CIMAC CONGRESS
HELSINKI | JUNE 6–10, 2016

2016 | 120

The Development of the Modern Low-Speed Two-Stroke Marine Diesel Engine

01 Product Development - Diesel Engines

Andreas Kyrtatos, Winterthur Gas & Diesel

Marc Spahni, Winterthur Gas & Diesel
Sebastian Hensel, Winterthur Gas & Diesel
Reto Züger, Winterthur Gas & Diesel
Gregory Sudwoj, Winterthur Gas & Diesel

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.

The copyright of this paper is with CIMAC.

ABSTRACT

The prime mover of merchant ships has been for more than a century the marine Diesel engine, which is nowadays predominantly a low-speed, two-stroke, crosshead-type, reversible, uniflow-scavenged, turbocharged, electronic engine. The low speed engine is, because of its size, the most efficient thermal machine and, due to its fewer cylinders and consequently fewer moving parts, particularly reliable. The two-stroke cycle is applied to maximise the power to weight ratio and minimise engine size. Due to the low speed required and the finite maximum piston speed achievable, the marine engine features a very high stroke to bore ratio, which in turn is the main reason for the crosshead design and the uniflow-scavenging concept. Since the long two-stroke engine needs to be force-scavenged, the two-stroke marine Diesel engine is invariably also turbocharged. Finally, the demand for flexible engine tuning, optimised throughout the load range dictates the use of electronic engine control.

The market requirements for a two-stroke marine Diesel Engine differ profoundly from engines in other segments of the marine and automotive industries. The engine designer is obliged to correctly select the power output and the speed of the engine independently, since the engine is directly connected to the propeller without a gearbox. This attribute of the low speed engine, added to the low production volumes characteristic of the merchant shipbuilding market, makes the low-speed engine a highly customised product, tailor made for each application.

The purpose of this paper is to give an overview of the development of the modern low-speed two-stroke marine Diesel engine from the engine designer's point of view starting from the market requirements for power and speed, with insights that are relevant for the whole industry. A presentation will be made of

- the basic thermodynamic layout of the engine,
- the dimensioning and design criteria determining the size and design concept of the powertrain and structure parts,
- the design concepts for the hot parts and cylinder lubrication,
- the choices the engine designer has to make regarding core ancillary systems such as the fuel injection and exhaust valve actuation systems,
- the automation & control systems governing the modern electronic engine.

Particular focus will be placed on the trade-off the engine designer (licensor) is challenged with in terms of reliability, cost, manufacturability and serviceability of the engine, in order to make a competitive product for his customers, which are obviously shipowners, but also engine makers (licensees) and shipyards. Additionally, the tools and methods that the engine designer of the modern two-stroke marine Diesel engine has at his disposal will be described in the context of the development process.

INTRODUCTION

The prime mover of merchant ships has been for more than a century primarily the marine Diesel engine, which is nowadays predominantly a low-speed, two-stroke, crosshead-type, reversible, uniflow-scavenged, turbocharged, electronically controlled engine. The low speed engine is, because of its size[1], the most efficient thermal machine and, due to its fewer cylinders and consequently fewer moving parts, particularly reliable. The two-stroke cycle is applied to maximise the power-to-weight ratio, minimise engine size and for reversibility.

Due to the low speed required and the finite maximum piston speed achievable, the marine engine features a very high stroke-to-bore ratio, which in turn is the main reason for the crosshead design and the uniflow-scavenging concept. Due to the two-stroke concept and the typically long stroke, the marine Diesel engine is invariably turbocharged. Finally, the demand for flexible engine tuning, optimised throughout the load range dictates the use of electronic engine control.

The market requirements for a two-stroke marine Diesel Engine differ profoundly from engines in other segments of the marine and automotive industries. The engine designer is obliged to correctly select the power output at the correct speed, since the engine is directly connected to the propeller without a gearbox. This attribute of the low speed engine, added to the low production volumes characteristic of merchant shipbuilding market, makes the low-speed engine a highly customised product, tailor made for each application.

The purpose of this paper is to give an overview of the development of the modern low-speed two-stroke marine Diesel engine from the engine designer's (Winterthur Gas & Diesel, WinGD) point of view starting from the market requirements for power and speed, with insights and design choices that are relevant for the whole industry. A presentation will be made of:

- The basic thermodynamic layout of the engine,
- The dimensioning and design criteria determining the size and design concept of the powertrain and engine structure parts,
- The design concepts for the hot parts and cylinder lubrication,
- The choices the engine designer has to make regarding core ancillary systems such as the fuel injection and exhaust valve actuation systems,
- The automation & control systems governing the modern electronic engine,
- A simplified view into the engine development process divided in the above groups.

Particular focus will be placed on the trade-off off in terms of reliability, cost, manufacturability and serviceability of the engine that the engine designer (licensor[2]) is challenged with in order to make a competitive product for his customers, which are obviously shipowners, but also engine makers (licensees) and shipyards. Additionally, the tools and methods that the engine designer of the modern two-stroke engine has at his disposal will be described in the context of the development process.

MARKET REQUIREMENTS

The most important market requirements for marine Diesel engines, which shape the basic engine design, are the following:

- **Engine Power:** the fundamental requirement in order to propel a certain vessel at a certain speed is engine power

- **Engine speed:** required engine speeds are very low compared to other segments in order to optimise propulsion efficiency with propellers as large as allowable by the vessel design. To obtain these low speeds, ranging 1-2 Hz, the engine designer chooses high stroke-to-bore ratios

- **Expected lifetime**: the expected lifetime of a marine two-stroke engine is at least an order of magnitude higher than an automotive engine[3]

- **Engine footprint:** the engine footprint refers mainly to the engine base and the Piston Dismantling Height (PDH). Slim engines can be installed far back into the vessel hull and lower PDH accounts for lower engine room height; both characteristics optimising cargo capacity

Equally important requirements shared with other engine industries include emissions (adhering to regulations), fuel and lube oil consumption, reliability, service friendliness and manufacturing cost. During the engine development, the engine designer tries to balance these requirements optimising the trade-offs arising such as reliability vs. manufacturing cost, fuel consumption vs. emissions, simplicity and service friendliness vs. modern, sophisticated and flexible technology.

[1] Large size means that the lower surface-to-volume ratios in the combustion chamber will result in lower heat transfer losses.

[2] Marine two-stroke engines are typically made by engine makers who license the designs from a licensor (engine designer).

[3] Typical engine lifetime: auto 6'000 hours, marine 150'000 hours.

THE TWO-STROKE LAYOUT FIELD

The two-stroke engine basic characteristic, which is derived directly from market requirements (power and speed), is the layout field. The layout field is defined by four straight lines (on a power-speed diagram with log-log scales) which represent the major design limitations as shown in Figure 1 and described below:

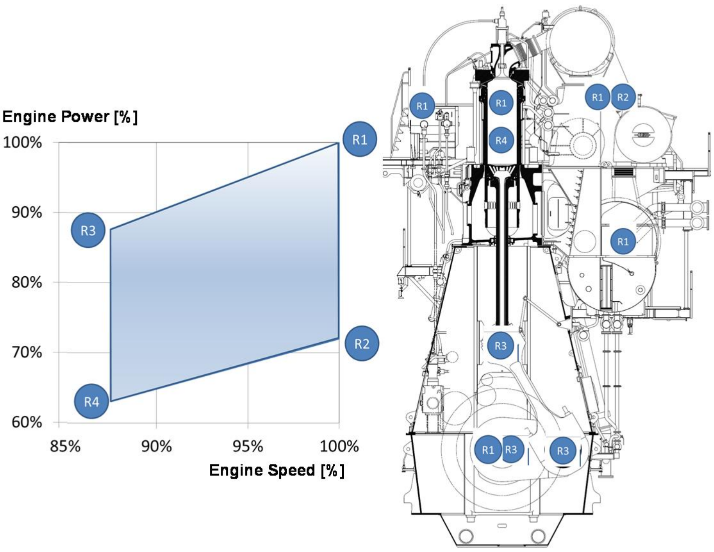


Figure 1: The two-stroke engine layout field and certain important influences to basic engine dimensions and designs

- **Torque limit (R1-R3):** a constant Mean Effective Pressure (MEP) line which determines the dimensioning of the crankshaft (static & dynamic torque), main bearing and hot parts
- **Speed limit (R1-R2):** the highest permissible speed, limited by tribological impacts for piston running (Mean Piston Speed (MPS) max 9.5m/s in modern engines) and rotating masses which also affect the size of the main bearing
- **Bearing layout limit (R3-R4):** Crosshead bearing and crank-pin bearing layout at maximum firing pressure and lowest speed (lube oil film thickness is at its lowest value at R3)
- **Scavenging limit (R2-R4):** also a constant MEP line, which represents the lower layout field limit due to thermodynamic and scavenging effects: high residual gases and diminishing benefit of derating.

