CIMAC CONGRESS
HELSINKI | JUNE 6–10, 2016

2016 | 233

The 2-stroke Low-Pressure Dual-Fuel Technology: From Concept to Reality

02 Product Development - Gas & Dual Fuel Engines

Marcel Ott, Winterthur Gas & Diesel

Ingemar Nylund, Intec Ab
Roland Alder, Winterthur Gas & Diesel Ltd.
Takayuki Hirose, IHI Corporation
Yoshiyuki Umemoto, Diesel United Ltd.
Takeshi Yamada, IHI Corporation

This paper has been presented and published on the occasion of the 28th CIMAC World Congress 2016 in Helsinki. The CIMAC Congress is held every three years, each time in a different member country.
The Congress programme centres around the presentation of Technical papers on engine research and development, application engineering on the original equipment side and engine operation and maintenance on the end-user side. The topics of the 2016 event covered Product Development of gas and diesel engines, Fuel Injection, Turbochargers, Components & Tribology, Controls & Automation, Exhaust Gas Aftertreatment, Basic Research & Advanced Engineering, System Integration & Optimization, Fuels & Lubricants, as well as Users' Aspects for marine and land-based applications.

ABSTRACT

Gas as a fuel for merchant shipping has gained considerable interest during the last years, primarily driven by its inherent potential to reduce emissions, and secondarily due to the fact that gas is a sulphur free fuel, with the potential to be readily available at a feasible price.

Winterthur Gas and Diesel Ltd. (WinGD) has developed a low-pressure Dual-Fuel technology for slow speed 2-stroke engines which is capable of meeting emission requirements far below the IMO Tier III limits, utilizing a low cost, highly efficient and reliable low pressure gas admission system. The basic concept was presented in the 2013 CIMAC conference in Shanghai and the technology is now implemented on several bore sizes in the Generation X-engine range and is taken into commercial use.

The technology is based on the lean-burn Otto-cycle combustion process in order to exploit the full emission reduction potential of gas as a fuel and thus fully complying with IMO Tier III emission limits without any exhaust gas after treatment. Additionally, only low pressure gas supply to the engines is required, in contrary to the competing technologies. These advantages allow substantial simplifications on the plant installations, resulting in considerable savings of first time installation cost and very competitive operating cost. The first application of this technology was made on a 50cm bore engine type, where a number of engines have been produced and in service for some time. In the meantime, the Dual-Fuel technology has been developed and released for larger bore, lower speed engines like the X62DF and X72DF. A joint development project with WinGD’s Japanese licensee Diesel United Ltd. (DU) has resulted in the installation of a 6 cylinder X72DF test engine at DU’s Aioi works, and extensive testing has taken place, ensuring that the Dual-Fuel technology is fulfilling the market requirements. Preparations to extend the technology to even larger bore X engines have been taken.

The present paper gives an overview of the concept and describes the application of the technology on larger bore engines. Besides an outline of the test bed setup, test results of the X72DF test engine are being presented. Additionally, a number of market references of 2-s Dual-Fuel engines are shown.

INTRODUCTION

When the development of the two-stroke low-pressure DF lean-burn technology started, the extensive experience from Wärtsilä's four-stroke developments could be utilized as a base in several technology areas. These include engine automation and control, pilot injection, gas supply system and engine testing to mention a few. Still, while having some of the technologies in place, the characteristics of the two-stroke engine required additional fundamental developments, which at the same time turned out to open up new opportunities.

One example is the differences introduced by the low engine speed, giving more time for gas mixing but also requiring better control of the combustion speed. Another difference is the uniflow scavenging process of the two-stroke engine, which on one hand is enabling certain stratification of the air/gas mixture, fresh air and exhaust gas. On the other hand, it represents a challenge to minimize the rest gas quantity in the large cylinders compared to four-stroke engines, which have a separate scavenging stroke. One example of a benefit of the two-stroke engine is the fully variable exhaust valve timing, which can be utilized to flexibly adjust the effective compression ratio and by this control the combustion via the influence on air excess ratio and compression temperature.

Understanding the boundaries while further developing the features has required some iterations, but at the same time has been a key to achieve today's results with the X-DF technology. The most encouraging achievement is the successful transfer of the technology from the 50cm bore research engine to the X72DF engine tested at Diesel United Ltd., where the expected results were confirmed or even exceeded.

REVIEW OF THE X-DF DEVELOPMENT

It was already in the 70ies and 80ies of the last century when first trials with gas fuelled two-stroke engines took place in the former Sulzer Diesel company. Both combustion principles, low-pressure Otto cycle and high-pressure Diesel cycle were tested and evaluated, but none of the concepts achieved a breakthrough in the market at that time.

After a new assessment of the market requirements and available technologies, the development of the low-pressure dual-fuel technology for low-speed two-stroke engines was restarted in 2011, and a designated test engine was installed in the engine laboratory in Trieste, Italy. As a first step, one cylinder of the RT-flex50 based diesel engine was converted to enable it to operate on gas, while the other five cylinders remained running on diesel with the original equipment. This setup was used to develop and validate the basic concepts for gas admission and ignition, and to define the control principles. This process work and initial results were presented in a paper at the last CIMAC Congress in Shanghai in 2013 [1].

In 2013, the full engine was equipped with more industrialized equipment for gas operation, upgrading it to an excellent development tool for full scale engine operation tests. Main focus was directed to the development of engine components, performance and control system features, finally resulting in the X-DF solution which is commercially available today.

As an imminent need for larger DF engines was recognized in the market, the technology was transferred to the larger bore engine types X62DF and X72DF, and a full scale 6X72DF technology demonstrator engine was set up at WinGD's licensee Diesel United Ltd. (DU) in Aioi, Japan (see Figure 1). DU has been the cooperation partner in this development since the beginning of the project. The engine went into operation in early 2015 and was demonstrated to numerous representatives of the shipping world in April 2015. The fact that the low-pressure technology has proven to also be perfectly suitable for larger bore engines has triggered an enormous interest for the X-DF solution in the market.

