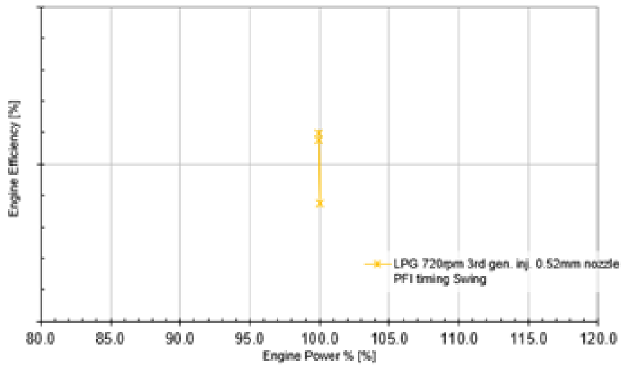


Figure 47. Engine efficiency during PFI swing.

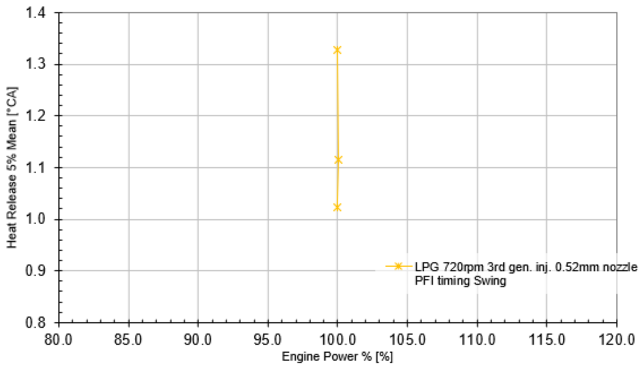


Figure 48. Heat release 5% during PFI swing.

Based on the engine calibration results, the following parameters were selected:

- MFI pressure: 1500 bar
- MFI timing: 9 deg bTDC
- PFI timing: 11 deg bTDC
- PFI pressure: 1000 bar
- PFI duration: 1000 µs

Table 10 summarises the rules of thumb for engine tuning, which were obtained by observing the engine response during the experiments.

Table 10. Rules of thumb for engine tuning.

		Engine response		
Changed parameter	Step	NO_x emissions	Engine efficiency	Combustion duration
MFI pressure	+ 100 bar	+ 50 ppm	+ 0.4% unit	- 2 degrees CA
MFI timing	- 1 degree CA	+ 30 ppm	+ 0.2% unit	- 1.5 degrees CA
PFI timing	- 1 degree CA	+ 10 ppm	+ 0.1% unit	< - 0.5 degree CA

Comparison of 0.58 mm and 0.52 mm nozzle variants

To select the optimal LG injector hardware, different nozzle configurations were tested on the laboratory engine. This chapter presents the performance comparison of two tested injector nozzles, which differed in their nozzle hole size (0.58 mm and 0.52 mm). In order to have a clear understanding about the performance differences, the parameters which affect the combustion process were kept constant and the same fuel was used (LPG). The explored engine load ranged from 10% to 100% load. The test results are reported in the graphs below, focusing on:

- Main fuel injection duration
- Fuel consumption
- NO_x and smoke (FSN) emissions
- Firing pressure
- Rate of heat release

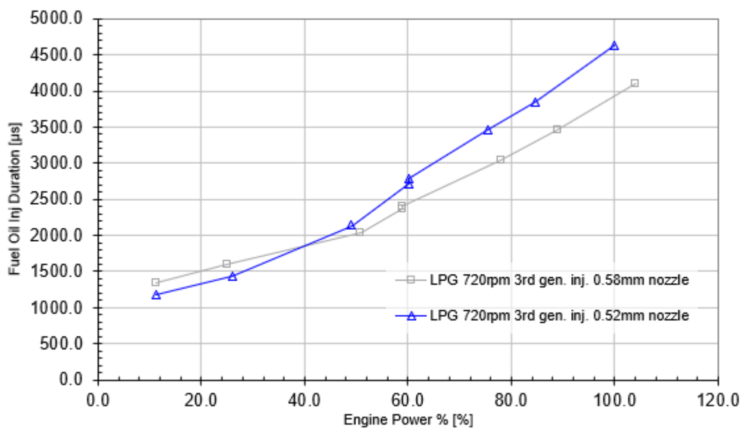


Figure 49. Fuel injection duration comparison for 0.52mm and 0.58mm nozzles with LPG.

Main fuel injection duration results showed that the smaller nozzle is “throttling” the fuel flow. This means that smaller nozzle holes need longer time to inject the required fuel amount. Consequently, fuel is delivered later relative to the piston position, when pressure and temperature inside the combustion chamber are lower, as the piston has already moved further down from TDC. In this environment, combustion is shifting more towards diffusion/rate-controlled combustion. The combustion work is less efficient the further the piston moves away from TDC. During diffusion combustion, the fuel and air have mixed more completely and there are fewer lean pockets of air (including nitrogen). At this point, the combustion temperature is lower, resulting in less NO_x emissions, but usually more soot/PM due to a lower lambda in comparison to premixed combustion. This phenomenon is visible in the combustion duration graph below.

