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Hardware-in-Loop Simulation Technology of High-Pressure Common-Rail Electronic Control System for Low-Speed Marine Diesel Engine

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Additional information is available at the end of the chapter

1. Introduction

1.1. Energy crisis and emission regulations

Oil price is increasing rapidly due to the oil reserve limited as a non-renewable resource. The proportion of the fuel cost rises in the total operating costs of ocean transportation companies. Compared with those from vehicles, the emissions of NO_x and SO_x from marine diesel engines are more serious, but the ones of CO_2, CO and HC are lower relatively. According to the statistics from International Maritime Organization (IMO), tens of millions tons of NO_x and SO_x are expanded into the atmosphere annually. An international protection regulation for the prevention and the control of marine pollution from ships— *Amendments to MARPOL Annex VI*—is adopted at Maritime Environment Protection Committee (MEPC) 58th meeting in October 2008. The detailed NO_x emission targets of marine diesel engines are shown in Table 1. At present the techniques of energy conservation and emission reduction are the future technical developing direction of marine diesel engines and the research highlights for producers and institutes.

Rated revolution n (r/min)	n<130	1≤n≤2000	n>2000
Tier I (2000) g/kW*h	17.0	45*n-0.20	9.84
Tier II (2011)g/kW*h	14.36	44*n-0.23	7.66
Tier III (2016)g/kW*h	14.36	44*n-0.23	7.66
Tier III (2016)g/kW*h(ECA)	3.40	9*n-0.20	1.97

Table 1. IMO NO_x emission standard

The primary technical methods to improve performance and to reduce emissions for marine diesel engine are shown below:

1. The density of intake air is increased through the turbocharging technology.
2. To raise the volumetric efficiency of the cylinder, the advanced techniques such as variable exhaust valve timing are used for improving the performance of intake and exhaust.
3. High pressure fuel injection is used to optimize fuel atomization and mix-ability of fuel and fresh air.
4. Fuel injection, mix and combustion are improved under variable working conditions by using suitable injection timing (VIT) based on fuel injection control system.

High-pressure common-rail (HPCR) electronic control technology is one of the most effective techniques to increase heat efficiency and to reduce emissions and noise for diesel engine. The fluctuation of the fuel injection pressure is not associated directly with diesel engine's speeds and loads. It means that the fuel injection pressure is independent in different conditions, especially the low-load condition. The performance parameters of the diesel engines, including excess air coefficient, combustion starting pressure, fuel atomization, are optimized through the flexible control of the fuel injection quantity, the injection plus width, the injection timing, the variable exhaust valve timing and etc. The fuel consumption and NO_X, PM emissions can be decreased substantially with the techniques implementation.

1.2. Hardware-in-loop simulation technology

Hardware-in-loop (HIL) simulation system is a close-loop dynamic testing system, composed of mathematic models and real-world physical parts. The mathematic models are established to simulate some other real-world physical parts which are non-existence or non-available for experiments in the system. HIL simulation technique is contributed to testifying the validity and feasibility of a designed proposal about the control system due to its flexible configuration. While some system settings change, the performance variation of the object measured can be observed simultaneously. Recently HIL simulation technique is widely used in developing the electronic control system of diesel engine, and it is acting as an important role during developing process. The benefits of HIL simulation system are shown in the following aspects:

1. A comprehensive emulation experimental environment is provided for the hardware and the control strategy certification. In addition, the experimental data can be as the evaluation index for the hardware being selected and matched.
2. The different working conditions of the engine can be simulated for testing software functions and control strategies. And a HIL simulation system can be applied to different projects due to its repeatability. The experimental cost on a HIL simulation system is reduced compared with that on a real diesel engine bench.
3. The initial calibration of electronic control system can be realized. The simulated engine model can be used to match and to demarcate basic control parameters to reduce developing period.
4. HIL simulation system is available for mass tests of system reliability.