Figure 1: W6X72DF technology demonstrator

As of February 2016, the order book for X-DF engines consists of 35 firm orders for RT-flex50DF, X62DF and X72DF engines, excluding potential project options. While a number of RT-flex50DF engines have been factory acceptance tested and delivered to shipyards, also testing of the first X62DF customer engine has been completed. An additional milestone has been reached with the Type Approval Testing of the RT-flex50DF engine type in December 2015, witnessed from representatives of all major classification societies (see Figure 2). The successful completion of this test demonstrates that the technology fulfils all the safety standards required and is now fully ready for commercial use in marine propulsion applications.

The first RT-flex50DF engines will go into commercial operation within first half of 2016, both on product tankers sailing in the Baltic Sea, as well as on a small LNG carrier operating in Chinese waters. Engine deliveries from several licensees will continue, and sea trials of the first large LNG carriers with twin-engine propulsion are scheduled for 2017.

Figure 2: Type Approval Test of the RT-flex50DF

DESCRIPTION OF THE X-DF TECHNOLOGY

THE X-DF CONCEPT

The low-pressure dual-fuel technology developed by WinGD for its X-DF engine series builds on Wärtsilä's long experience with what has become a well-proven industry standard on medium-speed dual-fuel engines. In contrast to high-pressure gas injection engines, which operate on the Diesel cycle, WinGD's low-pressure X-DF engines work on the lean-burn Otto cycle when operated in gas mode – i.e. ignition of a compressed lean air/gas mixture by injection of a very small amount of liquid pilot fuel.

As demonstrated on test and customer engines, WinGD's X-DF engines are characterised by stable combustion, inherently low NO_x emissions and high overall system efficiencies as well as safe gas operation. In terms of NO_x, WinGD X-DF engines undercut IMO Tier III limits for Emission Control Areas (ECAs) [2,3] by considerable margins without any additional measures, such as EGR or SCR while running on gas.

Moreover, with the low-pressure gas admission, the gas fuelling system does not require any high-pressure compressors, considerably reducing equipment costs, on-board energy consumption and maintenance during operation. Additionally, a large supplier base is available for gas supply systems, as the components are similar to systems installed on numerous four-stroke DF engines, proven in thousands of hours in field operation.

Due to possible fast combustion or pre-ignition, the maximum brake mean effective pressure (BMEP) of the X-DF engines has been limited to 17.3 bar, as explained in the chapter about fundamentals of combustion. This results in a reduced power density of the DF engines compared to the 21 bar maximum achievable BMEP on today's modern two-stroke diesel engines. However, the market response shows that the engines are commercially attractive, due to low investment cost and the superior overall system efficiency.

Three of the key systems of the X-DF concept are described in more detail:

GAS ADMISSION SYSTEM

The gas distribution and admission system has been specifically designed for the requirements of the two-stroke engine. Gas is supplied from aft- or free end and distributed in gas manifolds along both sides of the engine to feed gas to each cylinder. In order to ensure fast depressurization of the distribution pipes in case of a malfunction, both gas manifolds are equipped with a fast shut-off valve on the inlet side, and a fast vent valve at the end of the pipe (see Figure 3). All gas-containing pipes are of double wall design and made from stainless steel, in order to fulfil requirements of marine classification societies.

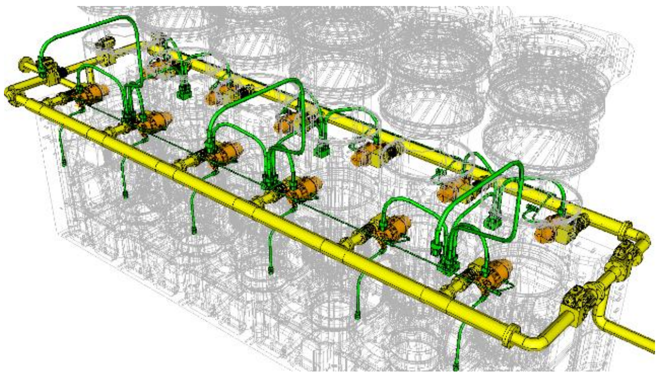

Figure 3: Gas distribution and admission system

Each cylinder is equipped with two hydraulically actuated gas admission valves (GAV), injecting gas through the cylinder liner wall directly into the cylinder. Since the gas is injected before the actual compression starts, a low gas pressure of 10…12 bar is sufficient to achieve a homogenous air/gas mixture, even at full engine load. The system is designed for a maximum pressure of 16 bar to cope with lower heating values down to 28 MJ/Nm^3, while maintaining similar gas injection durations and ensuring unchanged maximum power output.

A new design, shown in Figure 4, was developed in-house for the gas admission valves, in order to satisfy the specific requirements of the two-stroke engines. Particular attention was given to simplify the valve concept in order to maximize reliability and ensure simple maintenance. Hydraulic power is delivered from the existing servo oil circuit, also used for actuation of the exhaust valves, eliminating the need for an additional hydraulic system. For good long-term stability and tightness, a robust valve design with a spherical valve seat has been chosen. Sealing oil is used to separate the gas side from the servo oil side, as well as to lubricate the valve guide. For increased operational safety, the valves are equipped with a lift measurement sensor, which allows the engine control system to take immediate action in case of a malfunction of a valve or a related control signal. The cartridge-type design allows replacing all the moving parts without removing the valve housing or the gas piping system. This ensures short downtime periods for repairs or replacement of the valves and reduces the risk of gas leakages induced by maintenance.

