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Dynamic 2-stroke engine model for Hardware-in-the-Loop testing

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ABSTRACT

Winterthur Gas & Diesel Ltd. has developed a dynamic engine model of 2-stroke diesel and dual-fuel engines. The presented dynamic engine model provides the possibility to test the complete engine control software in Hardware-in-the-Loop fashion. The engine simulator is equipped with the real hardware modules needed to control an up to twelve cylinder diesel engine or an up to six cylinder dual-fuel engine, where the actuator outputs are connected to realistic electrical loads. The simulator also emulates external systems like the remote control system and gas valve unit, which makes it possible to test the entire automation setup as installed on the ship.

The objective of this article is to describe the dynamic engine model designed in Matlab Simulink and embedded in a NI-Veristand environment that executes all necessary tasks in real-time. The engine model consists of a physical common rail system with multiple pumps and injectors, a thermodynamic model of the cylinder pressure with a simplified combustion as well as a mechanical model for the load computation related to operating point and engine specific inertia. It also computes all relevant engine feedback signals in real time, such as exhaust valve position, gas admission valve position, lubrication pressure feedback, cylinder pressure and more.

Regarding the utilization of the dynamic engine model, the paper will also focus on the new possible control software tests that can be performed compared to the previously used steady state test rigs. The main purpose of this engine simulator is to test the behaviour of the engine control system in normal operation as well as in emergency conditions. Typical emergency cases are malfunctions of control modules, communication buses, actuators or sensors. It is also used for testing control sequences crucial for the correct engine operation, e.g., engine starting, fuel transfer from diesel to gas operation, redundancy functionalities. Additionally it enables the possibility to test and monitor the interaction of independently controlled closed loop systems, e.g., the interaction between the fuel pressure controller and the speed controller of the engine.

By using an engine simulator which is able to represent engine hardware and its dynamic behaviour it is possible to develop a more robust engine control software in much shorter time.

INTRODUCTION

With the introduction of the very first electronically controlled Common Rail System on a 2-stroke marine diesel engine (Sulzer 6RT flex58T B) in 2001 [1] the electronic control started its success also in this field of application for marine deep-sea shipping. The biggest advantage of the electronic control compared to the classic mechanical one is the wide range of adjustability and flexibility. This enables the full performance of e.g. a common rail fuel injection system as well as a hydraulic activated exhaust valve to optimize both fuel consumption and exhaust emissions over a broad operational range. Today, all the new X-series engines, diesel as well as dual fuel engines, from WinGD are available only in the version of an electronically controlled engine. This type of engine is fully integrated into the electronic control and monitoring system of the vessel. The potential of the electronic control devices is most often used to adapt the engines in a cost efficient way to a broader range of operations. Also, the ease in acquiring operational parameters for further investigation, either manually or automatically, have made the systems more and more complex. An end is not visible, it will rise further proofing Moor's Law.

However, the time to market is limited. The available time for development is becoming shorter and the number of development cycles must be kept low. Additionally, development is facing the challenge of the continuously rising complexity. These requirements are difficult to tackle with a more excessive final testing on the real applications since access to engines on the test bench as well as engines in ships is limited, expensive and late in the development cycles. Simple reduction of tests is no option, since a safe and reliable operation of the final product has to be ensured meeting with the quality standards and classification requirement.

Consequently, possible faults should be detected as early as possible. For the control software the testing starts on the computer itself, however limited to the constraints of this environment. A more complex and real testing environment is given by the usage of a hardware-in-the-loop (HIL) set-up, where the real control hardware is running against virtual systems, in our case the virtual 2-stroke engine.

A very high level of real world simulation is already given with the real control hardware interacting with its major control counterparts in the remote control system of the ship. The only abstraction herewith is an engine model developed in MATLAB Simulink on a standard computer and executed on a Real-Time System of National Instruments (NI), which represents the real engine, the "real process". With this environment, an already high level of validity of the final system behavior is possible. Especially, the behavior of the control system in transient conditions can be investigated in detail and in a representative manner. None the less, the real engine in the real application will always have the final say.

Because real time engine simulations for control development have been in development and use since several decades, the listed references can be reliable examples [2], [3]. Normally, the chosen solutions are specifically adapted to the needs of the development area they are applied to. It has to be taken into consideration that always the more detailed a model the more computation time is consumed.

The implemented engine models used in real time simulation are divided:

- physical or theoretical models, so called white-box models
- simplified theoretical models
- theoretical black-box or grey-box models
- experimental black-box models

The models presented here, are from the class of simplified theoretical model, developed for the specific needs of the 2-stroke applications: The model should be easy to maintain, it must represent the range of operation the control system software is used on, and it must be accurate enough to give the correct feedback occurring in transient operations of the process.

