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COMBUSTION ENGINE DIGITAL TWIN DEVELOPMENT AND APPLICATION METHODOLOGY FOR REAL TIME DIAGNOSTIC SYSTEMS

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RAZVOJ DIGITALNIH BLIZANACA MOTORA SA UNUTARNJIM IZGARANJEM I APLIKACIJSKE METODOLOGIJE ZA DIJAGNOSTIČKE SUSTAVE U REALNOM VREMENU

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1 INTRODUCTION

Internal combustion engines have been around since 1860. [1] where Etienne Lenoir created the first commercially used combustion engine. This combustion engine was based on the cycle which contained no compression stroke and the mixture of coal gas and air was spark ignited by Ruhmkorff coil [2]. The theoretical Lenoire cycle consisted of isobaric heat release, isochoric combustion and adiabatic expansion stroke. At 1876. Nikolaus Otto invented the first modern conventional internal combustion engine. The improvement compared to Lenoire cycle was introduction of the compression stroke which lead to the higher combustion pressures and temperatures and thus to the higher combustion efficiency. The theoretical Otto cycle consists of isentropic compression, isochoric combustion, isentropic expansion and isochoric heat release. This cycle served as the foundation of the working process of all modern gasoline internal combustion engines. At 1892. Rudolf diesel invented the diesel engine where ignition was caused primarily by increasing the temperature by piston moving towards the top dead center during the compression stroke. This increased the temperature of the mixture inside the combustion chamber, which initiated start of the combustion effect. These engines are also nowadays called compression ignition engines. The theoretical diesel cycle consists of an isentropic compression, isobaric combustion, isentropic expansion and isochoric heat release. This diesel cycle serves as the base working cycle for all modern diesel engines. There are also number of variations of these cycles especially in gasoline engines. Nowadays popular in downsized turbocharged gasoline engines is the Miller cycle whose characteristic is shorter open duration of the intake camshaft. For the hybrid applications Atkinson cycle applied mostly on naturally aspirated engine is used whose characteristic is longer opening duration of the intake camshaft.

Internal combustion engines have many different fields of application such as automotive, motorcycle, commercial, aviation and different kind of other industries. Depending on the application internal combustion engines can also be divided on 2 stroke or 4 stroke engines. 2 stroke engines perform the gas exchange and the rest of the standard cycle in one movement of the piston from bottom dead center to top dead center and back. 4 stroke combustion engines perform the gas exchange in the additional stroke of the piston which together with standard cycle leads to overall count of 4 stokes per cycle. 2 stroke piston engines are mainly used in light machinery applications such as portable chainsaws and similar, motorcycles with the low swept volumes, small aircraft applications, large heavy fuel oil marine diesel engine applications and remote power plants for electricity generation. 4 stroke piston engines are used in automotive industry, motorcycles with larger swept volumes, commercial vehicles, heavy duty applications, medium and high-speed marine engines, aircraft industry and portable power supply devices. There were also different concepts with 5 stroke, 6 stroke, Wankel, Scuderi, AMICES [3] and other engine concepts which did not enter into the large applications.

The internal combustion engines have severely evolved in the last 50 years mainly driven by emission legislation. They went from fully mechanical control of injection and ignition to fully electronic control. As the computational processors evolved, the control units of the internal combustion engines also got more advanced. The more advanced control strategies were followed by higher number of the sensors and actuators implemented on the engines. Today's conventional engines have electronically controlled injection, ignition, high pressure fuel pumps, thermostat or cooling rotary valves, throttle valve, wastegate, canister purge valve, coolant and oil pumps, camshaft position and lift, cylinder deactivation system, EGR valves, secondary air systems, port flaps, etc. This brings complexity into the control softwares. In order to reduce the function calibration complexity, the simple models are integrated into the control units together with learning and sensor and actuators aging corrections functions. This approach is widely applied in passenger and commercial vehicles. These functions and models are calibrated in the engine calibration phases prior to vehicle homologation to meet the targeted emission legislations, performance and onboard diagnostic requirements. The technology trends are leading to vehicle to vehicle, vehicle to road and to cloud communication. This opens opportunity to optimize the powertrain settings during vehicle lifetime depending on the state of the vehicle, traffic and weather conditions. It would be able to simulate predefined route on the cloud and search for emission friendly routes by using digital twins of the specific vehicle. This would open the possibilities of optimizing the powertrain parameters and hybrid strategies for meeting the optimal fuel consumption, emission output and keeping the optimal health state of the powertrain by using powertrain models in the cloud in communication with control units in the vehicle.

When talking about two stroke low speed marine engines, the control complexity is less than mentioned above for road vehicles. The control strategy is mainly focused on injection control and occasionally for low speed 2 stroke engines on hydraulic exhaust valve actuation Four stroke marine engines have more complex technologies such as Caterpillar-ACERT and MTU-Blue Vision. Large marine engines are higher producers of pollutant due to the higher mass flows of fuel and air which they consume as well as the type of the fuel which they use.

During component wearing and degradation consumption and pollutants are increasing drastically. It is important to mention that lifetime of large marine engines is about 30 years. In this periode the engines are many times retrofitted and the engine calibration does not fit to the one initially performed on the engine testbed. For these reasons the methodology of the digital twins or the engine models in use case of predicting the engine failures and optimizing the engine calibration by using models in the cloud solutions is chosen to be developed in this doctor thesis.

This qualifying work is focused on investigating the current marine engines control setups, engine control units, actuators and sensors. The available physical real time models are going to be evaluated in order to be used for engine model calibration and simulation of the early symptoms of the engine component failure. The evaluation of existing monitoring and diagnostic systems and methods for the marine engines which are existing on the market will be processed. Finally, the benefit of applying the real-time physical models on the cloud which are communicating with the real engine which is operating on the real-world condition will be evaluated. In order for this approach to work the engine model calibration methodology and failure simulation needs to be developed. When proven successful, this would be the first methodology of a kind where parallelly to a real engine it would be able to simulate how the engine should work in the current real-world conditions and thus, by comparing the results of the digital twin simulation to a real engine with the help of presimulated failure matrices, it would be possible to determine root cause and location of possible failure as well as suggest the engine optimal actuator settings for longer lifetime, less fuel consumption and lower emissions.

2 MARINE ENGINE OVERVIEW

2.1 Marine engine types

Marine diesel engines can be divided into a few subgroups based on working cycle, filling mechanism, construction, engine speed, etc. [4]. Based on working cycle they can be distinguished between 2 stoke and 4 stroke engines. 2 stroke diesel engines are mainly applied on large vessels such as tankers, bulk carriers etc, and they are fueled with heavy oil. 4 stroke engines are also used in large vessels but mainly as a part of generator sets widely applied on a cruise vessels. 4 stroke engines are very common on medium and small length vessels such as ferries, touristic boats, speed boats, fishing vessels, etc.

Engines with different filling mechanisms are divided into the turbocharged and naturally aspirated engines. Turbocharged engines increase the pressure of the fresh air entering cylinder through the cylinder ports or intake valve by boosting the air with compressor side of the turbocharger. The pressure at the start of cylinder compression phase is higher than ambient pressure. Naturally aspirated engines do not increase the pressure of the air or the mixture before the cylinder compression phase. The pressure at start of compression phase is lower than ambient in most of the operating points with maximum achievable pressure of the one at the ambient level. In some cases, naturally aspirated can achieve pressures at cylinder inlet slightly higher than ambient due to pressure pulsation effects.

Regarding to construction the marine diesel engine can be roughly divided by the engines with and without the crosshead. The engines with the crosshead are mainly 2 stroke slow speed large engines shown on the figure 1.

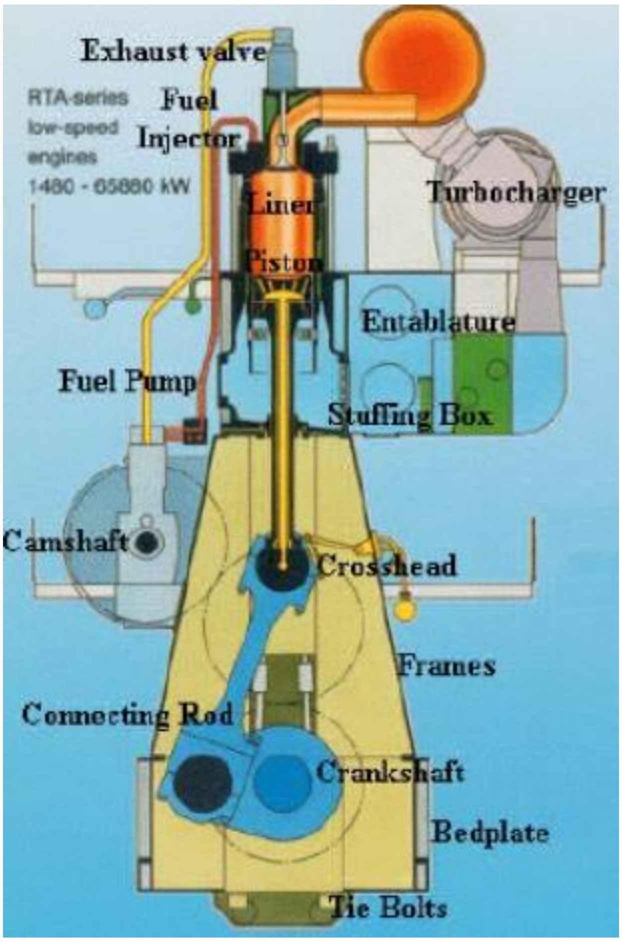

Figure 1. Construction scheme of 2 stroke slow speed marine diesel engine [5]

Tie bolts are used to tighten the tie rods which connect entablature, frame and crankcase together and serves as transmitter of the load to bedplate. The bedplate is the structure upon which frame and bearings are mounted. Crankshaft together with flying bearings converts linear piston movement to rotation. This rotation is further directly or over the gearbox transmitted on the propeller rotation. Connection rod is the rod which connects piston rod to the crankshaft. It transmits the force taken from piston rod and created by piston moving downwards during the combustion and expansion process. The frame is the structure which supports entablature and serves as a guide for the crosshead. Crosshead is the connection joint between piston rod and the connection rod. It is mainly used in slow speed engines where piston stroke is very high. Due to the long stroke, standard connection rod connected to the piston would hit the walls of the cylinder liner. For this reason, the piston rod with crosshead was introduced allowing connection rod to move freely inside the frame. Camshaft is the shaft with manufactured lobe on lobe on it. The lobe represents the lift profile of the exhaust valve. The camshaft is driven by gears from the crankshaft. The lobe is actuating the hydraulic oil under pressure which directly or influenced by electro-controlled hydraulic valves is initiating the opening of the exhaust valve. Fuel pump as well is driven by gears from the crankshaft. It is boosting the pressure in the fuel line towards the injectors. The stuffing box is the not allowing the oil from the crankcase to enter into the scavenging area and vice versa. The entablature or widely known as cylinder block. It is a housing for the scavenged air, cylinder liner and cooling space around the liner. Piston is the movable part of the combustion chamber. With piston rings it is in the contact with the liner and moves across the liner from bottom to top dead center. It transports the work produced by combustion and expansion towards the crankshaft as well as it compresses the air in the cylinder during the compression phase of the cycle. Liner is the stationary part of the combustion chamber which serves as a guide for the piston. The liner is water cooled on the outer side of the combustion chamber in order to obtain the material defined, safe operating temperature. Turbocharger is a thermal machine used to increase the pressure at the cylinder inlet ports in order to achieve higher engine power outputs as well as higher operating efficiency. It consists of the turbine and the compressor part with the shaft connecting them. Often it also contains the bypass valve for the regulation of the scavenging pressure. The turbine is driven by the enthalpy of the exhaust gasses from the exhaust collector and power produced on the turbine is delivered to the compressor. Compressor based on the flow and efficiency characteristics, is increasing the pressure and the temperature of the air at its outlet. Fuel injector is hydraulically driven nozzle which injects the fuel into the combustion chamber. The fuel needs to be at the desired pressure and temperature as well as it needs to be injected at the specific timing in order to increase the combustion efficiency and reduce the emissions. For better injection control, multiple injectors are installed in the single cylinder. The exhaust valve releases the combustion product out of the combustion chamber into the exhaust collector. It is indirectly camshaft driven and the opening is hydraulically controlled over the electro-hydraulic valve. The timing of the exhaust valve opening and closing is of high importance to the working cycle in terms of having better control over the gas exchange and residual gas content.

4 stroke marine engines differ to the 2 stroke engine in the speed range they operate which leads to construction differences, figure 2. Generally, 4 strokes engines have shorter stroke, meaning there is no need of using the piston rod and crosshead. Connection rod is connected directly from the crankshaft to the piston. This also means that the design of the frame, bedplate and the whole crankcase will be performed differently. One of the main difference is that there are no inlet ports but air filling is performed over the intake valve. The opening and closing of

the intake and exhaust valve is performed over rocker gear and is driven mainly by physical contact to the camshaft lobe.

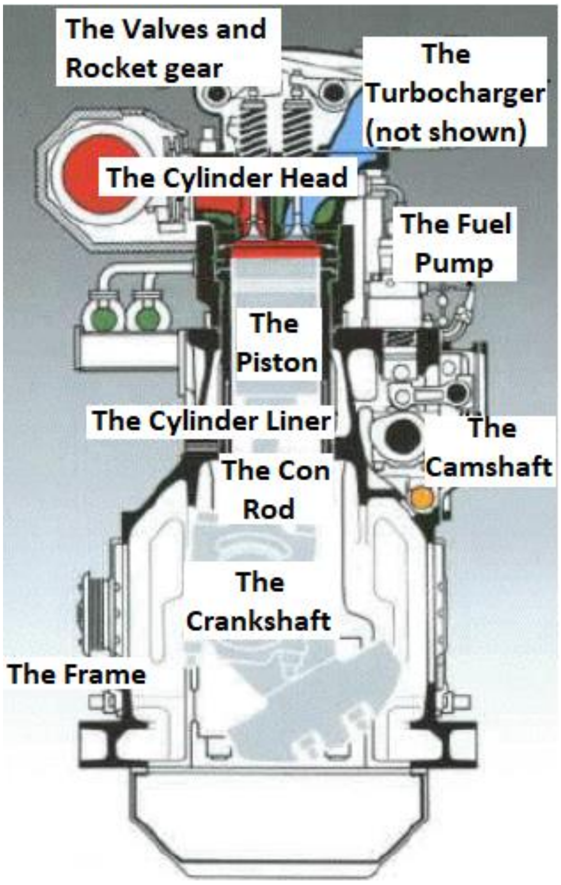

Figure 2 Construction scheme of 4 stroke marine diesel engine [5]

Marine engines can also be classified based on the operational engine speed range.

- Low speed engine operate on the speeds between 50 and 300 rpm
- Middle speed engines operate on the speeds between 300 and 750 rpm
- High speed engines operate on the speeds between 750 and 7500 rpm

2.2 Marine engine auxiliary systems

2.2.1. Hydraulic system for injection, exhaust valve and lubrication actuation

Four stroke, especially medium and high speed engines have solenoid or piezo injectors who are actuated by current, directly on the actuator. The exhaust valve actuation is performed also directly with the camshaft mechanically driven by the crankshaft rotation. The pressurized lubrication oil is delivered with the pump driven by the crankshaft rotation over the gear rotational translation. In two stoke large engines, all three above mentioned systems are actuated over the pressurized hydraulic oil with the exception of Wartsila two stroke engine which has common rail fuel system. The scheme of the marine engine hydraulic system is shown on figure 3.

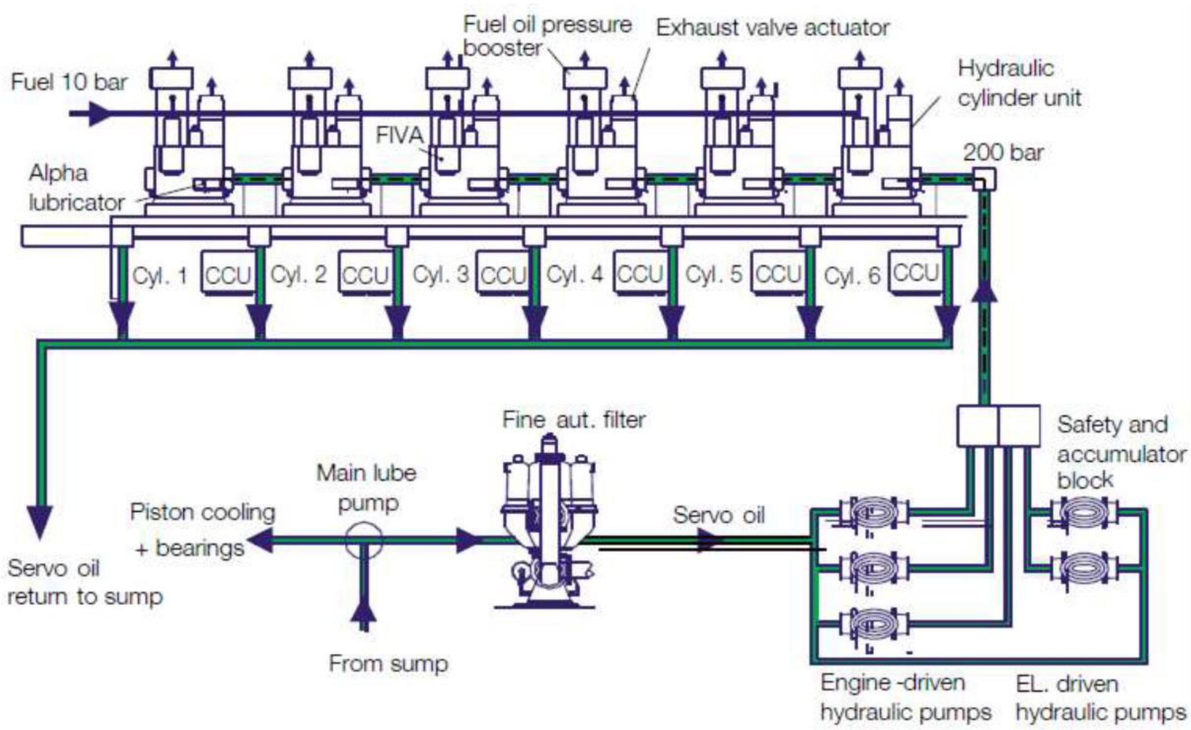

Figure 3 Marine engine high-pressure hydraulic system [6]

The main lube pump is supplying the oil towards the piston and bearing cooling circuit and fine automatic filter. After the filtration on the filter oil is compressed to 200 bar with the engine and electrically driven hydraulic pumps. The pressurized oil is than delivered over the safety and accumulator block to hydraulic cylinder units. There is one hydraulic cylinder unit per each engine cylinder. Hydraulic cylinder unit has the task of actuating fuel injection and exhaust valve opening and closing angles together marked as FIVA (Fuel Injection and Valve Actuation). Hydraulic cylinder units are linked between each other with high-pressure lines and are controlled by CYL-EU (cylinder electronic units). After the performed task in hydraulic cylinder units, oil return to the oil sump and closes the circuit loop.

Hydraulic power supply system consists of filter, pumps, hydraulic accumulator blocks and electronic pressure control system. The filter unit consists of fine automatic filter who filters the particulates with diameter higher than 6 µm or 10 µm and secondary or redundant filter who blocks the particulates with diameter higher than 25 µm [7]. The pumps used in hydraulic system are axial pumps. At least 2 pumps are driven by electric engine and are used during the startup of the hydraulic system. The rest of the pumps are driven by the combustion engine over the gear set mounted on the crankshaft. Electrically driven pumps have operating pressure to up to 175 bar and mechanically drive to 250 bar. Operation of the pumps is balanced by second order moment compensator shaft.

Hydraulic cylinder unit is electronically controlled hydraulic unit which consist of InFI Unit (Intelligent fuel injection), InVA (Intelligent valve actuation unit) and cylinder lubrication system called Alfa.

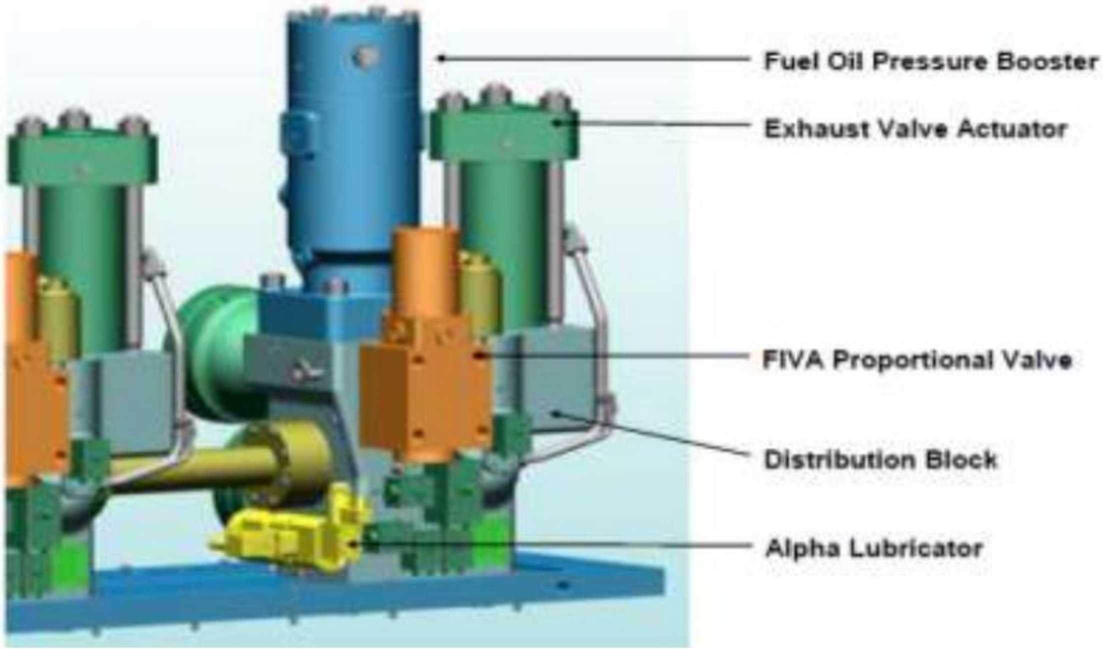

Figure 4 Hydraulic cylinder unit [8]

Additionally it contains high pressure fuel pump or fuel oil pressure booster which is a part of InFI Unit, hydraulic accumulator, FIVA valve and distribution block.