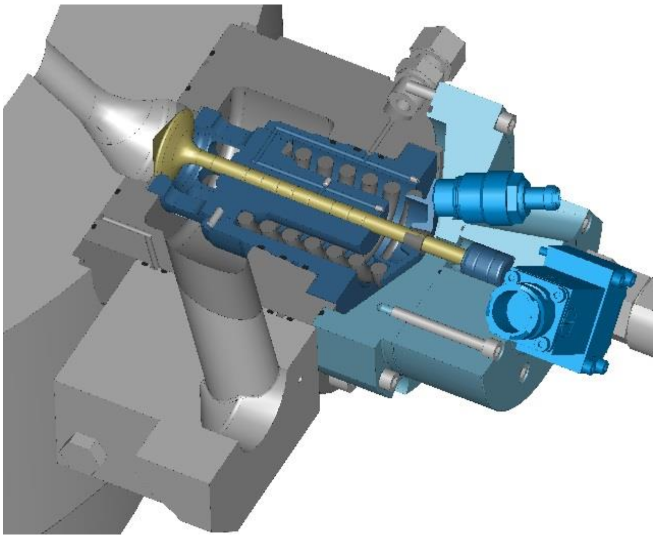

Figure 4: Gas admission valve section

Excellent results have been achieved in terms of functionality and reliability of the valves in rig and engine testing. After passing the gas admission valve, the gas fuel is injected into the cylinder by gas admission bores, which pass through the cylinder liner wall. Both, CFD simulations and engine testing have confirmed that a careful design of the admission bores in terms of dimensioning and geometry is essential for an optimized performance of the engine. The nozzles have a strong influence on mixture homogeneity.

PILOT FUEL IGNITION SYSTEM

To ignite the lean air/gas mixture, with its high self-ignition temperature, the compression end temperature in the engine is not high enough to create sufficient reactivity. Direct injection of pilot fuel, based on common rail injection, has proven to be a reliable solution for medium-speed four-stroke engines. It provides a sufficient ignition stability with a minimum fuel quantity. For the two-stroke engines, a solution with a pre-chamber, shown in Figure 5, was developed. The pre-chamber concentrates the energy of the pilot fuel, leading to higher temperatures than in the main chamber. As a result, there is more margin for the self-ignition of the pilot fuel, giving a shorter ignition delay with less stochastic variations of the timing. For the same reason, the combustion duration is short, resulting in a rapid pressure increase in the pre-chamber. This creates a powerful torch, penetrating well into the main combustion chamber. This concept provides ignition energy well distributed in the combustion chamber, with minimized pilot fuel quantities of below 1% of the total energy input at full engine load, a level considerably lower than of any other system available on the market. Pilot fuel is supplied to each injector by a small-size common rail system.

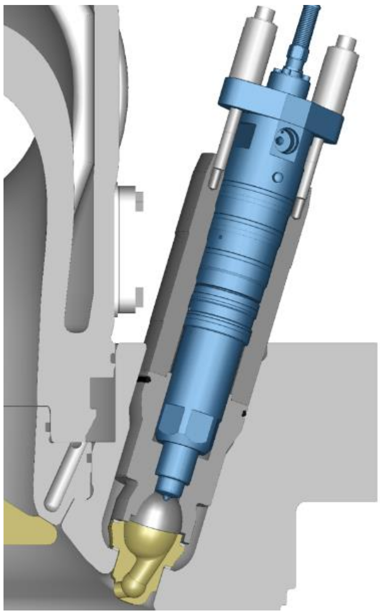

Figure 5: Pilot fuel injector and pre-chamber

ENGINE CONTROL SYSTEM

All WinGD two-stroke dual-fuel engines are operated by a UNIC-based engine control system. The UNIC control system is an embedded engine management system of modular design and is applied to a range of diesel engines in the latest X-engine series. For dual-fuel engines, the system is equipped with the following additional main functionalities for gas operation:

- Fuel mode transfer to change from liquid to gas mode and vice-versa during engine operation
- Control of timing, duration and pressure for gas admission and pilot fuel injection

- Control of the exhaust waste-gate for adjustment of scavenge air pressure and lambda
- Knock and misfire detection and control, for combustion control and engine safety
- Compression and firing pressure balancing to optimize engine performance by balancing individual cylinders
- Additional monitoring features for enhanced operational safety

FUNDAMENTALS OF COMBUSTION

Unlike with the Diesel process, the heat release in the Otto process cannot be controlled by the rate of injection since all the fuel is admitted and mixed with air before ignition. In the Otto process, combustion is to some extent guided by the turbulence and swirl level during compression and combustion, but the basis for combustion control on a given engine design is the air/fuel (A/F) ratio of the mixture. Correct adjustment of the A/F ratio ("lambda") together with correct ignition timing gives the optimum combustion speed, resulting in an operating point with best efficiency and lowest NO_x emissions. The correct A/F ratio also enables the engine to run on high BMEP levels without pre-ignition, knocking or misfiring.

As shown in Figure 6, there is a "rich limit" for the A/F ratio, which is to a certain extent depending on residual gas and lube oil content. If this limit is exceeded, the combustion will accelerate, increasing firing pressures and NO_x formation. Further reduction of the A/F ratio may lead to spontaneous ignition prior to the pilot injection ("pre-ignition") and/or may develop to a "knocking" combustion. The rich limit is defined by the firing pressure limit and the required knock margin.

Increasing the A/F ratio towards the "lean limit" will gradually slow down combustion, reducing efficiency and NO_x, increasing combustion instability with growing risk for partial or complete misfiring (see Figure 6). At the same time, eventually, emission of unburnt fuel ("methane slip") is increased. The lean limit is defined by the combustion instability, which is quantified as COV (Coefficient of Variation), the average variation of IMEP. A COV value of 2.5% is considered to be an industry standard. This limit can be shifted considerably by applying a strong ignition system as described on the next page.