These limitations will be described in more detail in the chapters that follow.

MAIN PARAMETERS FOR ENGINE LAYOUT

The stroke-to-bore ratio is the basic parameter that is derived from the layout field and determines the engine's main dimensions. Generally, a trend towards increasing stroke-to-bore ratios and mean piston speed, needed to optimise propulsion efficiency with minimised propeller speeds, can be observed throughout the last decades, as shown in Figure 2 below:

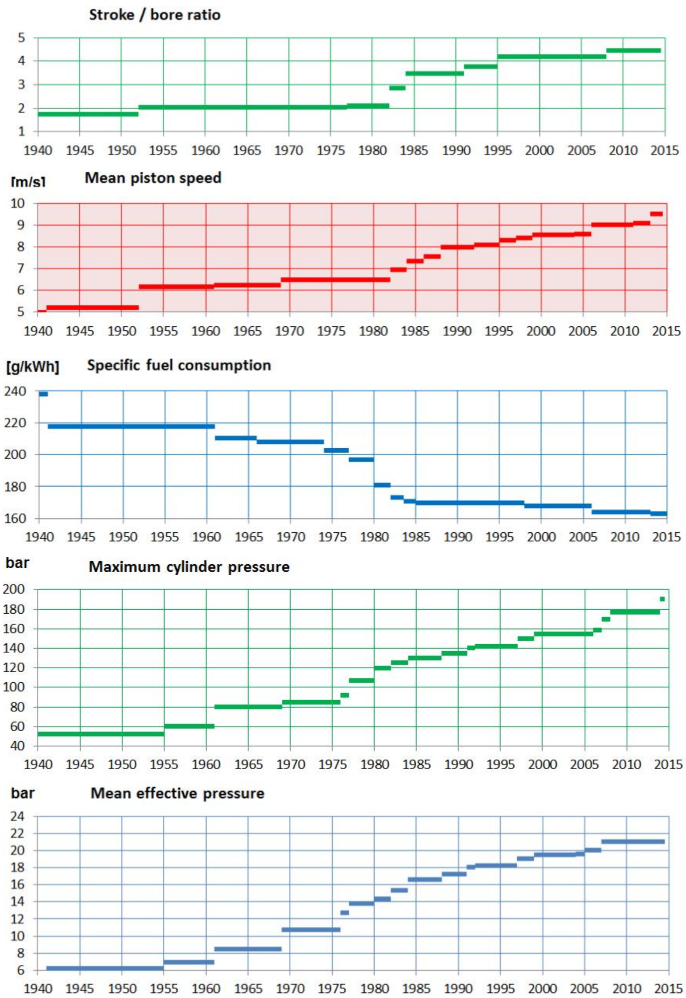


Figure 2: Historical Development of characteristic two-stroke engine parameters

This trend leads, in principle, above a certain stroke-to-bore ratio[4], to a compromise in Specific Fuel Consumption (SFC) due to increasing scavenging (throttling), friction (e.g. piston rings) and heat losses (due to higher surface to volume ratio). However, this tendency for increased SFC has been more than compensated on the engine side by a steady increase in Maximum cylinder Pressure (MCP) and MEP, which can be observed throughout the past 60 years[5].

[4] The optimal stroke-to-bore ratio is difficult to pinpoint and few studies have been made to that effect. In the most recent known study in 2015 [1], Parravicini concludes that the optimum is around 3.5.

[5] The large step in SFC improvement was made in the 1980's with the introduction of the uniflow-scavenged engine. Another major step, albeit influencing mainly part-load SFC has been the use of electronically controlled fuel injection.

THE ENGINE DEVELOPMENT PROCESS: AN OVERVIEW

PECULIARITIES OF THE TWO-STROKE ENGINE BUSINESS

The two-stroke engine business is characterised by a few peculiarities which have a significant effect on the engine development and design. Firstly, due to decentralised production, the licensee model has been established over decades whereby the engine manufacturer, or licensee, builds the engine under license from the engine designer, the licensor. This presents several organisational and technical challenges to the engine designer which will be described below. Secondly, due to the size of the engines and the sheer investment needed for a prototype of each new model, the prototype testing often takes place at the licensee's premises, on a customer engine. Thirdly, the broad use of Heavy Fuel Oil (HFO) with significant sulphur content and viscosities up to 2000cSt at 20°C, requires several measures such as trace heating, specially developed fuel pumping and injection equipment and special alkaline oils for piston lubrication to deal with acidic combustion residues. Finally, the constantly changing nature and volatility of the marine shipping business and the variety of vessels where the engines are applied require a high degree of engine customisation, which is made possible by the low production volumes of manufacturers.

THE ENGINE DEVELOPMENT PROCESS

The two-stroke engine development process is, similarly to development in other engine segments, highly iterative. The process will be simplified below for the purposes of this paper, and for better understanding will be separated in thematic and/or system sections. The most important development steps that take place are the following:

- Predevelopment of technologies
- Design concept selection
- Determination of cylinder distance, as a basic dimension influencing all design groups along the engine length
- Main engine development, comprising:
    - o Thermodynamic & scavenging layout
    - o Powertrain & engine structure development
    - o Hot parts & piston running development
    - o Fuel Injection & exhaust valve actuation application development[6]
    - o Engine control system application development
- Adaptation and preparation of basic engine design for emission control methods
- Full engine vibration calculations to check for inadmissible natural frequencies
- Component and sub-system testing on dedicated test benches and field tests
- Prototype full engine test at the engine manufacturer's (licensee) premises
- Factory Acceptance Test by the licensee
- Sea trial on the first vessel

The following diagram displays some of the most important process steps, which will be described in detail in the chapters that follow:

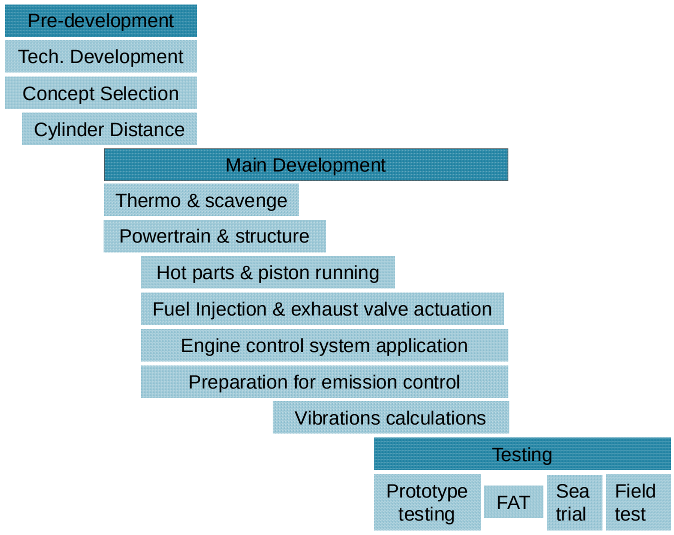


Figure 3: Overview of the engine development process

For reference throughout the paper, Figure 4 below gives an overview of the engine components of a Wärtsilä 6 cylinder X62 engine:

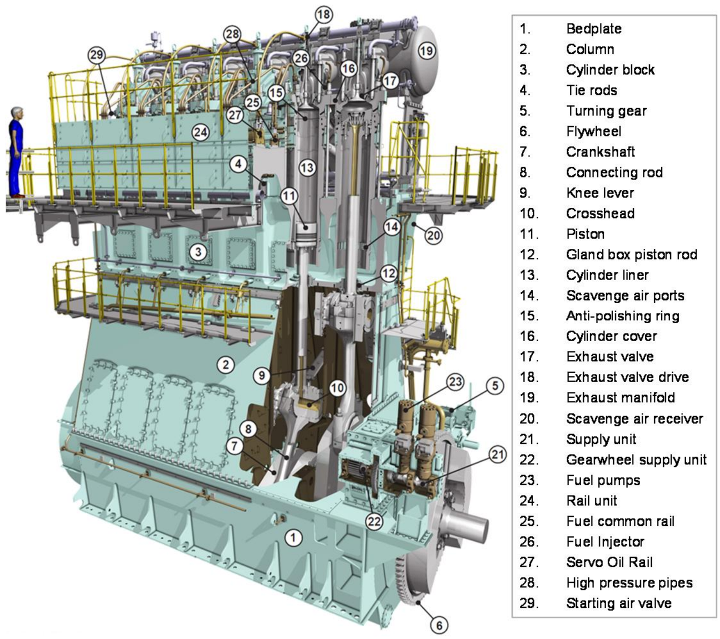


Figure 4: Overview of the basic engine components for reference

[6] Technology developments, e.g. fuel injection or engine control systems, take place in parallel or even in advance of the engine development. Shown here is the application development, which is the application of an existing technology to a new engine size.

