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Master of Science in Automotive Engineering

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Review of the certification process, homologation testing and inspection for a diesel marine engine.

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Confidentiality Agreement

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1 INTRODUCTION

The thesis project was developed as part of an internship held at the FPT Industrial research and development center located in Turin, Italy. The objective was to review the naval standards and directives for marine diesel engines and to follow the activities related to the certification process of a new developed marine diesel engine which include the homologation tests and the engine inspection. The commercialization of marine engines must be accompanied by a certification procedure according to a classification society that in this particular case was the RINA (Registro Italiano Navale). The certification procedure must be followed for marine applications in order to assure the correct engine design and construction to attain a sound fabrication and the safety of life at sea.

The studied engine is based on an off-road application and thus, a marinization process was followed to apply modifications and develop specific components in order to adapt the base engine for its use in marine applications considering the harsh conditions of a marine environment, the new operating conditions, and the more stringent safety and environmental regulations.

An analysis of the naval classification standard was done for controlling the impact of the regulations on the engine design and materials used for its fabrication. Then, a theoretical analysis of the propulsion and engine matching for marine applications was followed. The parameters that have an effect on the engine performance, such as the vessel resistance, type of hull and engine – propeller matching were studied. Also, a review of the main systems present in the engine test cell for the tests development and for the data acquisition, including the software for the test bench and engine control was done.

As part of the certification process, different homologation tests should be followed on the evaluated engine. These tests subject the engine to demanding and to fast load variation working conditions to evaluate the durability, fatigue and correct operation of the engine. During all the tests, the monitoring of different parameters such as temperatures, pressures and speeds of engine fluids and main components was done through the use of alarms and indications established according to the threshold values for attaining a safe and sound engine operating conditions and as required in the RINA regulations. Additional tests for verifying the lowest propulsion engine speed which controls the engine operation during the vessel/boat departure and for checking the compliance of the engine speed governor were performed.

Before the official certification tests, different experiments simulating the same engine homologating working conditions were performed in order to verify the correct functioning of the engine, the sensors and all the systems present in the test cell. During this step and as part of the followed engine tuning, the injection characteristic curve was modified, the ECU dataset was updated and the impact of these changes on the engine operation was studied.

Before the official certification tests, different experiments simulating the same engine homologating working conditions were performed in order to verify the correct functioning of the engine, the sensors and all the systems present in the test cell. During this step, the fuel delivery characteristic curve present in the base engine ECU's dataset was modified to increase the engine's maximum delivered power and as a consequence, to be able to develop the power levels requested for all the engine operating points of the homologation tests of the marine regulations. The effect of these more demanding engine operating conditions was studied, evaluating the new reached levels of different engine parameters such as start of Injection, black smoke, rail pressure and fluids temperatures and pressures; always considering the threshold values for attaining a safe and sound engine operation, avoiding any mechanical damage, material degradation or fatigue problems.

Finally, after the homologation tests, the engine was disassembled and tightening torque checks followed by a visual inspection on the main components was held to control for possible overload working conditions, fatigue issues or premature wear not allowed by the RINA regulations.

2 RINA (REGISTRO ITALIANO NAVALE) ENGINE CERTIFICATION

RINA – Registro Italiano Navale – group is a global corporation that offers assessment, control, certification, rule-making and research services in Energy, Marine, Transport and Infrastructure and Industry sectors. RINA is one of the founding members of the IACS (International Association of Classification Societies) association consisting on twelve member classification societies covering the 90% of world's cargo carrying tonnage [1] dedicated to maritime, and ships safety, and clean seas by the classification design, construction and through-life compliance Rules and standards set by the member Societies. Marine services have been the core of RINA's business since its inception representing the highest turnover percentage [2] demonstrating broad experience in the marine field.

2.1 General Requirements

RINA stipulates some requirements for the design and construction of marine diesel engines to attain a sound fabrication and the safety of life at sea.

2.1.1 Hot Surfaces and Fire Protection

Hot machinery surfaces having temperatures exceeding 220 °C, as turbochargers and exhaust pipes, are to be suitably insulated or protected in order to prevent ignition of combustible materials in case of contact.

2.1.2 Safety on Moving Parts

Moving parts such as, but not limited to, flywheels, couplings, belts, etc. are to be protected with screen or other suitable devices to prevent injuries to personnel.

2.1.3 Fuel Systems

- The fuel systems for propulsion machinery are to be fitted and arranged with filters that ensure uninterrupted supply of filtered fuel during cleaning operations of the filter equipment.

- All the external high pressure fuel lines between the high pressure fuel pumps and the fuel injectors are to be protected with a shielded piping system capable of containing fuel leakage coming from a possible high pressure line failure. Means for collecting the fuel leaked must be provided and an alarm is to be given in the event of a fuel line failure.

2.1.4 Lubricating Oil Systems

- Oil filter arrangements are to be provided so that the filter equipment can be cleaned without the interruption of the oil flow to the engine.

- The lubricating oil system shall be fitted with alarms that give audible and visual warning in the event of an appreciable reduction in pressure of the lubricating oil supply.

2.1.5 Safety Devices

- If engine overspeeding risk exists, means are to be provided in order to ensure that the engine safe speed is not exceeded. Therefore, a speed governor is to be fitted to the engine and adjusted so that the engine is automatically controlled for not exceeding the rated speed by more than 15%.

- Automatic shut-off arrangements shall be installed and act in case of failures, such as lubricating oil supply failure, which could rapidly lead to complete engine damage.

2.1.6 Communications

At least two means for communicating orders to the engine shall be available, that is, the regularly (main) installed boat instrument panel and an additional controller. The second controller must be able of regulate the engine speed and shall be located in the engine compartment. Both controllers must provide visual indication of the orders given to the engine and its responses; and are to be equipped with an alarm informing the presence of a mismatch between orders and responses. All the alarms explained in the previous points are to be visual and audible. The indicators are to be fitted at a normally attended position.

2.2 Type Approval Certification

The society takes part in the implementation of national and international rules and standards as delegated by various Governments. Is an EU recognized organization and may offer EU Type Approval Certificates. The “Rules” for the classification of ships includes the certification of its components such as the machinery inside the ship and as such, for the new engine developed and fabricated by FPT, a type certification process is compulsory and the type approval procedure should be followed. The type approval is the approval procedure of the product design and the prototype test performance for compliance with classification and requirements. The type approval is particularly suitable for products that can be consistently manufactured to the same design and specification such as the new

16L – 600 hp marine diesel engine developed by FPT. This implies that the design of the product is assessed once and the approval is made valid for all subsequent products of identical design.

2.2.1 Engine Type

It has to be considered that for each new engine of a type that is required to be approved, a type approval certificate is to be obtained by the engine designer and thus, it is necessary to specify the characteristics that define an engine type. Engines may be considered the same type if they do not differ from any of the following items.

- The cylinder bore
- The stroke
- The working cycle (2-stroke, 4-stroke)
- The cylinder arrangement (in-line or V)
- The rated power per cylinder at rated speed and mean effective pressure
- The kind of fuel (liquid, dual fuel, gaseous)
- The method of fuel injection (direct or indirect)
- The valve and injection operation (by cams or electronically controlled)
- The turbocharging system (constant or pulsating pressure)
- The charged air cooling system (e.g. with or without intercooler)

2.2.2 Type Approval Certification Process

The type approval certification procedure consists of the following steps [3]:

2.2.2.1 Drawing and specification approval

Provide an overview of the engine's design, characteristics and performance. The specific information that shall be sent to RINA is listed in different table available in [3] and includes the engine parts descriptive information such as connecting rod, flywheel, oil pump, crankshaft, crankcase, piston assembly, fluid circuit systems, etc.

2.2.2.2 Definition of the test program

The definition of the loads, residence times and number of cycles to be followed for the type tests is discussed between the designer/manufacturer and the Classification Society considering the information provided in section 2.2.3.2.

2.2.2.3 Execution of the type tests

The engine is subjected to an endurance test of 100 h as agreed with the Classification Society, considering the test parameters resulting from the previous step and is to be followed in presence of Classification Society personnel. There are additional tests, called functional tests, for verification of different engine specific operating conditions as it is reviewed in detail in section 2.2.3.3 and 6.3.

2.2.2.4 Review of the obtained type testing results

All the measurements and recordings taken during the type test are provided to the surveyor for its further analysis. After the test, the engine shall be disassembled for a subsequent inspection of the engine components by the inspector.

2.2.2.5 Issuance of a type approval certificate

Upon satisfactorily meeting the Rules requirements.

2.2.3 Type Tests

It is necessary to perform a type test for every new engine of a type according to the conditions presented in this section. Type testing is to be arranged to represent foreseen service load profiles considering the engine family. In this specific case the 16L – 600 hp engine is a heavy duty commercial type of engine which gives the possibility to a continuous engine usage at maximum rating. The test shall also cover the mechanical fatigue scattering present during service operation that includes:

- High cycle fatigue (HCF) in parts such as con rods, cams, rollers, springs, etc. where high stresses are present due to elevated injections pressures, cylinders maximum pressure, etc. and;

- Low cycle Fatigue (LCF) that could be present in hot parts when the engine is operated under load profiles that consider steep ramps such as: idle→full load→idle operation.

2.2.3.1 Ambient reference conditions and power correction

According to the RINA Rules, the testing of the engine is to be conducted at the ambient conditions shown in Table 1 [3].

Table 1. Ambient reference conditions.

	For RINA	For ISO
Barometric pressure	1 bar	1 bar
Relative air humidity	60 %	30 %
Air temperature	45 °C	25 °C
Seawater Temperature (Charging air coolant inlet)	32°C	25 °C

The engine manufacturer is not expected to provide simulated ambient reference conditions at any test, but if the tests are performed under different ambient conditions, and considering that substantial changes in engine performance can arise from changes in atmospheric conditions, power corrections are to be made in agreement to the procedures described in the international standard ISO 15550 [4] and ISO 3046-1 [5] that establish the normalizing and correcting factors to be adopted to adjust the different power rates. The power normalization and correction are to be performed only when the engine is operating at full power, i.e. at its rated power.

Since the type tests were performed under different ambient conditions than those stipulated by the RINA Society, a procedure for correcting the rated power must be followed. The first step of the ISO procedure consists on normalizing the engine power measurements from the test bed to the ISO standard reference conditions of Table 1, using the model shown in the next equations. This model has been verified by ISO by testing a representative number of pre-set engines.

The observed power shall be multiplied by a factor α_c which is specific for compression ignition engines:

$$\mathrm{P}_r = \alpha_c P_y \tag{1}$$

Where

- P_r: Brake power under standard reference conditions; [kW]
- α_c: Power correction factor for compression-ignition engines ; [-]
- P_y: Brake power under ambient conditions during test; [kW]

For calculating the normalization factor α_c the following equation is used and considers other two different factors, the atmospheric f_a and the engine f_m factors.

$$\alpha_c = (f_a)^{f_m} \tag{2}$$

With

- f_a as the athmospheric factor which indicates the effects of the environmental conditions (pressure, temperature and humidity) on the air drawn in by the engine. For the specific case of this study, i.e., for a turbocharged diesel engine with charge air cooled by air-to-liquid charge air cooler:

$$f_a = \left(\frac{p_r - \phi_r p_{sr}}{p_y - \phi_y p_{sy}}\right)^{0.7} \left(\frac{T_y}{T_r}\right)^{0.7} \tag{3}$$

Where

- p_r and p_y are the ISO standard reference total barometric pressure and the ambient total barometric pressure during test, respectively. [kPa].
- p_{sr} and p_{sy} are the standard reference and test saturated water vapor pressures, respectively. [kPa].
- ϕ_r and ϕ_y are the ISO standard and test relative humidity, respectively. [%].
- T_y and T_y are the standard reference and test ambient air thermodynamic temperatures, respectively. [K].

- And f_m as the engine factor and is dependent on the type of engine and the air/fuel ratio.

$$f_m = 0.036q_c - 1.14 \tag{4}$$

With the normalized specific fuel delivery q_c defined as

$$q_c = \frac{q}{r_r} \tag{5}$$

Where

- q is the fuel delivery parameter in milligrams per cycle per liter of the engine's swept volume. [mg/(L·cycle)]

$$q = \frac{120000\, \dot{m}_f}{V \cdot n} \tag{6}$$

- $\dot{m}_f$ is the fuel mass flow, [g/s]
- V is the engine displacement, [L]
- n is the engine speed, [1/min]
- r_r is the ratio between the absolute static pressure at the compressor outlet and compressor inlet under standard reference conditions. [-]

Equation (4) is valid only for $37.2 \leq q_c \leq 65$. Figure 1 shows the engine factor f_m as a function of the normalized fuel delivery q_c and it can be appreciated that for values of q_c lower than 37.2, f_m takes a constant value of 0.2. For values of q_c higher than 65, f_m takes a constant value equal to 1.2.

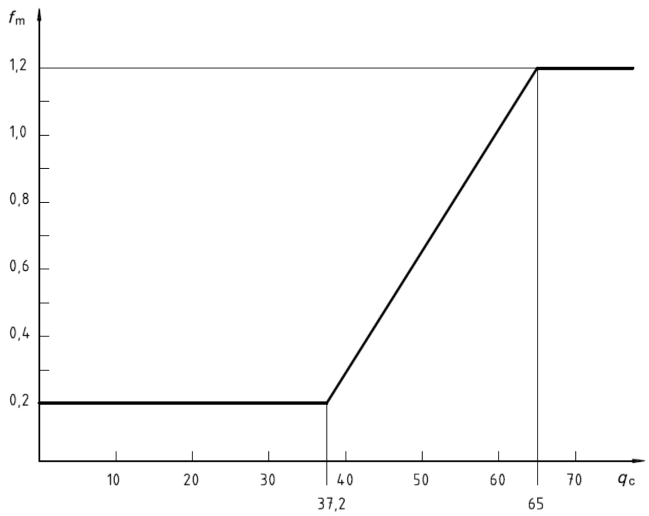


Figure 1. Engine factor f_m vs. corrected fuel delivery q_c

There's also a usable range for the normalization factor α_c and the equation (2) is only applicable if $0.96 \leq \alpha_c \leq 1.06$. If the limits are exceeded, the normalized power value that is obtained shall be reported together with the test ambient conditions.

Now that the value of α_c can be obtained, the calculation of the "engine power normalized to the ambient conditions for the ISO standard (P_r)" can be done using equation (1). The next step is to pass from the engine outpower under the ISO standard reference conditions P_r to the power under the ambient conditions required by the RINA P_x as shown inTable 1 and as described in [5]. Therefore:

$$P_x = \alpha P_r \tag{7}$$

With the engine power correction factor α given by

$$\alpha = k - 0.7(1-k)\left(\frac{1}{\eta_m} - 1\right) \tag{8}$$

And

$$k = \left(\frac{p_x - a\phi_x p_{sx}}{p_r - a\phi_r p_{sr}}\right)^m \left(\frac{T_r}{T_x}\right)^n \left(\frac{T_{cr}}{T_{cx}}\right)^s \tag{9}$$

Where,

- η_m is the mechanical efficiency. In the absence of any such statement, the value of 0,8 shall be assumed; [-]
- P_x is the ambient total barometric pressure required by the RINA rules, [kPa]
- P_r is the ISO Standard reference total barometric pressure, [kPa]
- ϕ_x and ϕ_r are the relative humidities as requested by RINA and for reference conditions of ISO, respectively. [%]
- p_{sx} and p_{sr} are the satured water vapour pressures for RINA and ISO, respectively. [kPa]
- T_r and T_x are the ambient air thermodynamic temperatures for ISO and RINA, respectively. [K]
- T_{cr} and T_{cx} are the charge air coolant thermodynamic temperatures for ISO and RINA, respectively. [K]
- The a factor and m, n, s exponents depend on the engine and fuel type, and on the conditions. For this specific case: a medium speed, four-stroke compression ignition (diesel) engine, turbocharged, with charge air cooling; takes the numerical values of $a = 0, m = 0.7, n = 1.2$ and $s = 1$ as stated in [5].

Considering the values specified previously, equation (9) is reduced to:

$$k = \left(\frac{p_x}{p_r}\right)^{0.7} \left(\frac{T_r}{T_x}\right)^{1.2} \left(\frac{T_{cr}}{T_{cx}}\right) \tag{10}$$

Finally, with the value of the variable k it is possible to calculate the power correction factor α and the power corrected to the ambient conditions stated by the RINA rules (P_x) employing equation (7).

2.2.3.2 Load points

- The maximum rated power for continuous operation (MCR – Maximum continuous rated power - is the maximum power at the RINA ambient reference conditions of Table 1 which the engine is capable of delivering, at nominal maximum speed, in the period of time between two consecutive overhauls), i.e., 100% output at 100% torque and 100% speed corresponds to the load point 1 in Figure 2.
- Load point 2 corresponds to 100% power at the maximum permissible engine speed.
- A 110% of the engine power at 103.2% of the engine speed according to the nominal propeller curve is shown as load point 3a.
- Minimum permissible speed at 100% torque corresponding to load point 4.
- Minimum permissible speed at 90% torque corresponding to load point 5.

- Load points 6, 7, 8 are related to partial loads, i.e., 75%, 50% and 25% of the rated power, respectively; and with a speed according to the nominal propeller curve, i.e. at 90.8%, 79.3% and 62.9% speed; respectively. Instead the load points 9, 10, 11 are related to the same partial loads, but at constant rated speed setting.

The conditions to choose depend on the intended application of the engine and the homologation test conditions are the result of an agreement between the Classification Society and the designer/manufacturer. During the operation on these load points the engine parameters shall be the specified.

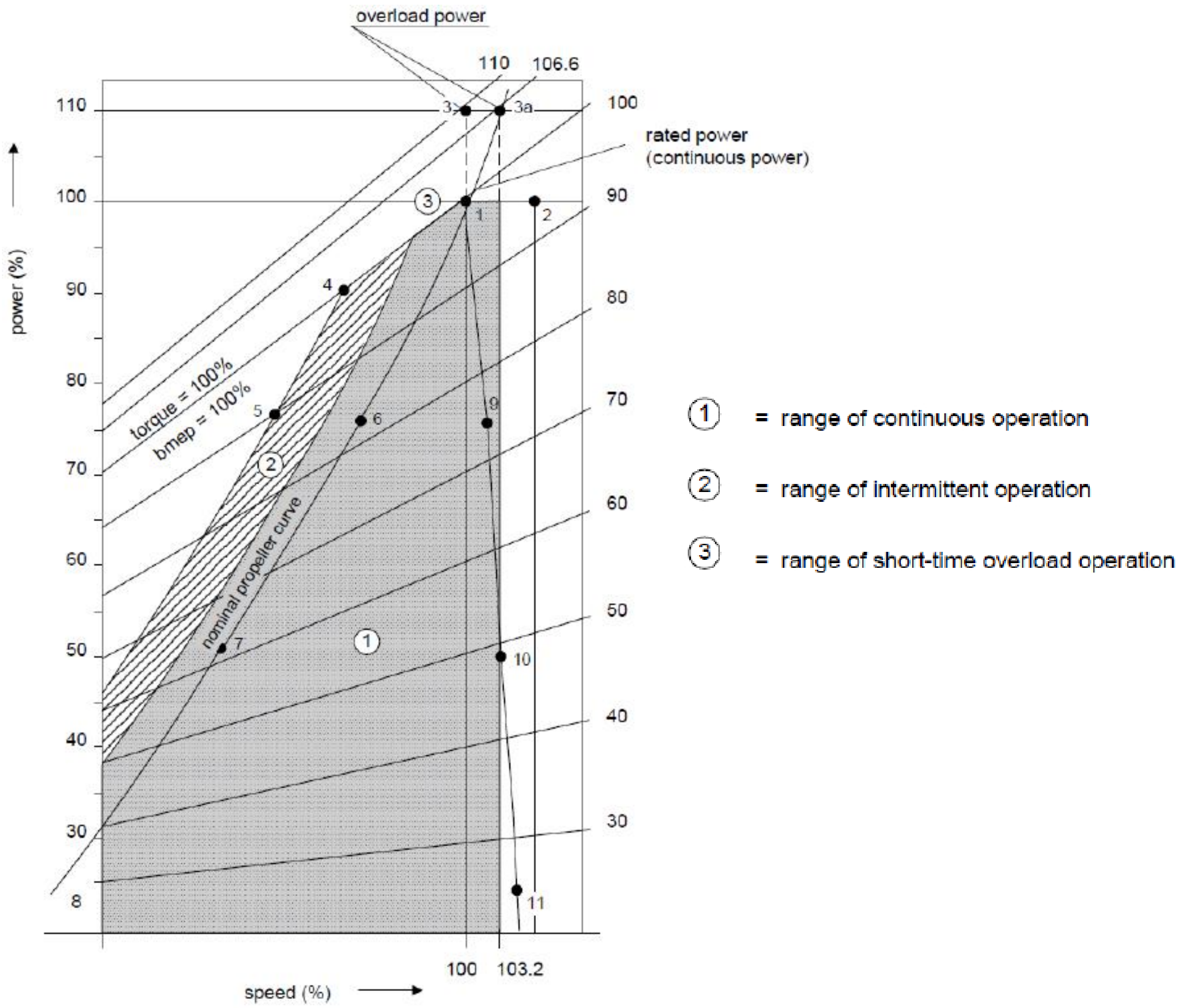


Figure 2. Power – Speed diagram

2.2.3.3 Functional tests

- Verification of the lowest propulsion engine speed according to the nominal propeller curve. No alarm shall occur during the operation.

- Tests for verifying the compliance of the speed governor which shall avoid the rated speed to be exceeded by more than 15%.

3 CASE STUDY

3.1 FPT Industrial – Powertrain for Industrial Vehicles

FPT Industrial is the brand of CNH Industrial dedicated to the development, production, sale and assistance of powertrains for all industrial applications: truck, bus, marine, energy solutions, defense, special vehicles, construction and agriculture, or more generally, divided in: On Road, off Road, marine and power generation application divisions. FPT industrial produces seven engine families with displacement from 2.2 to 20 liters with maximum powers between 42 and 1000hp. Besides diesel engines, the company offers solutions for engines that use alternative fuels, including natural gas and engines compatible with biodiesel up to 20%.