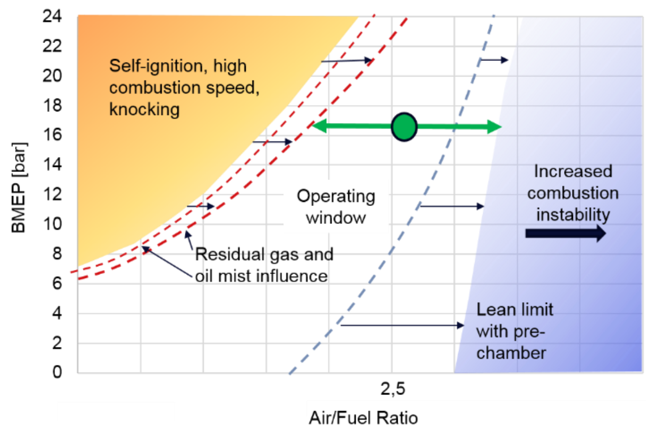

Figure 6: Lean-burn Otto combustion limits

The gas composition has an influence on the operating window. The X-DF combustion layout is made to cover gas with a minimum Methane Number (MN) of 80 without a reduction of the nominal engine output. Lately, engines have however been tested with full BMEP on MN 65.

Oil mist from cylinder lubrication has proven to have a minor influence on the combustion, indicated in Figure 6. Even if some of the lube oil droplets are igniting during compression, the ignition energy of individual droplets is too small to ignite the lean air/gas charge and have an influence on the combustion. Therefore, standard cylinder lubrication oils with a low base number (TBN 15…25) can be used for gas operation.

The area between the rich and lean limit is defined as the operating window. The optimum operating point has the best trade-off between efficiency and NO_x, while leaving sufficient margin to both limits. In Figure 7, typical cylinder pressure curves are presented for "rich", "normal" and "lean" combustion.

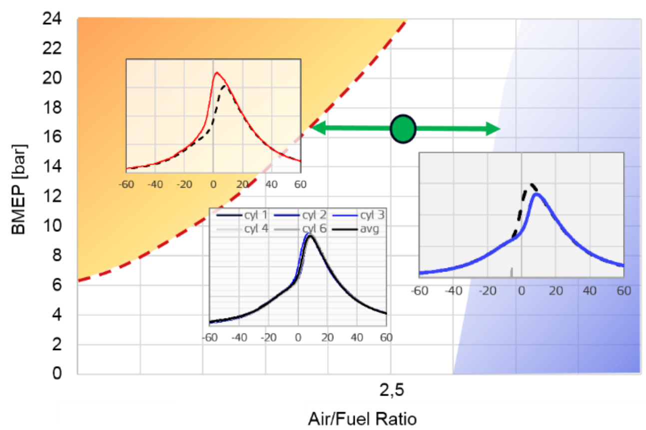

Figure 7: Examples of 'rich', 'normal' and 'lean' combustion

TURBO CHARGING AND CONTROL OF AIR/FUEL RATIO

Practically, the A/F ratio is controlled by the scavenge air pressure, which is adjusted by a by-pass valve for the turbo charger turbine (“exhaust waste-gate”), as shown in Figure 8. The required scavenge air pressure depends mainly on the actual gas quantity injected – which is dictated by the engine load – biased with adjustments based on actual operating conditions like ambient temperature, pressure, etc. The target values for scavenge air pressure are defined in the parameter maps of the engine control system. Besides this, also the exhaust valve closing timing directly affects the A/F ratio (trapped air mass).

The engine control system detects, if the engine for some reason is not operated within the target operating window. By constant monitoring of firing pressures, fast combustion, excessive firing pressures and misfiring is detected. In addition, the engine is equipped with conventional knock sensors to provide additional safety.

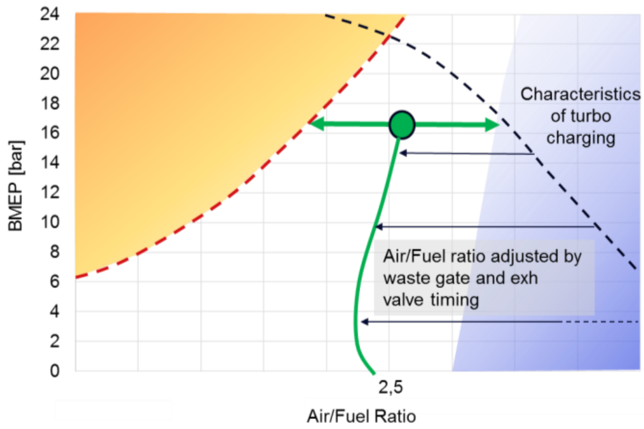

Figure 8: Control of Air/Fuel ratio

PILOT INJECTION/IGNITION

With a powerful ignition source as described above, the lean limit is extended and the engine can be run at a higher A/F ratio. This results in unchanged high efficiency while both, unburnt hydrocarbon emissions (methane slip) and NO_x formation are reduced compared to an open chamber pilot injection. A graphic illustration of this is shown in Figure 9.

With an optimized configuration of the pre-chamber and the pilot injection nozzle, the combustion is relatively insensitive to changes of the pilot fuel injection pressure and duration/quantity. Since the pilot mainly serves as a trigger for the start of combustion, any excessive pilot fuel quantity will not help to speed up the combustion or make it more complete.

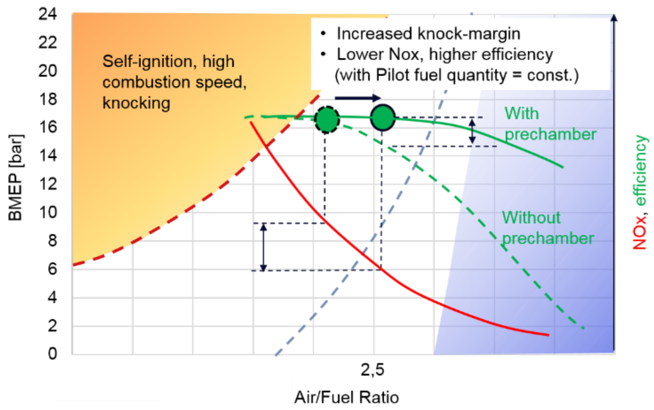

Figure 9: Comparison of pre-chamber vs. open chamber pilot

Figure 10 shows an example of a typical pressure trace in the pre-chamber. It can be seen that a strong pressure peak is created, which then shoots the ignition flame jet into the main chamber for ignition of the main air/gas charge.