THERMODYNAMIC & SCAVENGING LAYOUT

The optimisation of engine efficiency has always been one of the most important targets of the internal combustion engine developer. Therefore, the engine development starts with the thermodynamic layout.

THERMODYNAMIC & SCAVENGING LAYOUT PROCESS

The thermodynamic and scavenging layout determines fundamental dimensions of the engine such as the shape of the combustion chamber, size and location of the inlet ports, turbocharger selection, cooler sizing, valve and injection timing, etc. The most important steps in this development are described below:

- Firstly, a semi-empirical, basic layout takes place based on assumed inlet port and exhaust valve dimensions, firing pressure, scavenge pressure and valve timing. These parameters are often taken from engines with similar stroke-to-bore ratios
- In parallel, derating[7] strategies are developed applying state of the art tuning methodologies
- A zero-dimensional process calculation is then made for the entire rating field, which gives as output the first approximation of cylinder pressure, exhaust flows, temperatures, injection & valve timing, etc. This is used as input for the layout of powertrain, hot parts, fuel injection & hydraulics, scavenge system
- In an iterative process, with the above as input, the first dimensioning of the scavenge system can be made, which is in turn used as an input for the first engine outlines and hot parts. The combustion chamber is thence drafted in CAD
- Subsequently, CFD simulations of the scavenging and combustion processes are carried out for a single-cylinder geometry. The injection parameters[8] and injector geometry are optimized and an atomizer pre-selection is made in order to keep the combustion chamber temperatures within the required limits and optimize the SFC/NO_x trade-off
- Further, the Turbocharger (TC) allocation, auxiliary blower selection and Scavenge Air Cooler (SAC) calculations are done and optimised in an iterative process in parallel with the scavenge system design
- The latter is subsequently checked with a stress calculation, taking into account the temperature, pressure and weight of the parts
- After the design is finalised, the tuning is verified on the prototype test engine where appropriate pressure and temperature measurements are made. These measurements are used to check the process calculation and ensure that the boundary conditions have been correct.

The process simulation tools used throughout this process have been developed and maintained over decades within WinGD[9] and are specifically optimised for the application on large marine 2-stroke engines.

CONCEPT, DESIGN AND COMPONENT CHOICES

Throughout the thermodynamic & scavenging layout, the engine designer has several concept, design and component choices to make, some of the significant ones are listed below:

- **Maximum firing pressure:** generally higher firing pressures lead to higher efficiency at the expense of higher NO_x emissions and higher loading on crosshead and crank pin bearings, combustion chamber parts and engine structure
- **Compression ratio:** higher compression ratio leads to increased thermal efficiency, limited by heat and scavenging losses, tribology aspects and increasing NO_x emissions due to higher combustion temperatures
- **TC-selection:** The required volume flow of the TC is given by the scavenging demand, whereas the desired scavenging pressure results from the desired firing pressure, design criteria for pressure gradients and the given compression ratio. A range of TCs is thus selected to fulfil these requirements and optimize efficiency in different parts of the rating field
- **SAC layout:** The SAC is laid out for efficient cooling in order to minimise pressure losses and increase overall engine efficiency while minimising the volume for cost and space reasons. The scavenging air temperature is limited by the temperature of the available cooling fluids
- **Inlet port position & height:** A higher port height leads to higher scavenging efficiency but to lower effective compression ratio (need to open exhaust valve earlier) and hence reduction of engine work
- **Exhaust valve diameter:** a larger valve enhances the scavenging process whereas a smaller diameter reduces the hydraulic forces to open it and the cost of the expensive valve stem
- **Exhaust duct angle:** generally as small (parallel to liner) as possible to reduce scavenging back pressure, albeit a trade-off exists with overall engine height and position/stability of exhaust manifold
- **Exhaust valve and injection timing:** The exhaust valve and injection timing is used to control the combustion phasing and to adjust the effective compression ratio. Electronic engines offer the possibility of Miller timing and sequential injection to optimize efficiency and control emissions

[7] Whereby the engine is operated at same speed and firing pressure but lower MEP leading to higher overall efficiency and reduced friction losses.

[8] Extended studies also allow combined analysis of the injection and combustion parameters together with the scavenging process to optimize the inlet port shape since this highly influences the swirl motion at TDC, the heat transfer and also the scavenging efficiency.

[9] WinGD and predecessor companies.

POWERTRAIN & ENGINE STRUCTURE

CYLINDER DISTANCE

In slow-speed two-stroke engines the cylinder distance – and consequently engine length – are invariably determined by the powertrain layout, which takes place in parallel with the thermodynamic layout. The cylinder distance determination is central to the engine development since it gives the geometrical boundary conditions for most other design groups. Figure 5 below shows a cross section of the crankshaft and the cylinder distance, which comprises:

- The width of the crankpin bearing ($w_{\text{crankpin bearing}}$), which is mainly determined by the maximum firing pressure and minimum engine speed
- The width of the main bearing ($w_{\text{main bearing}}$), which is primarily laid out according to engine power, speed and rotating masses of crankshaft and connecting rod
- The shrink fit length, which depends on the required transmissible torque, web geometry, journal diameter and crankshaft material.

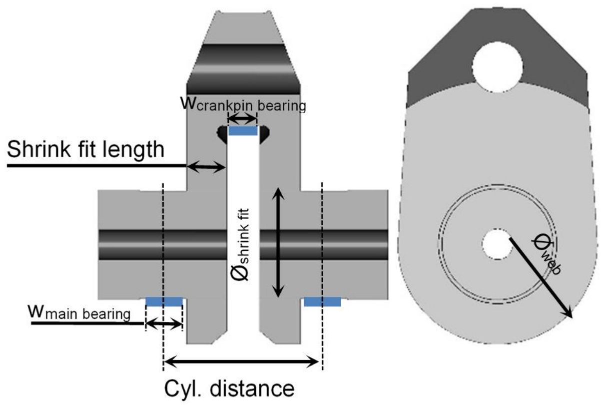


Figure 5: Cross-section of crankshaft throw; properties influencing the cylinder distance

The selected cylinder distance is checked against other possible influences, such as connecting rod thickness, crosshead bearing or cylinder cover arrangement (bolts, valves). Historically, two-stroke engines have had uniform cylinder distance, but modern engines such as the Wärtsilä-X92 feature variable cylinder distance[10].

DEVELOPMENT PROCESS

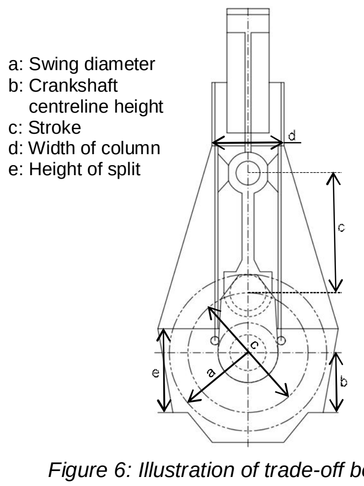
The development of the engine powertrain goes hand in hand with the engine structure, since the forces from the crankshaft influence the structure and in turn the behaviour of the structure influences the powertrain. A simplified description of the development process for the powertrain and engine structure is laid out in the following steps:

- Based on the bore, stroke, rating field and cylinder pressure specifications an initial estimation is made for width of main bearing, crank web (shrink fit length) and crankpin bearing, which gives the first cylinder distance
- In parallel, gas excitation files[11] for the new engine are created from the cylinder pressure curves at various points on the rating field
- Subsequently, with an assumed shaft line and propeller for a suitable marine installation, first dynamic calculations are carried out to evaluate the main powertrain dimensions
- A first stress calculation of the new engine structure (bedplate and column) can then be performed in order to determine the main structure dimensions
- If evaluation of dynamic behaviour for the crankshaft is accepted, then detailed layout and design of crank train are carried out
- Based on final crank train data and updated engine structure, the optimization of firing order for crankshaft in regards to free forces and moments and crankshaft vibrations is started. Results of Elasto-hydrodynamic[12] (EHD) calculations for the bearings are taken into account
- After determination of all crank train parameters, a final stress and strain calculation for the engine structure can be carried out

CONCEPT, DESIGN AND MATERIAL CHOICES

Throughout this development process the engine designer is confronted with numerous choices for design concepts, trade-offs and material choices, the most important of which are described below:

- **Engine height vs. width:** There exists a clear trade-off between crankshaft centreline height (which affects total engine height) and bedplate width for a given stroke, illustrated below:



A higher crankshaft centreline height (b) allows a narrower (lower part) bedplate width but results in a higher overall engine height. The bedplate side walls are often slanted in order to optimise the width at the bottom but still allow for enough space for the swing radius (a) at centreline height.