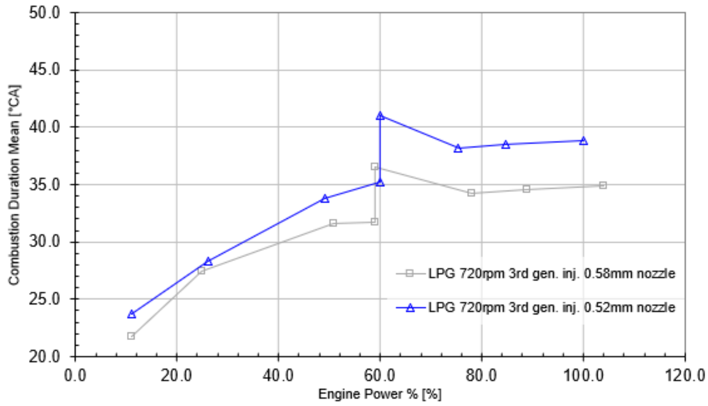


Figure 50. Combustion duration comparison for 0.52mm and 0.58mm nozzles with LPG.

As presented, longer main fuel injection duration leads to less efficient combustion cycles, identified by longer combustion duration. This phenomenon is more visible at higher load (above 50%), where the smaller nozzle holes have a more significant impact compared to the low load, where the amount of fuel is much smaller. The following heat release graphs (5% and 90%) demonstrate the difference in start of combustion as well as its length.

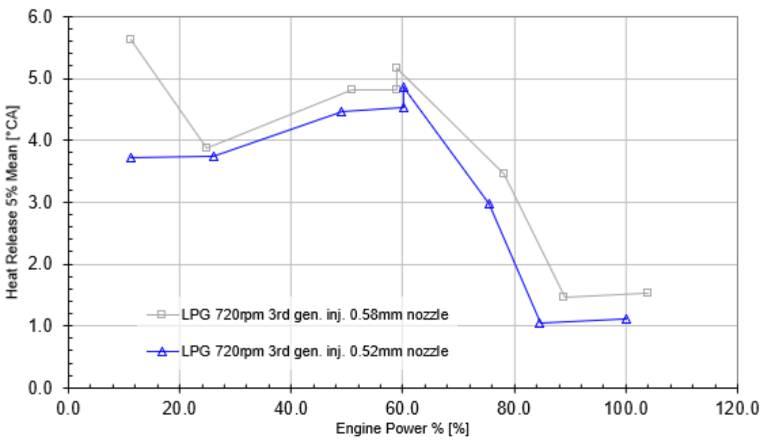


Figure 51. Heat release 5% with 0.58 mm and 0.52 mm nozzles.

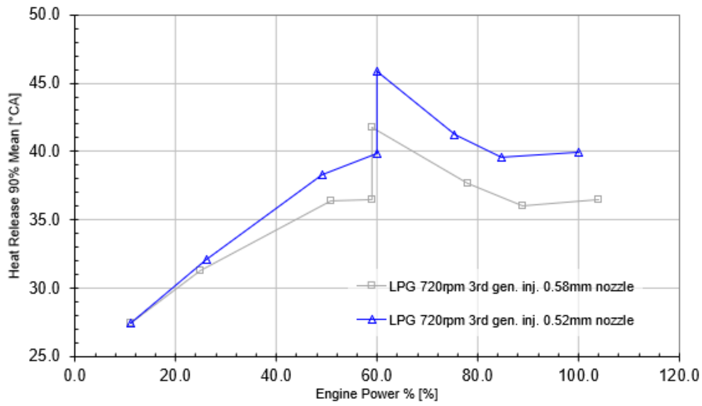


Figure 52. Heat release 90% with 0.58 mm and 0.52 mm nozzles.

Considering these results, the 0.58 mm injector nozzle provides more efficient combustion, as visible in the engine thermal efficiency graph below.

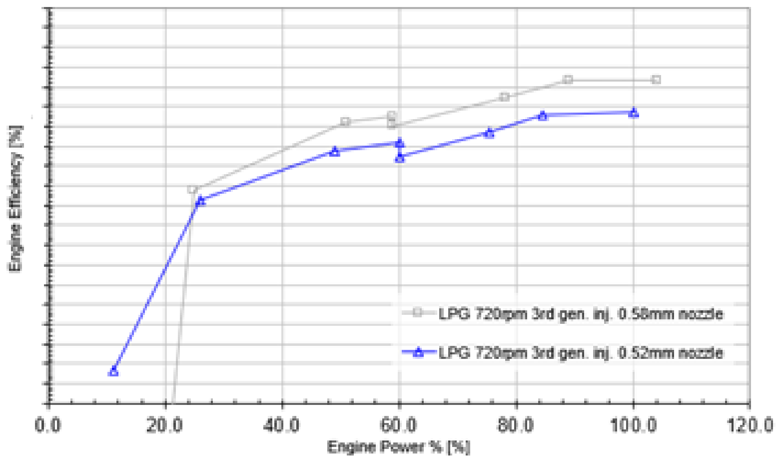


Figure 53. Engine efficiency with 0.58 mm and 0.52 mm nozzles.

From the graph above, as already seen in the combustion duration, the 0.58 mm injector nozzle provides improvements. Engine thermal efficiency is approximately 2% unit higher in the power range 50% to 100% load. As expected, the 0.58 mm injector nozzle guarantees a pre-mixed combustion that leads to higher NO_x formation, as visible in the graph below.