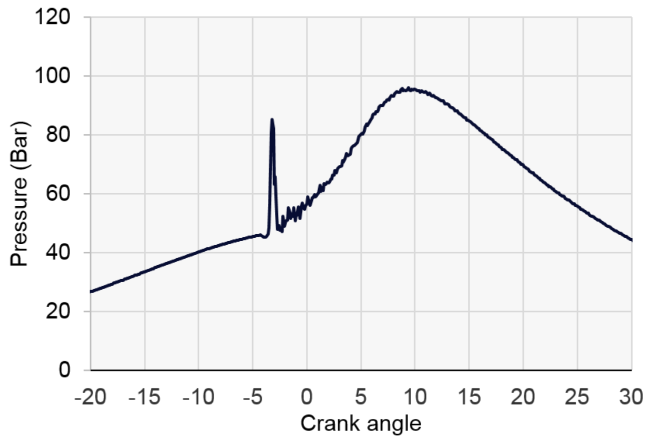

Figure 10: Example of pressure trace in pre-chamber

GAS ADMISSION

The geometry and location of the gas injection nozzle is designed with the target to maximise the homogeneity of the air/gas mixture, hence avoiding rich zones that would be prone to self-ignition and result in high combustion temperatures and NO_x formation levels.

On the other hand, a certain stratification is also strived for, for the following reasons:

- starting the gas injection as late as possible to minimise the risk that gas is getting in contact with hot rest gas in the upper end of the combustion chamber, which could result in self-ignition and early combustion

- by the above, direct methane slip into the exhaust can be avoided

- ending the gas injection early enough to create an air buffer on piston top. By this, the piston top land is not filled with air/gas mixture during compression, and also this potential source of methane slip is minimised

In addition to creating an air/gas charge with maximum homogeneity (yet stratified), the gas admission needs to:

- be accurate and repeatable from cycle to cycle
- have shortest possible opening/closing ramps
- have a minimum dead volume downstream the valve

Typical timings and durations of the gas admission process are shown in Figure 11.

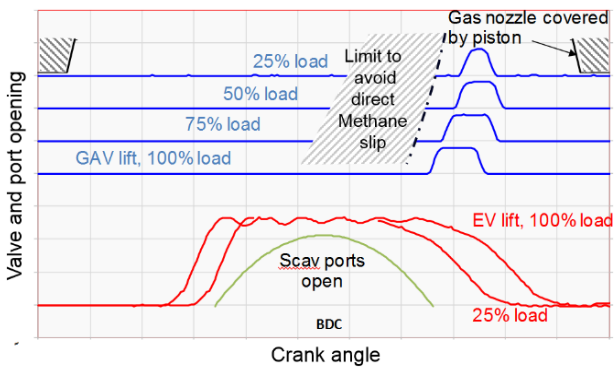

Figure 11: Gas admission valve lift curves related to boundaries

FUTURE DEVELOPMENTS

While the X-DF technology is released for commercial engines, the development work is continuing with the target to evaluate future potential for increased fuel efficiency in gas and diesel mode, as well as for higher maximum power output. Additionally, new features will be introduced, giving further benefits to operators, such as a fuel sharing mode. This development – to only mention one example – will allow LNG carriers to burn gas and liquid fuel simultaneously and by this achieve the desired speed even with limited amount of natural boil-off gas, eliminating the need to additionally force boil-off. Because a considerable part of the fuel is combusted in a diffusive regime, the fuel sharing mode will be tuned to be IMO Tier II compliant.

In Figure 12, a comparison of cylinder pressures and rate of heat releases is shown for different fuel ratios. It can be seen that the gas combustion in mixed mode still reaches the same maximum rate of heat release, while the diesel fuel portion leads to an increased total combustion duration. Overall, it can be concluded that the combustion in the fuel sharing mode is an interpolation of full gas and full diesel combustion, which is also reflected in the cylinder pressure curves. The same is valid for NO_x emissions, which range from well below IMO Tier III levels in gas mode to Tier II levels in diesel mode.

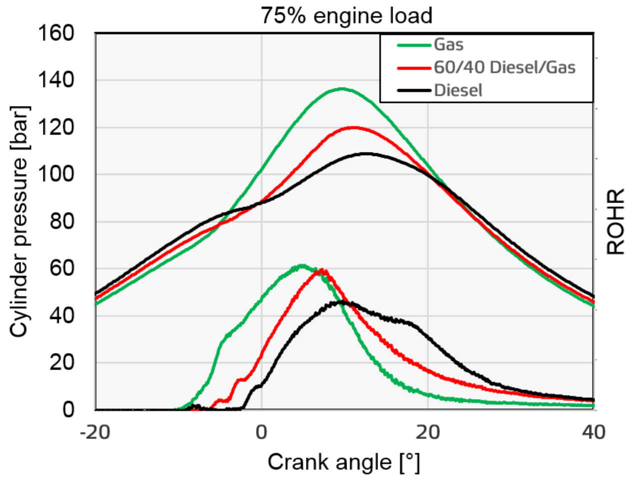

Figure 12: Comparison of cylinder pressures and rate of heat release with different fuel modes under comparable conditions

X72DF TEST RESULTS

TEST ENGINE AND PLANT

The main parameters of the 6X72DF test engine are shown in Table 1. The earlier described X-DF technology is applied to this engine. For testing purposes, this engine was designed to have some flexibility to vary the scavenging port position and size, the gas injection nozzle, the number of pilot injectors and the turbocharger type.