MODELLING OF THE ENGINE

The engine model described here is designed to generate all relevant feedback signals that are necessary for an advanced testing of the Engine Control System (ECS). It consists of closed loop simulated parts, where the model represents the plant part of the control loop to generate the necessary feedback signals. This approach is done for:

- speed, load and torque computation
- fuel pressure computation
- pilot fuel pressure computation
- cylinder pressure computation
- fuel injection computation

Other parts of the engine model generate the dynamic feedback signal based on online configurable characteristic parameters. This approach is done particularly for signals where - from the control system point of view - only monitoring and compensation tasks are executed. This regards the feedback computation of:

- exhaust valve position feedback
- gas admission valve position feedback
- cylinder lubrication pressure feedback
- servo oil pressure feedback
- scavenge air pressure feedback
- exhaust gas receiver pressure feedback
- turbo charger speed feedback

General Hardware description of the simulator

The dynamic engine model is embedded in a test rig that is used for testing new functionalities of the engine control system as well as to perform standard tests, which are executed for every control software released. In particular, it is possible to test the control software for Diesel and Dual Fuel 2-stroke engines. The test rig has the same ECS layout as a real engine with up to twelve cylinders. In addition to the UNIC engine control system itself there are several auxiliary systems connected like on a real installation (figure 1). For the testing of the interaction between the ECS, the Remote Control System (RCS) and the engine itself a complete Kongsberg system with control panel and telegraph for the bridge control as well as for the engine control room is installed. The communication between the RCS and the ECS is realized by a CANopen connection.
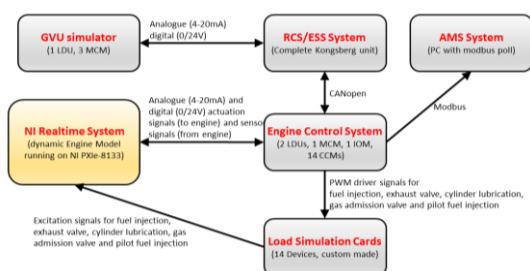

Figure 1: System layout of the test rig including auxiliary systems

For a proper testing of Dual Fuel engines it is also necessary to take the Gas Valve Unit (GVU) into consideration. The GVU is a standalone device, which is installed between the gas feed system and the engine and is responsible for the supply, cut-off and regulation of gas fuel and for the supply of inert gas. Since there is a lot of interaction between the

GVU, the RCS and the ECS a GVU controller and an additional feedback simulator for the GVU are also installed.

The dynamic engine model itself is developed in MATLAB Simulink and runs on a NI Real-Time system. To generate a sufficient accuracy of the simulation the engine model is executed with a fixed sample time of 1 ms which leads to a resolution of minimal 1° crank angle (CA°) for a maximal expected engine speed of 166 rpm. Since the nominal speed of the engine tested here is significantly smaller, it is possible to generate a resolution of approx. 0.7° CA.

The majority of the control valves installed on the engine are solenoid driven vales, which are activated by a current controlled pulse width modulation output (PWM) of the ECS. Since the ECS is designed to drive channel specific pull-in and hold currents and because of the high input impedance of the digital input module of the NI-System it is not possible to connect the ECS outputs directly to the inputs of the data acquisition. To solve this problem load cards are integrated in the test rig. These load cards have channel specific inductors and resistors installed which match the electrical characteristics of the real valves as shown in figure 2.
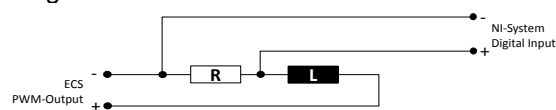

Figure 2: Load card schema for PWM output data acquisition

An additional benefit of the load card implementation is realistic behavior of the output current control and the possibility for testing the open- and short circuit detection algorithms of the ECS. This solution is implemented for exhaust valves, cylinder lubrication valves, main fuel injectors, gas admission valves and pilot fuel injectors.

Modular approach and parameter configuration

The single feedback generating functionalities for a specific purpose like fuel pressure computation, speed / load computation or cylinder pressure simulation are developed and executed in single functional groups. To have the possibility of an easy redesign of single parts of the engine model this approach was chosen. Another benefit is the simplification in case of an engine model extension with a new functionality group, like it was done for the implementation of the gas admission. This was just an add-on to the previous developed diesel part. Due to this it was possible to develop the gas part individually. Afterwards it was just necessary to

add the addition heat caused by the gas and pilot fuel injection to the existing cylinder pressure system.

With the previous generation of steady state simulators it was also a big effort needed to change the engine type or to react on new sensor scaling. With new engine model all needed parameters are directly read out of the ECS software package. That regards sensor scaling, pin partition, actuator types, engine rating or injector characteristics. This enables a fast generation of new engine models and lead to a significant speed-up of the testing.

Cylinder Pressure Computation

The computation of the pressure in the combustion chamber is based on a simplified thermodynamic model where the main parts of the cylinder are computed. Here the model is taking into account the volume change and the corresponding pressure deviation related to compression or expansion of the gas. Additionally all mass and enthalpy flows over the inlet port, exhaust valve and start air valve are taken into account. Furthermore the energy input related to the injection and combustion of the fuel is considered to compute the cylinder pressure at every time step of the simulation.