2.2.1.1 Injection system

Because of the combustion efficiency, pollutant formation and component protection, injection system needs to comply to specific injection strategies which consider delivering specific amount of fuel, at specific timing and under specific injection pressure. 4 stroke medium speed engine have fully electronically controlled opening and closing angles of the piezo injectors as well as variable control of injection pressure. The 2 stroke large engines have slightly more complex hydraulically control system. As previously mentioned MAN ME InFI unit consists of fuel oil pressure booster, high pressure pipes and slide-type fuel valves.

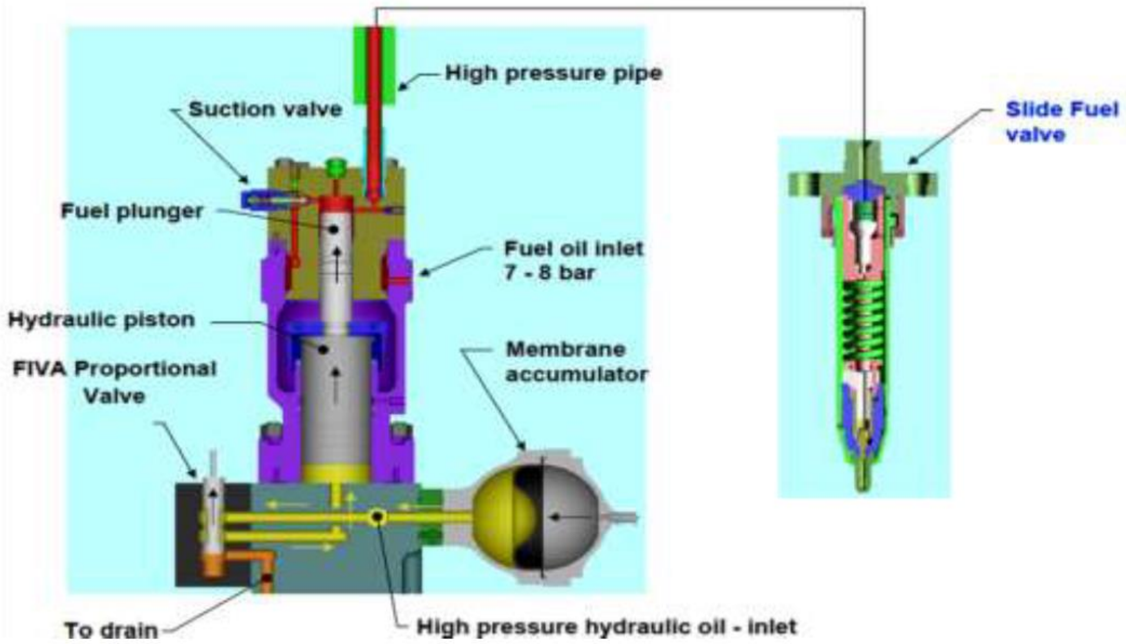

Figure 5 InFI Unit [8]

Based on the engine operating state, load and crankshaft position signal, FIVA proportional valve is actuated and directs the hydraulic oil towards the hydraulic piston located in fuel oil pressure booster. Hydraulic piston over the fuel plunger compresses the fuel which is supplied through suction valve with delivery pressure around 10 bar to the operating injection pressure to up to 1000 bar. When the pressure inside the high pressure pipe outpowers the spring force, the slide fuel valve starts to inject the fuel into the engine cylinder and FIVA valve releases the hydraulic fuel towards the oil tank. FIVA valve is controlled by cylinder control unit. The control unit considers the feedback position of the FIVA valve and fuel booster hydraulic piston position. It also gives the possibility to adjust the injection timing.

2.2.1.2 Exhaust valve actuation system

In 4 stroke medium speed engines exhaust valve actuation is rather simple. The opening and closing of the intake and exhaust valve is performed over rocker gear and is driven mainly by physical contact to the camshaft lobe. In 2 stroke large engines the exhaust valve is actuated with the electronically controlled hydraulic system which is on MAN ME example called InVA, figure 6.

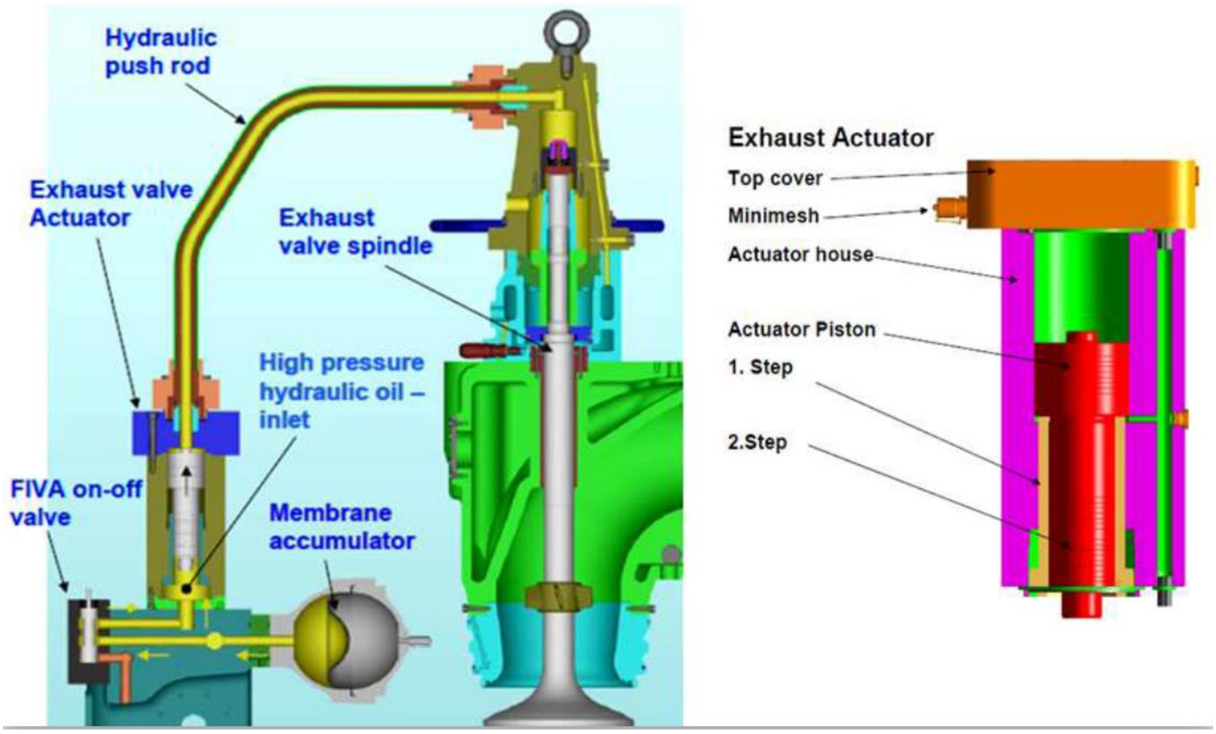

Figure 6 InVA Unit [8]

InVa consists of FIVA valve, exhaust valve actuator, hydraulic rod and exhaust spindle. At the end of expansion the exhaust valve start to open in a way that FIVA valves open and pressurized hydraulic oil is released towards the exhaust valve actuator who translates the hydraulic force through hydraulic push rod to the exhaust valve. After the gas exchange phase, the exhaust valve closes by reversing the FIVA valve who lets hydraulic oil circulate back to the oil thank and exhaust valve is by air spring returned to the initial position. The adjustment of exhaust valve opening or closing angle is done by cylinder control unit and commanded by multi purpose controller.

2.2.1.3 Lubrication system

At 4 stroke engines lubrication system is mainly based on mechanically driven oil pump, filter, oil cooler and lubrication lines leading to cylinders, camshafts, etc. In 2 stroke engines each cylinder has its own lubrication which on example of MAN ME engine is electronically controlled and called Alfa lubrication system. Alfa lubrication system together with exhaust valve and injection actuation is a part of hydraulic cylinder unit.

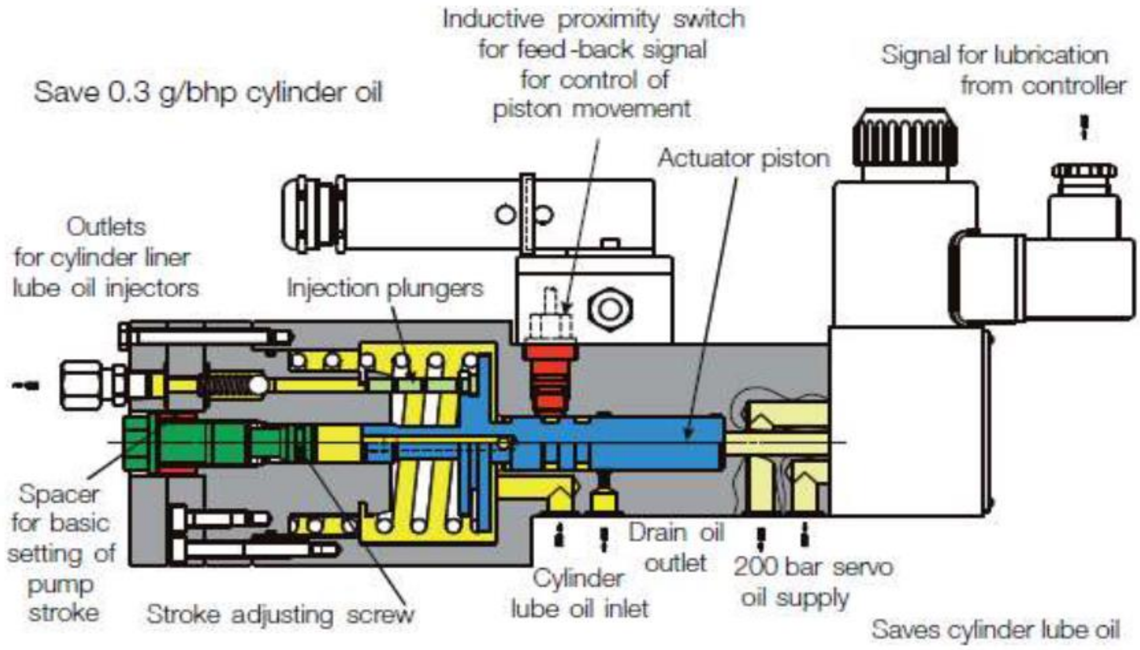

Figure 7 Alfa lubricator [8]

Signal for lubrication from controller lets the hydraulic oil to move the actuator piston and over the injection plungers to inject lubrication oil. After the signal has been terminated the spring moves actuator piston to its initial position and oil injection is terminated. Movement length of the actuator piston is adjusted manually over the stroke adjusting screw. Control of the lubrication system is also demanded by multi purpose controller and is based on the feedback signal from the inductive proximity switch.

2.2.2 Fuel supply and preparation system

The four stroke marine engines mainly use diesel oil which is delivered in the fuel tank already prepared for the injection. The fuel is further filtered and delivered to the high-pressure fuel system with low pressure fuel delivery pump. The fuel pressure is than increased by lobe driven high pressure fuel pump and delivered to the rail, which is connected to the injectors over the high pressure fuel pipes. Large two stroke engines operate constantly on heavy fuel oil and diesel fuel is only used for the maneuvering or engine start. Heavy fuel oil or otherwise called bunker oil is remnant from the distillation of the crude oil. It is low quality fuel substance and it is used in the marine engines mainly due to costs. Heavy fuel oil also consists of sulfur, nitrogen and aromatics which produce harmful pollutants during the combustion process. It is important to mention that marine engines in some sea areas are not allowed to use heavy fuel oil as the only fuel for powering the engine, but rather mixture with diesel or purely diesel fuel. One of the important characteristics of the heavy fuel oil is the high viscosity, which does not allow the fuel to be used directly into the injection system but requires fuel preparation. The fuel preparation system consists of filtration tanks, heaters and purifiers in order to get the heavy fuel oil transportable and usable for combustion process.

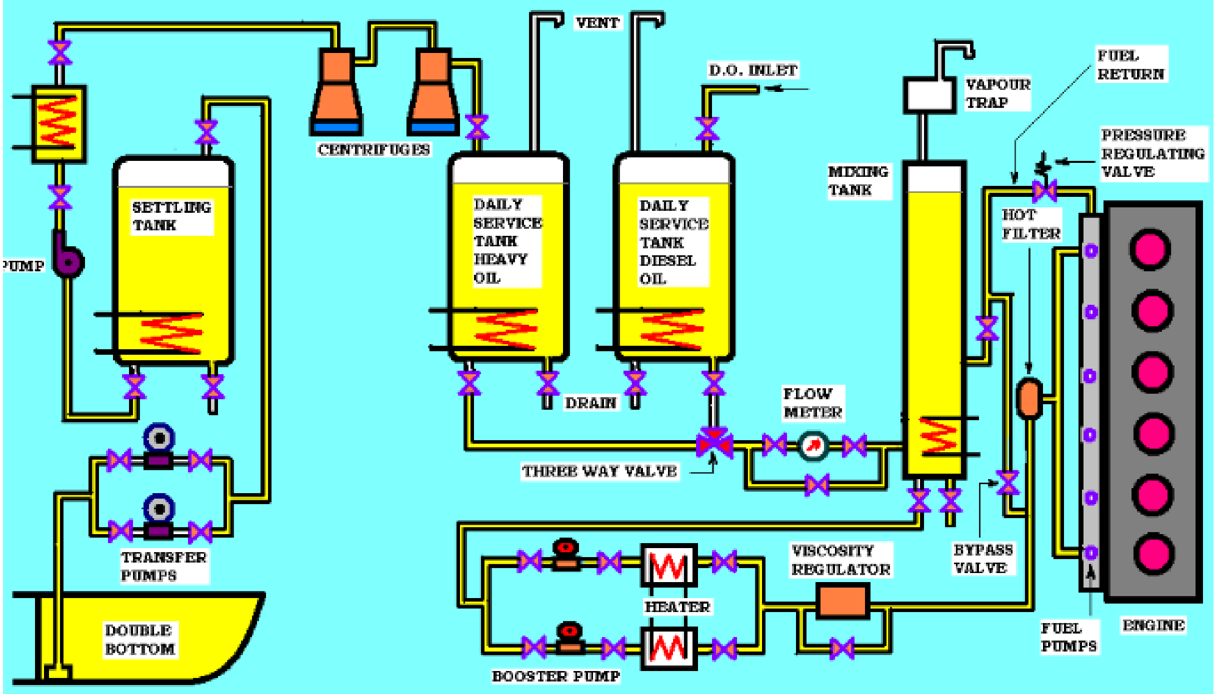

Figure 8 Heavy fuel oil supply system [9]

Heavy fuel oil when boarded is stored in bunker tanks where it is heated to achieve the viscosity level which allows the oil to be pumped by transfer pumps into the settling tank. Settling tank has the volume capacity for 24 hours engine fuel demand. In settling tank heavy fuel oil is heated as well and water as well as the heavy impurities are separated by gravity on the bottom of the tank. The impurities need to be drained regularly. Heavy fuel oil is than pumped into the centrifuges where all impurities are removed from the fuel. The fuel oil is than pumped into the daily service tanks and over the three way valve and flow meter to mixing tank. Over the three way valve it is possible to choose if heavy fuel oil, diesel fuel or mixture will be used. The mixing tank task is to provide smoother transition from diesel fuel to heavy fuel oil operation by allowing gradual change in temperature and viscosity of the fuel. It is also used as fuel recirculation tank. Booster pumps boost the fuel to desired pressure and over the viscosity regulator by adjusting the temperature, the optimal viscosity for combustion is controlled. Pressure regulating valve controls the supply pressure for the pumps who boost the fuel pressure which is used in InFI system. Bypass valve with hot filter is used to preheat the fuel before stating the engine.

2.2.3 Cooling system

The engine cooling system onboard ships is separated into the sea water and fresh water circuit. Fresh water is used to cool machinery and sea water as secondary circuit to cool down the fresh water circuit. Four stroke medium speed engines get cooled cylinder jackets, lubrication circuit and charge air cooler by the fresh water circuit. In large two stroke engine the cooling circuit is slightly more complex. Figure 9 shows one typical schematic of the fresh water cooling circuit of the large two stroke low speed diesel engine. Fresh water cooling circuit in this application cools down the jacket, piston, lubrication oil and fuel valve. For each of the stated there is a cooling circuit applied. Additionally the blower and charge air cooler can be cooled by fresh water or directly with sea water. Jacket cooler water pump pumps the water in the compartments around the cylinder liner and head, turbocharger and exhaust valves. The water is circulated further towards jacket cooler where the temperature is lowered by heat exchange with sea water. The expansion tank is also mounted onto the circuit to compensate the water loss and to minimize the pressure oscillations. There is also the control valve which regulates the water flow for each cylinder individually.

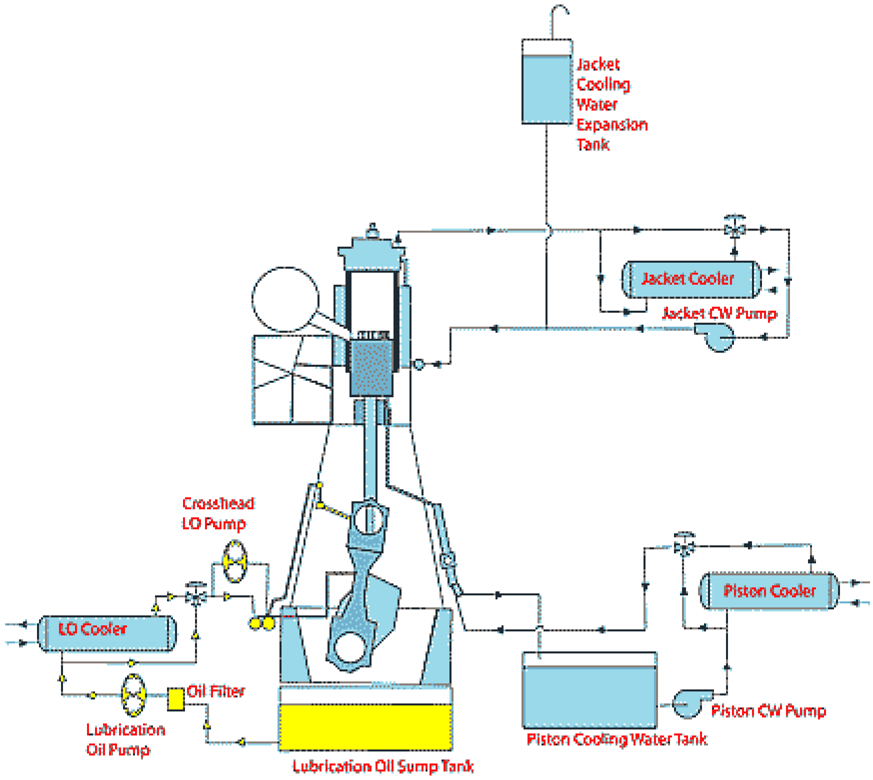

Figure 9 Fresh water cooling circuit [10]

The piston cooling pump pumps the water either to piston cooler which drops the water temperature by heat exchange with the sea water or towards the piston depending on the position of the control valve. The water flows towards the piston through telescopic and stand pipes and after cooling the piston it flows out to piston water tank.

Lubrication oil pump pumps the oil into the oil cooler to cool it to operating temperature or directly to the moving parts of the bearings depending on the position of the control valve. After the lubrication the oil will become heated and will drop into the oil sump tank.

The injected fuel temperature is important for the combustion efficiency and pollutant formation. For this reason injectors are drilled with passages through which the water circulates and controls the temperature of the injected fuel. This system as well as jacket cooling system is closed and requires the expansion tank.

2.2.4 Air starting system

The pneumatic air starting system is specific for large marine engines. The starting of the engine is done by injecting pressurized air into the engine cylinders at TDC until exhaust valve opens and by this pushing the cylinder downwards. The air is pressurized by the compressors and stored into the receivers. The pressure inside the receivers is around 30 bar. The air from the receivers is delivered over the automatic non-return valve to the air start valve which is mounted on each engine cylinder. The opening of the valves is controlled by pilot air system.