Figure 6: Illustration of trade-off between crankshaft centreline height and bedplate width

---

[10] Variable cylinder distance reduces the engine length by optimising the cylinder distance on the free end, taking advantage of the reduced torque that the crankshaft is carrying on that end. For more details refer to [2].

[11] Files containing the force vectors (magnitude, distance) on the crankshaft (from combustion) over time.

[12] EHD calculations are used to determine bearing load taking into account the forces, bearing hydrodynamic behaviour of the bearings and elasticity of the engine structure.

The total height of a modern 2-stroke marine Diesel engine is normally approximately 3.5 times engine stroke, and depends on: stroke, crankshaft centreline height, connecting rod and stuffing box design. The piston dismantling height is about 4.5 times the stroke.

- **Crankshaft:** Low speed 2-stroke engine crankshafts differ from crankshafts in other engine segments in two main characteristics: i) the semi-build construction due to limitations of casting and forging for such sizes and strokes, ii) the materials used. The crankshaft layout is first and foremost based on basic guidelines given by classification societies for semi-built crankshafts based on input parameters such as engine power and torque, torsional vibration limits, geometrical parameters such as shrink fit diameter, etc. There are two basic materials typically used for semi built crankshafts, namely a carbon steel (M60.6) and an alloyed steel (34CrNiMo6) with a higher yield strength.

- **Bedplate:** The 2-stroke marine Diesel engine mostly comprises a welded structure, which is preferred to cast structures for reasons of size and production volume. The most important design choice for the bedplate, the lower structure part, is single or double wall design. A single wall bedplate is more flexible to tilting of the main bearing housing, but introduces other challenges such as the fixing of the tie rods to the girder.

  An important design feature of the bedplate is the thrust bearing, laid out to withstand the vessel's thrust, which consists of a certain amount of pads arranged in an incomplete circle. The angle where the pads can be placed is limited by the flexibility of the structure, and the total bearing area is determined by the lining material used and the expected dynamic thrust. Importantly, the vertical position of the geometrical centre of the thrust bearing influences the tilting of the thrust bearing shaft and thus also the load distribution of the neighbouring main bearings as well as the intermediate bearings of the propeller shaft.

- **Column:** Similar to the bedplate, the column, which is "sandwiched" between the bedplate and the cylinder block, can be single or double walled. The single wall design, which is generally simpler to produce, is heavily influenced by the guide shoe lateral forces. An additional design choice is the fixation of the tie rods, which are four per cylinder for single wall column design and two per cylinder for double wall design (placement of tie rods between the walls).

- **Bearings:** The bearing forces are the main drivers of the stresses in the bearing, whereas conversely the stiffness of the housing is influencing the bearing behaviour. The interaction between bearing housing and the crankshaft is calculated by EHD calculation. Depending on the bearing location there are different influence parameters and challenges. First and foremost, the main bearing can be subject to high edge load such that a certain flexibility for tilting of the bearing housing is beneficial. On the other hand, crosshead and crank pin bearings do not face any inclination of the bearing pin but some other challenges such as low hydrodynamic speed and pin lift-off. For the crosshead bearing there are some additional restrictions caused by the space availability for part dismantling (e.g. guide shoe) and removal out of the engine (e.g. the maximum diameter of the crosshead pin is restricted by the width of the column doors)

- **Oil supply to piston & crosshead bearing**: The oil supply for the connecting rod bearings and for the piston cooling system is ensured on most Wärtsilä engines by the knee lever[13]. The design of the knee lever can accomodate a twin-circuit system, which permits the differentiation of the pressure level for piston cooling from crosshead bearing lubrication oil. This feature is increasingly important since higher firing pressures are needed to optimise SFC, which increase the load on the crosshead bearing above the limit for a given diameter. This limit is increased by applying high pressure[14] crosshead bearing oil supply.

<u>MANUFACTURING LIMITATIONS</u>

Due to the sheer engine size of the largest engine models (such as the Wärtsilä X92), manufacturers often come to size limitations with several components, such as, but not limited to:

- **Crankshaft size & weight:** foundry capacity (the crankshaft webs are free-forged out of one ingot, and capacity of the largest foundries is approximately 60 tonnes nowadays), rough and final machining lathe size (limited by the stroke), and crankshaft weight due to crane capacity limitation for crankshaft assembly at manufacturer and engine assembly at licensee

- **Bedplate size:** the welded two-stroke bedplate is Post-Weld Heat Treated (PWHT) to remove residual stresses from welding, creating a limitation for large engines due to the size of the PWHT oven

- **Column height:** the column height, which is determined by the engine stroke, is limited by the size of the Plano-miller for the milling of the crosshead guide rails (the counterpart of the guide shoe).

[13] The alternative design being the telescopic pipe, as applied for example on the Wärtsilä RT-flex50.

[14] Due to the hydrostatic effect from the higher pressure.

HOT PARTS & PISTON RUNNING

REQUIREMENTS

The requirements for the hot part components are diverse and often conflicting with each other. Component temperatures of 600 °C and above presuppose the correct material selection and a proper cooling. The thermal stresses, caused by the temperature gradients, have to be considered in the low-cycle behaviour analysis. The components around the combustion space are bolted together with elastic studs, which are pre-tensioned such that no leakage and no dynamic sliding between the components occurs. Finally, the level of dynamic stress in the different components, induced by the cylinder pressure, may not exceed the set limits.

One of the main contributors to the reliability of a two-stroke marine Diesel engine is the piston running behaviour, which is directly linked to the design of the hot parts. The standard measures applied on modern Wärtsilä engines have already been verified for many years. They include:

- Liner of the appropriate material
- Careful turning and deep plateau honing of the liner running surface for fast and trouble-free running-in
- Three chromium-ceramic coated, pre-profiled piston rings
- Chromium layer with ample thickness in the piston-ring grooves
- Anti-polishing ring at the top of the cylinder liner to scrape off deposits on the piston crown
- Removal of condensed water from the scavenge air by underslung and water separator
- Cylinder lubrication by the Wärtsilä Pulse Lubricating System (PLS)

DEVELOPMENT PROCESS

- In a first step the concepts (e.g. cylinder liner with or without bore-cooling) and main dimensions are defined based on existing similar engines with inputs such as stroke-to-bore, compression ratio, port geometry and cylinder pressure from thermodynamic layout
- The concepts are then verified and the dimensions defined in several iteration circles. With first input from the piston running experts, proposals are made for combustion chamber design, exhaust duct angle, cylinder jacket, piston height, piston cooling, etc
- Starting from the cylinder liner and cover only, quasi-static stress calculations with forces and temperature influence are made and gradually refined to include the surrounding components such as cylinder jacket and exhaust valve. An example of a such a calculation model is shown in figure 7:

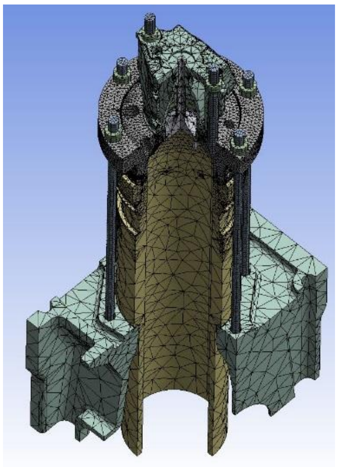


Figure 7: Finite Element (FE) model of the combustion chamber components, i.e. hot parts

- The stress calculations are iterated with CFD calculations for cooling efficiency of the parts
- Stress and hydrodynamic (piston rings) calculations of are performed on piston running components
- The layout of the entire exhaust valve drive is done in parallel with the hydraulic layout to determine the dimensions of the actuator piston and the maximal hydraulic pressures, which are used for the mechanical calculations of the actuator housing. The valve cage is optimised taking into account mechanical loads and deformations by means of FE-calculation
- As a final step, the hot parts are checked for compatibility with the surrounding engine structure components

CONCEPT, DESIGN AND MATERIAL CHOICES

- **Cylinder liner & cover:** The uppermost part of the cylinder liner is highly thermally and mechanically loaded since it is directly exposed to the combustion. The required cooling effect and part rigidity can be achieved by a thick collar with a row of cooling bores and water as coolant. Only the smallest engines have a more simple cylindrical cooling water space. The figure below illustrates the two cooling principles.

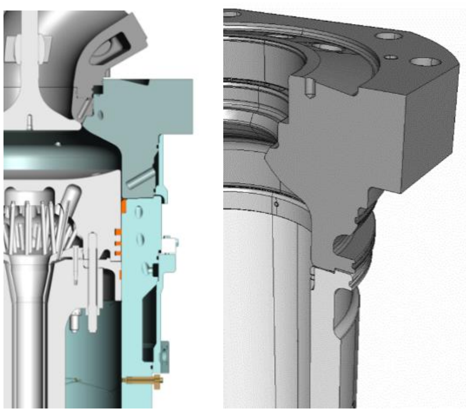


Figure 8: Upper part of cylinder liner with bore cooling (left) and cylindrical cooling water space (right)

The cylinder liners are made from a grey cast iron alloy which offers the best compromise between manufacturability, piston running behaviour and mechanical properties.