Simulated parts

For the simulation of the combustion chamber the engine model computes the air flow through the inlet port into the cylinder, the exhaust gases out of the exhaust valve and for starting the air injection through the start air valve for every cylinder as shown in figure 3.
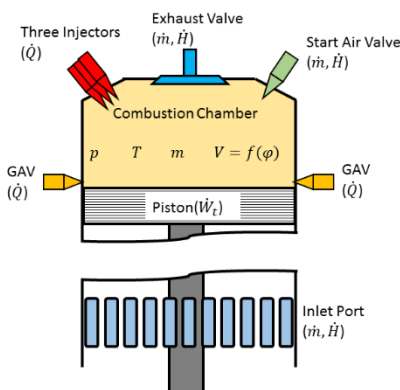

Figure 3: Simulated parts for the cylinder pressure computation

The injection of the fuel is computed in relation to the characteristics of the diesel fuel injector or the gas admission valve installed on the specific system.

Since the model is used for ECS tests and not for testing the engine behavior in detail the pressure

computation is simplified to a reasonable level. Neglected parameters for the computation:

- heat losses on the combustion chamber surface
- temperature dependence of the physical constants of air and exhaust gas mixture in the combustion chamber (c_v, κ etc.)
- pressure losses at the piston rings

Simplified Thermodynamics

To calculate the state of the cylinder volume the first law of thermodynamics for an open system is used. The differences in height and velocity between the volume inlet and the outlet are assumed to be negligible. This simplification results in the shown equation which is used to compute the derivative of the inner energy for one cylinder volume [4].

$$\dot{U} = \sum_i \dot{Q}_i + \sum_j \dot{W}_j + \sum_k \dot{m}_{in,k} * h_k - \sum_l \dot{m}_{out,l} * h_l$$

The sum of the input of heat to the combustion volume only depends on the injected fuel amount and is simply computed by the product of the injected fuel mass multiplied by the lower heating value of the used fuel.

$$\dot{Q} = \dot{m}_{Fuel} * H_{i_{Fuel}}$$

The technical work W_t which is performed by the gas in the cylinder can be calculated by the following equation for the pressure-volume work:

$$\dot{W} = -p_V * \dot{V}$$

V is the volume in regard to the movement of the piston. With the assumption that air is an ideal gas, it is also possible to compute the derivative of the internal energy by the following equation:

$$\dot{U}_V = u_V * \dot{m}_V + m_V * c_v * \dot{T}_V$$

with: $\dot{m}_V = \dot{m}_{in} - \dot{m}_{out}$

results in:

$$u_V * (\dot{m}_{in} - \dot{m}_{out}) + m_V * c_v * \dot{T}_V =$$

$$\dot{m}_{Fuel} * H_{i_{Fuel}} - p_V * \dot{V} + \dot{m}_{in} * h_{in} - \dot{m}_{out} * h_V$$

And rephrased:

$$\dot{T}_V = \frac{\dot{m}_{Fuel} * H_{i_{Fuel}} - p_V * \dot{V}}{m_V * c_v}$$

$$+ \frac{\dot{m}_{in} * h_{in} - \dot{m}_{out} * h_V - u_V * (\dot{m}_{in} - \dot{m}_{out})}{m_V * c_v}$$

With the knowledge about the initial state of the volume it is possible to compute the current state at every time step:

$$T_V = T_{V_{ini}} + \int_0^t \dot{T}_V dt \quad \text{and}$$

$$m_V = m_{V_{ini}} + \int_0^t \dot{m}_V dt \quad \text{and} \quad V_V = f(CA)$$

With help of the ideal gas law the pressure of the volume can also be calculated:

$$p_V = \frac{m_V * R_{Air} * T_V}{V_V}$$

The air flow of start air bottle, scavenge air receiver and combustion chamber are based on the equation of a compressible gas flow over a valve (figure 4). Therefore the equation related to critical and under critical pressure ratios were taken into account.
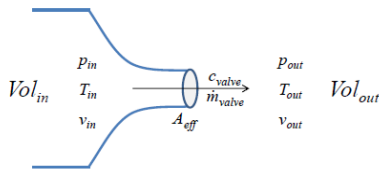

Figure 4: Subsonic gas flow through a nozzle

With: $\dot{m}_{Valve} = \frac{c_{Valve} * A_{eff}}{v_{out}}$ & $c_{Valve} = f\left(\frac{p_{in}}{P_{out}}, v_{in}\right)$

The opening of the scavenge air inlet port and the corresponding effective flow area to the combustion chamber is computed depending on the crank angle and the geometrical characteristics of the engine. The computation of the effective flow area of the exhaust valve is based on the activation signals, geometrical data and the characteristic dynamic behaviour of the valve.