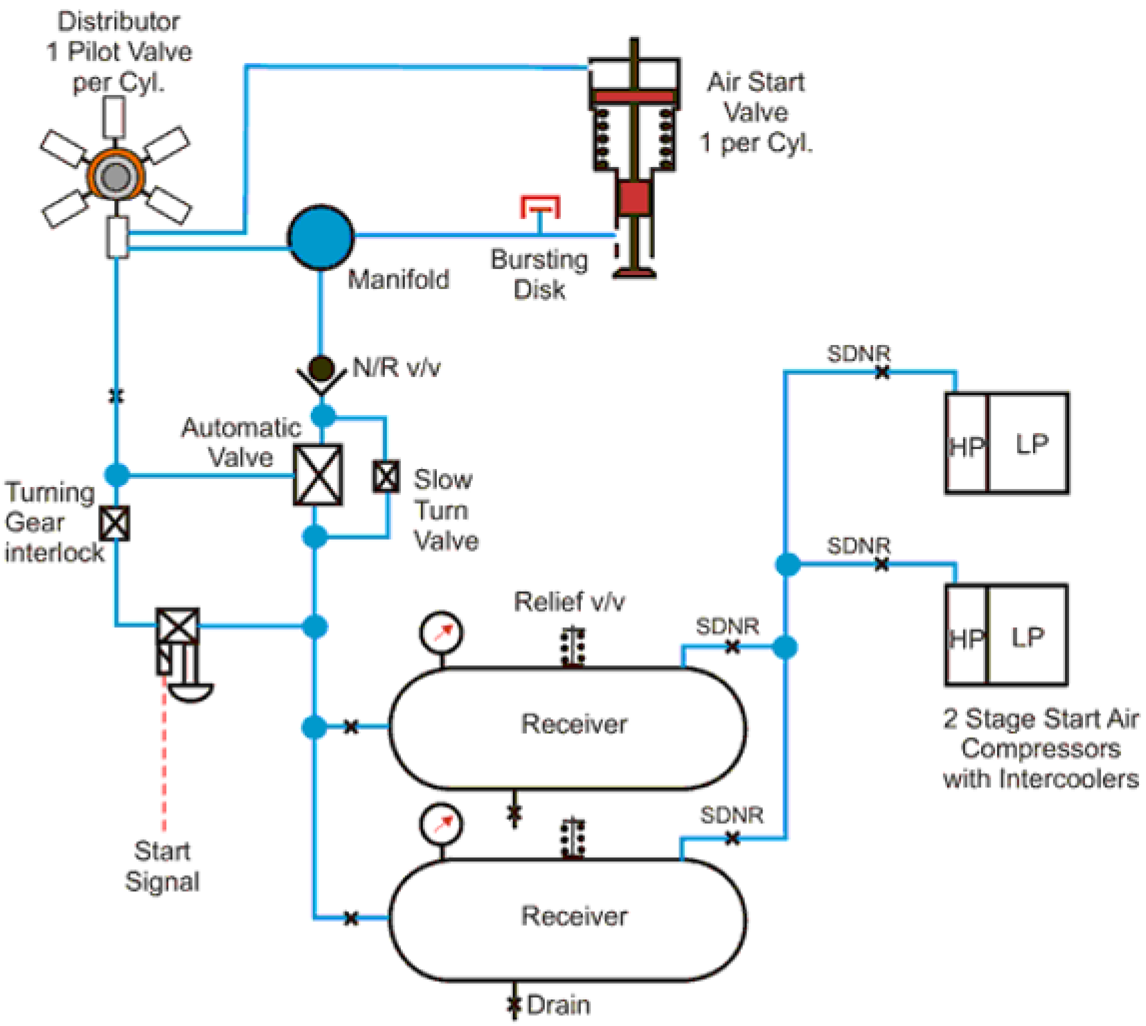

Figure 10 Air starting system [11]

After the start signal is applied the pilot air activates the automatic valve. The distributor receives the pressurized air as well, which will be based on the activation order will activate the air start valve. The distributor is synchronized with the crankshaft position by receiving the crank angle signal to activation control unit or mechanically over the camshaft, aiming to activate the air start valve when cylinder is at TDC. There are some safety measures to be applied on the starting system such as interlock which stops the activation of automatic valve. Compressor lubrication oil will accumulate on the pipelines of the air system and under the air start valve leakage the high temperature gasses from cylinder can cause oil fire. For this reason flame traps together with non-return and relief valves are installed.

3 MARINE ENGINE CONTROL LAYOUT

Driven by emission legislation, engine manufacturers were forced to develop and implement newer types of actuators who are able to adjust the fuel injection, air filling and other emission relevant parameters with more degrees of freedom and more accuracy. As in the previous chapters, the difference between 4 stroke and 2 stroke is rather noticeable.

Four stroke engine control module based on the load demand and sensor information listed below:

- Engine speed and timing
- Fuel pressure and temperature
- Gear oil pressure and temperature
- Engine oil pressure and temperature
- Intake manifold air pressure and temperature
- Coolant temperature and level

is determining how much fuel needs to be injected in order to achieve desired torque. After fuel determination, the emission and component protection parameters are defined by setting the desired injection pressure, number of injections and injection timing. Based on desired rail pressure the high-pressure fuel pump actuator angle and injection timing for injectors is calculated and finally control module releases current towards the actuators.

There are also additional actuators and sensors installed depending on engine concept. For example, if exhaust gas recuperation (EGR) is used, the EGR valve is controlled and fresh air mass flow meter is installed. For variable turbine geometry the control for the blade position is being used.

For 2 stroke engines the control layout is more complex. The complexity comes due to large stroke engine design which is more exposed to cylinder liner wearing and thus has more necessity for cylinder balancing and as well more complex auxiliary hydraulic system for injection and exhaust valve control. The main control module also controls starting, stopping and reversing engine operation. The control layout is shown on figure 11.

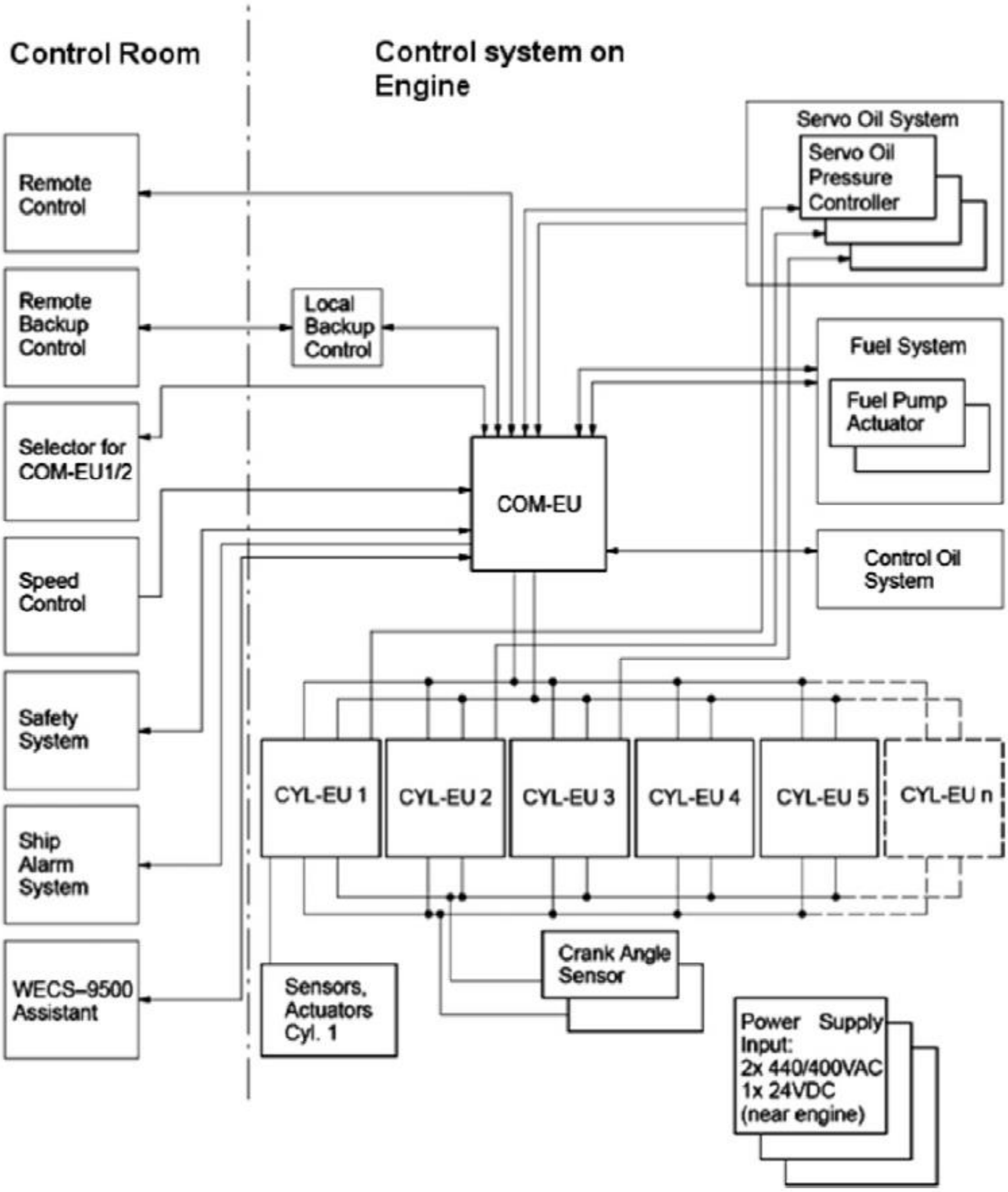

Figure 11 Large marine engine control layout [12]

The central part of the control layout is common electronic unit (COM-EU) which coordinates lubrication system, fuel supply system, hydraulic oil system and CYL-EU. It takes commands of speed and load from engine room and communicates with engine safety and alarm system. There is a single COM-EU per engine and per each cylinder there is one CYL-EU. The thermodynamic control logic is similar as described in 4 stroke engines with higher complexity on actuation periphery condition preparation in hydraulic servo system. In 2 stroke engines there is larger number of sensors installed. The pressure and temperature sensor is installed after or in each engine component from perspective of the cylinder block, air path, lubrication, cooling and hydraulic system. With different engine or driveline concepts such as dual fuel engines, gas engines, engines with utilization boilers, diesel generators, etc., the additional sensors are added. Bellow listed is the example of most important sensors on conventional 2 stroke large engine:

- Ambient pressure and temperature
- Pressure and temperature after the air filter
- Pressure and temperature after the compressor
- Scavenge pressure and temperature
- Exhaust collector pressure and temperature
- Pressure and temperature after the turbine
- Turbocharger speed
- Engine speed or crankshaft position
- Incylinder pressure indication traces
- Coolant circuit temperatures
- Oil pressure and temperature
- Oil mass flow
- Fuel booster position
- Fuel pressure
- Hydraulic oil pressure
- Exhaust valve actuator position
- Cylinder liner temperature
- Main and crosshead bearing temperatures

Based on sensor information, speed and load demand, the COM-EU based on precalibrated control functions sends information to CYL-EU who actuates FIVA system as described in chapter 2.2.1.1. and 2.2.1.2.

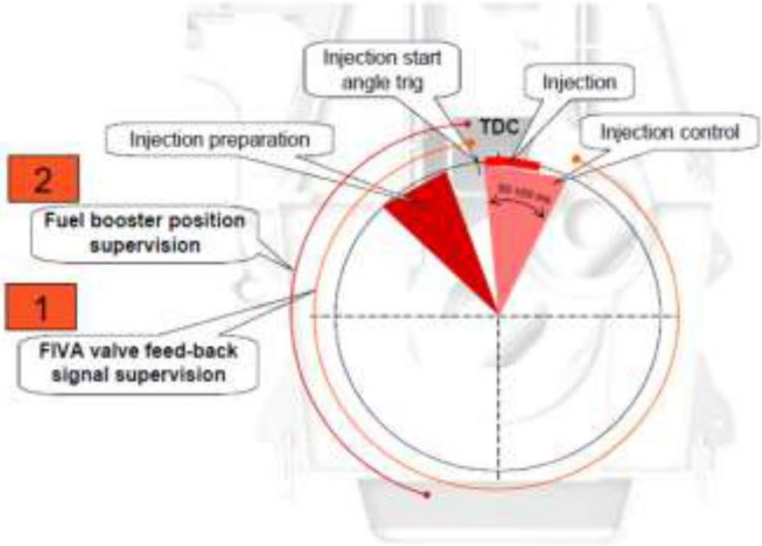

Figure 12 Fuel injection process [8]

The injection process consists of preparation phase where fuel reaches desired injection pressure, start of injection where injector starts to release the fuel into the cylinder, injection phase and injection control of combustion monitoring. Prior to injection, control unit is diagnosing if FIVA feedback current signal or fuel booster position are faulty. The injection parameters, injection pressure, start angle and duration are calibrated inside the control unit in order to achieve optimum cylinder balancing, desired in-cylinder peak pressures, optimal combustion efficiency and desired emission targets.

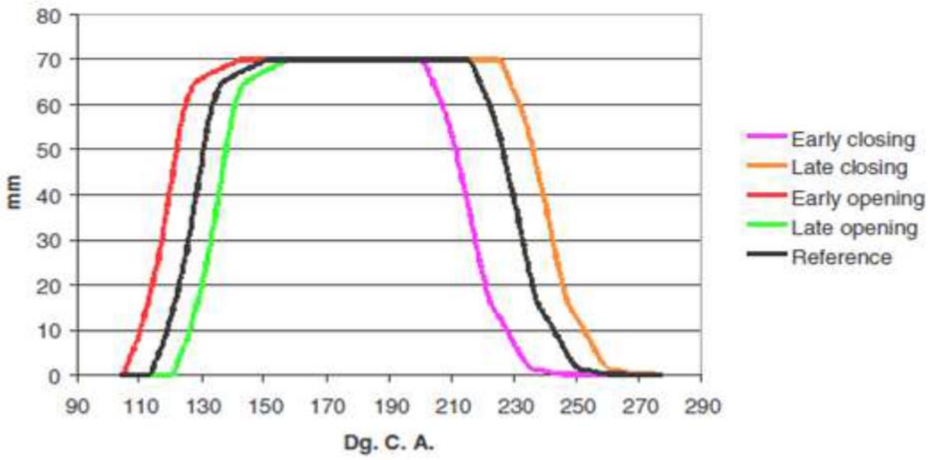

Figure 13 Exhaust valve opening and closing profiles [6]

The exhaust valve is also variably controlled over FIVA system with the control unit. It is possible to do variations of opening and closing angles of the exhaust valve. These variation are mainly used to improve gas exchange, control the compression pressure, exhaust temperature and compensate leakage over the exhaust valve.

4 ENGINE PROTECTION AND TYPICAL FAILURES

Engine is the most expensive and safety critical item on the vessel. Due to aging and different operating load profiles due to rough sea conditions it is also subject to failures. For the exact reason, engine safety system are introduced. Safety systems can be delivered by engine manufacturers or external tools can used as well.

4.1 The safety features and typical failures for the slow speed large marine engines

The safety features can be grouped in:

- Alarms
- Slow down
- Shut down
- Start Interlock

Alarms are sound and light warnings when one or multiple sensor values are over or under exceeding the nominal value of the specific sensor with predefined tolerance windows, Figure 14. If the reactions were not sufficient to obtain the problem and the parameters did not return to tolerance windows, the slow down procedure is initiated in order to protect the engine. The slow down procedure is related to reduction of engine speed and load. The example of scenarios which activate slow downs for the large engines are [5]:

- Lube oil pressure drop
- Cam shaft oil pressure drop
- There is no flow of piston cooling media (water or oil)
- Oil mist detector or Main bearing sensors has been activated
- Lube oil temperature at the inlet of engine is high
- Piston Cooling temperature is high
- Jacket water Temperature is high
- Engine outlet cylinder exhaust temperature is high
- Scavenge air temperature is high
- Thrust block temperature is high
- Low flow or pressure drop of the Cylinder lube oil
- Control air pressure is low
- Turbocharger oil pressure drop

System	Location of measurement	Gauge pressure limit values [bar]		Temperature limit values [°C]		
		Min.	Max.	Min.	Max.	Difference
Freshwater						
Cylinder cooling	Inlet	3.0	5.0	70	-	max. 15
	Outlet each cyl.	-	-	set point 90 *1)		
SAC LT circuit (single-stage SAC)	Inlet Cooler	2.0	4.0	25	36	*2)
	Outlet Cooler	-	-	-	80	
Cylinder lubrication						
Cyl. lubricating oil, static pressure	Inlet Engine	0.4	-	35	50	-
Crosshead lubrication Crosshead LO booster pump **needed** for engines with CMCR speed range 102 rpm ≤ CMCR speed < 109 rpm Crosshead LO booster pump **optional** for engines with CMCR speed range 109 rpm ≤ CMCR speed ≤ 127 rpm						
Crosshead bearing oil	Inlet Engine *3)	10.0	12.0	40	50	-
	Inlet *4)	3.6	5.0	40	50	
	Outlet	-	-	-	65	-
Main lubrication						
Servo oil	Servo oil pump inlet	3.6 *5)	5.0	-	-	-
Main bearing oil	Inlet	3.6	5.0	40	50	-
	Outlet	-	-	-	65	-
Piston cooling oil	Inlet	3.6	5.0	40	50	max. 30
	Outlet	-	-	-	80	
Thrust bearing oil		-	-	-	65	-
Torsional vibration damper (steel spring damper)	Supply	3.6	5.0	-	-	-
	Inlet Damper *6)	2.8	5.0	-	-	-
Axial vibration damper (chamber pressure)	Supply	3.6	5.0	-	-	-
	Monitoring	1.7	-	-	-	-
TC bearing oil (on engine lub. oil system), ABB A100-L	Inlet	1.0	2.5	-	-	-
	Outlet	-	-	-	110	-
TC bearing oil (with separate lub. oil system), ABB A100-L	Inlet	1.3	2.5	-	85	-
	Outlet	-	-	-	130	-
TC bearing oil MHI MET MA	Inlet	0.7	1.5	-	-	-
	Outlet	-	-	-	85	-
Fuel oil						
Booster (injection pump)	Inlet	7.0 *7)	10.0 *8)	-	150	-
After pressure retaining valve	Return	3.0	5.0	-	-	-
Scavenge air						
Intake from engine room (pressure drop, max)	Air filter / silencer	max. 10 mbar		-	-	-
Intake from outside (pressure drop, max)	Ducting and filter	max. 20 mbar		-	-	-
Cooling (pressure drop)	New SAC	max. 30 mbar		-	-	-
	Fouled SAC	max. 50 mbar		-	-	-
Air						
Starting air	Inlet Engine	12	25/30	-	-	-
Control air	Engine inlet (engine internal)	7.0	9.0	-	-	-
Air spring air for exhaust valve	Main distributor (engine internal)	6.0	7.5	-	-	-
		normal 6.5				
Exhaust gas						
Receiver	After each cylinder	-	-	-	515	Dev. ±50 *9)
	Before each TC	-	-	-	515	-
Back pressure in manifold after turbocharger	Design maximum	30 mbar		-	-	-
	Fouled maximum	50 mbar		-	-	-

*1) Tolerance steady state condition: ±2 °C. Tolerance transient condition: ±4 °C.
*2) The water flow has to be within the prescribed limits.
*3) Pressure range for engines **with** Crosshead LO booster pump.
*4) Pressure range for engines **without** Crosshead LO booster pump.
*5) Pressure can be 0.8 bar lower than indicated due to the specified max. allowable pressure difference over the fine filter.
*6) The alarm value can be different.
For the applicable setting value, refer to the specification of the damper manufacturer.
*7) At 100 % engine power.
*8) In stand-by condition; during commissioning of the fuel oil system the fuel oil pressure is adjusted to 10 bar.
*9) Maximum temperature deviation among the cylinders.

Figure 14 The example of Pressure and temperature ranges [13]Wärtsilä RT-flex48T-D

If there are high deviation for the two stroke large engines in sensor parameters such as engine oil pressure, the shut down procedure is started and all fuel supplies are cut down. The scenarios which activate shut downs for large engines are [5]:

- Lube oil inlet pressure to engine is very low <1 bar
- Cam shaft Lube oil pressure is very low < 1.5 bar
- Very high Jacket cooling water temperature >95 deg C
- Low Jacket cooling water pressure < 0.1 bar
- No flow of Cylinder lube oil
- Thrust block temperature very high > 90 deg C
- Lube oil inlet pressure for turbocharger is low < 0.8 bar
- Over speed of the engine which activates shut down at 107 % of Max. continuous rating MCR
- Oil mist detector
- Turbocharger oil pressure

Start interlock will not allow engine to be started if all the systems are not initiated or operated properly.

One of the ways to monitor the performance and engine health is the analysis of the in-cylinder pressure traces indication diagrams which commonly are not available for the medium speed four stroke engines but exclusively for the large engines, Figure 15.

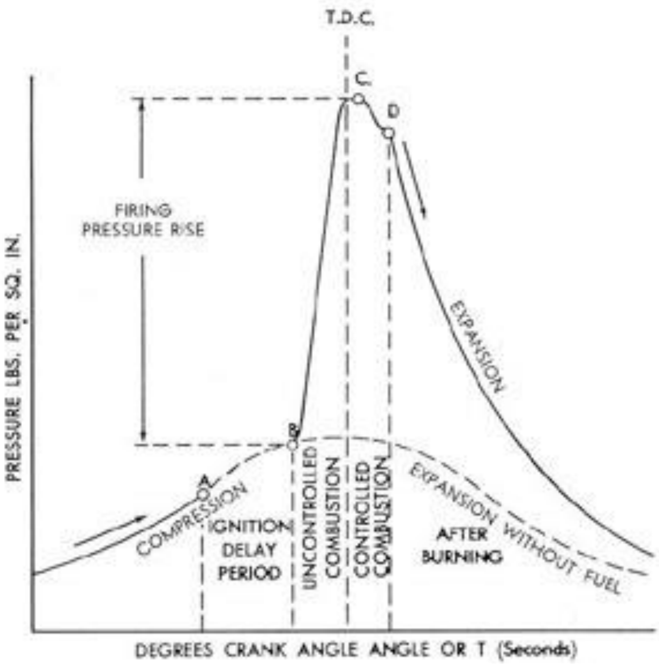

Figure 15 In-cylinder pressure trace [5]

When the measured diagram is compared to the reference one of the healthy engine a few types of typical deviations can be observed:

1. Compression pressure shows no deviation but peak firing pressure is too high. This indication on false injection setting pointing to the position of the fuel timing camshafts or settings with possibility of the failure of FIVA system. The high peak firing pressure is caused by too early injection and if all fuel injection control timing setting are valid, the only cause left is damaged injector leakage.
2. Compression pressure and peak firing pressure are low. This indicate on air and mixture leakage through exhaust valve, piston rings, wearing of the cylinder liner, burnt piston crown or low scavenging pressure.
3. Compression and peak firing pressure are high. This indicates on false exhaust valve timing. Late opening of exhaust valve means prolonged expansion which bring more work into the compression stroke of upcoming cylinder in the firing order. This can also indicate overload of the engine.

The mentioned symptoms or failures are cylinder individual. When monitoring the whole cylinder group, it is important to balance the performance between the individual cylinders. The cylinder or power balancing is mainly done by adjusting the fuel amount per each cylinder and by not exceeding safety measures such as too high exhaust temperature. Unbalanced operation can lead to overload of bearings and gears, vibrations, overheated pistons, cracked crankshaft, bolts failure, etc.