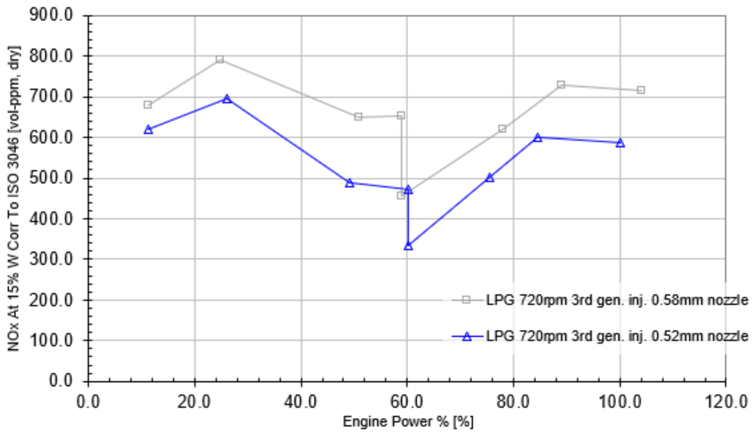


Figure 54. NO_x emissions with 0.58 mm and 0.52 mm nozzles.

Despite higher NO_x emissions, the level obtained is not a significant issue for the scope of this project, because it remains within the World bank limits (710 ppm) at full engine load. In case further reductions in NO_x emissions are required, SCR can be used.

Other fuels investigated

While LPG was the main fuel for engine development in this project, other fuels were also tested to investigate the LG fuel range. These additional tested fuels were LFO and Liquefied Volatile Organic Compounds (LVOC). LVOC fuel was tested to evaluate the combustion process with the worst quality fuel type in the LG range, due to its high composition of heavy hydrocarbons. On the other side, LFO was tested to evaluate the fuel injection system and overall engine performance, when using a fuel on the upper viscosity limit.

This test was performed with a prototype fuel injector that was used during the initial phase of the project. The purpose of this test was to evaluate the possibility of delivering the full power output without any hardware change (mainly fuel injector specification) throughout the whole LG range. The analysed parameters were related to injection duration, fuel ignitability and power output. While these results were obtained with a prototype injector version, they helped to define the trade-off in terms of engine performance between LVOC, LPG and LFO. Based on this data, LPG was used as main fuel for the whole test session, without switching between LVOC and LFO, as the performance difference could be scaled, based on the previous test.

LVOC results and LPG comparison

This test consisted of evaluating the engine capability of using LVOC as main fuel. LVOC was the chosen fuel to evaluate engine performance in worst case scenario from composition point of view, because of its low-quality fuel properties. This fuel has a low MN, due to a high content of heavy hydrocarbons. Table 11 presents the composition analysis of tested LVOC fuel and its calculated MN, which was done by using Wärtsilä algorithm. According to this method, MN was 17.

Table 11. Composition analysis and MN of tested LVOC and MN based on Wärtsilä algorithm.

Fuel type	**Chemical formula**	**Mole % in analysed sample**
Methane	CH_4	5.67
Ethane	C_2H_6	5.67
Propane	C_3H_8	10.55
Iso-pentane (2-Methylbutane)	i-C_5H_{12}	18.58
N-pentane	n-C_5H_{12}	22.31
Propylene	C_3H_6	10.24
Neo-pentane (2,2-Dimethylpropane)	Neo-C_5H_{12}	0.06
Mix-hexane	mix-C_6H_{14}	0.77
Nitrogen	N_2	0.01
		Total mole % = 73.85 (mol% normalised to 100%)
Output	**Wärtsilä Knock Index (WKI)**	28.0
	Propane Knock Index (PKI)	Above 100
	Wärtsilä Methane Number (WMN)	17

For this test, a limited test series was run, because such fuel is not available on the market as a product. This means that specific authorisations are needed to collect, transport and store it. From utilisation point of view, specific permits are needed. This is due to the fuel composition, which does not fulfil the land-based fuel requisition and therefore requires a dedicated authorisation for laboratory engine testing purpose.

One test series was carried out to explore the whole load range. The main purpose was to evaluate whether the full engine output can be delivered. The figure below represents the firing pressure curve and comparison of main combustion parameters at full load for LVOC and LPG.

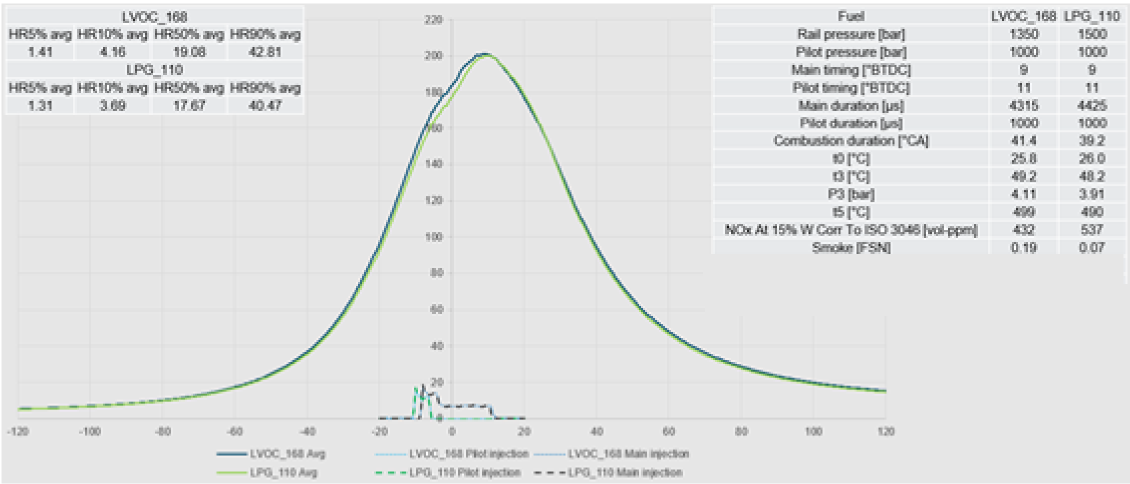


Figure 55. Comparison of LVOC and LPG combustion characteristics.