FPT industrial is the 4th largest industrial engine, axles and transmissions manufacturer in the world with a global footprint and 10 plants located in Argentina, Brazil, Italy, France and China and research centers also in USA and Switzerland; able to develop and produce any engine adapted to local requirements and specifications.

3.2 16 Liter - 600 hp Marine Diesel Engine

The FPT 16-liter marine diesel engine studied in this thesis project is a new developed marine engine recommended for fishing applications and heavy duty applications, giving the possibility to use the engine continuously at maximum rating (full throttle) and, with utilization at maximum power up to 100% of time and for unlimited hours per year. The time between overhauls (TBO), that is the manufacturer's recommended number of running hours before the engine or any component require to be removed, disassembled, cleaned, inspected and repaired; should be TBO ≥ 15000 h.

The engine offers high performance and high durability with a minimum displacement. It Is characterized by a class leading power and torque density and, a high compactness and lightness, fitting in the packaging of a regular 13-liter engine; giving the final customer the possibility of easier installation and wider space on board. This power unit features the latest generation of common rail injection systems, with important injection pressures and multi-point fuel injection; increased fatigue strength and new material components (cylinder block, cylinder head made of compacted graphite iron – CGI–, crankshaft & bearings, con-rods, steel pistons, rings, increased flows for oil and seawater circuits with a professional bronze body and impeller seawater pump). The induction system is composed of a single stage water cooled waste gate turbo with water cooled aftercooler. Considering an easier installation and in order to prevent any potential leakage, the oil cooler, oil pump, water pump and blow-by system have been integrated into the engine design. For maximizing the reliability and resistance to salt and water, the engine has been designed with specific air, water, oil and fuel circuits. The marinization set includes: heavy duty, high flow seawater pump, a new heat exchanger for improved heat rejection, and twisted wirings for maximum resistance to dirt, water and sludge.

The reduced gas emissions and a blow-by system designed for a great oil vapors recovery together with the green filters for better treatment of engine oils, provide a low environmental impact and compliance to international recreational and commercial emission standards such as IMO MARPOL (International Maritime Organization – Marine Pollution), EU-RCD (Recreational Craft Directive), EU-IWV (Inland Waterway Vessel), USA-EPA (Environment Protection Agency). The noise and vibrations reduction contributes to a higher navigation comfort. The engine is developed for heavy duty applications and recommended for work and passenger transport boats, fishing boats and tugboats. The engine is characterized by the data shown in Table 2.

Table 2. Engine characteristics

Thermodynamic cycle	Diesel 4 T
Air handling	Turbocharger + Aftercooler
Cylinders arrangement	6 L
Maximum Power - [kW] (HP)	441 (600) @ 1800 rpm
Maximum Torque - [Nm]	2650 @ 1300 rpm
Bore x Stroke - [mm]	141 x 70
Total displacement - [L]	15,9
Valves per cylinder	4
Cooling system	Liquid
Compression ratio	16,5:1
Direction of rotation (viewed facing flywheel)	CCW
Engine management	ECU
Injection system	CR (Common Rail)
Dimensions (L x W x H) - [mm]	1820 x 1074 x 1161
Dry weight - [kg]	1570
Electric system voltage - [V]	24

4 ENGINE

4.1 Marine Engine Propulsion

4.1.1 Power Definitions for Marine Propulsion

In the boat propulsion system there are different power levels depending on the losses encountered along the powertrain, from the engine up to the propeller.

4.1.1.1 Brake power (BP)

It is the power measured with the dynamometer brake at the drive shaft (flywheel) during the bench tests, i.e. the engine power available at the flywheel. BP should be greater than the IP in order to get a wide enough power margin for delivering the needs of power requested by the vessel's electrical installations, generators, compressors or other machinery that could be driven by the engine and not directly used to propel the vessel. See Figure 3.

4.1.1.2 Propeller shaft power (SP)

Is the power available on the output of the reducer-inverter and then transmitted along the propeller shaft to the propeller. SP is the power generated by the engine (BP) minus the power lost in the transmission and the losses resulting from the friction of the propeller shaft bearings. The shaft power is the actual power delivered to the propeller.

4.1.1.3 Effective power (EP)

Is the power required to overcome the vessel's resistance at a given speed without considering the power required to turn the propeller and overcome the internal friction of the powertrain system. This value is close to the power required to tow the vessel.

4.1.1.4 Indicated power (IP)

Power needed to drive the vessel at a given speed and includes the necessary power to overcome the internal friction of the machinery and the power to turn the propeller though the water. It is the power which is actually translated into thrust for the motion of the boat. The ratio of IP/EP is usually equal to 50% meaning that IP is commonly twice the EP.

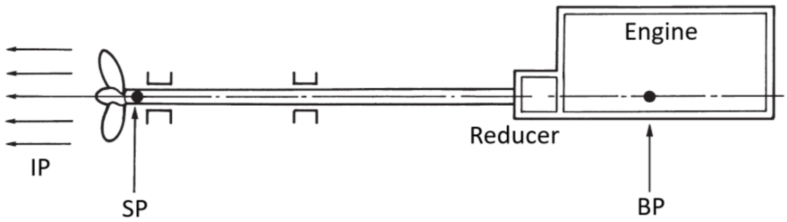


Figure 3. Boat power levels

4.1.2 Engine Setting

Marine engines are manufactured in a number of duty or power ratings. These ratings determine whether the engine is intended for continuous operation or for short term operation at maximum speed.

4.1.3 Engine Performance, Ship Resistance and Propeller Matching

The boat type, i.e. the type of hull together with the choice of the propeller, allows the possibility to identify the required engine performance for the application. Listed below there are other factors considered as "power losses" which must be considered as they also have an impact on the required performance and foresee a power "reserve" to compensate for this extra power absorption.

- As discussed previously, the environmental conditions have an impact on the power delivered by the engine. Environmental temperature, pressure and humidity.
- The power absorbed by the auxiliary components present in the boat such as, but not limited to: Generators, winches, pumps, inverters, reducers, etc.
- There's a difference between the conditions of a new boat and a used one due to the hull fouling related to the fact that organisms can attach themselves to the hull. There's a vegetation and incrustation growth on the submerged part of the hull and on the propeller. This phenomenon increases the boat's resistance and reduces the propeller efficiency.

In the design of marine propulsion systems, the correct matching of the engine to the propulsor and the vessel is of great importance. If the correct matching is not performed correctly the system could present excessive fuel consumption, maximum vessel speed different from the designed ones, engine overloading, etc. The problem of moving a vessel though the water involves the proportions and shape of the hull, the size and type of engine, and the system to transform the power into effective thrust. It is necessary for the ship to sail at the required speed with the minimum of shaft power. This is done attaining the best combination of low resistance and high propulsive efficiency that can only be accomplished by a proper matching of hull and propeller.

4.1.3.1 Hull types

All hull types discussed in this section refer only to the portion of the hull below the waterline. What is above the waterline has little impact on the propulsion machinery apart from the *Air Resistance* that will be explained in the next section, but still has a negligible effect compared to the other resistance figures present while the boat is sailing.

- **Displacing hulls**

  Is usually characterized by a round bottom and narrow stern - see (Figure 4. a) -. During sailing and when the boat speed is increased, this type of hull maintains its static **trim**[1] and does not reduce its **draught**[2]**.** They move through the water by pushing the water aside**.**

- **Gliding hulls**

  Due to the shape of the bottom and the power installed they skim over the water surface with relatively little disturbance of the water (ride on top of the water rather than pushing it aside). They displace the water only when stationary or travelling at low speeds, but as soon as the speed is increased, the trim changes and the water pressure lifts the boat stem. The pressure increases with the square of the speed. The main resistance to planing hull speed is the skin friction, for this reason hull of this type are very sensitive to the smoothness of the hull, making good hull maintenance essential for top performance. Gliding hulls are also very sensitive to boat weight. Flat-bottomed and V-bottomed hull shapes act as planing hulls - see Figure 4. c) -.

- **Semi-displacing hulls**

  Can be described as having characteristics of both planing and displacement hulls, but are not one neither the other. They can change their trim during sailing as they lift the stem and, as a result of the incidence of the bottom plane, they

[1] The **trim** of a vessel is defined as the difference between the bow draught and the stern draught. It indicates the "inclination" of the vessel in the longitudinal direction as the pitch does it in the case of automobiles. A boat could be trimmed by the stern or trimmed by the bow. The **static trim** is the trim when the boat is considered to be at rest (not sailing).

[2] **Draught** is is the vertical distance between the waterline and the bottom of the hull. It determines the minimum depth of water a ship or boat can safely navigate.

can use a small part of the water dynamic pressure to obtain a partial gliding effect - see Figure 4. b)-.

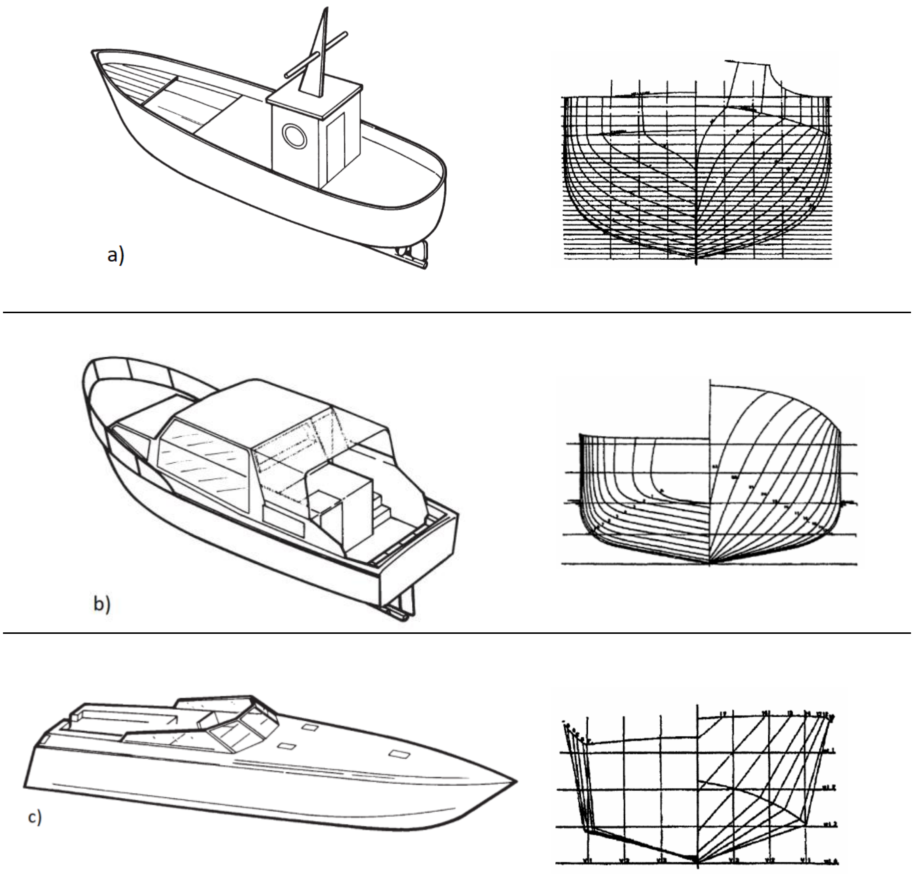


Figure 4. Different hull types. a) Displacing hull. b) Semi-displacing hull. c) Gliding hull. [6].

4.1.3.2 Ship resistance

The resistance of a ship at a given speed is the force required to tow the ship at that speed in smooth water, assuming no interference from the towing ship. If the hull has no appendages, this is called the bare-hull resistance. The power necessary to overcome the resisting forces ($\mathrm{R_T}$) coming from the water and air is called the effective power EP [kW] and is given by

$$EP = R_T V \tag{11}$$

Where

- $\mathrm{R_T}$ is the total resistance, [kN]
- V is the boat speed , [m/s]

The effective towing power, i.e. EP, can also be described by equation (12) according to [7].

$$EP = C_E \cdot \rho^{1/3} \cdot \Delta^{2/3} \cdot V^3 \tag{12}$$

With

- $\mathrm{C_E}$ is the specific resistance and depends on the boat speed, its hull shape, **fouling** [3], sea state and water depth.
- ρ as the seawater density
- Δ is the boat displacement and is defined as the weight of the vessel and its contents measured based on the weight of the water displaced by the hull. It is measured indirectly using Archimedes' principle by first calculating the volume of water displaced by the boat and then converting that value into weight displaced.
- V as the boat speed

The total resistance $\mathrm{R_T}$ from equation (11) is the sum of different kinds of resistance - equation (13) - caused by a variety of factors that interact with each other in an extremely complex way. For dealing with a less complex phenomenon, it is usual to consider the total calm water resistance as being made up of four main components.

$$R_T = R_f + R_w + R_v \tag{13}$$

a) **The frictional resistance ($\boldsymbol{R_f}$)**, due to the motion of the hull through a viscous fluid (water). R_f accounts as the largest single component of the total resistance of a boat. The roughness of the surface in contact with the water will increase the frictional resistance in a considerable way.

b) **The wave-making resistance ($\boldsymbol{R_w}$)**, due to the energy that must be supplied continuously by the ship to the wave system created on the surface of the water. When a boat travels on the surface of the water is subjected to fluid pressures acting normally on all parts of the hull. These normal pressures will vary along the length of the boat causing waves which alter the distribution of pressures over the hull resulting on a forwards and backwards net force which is the wave-making resistance. R_w greatly depends on the hull shape, waterlines and the transverse sections.

c) **Vortexes resistance ($\boldsymbol{R_v}$),** due to the energy carried away by eddies spread from the hull or appendages. When the water may be unable to follow the surface curvature it will break away from the hull giving rise to eddies and the separation

[3] Marine fouling describes a wide range of organisms that attach themselves to surfaces immersed in the ocean and accumulate with time. These organisms range in size from bacteria to seaweed to mollusks. Marine fouling generates surface roughness which increases the drag resistance of a vessel moving through water.

resistance. This resistance could be compared to a “sucking” effect generated downstream of large surfaces that are interrupted abruptly.

d) **Air resistance ($\boldsymbol{R_A}$)** experienced by the above-water part of the main hull and the structures due to the motion of the ship through the air. This resistance depends on the boat’s speed and the area and shape of the upper works. When the wind is blowing it depends on the wind speed and its relative direction.

The towing resistance $\mathrm{R_T}$ depends on the boat speed and also on the hull geometry, shape and characteristics; therefore it is of great importance to understand how these parameters are related with each other. The Froude number (FN) is a no-dimensional value used to determine the resistance of a partially submerged body moving though water and is dependent of the boat speed and dimension. Vessels having the same Foude numbers produce a similar wake and hence, their dynamics and the resistance to the hull motion can be easily compared even if their size or geometry are otherwise different.

$$FN = \frac{V}{\sqrt{gL}} \tag{14}$$

Where

- V is the boat speed, [m/s]
- g is the acceleration due to gravity, [m/s2]
- L is the length of the vessel at the water line level, [m]

If a single pressure point travels in a straight line over the water surface, it sends out waves that combine to form a characteristic patter as shown in Figure 5. This consists of a system of transverse waves following behind the point accompanied by a series of divergent waves from the point. The whole pattern is contained within two straight lines making an angle of 19 degrees with the vertex located at the pressure point.

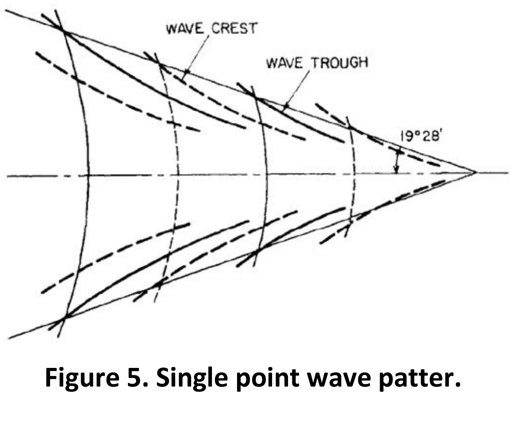


Figure 5. Single point wave patter.

A ship-wave system is shown in Figure 6. The most noticeable waves are a series of divergent waves that start at the bow of the vessel. Between these divergent waves on each side of the ship transverse waves are formed and have their crest lines normal to the direction of motion near the hull, bending back as they approach the divergent-system waves and finally coalescing with them.

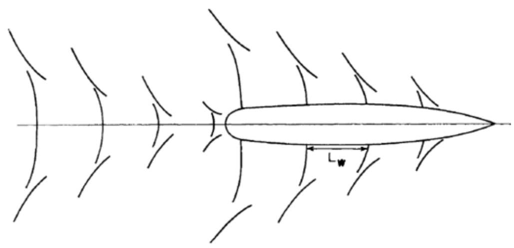


Figure 6. Schematic diagram of the bow and stern wave systems formed by a travelling vessel [8].

The general pattern described by the travelling single pressure point can be easily seen in the ship-wave system and explains many of its features. Since the wave pattern, as a whole, moves with the vessel, the transverse waves are moving in the same direction and at the same speed as the vessel speed V. According to [8], at the boat speed V, the wave length of these transverse waves ($\mathrm{L_w}$) is defined as:

$$L_w = \frac{2\pi\, V^2}{g} \tag{15}$$

It is worth to highlight that the wave length $\mathrm{L_w}$ increases with the square of the velocity.

When the vessel travels at a speed V that generates a wave length $\mathrm{L_w}$ equal to the value of its floating hull length L, i.e. when $\mathrm{L_w = L}$ in equation (14), a constant value of FN = 0.3989 ≈ 0.4 is determined as indicated in the next equation. This value applies only to boats with a displacing hull type since the boat maintains the same static trim and does not reduce its draught with the increment of the boat speed (for definition of static trim and draught see section 4.1.3.1).

$$FN = \frac{V}{\sqrt{gL_w}} = \frac{1}{\sqrt{2\pi}} \cong 0.4 \tag{16}$$

If FN < 0.4, the boat will travel over a wave patter in which the wave length is shorter than the length of the vessel ($\mathrm{L_w < L}$). If FN > 0.4, the boat will travel over a wave patter with a wave length greater than the length of the vessel ($\mathrm{L_w > L}$) and a condition with a great increase in sailing resistance will be generated. This condition will also tend to lift the bow and in the case of displacing hulls will often create dangerous sailing conditions, and will also generate expensive operation due to the high resistance increment. This phenomenon can be appreciated in Figure 7 where it is also possible to see that the resistance curves are dependent on the hull type. For displacing hulls, the frictional resistance $\mathrm{R_f}$ is predominant with respect to the wave-making resistance $\mathrm{R_w}$. On the other hand, for glinding hulls $\mathrm{R_w}$ component becomes predominant as the boat speed increases. Semi-displacing hulls will have a $\mathrm{R_w}$ component that will take importance with the boat speed increment.

Considering the Froude number, it is possible to distinguish and identify the different boat hull shapes [9]:

- FN < 0.4 for displacing hulls
- 0.4 < FN < 0.8 for semi-displacing hulls, and
- FN > 0.8 gliding hulls

Summarizing, the value of FN will result in different hull types and different resistance curves. In gliding and semi-displacing hulls, the pattern of the sailing resistance highlights a "hump" representing the transient state from the displacing sailing to the gliding sailing.

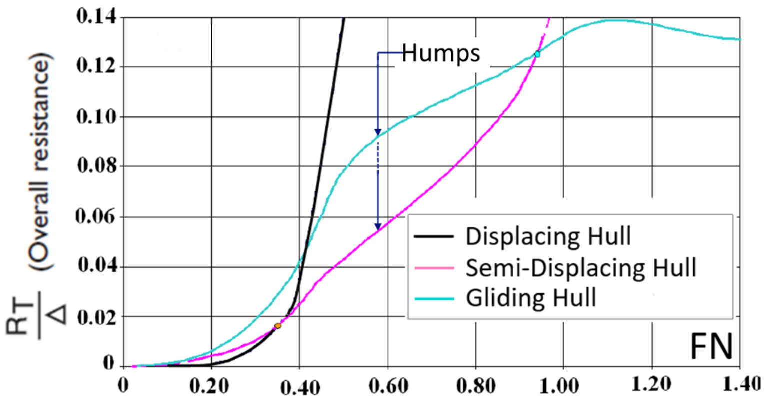


Figure 7. Different types of hull resistance characteristics [9].

The propulsion efficiency η_P is defined as ratio between the power used in overcoming the resistance to motion $\mathrm{R_T}$ at a certain boat speed (previously defined as EP) and the power that is actually delivered to the propeller, i.e. the propeller shaft power SP.

$$\eta_P = EP/SP \tag{17}$$

The power delivered by the main engine and available at the reducer/gearing output shaft is called (GP). The difference between GP and EP represents the power lost in friction in the shaft bearings and the **stern tube** [4]. The ratio described in equation (18) is called the shaft transmission efficiency η_s

[4] Stern tube is a hollow tube that runs through the bottom of the ship from the engine to the propeller. There's a shaft that runs inside the stern tube and connects the engine to the propeller.

$$\eta_s = SP/GP \tag{18}$$

4.1.3.3 Engine and propeller matching

For the ship to be able to move at its designed speed it is important to match the drive train, engine and reduction gear with the propeller required power. It should be considered that the engine is in charge of generating the power needed by the propeller in order to thrust the vessel through the water overcoming the sailing resistance R_T.

The propeller dimensioning is extremely important to establish the engine power that enables the boat highest speed. Its design depends on the hull shape and on the distribution of the weights onboard the vessel. The propeller design considers the selection of the correct diameter, the pitch, rotation speed, number of blades, blade shape; among other parameters. Propellers with the wrong dimensioning and design will not be able to transfer the maximum power supplied by the engine.

The theoretical propeller power (PP) curve is a representation of the average power absorption of the propeller at different speeds (rpm_s). PP is indicated in equation (19) [10]. This propeller law is useful for engine builders/designers at the test cells where engine loads can be applied with the dynamometer according to the load and revolutions calculated from this power law.

$$PP = C_{sm} w^n \tag{19}$$

With:

- $\mathrm{C_{sm}}$ as the sum matching constant. Chosen in order to make the propeller curve PP to cross the shaft power SP curve at near the maximum RPM and depends on the propeller pitch, diameter and blade area.
- w are the RPM
- n is the exponent that varies from 2.2 to 3. By experience, n has been found to be equal to 2.7 for almost all medium to high speed pleasure vessels, passenger vessels and light commercial vessels. For heavy commercial craft operating at low speed n should be taken as 3. [10].