4.2 The typical failures for the medium speed mainly four stroke engines

Common approach to group the failures based on symptoms and especially for medium speed 4 stroke engines where extensive monitoring devices on board are not available is [14]:

1. Engine not able to start
2. Overheating
3. Slow down
4. Oil pressure

1. Engine not able to start:

Engine not able to start is most probably due to a not sufficient air or fuel supply. This means all fuel and air paths in the engine must be unconstrained.

If the engine cannot overcome the cranking phase when starter motor is active the list of causes and action suggestions as stated by [14] are:

Battery capacity low:

- Check that electrolyte level is above the plates.
- Try to start the engine on the other bank of batteries. Failing this, try to start the engine on both banks of batteries. Never continue to use a battery if the starter motor is sluggish because high discharge rates will buckle the battery plates.
- Take the specific gravity of each cell of the battery. A fully charged battery would have a specific gravity reading in each cell of 1.26 where as a flat battery would give a reading of 1. 1 0. The specific gravity reading should not vary more than 0.030 between cells. A lower reading on one cell usually indicates the battery needs replacing

Battery connection dirty:

Check that the connections to and on the battery is clean and tight. A dirty or loose connection can be identified by the heat it generates.

Bad electrical connection to starter motor:

The starter motor draws the most load on the battery especially on diesel engines because of their high compression ratios. The electrical connections must therefore be tight and clean.

Faulty starter motor:

The starter motor could be burnt out or the pinion is not engaging with the ring gear on the flywheel.

Incorrect grade of the lubrication oil:

If the oil is too thick, the engine will not attain sufficient speed on the starter motor to generate the amount of heat required on the compression stroke to ignite the fuel.

Engine has been overhauled and it is tight:

The parts of an overhauled engine are brought back to their correct clearances. In these clearances there will be a number of high spots and they will be worn away as the engine is run in. When the engine is run in, it will turn easily. The engine will not attain sufficient speed on the starter motor to generate the amount of heat to ignite the fuel.

Air cleaner restriction:

The air cleaner is choked, blocking or restricting the air required for the engine.

Exhaust gas restriction:

- Could be caused by a bucket left on the outlet of a vertical exhaust pipe to prevent rain water entering the engine or by the automatic flap valve fitted for this purpose and is stuck in the closed position.
- Occasionally a baffle could come loose in a silencer and block the passage of exhaust gas.
- The air must be compressed to a high enough temperature to ignite the fuel. This is usually due to low or poor compression. Compression pressure can be checked by replacing each fuel injector in rum with a compression gauge.

Fuel tank empty:

Fuel piping could develop a leak emptying the contents of the fuel tank into the bilges.

Blocked fuel feed line:

The suction valve on the fuel tank could have vibrated closed or someone could have closed the emergency fuel shut off valve.

Faulty fuel lift pump:

Fuel is not being delivered from the fuel tank to the engine. If it is of the diaphragm type, the diaphragm could be perished or damaged. The drive to the pump could be damaged.

Choked fuel filter:

The fuel filter has choked up with foreign matter as to prevent the full flow of fuel. The filter may not have been changed at its recommended period. A bad batch of fuel may have been received. The filter may require changing at more frequent intervals until the system is clean.

Air in fuel system:

Air is compressible where fuel is not. Air in a fuel system will cause the engine to malfunction or not start. Air usually enters the fuel system when repairs are carried out or where there is a fuel leak. This air must be bled off until a bubble free fuel is obtained. Some fuel systems have a manual priming handle on the fuel lift pump or on the fuel injection pump. In addition, there are bleed valves throughout the system, such as on filters or water separators.

Faulty fuel injection pump:

The pump is not delivering the fuel to the injector

Faulty fuel injectors:

The valve pintle may be seized shut in its nozzle and no fuel is delivered to the cylinder. The holes or orifices in the nozzle may be blocked. The valve pintle may not be sealing on its seat.

Incorrect fuel pump timing:

The fuel is not being delivered to the fuel injector at the precise moment in the cycle. The engine could have been overhauled and the timing of the fuel pump was incorrectly carried out. It is possible for the timing to alter whilst the engine is running due to insufficient tension on the fuel pump coupling bolts.

2. Overheating:

Overheating is detected by reading high temperature values of the cooling, oil or exhaust temperature gauge. If the temperature increase gradually it is probably sign of wearing or strained flow surfaces and sudden increase in temperature would mean faulty thermostat, pump or engine overload.

The list of the failure root causes is as stated by [14]:

<u>Sea water intake grid:</u>

Could become clogged over a period of time so there would be a gradual increase in the fresh water cooling temperature. Reduce the engine speed until the normal operating temperature is obtained, or take steps to remove the obstruction.

<u>Clogged sea water strainer:</u>

- Could become clogged over a period of time so there would be a gradual increase in the fresh water cooling temperature. Reduce the engine speed gradually and stop the engine. Clean out the strainer. Start the engine and let it idle until temperature stabilizes.
- Alternately the vessel may have voyaged through matter which quickly clogged the strainer. Take the above action.

<u>Thermostat not opening fully:</u>

- When the engine is cold the thermostat is in the closed position. Water is circulated through the engine only. As the engine reaches its operating temperature, the thermostat opens and allows the water circulating through the engine to pass through the fresh water cooler or the keel cooling pipes.
- Should the thermostat stay in its closed position or not open fully, the engine will overheat. Feel the pipe from the thermostat housing to the fresh water cooler. This will indicate whether or not water is flowing through it. Reduce the engine speed gradually and stop the engine. When the engine has cooled down, replace the thermostat. Should you be at sea and have no replacement thermostat, the engine can be run with the a thermostat removed.

Faulty impeller in sea water pump:

- A faulty impeller in the sea water pump could be damaged. Damage usually occurs when the pump is run dry. The discharge pipe would be warm and not at the same temperature as the sea water. Also, there would be no sea water discharge overboard. Reduce the engine speed gradually and stop the engine. Replace the impeller.
- Should you be at sea and have no replacement impeller, it may be possible to reach port at reduced speed, if the impeller is only partially damaged and can still pump some water. Alternately, a sea water hose from the fire pump or the wash deck hose could be connected up to the system at the discharge.

Keel cooling pipes low efficiency due to marine growth (only for keel cooled engines):

This causes a gradual increase in the fresh water temperature. Reduce the engine speed gradually until the normal operating speed is obtained. The vessel will have to be slipped to clean the keel cooling pipes.

Air in a sea water cooling system:

On a lot of vessels, air is trapped in the sea water cooling system when the vessel re-enters the water after slipping. With the engine stopped, the air can be bled off by slackening off the backing plate on a pump or loosening a join in the seawater cooling pipe on the suction side of the pump that is below the water line. If it is a pump and it is run dry until the engine overheats, the rubber impeller will be severely damaged.

Low speed of the water pump:

On some vessels the sea water pump is belt driven from the engine. The adjustment of the belt may cause it to slip. Reduce the engine speed gradually and stop the engine. Adjust the belt tension.

Faulty impeller:

A faulty impeller in the fresh water pump could be damaged. Reduce the engine speed gradually and stop the engine. Replace the impeller. Should you be at sea and have no replacement impeller, it may be possible to reach port at reduced speed if the impeller is only partially damaged and can still pump some water.

Scale buildup on cylinder water jackets:

Fresh water contains impurities. They come out of solution at high temperatures and will adhere to hot surfaces. The scale will stop the transfer of heat from the combustion process to the fresh water cooling and, in the case of passages, will restrict the flow. This will be a gradual process. Reduce the engine speed until normal operating temperature is attained.

Tight engine parts causing friction:

A new or overhauled engine normally runs hotter because it is tight. As the engine is run in, the high spots disappear. The engine turns easily, thereby reducing the operating temperature. Reduce the engine speed so that it runs at its normal operating temperature.

Cooling water level too low:

- A leak has developed in the fresh water system causing a loss of water in the header tank. It could be a leak in the piping, seal in the pump or a blown cylinder head gasket. Reduce the engine speed gradually and if the fresh water system is the unpressurised type, very slowly top up the header tank to its correct level.
- If the fresh water system is of the pressurized type, reduce the engine speed gradually and stop the engine. Let the engine cool down. Start the engine and very slowly top up the header tank to its correct level. If there is very little water in the header tank, it is advisable to stop the engine and let the engine cool right down before adding fresh water. If possible, the leak should be repaired.

Blown cylinder head gasket:

A cylinder head gasket leaking will be indicated by bubbles in the header tank. The extent of the leak will determine the amount of bubbles. However, when the engine is stopped, there is no pressure in the cylinder. The header tank is above the cylinder thereby putting pressure (a head) on the water. The water would then flow through the leak in the cylinder head gasket into the cylinder.

Engine overload:

Engine in the overload can enter due to dirty hull, rope around the propeller, bent propeller blade, large pitch propeller. The engine speed should be reduced until the normal operating temperature is attained. To stop overheating, it would be necessary to

clean the hull of marine growth, or repair any damage to the prop.

Dirty or damaged fresh water cooler:

The sea water discharged overboard would be restricted. It is unusual for the cooler to be completely blocked. Reduce engine speed until normal operating temperature is attained. Stop engine and clean the cooler or return to port under reduced speed.

3. Slowdown:

Slowdown of the ship during voyage can lead to a dangerous outcomes especially under severe weather conditions. The most common cause of the slowdown is the fuel supply. The list of the possible failure root causes is as stated by [14]:

Fuel tank empty:

Isolate the damaged fuel tank and use another one

Fuel tank outlet pipe split or corroded:

Carry out temporary repair to pipe by placing a piece of rubber around it, then a thin bit of metal to give the rubber some support and attach them with hose clips. Isolate empty fuel tank and use another fuel tank.

Water in the fuel

Drain off the water at the fuel tank and, if fitted, at the water separator.

Clogged fuel filter

Depending on the type of filter, clean the filter or replace the disposable element.

Faulty fuel lift:

Pump failure. Replace the pump.

4. Oil pressure:

The oil pressure loss causes severe mechanical damage on the engine. The drop can happen suddenly or continuously. Over a sudden oil pressure drop, the engine needs to be shut off immediately and is usually followed by alarm sound., The list of the possible failure root causes is as stated by [14]:,

<u>Insufficient level of oil in the sump:</u>

May cause a fluctuation of the oil pressure as the vessel rolls, the pump could lose suction and air enters it. Reduce speed and top up the sump to the correct level.

<u>Lubrication oil pump strainer clogged:</u>

The additives in the oil keep the foreign matter and sludge in suspension for the filter to remove. Signs of a dirty filter would usually be a gradual drop in pressure. If possible, clean the strainer.

<u>Faulty lubrication oil pump:</u>

If the drive to the pump has sheared, there would be no oil pressure at all. The engine must be stopped immediately otherwise severe damage will occur. Should the gears or rotors of the pump be worn or too much clearance between them and the backing plate, there will be a gradual drop in oil pressure. If the lower oil pressure is sufficient, voyage at reduced speed back to port.

<u>Faulty relief valve:</u>

The pressure relief valve may be stuck in the open position or its spring may have broken. A cold engine when started, will have a high oil pressure which will cause the relief valve to open. The engine’s oil pressure drops as the engine reaches its normal operating temperature and the oil thins out. This results in the relief valve closing. Should the relief valve stick in the open position or the spring break, the oil pressure will drop below normal. Free up the sticking relief valve or replace the relief valve spring.

Filter partially blocked:

With the filter being partially blocked, the flow of oil will gradually be restricted. Lower oil pressure will occur and be indicated on the pressure gauge until the filter by-pass valve opens. Replace the filter element or clean the filter (centrifugal type).

Oil temperature too high:

A high oil temperature will thin the oil out causing it to run more easily with a resulting drop in oil pressure. This could be caused by a worn engine which would have fresh water overheating as well. Alternately, it could be caused by a dirty oil cooler on the sea water side. Run the engine at a slower speed until the normal operating oil pressure is obtained and voyage home. Alternately, clean the tubes in the oil cooler.

Faulty oil pressure gauge:

A faulty oil pressure gauge could indicate a low oil pressure where in fact the actual pressure is correct. If the oil pressure gauge is suspected, try another one.

Fracture lubrication oil pipes:

Will result in a gradual or sudden drop in pressure if the pipe splits. If it is possible, repair the leak. If the lower oil pressure is sufficient, steam at reduced speed back to port.

Water in the oil:

Water mixing with oil will result in emulsified oil. It is grey/white or sometimes described as milky in color. Emulsified oil loses its lubricating properties. When a certain amount of emulsification takes place, the oil pressure will drop below normal. Stop the water leak and change the lubricating oil.

Fuel in the oil:

Fuel contamination will thin out the oil and it will run easily off the dip stick. There will be a rise in the level in the sump. The dip stick will also have a fuel smell. Fuel contaminated oil loses its lubricating properties and the oil pressure will drop below normal. Stop the fuel leak and change the lubricating oil.

Regular maintenance is of high importance to keeping the engine healthy. It depends on the engine manufacturer suggestions such as engine hours, fixed dated due to fluid aging characteristics, oil grades, cumulated time on specific load, etc. The routine checks of engine performance parameters and states are done on daily basis, some on monthly, etc. and are logged into the logbook.
The intervals are engine and condition dependent and what follows is an average procedure recommended by [14] for a small medium and high-speed diesel engine.

Daily checks:
Check lube oil level in engine and gearbox, water levels, fuel level, drain water trap (or tank), visual check of all V belts for wear and tension, engine and hoses for water/oil leeks, listen for knocks or rattles. check gauges, grease any grease nipples on pumps, check battery
Every 50 Hours:
Clean fuel filters and remove condensation.
Every 250 hours:
Replace fuel filter, change lube and gearbox oil, replace lube oil filter, clean thermostat, check and adjust governor linkage, check and adjust valve clearance, re-tighten major bolts, clean air filters and replace if necessary
Every 500 hours:
Replace zincs in engine, check rubber impellors, check fuel pump, check injectors (spray pattern, injection pressure, clean)
Every Year:
Valve grind, check rocker arms and valve guides, overhaul, clean and check piston rings, check Con rod (bearings, bolts and torque), check crank arm deflection
Every two years:
Main bearings, crank shaft, clutch/gearbox

5 ENGINE DIAGNOSTIC METHODS

The influence of engine condition change can be reduced with regular maintenance, thus various condition monitoring systems are introduced. There are systems provided by engine manufacturers and also 3rd party systems. They can all be divided in a few system types:

- Measurement data trend monitoring
- Vibroacoustic analysis
- Endoscopic methods
- Exhaust gas composition analysis
- Data driven statistical methods and neural networks
- Model based diagnostic

The various types of fault detection methods are reviewed by [37] and the newest approaches using machine learning, artificial intelligence, fuzzy logic, etc. methods are demonstrated in detail by [38].

5.1 Measurement dana trend analysis

Most of the tools on the market are based on the engine sensor parameters monitoring. The parameters are usually obtained from the data network in the engine room or are available on the standardized diagnostic data ports which are usually CAN based. From the history of the parameter logging, it is possible to determine the parameter trend and gradient. Based on the determined trends, the experience based diagnostic warnings are implemented into the applied tools. The warnings are also based on predefined sensor warning or alarm threshold values. One of the segment representatives is MAN B&Ws CoCoS-EDS [15]. It is engine diagnostic system dedicated for diagnostic and monitoring of large marine engines. It is focusing on turbocharging, combustion, injection, airpath, oil and water pressures and temperatures, component temperatures, air and charge air cooler cleanness and exhaust gas boiler state. The system uses standard alarm monitoring values and additional sensors in order to make the analysis more precise. The short term trend analysis the data is stored in 5 minutes intervals and memory depth is two weeks. The long term trend analysis is based on short term daily based values accumulated over maximum 2 years. The data flow diagram is shown on figure 16. The data is logged every second. The data is processed in a form of the raw data. The raw data is then being filtered by calculating relevant information for further use. The data is than being grouped to the specific data branch. It is possible to add data manually if the measurement source was external. The data flow consists of a nominal values which serve as the reference for further diagnostic tasks. The reference is obtained by healthy engine measurements and stored in that way. The system monitors the current operating values and can display them in forms of charts, curves, etc. Based on the monitored values diagnostic is performed. The diagnose is mainly based on the detection of the wear or the failure detected by temperature differences and its rise, pressure drops, indication pressure traces change and its comparison to the nominal data. It also suggests further actions in order to eliminate the symptoms or the failure.

There are other data trend monitoring system under which the The Data Trend by Norcontrol [39], CC-10 by B&W [40], Comos and DMTAS by Mitsubishi [41]. There are others on the market as well for medium speed 4 stroke as well as large slow speed engines. The research of the market holders is shown in the Appendix 1 containing not only engine diagnostic but also engine optimization as well as trim position and route optimization which contributes to fuel savings and engine durability.

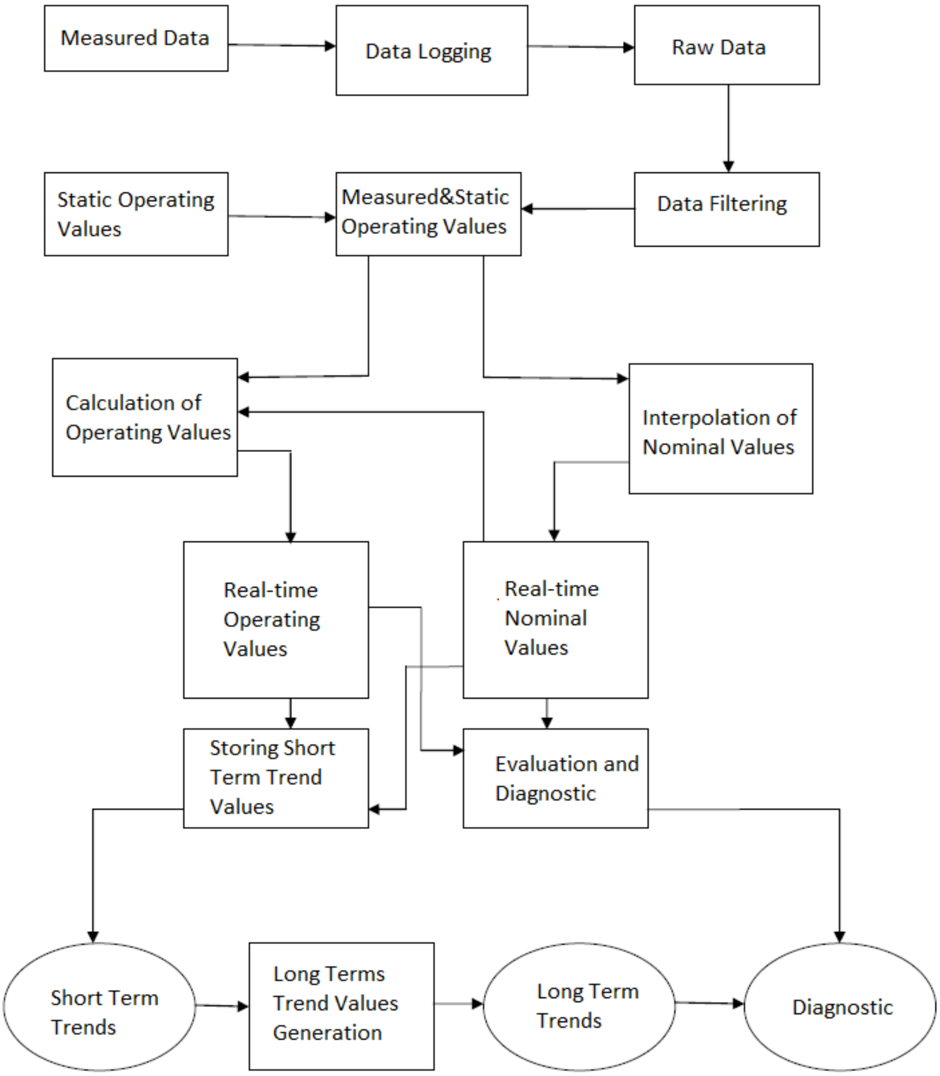

Figure 16 Data Flow CoCoS [16]

The benefit of this kind of a system is constant monitoring and having insight into the historic engine behavior. The downsize is that the only failure predictiveness is based on the trend monitoring which requires a lot of experience from user in order to make reasonable conclusions. Another downsize is that in order to detect or monitor component health, sensors need to be installed in or around it. There is weak failure symptom correlation for detection root causes on the location further than monitored ones.

5.2 Vibroacoustic analysis

This method is based on the measurement of the vibrations of the engine. Since every engine component vibrates, the biggest challenge is to isolate the component contribution during the spectral analysis. The change in the frequency spectrum is being monitored and by knowing the source of vibration different kind of anomalies can be detected. The most common are the thermal processes, combustion roughness caused by injection system failures, component wearing and cumulation of the composition on the specific friction relevant components. The method operates in real-time and uses Fast-Fourier analysis or digital filtering for the vibration signal processing.

The acoustic method is also present for the detection of the wearing, corrosion and other types of settlement or damage on the material. The acoustic wave is sent throughout the material which has a specific amplitude. The amplitude of the vibration decreases with the time and distance and is described by its envelope shape. When the envelope shape changes, it indicated the abnormalities in the material.

The benefit of this method is usage of a single or minor number of measurement locations to obtain the vibration signals for further processing. From this signal it is possible to diagnose the failures on locations other than the measurement points. The downsize of the method is complexity of component vibration detection in a spectrum which makes the method sometimes not reliable and there is no predictiveness of the failure detection.

5.3 Endoscopic methods

It is visual inspection of the engine components by using endoscopes. It is used for tight areas such as combustion chamber or turbocharger. In combustion chamber it is possible to inspect piston, liner, cylinder head, exhaust valves, etc. the endoscopy can be performed in 3 dimensions with single lens which gives us picture in 3 dimensions.

The benefit of this method is reliable detection of the damages on the material which online systems are not able to detect. The downsize is that it requires engine to be in shut down in order to perform the measurements. There is also no predictiveness of the failure into this approach.