As visible in Figure 56, different MFI pressure was required to reach full load. By maintaining all the combustion parameters constant (MFI/PFI pressure and timing), the engine was not able to deliver the full output, due to some unexpected combustion instability issues.

At higher engine loads, engine knock was present, similar to Otto or DF engine. These cycles were characterised by higher firing pressure (above engine design limit) and faster and uncontrolled combustion. These events are under investigation, as they are not expected in a diesel combustion process. This phenomenon will be investigated further to understand which elements are creating it and as well the effect of the different fuels.

Engine hardware components' evaluation

This activity typically consists of validating the engine components from different perspectives, such as: vibration, thermal load, lifetime, stress. In the LG project, no major changes were applied in the engine design apart from the fuel injection system. Therefore, the validation process was limited to the thermal load evaluation of the components that are exposed to the combustion chamber (inlet and exhaust valves, cylinder liner, cylinder head and piston). For each component, the temperature measurement points were defined. Temperatures at these points must not exceed the maximum allowed limits. These limits are set in order to guarantee the expected component lifetimes and they are measured at full load, which represents the point with the highest thermal stress. The figures below illustrate the temperature measurement points on the components.

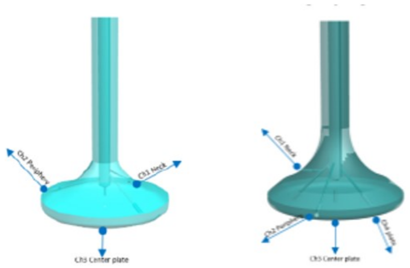


Figure 56. Temperature measurement points on the inlet (left) and exhaust (right) valves.

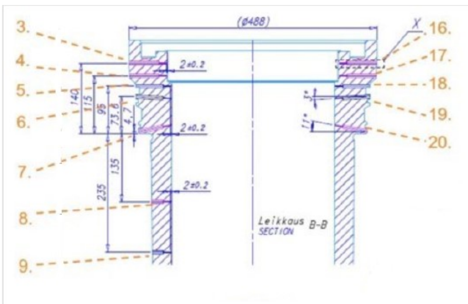


Figure 57. Temperature measurement points on the cylinder liner.

5 Conclusions

The objective of this project was to identify an alternative technology that enables the use of fuels with low-viscosity, poor ignitability characteristics and poor quality (low MN). To reach this objective, three different experimental stages were carried out: low-viscosity fuel testing in a CRU, rig testing to validate fuel injection components and laboratory engine testing.

During the CRU testing stage, two different conditions were tested. Engine idling (combustion chamber pressure 55 bar and combustion chamber temperature 550 °C) and low load operation (combustion chamber pressure 70 bar and combustion chamber temperature 590 °C) conditions were replicated, as this is the most critical environment for the ignitability of poor-quality fuels. Results obtained from CRU fuel tests provided a wide overview about ignitability and combustion response of different fuels in the low viscosity range, as summarised in Table 11. Additionally, CRU results provide the amount of pilot fuel needed to have a stable and repeatable combustion. These values are useful for the pilot nozzle injector dimensioning and pilot fuel injection mass flow requirement.

Considering the characteristics of these fuels, a new injector nozzle was developed for low-viscosity application. Rig testing activities were an important milestone in the project to validate mechanically the new design of the fuel injector nozzle for this application. Based on the inspection results, after a 500 hours test (and partially from the intermediate inspection after 250 running hours), one of the three tested materials was selected as candidate material for the final production. The outcome of the rig testing was shared with the injector supplier, in order to define the characteristic for the high volume injector production. Based on this selection, two sets of injectors with respectively 0.52 and 0.58 mm holes were ordered for engine testing.

Table 12. Main conclusions of fuel tests in CRU.

Fuel type	Pilot fuel requirement	Combustion stability
Reference fuel: LFO	Not needed	Very good
N-pentane	Not needed	Very good
N-hexane	Not needed	Very good
N-heptane	Not needed	Very good
Methanol	Needed	Good
Ethanol	Needed	Good
Butanol	Needed	Good
Propanol	Needed	Good
Cyclohexane	Needed (long ignition delay)	Unstable
Xylene	Needed	Unstable (some improvement with longer pilot injection)
Toluene	Needed	Unstable (some improvement with longer pilot injection)
Kerosene	Long ignition delay, can be compensated with engine calibration	Good
Isopentane	Long ignition delay	Pilot injection results in repeating pressure peaks
Naphtha	Long ignition delay	Pilot injection results in repeating pressure peaks