Fuel Injection

The injected energy (fuel) is computed based on the injection timing (duration), the fuel pressure and the characteristics of the injector nozzle tip. A simplified rectangular heat release shape in connection with a PT1 Filter is implemented related to the injected fuel amount at every timestep. Because the purpose of the engine model is automation testing, it is sufficient to use this simplification of the heat release to the combustion chamber.

Fuel oil system / Pilot fuel oil system

The Common Rail System on a combustion engine is responsible for the fuel supply of the injectors. The Common Rail System consists of a high pressure fuel pump, the rail and the injectors. The rail is like an accumulator of high pressured fuel (600 bar to 1000 bar), which is used for the injection. The electronic controlled fuel pumps supply the rail with fuel from the fuel reservoir. The injectors are also electronically controlled by a solenoid valve, to provide maximum flexibility in the injection timing. Due to the high pressure and the solenoid valve it is possible to generate very short injection durations, which improves fuel consumption and the composition of the exhaust gas. The pressure in the rail is controlled by the flow rate of the fuel pumps.

The goal of the Common Rail Model is to simulate behaviour of the dynamic pressure development of the rail and the injected fuel quantity according to the current pressure and the injection command of the control unit.

For modelling the rail pressure, it is assumed that the volume of the rail is constant. With the speed of the engine, the position of the actuator for the fuel pumps and the operating map of the pumps it is possible to calculate the flow rate of the pumps. This is the whole incoming flow rate to the rail. On the other side it is possible to calculate the outflow of the rail. Therefore, the current rail pressure, the injection command from the UNIC controller and the operation map of the injection valve is necessary. The calculation of the pressure sequence can be derivate by the following equations [6].

$$\dot{V} = C_h * \dot{p}$$

$$\dot{p} = \frac{\dot{V}}{C_h} \quad with \ C_h = C_{h_{Fluid}} + C_{h_{Pipe}}$$

Since the pipe represented here is small it is possible to neglect the hydraulic capacity of the pipe. The hydraulic capacity of the fluid only depends on its volume and the bulk modulus of the diesel oil.

$$C_h = C_{h_{Fluid}} = V_{rail} * \frac{1}{\kappa}$$

$$p = \int \dot{p} \ dt + p_{initial} = \frac{V_{rail}}{\kappa} * \int \dot{V} \ dt + p_{initial}$$

$$= p_{initial} + \frac{V_{rail}}{\kappa} * \int \left(\dot{V}_{in} + \dot{V}_{out} \right) \ dt$$

The calculation of the injection quantity is done by the injector map, which represents the behaviour of the main fuel injector used (figure 5). The characteristic points P1 to P6 of the injector maps are based on measurements and are selected for the calculation of this injector curve according to the current pressure. These points represent the response time, the ballistic time period and the completely open period of the injector at highest and lowest pressure of 1000 bar and 600 bar.

With the values of the characteristic points P1 to P6 it is possible to generate a simplified injector map. This simplified injector map enables the possibility of calculating an interpolated injector curve according to the current rail pressure as shown in figure 5. This is done by interpolating points P1 and P4 for the reacting time, P2 and P5 for the ballistic time and P3 and P6 for the open time. The results are points P7, P8 and P9, which represent the current injector curve related to the current rail pressure.
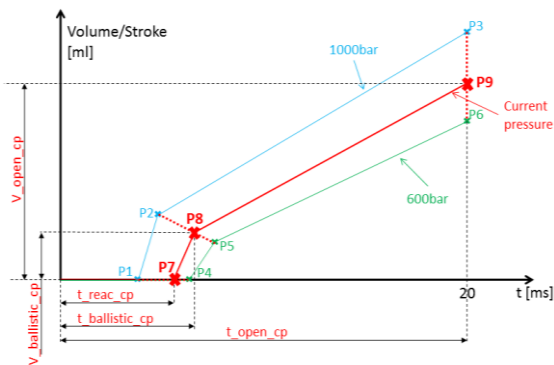

Figure 5: Interpolation of the injector curve depending to the characteristic at 600 bar and 1000 bar

Since the engine model is running with a fixed sample rate of 1 ms it is necessary to interpolate the pressure corrected injector curve in millisecond steps. Because the engine simulation needs a volume flow instead of the injected volume, it is additionally necessary to generate the derivative of this injector curve (figure 6). This procedure leads to a difference during the beginning of the injection. But since the injectors are normally operated with injection durations larger than the ballistic time the injected fuel amount per injection pulse is correct.
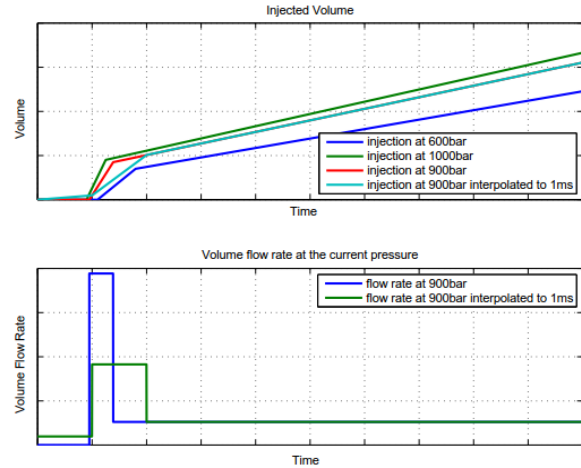