5.4 Exhaust gas composition analysis

Exhaust gas composition analysis is performed using portable emission device. Ideally the device should be placed as close to cylinder exhaust valves, normally on the pipe on the exhaust collector. This method measures exhaust emission components and by comparing it to engine shop data or latest healthy engine state, it is possible to bring the conclusion on the injector state, injection timing, wearing, etc. For example, higher NOx emissions mean higher in-cylinder peak temperatures which is probably cause by premature injection. CO emissions are corelated to combustion efficiency, HC to injector leakage, etc.

The benefit of the system is that it requires a single point for the measurement and is able to provide the data from which more detailed conclusion on combustion process can be made. The downsize of the method is that it hard or impossible to determine any root cause of the failure outside of the combustion chamber in a direct way. Another downsize is that has not failure predictiveness capabilities.

5.5 Data driven statistical methods and neural networks

Since modern ships stream the data over the standardized protocols direction engine room through ship communication network, the data became the valuable source for diagnosing the engine condition. When dealing with large amount of historic data, by using statistical methods it is possible to predict the probabilities of the specific component failures. The prediction quality depends mainly on the historic data amount, data quality and encountering similar events during the data collection period.

Except applying statistical methods on data, the artificial neural networks can be applied as well. The example is shown on the fishing vessel [17].

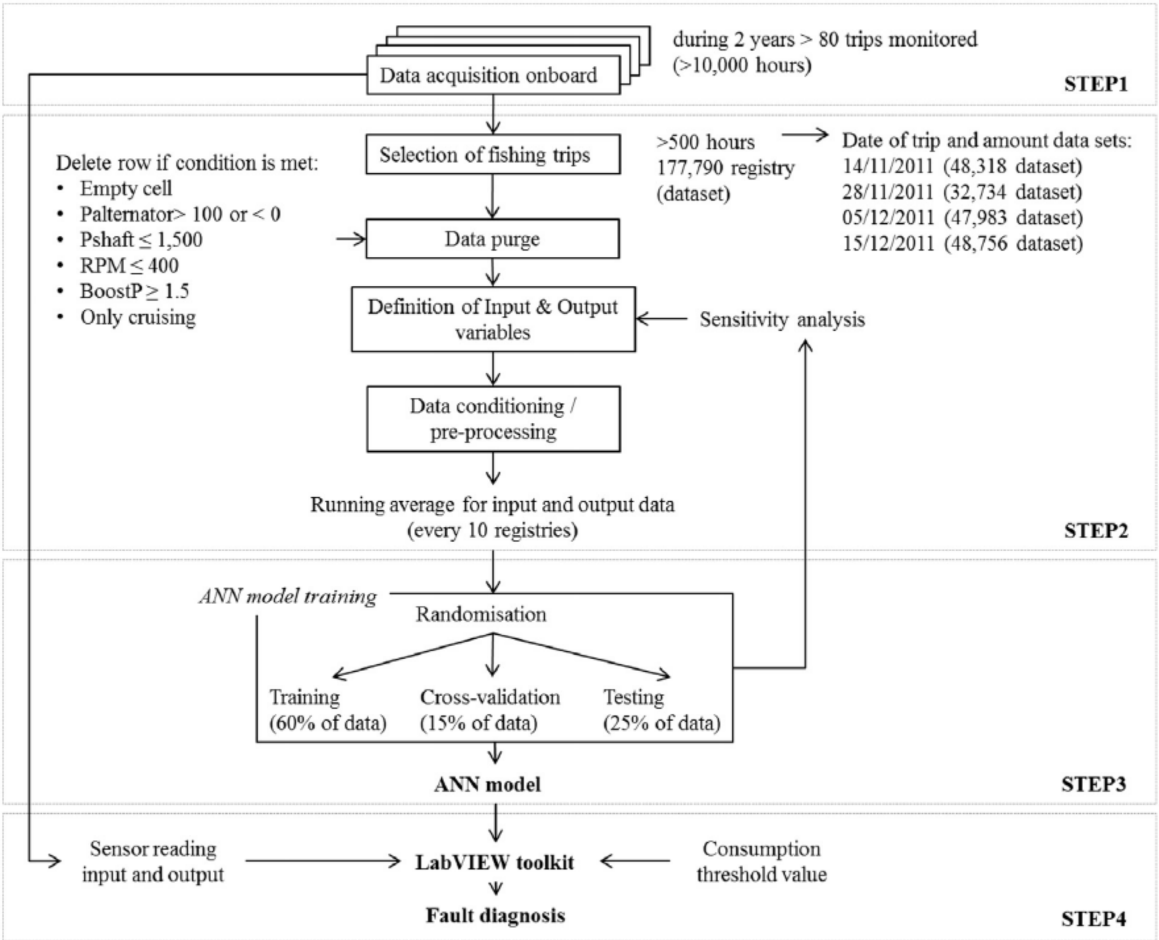

Figure 17 Application of Artificial Neural Networks for marine engine fault diagnosis [17]

The process on the figure 17 is described in the 4 steps. First step is dana acquisition, step 2 is dana selection, purging and conditioning, step 3 of the engine model and finally step 4 is fault diagnosis. In the data acquisition phase the combustion, turbocharger, fuel condition and the electrical consumption parameters were measured I time duration of 2 years with more than 10000 engine operating hours. In the second step the data was purged or filtered by removing implausibility as well as for modelling nonrelevant engine operation. After purging the data, all relevant input and output variables are defined and finally data is being processed by calculation some engine relevant parameters and averaging desired measured or calculated data.

After data preparation, sensitivity analysis is performed in order to determine the weighting factors for each variable. In step 3, three-layer feed-forward network is trained by data obtained during the measurement phase. The final phase, fault diagnosis, is based on dirty turbine, air filter, charge air cooler and bad fuel injection detection. The system frame is finally divided into sensors, neural network engine model and fault detection as in figure 18. Fault is detected when fuel consumption deviates more than 3% from the one obtained by the measurement.

Failure cause is detected by comparing simulated data from the one obtained by the measurement and if simulation deviates more than predefined threshold with reference to the measurement, failure root cause is detected.

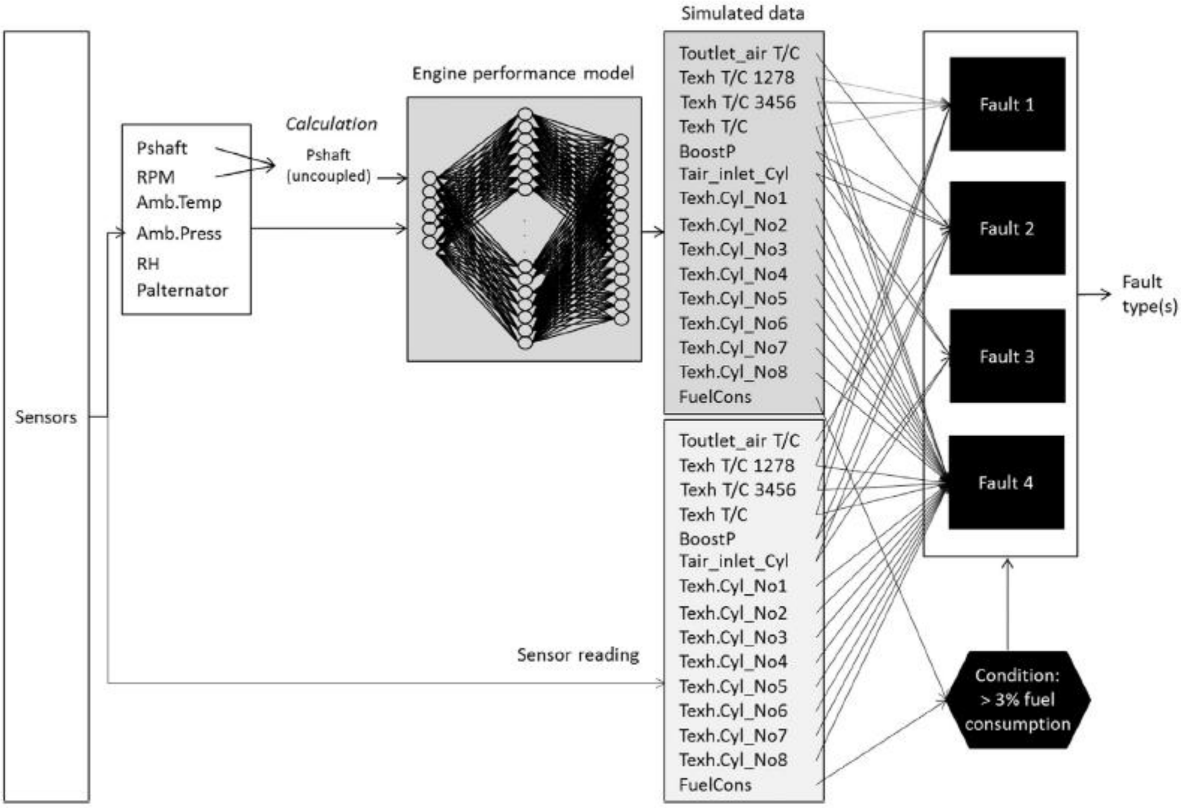

Figure 18 System overview of using neural networks for marine engine failure detection [17]

The benefit of this system is that it runs online and evaluates the engine states constantly. The downsize is the long measurement period for getting the data for the model buildup, model extrapolation capabilities are minor which means low predictability of the failures which happen out of the scenario range which was obtained during the measurement campaign.

5.6 Model based diagnostic

Model based methods use models as the representation of the physical component which is being observed. The simulation results are compared with the physical unit of interest and from its observations, the faults can be detected. The fault is the abnormal behavior of the component or the system which does not meet function specification and can lead to a failure [42]. By comparing the simulation to the measurement values the residuals are being calculated. Having the residual near zero point to no fault and if different than zero, indicates the component or system fault. There are different ways of handling the residuals. One of the is null hypothesis. By observing the residuals, if null hypothesis conditions are holding, there is no fault happening. The null hypothesis conditions or rules are defined by probability theory. The threshold is defined and when the residuals are withing there is no fault, when the threshold is exceeded, the fault is detected [43]. The most popular hypothesis used in diagnostics are Sequential Probability Ratio Test [44], Cumulative Sum algorithm [45] and General Likelihood Ratio Test [43]. The residuals estimation can also be conducted by using the technique called parameter estimation [46, 47]. The system is modeled using mathematical equations and in the equations relevant parameters are defined such as, blow-by coefficient, turbocharger efficiency, etc. the parameters are initially calibrated and during the evaluation phase re-adjusted to reproduce the current behavior of the component or the system. The parameters can be estimated using optimization techniques such as Kalman filter [48, 49], weighted least squares [50] and others. If the parameter differs from the nominally calibrated value, the fault is being detected and points to the location of the possible failure. This method was applied in the marine engine model based diagnostic approach by [19] and is widely used in gas turbine industry [51-56].

As the lack of predictability in the previous methods presented in chapter 5 shows difficulties when diagnosing the engine fault and specially over the lack of capability to consider engine retrofitting where the actual engine data is compared to the one obtained in the past during the measurement campaign, model based diagnostic is introduced. The central part of the diagnostic is the thermodynamic physical based engine model [18]. The engine model and its auxiliaries are calibrated on the engine shop or sea trial data. The modeled parameters have focus on in-cylinder combustion, exhaust temperatures and the turbocharger. The simulation model shows the reference engine behavior under the current conditions and based on the modifications needed to match the status of the currently running online measurements, the reasonable conclusions for the fault diagnostic can be done. The modifications are done in the engine model constants.

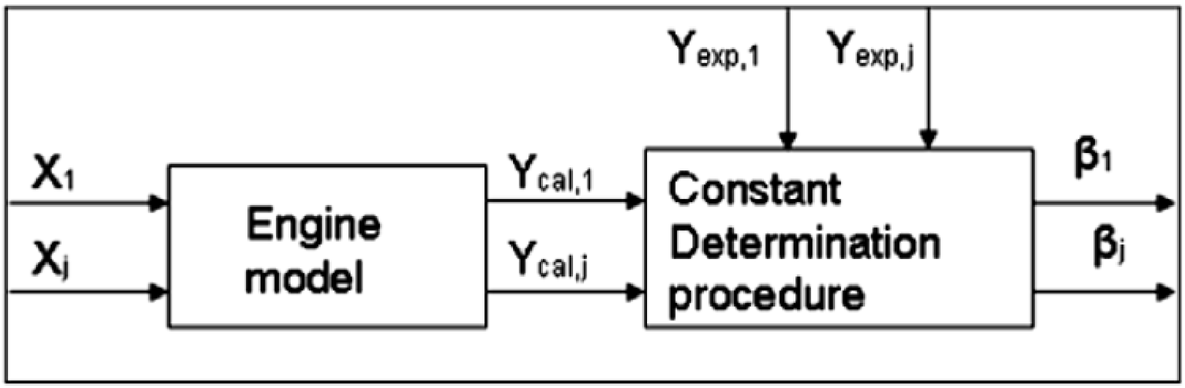

Figure 19 Constant determination philosophy [19]

In the Figure 19 x_j are the model inputs, $y_{cal,j}$ are the calculated data, $y_{exp,j}$ are the measurement data and β_j are the determined constants [19]. The constants are determined by minimization of the difference between the measured and calculated data by using least squares method.

The fault detection is established based on the equation:

$$\left|\frac{\beta-\beta_0}{\beta_0}\right| \cdot 100 \geq 3\% \qquad (1)$$

Where β is the actual engine constant and β_0 is the reference constant from the engine shop data. The in-cylinder constants evaluate compression pressure as a detection of the liner wearing, expansion pressure for the ignition delay, peak firing pressure for the injection dispersity, fuel rack for the pump state, temperatures and pressures around the turbocharger for the dirtiness or damages.

The benefit of this method is that is adaptable to the new engine states or fuel qualities, it can be applied online during engine runtime or in the postprocessing phase, requires the single time load sweep measurements only for the reference constants. The downsize of this method is that for setting it up it requires a lot of thermodynamic and modeling know-how and the constant deviation threshold of 3% is more experience based and can in reality differ between the constants.

6 MARINE ENGINE DIGITAL TWIN

From the diagnostic methods stated above it can be concluded that optimal diagnostic method should be non-intrusive, should operate online during engine running hours constantly, should be easily adaptable (after engine retrofit, different fuel characteristics, et.), require as less possible sensor information (ideally no additional sensors as engine already has installed by engine manufacturer) and needs to be predictive. The only method which can comply with all requirements stated above is the one based on the advanced digital twin. The digital twin must combine a few technologies used in nowadays diagnostic system such as data trending, advanced physical models including combustion, emissions, turbocharging and fuel path with easily adjustable calibration coefficients who are adapted by current sail condition or engine state by using machine learning methods.

The market is recognizing the importance of the digital twins in the marine applications. As stated by classification society DNV GL [20] the digital twin is an asset that can create the value for the all stakeholders in the marine industry. For the shipowners who are using more and more complex propulsion systems that coexist with other onboard power generation and consumer systems as a part of the onboard energy distribution, digital twins can serve to test, verify and certify such systems. It gives them the insights on operational data, possibility of optimization of the performance and reduction of operational costs especially when applying on the overall fleet. One of the most significant use cases is the possibility for the fault diagnostic or maintenance predictions as well as post-retrofit benchmark. The benefit when dealing with operational analysis is that the digital twins can simulate the historic operations, real-time as well as predict the system behavior for the future scenarios. The Original Equipment Manufacturers can gain by integrating the simulation models during the design phase ensuring that components perform properly as a part of the larger system. It also opens the possibility for the monitoring and predicting the component maintenance intervals as a part of condition based maintenance approach during the component lifetime. The classification societies have also noticed the high need for this technology in the future in terms of secure reporting followed by monitoring, digital twins applications in terms of condition based maintenance, vessel operation optimization, fuel and emission predictions. Lloyd's Register even introduced Digital Compliance framework under which the digital twins can be tested, verified and certified [21]. The application and development of digital twins need to go through quite a few challenges. The shipowners and other shareholders will need to get used to the new tools or toolchains, rise of the new business models which derive from the application of digital twins, standardization of the naming conventions, data quality, model connection interfaces, etc. One of the major challenges is the competence to build and calibrate predictable and robust simulation model. The simulation model or the digital twin of the specific physical asset can be ship hydrodynamic model, propeller model, ballast water treatment model, etc. As the machinery is the most critical and expensive part to fail on the ship, the thesis will be focused on the digital twin for the diesel propulsion engine applied on 2 stroke large engines and 4 stroke medium speed engines.

In order for model based diagnostic method to avoid triggering false fault detections, it requires the model to be robust and predictable. One of the most advanced system simulation tools, AVL Cruise M was chosen as the core platform for digital twin implementation, calibration and testing. Except advanced physical models, multiple degrees of freedom on model calibration process, it is 0 dimensional based simulation tool which is able to deliver results in real-time. Real-time simulation is one of the main model requirements for online diagnostic operations in closed-loop with the real engine and user.

6.1 Cylinder Model

Cylinder model is built from number of sub-models such as Port In –representing inlet port opening surface and flow characteristics, Direct Injector – representing rail and injector model, Port Out –representing exhaust valve opening surface with variable opening and closing angles, the Combustion Chamber contains geometrical data and combustion models and finally SW (solid wall) elements representing the temperatures of the components in the cylinder.

5.

For the prediction of the combustion, AVL´s Multi-Zone Combustion model is evaluated. As described in Cruise M Manual [22] the model divides the spray into a high number of zones, Fig 20. The benefits of using the multizone model towards more conventional approaches is described by [23].The sub-models of charge entrainment, evaporation, ignition delay and combustion are calculated for each zone. Each injection package is divided into the number of radial zones. The total number of zones depends on duration of injection, calculation time-step and discretization in the radial direction.

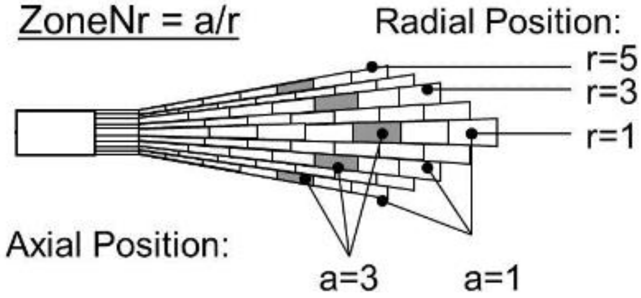

Figure 20 Spray Discretization in Multi-Zone Combustion Model [22]

Entrainment:

The mass of the zone at certain time after the injection of the package is:

$$m_i = m_{inj} \cdot \left[1 + \left(\frac{v_{inj}}{v_{tip}} - 1\right) \cdot C_{Entrain}\right] \qquad (2)$$

where m_i is zone mass at t, m_{inj} is injected package fuel mass, v_{inj} is velocity of injection, v_{tip} is tip penetration velocity and $C_{Entrain}$ model parameter.

Decrease of v_{tip} relatively to v_{inj} is resulting in the increase of mass in the zone and this can be tuned by $C_{Entrain}$ [5]. Velocity of injection comes from:

$$v_{inj} = c_D \sqrt{\frac{2 \cdot \Delta p}{\rho_{fuel}}} \qquad (3)$$

where c_D is injector nozzle discharge coefficient, Δp is injection pressure drop and ρ_{fuel} is liquid density of the fuel.

The tip penetration velocity is calculated by:

$$v_{tip,c} = \frac{2{,}95}{2} \cdot \left(\frac{\Delta p}{\rho_{ch}}\right)^{0{,}25} \cdot d_{inj}^{0{,}5} \cdot t^{-0{,}5} \cdot f_{rad} \cdot f_{ax} \qquad (4)$$

where ρ_{ch} is charge density, d_{inj} is injector nozzle diameter, t is time after injection of package, f_{rad} is correction functions to account for radial positon in the spray and f_{ax} is correction functions to account for axial positon in the spray.

Since the spray velocities at the borders of the spray are lower than in the centerline the f_{rad} correction function was introduced:

$$f_{rad} = e^{\left[C_{rad} \cdot (i_{rad}-1)^2\right]} \quad (5)$$

where i_{rad} is radial position of the package in the spray and C_{rad} is model parameter.

The influence of current injected package to the later injections due to the slipstream effect is described by:

$$f_{ax} = C_1 \cdot \left[1 + \left(\frac{i_{ax}-1}{i_{ax,max}-1}\right)^{C_2} \cdot \frac{\Delta t_{inj}}{C_3}\right] \quad (6)$$

where C_1, C_2, C_3 are model parameters, i_{ax} is index of axial zones, $i_{ax,max}$ is number of all axial zones and Δt_{inj} is number of radial zones.

In AVL´s Multi-Zone approach, heat and mass exchange between the reaction zones is taken into account. The burned-gas zone was introduced which absorbs all burned zones but also recycles part of them together with the fresh air back into the reaction zone, Fig. 21. The amount of re-entrainment gas into the fresh zone is calibrated by model parameter [23].

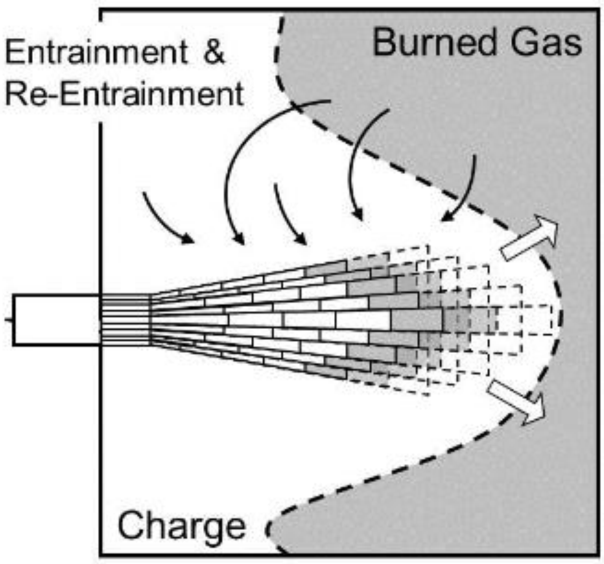

Figure 21 Re-entrainment of burned gas [22]

The heat exchange between the zones is calculated by:

$$\frac{dq_{ZtoCh}}{dt} = \alpha_{ht} \cdot A_i \cdot (T_i - T_{ch}) \qquad (7)$$

where α_{ht} is heat transfer coefficient, A_i is zone surface calculated from the zone volume with assumption of its spherical shape, T_i is temperature of zone and T_{ch} is charge temperature.