Engine type	**DU - WinGD 6X72DF**
Bore x Stroke	720 x 3086 mm
Max. Power (R1)	19.35 MW
Max Speed (R1)	89 rpm
Max. BMEP (R1)	17.3 bar
Turbocharger	2 x ABB A270-L or 2 x MET 60MB

Table1: Main parameters of the 6X72DF test engine

Fuel gas for operation of the test engine is supplied from the low-pressure fuel gas supply system (FGSS) shown in Figure 13. LNG, which is stored in an insulated tank, is pressurized by forced evaporation and volumetric expansion of the gas in the tank. The pressurized LNG is evaporated in a heat exchanger by hot water. The gas supply pressure can be changed flexibly up to 16 bar. During engine testing, fuel gas is typically supplied at a pressure of 12 bar. Two large buffer tanks are installed in the gas supply line to minimize the variation of the gas composition, which can be induced by a sudden change of the gas demand. The fuel gas composition is analysed by a gas chromatograph (SHIMADZU GC20B). Based on the composition, the lower calorific heating value (LHV) and the Methane Number (MN), which indicates the knock resistance, are calculated with the software 'AVL Methane'. The engine test results presented in this report are carried out with an approximate Methane Number of 67.

Figure 13: Low-pressure fuel gas supply system (FGSS) for test engine

PERFORMANCE TEST RESULTS

EXHAUST EMISSIONS

As shown in Figure 14, the NO_x emissions of the engine in Diesel mode complies with IMO Tier II NO_x regulations. In gas mode, the NO_x levels can be reduced to well-below-IMO Tier III NO_x limits without exhaust gas after treatment, which is a unique feature of the low-pressure DF technology.

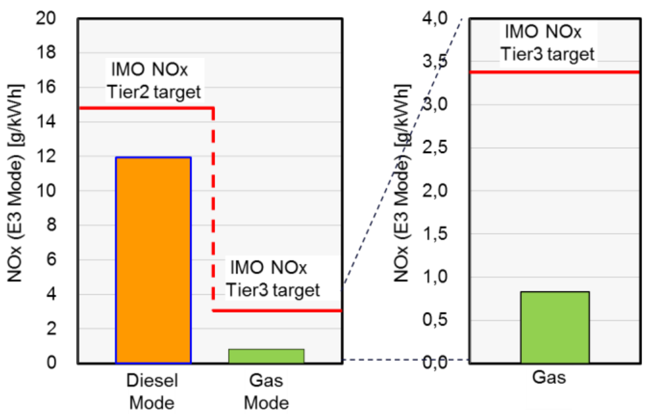

Figure14: NO_x emissions

Results of methane slip measurement are shown in Figure 15. Initially, the level was expected to be higher, due to the risk of direct methane slip during the scavenging of the cylinder. However, by adjustment of the gas injection timing, the scavenging process is completed (exhaust valve closed) before the air/gas mixture reaches the upper part of the cylinder, and direct methane slip is therefore eliminated. The remaining methane slip relates to flame quenching on combustion chamber walls and 'dead volumes' in piston top land etc. In comparison, the level is lower than on typical four-stroke lean-burn gas engines, as a result of lower engine speed, longer time for oxidation and a larger combustion space, with a favourable volume-to-surface ratio.

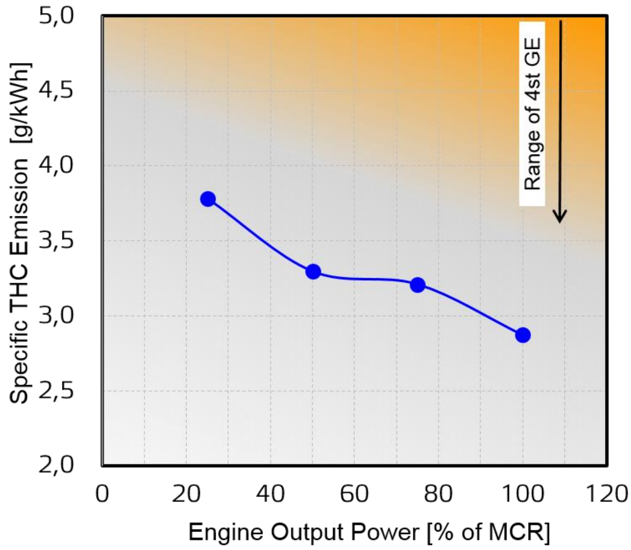

Figure 15: Measured THC emissions (methane slip)

FUEL CONSUMPTION

As shown in Figure 16, energy consumption in gas mode (including pilot fuel) was measured within tolerances as specified in GTD [4], WinGD's performance documentation tool. Only at full engine load, the limit is slightly exceeded. Further optimization is expected to bring the full load consumption to target values. At part load, consumption figures well below the specified values could be achieved, resulting in a benefit in real operation of a vessel.

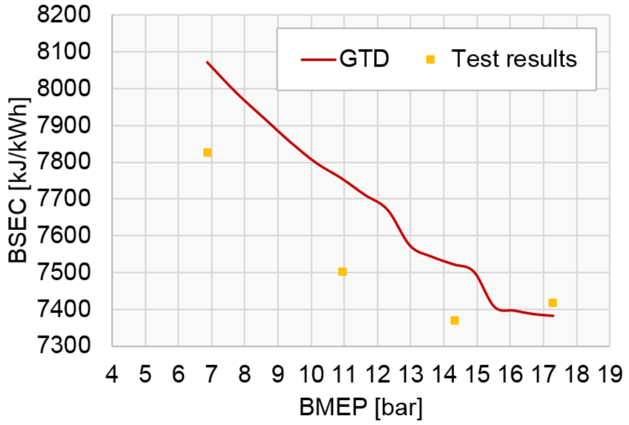

Figure 16: Measured energy consumption in gas mode

A major milestone was achieved in terms of pilot fuel consumption. As shown in Figure 17, it was possible to reduce the pilot injection quantity to record low levels of below 0.5% of the total fuel input at full load, while maintaining stable operation of the engine. When running in diesel mode, the pilot injection system is still running in order to keep the injector nozzle holes clean, though with further reduced injection quantity.