The type and operation of the vessel is related with bottom shape of the boat and so is the propeller absorbed power PP since it is directly related to the parameter FN and as a consequence it is also associated to the hull type. It has been found ([6] and [9]) that the propeller power absorption PP follows a square or cubic behavior depending on the selected propeller, but the hull type greatly affects this behavior law. For displacing hulls

$\mathrm{n} = 3$ in equation (19) giving a cubic absorption law, while for gliding hull types $\mathrm{n} = 2$ and the propeller power absortion is called the square (or quadratic) law.

The shaft power SP (for definition of SP see section 4.1.1) and the propeller absorption power PP curves have very different behaviors, as depicted in Figure 8, making both curves to cross only at a single point. Since the engine must be able to reach its maximum speed (or close to it), in the case of fixed pitch propellers, the propeller characteristics must be chosen in order to make the propeller absorbed power PP to match the engine power close to the maximum RPM. It is worth to note that the shaft power SP is related to the effective towing power EP by the propulsion efficiency η_P of equation (17).

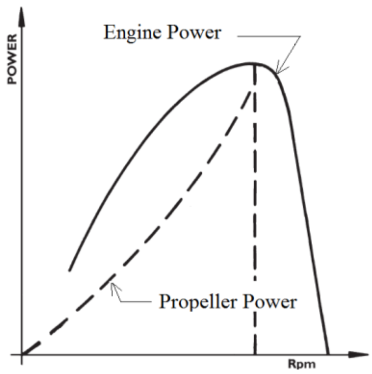


Figure 8. Engine power and propeller absorption curves.

When performing the engine and propeller matching it should be considered that the engine must be neither under loaded nor overloaded. In the case where the propeller characteristics are chosen in such a way that the propeller power curve crosses the shaft power curve at far less RPM than the max RPM, the engine will never be able to reach its rated speed and the engine will be overloaded at any higher speed (see Figure 9 - curve A) and as a result black smoke could be emitted from the exhaust and this situation could lead to long-term engine damage and wear. The contrary case, where the selected propeller has very low power requirements causes the propeller curve to never cross the shaft power curve (see Figure 9 - curve B) and the engine will be oversized for the application while being fuel, cost and space inefficient. In this latter situation, the engine cannot be operated to full throttle and consequently full power will never be used and could lead to a situation in which the propeller will be run heavily, spinning it ineffectually, leading the engine to easily overcome its maximum speed and to be often limited by the speed governor.

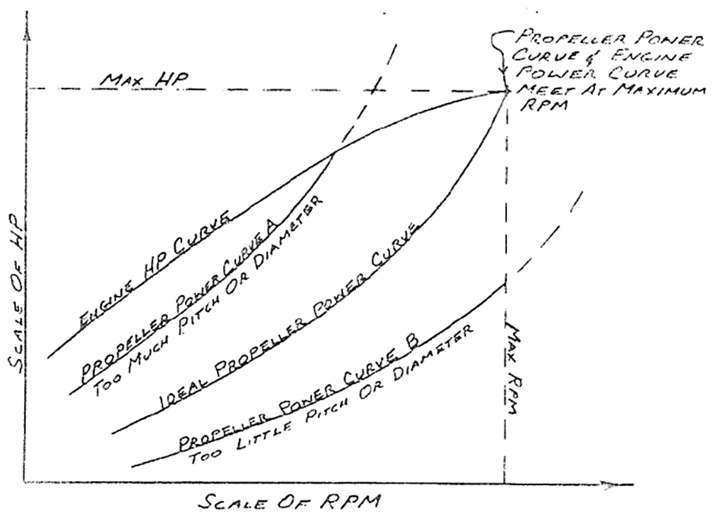


Figure 9. Engine and propeller power equilibrium [10].

It must be addressed that the SP curve shows the power availability for a given speed and not the actual output power, therefore the power difference between the shaft power SP curve and the propeller absorption PP curve is not generated by the engine since it is not requested by the propeller; that is, the engine only generates the power level requested by the propeller according to the propeller curve. Taking this into account, it must be said that for marine engine applications, the throttle does not directly adjust the fuel flow to the engine, but it adjusts a governor that regulates the fuel flow to maintain a constant rpm. Hence, if the propeller requests less power than that available from the engine, the governor limits the fuel flow and reduces the power generated by the engine at that specific speed. Figure 8 and Figure 9 show that as the RPMs increase, the unused power quantity decreases since both curves get closer.

To summarize, the matching of the engine and its propeller is basically finding an equilibrium between the power – rpm characteristics of the engine and the propeller. The power produced by the engine minus any loss in the transmission equals the power absorbed by the propeller.

4.2 Engine Marinization

The marinization process consists in the modification of a base engine followed together with the development of specific components in order to adapt the engine for its use in marine applications considering the harsh conditions of a marine environment, the new requested power output, the RPM for rated speed and the safety and environmental regulations. Usually, the marinization process is followed based on an Off-Road application and the engine is subjected to an increase in the power rating since in the original application the engine works at a higher profile, that is, the off-road engine is usually used at higher loads for long periods of time. The following sections detail the components design characteristics and restrictions.

4.2.1 Valve Cover

The use of non-metallic materials for any engine element or component is assessed in relation to the risk of fire associated with the component. Elements of machinery subjected to splash of inflammable liquids from the inside due to the action of machinery running and where there's risk of ignition due to the proximity to hot surfaces. This specific case is present on the valve cove and thus, the use of metallic material for its fabrication is required.

4.2.2 Intake System

The high salt concentration in the air creates a corrosive environment that has a big impact on the engine design and the materials selection. The air-intake system is made by anodized pipes and; an epoxy-based material, more resistant to salt and humidity, is used for the air filters.

The high heat exchange in the intercooler, caused by the seawater at low temperatures, causes condensation on the surface of the intake valves that, together with the high salt concentration in the air, could lead to the acceleration of the corrosion process hence, the valves material is to be examined and replaced with a high corrosion resistant material.

The corrosion of the valves, intake line pipes, the components in the combustion chamber (piston and piston rings) and the CAC; as a result of the high salt concentration in the intake air is reduced by avoiding condensation in the intake line or focusing the condensation in a specific area of the line and adopting a condensation accumulation drain pipe.

4.2.3 Blow-By Filter

The increase in the rated power consequently increases the blow-by flow rate and thus, the filtration capacity should be improved.

4.2.4 Oil Sump

The oil sump is to be redesigned and the addition of lateral pockets, as seen in Figure 10, help to reduce the component height according to the more stringent engine compartment layout constraints found in marine applications and for increasing the sump volume to improve the sump capacity and thus, reducing the number of refilling operations. It is also necessary to foresee threaded holes in order to connect an oil draining pump used in the maintenance service operations.

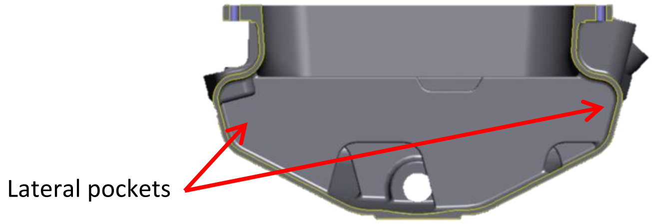


Figure 10. Oil sump design

4.2.5 Piston

Due to the increase in power rating, the compression ratio should be reduced by means of increasing the bowl's volume in order to maintain the same peak cylinder pressure (PCP) of the base engine. This way, the same reliability of the base engine is assured without the need of applying modifications to the crankshaft or to the connecting rods.

The clearance between the piston ceiling and cylinder head should be verified and modified. The clearance is to be increased to counteract the effect of the greater piston thermal elongation and the increase in the elastic deformation due to the higher rated speed compared to the base engine. Finally, also as a consequence of the power increase, the piston should deal with a higher thermal load and therefore, the oil flow required for the piston cooling is also to be enhanced.

4.2.6 Oil and Fuel Filters

A duplex type arrangement (Figure 11) for the fuel and oil filters was implemented as requested by the RINA Rules. The double switchable filters allow the performance of maintenance operations without the interruption of neither the oil flow nor the fuel flow and hence, avoiding the need of the engine shut down. The duplex arrangement consists of two separate filter baskets housings and a valve handle placed between the two gaskets to divert the liquid flow to one filter while the other is being cleaned.

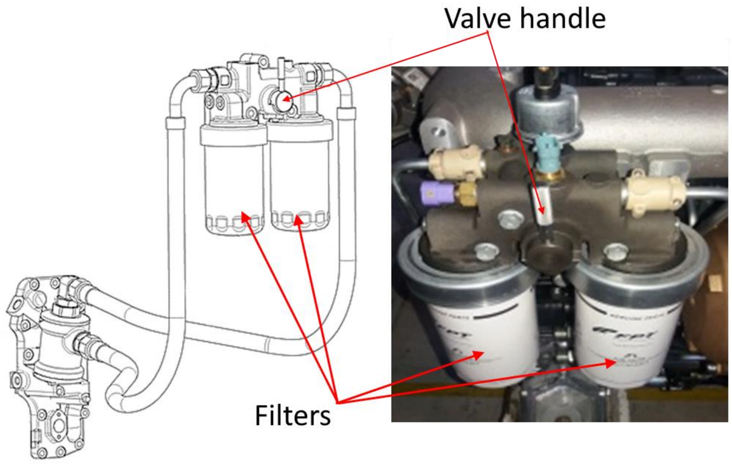


Figure 11. Duplex type filters for avoiding flow interruption during maintenance operations.

4.2.7 Injection and Fuel High-Pressure Pump

The maximum engine power may not be fulfilled while maintaining the same injected quantity and therefore, a new injector with enhanced flow capability is to be used. As a consequence, also a new fuel pump delivering a higher fuel flow rate shall be needed, providing the required fuel rate for maximum power and increased flow capability for the nozzle.

4.2.8 Fuel High-Pressure Piping and Common Rail

By increasing the fuel pressure with respect to the base engine, it is possible to have an increment in the amount of injected fuel (fuel delivery). Consequently, the common rail pressure regulation valve is also to be changed.

For safety purposes, it is required to adopt a solution to collect a potential fuel leakage along the high-pressure pipes line such as a jacketing system and to connect the jacket to a collector tank through an additional fuel pipe. This solution is not required on recreational applications but is usually requested on light commercial or heavy-duty applications for Naval Register Classified Engine as in the case of the 16L – 600hp marine diesel engine classified by the RINA Society.

The specific solution applied to the 16L - 600hp marine engine for collecting a potential fuel leakage from the high-pressure fuel line was done by adapting a collector box shown in Figure 12 containing the section of fuel pipes from the high-pressure fuel pump to the common rail. This leakage collector box fulfils two important functions: to act as a shield, avoiding fuel to come in contact with other hot components in case of fuel leakage and to collect the leaked fuel. In this case, it's also required to adopt a level sensor in the collector box to detect the presence of a fuel leakage and to avoid going over the collector tank maximum level.

Another solution was implemented for the common rail and for the high-pressure fuel pipes section from the rail to injectors. These were encased inside the cylinder head thus, the rail jacket is not required because the potential fuel leakage is collected in the cylinder head and consequently in the oil sump. A level sensor located in the oil pan is adopted to detect the fuel leakage.

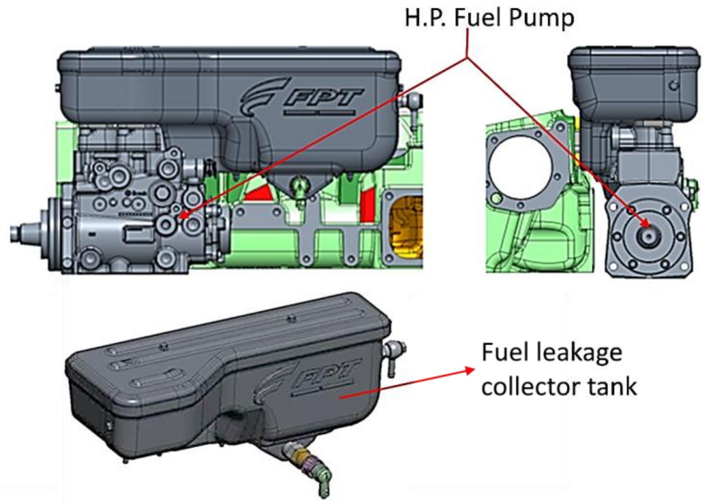


Figure 12. Fuel leakage collector box.

4.2.9 Exhaust Manifold and Turbocharger

According to regulation requirements, the addition of a heatshield is required on both the exhaust line and the turbocharger to limit the components external temperature. In this specific case, the exhaust manifold and turbocharger are cooled by means of a water-jacket in which the circulating engine coolant regulates the temperature. For minimizing the risk of leakages and of coolant mixing with the exhaust gases, the design of the manifold foresees a monolithic solution. The use of intermediate adapters between exhaust manifold and turbocharger is avoided and thus, the exhaust manifold and turbine water jackets operate separately.

4.2.10 Oil Pump

For providing an easier installation and preventing any possible leakages, the oil pump has been integrated into the engine. The pump is driven by the gear train of the timing system. The flow rate capability is to be reviewed and increased to keep the pace with the oil demand increase due to the higher heat rejection needed.

4.2.11 Front End Accessory Drive (FEAD)

Since the RINA regulations require that the exposed belt drivers, chain drivers and rotating parts in general are covered to avoid reaching through the regular openings and thus, preventing injuries during the engine normal operation, a FEAD cover was designed. The guard is to have openings to guarantee a correct aeration avoiding excessive local temperature increase and shall be designed to contain a normal but not necessarily catastrophic failure of the guarded parts as in the case of a belt breakage.

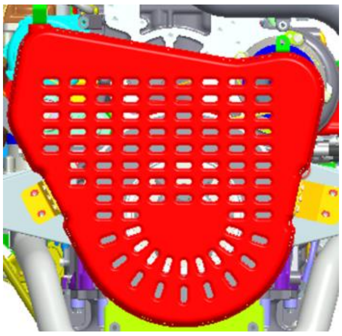


Figure 13. FEAD cover with openings.

4.2.12 Marine Engine Monitoring

Diesel engines for main propulsion in marine applications are to be equipped with extra monitoring equipment for safety reasons. Additional, and in some cases redundant sensors, are to be connected to a set of visual and/or audible alarms and indicators which must be fitted at a normally attended position. There's a dedicated safety panel called RINA panel which is directly connected with some of the additional sensors. This panel has also the specific function to stop the engine in case of an overspeed event (monitored by a specific additional speed sensor) and in case of very low oil pressure (monitored by specific additional oil pressure sensor). From the RINA panel is possible to control basic functions of the engine as the starting and turning off; and develops also a throttle function. It must be also able to provide visual indication of the orders given to the engine and its responses and; is to be equipped with an alarm informing the presence of a mismatch between those orders and their responses.

The additional sensors installed in the engine as part of the marinization process and according to the RINA regulations are:

4.2.12.1 Speed governor

A speed governor is implemented to ensure that the engine speed is automatically controlled in case of overspseeding conditions. The speed governor is to be adjusted so that the engine does not exceed its rated speed by more than 15%.

4.2.12.2 Overspeed sensor

Is a magnetic pickup sensor used to evaluate the engine rotational speed. This measurement is a redundant one since is an additional sensor, different from the one already present in the engine and connected to the main engine ECU. The overspeed sensor is mounted on the flywheel housing, as illustrated in Figure 14, for making measurements at the flywheel cogs. The sensor is directly connected to the RINA panel and must command the engine shut down as a safety function in case of exceeding the 20% of the engine rated speed (indicated as high engine speed in Table 3)

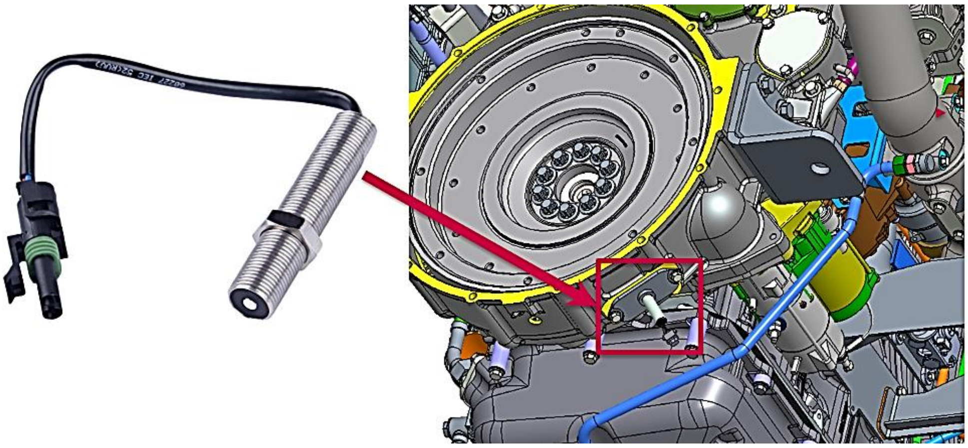


Figure 14. Overspeed sensor located on the flywheel housing

4.2.12.3 Engine oil pressure sensor

The oil pressure is monitored in order to avoid situations where the engine oil pressure decreases to dangerous low levels such as in cases of lubricating oil supply failure, which could rapidly lead to complete engine damage. Therefore, and as shown in Table 3, an indication accompanied with an alarm must be present in the case of a low oil pressure condition while a safety engine shut down action is preformed when a very low oil pressure condition is encountered. An additional (redundant) pressure sensor for safety considerations is required by the RINA directives and is to be directly connected to the RINA panel and should command the engine shut down in case of a very low oil pressure condition.

4.2.12.4 Engine oil level sensor

Is an ultrasonic sensor type used for controlling the maximum oil level present in the oil sump that could rise as a consequence of a fuel leakage coming from the common rail, or from the high-pressure pipes connecting the rail to the injectors, since these components were designed to be enclosed by the valve cover and therefore, in case of a fuel leakage from the high pressure fuel line, the fuel mixes with the oil and ends up in the oil sump.

The sensor provides a continuous monitoring of the oil level throughout the period of time the engine is operated and if the maximum level is reached, the ECU gives an alarm signal to prevent the engine damage due to the high oil dilution present in the oil sump (see Figure 15), probably as a consequence of a fuel leakage from the H.P. fuel line. The sensor is mounted into the oil pan from below as illustrated in Figure 16.

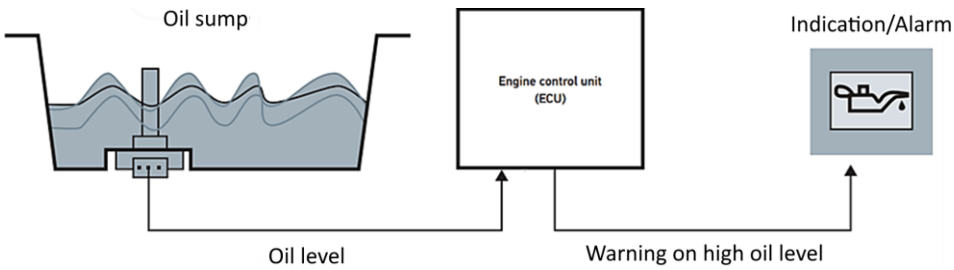


Figure 15. Oil level sensor alarm working principle. Modified from [11].

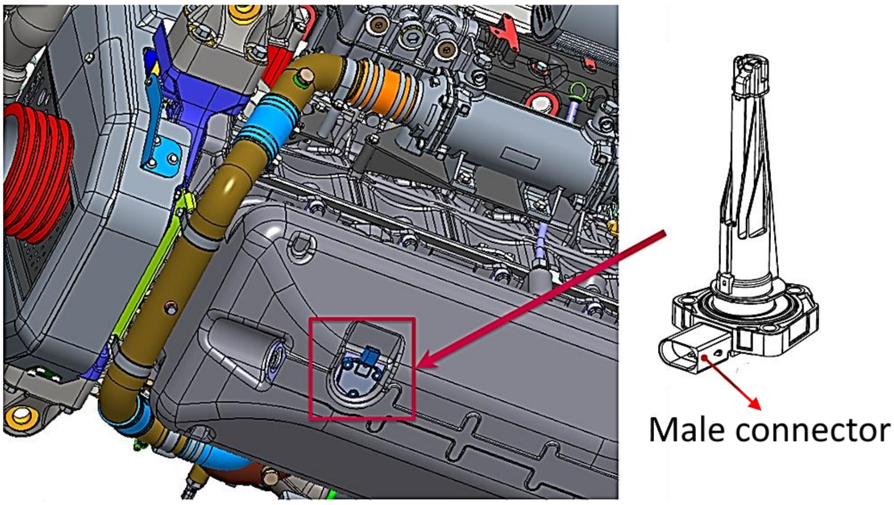


Figure 16. Oil level sensor located at the bottom of the oil sump.

4.2.12.5 Engine coolant level

A sensor must be installed in the engine coolant expansion reservoir in order to measure the coolant level. It must not be directly attached to the safety panel for value readings, but the alarm must be triggered if the level reaches a dangerous low level.

4.2.12.6 Fuel level sensor

If a potential fuel leakage occurs from the section of high-pressure fuel pipes between the H.P. pump and the common rail, the leaked fuel ends up in the fuel leakage collector tank (see Figure 12). A sensor to control the level of the fuel present in the collector box should be used and in case of reaching the maximum level, a signal is sent to the engine

ECU to turn the alarm/indicator ON and evidence the presence of a fuel leakage in the respective section of the high-pressure fuel line.

4.2.12.7 Fuel pressure sensor

To monitor the filter saturation level or possible clogging in the fuel line, the pressure before and after the switchable double fuel filter is measured.

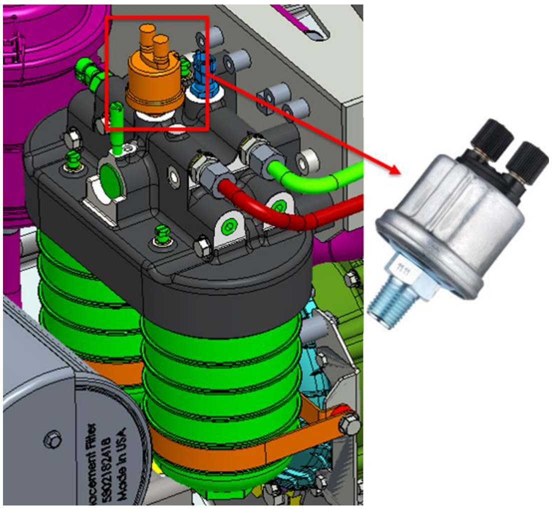


Figure 17. Fuel pressure sensor location.