Figure 6: Linear interpolation of the injector curve to match with the engine model sample time of 1 ms

Processing the injection signal of the ECS, the volume out of the fuel rail can be computed by the sum of all participating main fuel injectors. These main fuel injection flows are also used to compute the induced heat to the combustion chamber for the cylinder pressure computation:

$$\dot{V}_{out} = \sum_{i=1}^{N_{cyl}} \sum_{k=1}^{N_{inj}} \dot{V}_{i,k}$$

The simulated fuel pumps are plunger pumps with an upstream installed flow control valve (FCV) that enables the possibility to control inlet volume flow by the actuator position of this FCV. The fuel pumps are driven by a gear wheel connection to the crank shaft.

If the FCV is opened as far as the plungers of the pumps are filled completely with fuel and the flow rate of the pump depends only on the speed of the pump. As soon as the FCV is closed as far as the plungers the pumps are not filled completely and the flow rate of the pumps depends only on the actuator position of the FCV. These two dependencies determine the actual flow through the fuel pump:

$$\dot{V}_{in} = \min\left[\dot{V}(speed); \dot{V}(FCV)\right]$$

Data of the Wärtsilä pumps are shown in the measurements: Here it is a pump with a plunger diameter of 46 mm and an effective stroke of 45.1 mm.
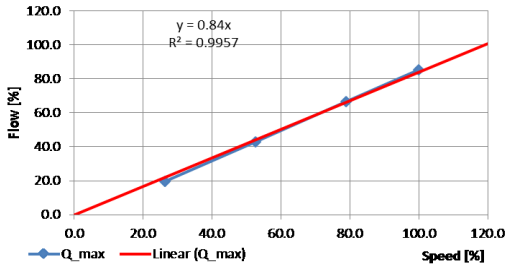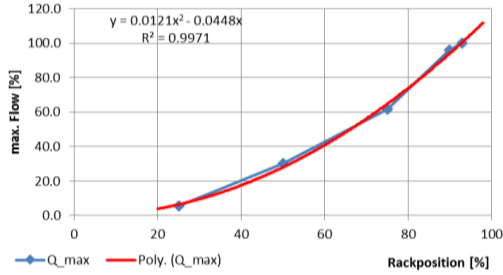
Figure 7: Fuel pump speed dependency

Figure 8: Fuel pump control valve position dependency

The characteristic values of the pump are extracted from the measurement data shown above in figure 7 and 8. For the speed dependency a linear relationship was determined and is shown below.

$$\dot{V}(speed) = 0.84 * n_{pump} * \dot{V}_{max}$$

$$\dot{V}(FCV) = (0.0121 * FCV^2 + 0.0448 * FCV) * \dot{V}_{max}$$

The UNIC control software creates a PWM signal to control the position of the FCV. The detection of the pulse width is done by a Field Programmable Gate Array (FPGA) application, which runs on the NI-system. This application converts the pulse width to a value between 0% and 100%. This percentage value of the FCV is used in the simulation model.

With due consideration of the above shown equation related to the volume flow into and out of the rail it is possible to compute the fuel pressure in the rail as well as the injected fuel amount in relation to the pressure and the injection duration for every time step.

The simulation of the pilot fuel system with one pump and up to three pilot fuel injectors per cylinder is implemented in the same way as described above for the fuel rail system. Since the pilot fuel pump is electrically driven it is not necessary to take the speed of the engine/pump into consideration here. The volume flow of the pilot fuel pump depends only on the on the actuation signal for the connected flow control valve.

Torque computation

For the computation of the torque acting on the crank shaft and leading to an acceleration, a balance equation of all acting torques is formed:

$$0 = M_{ind} + M_{Fric} + M_{Load} + M_{In}$$

For the induced torque caused by the pressure on the upper side of the piston, only the tangential component has to be computed considering the geometry of the crank shaft piston connection as shown below [5].

$$M_{ind_{Cyl}} = \left(\frac{(p_{Cyl} - p_{Scav}) * \pi * d^2 * r}{4} \right)$$

$$* \left(\sin \varphi + \frac{\frac{r}{l} \sin(2\varphi)}{2\sqrt{1 - \left(\frac{r}{l}\right)^2 \sin^2 \varphi}} \right)$$

The simulation of the induced torque is done independently for every single cylinder and the sum of these torques leads to the torque of the engine:

$$M_{Ind} = \sum_{n=1}^{N_{cyl}} M_{Ind_n}$$

For the frictional torque a constant mean effective friction pressure is used based on empirical data measured on different installations. In case the speed is zero a different mean effective friction pressure is used to represent the sticking friction which is slightly above the value during running operation [5].