The last experimental stage consisted of laboratory engine testing. This validated the previous stages and simulations which had been done for the injection system and for engine performance. During the testing phase, three different fuels were tested: LPG (used as a representative fuel for the lowest viscosity end of the fuel range), LFO (representative for the highest viscosity fuel) and LVOC (most challenging fuel from its own chemical composition due to high heavy hydrocarbon content, low MN and very wide range of composition, depending from source). From hardware perspective, two injector nozzles were tested, both providing good results. Firstly, nozzle with 0.58 mm hole diameters was tested, resulting in excellent engine thermal efficiency with LPG fuel within World Bank emission limit. However, this setup was not able to fulfil the emission limits when operating on LFO. Thus, with this setup the engine would require different combustion parameters, based on the used fuel. However, from the project requirements, the focus was to identify a solution that provides the operation of the engine within LG fuel range, without any change in the settings (hardware and software). For this reason, the second available injector nozzle (0.52 mm) was tested. Results obtained with this nozzle met the project requirements. While engine thermal efficiency dropped significantly (approximately 2% unit at full load compared to the 0.58 mm configuration), the emission requirements were met within the tested LG fuel range while using constant settings. Figures 59-60 summarise the engine thermal efficiency and NO_x emissions. In the efficiency graph, W34SG-LPG data is shown to highlight the benefit in terms of power output (and efficiency) by using diesel process with LG technology.

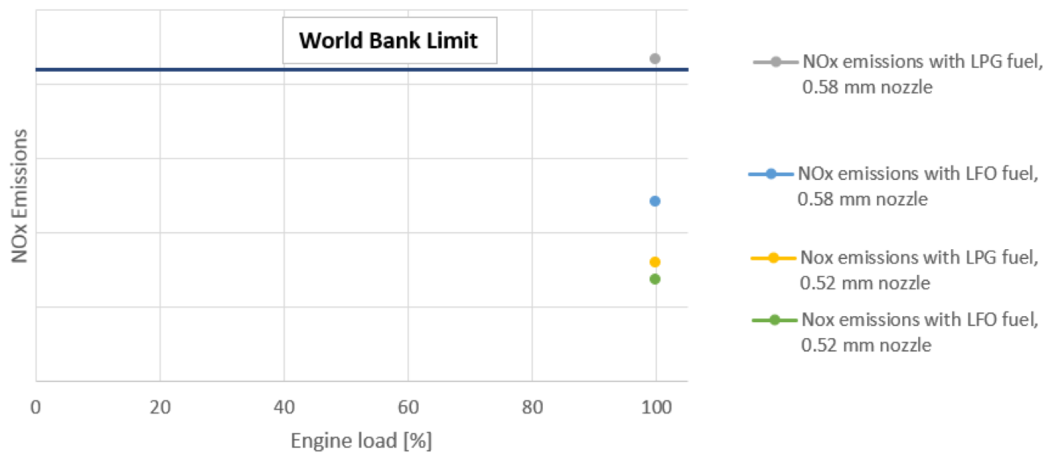


Figure 58. NO_x emission comparison with 0.58/0.52 mm nozzles and LFO/LPG fuel.

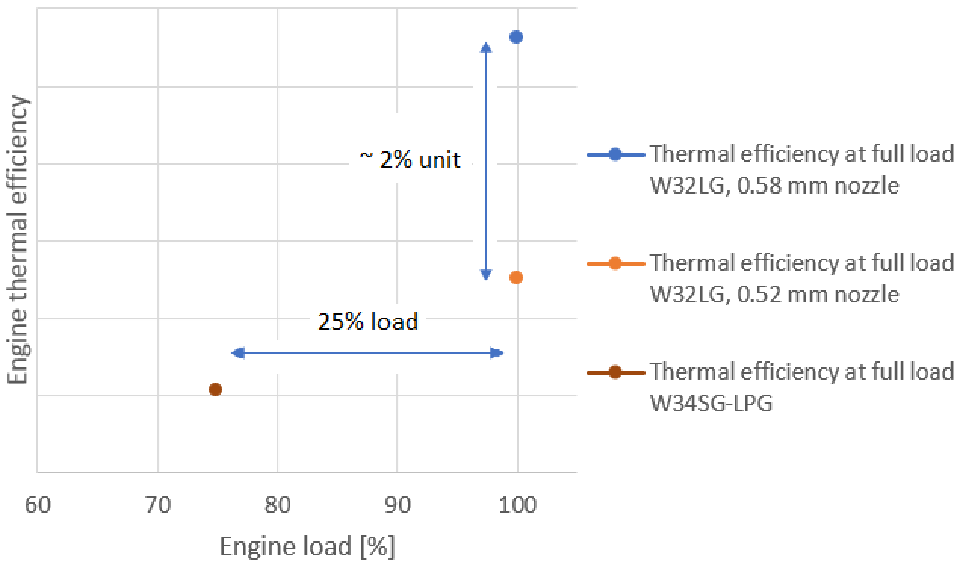


Figure 59. W32LG engine thermal efficiency comparison with 0.58/0.52 mm nozzles and with W34SG-LPG engine.

Figure 59 visually explains the 0.52 mm nozzle selection with an engine thermal efficiency that is in line with the previous Wärtsilä solution, based on W34SG-LPG. The main gain of W32LG is the 25% higher power output, without any restriction on LG fuel chemical composition.

The outcome of these tests was an engine able to meet the initial project targets, which consisted of defining a concept that can run freely with all LG fuels, deliver the full power output of the engine and expand the fuel range portfolio for Wärtsilä engines, without any changes in hardware (injector nozzle) or software settings (main fuel pressure and pilot settings). Additionally, following the laboratory engine testing stage, a customer W20V32LG engine was tested at Wärtsilä factory during May 2020. The result of this work enabled launching the W32LG technology into the market as part of a commercial project in which the carbon footprint of the power plant and its investment and operational costs are reduced in comparison to previously available technology.