All the monitoring ad safety equipment that is to be fitted on marine diesel engines for main propulsion according to the RINA regulations is detailed in Table 3. The set of alarms and indicators that are to be activated when required, depending on each system, are to be visual and audile and to be located on a normally attended position (main boat control panel and engine compartment).

Table 3. Monitoring of main propulsion marine diesel engines.

	Monitoring		Safety - Automatic Control
Identification of System Parameter	**Alarm**	**Indication**	**Shut down**
Fuel pressure after filters	–	Local	
Leakage from high pressure pipes – Fuel level	High	–	–
Lubricating oil main gallery pressure	Low	Local	
	Very Low	–	X
Lubricating oil main gallery temperature	–	Local	
Engine coolant inlet pressure	Low	Local	
Engine coolant outlet temperature		Local	
Coolant level in the expansion reservoir	Low	–	–
Exhaust gas temperature. Turbine inlet.	–	Local	
Engine speed	–	Local	
	High	–	X
Fault in the very low oil pressure sensor	X	–	–
Fault in the electronic governor system (overspeed governor)	X	–	–

NOTES :

- X = Function is required
- The "local" indication means that the read value for the parameter must be displayed on the RINA safety panel.

4.3 Engine Fluid Circuits

Unlike automotive applications where air is the primary element used for extracting engine heat through the radiator, in marine applications the engine is cooled by two separate but linked systems. An open system, in which water is taken from and returned to the sea, is used to control the temperature of the coolant which circulates through the engine in a closed circuit. The coolant of the closed system is cooled in a water to water heat exchanger which replaces the radiator normally seen in vehicles.

4.3.1 Open Cooling Circuit (Seawater Circuit)

The open cooling circuit is composed by a heavy duty - high flow seawater pump integrated into the engine and driven by the accessory gear train. The seawater is taken through a seacock valve (Figure 18.a.) located at the bottom of the hull which filters out large debris before entering into the suction pipe. Then, the seawater passes to the strainer (see Figure 18.b.) where it is filtered from smaller debris that managed to pass though the seacock and that could damage the pump's impeller or obstruct the heat exchangers. The pump draws the water overboard and circulates it throughout the cooling circuit which is composed of two heat exchangers, one for the cooling of the intake air coming from the compressor and one for reducing the temperature of the engine coolant. Finally, the seawater is discharged out of the engine to the sea. Figure 19 shows a schematic diagram of the open cooling circuit just described.

Figure 18. a) Seacock valve placed under the sea waterline and b) Strainer to filter out smaller debris.

Considering that the compressed intake air is cooled in a seawater-to-air intercooler, the global heat transfer coefficient could be considerably high, especially during engine cold start/cold operations and when the seawater is at very low temperature. This could lead to an excessive low temperature of the intake air which will produce white smoke out of the exhaust. Therefore, an electrical resistance is mounted at the exit of the intercooler, just before the intake manifold to heat the incoming air when needed. The white smoke is a mixture of partially vaporized droplets of water and fuel, the former being products of combustion and the latter arising because the droplets were incapable of reaching the temperature needed for ignition.

4.3.2 Closed Cooling Circuit (Engine Coolant Circuit)

A water pump, driven by the accessory gear train in front of the engine, circulates a glycol-based coolant in a closed-loop circuit. The coolant temperature is regulated in a water-to-water heat exchanger. After exiting this heat exchanger, the coolant flow is divided into two different cooling lines: one enters the engine cooling down the engine block, the cylinder head and the exhaust manifold. The other line is in charge of controlling the engine oil and turbo temperatures. After exiting the turbo, the coolant joins the first line in the return passage of the exhaust manifold as depicted in Figure 19.

The engine coolant absorbs heat rejected by the previously described components and passes it to the seawater circulating in the open circuit through the water-to-water heat exchanger. Figure 19 shows a schematic diagram of both open and closed cooling circuits.

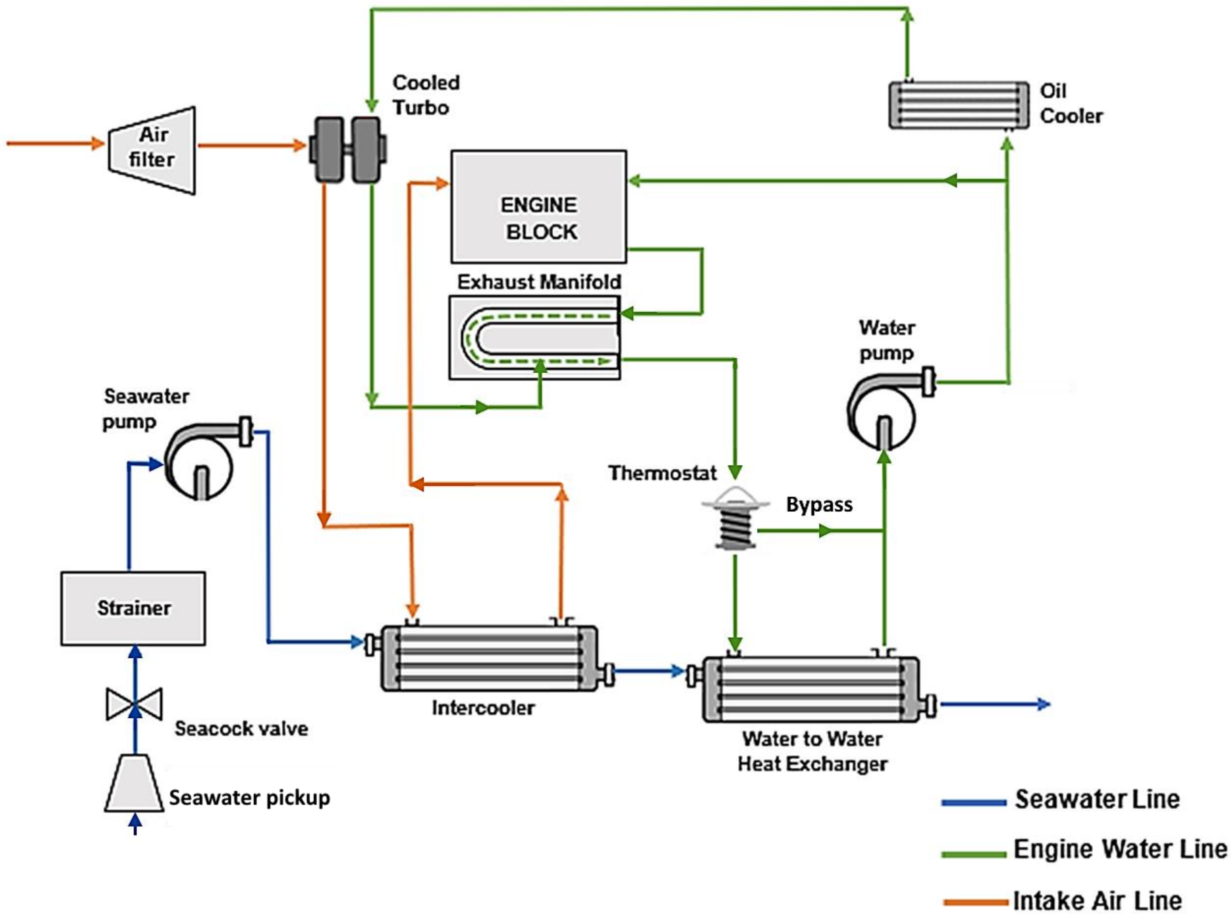


Figure 19. Seawater, engine coolant and intake air circuits.

4.3.3 Fuel Circuit

The fuel inside the engine fulfils two basic functions: one as source of energy to be transformed into useful power and another one for cooling down and lubricating the components of the injection system and the combustion chamber.

The system is composed of a dual-feed fuel pressure pump with capability for high fuel volume/flow demand found in high displacement engines, ensuring that enough fuel is delivered throughout the engine's entire operating range. The pump consists of two pumping stages at two different pressure levels. The low pressure transfer pump supplies the fuel from the fuel tank into the pumping chambers of the high pressure pump. The fuel volume quantity fed into the high pressure pump is regulated by the fuel metering control valve which is commanded by the ECU. Mounted on the fuel pump, there's also a pressure regulator valve which controls the fuel pressure delivery to the common rail and opens the return line for the excess fuel to come back to the fuel tank. The fuel is supplied to the common rail at high pressure. The rail stores the fuel at a determined pressure level and damps the pressure oscillations caused by the non-continuous fuel supply delivered by the high pressure pump and by the effect of the opening and closing of the injectors' nozzles during the injection process. Fitted at the end of the common rail there's a pressure limiting valve which relieves the rail's pressure in case of an abnormal high pressure level. This valve opens a fuel return port and the excess fuel is relieved back to the fuel tank through the fuel return line. Finally, from the common rail the fuel is distributed to the individual injectors. Each injector has its own return line which joins the main return line that goes back to the fuel tank. The injection system for the FPT 16L -600hp engine was developed together with Bosch. A schematic representation of the system is shown in Figure 20. It is possible to see that the common rail together with the high pressure pipes from rail to injectors are located inside the valve cover and that the high pressure pipes from the high pressure pump to the common rail are enclosed by the fuel collector box. Both solutions are developed according to the RINA requirements for fuel leakages as mentioned in the numerals 2.1.3 and 4.2.7.

The management of the common rail diesel fuel systems is operated on a closed loop basis. The ECU carries out many calculations to precisely control the fuel quantity and the injection timing though the electrical command sent to the injectors.

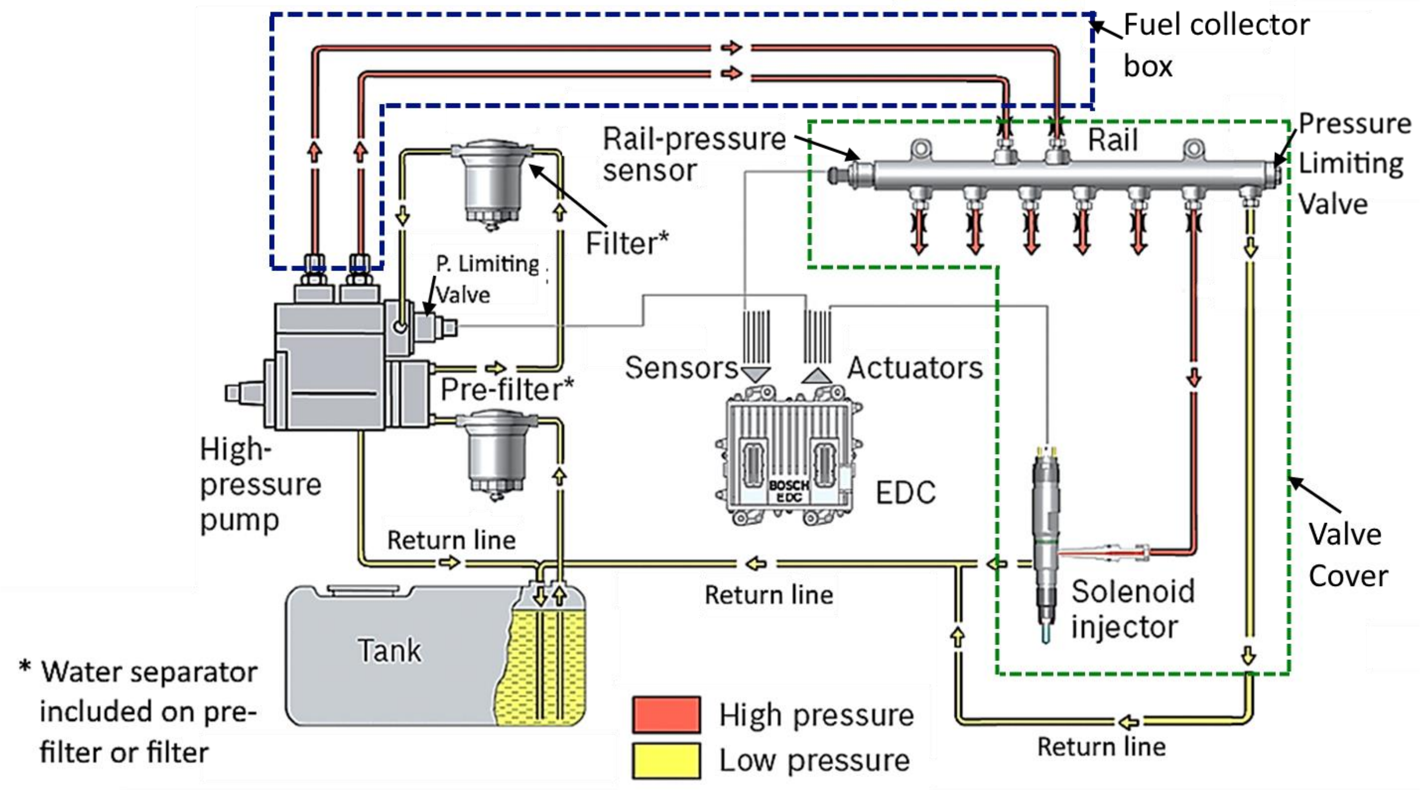


Figure 20. Common-rail diesel fuel injection system. Schematic adapted from Bosch diagram.

5 ENGINE TEST CELL

The FPT industrial’s Turin testing facilities are composed of 74 test cells for the development and certification of engines and after-treatment systems, as well as the development of axles and transmissions. The engine test cell consists of two main environments, the control room and the engine bench room. The general test facilities layout consists of multiple test benches built in side-by-side configuration that have a common undivided corridor. In some cases the test cells have corridors running at both ends. The control rooms for the different test cells are located in the frontal one together with a personnel access door, while the rear corridor provides the access for the engine and large/heavy equipment as shown in Figure 21; this arrangement keeps the control room quiet and free from external disruptions caused by the movement of machinery. Other type of arrangement considers only one frontal common corridor where the access for both personnel and equipment is located (see Figure 22).

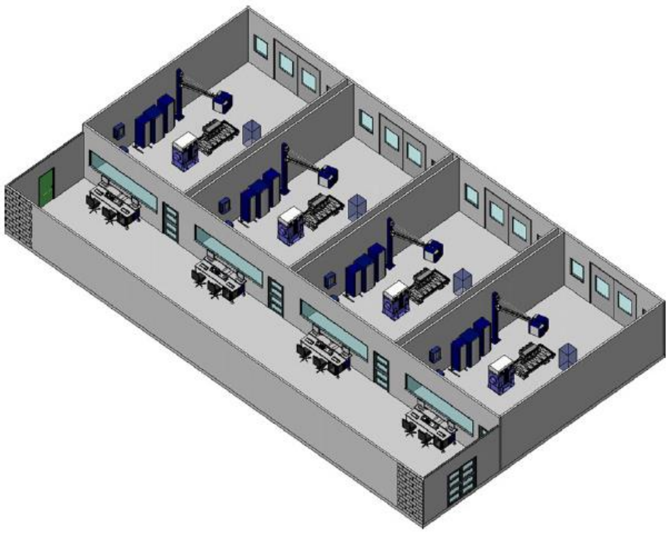


Figure 21. Test benches with common fontal and rear corridors [12].

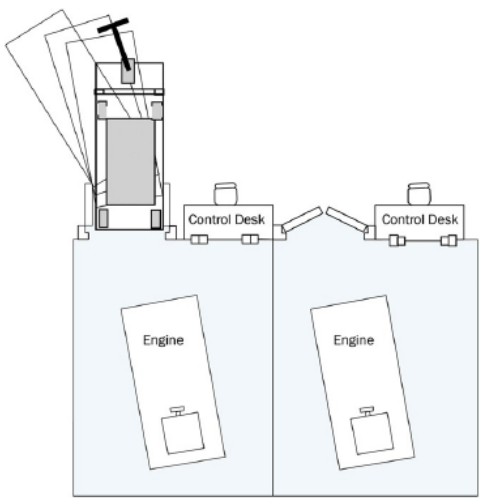


Figure 22. Test benches arrangement with frontal corridor only [12].

The engine present in the test cell is isolated from the control room and in general from the external facilities by means of special doors, walls and windows. The engine room acts as a hazard containment box meeting requirements for noise attenuation and fire resistance & containment. The confinement assures the safety of the personal in case of a fire event, explosions or the presence of flammable or dangerous gases such as exhaust or oil & fuel vapors. There's a window located in front of the control desk which separates the control room from the engine room allowing the engine to be monitored also from a visual point of view even if there are video cameras installed inside the test bench. The glass (including its frame) on the cell side should have the highest fire resistance and the control room side glass should be the most impact resistant.

Besides the physical barriers separating the control space from the unit being tested, there are some other safety systems present in the facilities. These systems include a fire and gas detection systems which automatically interrupt the electric and the fuel supply, shutting down the engine. The detection sensors are connected to the fire-fighting system and to a set of visual and audible alarms to alert about possible hazards in the powertrain facilities.

The test cell floor is provided with arrangements for bolting down the engine which was previously rigged. All the engine and bench necessary instrumentation and sensors are installed before performing the tests. These include pneumatic and electric type sensors that are connected to the cell signal management system which in turn is connected to the monitoring and control systems.

5.1 Test Cell Devices and Systems

Besides the instrumentation presented above, the test cell is provided by other different systems and devices used for monitoring and controlling the parameters needed to assure the correct working and development of the engine tests. These systems are used for the engine calibration, optimization and for controlling the test bench parameters (e.g. engine brake, seawater temperature, etc.). The system and devices present in the test cell are:

- Air ventilation
- Combustion air supply
- Exhaust gases extraction
- Fuel supply
- Engine coolant supply
- Test cell cooling water circuit
- Seawater supply
- Dynamometer (brake)
- Brake cooling water

The different inflows and outflows to and from the test cell are shown in Figure 23. It must be highlighted that the electricity is provided into the test cell for ECU, actuators, sensors and for the dynamometer operation. Considering that the dynamometer for this particular case is of the passive brake type, no electricity is taken out from it and put into the facilities electrical grid and therefore, the electricity supplied to the passive brake is transformed into heat that is finally rejected to the dynamometer cooling water.

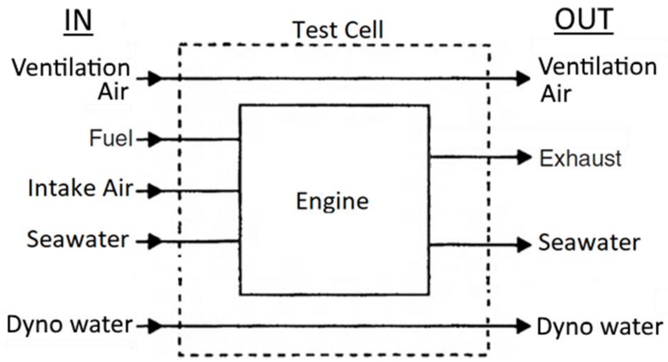


Figure 23. Fluid flows to and from the test cell.

5.1.1 Dynamometric Brake

The engine brake is an electric machine capable of producing the resistant torque that the engine would, for example, encounter when used for propelling the vessel. The dynamometer present in the test cell employed for the tests developed in this thesis project is of the Eddy Current type and uses the induction principle to develop resistant torque and dissipate power (Figure 24). The machine consists of a toothed disc coupled to a rotor shaft moved by the internal combustion engine as shown in Figure 25. The disc rotates with a narrow clearance (air gap) between magnetic poles placed in the stator. When a current is applied to the coil that is circumferentially wound on the stator, a magnetic field passing though the poles and rotor is generated as shown by the blue line in Figure 25. The rotation of the disc teeth in the magnetic field produces a continuous variation of the air gap and consequently the magnetic flux changes; this in turn produces eddy currents in the front areas of the stator poles. The eddy currents build up an opposite magnetic field that applies a braking moment to the rotor which tries to reduce its rotational speed, but the torque supplied by the internal combustion engine attached to it maintains the RPM. Eddy currents induced in the stator are dissipated in producing heat and therefore the dynamometer needs adequate cooling. Heat is transferred to cooling water circulating through passages in the stator, while some cooling is achieved by the radial and convective heat flow to the air present in the gaps between rotor and poles. The dynamometer load is controlled by varying the current supplied to the stator field coils with the possibility to apply rapid load changes. Eddy current dynamometers are not able to develop motoring torque so it is not possible to develop motored engine tests.

Figure 24. Dynamometric brake used in the tests.

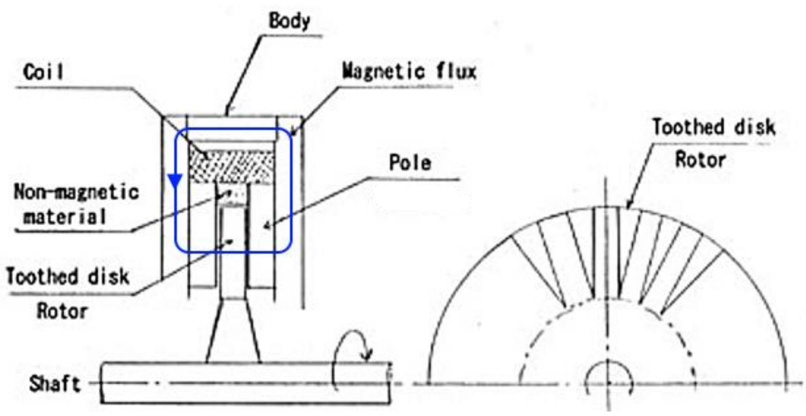


Figure 25. Working principle of the Eddy-current dynamometric brake.