<u>Evaporation:</u>

The evaporation mechanism is calculated in 2 areas of droplet temperature separated by critical temperature [24]. The evaporation rate below critical temperature is calculated by:

$$\left.\frac{dm_{fv}}{dt}\right|_{T_l<T_{cr}} = C_{Evap} \cdot \pi \cdot SMD \cdot D_v \cdot Sh \cdot \frac{p}{R \cdot T_m} \cdot ln\left(\frac{p}{p-p_s}\right) \qquad (8)$$

and above critical temperature:

$$\left.\frac{dm_{fv}}{dt}\right|_{T_l>T_{cr}} = C_{Evap} \cdot C_1 \cdot \pi \cdot SMD \cdot \left(\frac{p}{6}\right)^{C_2} \qquad (9)$$

where SMD is Sauter mean diameter, Sh is Sherwood's number, p_s is saturation pressure, p is cylinder pressure and C_{Evap}, C_1 and C_2 are model parameters.

Heat transfer to droplet is calculated by:

$$q = C_{eth} \cdot \pi \cdot SMD \cdot \lambda_m \cdot (T_i - T_l) \cdot Nu \cdot \left(\frac{z}{e^z - 1}\right) \qquad (10)$$

where λ_m is thermal conductivity, T_l is temperature of liquid, Nu is Nusselt number, z is fuel parameter and C_{eth} is model parameter.

The Sauter´s mean diameter at breakup is calculated by:

$$SMD_{tb} = \frac{12{,}392 \cdot d_{inj}^{0{,}44} \cdot \rho_{fuel}^{0{,}42} \cdot \left(\sigma_{fuel} \cdot v_{fuel}\right)^{0{,}28}}{\Delta p^{0{,}42} \cdot \rho_{ch}^{0{,}28}} \qquad (11)$$

where σ_{fuel} is surface tension of the fuel, v_{fuel} is fuel flow velocity and ρ_{ch} is charge density.

For model calibration 2 main parameters are used, C_{eth} to control the heat up of the droplet and C_{Evap} for controlling the evaporation rate.

<u>Ignition delay:</u>

The auto ignition starts when progress variable reaches the value of 1.

$$\int_{inj}^{inj+\bar{\tau}} \frac{1}{\tau} \cdot dt \geq 1 \tag{12}$$

where τ is characteristic ignition delay time and it is calculated by:

$$\tau = C_{IgnDel} \cdot p^{-1,02} \cdot e^{\frac{C_{IgnExp}}{T_{IgnDel}}} \tag{13}$$

where C_{IgnDel} is ignition delay parameter, C_{IgnExp} is ignition delay exponent and T_{IgnDel} is temperature for ignition delay calculation and it represents mean spray temperature.

<u>Combustion:</u>

The fuel burning rate is defined by the reaction kinetic approach by Jung [25]. The concentrations are set by the mass fraction of fuel vapor and air.

$$\frac{dx_{fb}}{dt} = K_b \cdot \rho_{ch}^{a_p} \cdot x_{fv} \cdot x_{O_2}^{b_{O_2}} \cdot e^{\frac{-c_{arrh}}{T_i}} \tag{14}$$

where $\frac{dx_{fb}}{dt}$ is the fuel burning rate, K_b is chemical reaction parameter, x_{fv} is fuel vapor mass concentration, x_{O_2} is oxygen mass concentration and a_p, b_{O_2}, c_{arrh} are model parameters.

NO_X formation is based on [26] with the reactions based on Zeldovich mechanism and Soot formation is based on [27].

6.2 Injection model

The injection model is made of rail and injector model. They are defined by geometry, injection signals and profiles. The injection process is crank angle resolved.

The rail model is a 0D model and considers the compressibility of fluid and is calculated by:

$$\frac{dp_{rail}}{dt} = \frac{E}{V_{rail}} \cdot \frac{1}{\rho_{rail}} \cdot \left(\frac{dm_{pump}}{dt} + \frac{dm_{inj}}{dt}\right) \tag{15}$$

where E is the bulk modulus for the working fluid, V_{rail} is rail volume, ρ_{rail} is density of the fluid inside rail, $\frac{dm_{pump}}{dt}$ is mass flow through pump and $\frac{dm_{inj}}{dt}$ is mass flow through injector.

Approach used for injector rate determination is based on injection velocity and nozzle area. The flow coefficients are representing the friction and in the model they are introduced over the hole and needle seat area, Fig. 22.

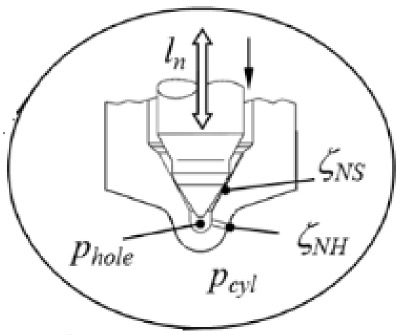

Figure 22 Injector hole and needle seat flow coefficients [22]

$$\frac{dm_{inj}}{dt} = \sqrt{2 \cdot \rho_{fuel} \cdot \left(p_{pipe} - p_{cyl}\right) \cdot \frac{1}{\left(\frac{\zeta_{NS}}{A_{NS}^2} + \frac{\zeta_{NH}}{A_{NH}^2}\right)}} \qquad (16)$$

where $\frac{dm_{inj}}{dt}$ is injected fuel mass flow, ρ_{fuel} is density of the fuel, p_{pipe} is pressure inside pipe, p_{cyl} is pressure inside cylinder, ζ_{NS} is needle seat flow coefficient, ζ_{NH} is nozzle hole flow coefficient, A_{NS} is area of needle seat and A_{NH} is area of the nozzle hole.

6.3 Cylinder wall temperature model

Cylinder wall temperature models is built from heat transfer between gas in the cylinder and solid wall and from solid wall to the coolant. The heat transfer is calculated by:

$$\dot{Q} = \dot{m} \cdot c_p \cdot (T_{inlet} - T_w) \cdot \left[1 - exp\left(\frac{-A_{trans} \cdot \alpha \cdot F_{HT} \cdot F_{target}}{\dot{m} \cdot c_p}\right)\right] \qquad (17)$$

where A_{trans} is heat transfer surface, T_{inlet} is inlet fluid temperature, T_w is wall temperature, α is heat transfer coefficient, c_p is specific heat capacity, F_{HT} is heat transfer multiplier, F_{target} is factor used to evaluate target pressure drop or target efficiency and $\dot{m}$ is mass flow rate.

The temperature in the wall is built up from relation:

$$m \cdot c_p(T) \cdot \frac{dT_s}{dt} = \sum \dot{H}_i \qquad (18)$$

where m is solid wall mass, $c_p(T)$ is temperature dependent specific heat, T_s is temperature of the solid, t is time and $\dot{H}_i$ are heat fluxes.

The port, exhaust valve, piston, cylinder head and liner temperature models are built with the same relations but different geometry characteristics.

6.4 Turbocharger model

The compressor power is calculated from enthalpy difference and mass flow defined by turbocharger speed and pressure ratio map.

$$P_C = \dot{m}_c \cdot (h_2 - h_1) \tag{19}$$

where P_C is compressor power consumption, $\dot{m}_c$ is mass flow rate in the compressor, h_2 is enthalpy at the outlet of the compressor and h_1 is enthalpy at the inlet of the compressor.

The enthalpy difference is given by:

$$h_2 - h_1 = \frac{1}{\eta_{s,C}} \cdot c_p \cdot T_1 \cdot \left[\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\kappa-1}{\kappa}} - 1\right] \tag{20}$$

where κ is ratio of the heat capacity at a constant pressure to the heat capacity at constant volume, $\eta_{s,C}$ is isentropic efficiency of the compressor, c_p is mean value of the specific heat at constant pressure between the compressor inlet and outlet, T_1 is compressor inlet temperature and $\frac{p_{02}}{p_{01}}$ is total to total compressor pressure ratio.

The same as for compressor the turbine power is calculated by:

$$P_t = \dot{m}_t \cdot (h_{03} - h_{04}) \tag{21}$$

where P_t is turbine power, $\dot{m}_t$ is mass flow rate in the turbine, h_{03} is total enthalpy at the turbine inlet and h_{04} is total enthalpy at the turbine outlet.

The enthalpy difference is:

$$h_{03} - h_{04} = \eta_{s,t} \cdot c_p \cdot T_{03} \cdot \left[1 - \left(\frac{p_4}{p_{03}}\right)^{\frac{\kappa-1}{\kappa}}\right] \tag{22}$$

Where $\eta_{s,t}$ is isentropic turbine efficiency, c_p is mean specific heat at constant pressure between turbine inlet and outlet, T_{03} is total turbine inlet temperature, $\frac{p_4}{p_{03}}$ is turbine expansion ratio, total to static and κ is ratio of the heat capacity at a constant pressure to the heat .

The turbocharger speed in stationary and transient behavior is given by:

$$\frac{d\omega_{TC}}{dt} = \frac{1}{I_{TC}} \cdot \frac{P_T - P_C}{\omega_{TC}} \quad (23)$$

where ω_{TC} is turbocharger wheel speed, I_{TC} is turbocharger wheel inertia.

6.5 Charge air cooler model

Pressure drop over the charge air cooler is defined by:

$$\Delta p = F_{fr} \cdot \zeta \cdot \frac{L}{D_{hyd}} \cdot \frac{\rho \cdot \nu^2}{2} \quad (24)$$

where F_{fr} is friction force, ζ is friction factor which a function of the Reynolds number, L is length, D_{hyd} is hydraulic diameter, ρ density and ν kinematic viscosity.

The described digital twin is a generic diesel physical model. The major challenge is the understanding and calibration of the model in order to achieve all the requirements that digital twin is demanding. In the thesis, the digital twin will be calibrated referenced to the engine shop measurement data for widely common 2 stroke large engine as well as 4 stroke medium speed engine. The boundaries of the model will be determined and the automatic model calibration method based on genetic algorithm optimization method will be assessed. The model will afterwards be exercised in order to create sensor data behavior fault pattern matrices by simulating the component failures with the digital twin. The patterns can be therefor used for the automatic failure detection applications. The model can also be used for the optimization purpose [57] as well as for emission monitoring [58].

7 DIGITAL TWIN APPLICATION METHODOLOGY

In recent years the digital technologies have rapidly evolved into the maritime sector. The wider coverage of the GSM and the cheaper and efficient satellite data transfer made the cloud computing technology accessible for the maritime sector. This opens the possibility to interconnect all the stakeholder and offers opportunity for the new business models. The cheaper sensors allowed wide spread of the measurable assets which can store the data locally or send it directly to the cloud towards the data analytics algorithms. The big data technologies coupled with the artificial intelligence offer faster and more precise conclusion extraction from large amount of data as well as training the algorithms for the maintenance, diagnostic and similar. The development of the simulation technologies puts the digital twins in the focus of the digital applications for maritime sector. In large variety of the digital twin applications, a few can be emphasized as a high value creators: maintenance requirement detection, sensors measurement plausibility checks, fleet benchmarking, retrofit evaluation, fuel consumption, emissions, failures predictions and complex system interaction predictions. The digital twin can also be used for operation optimization by simulating optimal machinery settings during the ship voyage.

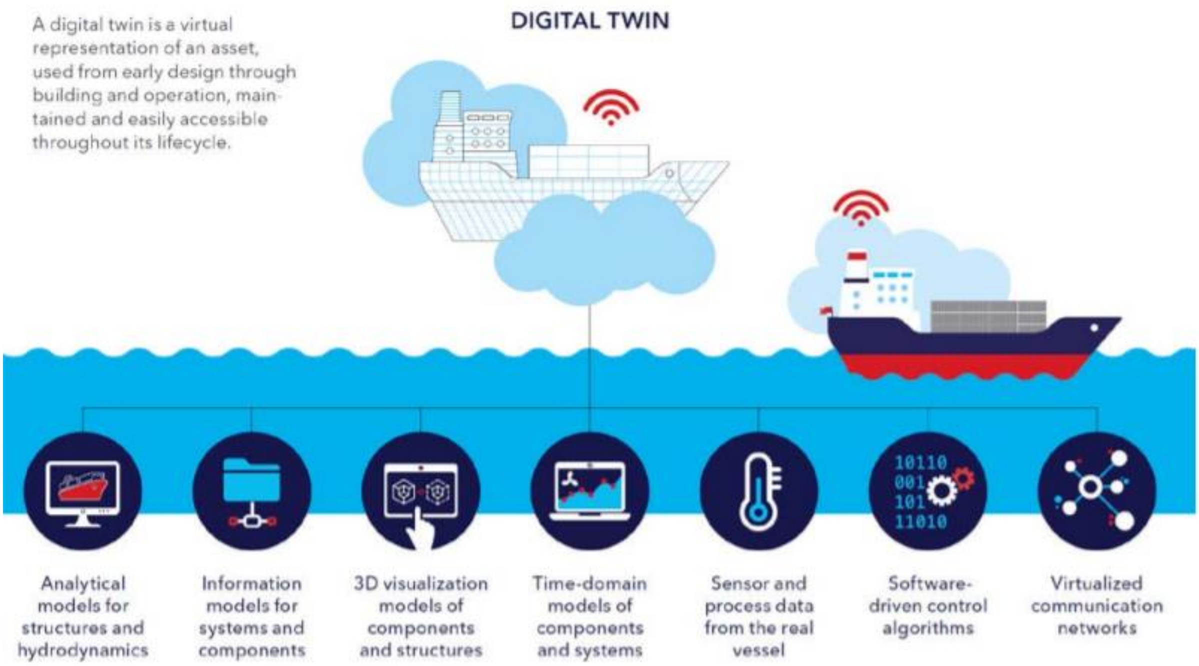

Figure 23 Digital Twin application directions [20]

In the thesis the possible engine digital twin application usecases will be conceptually created and are here introduced.

7.1 Onboard optimization tool

In 2009. Was stated by [28] that voyage optimization and energy management can reduce the CO_2 emissions from 2% to 20%. The need for the onboard optimization tool comes from the fact that the sea conditions cannot be fully reproduced on the engine testbed, the ship is sailing in different areas with different emission regulations so pollutants can be reduced for a specific trip. Finally, the aging and the condition change of engine components are one of the main factors for efficiency decrease thus the original calibration from the testbed may no longer be optimal.

The presented solution would allow the choice of 3 control strategies, Fig.24.:

1. Base engine map - map derived from the testbed. This is the base for any other optimization or the next 2 strategies.
2. Fuel consumption - actuator setpoint values in control unit derive from optimization software based on economical demand with tradeoff for NOx pollutants.
3. Low emission - the optimization and adjustment of optimal actuator values for low NOx and Soot emissions.

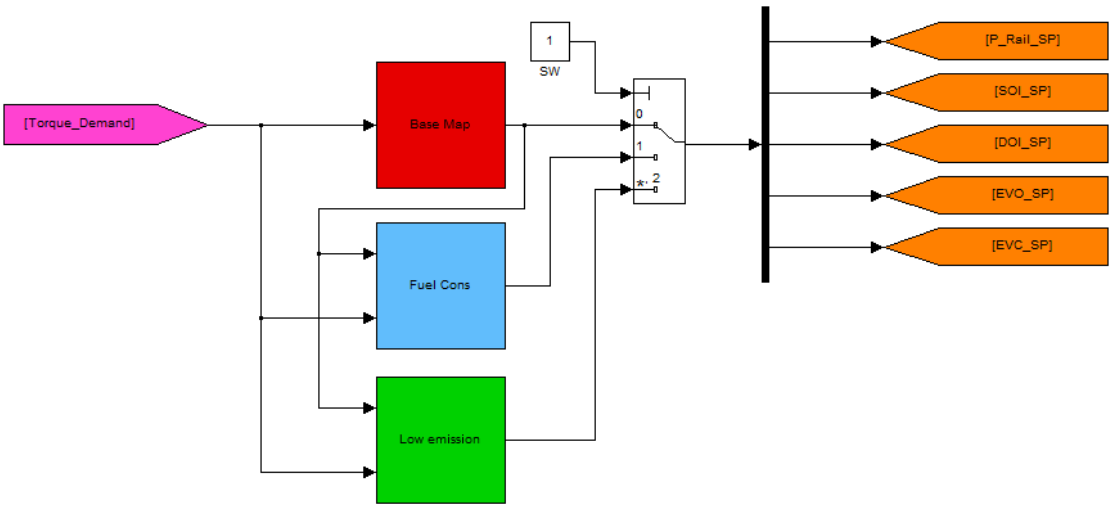

Figure 24 Optimization tool strategy selection

In the Fig. 24 it can be seen that the torque demand is the input into the control unit software and based on user chosen control strategy the optimizer gives the optimal setpoints such as rail pressure, start of injection, duration of injection and for more advanced engines, exhaust valve opening and closing angle. The setpoints are sent to the controllers as a demand value.

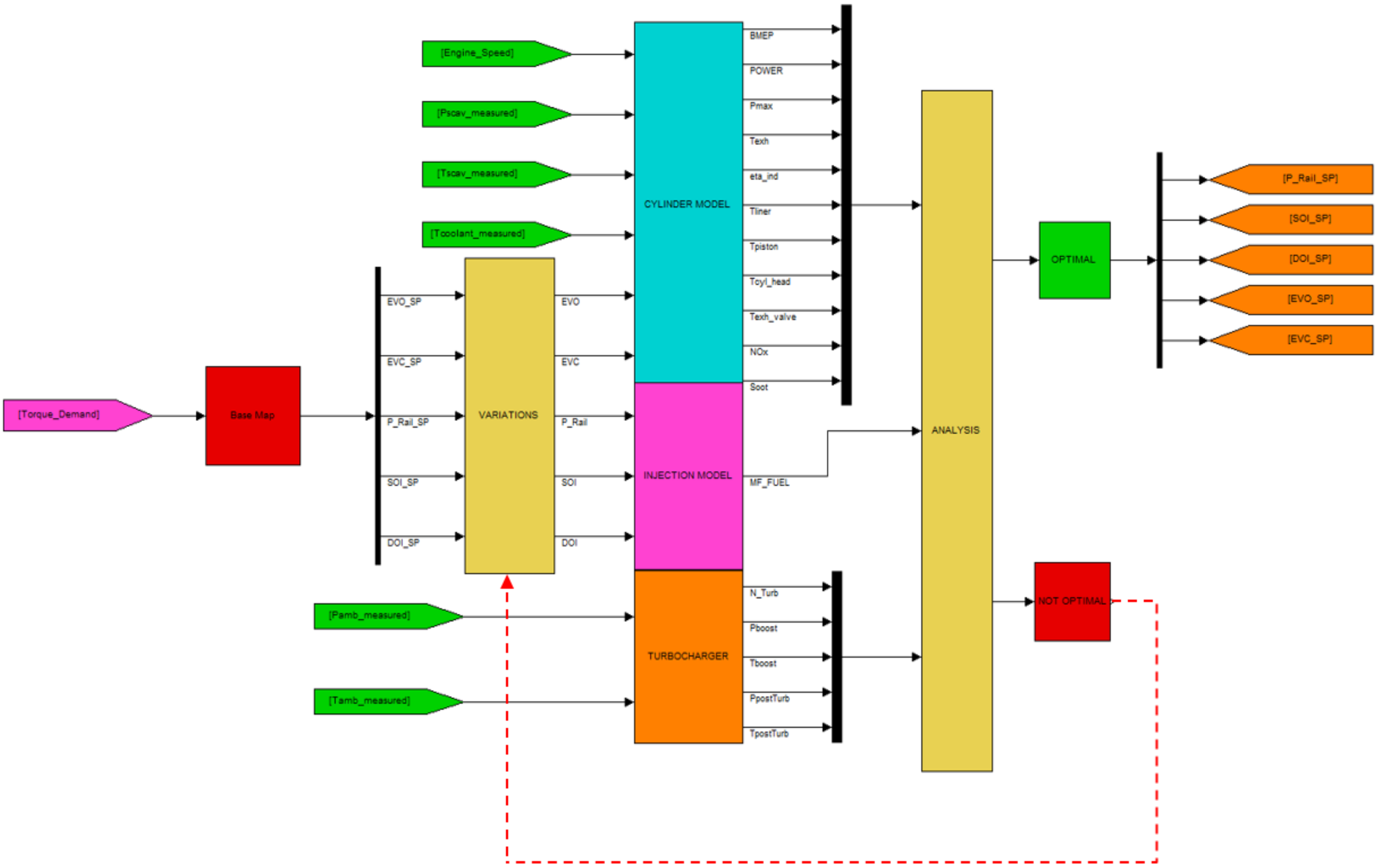

Figure 25 Digital twin engine optimization concept

The optimization tool is based on the described physical model and the optimization structure is presented in Fig. 25. The optimizer starting point is the calibration from the testbed to reduce the optimization calculation time and it is driven by torque demand. The engine actuator setpoints, where EVO_SP is exhaust valve opening setpoint, EVC_SP is exhaust valve closing setpoint, P_Rail_SP is rail pressure setpoint, SOI_SP is start of injection setpoint and DOI_SP is duration of injection setpoint, from the base engine map are sent to variation block where they are being varied and fed to the engine model. It can be seen that except the actuator inputs into the cylinder model, the real-time measured scavenged pressure Pscav_measured and temperature Tscav_measured are also fed to the cylinder. This means in case of dirty charge air cooler or deteriorated turbocharger the optimum setpoints will also be found even for these conditions. To predict the in-cylinder temperatures like liner temperature Tliner, piston face temperature Tpiston, cylinder head temperature Tcyl_head and exhaust valve temperature Texh_valve the measured coolant temperature is fed into the model. Due to the possible engine speed oscillations caused by rough seas, the measured engine speed Engine_Speed is sent as input into the cylinder model. The performance outputs from cylinder model that are used for optimization are BMEP, POWER, in-cylinder peak pressure Pmax, exhaust temperature Texh and indicated efficiency eta_ind. For the emission optimization, NOx, CO, CO_2, HC and Soot prediction is calculated.