6 Summary

The LG project was initiated during 2018 in order to evaluate the potential of using low-viscosity fuels in internal combustion engines. This fuel range consists of hydrocarbons with carbon number C3 to C20 (LPG to LFO), where a predominant part of these hydrocarbons is commonly available as side- or waste streams of other processes. Due to the characteristics and composition of this fuel, the focus was to develop a technology based on the diesel process to enable their utilisation. Because of the wide variety of heavy hydrocarbons in the low-viscosity fuel range, the Otto cycle was not taken into the development process, since the power reduction would have been too significant due to the fuel quality (extremely low MN, below 35). Starting from this requirement, a high-pressure injection system was developed for fuels with low viscosity (below LFO value of 2.9 mm^2/s at 40 °C). The engine selected for this project is the Wärtsilä 32 engine and the main development consisted of upgrading the existing jerk pump fuel injection system with an electronically controlled common-rail system, based on twin-needle injectors. This system required a specific development with reduced tolerances and different nozzle material due to the low-viscosity fuel. The input for the injection system design came from the fuel laboratory investigation and rig testing.

Based on CRU tests, the fuel laboratory investigation provided information about the fuels which can be used, by analysing their ignitability in a diesel process with or without pilot fuel (LFO). This test gave an initial estimation of the amount of pilot fuel needed for needle dimensioning. On the other side, the rig testing supported the feasibility evaluation and the mechanical validation of the fuel injection system, when operating with low-viscosity fuels. When these activities were finalised, the most important part of the project started, which was the engine testing with full-scale system. Engine testing consisted of verifying that a multi-cylinder engine (6L for laboratory and 20V for production engines) can operate safely and reliably, according to the project target. This was an important milestone, as it is a result of different testing at earlier stages, such as simulations, rig testing and CRU testing. During the engine testing stage, two different nozzles and three different fuels were tested to map the engine behaviour from performance perspective within the minimum and maximum viscosity limits. The resulting analysis provided the best functionality parameters to control the combustion process and, consequently, the emission level. At the same time, from the same testing session, other parameters such as heat balance, particulates, noise and vibration measurements, were collected to define the engine manual. This was an important milestone for W32LG technology development, because all the parameters needed for power plant dimensioning and for sales purpose were defined. The outcome of the project is aligned with the project targets. Wärtsilä 32LG is able to operate with low-viscosity fuels, by guaranteeing the full power output without any restriction on fuel composition in the LG range.

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Appendices

Appendix 1. LG fuel example.

Combined Components mass %	Normal	Set 1	Set 2	Normal	Set 1	Set 2	Normal	Set 1	Set 2	Normal	Set 1	Set 2
Aromatics	53,39	24,62	53,39	57,12	31,11	57,12	30,64	25,06	34,66	3,06	2,34	3,77
Olefins	0,20	0,20	0,20	0,02	0,02	0,02	16,48	18,36	16,23	76,81	77,92	75,70
Cyclic paraffins, Naphthenes	16,51	45,40	16,51	17,11	45,27	17,11	6,74	16,06	8,08	20,14	19,74	20,53
Paraffins	29,89	29,78	29,89	25,75	23,60	25,75	46,15	40,53	41,04	0,00	0,00	0,00
Oxygenates	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Components mass %												
C6 monocyclic aromatics	24,48	14,64	24,48	55,57	30,59	55,57	30,64	25,06	34,66	3,06	2,34	3,77
C7 monocyclic aromatics	17,04	6,24	17,04	1,55	0,52	1,55	0,00	0,00	0,00	0,00	0,00	0,00
C8 monocyclic aromatics	6,96	2,28	6,96	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
C9 monocyclic aromatics	3,91	1,15	3,91	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
C10 monocyclic aromatics	1,00	0,31	1,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Aromatics Sum	**53,39**	**24,62**	**53,39**	**57,12**	**31,11**	**57,12**	**30,64**	**25,06**	**34,66**	**3,06**	**2,34**	**3,77**
C5 monocyclic paraffin	0,87	0,87	0,87	1,98	1,82	1,98	3,72	13,29	5,34	3,95	3,59	4,30
C6 monocyclic paraffin	4,29	14,15	4,29	9,74	29,56	9,74	3,01	2,77	2,74	13,01	13,38	12,63
C7 monocyclic paraffin	6,07	17,01	6,07	5,36	13,82	5,36	0,01	0,00	0,01	3,19	2,77	3,60
C8 monocyclic paraffin	3,15	7,86	3,15	0,03	0,06	0,03	0,00	0,00	0,00	0,00	0,00	0,00
C9 monocyclic paraffin	1,94	4,71	1,94	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
C10 monocyclic paraffin	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
C11 monocyclic paraffin	0,16	0,72	0,16	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
C12 monocyclic paraffin	0,03	0,08	0,03	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Naphthenes Sum	**16,51**	**45,40**	**16,51**	**17,11**	**45,27**	**17,11**	**6,74**	**16,06**	**8,08**	**20,14**	**19,74**	**20,53**
C5 linear and iso olefin	0,00	0,00	0,00	0,00	0,00	0,00	1,62	1,75	1,63	37,60	39,49	35,70
C6 linear and iso olefin	0,01	0,01	0,01	0,02	0,02	0,02	14,68	16,37	14,49	28,82	30,33	27,30
C7 linear and iso olefin	0,00	0,00	0,00	0,00	0,00	0,00	0,17	0,24	0,11	10,40	8,10	12,70
C8 linear and iso olefin	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
C9 linear and iso olefin	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
C10 linear and iso olefin	0,19	0,19	0,19	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Olefines Sum	**0,20**	**0,20**	**0,20**	**0,02**	**0,02**	**0,02**	**16,48**	**18,36**	**16,23**	**76,81**	**77,92**	**75,70**
C4 linear and iso paraffin	0,85	0,85	0,85	1,94	1,78	1,94	0,00	0,00	0,00	0,00	0,00	0,00
C5 linear and iso paraffin	2,13	2,12	2,13	4,86	4,45	4,86	0,00	0,00	0,00	0,00	0,00	0,00
C6 linear and iso paraffin	5,36	5,34	5,36	12,23	11,21	12,23	46,00	40,50	40,92	0,00	0,00	0,00
C7 linear and iso paraffin	5,03	5,01	5,03	6,70	6,14	6,70	0,14	0,03	0,12	0,00	0,00	0,00
C8 linear and iso paraffin	3,09	3,08	3,09	0,03	0,02	0,03	0,00	0,00	0,00	0,00	0,00	0,00
C9 linear and iso paraffin	4,33	4,31	4,33	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
C10 linear and iso paraffin	3,51	3,50	3,51	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
C11 linear and iso paraffin	2,42	2,41	2,42	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
C12 linear and iso paraffin	1,49	1,48	1,49	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
C13 linear and iso paraffin	1,67	1,67	1,67	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Paraffins Sum	**29,89**	**29,78**	**29,89**	**25,75**	**23,60**	**25,75**	**46,15**	**40,53**	**41,04**	**0,00**	**0,00**	**0,00**