The braking torque is transferred via a bearing mounted body (carcase) to the load cell present at the end of a lever arm, thus the measured force is proportional to the braking moment. Knowing the magnitude of the lever arm and the rotational speed, the engine's torque can therefore be measured. The effective length of the arm from which the force F is applied to the load cell is the value r shown in Figure 26, measured from the center of the rotor. It is necessary that the force F acts perpendicular to that arm and therefore, the centers of its connections must be must be horizontal. Torque can be calibrated and measured in either rotational direction with the load cell in either compression or tension. The torque applied by the engine on the dynamometer is T:

$$T \;= F\,r \tag{20}$$

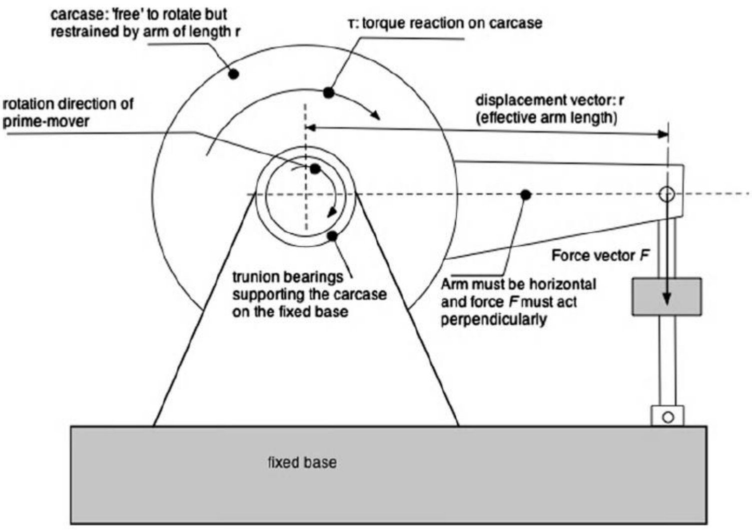


Figure 26. Schematic of a dynamometer [12].

Torque is the engine's ability to do work, whereas power is the rate of this work that is done. The power P delivered by the engine and absorbed by the dynamometer is the product of the torque and angular speed:

$$P[W] = 2\pi n\ [\text{rev/s}] \cdot T\ [\text{Nm}] \tag{21}$$

The value of measured engine power as described above is called brake power. This is the usable power delivered by the engine to the load.

Rotational speed of the dynamometer is measured either by using an optical encoder system or, a toothed wheel and a pulse sensor. The dynamometer control system is provided with a closed loop control system by giving feedback signal from load cell for torque and from the RPM sensor for speed. The dynamometer's closed cooling circuit is equipped with a flow control unit and thermal switches in the water outlet. The flow control unit controls a minimal amount of cooling water and the thermal switches a maximally permissible cooling water outlet temperature. Both systems avoid to a great extent damages due to insufficient cooling.

5.1.2 Test Cell Ventilation

An engine test cell is a very demanding environment with large amounts of generated power, high hot fluid flow and high electrical power supply; everything circulating and generated within a relatively small and confined space. These characteristics, together with rapid variations of the thermal load, give rise to surfaces at high temperature and a modification of the test cell environment. Even if there's a dedicated air circuit for the combustion air, the condition of the air surrounding the engine, temperature, pressure, and humidity has an indirect impact on the engine's performance and power output.

The sources of the heat present in the engine room are the engine itself, the eddy-current dynamometer that rejects the heat generated in it to its dedicated cooling medium passing inside the cell; the electrical power to lights, fans and the instrumentation in the test cell; the heat exchangers used for the seawater and fuel temperature control and the long lengths of not isolated pipework holding hot fluids. All of them appear as heat transmitted to engine room environment which in turn is controlled by transmitting the heat to the ventilation load and by regulating the quantity, temperature, and in some cases the humidity of the air passing through the test cell.

For the circulation of the ventilation load from the cell, forced ventilation is provided using conditioned outside air. Fans draw air through the cell entering through two low-level inlet grilles placed on opposite walls near the engine, and then the air is extracted though high-level outlet ducted system on the cell's celling as shown in Figure 27. Ventilation systems for test cells not only remove heat but also prevent the build-up of dangerous levels of gases and vapors. This is ensured by externally conditioning the air from the cell, by guaranteeing sufficient air flow through the cell even at times of low thermal load, and by having the cell space running at slightly below ambient pressure; thus preventing engine exhaust and hydrocarbon fuel fumes from being pushed into the control and other work areas.

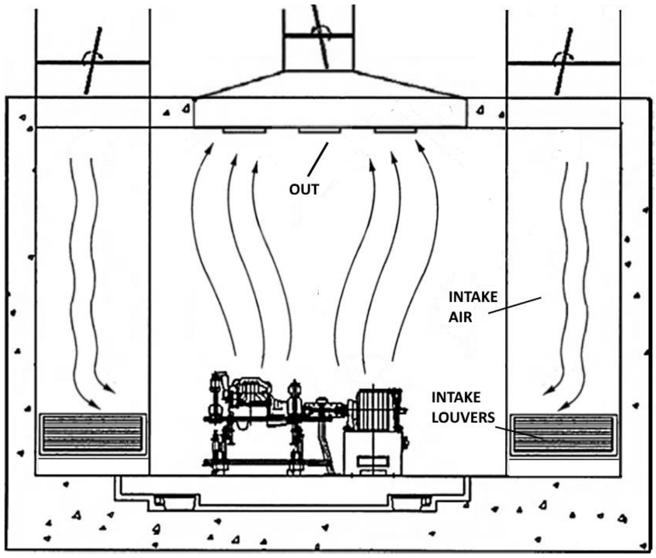


Figure 27. Test cell ventilation system layout. Modified from [12].

5.1.3 Secondary Fluid Circuits – Dynamometer cooling water and engine seawater

The temperature of some liquids flowing through the engine such as oil, engine coolant and fuel are not controlled externally, instead their temperature is regulated by the engine itself. The oil temperature is controlled in the oil cooler where heat is exchanged with the engine coolant as shown previously in Figure 19, the engine coolant in turn enters the water-to-water heat exchanger where its heat load is passed to the seawater and finally, the fuel from the return line is mixed with the fuel present in the test cell fuel tank. The seawater and dynamometer fluids are considered as secondary circuits and their inlet temperature to the engine is controlled and maintained (if needed) with the aid of an external primary circuit where water is supplied by the test facility at a determined and constant temperature, pressure and volumetric flows. This allows the system to provide adequate cooling capacity though the heat exchangers where it encounters the secondary circuits as shown in Figure 28.

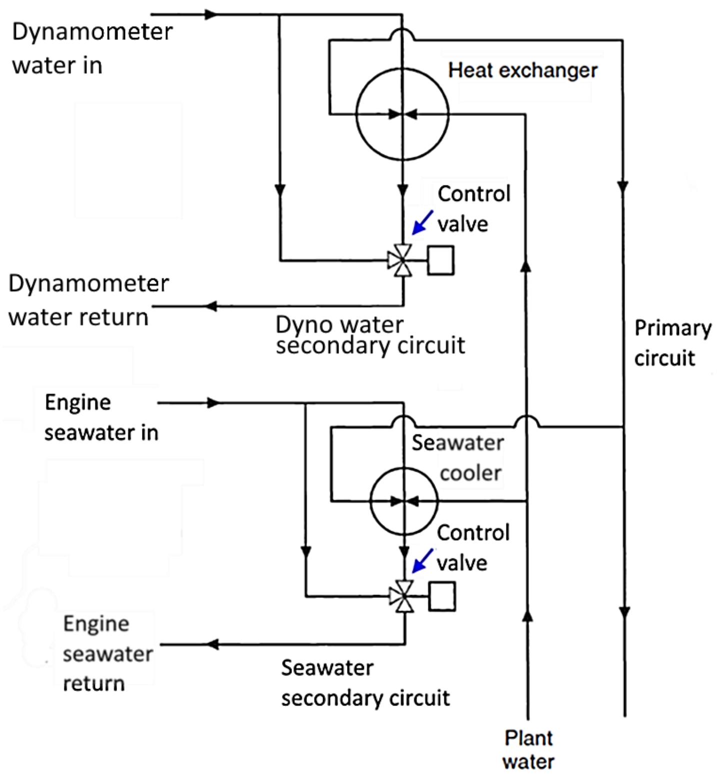


Figure 28. Test cell and engine primary and secondary cooling circuits layout.

There are different automatically PID controlled valves for each secondary circuit (depicted in Figure 28). They regulate the fluid flows present in the secondary circuits to each heat exchanger where heat is rejected to the facility water present in the primary circuit. Valves are required for each sub-circuit because they have their own particular thermal load and resistance, and therefore require a specific primary system flow rate. By regulating the flow rate using the PID controlled valves the temperature of the fluid present in the secondary circuit is controlled allowing the temperature to be maintained at a target value during the whole duration of the tests. As a matter of fact, the control of the opening or closure of the valves is done by a closed-loop proportional-integral-derivative (PID) control system depending on a given secondary fluid temperature target value. The feedback is done by means of the fluid temperature measurement performed in real-time by a dedicated sensor. Figure 29 shows the engine inlet seawater temperature variation over time while performing the homologation tests. It can be seen that the temperature value oscillates around the target value requested by the specific test conditions, due to the control performed on the seawater flow though the heat exchanger by acting on the control valve.

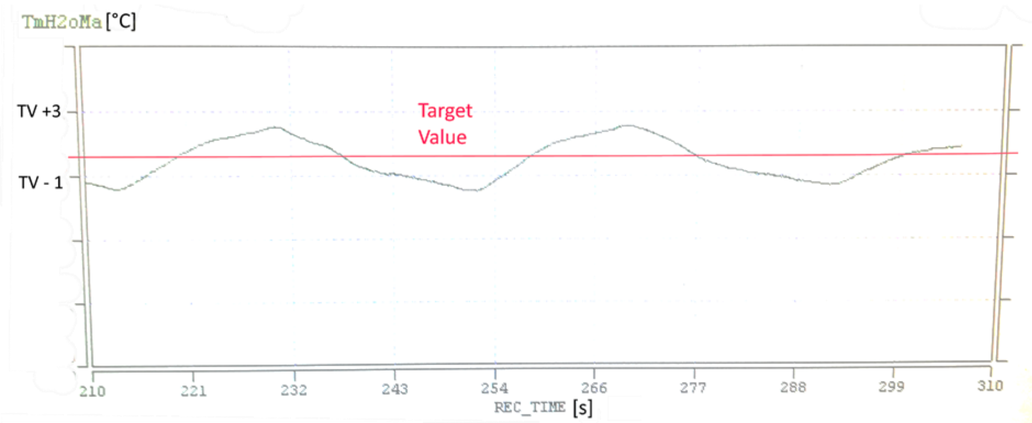


Figure 29. Recorded seawater temperature at the engine inlet (TmH2oMa) over time. (TV: Target Value)

5.1.4 Fuel Consumption Measurement

It is possible to perform two types of fuel consumption measurement: A "static" one where the time taken by the engine to consume a known mass of fuel is measured and another one determining the fuel mass flow in a continuous way. The static and continuous methods use different devices for the fuel consumption calculation, the first one employs a gravimetric gauge and the latter employs a Coriolis Effect flowmeter.

5.1.4.1 Gravimetric gauge (fuel balance) – discontinuous method

It meters the fuel mass rather than a fuel volume and is used to determine the specific fuel consumption. It consists of a vessel mounted on a weighing cell from which fuel is drawn by the engine while it is also simultaneously refilled by the engine fuel return line (Figure 30). The vessel is automatically filled and the fuel is weighed at the start and at the end of a given time interval previously defined by the user, calculating thus the engine fuel consumption. The main advantage of this system is that it is not necessary to estimate the fluid density since it is directly weighed and therefore the measurement is not affected by the variations in temperature with its consequently density variations.

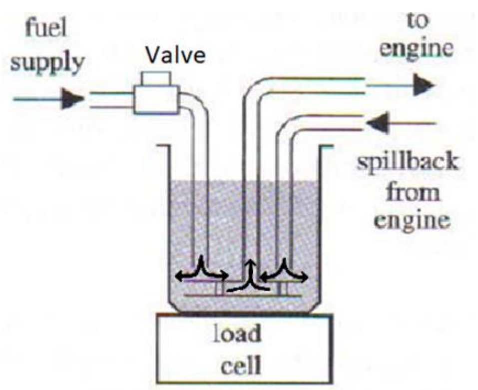


Figure 30. Gravimetric - direct weighing fuel gauge.

5.1.4.2 Coriolis Effect Flowmeters – Continuous method

The fuel consumption measurement employed in the tests was of the continuous type. It consists of two parallel U-shaped tubes that oscillate due to the action of an electromagnetic driver as depicted in Figure 31. This drive coil forces the tubes to vibrate in opposition to each other at the natural resonating frequency of the tubes. Two pairs of opposite sensors are positioned upstream and downstream of the tubes (Figure 31), and as they oscillate, the voltage generated from each pair of sensors creates sinewaves which indicate the motion of one tube relative to the other. When the fuel enters the flowmeter, the flow is divided by the splitter near the inlet into two streams; half of the fluid passes through each tube and are then recombined at the meter's outlet. When no flow is passing inside the tubes, the vibration caused by the driver coil results in identical oscillation of the tubes and therefore the displacements measured by the two sensing points result in two identical sinewaves. But when fuel is flowing through the tubes, Coriolis forces are induced causing the tubes to twist in opposition to each other (Figure 32) while they continue vibrating and as a result, the sinewaves present a small phase shift in the relative motions. It is important to highlight that the deflection of the tubes caused by the Coriolis force only exists when both axial fluid flow and tube vibration are present. The time delay (Δt) between the sinewaves is measured and is directly proportional to the mass flow rate through the tubes.

The calculation of the fluid's density can also be simultaneously performed by determining the tubes oscillating frequency. When the density of the fluid inside the tubes changes, the vibrating frequency of the tubes also changes and since the density of the fluid contained in the tubes is the only variable affecting the oscillating frequency, the density can be found. Both measurements of fuel's mass and density allow the calculation of its volumetric flow too.

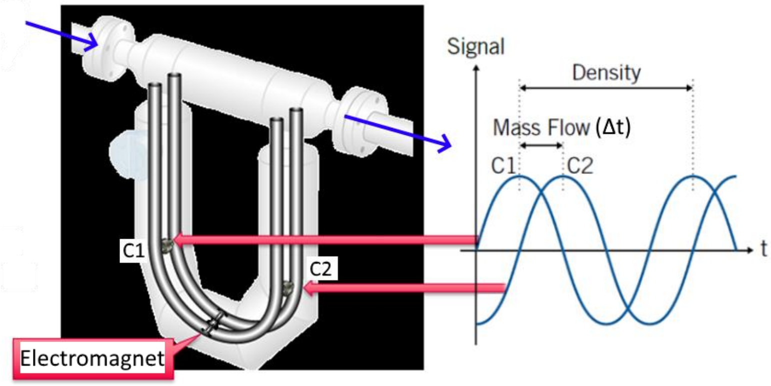


Figure 31. Coriolis effect flowmeter

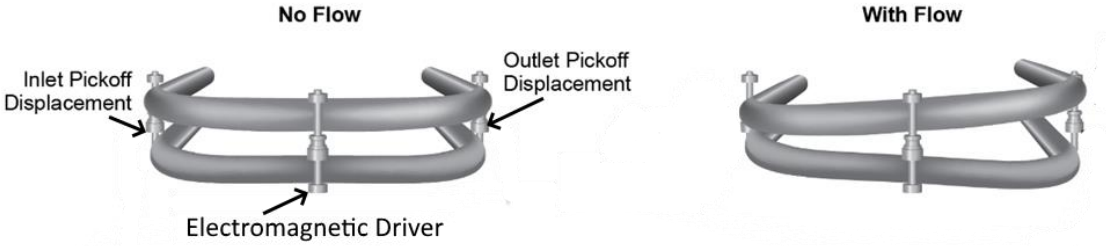


Figure 32. Tubes twisting caused by the Coriolis forces.

5.2 Control Room

The control and monitoring of all the devices and systems present in the engine test cell is performed in the control room. All the sensors and metering devices being analog, electronic or pneumatic, are connected to an AVL box which sends the signals values in V or mA to the different software used and therefore to the PC present in the control room.

The primary purpose of the control desk is to permit the operator to directly regulate either the engine and/or the dynamometer. The system must also allow him to monitor the functions of a control system while it runs automated test sequences. Figure 33 shows some main elements that can be found in the control room. The controls that govern the torque applied to the unit under test and its rotational speed are disposed centrally to the layout. The speed and torque display should be prominent in order for the operator to judge rate of change of the different parameters and “drive” the system if needed. Other important measurements taken in real time by the sensors positioned on the engine such as temperatures and pressures of the different fluids circulating through the engine (seawater, oil, exhaust gases, coolant, among many others), and also parameters that were not directly measured and were instead calculated by means of a formula or relation e.g. engine power; are displayed in real time on the control room screens in order for the operator to judge the rate of change during initial start-up and while running the tests. Together with these parameters, also possible warning signals indicated by the control software are shown.

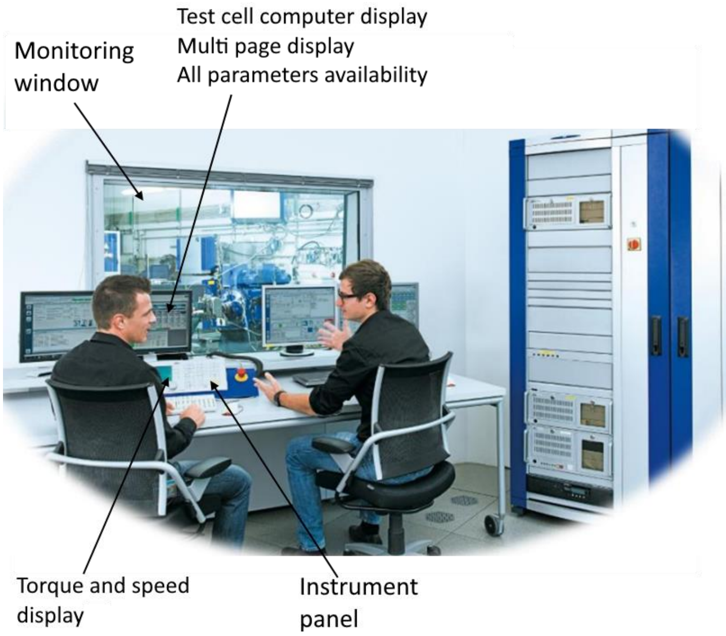


Figure 33. Powertrain test cell control room.

As indicated before, disposed centrally to the control room layout it is located the instrument panel where manual control of the test bed services is enabled. It shows the running status of each major circuit and should be visible from the operator’s normal work position. By means of the instrument panel, the operator can directly control different tasks including:

- The starting and stopping of the engine as depicted in Figure 34.

- Selecting the control characteristic for the test run.

  The developing of engine tests is done by imposing a sequence of desired values of engine speed and torque that are regulated by acting on the throttle actuator and the dynamometric brake, respectively. Speed control consists in the manipulation of the engine’s power output, usually by way of its throttle system where an electric signal is sent to the engine control unit which actuates on the injected fuel quantity. Torque control refers to the dynamometer torque absorption setting.

  There are different control characteristics modes for both engine and dynamometer. The position mode (%) which specifies the degree of throttle opening that regulates the fuel injection, the constant speed control mode (ω = K) and the constant torque (T = K) regulation mode. It is possible to combine couples of these characteristics in order to obtain different test operation modes. For example in the “Torque (T = K) / Position (%) Mode”  shown in Figure 34, the torque absorbed by the brake is set to a constant value that is maintained invariable independently of the engine speed with the help of the automatic controller on the dynamometer while the engine speed is regulated actuating on the position (throttle setting). The system employs a closed-loop control. Another example could be the  button in the characteristic selection area of Figure 34 where the **x** represents any parameter of interest selected by the user such as fuel consumption per hour, pme or turbo outlet temperature. Therefore, the control system will load the engine until having the target turbo outlet temperature maintaining the pre-set engine speed.

- Another task of the instrument panel is setting the demand (imposed) values and visualizing them together with the actual values.

  The three knobs present in the instrument panel are used to set the demand values. The variable adjusted with each knob depends on the control characteristic (mode) chosen for each test. Directly above the knobs, the demand (set) values are displayed while the current values are visualized above in the panel screen as shown in Figure 34 and Figure 35. Each knob has its own assigned function:

  a) Knob to set the dynamometric brake values (torque)
  b) Knob to set the engine values.
  c) Configuration-dependent, mainly used for speed difference and torque (e.g. test beds with two or more dynos). This knob was not used in the tests developed in this thesis.

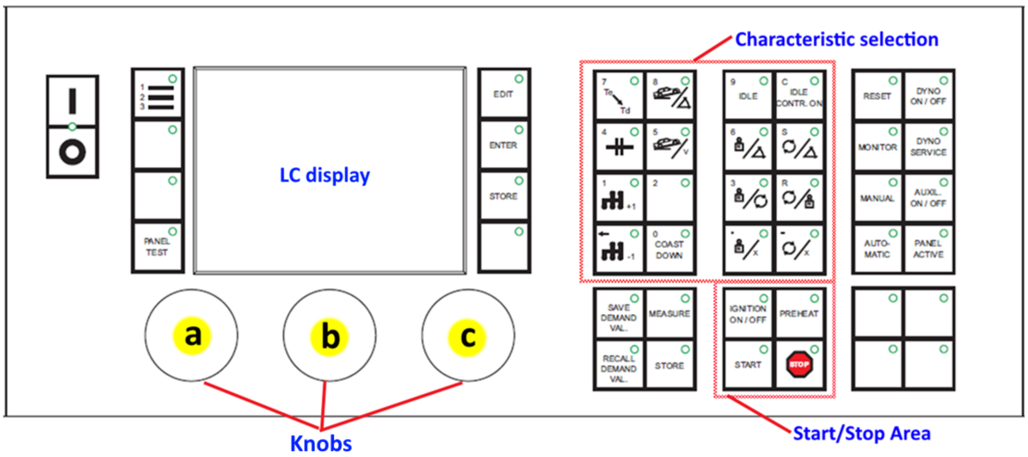


Figure 34. Operating panel.