The free-standing cylinder liner is „sandwiched" between the cylinder block and the cylinder cover by the cylinder cover studs. The bolts are pre-tensioned so that most of the dynamic stress induced by the cylinder pressure is experienced as a relaxation in the compression of the cylinder liner instead of a strain on the studs. The pre-tension force of the cylinder cover studs has to be determined very carefully as the cylinder cover acts as relief valve of the combustion space: it needs to be tight during normal operation but has to lift-off before major engine components (piston or connecting rod) are damaged due to excessive cylinder pressures (e.g. >130% of layout pressure). Additionally no dynamic sliding is allowed between the cylinder cover and liner.

- **Piston:** Finding a proper cooling is one of the main challenges for the piston designer. With the so-called jet-shaker cooling principle (see Figure 9 below), which is applied on most Wärtsilä engines, it is possible to keep the temperatures below the limit on the combustion space side as well as in the cooling bores. Too high temperatures at the piston top would lead to material losses due to corrosive attack from the combustion gases. Excessive temperatures in the cooling space can lead to carbon deposit build-up with negative consequences to the cooling efficiency. For a reliable engine operation, it is also necessary to keep the temperatures around the piston rings within a certain range.

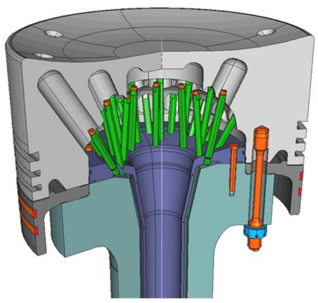


Figure 9: W-X62 section, showing the three piston rings and cooling concept

The piston is exposed to pressures up to 200 bar, so it is designed such that the stresses and deformations caused by the combustion forces remain within the limits. The main piston components: piston head, skirt and rod including their bolting have to be designed very carefully to avoid any dynamic sliding at the joint faces.

- **Exhaust Valve:** The diameter of the exhaust valve is one of the first parameters that is determined for a new engine type as it is significant for the thermodynamic layout. Today the diameter of the valve disc is typically in the range of 0.5 x bore diameter; a good compromise between engine performance, manufacturing cost and power demand for the valve drive.

The temperatures of the exhaust valve spindle typically reach more than 600 °C since there is no cooling of the spindle itself. Excessive material loss on the combustion space due to high temperature corrosion can only be prevented by the use of high-alloy steels (including high chromium content) or equivalent welded protection layers (seen mostly on recent engines). By employing a careful design of the exhaust valve spindle and seat, the relative movement between the two components under combustion pressure can be limited, else fretting would lead sooner or later to serious damages on both components.

- **Cylinder Lubrication System:** As described earlier in this chapter the cylinder lubricating system is one of the contributing factors to a reliable piston running behaviour. It is designed to bring the right amount of cylinder oil at the right time to the piston rings and the cylinder liner surface. The target is to achieve a satisfactory piston running behaviour at the lowest possible oil feed rate since the costs of the oil are a considerable part of the engine operating costs[15].

The cylinder oil is injected by four to ten – depending on the engine size – lubricating quills to the cylinder; about one third above the piston, the rest either in the piston rings or below the piston. This distribution pattern as well as the position of the quills is carefully selected to achieve the best possible oil distribution. The oil is provided to the quills by means of an electronically controlled, piston-type cylinder lubricating pump[16].

[15] Typically the cylinder lubricating oil is regarded as a maintenance cost (as opposed to operating cost), which is paid by the vessel owner, instead of the charterer. Thus cylinder oil consumption is often at least as important as fuel consumption in the choice of vessel and main engine (which is made by the owner).

[16] More information on Wärtsilä cylinder lubrication system in [2].

FUEL INJECTION AND EXHAUST VALVE ACTUATION SYSTEMS

SYSTEM REQUIREMENTS

The fuel injection and exhaust valve actuation systems are core ancillary engine systems responsible for optimising the engine operation in variable conditions of load and speed. The major requirements for these systems on the modern two-stroke marine engine are:

- Fully variable (timing and pressure), load-independent fuel injection
- Accuracy (cylinder-to-cylinder, injector-to-injector)
- Very low, stable injection quantities
- Multi-fuel & multiple injection capability
- Fully variable, load independent valve actuation
- Serviceability, initial cost, maintenance cost

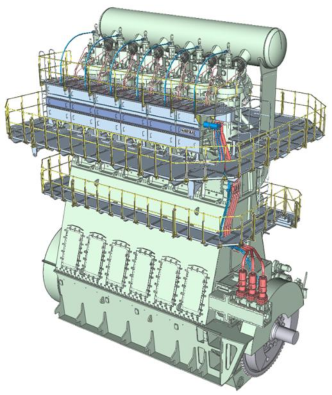


Figure 10: The fuel injection (red) and exhaust valve actuation (blue) systems highlighted on a 6 cylinder W-X62 engine

The above requirements present several important trade-offs that are balanced differently with different system architectures. The current changing market trends with very low fuel oil prices and very high uncertainty in the dominant future technologies for fuel consumption optimisation and emission abatement, make the flexibility and the multi-fuel capability of the systems key requirements for the future.

SYSTEM ARCHITECTURES

The available fuel injection system architectures for the engine designer range from pump-line-nozzle systems to advanced fully-flexible Common Rail (CR) systems [3]. Since flexibility and fully variability in timing and injection pressure are key requirements, CR systems are the obvious single alternative for modern two-stroke marine Diesel engines. WinGD (previously Wärtsilä Switzerland) has been developing CR systems since the late ‘90s, with the first engine equipped with an Injection Control Unit (ICU) in 2000. This system features an engine-driven high-pressure fuel pump unit, and an ICU located on the high pressure fuel rail feeding a spring-loaded injector on the cylinder cover.

Since then WinGD has been developing the next generation of CR systems based on the same HP fuel pump concept combined with a more advanced fuel-actuated CR injector, in order to bring the injection control closer to the cylinder cover. By doing so, the volume of high pressure fuel between the switching needle and the injector tip is minimised, enhancing the accuracy and multiple injection capability of the system, useful for BSFC/NOx optimisations and engine thermal management. An overview of the fuel injection system developments in the last 30 years can be seen below:

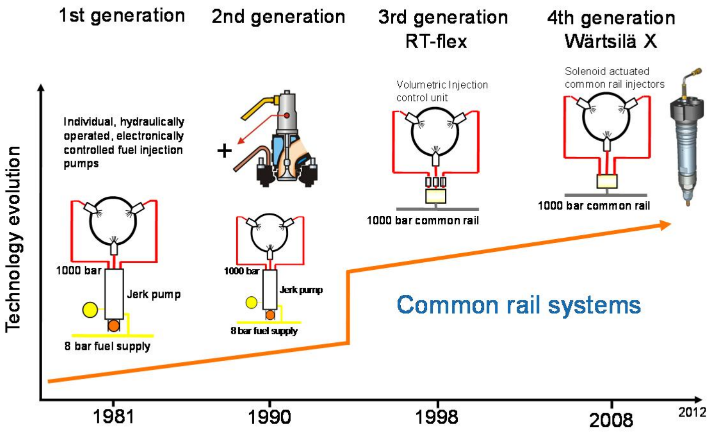


Figure 11: Fuel injection system architectures developed throughout the past 30 Years for (Sulzer and later) Wärtsilä engines

Modern Wärtsilä engines feature also a compact, powerful and cost efficient exhaust valve actuation system (servo oil pumps, high pressure pipes, valves, exhaust valve control unit). With an operating pressure of 200-300bar supplied by engine-driven servo oil pumps, this common rail system offers the designer full flexibility in exhaust valve timing, giving the possibility to offer several different tunings for customised applications.