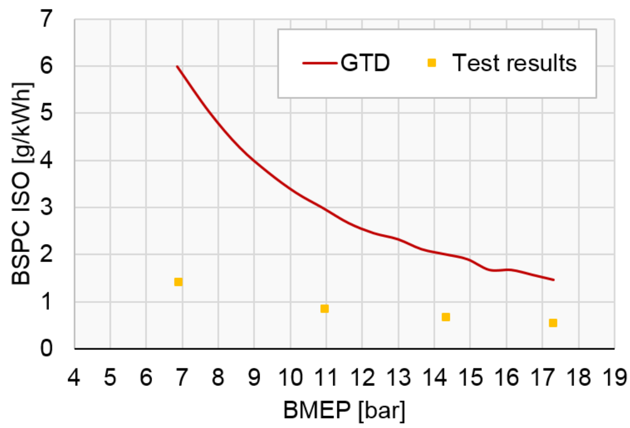

Figure 17: Measured pilot fuel consumption in gas mode

Diesel fuel consumption in the IMO Tier II compliant diesel mode was measured to be within the specified range, with margin at part load operation.

COMBUSTION BEHAVIOUR

Figure 18 shows an example of cylinder pressures over the load range in gas mode. The combustion is free of knocking or pre-ignition even with the low Methane Number of 67. Maximum cylinder pressure and the gradient of pressure rise are below design limits. Therefore, similar component reliability as on a diesel engine can be expected.

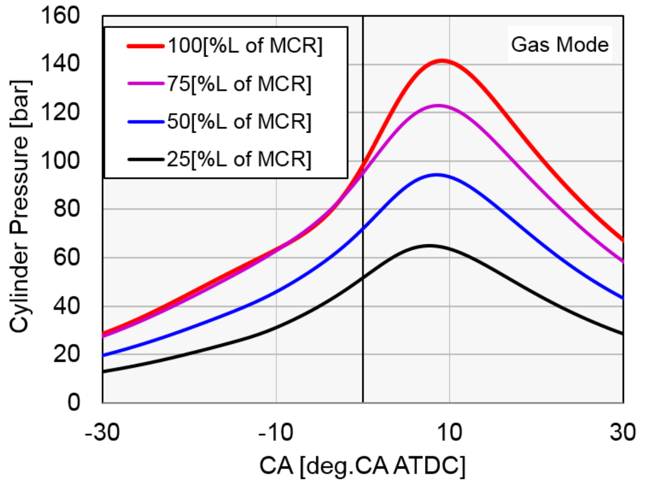

Figure 18: Cylinder pressures

Lean-burn pre-mixed combustion typically results in higher cycle-to-cycle fluctuation of cylinder pressures compared to diesel combustion, as also known from four-stroke lean-burn gas engines. Figure 19 shows that the cycle-to-cycle fluctuation of maximum cylinder pressures in gas mode is higher than in diesel mode. However, as shown in Figure 20, indicated mean effective pressure (IMEP) fluctuation in gas mode is as small as in diesel mode. Therefore, the engine speed deviation shown in Figure 21 is on well acceptable level, comparable to the deviations in diesel mode.

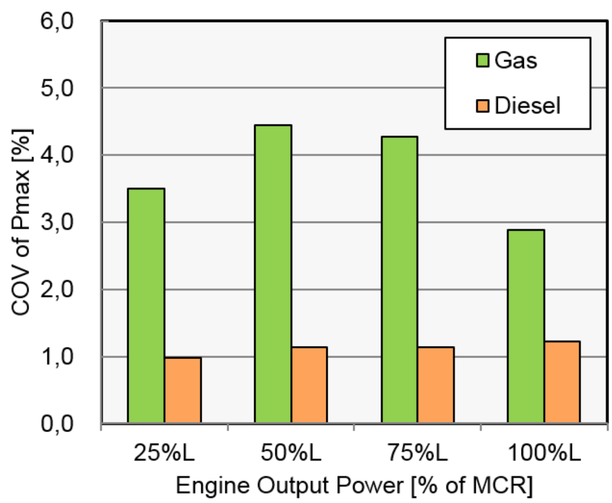

Figure 19: Cycle-to-cycle fluctuation of Pmax

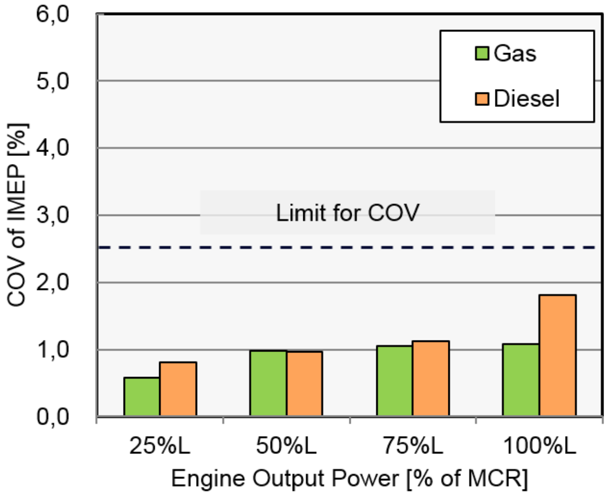

Figure 20: Cycle-to-cycle fluctuation of IMEP

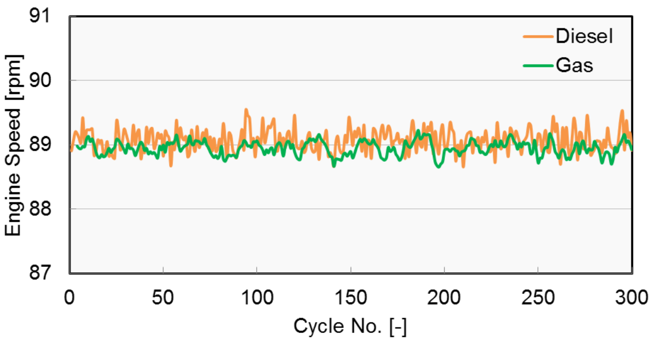

Figure 21: Engine speed fluctuation at 100% engine load

LOW LOAD AND WAVE LOADING OPERATION

Another unique feature of the low-pressure DF technology is the ability to operate at manoeuvring speeds (low load). Testing has confirmed that the X-DF engines can run stably at low engine loads without misfiring or instability. In order to achieve this, the strong ignition source from the applied prechamber system is essential. At the same time, the minimized pilot fuel amount results in a big enough gas amount for repeatable gas admission from cycle-to-cycle. The minimum engine load in gas mode for X-DF engines is currently set to 5% of MCR and is likely going to be further reduced in the future.