Appendix 2. Example of LPG fuel composition analysis.

Sample type	LPG

Hydrocarbon	**mol %**
Methane	0.02
Ethane + Ethylene	0.17
Propane + Propylene	96.60
Isobutane	2.83
1-butene + 1, 3-Butadiene	<0.01
n-Butane	0.37
Neopentane	<0.01
Isopentane	<0.01
n-Pentane	<0.01
n-Hexane	<0.01
n-Heptane	<0.01
n-Octane	<0.01

Appendix 3. Properties of LG fuels.

Fuel	Chemical formula /composition	LHV	LHV % of LFO	Melting point	Flash point	Boiling point	Auto-ignition temperature AIT	Flammability limit	Density at 25 °C	Dynamic viscosity at 25 °C	Kinematic viscosity at 25 °C	Cetane number
		MJ/kg		°C	°C	°C	°C	%	kg/m3	mPas*s	mm2/s	
LFO (reference fuel)	85.5%C, 11.5%H	42.95	100		56 - 82				833-960 (at 15 °C)		2.9 (at 40°C)	55
Propane	C3H8	46.4	108.03	-188	-104	-42	510	1.8-9.5	510-580			<3
n-butane	C4H10	45.3	105.47	-138	-60	-0.5	490	1.5-8.5	604			
Naphtha	C4 - C9				10	20-220	>240		712.5 (@15°C)		<1 (at 40C)	
MGO	C10 - C28	42.8	99.65		128	270	>220	0.6-6.5	835-845 (at 15C)		2-6 (at 40C)	67.7
Gasoline		43.4	101.05		-42.8	30-210	257	1.3-7.6	765 (@liquid)			
Tested in CRU (Chapter 4.1)												
n-pentane	C5H12	45.357	105.60	-129.8	-40	36.1	309	1.5-7.8	626	0.224	0.358	25.9
n-hexane	C6H14	44.572	103.78	-94	-26	68.5	234	1.2-7.4	660.6	0.254		46.4
n-heptane	C7H6	44.566	103.76	-90.549	-4	98.38	223	1.1-6.7	679.5	0.386		61
Methanol	CH3OH	19.93	46.40	-97.6	11	64.7	385	6.7-36.0	792	0.545	0.64 @ 20 C	1.6
Ethanol	C2H6O	26.7	62.17	-114.14	14	78.24	365	3.3-19.0	783.924	1.074	1.52 @ 20 C	-5.1
Propanol	C3H8O	30.68	71.43	-126	22	97	371	2.2-...	803	1.959		7.2
Butanol	C4H9OH	34.4	80.09	-89.8	28.89	117	345	1.7-12.0	806.315	2.544	3.64 @ 20 C	
Cyclohexane	C6H12	43.45	101.16	6.47	-20	80.74	245	1.3-7.8	778.1	0.896		
Xylene	C8H10	40.961	95.37	(-47.8) to 13.2	27-32	137-144.5	463-528	1.0-7.0	861-880 (at 20C)			
Toluene	C7H8	40.589	94.50	-95	6	111		1.2-7.1	866.67	0.59 @ 20 C		
Kerosene	C6-C16	43	100.12		min. 38	170-300		0.7-5.0	787.3 (at 15C)		<7 (at 40C)	70-100 (a)
Trimethylpentane	C8H18	44.31	103.17	-107.38	-12	99.3	396		692	0.5 @ 20 C	0.72 @ 20 C	5
Isopentane	C5H12	45.241	105.33	-159	-51	27.8	420	1.4-7.8	616	0.214 @ 20 C		10.4

Appendix 4. Previous research about LPG application in diesel engines.