SYSTEM DEVELOPMENT PROCESS

Fuel injection and exhaust valve actuation system development is typically highly validation-intensive with rigorous component and system rig testing and field testing before the validated parts can enter serial production. The main development steps are outlined below:

- Concept & architecture decisions are made with basic inputs such as power, speed, injection & valve timing, cylinder pressure, valve diameter & opening time and system flexibility
- First designs of the supply unit, rail unit, injection and valve drive are made taking into account geometrical considerations given from powertrains, structure, hot parts, etc
- Hydraulic and stress calculations are used to check the first designs
- Component and sub-system validation (functionality, endurance) on rigs and field engines for several thousand hours. Iterations of the design are made as needed
- Multi-disciplinary design reviews with hot parts, scavenge system and design finalisation
- Testing on prototype engine and final optimisation of system and component design

ENGINE CONTROL

BASIC REQUIREMENTS

The functional requirements for the Engine Control System (ECS) are dictated by thermodynamic and other (mechanical, tribological) functional requirements. Apart from several core functions which every ECS has to perform for the electronically controlled reciprocating engine, the ECS of the marine two-stroke low-speed engine has to fulfil additional requirements:

- Compliance with environmental and Classification Society rules
- A single ECS for both Diesel and DF engines
- Modular hardware layout allowing prefabrication of engine sub-assemblies (e.g. Rail Unit)
- Simple and cost efficient hardware installation for the engine builders and Shipyards
- User friendliness for commissioning personnel as well as for the Operators/Crews
- Local and remote diagnostic possibilities
- Redundancy and electromagnetic compatibility
- Low power consumption
- Low cost, long lifetime, retrofitability and a thoroughly planned obsolesce concept

ENGINE CONTROL SYSTEM SOFTWARE ARCHITECTURE

The engine control system software of a marine Diesel engine comprises four elements:

1. **System/platform software:** the interface between the engine hardware[17] and the application software
2. **Application software:** software controlling core engine functions
3. **Application parameters:** the engine-dependent values embedded in the application software
4. **User interfaces and software tools:** interfaces between engine and user and other software tools

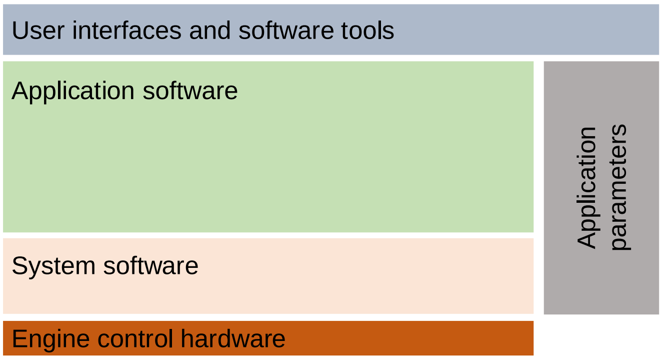


Figure 12: Architecture of the engine control system software

DEVELOPMENT PROCESS: FROM REQUIREMENTS TO SOFTWARE

The development process for control system software is carried out according to the so called V-process. The following graph visualises relations and dependencies between development steps from requirements gathering and formulation to final validation:

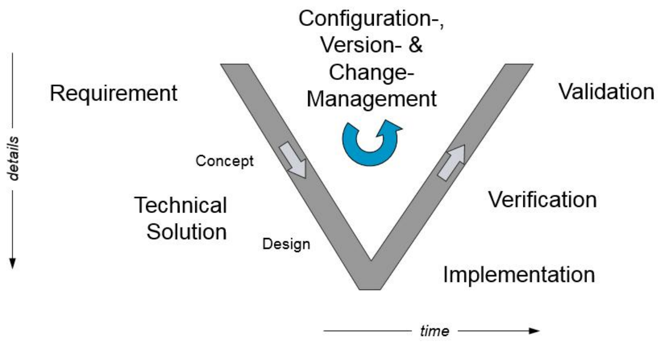


Fig. 13: Illustration of the V-process of software development

The process will be explained in more detail in the steps below, referring to the abovementioned four elements.

1. **System / platform software development**: the platform software development starts with requirements analysis and processes' definition. Particulars such as sampling rates (e.g. crank angle 1kHz), task rates (e.g. 100Hz starting air pilot valve control) and transmission priorities (e.g. highest priority on CAN-Bus is fuel command due to speed regulation) are decided at this stage for all co-existing applications and internal level processes. A wide range of parameters and dynamic behaviours have to be considered, such as engine speeds, acceleration/deceleration ratios, etc, since a single ECS covers an entire portfolio of engines.

   Initial tests and software performance optimisations are performed on the laboratory simulators. When µControllers, processors and transmission busload are optimised and software operations are validated the next level of testing takes place on the test engine and later on the production engines.

2. **Application software:** The application software is the core knowledge of the engine developer. The development process starts with requirements definition; based on this the definition of the control concept is started and the design is done in a Simulink model. This model is then optimised through several iterations and tests in Simulink environment. After compilation to the processor / µController code, the optimised Simulink model is tested on the laboratory simulator on the real hardware control modules. After successful testing on the simulator, the application is subject to final

[17] A differentiation should be made between engine hardware (e.g. fuel injectors, exhaust valve, etc) and engine control hardware (cylinder control modules, etc). Engine control hardware is typically not the core scope of the engine developer, as is the case in WinGD, and is, at least partly, outsourced to third parties.

validation on the test engine and later on the production engine.

3. **Parameters definition:** This element begins with process definition, including Simulink modelling and establishing a theoretically safe range of parameters for the particular process and application (e.g. overlap of fuel injection with exhaust valve opening not allowed). Subsequently, the first attempt to classify parameters into pre-defined groups takes place. Parameters are grouped in regards to i) their nature: engine (bore) size-specific, engine cylinder configuration-specific, and installation (i.e. tuning) specific, and ii) their accessibility: expert level editable, commissioning level editable, operator level editable.

   The initial setup of parameters is then cross-checked on the laboratory simulator and later optimised on the test engine where safe-ranging (allowed for certain level of accessibility) is finally validated. Final grouping of parameters is reviewed with a commissioning team and operational experts who contribute to the creation of commissioning procedures.

4. **Interfaces and software tools development:** The development of the operator's interface is based on available functions of the Graphical User Interface (GUI) and particular needs of an engine (e.g. a DF engine requires a much more complex interface than a Diesel engine). Particularly important for this element is classification rules, ergonomics and considerations about marine practice. Commissioning and debugging tools are developed and reviewed with relevant software experts as well as operational experts experienced in field activities. Finally, comprehensive manuals are created.

MODERN CONTROL SYSTEM HARDWARE ARCHITECTURE: PERCULIARITIES OVER OTHER INDUSTRIAL ENGINES

Due to the nature of its application as a single prime mover and the corresponding classification requirements[18], a marine engine has to demonstrate high redundancy and reliability; features that greatly affect the architecture of its ECS. A high level of redundancy is, however, accompanied by high complexity and cost; therein lies the main trade off in choice of engine control system architecture. The two extremes are split Intelligence Systems (SIS) and Centralised Intelligence Systems (CIS), examples of which are shown below.

The SIS is characterised by the absence of a particular hardware module[19] which plays a role of the main computational and communication interface. All modules are equal and are processing and controlling assigned own cylinder functions (e.g. fuel injection) and common tasks (e.g. fuel pressure regulation, external communication, etc).

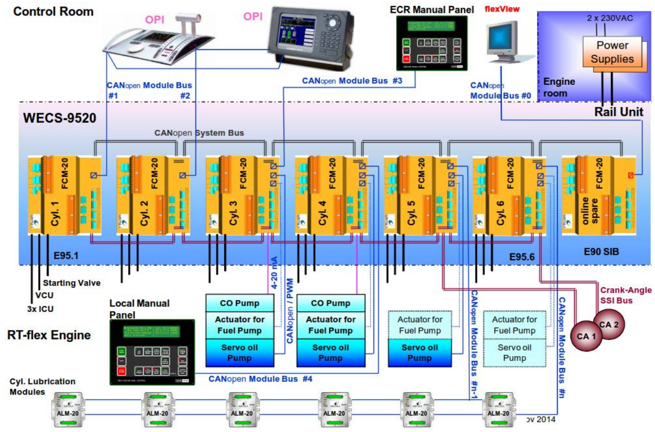


Figure 14: Example of a split intelligence control system (WECS-9520 on Wärtsilä 2-stroke engine)

In CIS there are at least two different type of hardware modules: main- and cylinder-control modules. Typically (for redundancy reasons), there are two main modules which are acting communication masters for internal Bus communication and as interfaces to external systems. Additionally these two modules store the software package for the entire system, process and sometimes also control common regulation tasks for the entire engine.