Reliable operation on gas under rough sea conditions is important for ocean going vessels. Under such conditions, waves and rolling motion of the ship induce load variations to the propeller, which directly transfer to the engine. For simulation of rough sea conditions, a number of tests were carried out on the engine test bed. Two examples are shown below:

- At a constant engine speed setpoint, a torque fluctuation of the water brake was set to 9% at a period of 20 seconds, which was limited by the water brake capability (Figure 22). As a result, the engine was able to operate in gas mode continuously without knock or misfire.

- In addition, a test with a variation of the engine speed setpoint was carried out in both, diesel and gas mode in order to further investigate the dynamic behaviour of the engine. The deviation of the speed setpoint was set to +/- 4.3 revolutions at 80.5 rpm (75% of MCR power) as shown in Figure 23. The engine speed follows the speed setpoint without knock or misfire. This means that the engine shows a good transient response behaviour and can therefore withstand to rough sea conditions in gas mode.

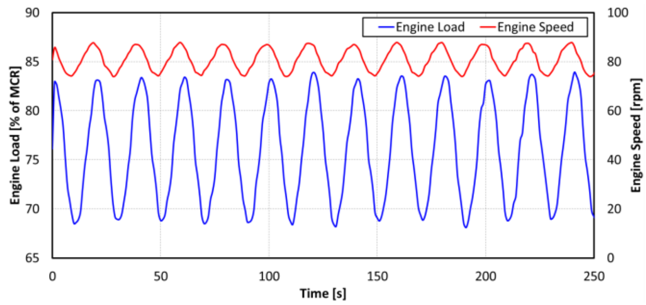

Figure 22: Engine response under wave load condition

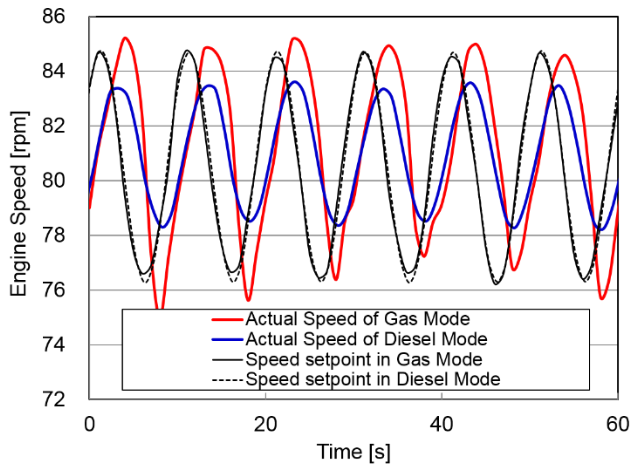

Figure 23: Engine response to a variation of the speed setpoint

CONCLUSIONS

For the first time, the low-pressure DF technology has been successfully implemented on a large two-stroke marine engine. Extensive R&D activities have brought the technology from a concept level to a solution ready for the market. This has been demonstrated in several Factory Acceptance Tests and the Type Approval Test of the RT-flex50DF engine. Additionally, it is shown in this paper that the technology could be transferred to the larger bore X72DF engine in a straightforward way and that the expected performance results were achieved.

Based on the experience gained by this development, it can be concluded that the results build a solid base for further extension of WinGD's low-pressure dual-fuel technology to the full engine portfolio.

DEFINITIONS, ACRONYMS, ABBREVIATIONS

DF: Dual-Fuel
LNG: Liquefied Natural Gas
UNIC: Unified Controls (engine control system)
CFD: Computational Fluid Dynamics
GAV: Gas Admission Valve
FGSS: Fuel Gas Supply System
GTD: General Technical Data (engine performance documentation)
ROHR: Rate of Heat Release
COV: Coefficient of variation
IMEP: Indicated Mean Effective Pressure
BMEP: Brake Mean Effective Pressure
MN: Methane Number
MCR: Maximum Continuous Rating
TBN: Total Base Number
ECA: Emission Control Area
EGR: Exhaust Gas Recirculation
SCR: Selective Catalytic Reduction
NO_x: Nitrogen Oxides
SO_x: Sulphur Oxides
THC: Total Hydro Carbon
A/F-ratio: Air/Fuel ratio

REFERENCES AND BIBLIOGRAPHY

[1] NYLUND I., OTT M. "Development of a Dual Fuel technology for slow-speed engines" CIMAC PAPER No. 284, CIMAC Congress, Shanghai, 2013.

[2] INTERNATIONAL MARITIME ORGANIZATION, IMO, "Revised MARPOL Annex VI", www.imo.org

[3] INTERNATIONAL MARITIME ORGANIZATION, IMO, THE MARINE ENVIRONMENT PROTECTION COMMITTEE, Resolution MEPC.177(58) "Amendments to the technical code on control of emissions of nitrogen oxides from marine diesel engines", NOx Technical Code 2008.

[4] GTD, Program for calculating the performance of WinGD engines, www.wingd.com, or https://www.wingd.com/en/media/general-technical-data/

CONTACT

This paper was written by:

- Marcel Ott, Winterthur Gas & Diesel Ltd., Switzerland
- Ingemar Nylund, Intec AB, Finland
- Roland Alder, Winterthur Gas & Diesel Ltd., Switzerland
- Takayuki Hirose, IHI Corporation, Japan
- Yoshiyuki Umemoto, Diesel United Ltd., Japan
- Takeshi Yamada, IHI Corporation, Japan

The authors can be contacted by writing an email to: marcel.ott@wingd.com