The injection model is fed with start of injection, duration of injection and rail pressure and predicts injected fuel flow MF_FUEL. The turbocharger model receives ambient pressure Pamb_measured and temperature Tamb_measured as inputs which gives the possibility to predict the engine behavior in the different environmental conditions. The rest of inputs for (20) and (22) are calculated from the air path volumes. The predictions from turbocharger model are turbocharger speed N_Turb, boost pressure Pboost, boost temperature T_boost, pressure at turbine outlet PpostTurb and temperature at turbine outlet TpostTurb. Air filter pressure drop is as well calculated in the turbocharger component.

After predicting the engine behavior with cylinder, injection and turbocharger models, the results are sent to the analysis block. The analysis block decides if the results are optimal. If yes, it forwards the new actuator setpoints to the controllers which actuate the actuators on the engine and if not, the variations of the inputs in the engine model are being varied until optimum is reached. The optimum depends on which control strategy was chosen, fuel consumption or low emission, Fig. 24.

During the variations the engine runs on the base map or on the last optimal value derived from the onboard optimizer, Fig. 25. This points that that the optimizer doesn´t have to be real-time capable and thus the AVL Multi-Zone Combustion model was used, which runs slightly slower than real-time. If a faster on board optimization and real-time diagnostic is needed, then the faster combustion models need to be applied such as AVL MCC 2014.

7.2 Model Based Diagnostic tool

In the year 2011, it was reported that more than 50% of the fleet is older than 15 years [31]. Engine long term operation causes engine condition change which results in lower efficiency, higher fuel consumption and thus increase in NO_x, CO and CO_2 emissions. The diagnostic system can detect the faults or aging and adapt to the new conditions or prevent the major failures. In the next chapters the application of the engine model on error simulation and a few malfunctions detections is presented.

7.2.1 In-cylinder model based diagnostic concept

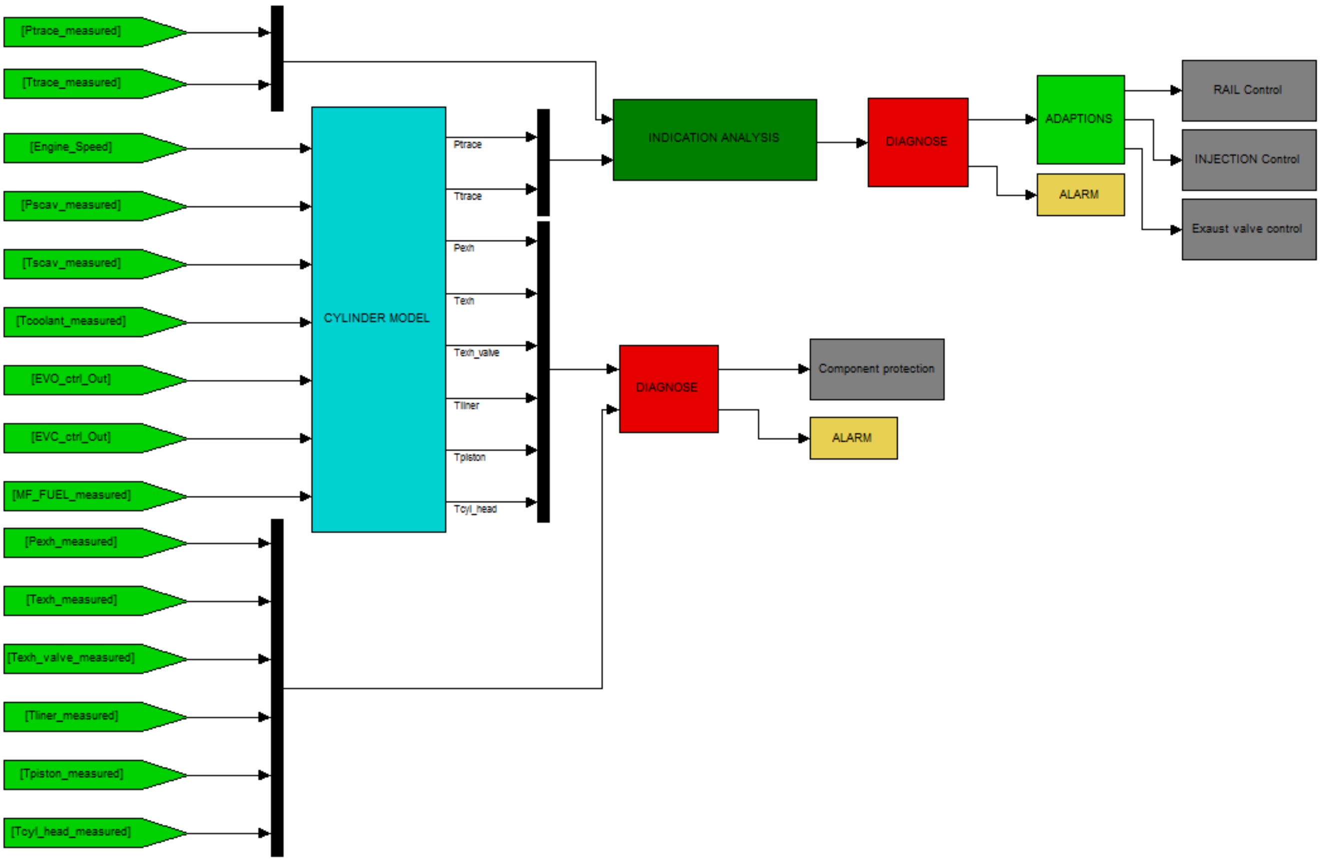

Figure 26 In-cylinder diagnostic concept overview

The base of the in-cylinder diagnose model is the cylinder model shown in chapter 6.1. The measured values from the engine in real-time are fed to the in-cylinder diagnostic tool and some of them are used as model inputs and others for the diagnostics. The measured inputs to the cylinder model are engine speed, scavenged pressure and temperature, coolant temperature, EVO_ctrl_Out is the real exhaust valve opening angle with included opening deadtime, EVC_ctrl_Out is the real exhaust valve closing angle with included opening deadtime and MF_FUEL_measured is the fuel amount measured on fuel metering unit. Based on these inputs the cylinder model gives two types of results. One is pressure and temperature information crank angle based and the other is mean value based. The mean value model outputs, such as exhaust temperature, exhaust pressure, temperature of exhaust valve, liner, piston and cylinder head temperature are compared with measured values and in diagnose block (Fig. 26.) The relative deviation is calculated. If the relative deviation is close to the predefined threshold value the component protection is activated which sends the control unit or the user the request to regulate the injection timing and duration and exhaust valve timing until the relative error falls below the threshold. The component protection is mainly based on in-cylinder components temperatures. If the error overshoots the threshold the alarm or shutdown is activated. The calculated pressure and temperature traces are sent to the indication analysis block (Fig.26.) and compared to the measured ones. In the indication analysis the heat release, start of combustion, combustion efficiency, peak firing pressure, emissions, etc. are being analysed and compared between simulation and measurement. The comparison is then sent to the diagnose block where the relative deviations are calculated and compared to the thresholds. The diagnose block can also contain the database of the pressure traces with simulated malfunctions from the office simulation and compare them to the measured pressure traces and thus lead to the malfunction source or incorporate the artificial intelligence algorithms or matrices for automatic fault detection. When measured pressure trace doesn´t comply with the simulated due to condition change, the adaptions block is activated and adapts the actuators to achieve desired performance or emission outputs. This comes on top of optimizer (Fig.25.) or base map (Fig.24.).

7.2.2 Injector and rail model based diagnostic concept

The injector and rail diagnostic concept has the same diagnostic workflow as the in-cylinder diagnostic concept. The injection and rail model from Fig. 27.

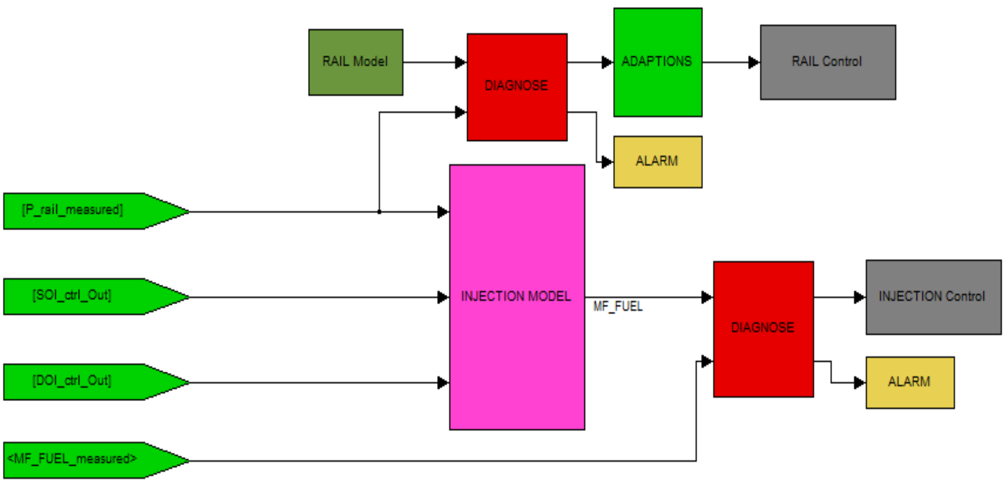

Figure 27 Injector and rail diagnostic concept

The inputs to injection model are P_rail_measured which is measured rail pressure, SOI_ctrl_Out is the measured or demanded by control unit start of injection, DOI_ctrl_Out is duration of injection measured or demanded by control unit and the fuel amount measured on fuel metering unit. The rail pressure submodel as an input, uses the mass flow of the fuel pump, injected mass flow and from it calculates the rail pressure. The calculated rail pressure is then compared to the measured on and then in diagnose block relative deviation is compared to the threshold. If under the same pump and injected mass flows the relative deviation overshoots the threshold value than there is high possibility for leakage. The injection model calculates the fuel and compares it to the measured fuel amount to find the malfunction. In this way it is not fully possible to find the source of the injector malfunction that is why the method is combined with the in-cylinder pressure trace analysis for localisation of the fault.

7.2.3 Turbocharger model based diagnostic concept

The Turbocharger diagnose model the same diagnose workflow as the in-cylinder and injector/rail diagnose model.

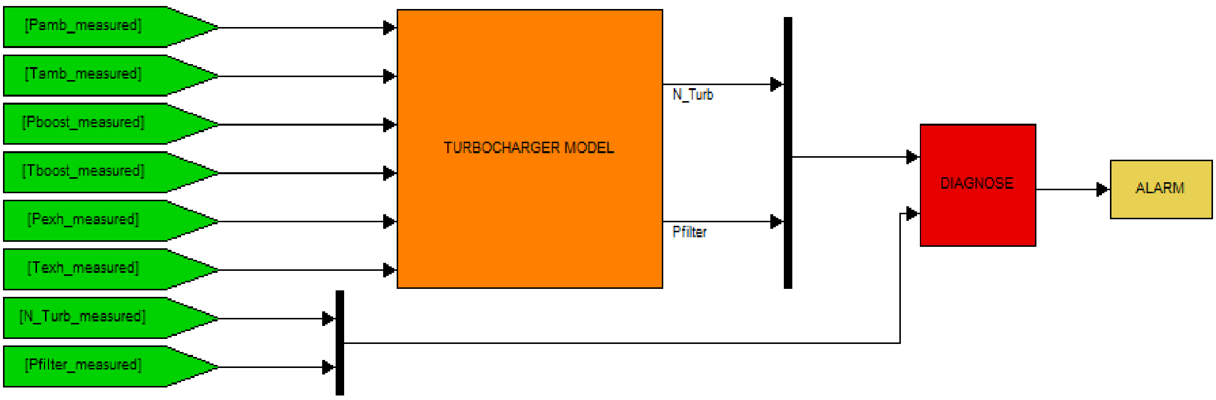

Figure 28 Turbocharger diagnostic concept

The inputs to the turbocharger diagnose model are values measured in real-time such as ambient pressure and temperature, Pboost_measured which is the measured boost pressure, Tboost_measured which is the measured boost temperature, exhaust pressure and temperature, N_Turb_measured which is turbocharger rotational speed and Pfilter_measured is the pressure after the air filter. The model calculates N_Turb turbocharger speed and Pfilter pressure after the air filter and comparing it to the measurement. After calculating relative deviations the turbocharger degradation, plugged or damaged air filter can be detected.

7.2.4 Charge Air Cooler model based diagnostic concept

The Charge Air Cooler diagnostic concept uses the same diagnostic workflow as the in-cylinder and injector/rail diagnose model, Fig. 29.

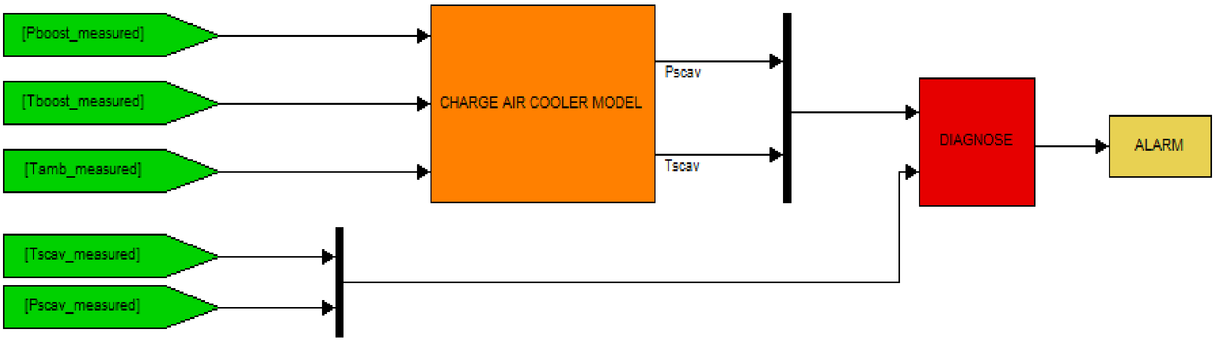

Figure 29 Charge Air Cooler diagnostic concept

The inputs to the charge air cooler model are measured boost pressure, boost and ambient temperature and for diagnose subsystem, measured scavenge pressure and temperature. The model calculates the scavenged pressure based on pressure drop equation and scavenged temperature based on cooler efficiency and comparing it to measurements the dirty or damaged charge air cooler condition can be detected.

The digital twin application can be applied as a standalone tool on board the ship or the cloud and/or coupled with other condition monitoring systems. If it is going to be coupled with another diagnostic tool it must have interface which allows the communication between the parties. The model can be used as well for the diagnostic software development and testing in software in the loop environment. The number of condition monitoring tools are based on artificial neural networks which require a lot of measurement data which demand 1-2 years of measurement onboard the ship. Since the model can be used for simulating malfunctions in the office environment, the model can be used for training the neural networks and significantly reduce time and cost of training them from the data obtained onboard the ship.

8 CONCLUSION

The marine engine is the main and most important segment of the machinery onboard the ship. It's not only observed from the performance, but also from safety perspective. The ship engine is a single highest initial investment and lifetime long maintenance cost item. Depending on the applications, it is also main operational cost producer in terms of the fuel consumption. This, together with the reduced limits in emission legislations has driven the development of the control layouts, in the last decade especially. The electro/hydraulic controls demanded some engine mechanical changes as well as the changes in the auxiliary systems as described in the first 3 chapters. The marine powertrain industry is currently migrating towards the hybrid applications which will make systems even more complex to handle and maintain.

Due to the powertrain complexity, the risks of having the engine breakdown at the open seas are increasing. Chapter 4 deals with the research of the most common engine breakdowns. Although there are measures to preserve an engine, most of them are active when the alarms take already place. Most of the safety measures are focused on the engine slowdown or shutoff. This produces the costly downtimes until the repairs are done, sometimes even not possible at the seas as well as increases the danger when running without propulsion on the sever weather conditions or in the port. In order to foresee the engine breakdowns and avoid costly and dangerous events, the placement of monitoring and diagnostic tools is of an essential need. The diagnostic approaches presented in the chapter 5 are the research reviews from most diagnostic products currently available on the market (Appendix 1). The benefits and drawbacks of each method are emphasized and the one who showed the least drawbacks is the digital twin method. The method was conceptually introduced in 1997. By [18], reintroduced in 2000. [35] and applied at 2010. by [19]. Due to its setup complexity has never been broadly applied, but has gained more interest with newly introduced technologies with connected machinery into the cloud services.

The highest benefit of the digital twin method is its predictability and the possibility to isolate and resimulate engine component behavior and by it, precisely detecting the root cause of the engine failure even at the early stage of possible breakdown. The biggest drawback of the method is the requirement of the specific knowledge for the digital twin generation and its calibration. The novelty in the thesis will focus on introducing the methodology for digital twin generation, demonstrating also the automatic method for the model calibration in order to bring the model results quality as close to the real engine measurements and still ensuring the desired model predictability. In order to make the model calibration method generally applicable for the different kinds of engine families such as 2 stroke large engines, 4 stroke medium and high speed diesel engines, the advanced engine model is selected as seen in chapter 6. The standard Vibe, single or 2 zones combustion models need to be drastically altered in order to cope with the model output quality requirements when applying it generically to different varieties of diesel engine families. The advanced multizone combustion model should cope with all needs in terms of combustion results quality as well as combined with emission submodels, the emission predictability [23, 24, 25, 26, 27, 29 ,30, 32, 33, 34, 36]. For the diagnostic purpose it is also important to be able to simulate the additional components such as injectors, fuel pump, turbocharger and charge air cooler as described by models in chapter 6.

The methodology developed of model generation and calibration in the thesis will be able to demonstrate the model or digital twin results quality referenced to the engine testbed measurements for 2 stroke large and 4 stroke medium speed diesel engine, using same model structure but with different model calibration parameter sets. When having the digital twin calibrated, it will be possible to develop the predictability of the engine failure simulation will be performed by artificially generating the engine faults on the engine model and creating from the failure simulation results which based on the failure specific sensor value patterns, automatically detects the engine fault while still being in its early stage.

Based on the demonstrated digital twin quality, the new approach in diagnostic method will be evaluated together with the engine optimization concept aiming to merge the digital twin with the engine control unit and chief engineer onboard the ship. The research is tending to demonstrate the usage of the smart approach in the digital twin calibration as well as demonstrate the new concept of using the digital twin in favor of engine safety, fuel consumption and emission reduction.

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LIST OF SYMBOLS

A_{NH}	area of the nozzle hole
A_{NS}	area of needle seat
A_i	zone surface calculated from the zone volume with assumption of its spherical shape
a_p	model parameter
A_{trans}	heat transfer surface
b_{O_2}	model parameter
COM-EU	common electronic unit
CYL-EU	cylinder electronic units
c_{arrh}	model parameter
c_D	injector nozzle discharge coefficient
$C_{Entrain}$	model parameter
C_{eth}	model parameter
C_{eth}	parameter to control the heat up of the droplet
C_{Evap}	parameter for controlling the evaporation rate
C_{Evap}	model parameters
C_{IgnDel}	ignition delay parameter
C_{IgnExp}	ignition delay exponent
c_p	specific heat capacity
c_p	mean value of the specific heat at constant pressure between the compressor inlet and outlet
c_p	mean specific heat at constant pressure between turbine inlet and outlet
$c_p(T)$	temperature dependent specific heat
C_{rad}	model parameter
C_1,C_2, C_3	model parameters
D_{hyd}	hydraulic diameter
d_{inj}	injector nozzle diameter
$\frac{dm_{inj}}{dt}$	mass flow through injector

$\frac{dm_{inj}}{dt}$	injected fuel mass flow
$\frac{dm_{pump}}{dt}$	mass flow through pump
$\frac{dx_{fb}}{dt}$	the fuel burning rate
E	the bulk modulus for the working fluid
EGR	exhaust gas recuperation
FIVA	Fuel Injection and Valve Actuation
f_{ax}	correction functions to account for axial positon in the spray
F_{fr}	friction force
F_{HT}	heat transfer multiplier
f_{rad}	correction functions to account for radial positon in the spray
F_{target}	factor used to evaluate target pressure drop or target efficiency
$\dot{H}_\iota$	are heat fluxes
h_{03}	total enthalpy at the turbine inlet
h_{04}	total enthalpy at the turbine outlet.
h_1	enthalpy at the inlet of the compressor
h_2	enthalpy at the outlet of the compressor
InFI	intelligent fuel injection
InVA	Intelligent valve actuation unit
$i_{ax,max}$	number of all axial zones
i_{ax}	index of axial zones
i_{rad}	radial position of the package in the spray
I_{TC}	turbocharger wheel inertia
K_b	chemical reaction parameter
L	length
m	solid wall mass
$\dot{m}$	mass flow rate
$\dot{m}_c$	mass flow rate in the compressor
$\dot{m}_t$	mass flow rate in the turbine
m_i	zone mass at t
m_{inj}	injected package fuel mass
P_C	compressor power consumption

p_{cyl}	pressure inside cylinder
p_{pipe}	pressure inside pipe
p_s	saturation pressure
P_t	turbine power
$\frac{p_{02}}{p_{01}}$	total to total compressor pressure ratio
$\frac{p_4}{p_{03}}$	turbine expansion ratio, total to static
t	time
TDC	top dead center
T_{ch}	charge temperature
T_i	temperature of zone
T_{inlet}	inlet fluid temperature
T_{IgnDel}	temperature for ignition delay calculation and it represents mean spray temperature
T_l	temperature of liquid
T_s	temperature of the solid
T_w	wall temperature
T_{03}	total turbine inlet temperature
T_1	compressor inlet temperature
V_{rail}	rail volume
α	heat transfer coefficient
α_{ht}	heat transfer coefficient
β	actual engine constant
β_0	the reference constant
Δp	injection pressure drop
Δp	pressure drop over the charge air cooler
Δt_{inj}	number of radial zones
ζ	friction factor which a function of the Reynolds number
ζ_{NH}	nozzle hole flow coefficient
ζ_{NS}	needle seat flow coefficient
$\eta_{s,t}$	isentropic turbine efficiency
$\eta_{s,C}$	isentropic efficiency of the compressor

κ	ratio of the heat capacity at a constant pressure to the heat capacity at constant volume
λ_m	thermal conductivity
ν	kinematic viscosity
ρ	density
ρ_{ch}	charge density
ρ_{ch}	charge density
ρ_{fuel}	liquid density of the fuel
ρ_{fuel}	density of the fuel
ρ_{rail}	density of the fluid inside rail
σ_{fuel}	surface tension of the fuel
τ	characteristic ignition delay time
ω_{TC}	turbocharger wheel speed
SMD	Sauter mean diameter
Nu	Nusselt number
Sh	Sherwood's number

REFERENCES

[1]*"History of Technology: Internal Combustion engines". Encyclopædia Britannica. Britannica.com. Retrieved 2012-03-20*

[2] Georgano, G.N. *Cars: Early and Vintage 1886–1930*(London: Grange-Universal, 1990), p.9.