$$M_{Fric} = \frac{p_{Fric} * N_{Cyl} * d^2 * r}{4} * -sign(n)$$

The torque related to the load at the propeller is computed based on the propeller law for fix pitch propellers:

$$P_{Load} = k * n^3 = M_{Load} * \omega = M_{Load} * n * \frac{\pi}{30}$$

leads to: $M_{Load} = \frac{k*30}{\pi} * n^2 * -sign(n)$

Additionally, there is the restriction that at engine standstill the absolute value of the friction torque can maximally be the absolute value of the induced torque. This is done to prevent an engine acceleration at standstill caused only by the frictional torque.

$$M_{Fric} = \begin{cases} min(M_{Fric}, M\_ind), & n = 0 \\ M_{Fric}, & n \neq 0 \end{cases}$$

Load / Speed computation

The inertia of the whole drive train and the torque caused by the acceleration of the engine allow the balance equation of the torque acting on the crank shaft to be completed. Hereby the torque caused by the inertia is computed by the following equation:

$$M_{In} = J_{tot} * (-\dot{\omega}) = J_{tot} * \left(-\frac{\pi * \dot{n}}{30}\right)$$

Due to reasons of simplification a constant inertia is used, which is independent of the current crank angle.

With the implementation of the above described equation and with the known initial condition it is possible to compute acceleration, speed and load of the engine for every time step:

$$\dot{n} = \frac{M_{ind} + M_{Fric} + M_{Load}}{J_{tot}} * \frac{30}{\pi}$$

Validation of the Engine Model

For the validation of the engine model behaviour several tests where done on the real engine as well as on the simulator to prove the concept of the engine model and there auxiliary systems. The results presented here are for a W-X62DF engine. This engine is a six cylinder dual fuel engine with a bore diameter of 62 cm, a nominal speed of 97 rpm and a nominal power of 13'450 kW.

For the first test the starting behaviour from a specific crank angle (212 CA° in this case). The main focus is set on speed, fuel pressure, firing pressure and the injection duration. The speed set point for the start test was set to 36 rpm. The results for the engine start is shown in the graph below:
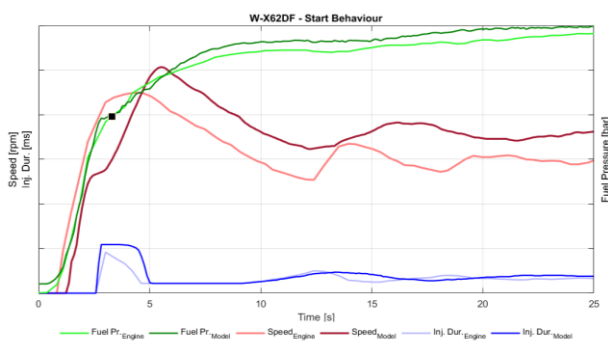

Figure 9: Validation test of engine start behaviour for a W-X62DF engine

Related to the speed behaviour (figure 9) it can be observed that the characteristic shape of the model

speed is quite similar to engine speed. Although the overshoot of the engine speed of the model is slightly higher as on the real engine, the accuracy is sufficient for a software test. This difference in the overshoot is caused by the simplification in the torque and friction computation in the engine model. Comparing the fuel pressure of the real engine and the engine model we see that the model represents the real behaviour in a very accurate way over the entire starting procedure. The sharp bend of the fuel pressure at approx. 3 s (see mark in figure 9) is explained by the activation of the fuel injector at that moment. Related to the bigger fuel command in the engine model at this time the effect on the pressure trace is also more distinct.
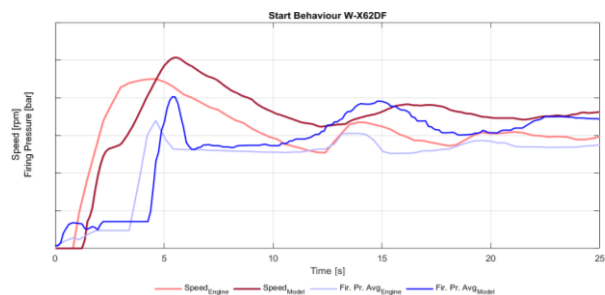

Figure 10: Validation test of engine start behaviour for a W-X62DF engine regarding firing pressure

In figure 10 the firing pressure of the engine is monitored for the same test. The firing pressure evaluation in the UNIC system is done by the digital signal processor and sent to the main application once per revolution. To have a clearer view to the characteristics of the firing pressure values, the pressure values shown here are the average of all six cylinders. From the graph in figure 10 it is visible that the absolute values of the firing pressure in the model are constantly above the values measured on the real engine. The observed differences are explicable by the simplifications done in the pressure- and torque computation in the engine model. In spite of a small offset in the firing pressure measurement the characteristics of the trace matches the reference measurement of the real engine and the accuracy achieved here is sufficient to test the starting behaviour from a software point of view.