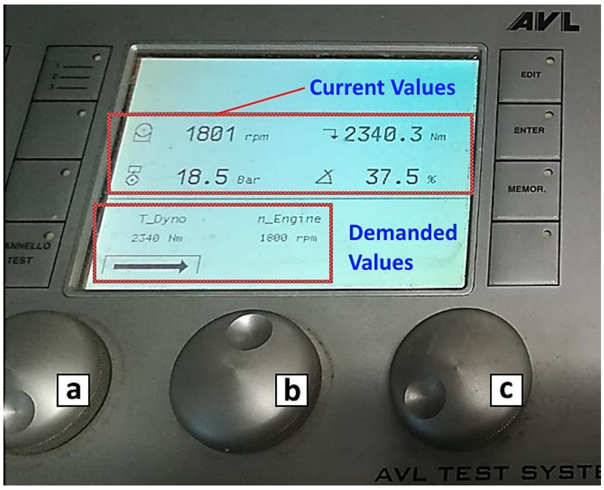


Figure 35. Knobs and display of the instrument panel

5.3 Test Control Software

There are two main necessary systems to be able to carry out the development of tests and the calibration of the engine:

- Systems for the control of the test bench, i.e. engine brake, devices for the regulation of the test room parameters and all the cell sensors. The control of the cell systems where all the tests were performed was done though the software PUMA by the AVL company.
- Systems for control and calibration of the engine control unit (ECU). Performed using INCA software by ETAS.

All the measurements taken during the tests are recorded and saved in a computer and could be consulted using another dedicated software named concerto from the AVL company where all the results can be managed and visualized in tables or different graphs.

The typical structure of an engine test cell showing the previously described software is represented in Figure 36. The communication between the systems and therefore between all its associated hardware is done though an additional software from AVL called CAMEO which joins all the received data. The most important CAMEO feature is that it allows the direct connection with PUMA and therefore it is possible to generate automated engine tests.

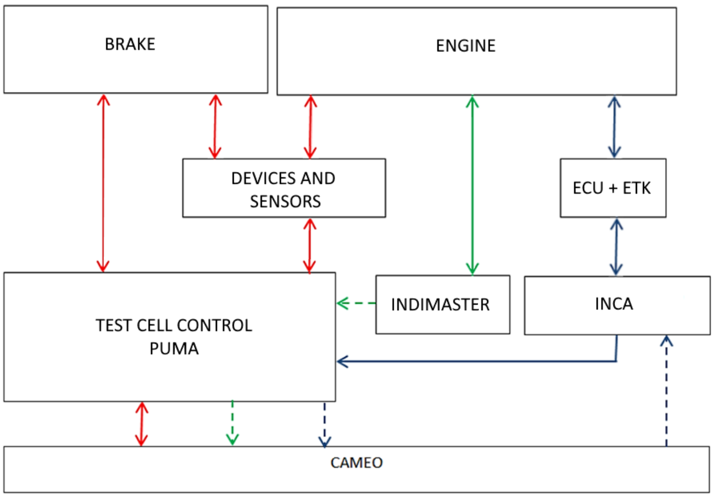


Figure 36. Test cell management system layout

There is a relatively small number of companies in the engine test industry that provides services and software products designed specifically for automating engine tests, acquiring test data, and displaying results. The software used for the tests followed in this thesis project are described in the following.

5.3.1 PUMA-OPEN

Puma-Open is an AVL commercial hardware-software architecture that allows total control of the test cell, either in manual or automatic operation mode. The management of the test cell and of the alarm limits is done employing a Programmable Logic Controller (PLC). PUMA regulates the engine torque and speed through a PID controller that acts on the dynamometric brake and the throttle actuator, respectively. This software also allows setting the different PIDs that act to automatically regulate the temperature of the fluids flowing though the engine and the test cell, and to continuously monitor its magnitudes in order to avoid possible damage to the engine or to the dynamometer by applying predefined reactions depending on the encountered situations, for example, shutting the engine down or taking it to idle speed.

The PUMA Windows interface allows the use of the computer for the interaction with the whole PUMA application system, interacting and controlling the brake, the engine room equipment and the motor actuators. The PC where the PUMA software is installed is equipped with two screens on which the values of the analyzed parameters of the engine and test cell are displayed in real time. It is possible for the user to choose which parameters to display and arrange the digital indicators on the interface as pleased (see Figure 37). These characteristics facilitate the monitoring of the test and of all the parameters of interest for the safety of the engine and of the devices present in the cell. A group of maximum or minimum threshold values for the interested parameters are set in the software before the start of any test. These limits are intended to provide an automatic monitoring of the test cell and of the specific test. Each alarm could be associated to a different action such as stopping the engine, driving it to idling speed, message pop up in the display, acoustic alarm, etc. in case of exceeding the limits imposed by the user at the beginning of the tests.

Figure 37. Puma operator user interface.

5.3.2 INCA

The management of the engine actuators is delegated to the ECU which is equipped with a specific program that deals *with inputs, outputs and control parameters*. The *inputs* correspond to signals acquired by the numerous sensors present on the engine, the *outputs* represent the commands given by the ECU to the control actuators available on the engine (e.g. ignition advance, injection timing, valve opening, etc.) to manage its operation. Finally, the *control parameters* are the variables stored in the ECU which allow the engineer to decide which reactions (outputs) the ECU should execute under certain operating conditions (inputs).

Using the software INCA by ETAS it is possible to modify the *control parameters* during the "calibration" phase when working on *development ECUs* which are "open" ECUs that allow, through the use of software, the management and modification of its content. "Calibration" is the term used in the engine testing industry for referring to the optimization process of the operational and control characteristics of a powertrain system through the manipulation of all its control mechanisms to suit specific engine performance, emissions and/or fuel economy while remaining within the limits imposed by the environmental and safety regulations. Once these variables are optimized, are to be implemented in production ECUs and installed on production engines. These ECUs are characterized by a Read Only Memory (ROM) that does not allow any kind of modification of the stored values by the user. Therefore, INCA allows the engineer to read the values of the inputs and outputs (defined as Measurements) and to act on the control parameters (called Calibrations) in order to identify the values that allow the best/optimized engine operation. Once the optimized control parameters are found it is possible to store their values through INCA in the engine map (also called ECU dataset). The engine map is essentially a multi-axis look-up table held in the memory of the engine control unit. Employing the interface ASAP3 used for the interaction between PUMA and the ECU software (INCA) it is possible to see the variation of the parameters magnitudes in real-time while running engine tests.

6 EXPERIMENTAL RESULTS

During the tests, the monitoring of different parameters was done through the use of alarms and indications according to threshold values specific to the studied engine and as requested by the RINA regulations. Table 4 shows the monitoring action that takes place when a limit value is reached according to the RINA directives for main propulsion engines in marine operations.

Table 4. Monitoring of the 16L – 600 hp main propulsion marine diesel engine with threshold values.

	Monitoring		Safety - Automatic Control	Threshold Value
Identification of System Parameter	**Alarm**	**Indication**	**Shut down**	
Fuel pressure after filters	–	Local		14 bar
Leakage from high pressure pipes – Fuel level	High	–	–	–
Lubricating oil - main gallery pressure	Low	Local		1 bar
	Very Low	–	X	0.8 bar
Lubricating oil - main gallery temperature	–	Local		100 °C
Engine coolant outlet temperature	–	Local		108 °C
Exhaust gas temperature before turbo	–	Local		730 °C
Engine speed	–	Local		2070 rpm
	High	–	X	2160 rpm
Fault in the very low oil pressure sensor	X	–	–	–
Fault in the electronic governor system (overspeed governor)	X	–	–	–

NOTES :

- X = Function is required
- The “local” indication means that the read value for the parameter must be displayed on the RINA safety panel.

6.1 Tuning Tests

Considering that the engine was a prototype it was necessary to develop different engine tests before proceeding with the official homologation tests in order to perform tuning actions and to check the correct working of the engine, the sensors and all the devices present in the test cell. The tuning tests simulate the same working points of the official homologation test shown in Table 5, but have a duration of 3 minutes for each individual step instead of the times reported in the table.

All the tests cycles were performed in an automated way by executing predefined tests run sequences using an AVL tool called BSQ-SSQ editor present in the PUMA installation of the test cell. The tool is used to write the program that will be fed to the PUMA software and through it, control all the characteristics for the cycles described in Table 5 and Table 6. Some of the characteristics include the ramp rate (i.e. rate of change) of variables such as speed or power, the residence time for each cycle step, data recording time, etc. In essence, the automatic program controls the setting of the engine in a specific operating point dictated by each of the cycle steps and how the engine is brought to that point.

During the tuning tests it was observed that engine was not able to deliver the power level requested on steps 5, 6, 11 and 12 of Table 5, more specifically, the engine was not able to reach the operating points of rated power (step 5 and 11) and of 110% rated power (step 6 and 12) at the engine speeds imposed by these specific steps of the RINA homologation tests. This issue was verified when trying to take the engine to those operating points while working on manual control mode of the PUMA test bed. This issue was associated with two different phenomena. The first one, related to step 5 and 11 (where engine must reach 100% rated power and 100% engine speed) is due to the fact that the ambient power correction factor α was being applied since there was a difference between the test ambient conditions and the reference ambient conditions, as mentioned in section 2.2.3.1 and as required by the RINA directives. The correction factor α acts as a penalty factor to the power developed by the engine and therefore, it is necessary for the engine to deliver more power than the 100% rated power to reach the official 441kW plus the power penalization. Since the engine was a prototype, the **dataset** present in the **base engine**'s ECU didn't consider the new power request deriving by the application of the RINA standards and thus, a **tuning procedure** was required to take the engine to these new power levels. It is worth to highlight that according to the RINA standards, the ambient correction factor α is only to be applied to the engine 100% rated power operating points.

The second and most important phenomenon considers the step 6 and 12 of Table 5. Here, the power requested by the RINA standards is 110% of the rated power and at step 6 it is even coupled to a more demanding condition of 103% of the rated speed hence, it was not possible for the engine with the base dataset to develop such performance levels and therefore, modifications to the ECU dataset must be followed. The points related to 110% rated power give the most demanding operating conditions to the engine.

For the engine to be capable of reaching the engine operating points of the RINA homologation tests, the power limit was raised by changing the fuel delivery (injected fuel quantity) and monitoring its influence on other engine variables and hence, controlling to be within the limits shown in Table 4 and for the components correct operation. The results and analysis will be focused on different engine parameters data for the full load curve (obtained with an engine full load test) and on the most demanding operating point of the RINA homologation tests, that is, on the step 6 of Table 5, where the 110% rated power at 103% rated speed should be developed. It is worth to remember that the engine maximum power is 441 kW @ 1800 rpm as mentioned earlier in section 3.2 therefore, the operating point to be analyzed is where the engine delvers 485 kW @ 1854 rpm. Figure 38 shows the fuel delivery related to the engine full load curve with the base dataset and the modification done to the fuel injected quantity for the most demanding point. The injection quantity for developing 485 kW was increased by 13% with respect to fuel delivery of the base dataset curve at the engine's rated speed (1800 rpm). This significant increment will have an impact on the operating conditions of the engine and will be analyzed in the following.

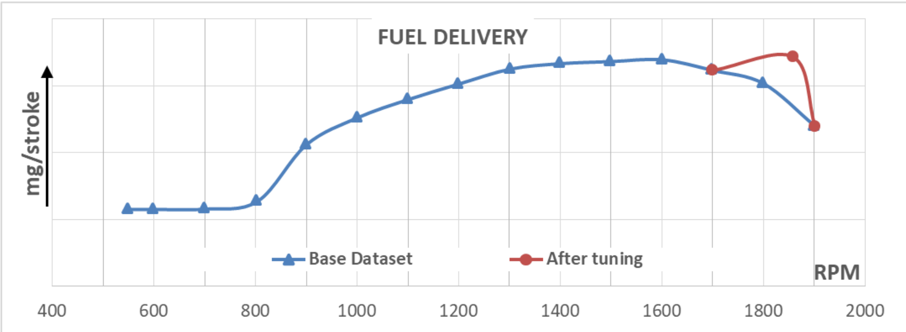


Figure 38. Fuel delivery before and after tuning tests

To examine the power and torque levels achieved with the variation of the fuel delivery, the full load curve is plotted in Figure 39. As observed, the maximum power developed is indeed 485 kW @ 1854 rpm. For the analysis of the engine power at rated speed, i.e. 1800 rpm, it is considered that the correction factor α has a variability depending on the actual ambient conditions encountered in the test cell while performing the homologation test. An average factor α of 0.944 was found for all the cycles, corresponding to a delivered power of 467 kW which is in average 5.9% more power than the engine's rated power of 441 kW.

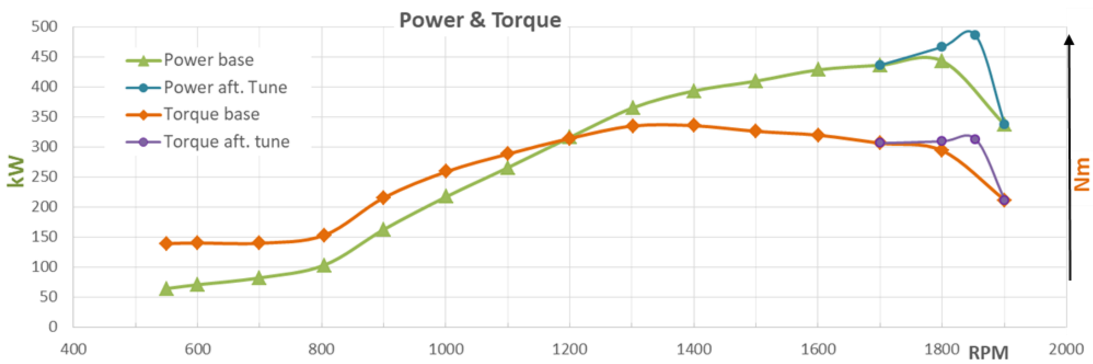


Figure 39. Power and Torque behavior – Full load

From Figure 40 it is possible to see how the fuel delivery is directly related to the load imposed to the engine. As observed in Figure 40 the increment in torque comes accompanied by an increase of the fuel delivered and vice versa. At the most demanding operating point, i.e. at 1854 rpm, there's a peak in injected fuel quantity that allows the engine to increase the delivered torque by making more fuel to be available in the combustion chamber for mixing with the air. The fuel delivery increase followed with the tuning procedure brings a rise of almost 6.5% of the engine torque at 1854 rpm.

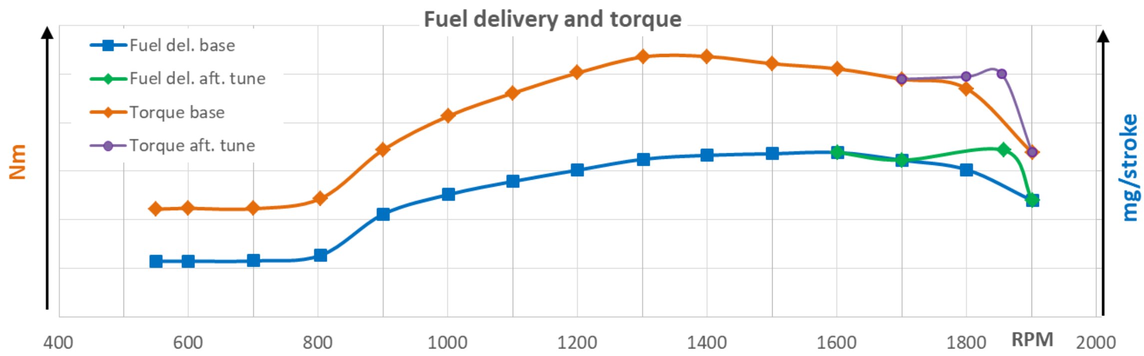


Figure 40. Relation between load and fuel injection

The fuel injected quantity depends on the injection duration (energizing time) and on the rail pressure. The more time the injector nozzle remains open, the more fuel will be injected. The more pressure is applied, the more the velocity of the injected fuel which results in more fuel delivered per unit time. Another important variable to be considered together with the fuel delivery is the Start of Injection (SOI). More time will be needed to inject a greater amount of fuel therefore, the fuel injection will have to start earlier so that the all the fuel has been injected before reaching the TDC avoiding the decrease in engine efficiency. From the graph of Figure 41 it can be observed that the injection will continue to be advanced even if the fuel delivery (shown in Figure 38) decreases. This can be explained considering the fact that the phenomena of fuel jet atomization, liquid fuel evaporation, fuel vapor mixing with surrounding air and the chemical reactions that lead to the spontaneous ignition are not scaling with the revolution speed in diesel engines. These phenomena are the cause of the so called ignition delay and they require roughly the same amount of time at different revolution speeds thus, with the speed increment there's less time availability per cycle for the completion of these phenomena and the SOI has to compensate for that time. It should be mentioned that the increase of the rail pressure can shorten the ignition delay by improving the fuel atomization giving a faster fuel evaporation.

Talking about the data of base curves, it is also worth to notice in Figure 41 that there's a sudden increment in the rail pressure at 1800 rpm even if there's no such behavior for the fuel delivery. This is done for controlling the amount of particulate matter (measured as black smoke) emitted at the rated power. While approaching the 1800 rpm there's a black smoke increase trend that is rapidly lowered by the significant increment in rail pressure.

The rail pressure improves the fuel-air mixing, lowering the formed particulate matter by favoring the fuel atomization and the air entrapment in the injection jet plume.

Now, for the most demanding point of the homologation tests (1854 rpm) reached with the modifications done in the tuning procedure, the rail pressure was further increased to help the increment in the fuel delivery and to counteract on the emitted particulate matter. Even with the increment of fuel in the combustion chamber and without the increase of injection advance, the FSN for black smoke remained almost unvaried at 1854 rpm.

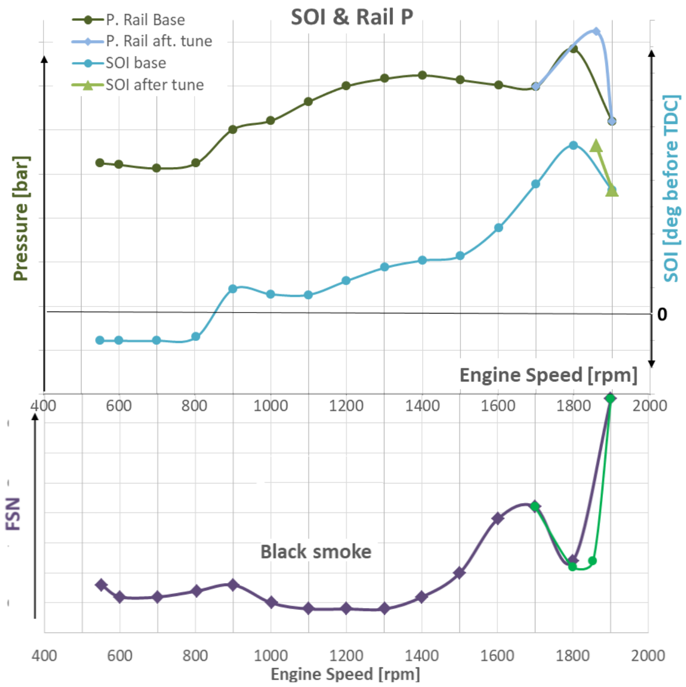


Figure 41. Development of SOI, Rail pressure and Black Smoke curves related to the full load test.

In the following the agreement between some of the parameters behavior during the full load test, their modifications done through the engine tuning for the most demanding operating point, and the engine design limit values is studied. These limits were established during the engine design phase for assuring a sound engine operation, avoiding overloads that generate mechanical stresses which may lead to the components failure, and maintaining the components within their working operation temperature range.

Unfortunately, the cylinder pressure value for the most demanding operating point was not available, but considering the increment percentage of the fuel delivery with respect to the point at 1800 rpm and hypothesizing that the same increment percentage will be experienced by the cylinder pressure; it is possible to evaluate if the limit value was reached. As mentioned in the previous paragraphs , the fuel delivery was risen by 13% and considering that the maximum peak firing pressure found during the full load test (at 1800 rpm) is 19% smaller than the limit value as shown in Figure 42, there’s enough margin to avoid exceeding the limit value for the allowed cylinder pressure.

Checks for the limit values were also done for the compressor outlet, turbo inlet and engine coolant temperatures. All these parameters presented the maximum value for the operating point of 110% rated power (@ 1854 rpm) as shown in Figure 42. The limit values for the compressor outlet, turbo inlet, coolant outlet and coolant inlet temperatures were 3.7%, 20%, 9% and 10% greater than the maximum experienced value.

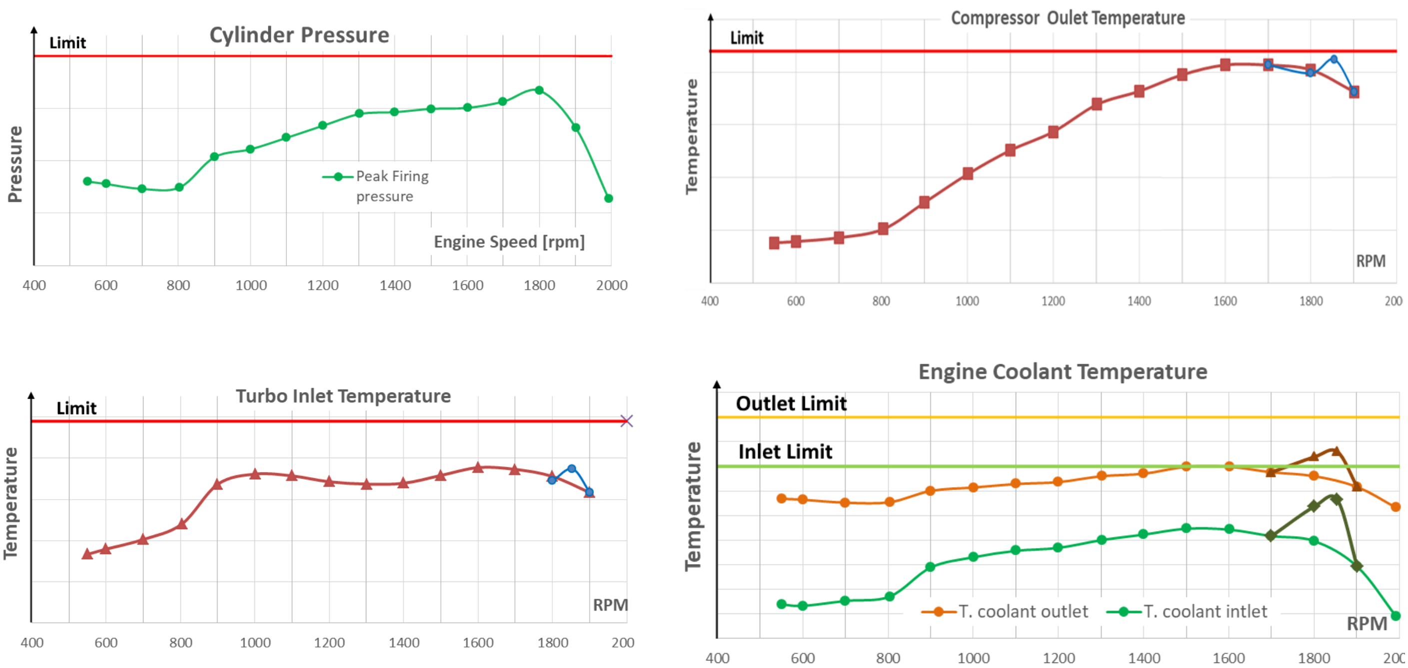


Figure 42. Cylinder pressure and, compressor outlet, turbo inlet and engine coolant temperatures with limit values.