5. When a electronic control system is tested by a HIL simulation system, it is easy to
    control and to record the testing process, and it is convenient to analyze and optimize
    the objects via calculating results from the models.

Normally, there are mainly four types (Figure 1) of HIL simulation system for different uses.

1. The first type HIL is mainly used in ECU developing and testing. The engine and
    mechanism are simulation models, but the ECU is real. This type
2. The second type HIL is mainly used in mechanism testing. The engine and ECU are
    simulation models, but the mechanism is real.
3. The third type HIL is mainly used in testing the combination of ECU and mechanism.
    The engine is simulation model, but the ECU and mechanism are real.
4. The fourth type HIL is mainly used in ECU developing and testing. The engine and
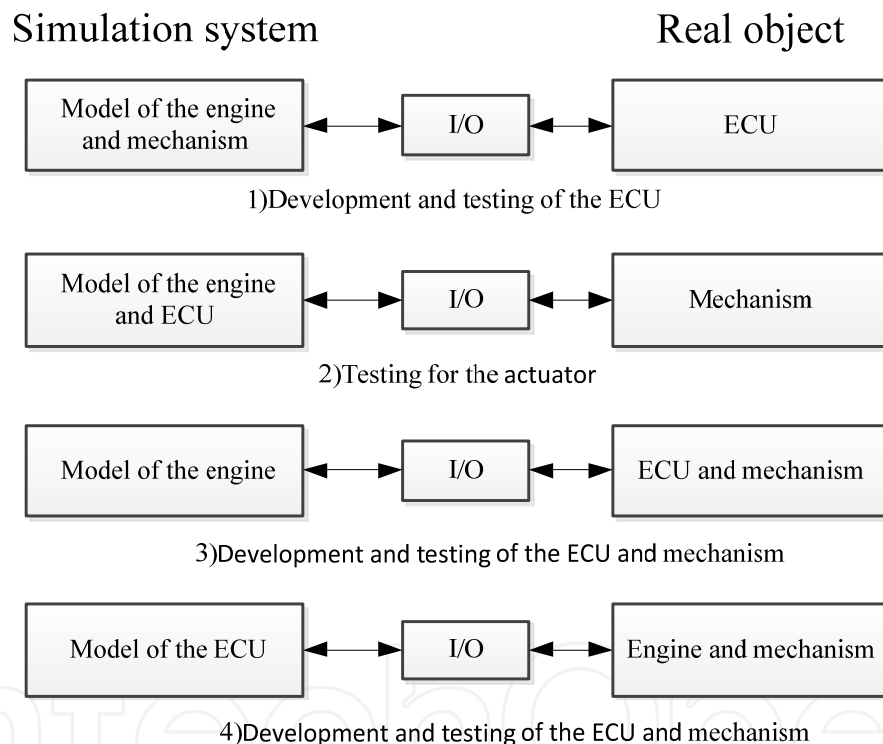   mechanism are simulation models, but the ECU is real.

Simulation system   Real object

| Model of the engine and mechanism | ◄► | I/O | ◄► | ECU |

1)Development and testing of the ECU

| Model of the engine and ECU | ◄► | I/O | ◄► | Mechanism |

2)Testing for the actuator

| Model of the engine | ◄► | I/O | ◄► | ECU and mechanism |

3)Development and testing of the ECU and mechanism

| Model of the ECU | ◄► | I/O | ◄► | Engine and mechanism |

4)Development and testing of the ECU and mechanism

Figure 1. Application type of the simulation technique

The first type HIL cannot involve the entity of the engine and the mechanism, and the
hardware costs of HIL are low relatively, besides the modeling methods are mature
relatively, so the real time simulation technique is developing rapidly and widely used. The
main corporations of real-time simulation system platform are dSPACE (Germany) , ADI
(U.S.A) and NI (National Instrument, U.S.A) platform. Since 2005, NI corporation (National
Instrument, U.S.A) has introduced PXI (PCI eXtensions for instrumentation) hardware
platform with the RTOS (Real Time Operation System), which have combined its abundant
I/O hardware resources, and have provided more choices to realize HIL real-time simulation
system. There are many investigations on the first type HIL. F. R. Palomo Pinto realized HIL

simulation of a fuel system using two PXIs with RTOS[1]. R. Isermann and J. Schaffnit realized HIL simulation system for ECU design and test using dSPACE[2]. Zhang Jie realized HIL simulation of high pressure common rail diesel engine on a common PC, using Linux RTAI[3]. The satisfactory effects have been obtained. The real-time feature of the model is required strictly in the first type HIL system.