As a second validation test the engine behaviour during deactivation and activation of one Cylinder Control Module (CCM) is compared. The CCMs control all cylinder specific functions like fuel injection, exhaust valve operation and cylinder lubrication. Additionally, there are also engine common functions redundantly controlled by single CCMs like fuel pressure control or servo oil pressure control. In the test shown here the deactivation and activation is done for CCM Two, where the fuel pressure controller and the

corresponding hardware output are located. The test performed here was done at low load.
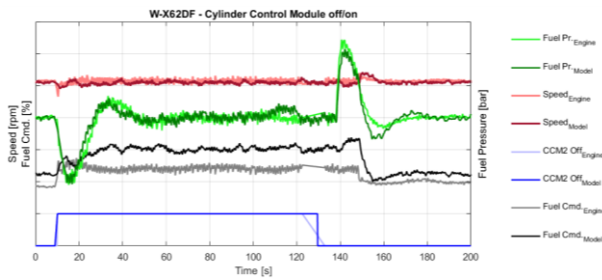

Figure 11: Validation test of deactivation and activation of CCM two for a W-X62DF engine

Figure 11 shows the effects of deactivating the control module. In this chart the CCM-Off signal is scaled with factor ten to provide a better view.

In figure 11 it is visible that the fuel pressure response caused by the missing volume flow from one fuel pump matches the reference measurement taken on a real engine with very high accuracy. Related to the speed behaviour during the deactivation of one unit the speed drop and the stabilisation time for the real engine as well as for the engine model is very similar. The same accuracy is observed for the reactivation of the CCM of Cylinder Two. Related to the absolute value of the fuel command for the engine model it can be observed that the simulated engine needs a high fuel command at the same steady state operation point as the real engine. This difference is explicable by the simplifications done for the torque computation caused by the fuel injection as well as for the friction computation. Since the characteristics of the fuel command during the deactivation as well as during reactivation of CCM is similar to the real engine this accuracy is also adequate for software tests.

As a third test for the validation of the engine model the cylinder balancing functionality of the UNIC control system was tested at an engine load of 50 % and at a speed of 61 rpm. The cylinder balancing application compensates differences in the firing pressures between the single cylinders by an individual adaptation of the start of injection. As shown in figure 12 the cylinder pressures of the single cylinders were already quite balanced at the beginning of the test. Therefore it was necessary to unbalance the engine manually by adopting the start of injection offsets.
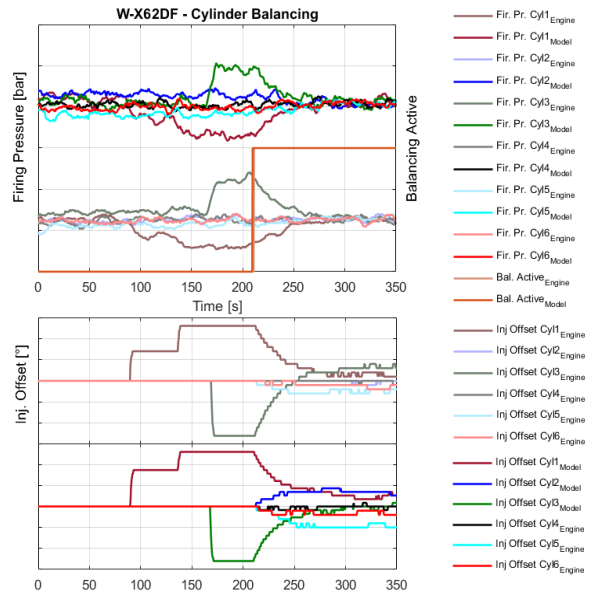

Figure 12: Validation test of the cylinder balancing functionality for a W-X62DF engine

In the graph for the firing pressure values it is apparent that the firing pressures in the engine is slightly higher than on the real engine. This is caused by the simplifications done for the pressure and load computation in the engine model. Nevertheless, it is also visible that the relative increase and decrease of the firing pressures related to the change of the start of injection are the same in the engine model as on the real engine. As shown in the chart the characteristics of the firing pressure with activated balancing functionality matches the reference measurement on the real engine very well.

CONCLUSIONS

With the launch of electronically controlled marine diesel 2-stroke engines, the requirements for the ECS were constantly rising. With the associated increasing complexity the need for a better testing environment was given. Therefore, the development of capable simulators was driven by WinGD and resulted in the dynamic engine model described here. The previous generation of simulators used for hardware-in-the-loop (HIL) testing were steady state simulators with static feedback signals.