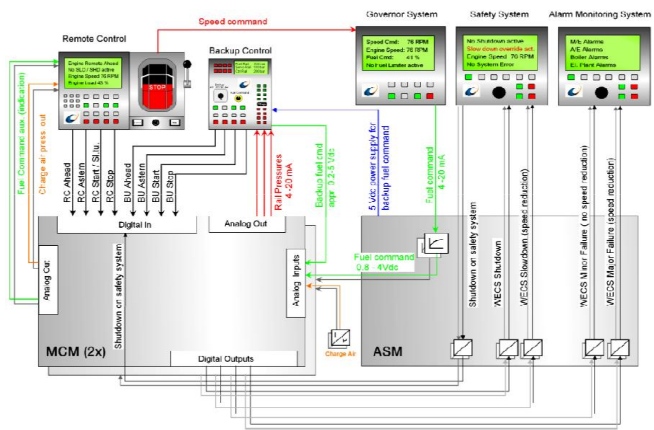


Figure 15: Example of a centralised intelligence control system (WECS-9500 on Wärtsilä 2-stroke engine)

Overall, SIS offer very good redundancy, however that is often at the expense of hardware utilisation[20]. On the other hand, CIS[21] systems usually offer better utilisation of hardware but require more data to be transmitted across the control system. Appropriate combination of specific features of these two architectures (and other, such as commonality of modules, obsolescence management, etc) provides a good base for achieving competitive cost level for a specified redundancy level.

[18] (e.g. in the event of certain severe but non-critical failures, a single prime mover will go into slowdown rather than shutdown on a multi prime mover vessel)

[19] Modules are equally important in the sense of functionality and each function is carried by at least two modules. Common functions are distributed to cylinder modules as equally as possible.

[20] Hardware utilisation is low for split intelligence systems because the functions are distributed for redundancy but only few modules are fully utilised.

[21] Where functionalities are concentrated and common functions are controlled by central modules.

EMISSION CONTROL CONCEPTS: EFFECTS TO BASIC ENGINE DESIGN

Modern two-stroke marine Diesel engines need to comply with the emission legislation of the International Maritime Organization (IMO). Regulations comprise limits for the emission for NO_x and Sulphur Oxides (SO_x) globally and more stringent limits inside Emission Control Areas (ECA). A detailed description of the regulatory framework can be found in Annex VI of MARPOL 73/78, Regulation 13 and 14 and in several publications [4].

SO_x emissions, for a given fuel sulphur content, can be controlled by exhaust gas after-treatment based on scrubber technology, which has limited impact on the engine design. Alternatively, low sulphur fuels can be used. The increasing use of low sulphur fuels dictates an adaptation of the fuel injection, cylinder lubrication and the engine control systems to take into account different evaporation, ignition or lubrication characteristics of the fuel type.

The Tier II NO_x emission limits[22], which came into force in 2011, can be fulfilled with optimisation of electronically controlled engines. By adjusting the scavenging air pressure, the fuel injection parameters and the exhaust valve timing, the NO_x limitation can be fulfilled without significant increase in SFC. On the other hand, the Tier III NO_x legislation can hardly be achieved only by adjusting the combustion and injection-relevant control parameters without a substantial fuel consumption penalty. More economical solutions are exhaust gas after treatment (SCR), other internal measures (EGR) or the WinGD low-pressure Dual-Fuel (DF) technology, the effects of which to the basic engine design will be described below.

SELECTIVE CATALYTIC REDUCTION (SCR)

Currently, WinGD's preferred solution for Diesel engines to comply with Tier III legislation is exhaust gas after-treatment based on SCR technology since it is technically the most advanced, proven and currently most cost efficient solution. The SCR reactor can either be mounted upstream or downstream of the turbine. In the Low-Pressure (LP) SCR system, the reactor is placed downstream of the turbine such that only minor modifications are required on the basic Diesel engine. However, the LP SCR has the disadvantage of the lower temperatures, pressure and density of the exhaust gases at the catalyst (which thus requires a different composition and/or heating of the exhaust gas). This leads to large reactor volumes and the risk of formation of ammonia sulphates, in particular for operation with high-sulphur fuels, since the temperatures are mainly below the dew point of ammonia sulphates.

The High-Pressure (HP) SCR [4] requires more significant modifications on the engine since the reactor is located between exhaust gas receiver and turbocharger. In this configuration, higher temperatures before the TC reduce the efforts on the engine process and control to keep the temperatures above the dew point of ammonia bi-sulphate. Furthermore, the higher density of the gas allows a more compact arrangement of the SCR system and reduced volumes of the reactor and the mixing devices.

The SCR system can be bypassed in order to enable the operator to switch between Tier II and Tier III modes when entering or leaving ECA. The required piping and the flow control valves are integrated in the engine design and supported by the engine structure, but the system itself is designed to be supported by the engine room walls and surroundings. The valves act as the interface between SCR and engine system, such that the vast forces induced due to the high pressure and the momentum change from flow redirections have to be considered in the engine structure as well as on ship hull side. Additionally, the SCR reactor thermal capacity needs to be considered during transient operation as well as in the heating up phase to guarantee stable operation of the engine.

WinGD is currently also in the later stages of development of a fully integrated HP-SCR system, whereby the catalyst and all reactant injection and mixing equipment and all flow control valves are integrated on the engine. This configuration presents obvious space and arrangement challenges, since the reactor and mixing pipe need to be extremely compact in order to be placed below the exhaust manifold. Figure 16 below shows such a system on a 6RT-flex50B engine:

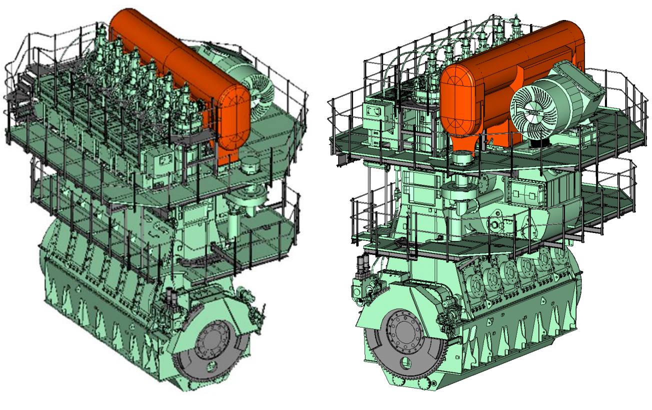


Figure 16: Integrated HP-SCR on a 6RT-flex50B

[22] Global NO_x limit at 14.4 g/kWh weighted average (engine speed dependent above 130rpm).

EXHAUST GAS RECIRCULATION (EGR)

Another technology used by WinGD to decrease the NO_x emissions is High Pressure Exhaust Gas Recirculation (HP-EGR). By recirculating exhaust gas into the combustion chamber, the heat capacity of the gas in the cylinder is increased and the oxygen concentration is reduced, thus reducing combustion temperatures and NO_x formation. In order to reduce corrosion, mechanical damage and deposit formation, the exhaust gas has to be cleaned before it is mixed with the fresh charging air. Besides the exhaust gas cleaning device (scrubber), a cooler, a blower (to overcome the pressure gradient) and a water separator are required on the engine, such that these components can be integrated into the engine design and structure. Apart from the components installed on the engine, complex waste water and sludge treatment devices need to be integrated into the vessel system.

LOW PRESSURE DUAL FUEL (DF) ENGINE

Despite the recent drop in liquid fuel prices, the increasing availability of gaseous fuels has made the latter attractive for application in the marine sector, even beyond the LNG carrier sector. On this basis, WinGD has decided to utilise their long-standing experience in the field of low-pressure dual-fuel technology on 4-stroke engines to develop a new product range of 2-stroke dual-fuel engines. These engines do not only enable the use of gaseous fuels but they are inherently compliant with the most stringent future emissions standards: NO_x, SO_x and particulate matter emissions.

The low pressure DF engine development has presented additional challenges to WinGD which have been duly overcome, such as:

- Stable pre-mixed combustion control, influenced mainly by mixing, ignition and a lower geometric compression ratio required to avoid knocking
- Higher maximum combustion pressures compared to the Diesel layout in order to increase the thermodynamic efficiency which, though, can be used to further optimise Diesel SFC
- Accurate and reliable operation of gas admission systems and valves, which have been validated thoroughly on test rigs
- A CR pilot fuel system, which is independent of the main fuel system. The pilot fuel injector is integrated into a pre-chamber which is used to achieve a stable combustion and low unburned Total Hydrocarbon (THC) emission

By carefully integrating the above measures into the basic engine design, WinGD is able to offer its entire Diesel portfolio as DF-ready engines without major changes to the basic Diesel engine itself. Refer to [5] for more details.

THE CHOICE OF EMISSION CONTROL CONCEPT

The choice of the right technology depends on several parameters such as the vessel type, trade routes (time share spent in and out of ECA), scenarios of fuel and urea price development and expected future industry trends. Whilst for LNG carriers the low-pressure DF engines are the obvious choice, for other vessel types equipped with Diesel engines the choice is by far more complicated. SCR and EGR systems are used primarily to achieve Tier III emissions limits; however both solutions can also be operated outside ECA to enhance the efficiency of the engine in Tier II mode. Additionally, DF engines can be used for vessels other than LNG carriers running in gas mode within ECA and on liquid fuel outside ECA.