6.2 Homologation Tests

The homologation procedure evaluates the engine durability by means of different tests with a total duration of 100 h which are divided in different sub-tests. The so-called core tests and the variable load tests.

6.2.1 Core Tests

The first part of the core tests consists of 12 steps with a duration of 24.5 h per cycle (see Table 5). The engine speed and power are increased in each step until arriving to the step 5 where the engine stays for 10h at rated power and rated speed. After this, the engine stays for one more hour at 110% rated power with a little increase in engine speed (103% rated speed - point 6). For the step 7, the power is once again brought down to the lowest power value and then the power is increased at each step as done before, but this time the rated speed is maintained at its maximum value for all the steps. The previously described cycle is repeated four times obtaining a total of 98 h test duration. The automation for all the tests was performed in the same way as explained previously in section 6.1. It must be considered that due to the long duration of the test, the automation program will perform the data recording only for the last 30 s of each step. Since the step 5 and 11 last 10h each, the program takes 30 s recording each hour, i.e. 10 recordings for each of these two points (as depicted in Figure 43). Additionally, an average of all the data measured during the recording times is performed by the program, generating one value for each step which will be used later for data manipulation for analysis or graphic representation.

The residence time, number of steps and engine operating conditions are the result of an agreement between the RINA Society and FPT considering the load points criteria given in section 2.2.3.2.

Table 5. Homologation test operating points and duration.
Total for 4 cycles → 98h

	Time		% Rated Power	% Rated Speed
	Step	**min**		
	1	20	25	63
	2	20	50	79
	3	20	75	90
	4	15	90	96
	5	600	100	100
	6	60	110	103
	7	20	25	100
	8	20	50	100
	9	20	75	100
	10	15	90	100
	11	600	100	100
	12	60	110	100
Total/cycle	**[min]**	1470		
Total/cycle	**[h]**	24,5		

Figure 43 shows the power level developed by the engine throughout the whole 98 h test, considering the power correction according to the discrepancies between the ambient conditions found in the test room and the ones provided in the RINA standard. It is worth to highlight that according to the RINA directives, the power correction is only applied to the operating points regarding the 100% of engine rated power (steps 5 and 11). As can be seen there's little variability in the power level between the four cycles, meaning that the test cell and engine conditions were steady and highly controlled for the whole duration of the tests. This is supported by the fact that the power correction factor α, that depends on the barometric pressure, relative air humidity, ambient air temperature and seawater temperature (as indicated in Table 1); suffered minimum variations during the 98h test duration, ranging from 0.935 to 0.955. For a more detailed view of the data, only one cycle of the core test was depicted in Figure 43.b). Each individual cycle is divided in two half parts. The first part of the cycle is related to the steps where there's a gradual increase in the engine speed (steps 1 to 6 of Table 5) and in the second part of the cycle the speed is maintained constant at the maximum engine speed (steps 7 to 12) . Even considering these speed variations, the engine power is maintained approximately the same between correspondent points within the cycle, i.e. between steps with the same percentage of rated power (steps 1&7, 2&8 and so on), by varying the dynamometric brake torque as illustrated in Figure 44. a) and b). In the latter figure it is possible to appreciate more clearly that the braking torque is lower for the steps related to the second half of the cycle where the speeds are higher.

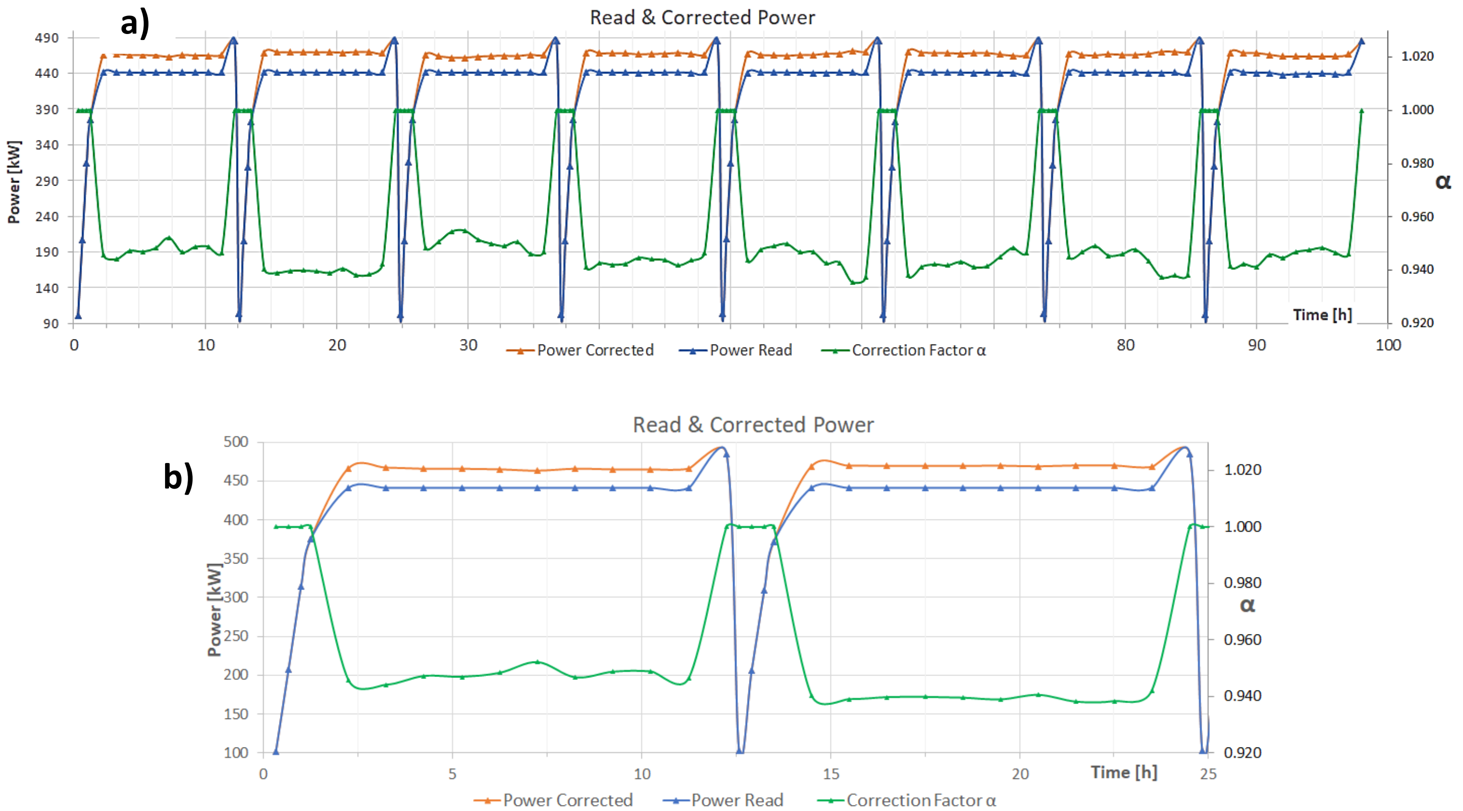


Figure 43. Read and corrected power for a) the full 98h core tests and b) one cycle

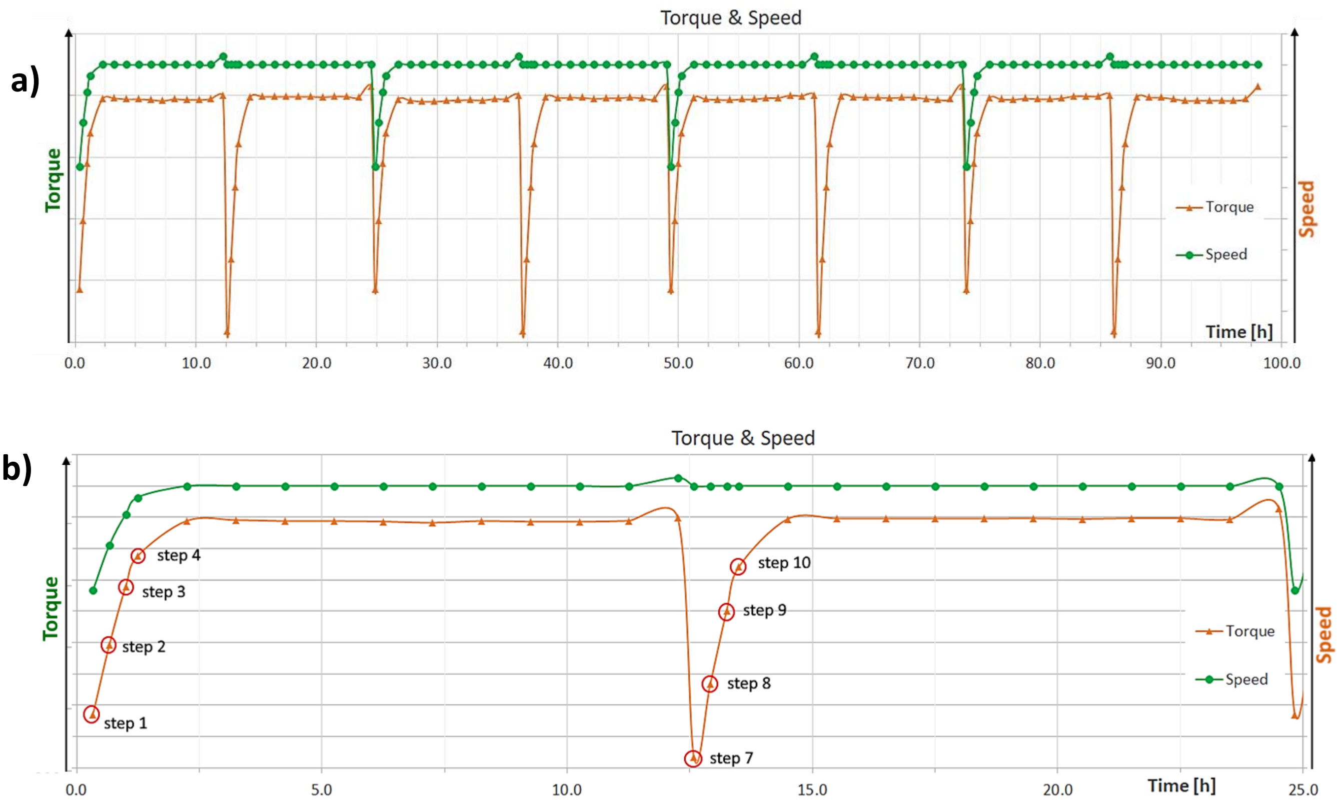


Figure 44. Torque and speed for a) all four cycles and b) one cycle

6.2.2 Variable Load Tests

The second part of the assessment for the completion of the RINA 100 h mandatory tests considers an extra two hours of intermittent operation where the engine is subjected to idle–full load–idle cycles that subject the engine to variable load cycles. Each intermittent load cycle is divided in two steps. One where the engine is maintained at low idle with no load for a brief time interval and the second one where the load is increased to the maximum engine power at rated speed as indicated in Table 6. This cycle is repeated 12 times to complete the total required time of two hours. It should be considered that the rate of change between both operational points is performed at a fast pace and thus, the automated program was set with a steep ramp. Due to the short duration of each operational point, the program is set to follow a continuous data recording for each step with a subsequent averaging of the data. For the variable load tests no correction factor is applied to the engine power when developing 100% rated power since is not requested by the RINA directives.

Table 6. Intermittent tests operating points and duration.

	Total for 12 cycles → 2h			
	Time		% Rated	% Rated
	Step	min	Power	Speed
	1	4	0	Low idle
	2	6	100	100
Total/cycle	[min]	10		
Total/cycle	[h]	0,167		

Figure 45 shows the power and torque values for 1h of test duration (total duration = 2h). All the values of the points belonging to the full load have virtually the same value determining that the engine can consistently develop the requested power and torque even when subjected to abrupt load changes. During the test, the engine showed also a rapid response to these load changes and the engine inertia, when passing from one operating point to another at a fast pace, was not an issue.

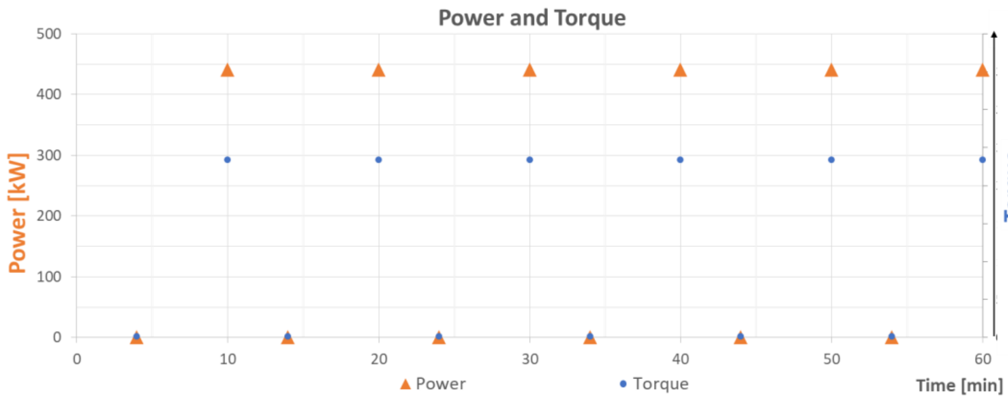


Figure 45. Power and Torque behavior

6.3 Functional Tests

Finally, for the completion of the engine certification tests, two further engine assessments called functional tests should be performed as described in section 2.2.3.3. The functional tests verify the lowest propulsion engine speed and the compliance of the engine maximum speed regulator.

6.3.1 Lowest Propulsion Engine Speed Test

This test is used for verifying the correct engine operation during the departure of the vessel, that is, when the engine is operating at idling speed and little throttle is given in order to start the movement of the vessel. In marine applications the engine is subjected to significant loads at idling and near idling speeds due to the turning of the propeller through the water which has a significant impact on the power required to thrust the vessel according to the propeller power curve. This situation could be compared to a car uphill start where the engine is subjected to a high load due to the grade of the hill. The available torque at near idling speeds should be enough to avoid the engine from shutting down and to accelerate the vehicle uphill or in the case of marine applications, move the vessel through the water.

The verification is done by means of the *lowest propulsion engine speed test* which begins maintaining the engine at idling speed while the braking torque is gradually increased starting from a null requested torque until having a reduction of 50 rpm from the initial engine idling speed. The engine shall not stall and no alarm or indication should be activated during the whole test extent. For the braking torque increment the automated program uses a gentle slope ramp assuring gradual and mild rate of change. Figure 46 shows the developed engine power during the test. While increasing the requested power by means of the increment in braking torque, it is possible to see that the idle speed is not maintained perfectly constant, but instead, there's a small oscillation on the read engine speed. These variations can be attributed to engine behavior because of high the high loads requested at such low engine speed, due to the intrinsic behavior of the dynamometric brake or due to a combination of both effects. After reaching the maximum power that the engine can produce at idling speed, and since the braking torque continues to increase, the engine is not able to maintain the requested load and the speed starts to decrease, as a consequence so does the developed power. It is worth noting that after reaching the maximum power developed at idling speed, the engine power follows the engine "full load curve" while it continues to lose speed until arriving to the end of the test, i.e. when the engine speed is reduced by 50 rpm.

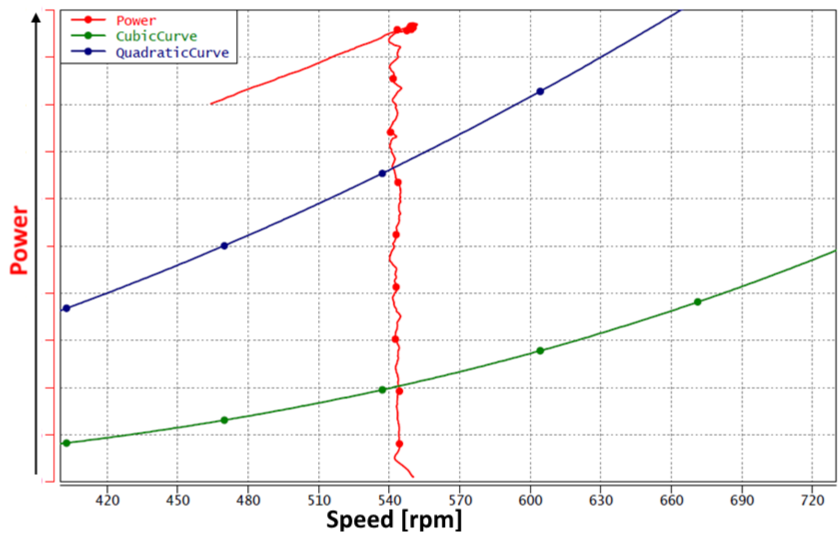


Figure 46. Lowest Propulsion Engine Speed Test Curve

The test results show that even at the lowest speed, the engine is able to deliver enough power to propellers mounted on either displacing hull vessels characterized by a cubic propeller power absorption law or, gliding hull vessels characterized by a quadratic law (power absorption laws were discussed in section 4.1.3.3). Even considering the most power demanding situation, there's enough engine power availability for the vessel propulsion plus a margin for extra power requested from the engine such as power take offs, electrical installations, generators, compressors, etc. which are not directly used to propel the boat.

6.3.2 Speed Governor Compliance Test

The compliance of the speed governor functional test studies a specific situation where the governor should take action and avoid the engine speed to be exceeded by more than 15% of the rated speed. This situation could be present in marine applications when the boat propeller is exposed out of the water after the boat takes-off after taking a wave while travelling at full load at maximum rated speed. This specific situation leads to a sudden change in the engine operating conditions. The test simulates this event by abruptly passing the engine working at full-load condition at maximum rpm to a no-load condition where the engine speed tends to rapidly increase, but should be immediately controlled and regulated by the speed governor that sends a signal to the ECU which in turn actuates on adjusting the fuel injection.

Figure 47 illustrates the engine speed and power responses during the test. The experiment starts with the engine revving at maximum speed and it is possible to notice how after the sudden decrease in requested power by means of the dynamometric brake torque reduction, the speed abruptly increases but it is rapidly regulated reaching a maximum speed of approximately 11% of the rated speed after the stabilization period. This percentage value was calculated by the ratio between the maximum reached speed during the test and the initial engine speed. The reaction of the governor was in agreement with the RINA directives since the maximum threshold of 15% rated speed was never exceeded, not even before reaching the speed stabilization. The governor speed control demonstrated a fast response and no oscillation phenomenon was observed meaning a high damping characteristic.

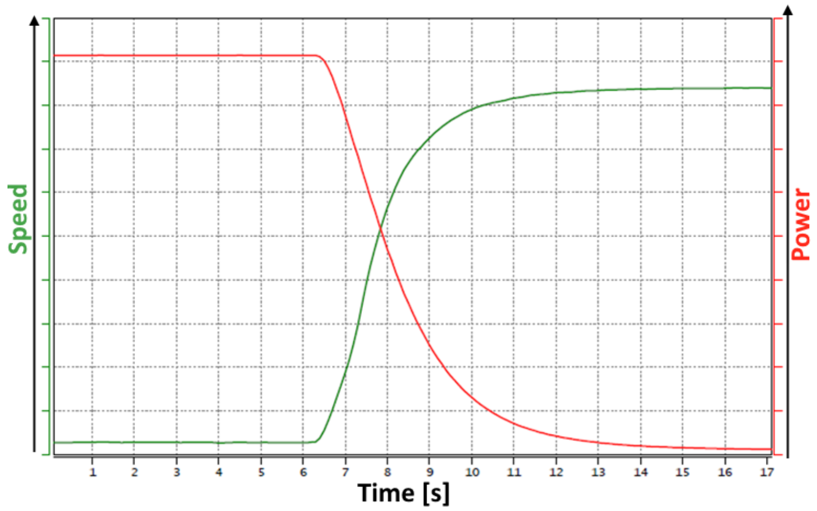


Figure 47. Speed Governor Compliance Test

6.4 Engine Inspection

After all the certification tests, the engine components must be inspected for premature wear, and overheating and fatigue problems resulting from overload working conditions and/or by the presence of components or materials defects. The inspection was performed by disassembling the engine, cleaning and performing a visual inspection on the main components and by proceeding with a tightening torque check of some relevant components nuts or bolts.

6.4.1 Torque Tightening

The tightening torque applied to the bolts and nuts of the different components when building the engine is related to the force that clamps the joint members together, known as pre-load tension. Checking the pre-load of main fasteners is an important tool for controlling if the engine components have been submitted to overload or fatigue problems due to fabrication (component defects), assembling mistakes, material imperfections or harsh operating conditions out of the engine design limits.

For controlling the state of the pre-load, marking torque tests were followed while disassembling the engine. It should be considered that there’s no practical way to measure the clamping force and consequently, the value of the clamping force is usually referred to the tightening torque. The procedure illustrated in Figure 48 involves the marking of the fastener and the surface being clamped before loosening it, proceed with the loosening of the fastener then and slowly re-tighten it until aligning the marks on the fastener and surface. The torque value read on the torque wrench is the torque required to return the fastener to its original location and is the reference of the original torque applied to the fastener during the engine assembling and should match or almost have the same value as the tightening torque shown in the different components assembling designs. If both values match, it means that the loads experienced by the components while performing the certification tests were smaller than the joint clamping force and it could be assumed that no overstress or fatigue problems were present during the tests. On the other hand, if the joint is exposed to higher loads than the fastening pre-load, the torque tightening values will present considerable variations meaning that the components were subjected to overload or fatigue problems during testing resulting in bolts with permanent elongation or damaged threads which prevent the joint to remain tight enough to avoid the joint loosening.