With the previous simulator it was not possible to test closed loop functionalities of the ECS, because it was necessary to set every single feedback signal manually. For this reason it was not possible to test the engine behaviour in transient states; only a verification of isolated signals related to their plausibility was feasible. By simulating the complete plant side, it is now possible to test several closed

loop systems in a reliable way. This is done for the speed and load computation, for fuel- and pilot fuel pressure computation and for the cylinder pressure computation. With this method it is now directly possible to see the effects related to changes done in the control functionality in steady state operation points as well as in transient engine states. This allows more realistic testing, because it reveals unintentional side effects on other systems or interactions between different closed loop controllers.

Because of the simple feedback signals provided by old simulators, it used to be very difficult to check the results computed by the digital signal processor of the ECS. In the engine model presented here the shape of the feedback signals are configurable online for different applications like exhaust valves, cylinder lubrication or gas admission valves. This enables a very easy and reliable testing of the monitoring and dead time compensation functionality implemented in the UNIC system.

Another problem of the previous simulator generation was the low flexibility in the usage. The wiring and corresponding feedback generation was designed for only one engine type. Requests to test a different engine type required several modifications related to wiring and sensor characteristics. With the new engine model the majority of the engine parameters are automatically read from the configuration of the test software. This is related to sensor scaling, pin partition, actuator configuration (bi- or monostable) and engine rating. This provides the possibility to test, for example, a software of a 12-cylinder diesel engine directly after the software of a 5-cylinder dual fuel engine without any rewiring or manual reconfiguration.

It was proven in several different tests that the accuracy of the engine model is sufficient for the purpose of software validation tests. Even more, it was visible that the dynamic behaviour for the speed and for the fuel pressure values match the real engine in a very accurate way for steady state operation points as well as for transient states. In the cylinder balancing test higher cylinder pressure values of the engine model in relation to the real engine were observed. These differences can be ascribed to the simplifications done for the pressure and torque computation. It has to be investigated, if this error can minimized by considering the heat exchange over the cylinder wall and the blow by losses.

DISCUSSION

For the further development of the dynamic engine model for HIL tests of the control software it is planned to expand the supported closed loop simulation to the gas fuel system, too. Additionally, there is the intention to add a simplified feedback generation for the exhaust gas temperatures, scavenge air pressure feedback and the Gas admission valve sealing oil pressure. Furthermore, WinGD aims to take the next step towards automatic testing, where several standardised tests will be performed and evaluated by the NI-system. Therefore, it will be necessary to simulate a part of the Remote Control System logic in the NI-environment to generate automatic command to the ECS. The dynamic engine model provides the biggest precondition for this next big step towards this target.

NOMENCLATURE

Symbols

A	Area	m^2
c	Hydraulic capacity	$\dfrac{m^3}{bar}$
c	Heat capacity	$\dfrac{J}{mol\,K}$
c	Velocity	$\dfrac{m}{s}$
d	Diameter	m
H	Heating value	$\dfrac{MJ}{kg}$
h	Specific enthalpy	$\dfrac{J}{kg}$
J	Inertia	$kg\,m^2$
k	Constant of the propeller law	
l	Connection rod length	m
M	Torque	Nm
$\dot{m}$	Mass flow	$\dfrac{kg}{s}$
n	Engine speed	rpm
n	Cylinder number	
$\dot{n}$	Engine acceleration	$\dfrac{rpm}{s}$
p	Pressure	bar
$\dot{p}$	Pressure derivation	$\dfrac{bar}{s}$
$\dot{Q}$	Heat derivation	$\dfrac{J}{s}$
R	Gas constant	$\dfrac{J}{kg\,K}$

r	Crank shaft radius	m
$\dot{T}$	Temperature derivation	$\dfrac{K}{s}$
t	Time	s
$\dot{U}$	Internal energy derivation	$\dfrac{J}{s}$
u	Specific internal energy	$\dfrac{J}{kg}$
$\dot{W}$	Work derivation	$\dfrac{J}{s}$
$\dot{V}$	Volume derivation	$\dfrac{m^3}{s}$
v	Specific volume	$\dfrac{m^3}{kg}$

Greek Letters

κ	Bulk Modulus of the fluid	bar
π	Pi: 3.14159	
φ	Crank angle	rad
ω	Angular velocity	$\dfrac{rad}{s}$
$\dot{\omega}$	Angular acceleration	$\dfrac{rad}{s^2}$

Subscripts

Cyl	Cylinder
cp	Current pressure
$Fric$	Friction
h	Hydraulic
In	Inertia
i	Lat. inferior (lower)
in	Inlet
ind	Induced
ini	Initial
out	Outlet
$Scav$	Scavenge air receiver
tot	Total
V	Volume
v	Isochoric

Abbreviations

CA	Crank angle
CCM	Cylinder control module
ECS	Engine control system
FCV	Flow control valve
GVU	Gas valve unit
HIL	Hardware-in-the-loop
NI	National Instruments
PWM	Pulse width modulation
RCS	Remote control system
UNIC	Control system of Wärtsilä
WinGD	Winterthur Gas und Diesel

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