Reference	Idea and purpose of the research	Conclusions/ Findings
[9]	Conversion of traditional diesel engines into dual fuel engines. In this case, dual fuel refers to mixtures of diesel and LPG (or CNG). The engine that was developed has a new and economical type of dual-fuel supply system. These engines have been successfully implemented in public transportation buses in China, Guangzhou.	**Benefits:** - Decreased NO_x, soot and CO emissions in diesel and LPG/CNG dual fuel mode, in comparison to only diesel mode. - Improved fuel consumption. - Slightly improved power output. - Low-cost engines. **Drawbacks:** - Decreased volumetric efficiency. - HC emissions quite high on partial load in diesel-LPG mode with higher content of LPG. - CO emissions increased at diesel-LPG mode with partial load (but decreased at full load).
[10]	Evaluation of combustion process and knocking behaviour in LPG (gas phase) and LFO dual fuel engine, with pilot LFO and compression ratio 16.	- Pilot fuel dose and injection timing strongly influence the engine's efficiency, maximum torque, emissions and combustion process parameters. - Optimal results for dual fuel mode were achieved with injection at 20°BTDC and pilot fuel dose at 30% of total consumption at full load. These settings achieved a lower maximum pressure and average pressure rise rate was achieved and without knocking.
[11]	Modification of a single cylinder, naturally aspirated, direct injection CI-engine for dual fuel operation (LPG and diesel).	- It was possible to achieve significant reduction in NO_x emissions. - Beyond half load operation, thermal efficiency was improved in comparison to diesel mode.
[12]	Exhaust emissions and engine performance analysis of a single cylinder CI engine in dual fuel mode with an air-LPG mixture.	- Improved engine performance and reduced BSFC with air-LPG mixture, in comparison to diesel fuel.
[13]	Evaluation of the combustion process of LPG for heavy duty applications, based on tests in constant volume combustion chamber (CVCC).	- Maximum flame propagation speed was noticed with stoichiometric air-fuel mixtures. - The composition of the mixture had a more significant effect on flame propagation than initial pressure and temperature conditions.
[14]	Experimental and numerical study of spray and atomisation characteristics of LPG and LPG-diesel blends at high-pressure injection. Experiments carried out in a high-pressure, constant volume chamber.	- Experiments showed that at same injection conditions were applied, 100% LPG had the shortest spray tip penetration, in comparison to blends. Increased ambient pressure also helped for all fuel blends tested. - Generally, numerical approach predicted experimental results well, except at late injection stage, due to a low sensitivity of the high-speed camera.

Reference	Idea and purpose of the research	Conclusions/ Findings
[15]	Comparison of performance and emissions between diesel-LPG blends (0%, 10%, 20%, 30%, 40%) in a single-cylinder, direct injection (DI) CI engine. Fuel was injected in liquid form with a high-pressure fuel feed pump, compressed nitrogen tank and special injector nozzle.	- Results showed that blending diesel with LPG resulted in lower maximum cylinder pressure. In blends with more than 20% LPG: peak heat release rate was lower, NO_x emissions lower and combustion duration longer, in comparison to diesel operation.
[16]	Analysis of EGR in a homogeneous charge compression ignition single-cylinder engine (HCCI) with LPG fuel and diethyl ether as pilot fuel for ignition.	- With the use of EGR, part load brake thermal efficiency increased by 2.5%. At full load, NOx emissions reduced by 68% and decreased peak cylinder pressure. - Slower combustion rate when EGR was used at higher percentages.
[17]	Various loads were tested on a CI engine in dual fuel mode with diesel and LPG fuel.	- Results showed that by blending LPG and diesel, up to 80% of diesel can be saved. However, in these tests, only 45% could be achieved, due to high engine vibrations. - Engine break power was increased by 15% and specific fuel consumption was decreased by 30% in dual-fuel mode. The reason for this can be better mixing of air and LPG and an improved combustion efficiency in comparison to diesel mode.
[18]	Performance testing of a dual-fuel (diesel + LPG) CI engine at various loads.	- At lower loads, blends with up to 50% LPG could be used. At higher loads, only 20% LPG could be used. - At loads up to 35%, engine performance was better with diesel. At higher loads, fuel blends performed better.
[19]	Investigation of the effect of propane to butane ratios in LPG fuel on the performance of a single cylinder, naturally aspirated, indirect injection Ricardo E6 CI engine. Ratios tested were: 100:0, 70:30, 55:45, 25:75 and 0:100).	- Experiments showed that engine parameter tuning has a significant effect on engine performance. - Efficiency increased at higher compression ratio, which was limited by the engine's knock limit. - Increased engine speed and advanced pilot timing both resulted in lower efficiency and higher combustion noise. - All LPG blends achieved a higher efficiency at higher mass flow rates. The performance of the fuels was similar, but the noise levels changed.
[20]	Study of the knock characteristics of a DI CI engine with diesel pilot fuel and LPG main fuel. LPG was introduced into the cylinder at a pressure slightly higher than ambient.	- Combustion process was slower to start but speeded up and completed faster than with diesel fuel. - Main factors affecting engine knocking were: pilot fuel quantity, engine load and speed, gas flow rate and time interval of secondary ignition.