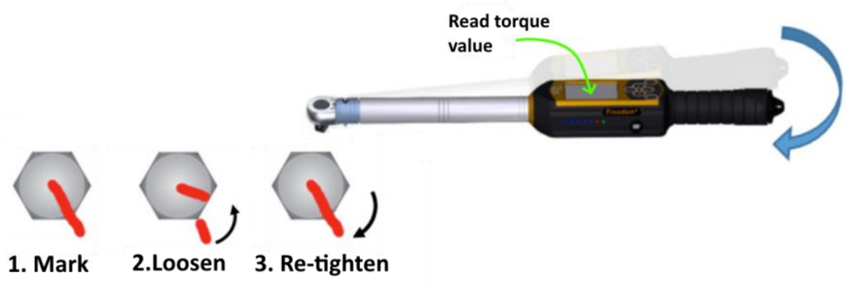
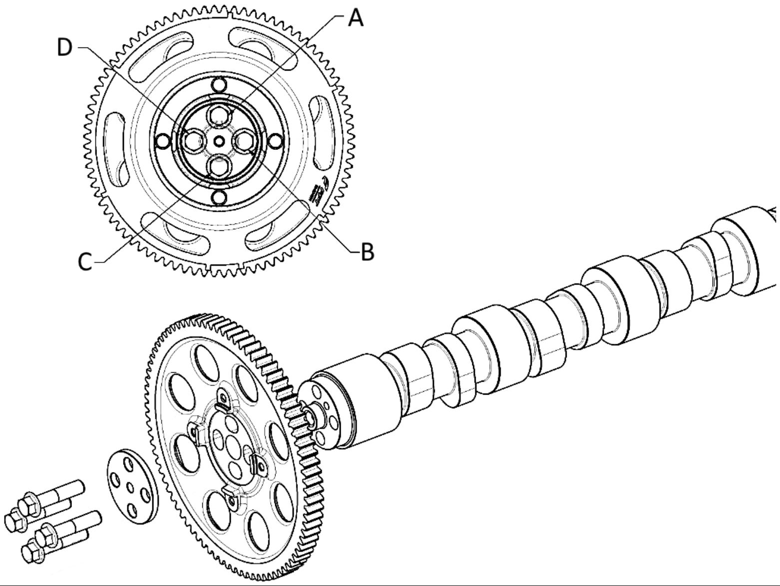


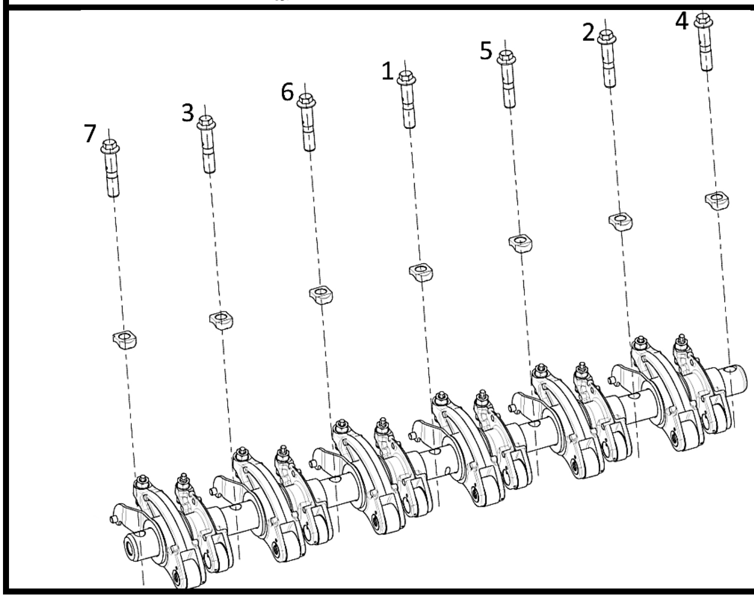
Figure 48. Marking torque test procedure.

No mismatches were found for any of the evaluated fastenings, that is, all the tightening torque values reported during disassembling found following the marking torque tests were included inside the tightening torque range reported in the components' assembling designs. Table 7 shows some examples of the results obtained for some of the evaluated fastenings. The tightening torque values reported come from the marking torque tests and the target values are the check torque values written in the component drafts developed during the engine design phase.

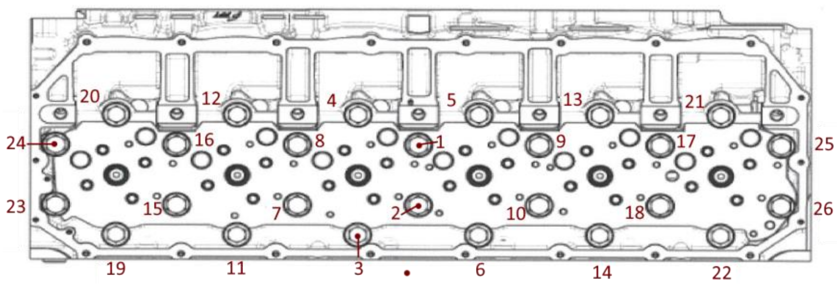
Table 7. Torque tightening results from marking tests.



Camshaft Timing Gear Retaining Bolts	
Target [Nm]:	160 - 300
Bolt	**Read torque [Nm]**
A	243
B	253
C	248
D	255



Rocker Arm Shaft Bolts	
Target [Nm]:	160 - 280
Bolt	**Read torque [Nm]**
1	235
2	240
3	237
4	245
5	250
6	238
7	242



Cylinder Head Bolts			
Target [Nm]:	440 - 680		
Bolt	Read torque [Nm]	Bolt	Read torque [Nm]
1	473	14	501
2	485	15	495
3	490	16	485
4	480	17	490
5	475	18	488
6	505	19	475
7	485	20	484
8	473	21	504
9	486	22	496
10	472	23	499
11	480	24	495
12	493	25	501
13	473	26	490

6.4.2 Components Visual Inspection

6.4.2.1 Connecting rods

All the connecting rods presented normal polishing on the small eye surface. Also, all of them showed two darkened areas on the small eye, one on the thrust side surface and another one on the opposite side as represented in Figure 49.b) and a), respectively. The darkened areas are the result of highly demanding lubricating conditions due to high local contact pressures with a consequent local temperature increase resulting in an oil degradation phenomenon that gives a slight difference in the surface appearance compared to the surrounding areas. It must be highlighted that this phenomenon is considered normal and does not have any negative impact on the engine's performance or represents any abnormal material degradation.

For the thrust side, the most demanding lubricating condition is present when the combustion pressure, coming from the piston and transmitted as a net force to the conrod though the contact area between wrist pin and conrod's small eye, reaches its maximum level. The peak pressure experienced in the combustion chamber is developed some crank angle degrees after top dead center. This explains why the darkened area is not strictly centered but it is instead a little bit moved to the side.

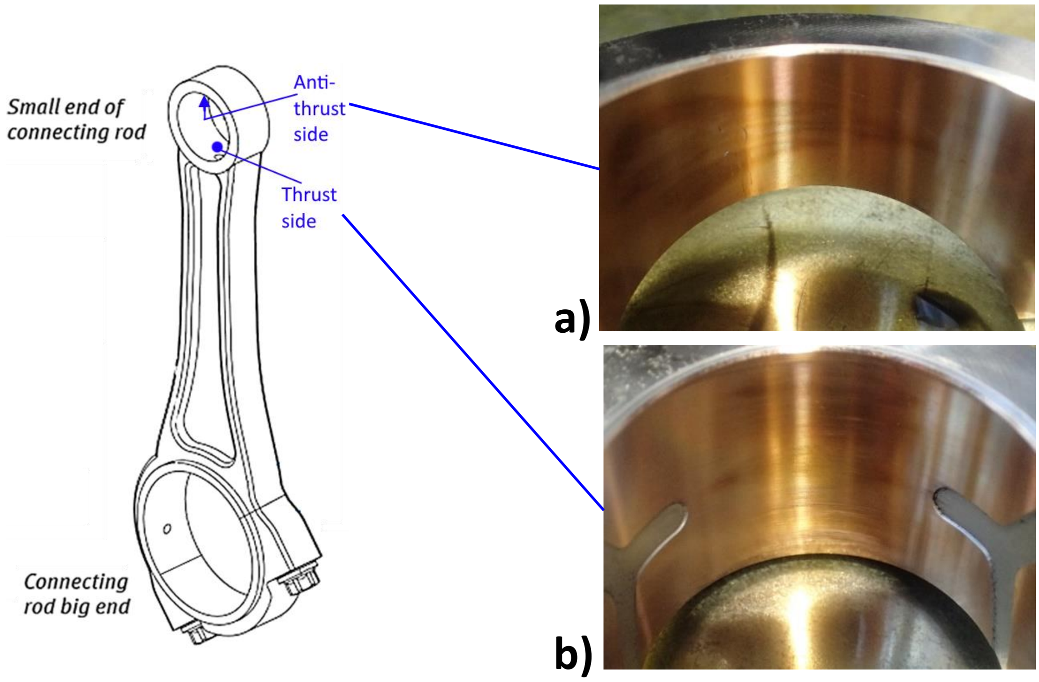


Figure 49. Darkened surface areas present on the conrod's small eye. a) Thrust side surface. b) Opposite to thrust side surface.

In the case of the anti-thrust side, the maximum stress condition on the contact area is developed during the exhaust stroke; specifically, when the force acting on the conrod's small eye passes from the thrust side (force exerted by the crankshaft) to the antithrust side. From this moment, the high inertia carried by the piston and the wrist pin bodies during their upward motion is transmitted to the anti-thrust side resulting in a high resistance force (see Figure 50) which decelerates both bodies until coming to complete stop at TDC before the start of the intake stroke and the subsequently change of the piston movement direction. Since this resistance force is only present during the piston upward motion, the darkened area shown in Figure 49.a) is not symmetrical and on the contrary, is only present on one side. The distinctive "drop" or "cam" shape of the darkened area is the result of the force F value variation along the piston travel from the moment the force starts to be applied on the small eye anti-thrust side surface until the piston reaches TDC. The force F is dependent on the piston speed which is in turn related to the position of the crankshaft. For the diagram depicted in Figure 50, the maximum speed and consequently the maximum force, is reached when the angle $\theta = 90°$ and minimum when $\theta = 0°$. That is, when the piston is at halfway stroke and when at TDC, respectively.

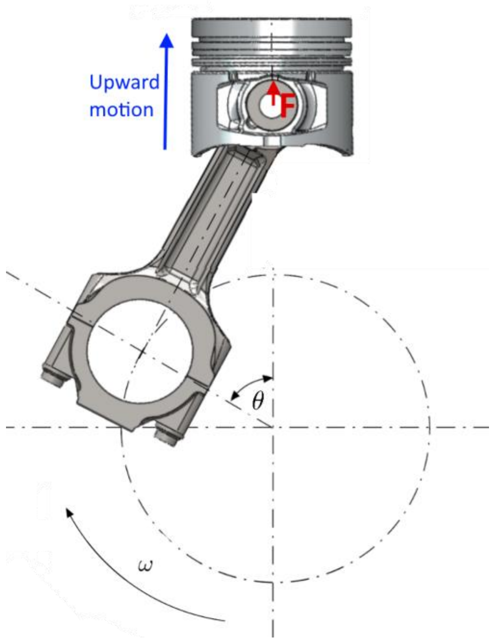


Figure 50. Force F applied to the anti-thrust side of the conrod's small during piston upward motion.

6.4.2.2 Crankshaft bearing

The crankshaft bearings show a particular surface appearance on a very specific area located on the conrod's big eye upper half bearings as shown in Figure 51.a) while there's no evidence of surface anomalies on any of the cap bearings (Figure 51.b)). The surface polishing present in all of the 6 bearings upper halves is the consequence of the same phenomenon present in the small eye, where the combustion chamber pressure reaches its maximum level at near TDC. In this position of the crank mechanism, the peak force coming from the piston and passing through the conrod is finally transmitted to the crankshaft through the contact area between upper bearing of the conrod's big end and the crankshaft rod journal (see Figure 52). This explains why the polishing appears only on the upper bearing and not on the cap bearing. The surface appearance of the bearings upper halves does not represent any damage to the bearings or premature wear.

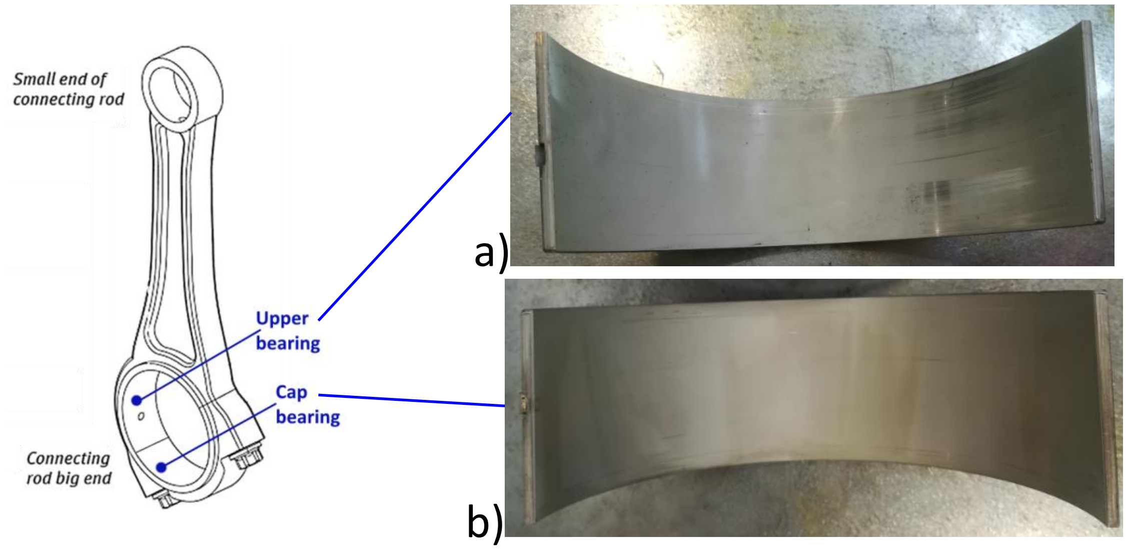


Figure 51. Crankshaft bearings appearance. a) Upper half (thrust side). b) Lower half (cap side)

From Figure 51.a) it is possible to notice that the surface polishing is not centrally located but it is present only on one side of the upper bearing. This is because the connecting rods for this specific engine have an oblique face junction geometry between rod and cap. When the maximum force is developed, i.e. some crank angle degrees after TDC, the conrod is situated at an almost vertical position and thus, the contact area between the upper half bearing and the crankshaft rod journal is the one shown in Figure 52 and described as thrust side.

Figure 52. Maximum force developed on the thrust side of the connective rod's big eye.

6.4.2.3 Cylinder liners

The cylinders inspection revealed that there were some light marks running parallel to each cylinders axis and were present on both the thrust and anti-thrust sides of all the cylinders as shown in Figure 53. The marks are very light and don't represent any risk for the engine integrity, but they are the consequence of external bodies present in the lubricating oil film between the liner and the piston rings which caused the formation of some vertical lines due to the upward and downward piston motion. Together with these marks, it is also possible to note the existence of a great number of crossed parallel lines disposed at approximately 45° with the cylinder axis which are the result of a honing process followed on the cylinder liners during its fabrication. If the liner's surface is too polished and smooth it will be difficult for the oil to form lubricating film on the surface and the piston rings will slide on a too dry surface. Hence, the marks disposed in a cross-hatch pattern will enhance the oil retention on the liner's surface, acting as oil "pockets" (see Figure 54) improving the lubrication and as a consequence providing good tribological coupling between liner internal surface, piston external skirt and the piston rings limiting the wear during the relative motion of the piston.

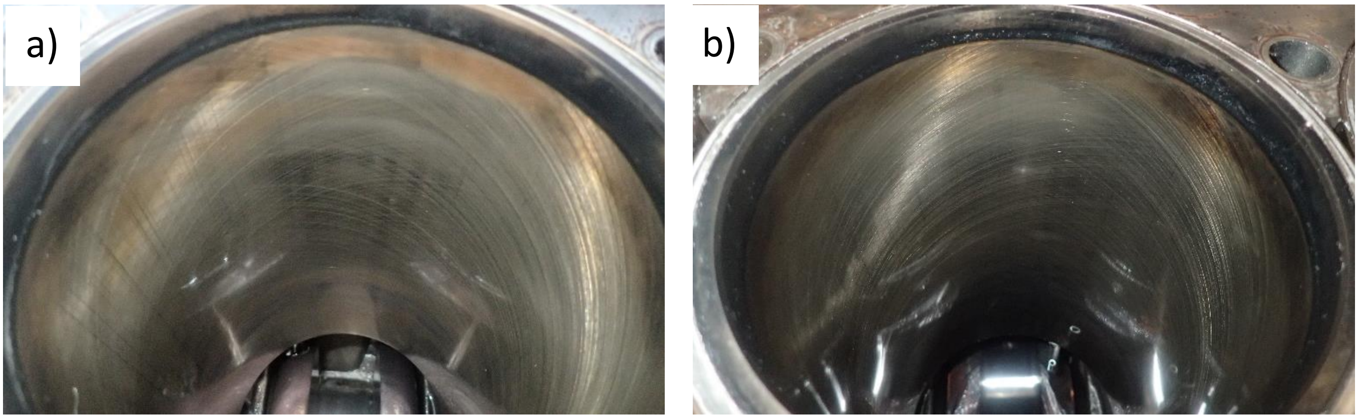


Figure 53. Cylinder walls visual inspection. a) Major thrust side b) Minor thrust side

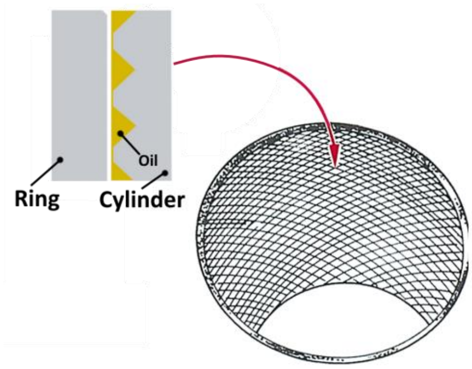


Figure 54. Lubrication improvement achieved with honing of the cylinder liners.

Finally, from Figure 53. a) it is also possible to observe a polished area at almost midspan of the major thrust side of all the cylinders. This characteristic is the result of high demanding stress conditions that occurs cyclically and precisely on that area. As illustrated in Figure 55, during the expansion stroke the force F coming from the combustion pressure can be decomposed into two forces: the axial force $\mathrm{F_{cr}}$ acting on the connecting rod and directed along its axis and, the thrust force $\mathrm{F_n}$ which is transmitted from the piston perpendicularly to the cylinder wall. The latter is the responsible of generating the characteristic polished area found on the major thrust side and its value depends on both the combustion force (F) and the position of the connecting rod (β) as described in equation (22). The exact location of the polished area at almost midspan of the major thrust side of the cylinder is due to fact that the effects superposition that both variables - F and β - have on the thrust force $\mathrm{F_n}$ reaches its maximum at that precise location of the piston travel because the combustion pressure is still significant and the inclination of the conrod is high enough to generate a considerable perpendicular component of the force F.

$$F_n = F\ tan\beta \tag{22}$$

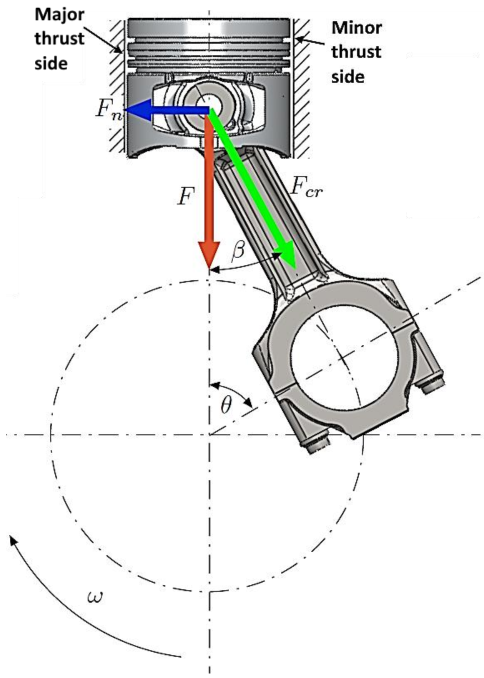


Figure 55. Forces present on the crank mechanism in the expansion stroke.

6.4.2.4 Other main components

Such as piston (thrust and anti-thrust side), piston rings, camshaft, crankshaft, turbocompressor (blades), cylinder head gasket, injectors, valves and cylinder head didn't present any signs, marks, high polishing areas, blackened areas or any other abnormal characteristics that represent damage, premature wear or overheating problems on the component.

7 CONCLUSIONS

- As seen in this thesis project, the certification associations' guidelines and directives lead to modification in the engine hardware and software, mainly due to safety requirements and to the operating conditions requested for the homologation tests. There were components added, replaced or completely removed, sensors and electrical cables installed, software developed and ECU data modified. It is therefore advised to have a deep knowledge of the engine to be certified for proceeding with a thorough review of the certification guidelines and standards, looking for the specific requirements for the application.

- Due to the tuning activities followed, it was possible to bring the engine to the operating conditions requested for the homologation tests. Since the ECU base dataset was developed before starting with the certification process, the increment in the power requirements due to the penalization power factor and due to the high demanding test parameters was not considered.

- A good communication between the engineering departments should be maintained before and during the whole certification period. The product development, design and testing engineers should have a close and direct contact because specific values or data from the engine design are permanently requested from the testing facilities.

- Since there are many variables to be controlled during engine testing, the alarms and indications system present in the test cell help with the automatic monitoring of the engine parameters through the set up of threshold values coming from engine design and from the certification regulations.

- The inspection process revealed the presence of small foreign bodies in the cylinders. Considering that the engine was a prototype it is possible that there were assembling conditions far from the ideal ones found at the engine assembling line.

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