SCR systems require certain temperature levels to avoid the formation of deposits and to ensure an efficient conversion of NO_x, hence the tuning for Tier III mode is adapted in such a way that the exhaust temperatures are above the required limits. In order to optimise the engine operating cost, the engine is tuned to lower SFC and higher engine-out NO_x emissions, which are in turn reduced by the SCR to meet Tier III legislation. SCR systems are generally suited better to smaller engines, where a single catalyst of a moderate size can be used. There exist technical solutions for very large catalysts and also for multiple catalyst configurations, but these introduce challenges such as structural and control issues respectively.

EGR leads, in principle, to a decreased scavenging efficiency of the engine. Nonetheless, the recirculation of residual gas also allows different tunings of the engine towards lower fuel consumption to compensate the scavenging effect of the EGR. EGR systems are generally better suited to large engines since the complexity of the scrubbing system may not be particularly suited to small engines. A simple version of EGR can also be used to increase power output and/or decrease SFC in DF engines, where no scrubber is needed as long as the EGR is only running in gas mode.

The choice of technology also depends markedly on the standpoint. For the vessel owner the initial and operating costs and the reliability are most relevant. Conversely, for the engine builder, the focus is on manufacturability, ease of assembly and initial cost including material and testing. For the shipyard, the best solution is one placed entirely on the engine so that both costs and technical responsibility are taken by the engine maker. Finally, the engine designer strives to achieve the best compromise of the above demands.

CONCLUSIONS

THE ENGINE DEVELOPMENT PROCESS

As it has been described in the chapters above, the low-speed marine two-stroke engine development process is characterised by a few particularities such as decentralised production, the sheer size of the engines, broad use of HFO and the volatility of the shipping business as a whole, which constitute important boundaries for the engine designer. Still, overall the process follows a few familiar steps found in the development of any product:

- Assumptions
- First designs
- Calculations
- Detailed design
- Iterations
- Testing
- Final verification/prototype testing
- Endurance testing

Throughout the development, the engine designer is presented with numerous concept, design and material choices, which need to be decided upon based on input from all customers: the vessel owner, who ultimately chooses the vessel and frequently influences the engine type, the shipyard, which specifies the engine type and configuration and the engine builder (licensee), who procures/manufactures the parts and assembles the engine. The most basic choices are concentrated in the thermodynamic layout and the layout of the powertrain, which determine the size and design of the combustion chamber and the cylinder distance respectively, affecting consequently the whole engine. Subsequently, the engine structure concept is chosen and the design is iterated and refined in steps with all peripheral parts such as hot parts, scavenge system, fuel injection and hydraulics, and ancillary systems such as platforms, piping and electrical components. The whole engine is then subject to several checks, such as vibrations analysis, component and system tests, before a prototype is built and tested on test bed and later in a vessel. The result is today's modern two-stroke Diesel engine.

THE NEXT GENERATION TWO-STROKE ENGINES

The major trend shaping today's marine engine business and the requirements for the next generation of engines is the development of the fuel and gas prices. With crude oil prices at record low levels and no significant indication of a reversal of this trend, there is likely to be less imminent pressure on fuel consumption and multi-fuel compatibility in the near future. However these requirements are still expected to remain a medium to long term trend. The second major influence are the climate change initiatives. At several instances in the recent past, leading to the United Nations conference on climate change in Paris in November 2015, it has been made increasingly clear that more stringent measures need to be taken to decelerate global warming, a tendency that is bound to affect the shipping industry with more stringent emissions legislation in the near future.

For these reasons, the next generation of marine Diesel engines are likely to feature, among other:

- Multi-fuel capability
- Higher fuel injection pressures
- Higher firing pressures
- Combined emission control mechanisms integrated on the engine
- Condition Based Maintenance (CBM)
- Big Data: vessel/engine variables acquisition, transmission and statistical processing in order to optimise vessel performance
- Remote data connection allowing remote problems diagnostics, software download and engine performance optimisation

The rate of adaptation of such technologies, and consequently the rate of evolution of marine two-stroke engines, is dictated by trends in the marine business (market pull) and technology developments in other segments which can be adopted on marine engines (technology push). In any case, the modern two-stroke Diesel engine, in a form akin to the current state-of-the-art design (see Figure 17 below), is very likely to survive the challenges of the next decades and remain the prime mover of most cargo vessels in the world in the long term.

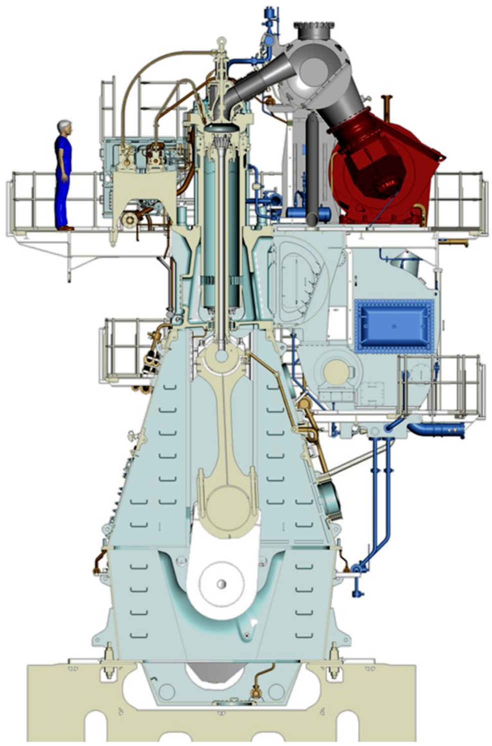


Figure 17: State-of-the-art Wärtsilä X72 Diesel Engine

NOMENCLATURE

BMEP	Brake Mean Effective Pressure
CAPEX	Capital Expenditure
CAD	Computer Aided Design
CBM	Condition Based Maintenance
CFD	Computational Fluid Dynamics
CHB	Cross Head Bearing
CIS	Centralised Intelligence Systems
CPB	Crank Pin Bearing
CR	Common Rail
DF	Dual Fuel
DWI	Direct Water Injection
ECA	Emission Control Areas
ECS	Engine Control System
EEDI	Energy Efficiency Design Index
EGR	Exhaust Gas Recirculation
FAST	Fuel Actuated Sackless Technology
GUI	Graphical User Interface
HP	High Pressure
IACS	International Association of Classification Societies
IMO	International Maritime Organisation
MB	Main Bearing
NOx	Notrogen Oxides
OPEX	Operating Expenditure
PDH	Piston Dismantling Height
PLS	Pulse Lubricating System
SAR	Scavenge Air Receiver
SCR	Selective Catalytic Reduction
SFC	Specific Fuel Consumption
SO_x	Sulphur Oxydes
VVT	Variable (exhaust) Valve Timing
VIT	Variable Injection Timing
SAC	Scavenge Air Cooler
SIS	Split Intelligence Systems
TEU	Twenty-foot Equivalent Unit
TC	Turbocharger
THC	Total Hydrocarbon
TCO	Total Cost of Ownership
TDC	Top Dead Centre
WinGD	Winterthur Gas & Diesel

REFERENCES

[1] PARRAVACINI M. “Determination of the optimum stroke-to-bore ratio of a large two-stroke engine using simple models”, Thesis, ETH Zürich, 2015.

[2] SPAHNI M., KYRTATOS A., DE JONG R. “The New Generation of Low-Speed Engines from Wärtsilä" CIMAC, 2013.

[3] BOLETIS E., KYRTATOS A., DE JONG R. “A new fuel injection and exhaust valve actuation system for a two- stroke engine family in the 30 to 50 cm bore segment " CIMAC, 2010.

[4] SANDELIN K., PEITZ D., “SCR under pressure - pre-turbocharger SCR for marine 2-stroke diesel engines“ CIMAC, 2016.

[5] OTT M., “The 2-stroke low-pressure Dual-Fuel technology: from concept to reality “ CIMAC, 2016.

BIBLIOGRAPHY

‘THE NEW WÄRTSILÄ 820 MM-BORE ENGINE SERIES – ADVANCED DESIGN AND FIRST SERVICE EXPERIENCE’, Marc Spahni, Heinrich Brunner and Ronald de Jong, CIMAC 2010, Bergen.

CONTACT

Andreas Kyrtatos: General Manager, Project Management Office, R&D, WinGD.
andreas.kyrtatos@wingd.com

Marc Spahni: General Manager, Engine Design, R&D, WinGD.
marc.spahni@wingd.com

Sebastian Hensel: General Manager, Engine Performance, R&D, WinGD.
sebastian.hensel@wingd.com

Gregory Sudwoj: General Manager Technical Experts, Operations, R&D.
gregory.sudwoj@wingd.com

Reto Züger: Development Expert Powertrains, R&D, WinGD.
reto.zuger@wingd.com