[3] Matulić, Nikola; Radica, Gojmir; Nižetić, Sandro. Thermodynamic analysis of active modular internal combustion engine concept: targeting efficiency increase and carbon dioxide emissions reduction of gasoline engines *// International journal of energy research,* SI (2018), 1-13. doi:10.1002/er.4055

[4] Milić, L.: Brodski dizelski motori I, Pomorski odjel, Dubrovnik, 2002.

[5] Vamshikrishna Kuruva, Marine Diesel Engine Introduction And Theory, Scribd, 29.12.2015.

[6]ME Engines-the New Generation of Diesel Engines;
http://www.mandiesel.com/files/news/filesof2810/p412-0503.pdf

[7] Radica, G.: Inteligentni brodski dvotaktni dizel motori, Pomorski fakultet, Split 2011.

[8] ME Control System;
http://download.odessacrewing.com/test/MAGIC%20FINISH%20PLAN/MANBW%20Files/0603%20ME%20Control%20System.pdf

[9] MarineInfoSite;

http://marineinfobox.blogspot.com/2017/06/properties-and-working-system-of-marine.html Retrieved 2019-07-14

[10] Marine Engineer World;

http://www.free-marine.com/i3dieselcooling.htm Retrieved 2019-07-17

[11] Faculty of Maritime Studies University of Rijeka;

https://www.pfri.uniri.hr/bopri/documents/15_StartingAirSystem.pdf Retrieved at 2019-07-17

[12] M. Morsy El Gohary , Khaled M. Abdou: Computer based selection and performance analysis of marine diesel engine; Alexandria Engineering Journal (2011) 50, 1–11
[13] WinGD_Marine_Information_Manual, DBAB646817, 2017-04-06
[14] DIESEL ENGINEERING for pacific island mariners:
https://spccfpstore1.blob.core.windows.net/digitallibrary-docs/files/23/23e06da86f1ee89e80f8a4219f16fefe.pdf?sv=2015-12-11&sr=b&sig=4MRAj4NvoY3%2BvM%2B6FAvIMN8RDt96qEzOHMb6%2FdxkcFE%3D&se=2020-05-11T05%3A35%3A20Z&sp=r&rscc=public%2C%20max-age%3D864000%2C%20max-stale%3D86400&rsct=application%2Fpdf&rscd=inline%3B%20filename%3D%22diesel_lg.pdf%22
Retrieved 2019-11-13
[15] https://maritimeexpert.files.wordpress.com/2016/08/b1-operatinginstructions1.pdf
Retrieved 2019-12-15
[16] B.Salic: Primjena ekspertnih sustava za odrzavanje suvremenih brodskih motora, Split, 2016
[17] Oihane C. Basurko , Zigor Uriondo : Condition-Based Maintenance for medium speed diesel engines used in vessels in operation; Applied Thermal Engineering, 07.02.2015

[18] D.T. Hountalas, A.D. Kouremenos: Development and application of a fully automatic troubleshooting method for large marine diesel engines; Applied Thermal Engineering 28.10.1997
[19] V.T. Lamaris, D.T. Hountalas: A general purpose diagnostic technique for marine diesel engines – Application
on the main propulsion and auxiliary diesel units of a marine vessel Energy Conversion and Management 51 (2010) 740–753
[20] https://www.dma.dk/Documents/Publikationer/Digital%20Twin%20report%20for%20DMA.PDF Retrieved at 2020-05-03
[21] https://thedigitalship.com/download-digital-ship/send/16-2018/2658-digital-ship-132-october-2018 Retrieved at 2020-05-03
[22] Cruise M Manual, 2015
[23] Poetsch, C., Ofner, H., Schutting, E., "Assessment of a Multi Zone Combustion Model for Analysis and Prediction
C0I Engine Combustion and Emissions", SAE 2011-01-1439, SAE World Congress 2011, Detroit, USA
[24] Reinhardt, H., "Modellierung des Zuendverzuges bei Mehrfacheinspritzungen aufgeladener Dieselmotoren mit Direkteinspritzung innerhalb der thermodynamischen Prozessrechnung". Abschlussbericht. FVV Heft 870, 2008
[25] Jung, D., Assanis, D.N., "Multi-Zone DI Diesel Spray Combustion Model for Cycle Simulation Studies of Engine Performance and Emissions", SAE 2001-0-1246, SAE World Congress 2001, Detroit, USA.
[26] Pattas K., Haefner G., 'Stickoxidbildung bei der ottomotorischen Verbrennung', MTZ Nr. 12, 397-404, 1973.
[27] Hiroyasu, H., Kadota, T., Arai M., "Development and Use of a Spray Combustion Modeling to Predict Diesel Engine Efficiency and Pollutant Emissions, Part I": Combustion Modeling, Bulletin of the JSME, Vol. 26, pp.569-575, 1983
[28]Ø. Buhaug, J.J. Corbett, Ø. Endresen, V. Eyring, J. Faber, S. Hanayama, D.S. Lee, D. Lee, H. Lindstad, A.Z. Markowska, A. Mjelde, D. Nelissen, J. Nilsen, C. Pålsson, J.J. Winebrake, W. Wu, K. Yoshida, Second IMO GHG Study 2009, International Maritime Organization (IMO), London, United Kingdom, 2009, p. 240.
[29] V.T. Lamaris *, D.T. Hountalas: A general purpose diagnostic technique for marine diesel engines – Application
on the main propulsion and auxiliary diesel units of a marine vessel, Energy Conversion and Management 51 (2010) 740–753
[30] Hiroyasu, H., Kadota, T., Arai M., "Development and Use of a Spray Combustion Modeling to Predict Diesel Engine Efficiency and Pollutant Emissions, Part I": Combustion Modeling, Bulletin of the JSME, Vol. 26, pp.569-575, 1983
[31] The World Merchant Fleet in 2011, Statistics from Equasis, Lisbon, Portugal, 2012.
[32] Varde, K.S., Popa, D.M., Varde, L.K., "Spray Angle and Atomization in Diesel Sprays". SAE 841055, 1984 SAE World Congress, Detroit, USA
[33] Bozza F. and Gimelli A., "A Comprehensive 1D Model for the Simulation of a Small-Size Two-Stroke SI Engine", SAE Paper 2004-01-0999, 2004.
[34] Poetsch, C., Ofner, H. and Cartellieri, W., "Analysis of Thermodynamic Characteristics of Diesel Engine Emission Control Strategies Using a Multi-Zone Combustion Model," *SAE Int. J. Engines* 5(3):2012, doi:10.4271/2012-01-1340.
[35] Dimitrios T. Hountalas," Prediction of marine diesel engine performance under fault Conditions", Applied Thermal Engineering 20 (2000) 1753±1783

[36] Fabio Scappin , Sigurður H. Stefansson , Fredrik Haglind, Anders Andreasen, Ulrik Larsen, “Validation of a zero-dimensional model for prediction of NOx and engine performance for electronically controlled marine two-stroke diesel engines”, Applied Thermal Engineering 37 (2012) 344e352
[37] Dubravko Miljković,“ Fault Detection Methods: A Literature Survey”, MIPRO 2011/CTS, Page 110-115
[38] Skliros C, Esperon Miguez M, Fakhre A, Jennions IK. A review of model based and data driven methods targeting hardware systems diagnostics. Diagnostyka. 2019;20(1):3-21. http://dx.doi.org/10.29354/diag/99603
[39] Foyen, A., „Aspects of condition monitoring and maintenance of machinery“,Norvegian Shipping news, No11C, 1977.
[40] Ostergard A, Fischer P, „B&W condition check system CC-10 for 2-storke K-GF diesel engines“, Proceeding of 2-nd IFAC/IFIP Symposium, Washington 1976.
[41] Sagawa, R., Nakamura, Y., Development of new diagnosis and trend analysis system for marine diesel engine, Bulletin of the Mar. Eng. Soc., Vol. 18, Japan 1980.
[42] Matthew Daigle, Indranil Roychoudhury, „Mode-Based Diagnostics“, Diagnostics & Prognostics Group, Intelligent Systems Division NASA Ames Research Center, PHM 2016: Model-based Diagnostics
[43] Shi N-Z, Tao J. Statistical hypothesis testing. Toh Tuck Link, Singapore: World Scientific Publishing Co. Pte. Ltd.; 2008.
[44] Siegmund D. Sequential Analysis. New York: Springer; 1985.
[45] Mohanty SR, Pradhan AK, Routray A. A cumulative sum-based fault detector for power system relaying application. IEEE Trans Power Deliv. 2008;23(1):79–86.
[46] Isermann R. Fault diagnosis of machines via parameter estimation and knowledge processing-Tutorial paper. Automatica. 1993;29(4):815–35.
[47] Isermann R. Process Fault Detection Based on Modeling and Estimation Methods-A Survey. 1984;4(4).
[48] Volponi AJ. Foundations of Gas Path Analysis I. In: Gas turbine condition monitoring and fault diagnosis - Lecture series 2003-01. von Karman Institute for Fluid Dynamics; 2003.
[49] Volponi AJ. Foundation of Gas Path Analysis II. In: Gas turbine condition monitoring and fault diagnosis - Lecture series 2003-01. von Karman Institute for Fluid Dynamics;
[50] Doel DL. An Assessment of Weighted-Least-Squares Based Gas Path. In: International gas turbinbe and aeroengine congress and exposition. Cincinati, Ohio: ASME; 1993.
[51] Urban LA. Gas path analysis applied to turbine engine condition monitoring. J Aircr. 1973;10(7):400–6.
[52] Doel DL. Development of baselines, influence coefficients and statistical inputs for gas path analysis. In: Gas turbine condition monitoring and fault diagnosis - Lecture series 2003-01. von Karman Institute for Fluid Dynamics; 2003.
[53] Stamatis AG. Optimum Use of Existing Sensor Information for Gas Turbine Diagnostics. In: Proceedings of the ASME Turbo Expo 2008 http://proceedings.asmedigitalcollection.asme.org/proceeding.aspx?articleid=1623480
[54] Volponi AJ. Gas Turbine Engine Health Management: Past, Present, and Future Trends. J Eng Gas Turbines Power. 2014;136(5):051201. http://gasturbinespower.asmedigitalcollection.asme.org/article.aspx?doi=10.1115/1.4026126
[55] A.Stamatis, K. Mathioudakis, G. Berrios KP. Jet engine fault detection with discrete operating points gas path analysis. J Propuls Power. 1991;7(6).
[56] Doel DL. TEMPER-A Gas-Path Analysis Tool for Commercial Jet Engines. J Eng Gas Turbines Power 1994;116(1):82. http://link.aip.org/link/JETPEZ/v116/i1/p82/s1&Agg=doi
[57] Lifetime Project, Athens, October 8, 2003

[58] N. P. Kyrtatos*, G. G. Dimopoulos, G. P. Theotokatos, E. I. Tzanos and N. I. Xiros, „NOx-BOX: A SOFTWARE SENSOR FOR REAL-TIME EXHAUST EMISSIONS ESTIMATION IN MARINE ENGINES“, Laboratory of Marine Engineering Department of Naval Architecture & Marine Engineering National Technical University of Athens
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Appendix 1

Company	Engine optimizaton	Engine diagnostic	Trim	Routing	Product	Link
ABB/Amarcon	✓	✓	✓	✓	OCTOPUS	http://new.abb.com/marine/systems-and-solutions/automation-and-marinesoftware/advisory
	✓	✓	✓	✓	OCTOPUS	http://www.abb.com/cawp/seitp202/dfc860263654b00bc1258026004bca2f.aspx
	✓	✓	✓	✓	OCTOPUS	https://library.e.abb.com/public/abed9ca5b2ae4446b97f8fec538ead62/OCTOPUS%20for%20Cargo%20vessels.pdf
	✓	✗	✓	✓	OCTOPUS	https://library.e.abb.com/public/41ae773717a54c12a8aaf50309390060/OCTOPUS%20for%20Cruise%20vessels.pdf
	✓	✗	✗	✗	OCTOPUS	https://library.e.abb.com/public/4ba613b3e2024083ba77ae0a88949a45/OCTOPUS%20for%20LNG%20vessels.pdf
	✓	✗	✗		OCTOPUS	https://library.e.abb.com/public/f769455b564c469195196df3d4945285/OCTOPUS%20for%20Offshore%20vessels.pdf
ABS	✗	✓	✗	✗	OEP	http://ww2.eagle.org/en/what-we-offer/asset-performance/vessel-performance/performance-monitoring.html
	✓	✗	✗	✗	APM	http://ww2.eagle.org/en/what-we-offer/asset-performance/vessel-performance/performance-optimization.html
Applied Weather Technology (StormGeo)	✗	✗	✗	✓		http://www.stormgeo.com/shipping/shore-to-ship-services/awt-ship-routing/
AtoBviaC	✗	✗	✗	✓		https://www2.atobviaconline.com/Pages/MarineDistanceTables?Length=5 https://www2.atobviaconline.com/Pages/RouteingCriteria https://www2.atobviaconline.com/Pages/TheNeedForAccuracy
BMT SMART	✗	✓	✓	✗	SMART FLEET	http://www.bmtsmart.com/our-products/smartfleet/
	✗	✗	✗	✗	SMART VESSEL	http://www.bmtsmart.com/our-products/smartvessel/
	✗	✗	✗	✓	Metocean	http://www.bmtsmart.com/our-products/metocean-data/
ClassNK-NAPA	✓	✗	✓	✓		https://www.napa.fi/Ship-Operations/ClassNK-NAPA-GREEN-Optimization/Voyage-Optimization, https://www.napa.fi/Ship-Operations/ClassNK-NAPA-GREEN-Optimization
Jeppesen-C-Map	✗	✗	✗	✓	VVOS	http://commercialmarine.c-map.com/en/route-planning-and-fleet-management/c-map-vessel-and-voyage-optimization-solution-vvos
Danaos Information Service	✗	✓	✗	✓	WAVES	http://web2.danaos.gr/wp-content/uploads/2015/07/Danaos_Waves-Fleet-Performance.pdf http://web2.danaos.gr/maritime-software-solutions/optimal-routing-system/
DNV GL	✓	✗	✓	✗	ECO Assistant	https://www.dnvgl.com/services/energy-system-simulation-81199 https://issuu.com/dnvgl/docs/dnv_gl_pospaper__6-2014_beyond_cond https://www.dnvgl.com/services/eco-assistant-effective-trim-optimisation-1422 https://www.dnvgl.com/software/ship_management_and_operations/index.html
ENIRAM	✗	✗	✓	✗	TRIM	http://www.eniram.fi/product/eniram-trim/
	✓	✗	✗	✗	Speed	http://www.eniram.fi/product/eniram-speed/
	✗	✓	✗	✗	Engine	http://www.eniram.fi/product/eniram-engine/
	✗	✓	✗	✗	Performance	http://www.eniram.fi/product/eniram-performance/
	✗	✓	✗	✓	Fleet	http://www.eniram.fi/product/fleet-performance-manager/
ESGR	✓	✓	✗	✗	Ostia Edge	
Force Technology	✓	✗	✗	✗	Sea Engine	https://forcetechnology.com/en/maritime-industry/cases/seasuite
	✗	✗	✗	✗	Sea Logger	
	✗	✗	✗	✓	Sea Planner	
	✗	✗	✗	✗	Sea Trend	
	✗	✗	✓	✗	Sea Trim	
Global Navigation Solutions	✗	✗	✗	✓	Voyager	http://www.globalnavigationsolutions.com/voyager/
Green Steam	✗	✓	✓	✗	GreenSteam	http://greensteam.com/how-it-works.html
INTERSCHALT Maritime Systems	✗	✗	✓	✗	TROP Module	http://www.interschalt.com/software/macs3-loading-computer.html
	✓	✓	✗	✗	Bluetracker	http://www.interschalt.com/fileadmin/user_upload/Brochuere_GoodSeamanship2.0.pdf
Kongsberg	✗	✓	✗	✗	AVL_EPOS	https://www.km.kongsberg.com/ks/web/nokbg0240.nsf/AllWeb/89547440DCE18A7CC12579EB0043FD93?OpenDocument
	✗	✗	✓	✓	N/A	https://www.km.kongsberg.com/ks/web/nokbg0240.nsf/AllWeb/8200D31F7A44C8FEC12579C60043D84B?OpenDocument
Kyma	✓	✓	✗	✗	KSP system	https://www.kyma.no/#products/ksp https://www.kyma.no/#products/mip https://www.kyma.no/#products/ksp/module-diagnostics
	✗	✗	✓	✗	KSP system	https://www.kyma.no/#products/ksp/module-trim
MAC system solutions	✗	✗	✗	✗	Data transfer	https://www.macsystemsolutions.com/maritime
	✗	✗	✓	✗	Trimp/List measurement	
	✗	✗	✓	✗	Data sampling	
	✗	✗	✗	✗	Shaft Power measurement	
MACSEA	✗	✓	✗	✗	DEXTER	http://macsea.com/dexter/new-dexter-diesel-expert-module-helps-keeps-engines-fuel-efficient/ http://macsea.com/systems-solutions/dexter/overview/ http://macsea.com/systems-solutions/hull-medic/features-benefits/
MARORKA	✗	✓	✗	✗	Machinery	http://www.marorka.com/products/#marorka-onboard
	✗	✗	✗	✓	Voyages	http://www.marorka.com/products/#marorka-onboard
	✗	✗	✓	✗	Trim	http://www.marorka.com/products/#marorka-onboard
WinGD	✓	✓	✗	✗	WiDE	https://www.wingd.com/en/digital-solutions/wide/
	✗	✓	✗	✗	EDS	https://www.wingd.com/en/digital-solutions/wide/eds/
	✗	✗	✗	✗	DCM	https://www.wingd.com/en/digital-solutions/wide/dcm/

ABSTRACT

The marine engine is the most important segment of the machinery onboard the ship. It's not only observed from the performance, but also from the safety perspective. The marine engines are getting more complex driven by emission legislations in terms of control configurations, complex auxiliary systems and hybridization. The complexity increases the risk of failures which are not just the cost parameter but also a significant safety factor.

The monitoring and diagnostic systems became the central tool for engine maintenance strategies and are a ground upon which the engine optimization can be performed. However, most of the diagnostic methods on the market have under its benefits also a significant drawbacks in terms of unreliable or not detectable engine faults or costly and timely preparation phase.

In the qualifying paper different kind of diagnostic methods were investigated where the digital twin diagnostic concept is isolated as the most rounded method but currently with very less application activities on the market. One of the main reasons is the required expert know-how for digital twin generation and skills for the digital twin calibration. The advanced digital twin engine model is physically described together with all diagnostic relevant auxiliary models. The new concepts of the digital twin application concepts were demonstrated driven by the nowadays available engine control unit communication and satellite data transfer technologies. If the digital twin is able to demonstrate its easy adaption/calibration to the new engines or data and keeping its required results quality, it would open the door for presented optimization and diagnostic concepts to be applied in the real-world applications. This would lead to significant fuel consumption reduction, cleaner exhaust emissions and decrease in the machinery breakdowns.

SAŽETAK

Brodski motor je najznačajniji stroj na cijelome brodu ne samo iz perspektive njegovih performansi već značajno i u pogledu sigurnosti same plovidbe. Vođeni legislativom, brodski motori postaju sve kompleksniji u vidu kontrolnih strategija, kompleksnih pomoćnih strojeva i uređaja te same hibridizacije pogona. Njihova kompleksnost uvećava rizik pojave kvara koji nisu samo financijski parametar već i značajni sigurnosni faktor.

Dijagnostički i nadzorni sustavi su postali centralni alat za planiranje održavanja brodskog motora i baza na kojoj se izvršava optimizacija rada brodskog pogona. Međutim, većina dijagnostičkih alata na tržištu osim svojih prednosti posjeduju i značajne nedostatke u smislu nepouzdanih ili ne detektiranih kvarova ili su metode koje zahtijevaju skupu i dugotrajnu pripremu.

U kvalifikacijskom radu istražene su različite dijagnostičke metode te dijagnostički koncept uporabe digitalnih blizanaca ili virtualnih simulacijskih modela fizičke komponente u svrhu detekcije kvara motora je istaknut kao najpotpunija metoda. Metoda se još ne koristi u brodskoj industriji. Jedan od glavnih razloga tome je potrebno znanje u razvoju digitalnih blizanaca i vještine pri kalibraciji istih. Opisan je napredan fizikalni model motora zajedno sa svim dijagnostički relevantnim pomoćnim ili dodatnim modelima. Demonstrirani su novi koncepti uporabe digitalnih blizanaca vođeni tehnologijama trenutnih komunikacijskih sučelja motornih kontrolnih jedinica te satelitskim tehnologijama prijenosa podataka sa broda na kopno. Zaključeno je da ukoliko bi razvoj digitalnog blizanca za specifični motor bio brz i jednostavan uz zadržavanje željenih kvaliteta rezultata, bilo bi omogućena primjena opisanih koncepata optimizacije i dijagnostike motora na samome brodu. To bi vodilo značajnom smanjenju potrošnje goriva, smanjenju emitiranja ispušnih emisija te povećanju sigurnosti sa smanjenjem broja kvarova motora.