As for the second type HIL, Bosch (Germany) in cooperation with EFS (Germany) developed a system testing platform, which has become more mature and widely used in recent years. According to the test requirements, users can set ECU model control parameters on the testing platform, such as the common rail pressure, the rail pressure control parameter, the pulse width of the fuel injection, the injection rate and so on. The research institutions and universities also successfully developed many testing platforms according to their research content and object. Wang Zhigang developed a middle pressure common rail diesel engine system testing platform[4]. Catania A. E. and Ferrari A. replaced the common rail ripe of Moehwald-Bosch testing platform by a high-pressure oil pipeline of smaller diameter. And the effect of the change to the fuel injection process is also accounted [5]. Shanghai Jiao Tong University and Shanghai Marine Diesel Engine Institute cooperated to develop the high pressure common rail fuel injection system reliability test platform[6]. The second type HIL has little demand of the engine models. ECU can only realize simple pressure control and electromagnetic valve control pulse output. Since there is no MAP of the fuel injection pressure and the fuel charge matching the engine load, it is impossible to combine the performance of the high pressure common rail system and the diesel engine load to research deeply on.

As for the third type HIL, Song Enzhe developed a semi-physical simulation platform according to type 16V396TE94 marine diesel engine and electronic control system[7]. The results of the experiments show that semi-physical simulation platform can be used for testing the diesel engine electronic control system in laboratory conditions, and the development cycle time can be shorten, and test costs can be reduced. Ou Dasheng designed a high-pressure common rail system test platform and carried out a series of tests on the platform[8], which includes the response characteristics of a high-speed solenoid valve, the hydraulic response of a electronically controlled fuel injector, the atomization characteristics of a high-pressure common rail system, the control of a common rail chamber pressure volatility, the common rail high pressure pump seal and seal ability, the solenoid valve fuel injector fuel injection characteristics and ECU performance tests. A higher reliability of the engine model is required in this kind of application.

The fourth type HIL is very close to a complete and original diesel engine. Shanghai Jiao Tong University designed a electronically controlled high pressure common rail diesel engine test bench system basing on GD-1 HPCR diesel engine, using the designed system to match and to calibrate the electronic control system of HPCR[9].

In summary, almost all the test benches of HPCR electronic control system are suitable for high and middle speed diesel engines, but there are rare reports and papers to introduce the test bench for the low-speed marine diesel engine.

2. HPCR electronic control system of low-speed marine diesel engine

2.1. Introduction

Nowadays the low-speed marine diesel engines with HPCR have successfully been put into commercial operation. These marine diesel engines with HPCR mainly include Wärtsilä RT-flex (Flex control) type marine diesel engine, MAN ME (Electronic control) type marine diesel engine and Japan Mitsubishi UEC Eco (emission control) type marine diesel engine. The first two corporations have accounted for more than 90% marine diesel engine with HPCR market shares.

2.2. RT-flex type marine diesel engine

The system architecture is shown in Figure 2. Comparing with a traditional diesel engine, RT-Flex marine diesel engine removed some components, such as exhaust valve drive devices, fuel pump, camshaft, servo motor for direction change, fuel connecting rod, starting air distributor and camshaft drive. These components are replaced with a fuel supply unit, a common-rail unit, an injection control unit, an exhaust control unit and WECS (Wärtsilä Engine Control System). WECS outputs 24V DC pulse signals to control the mechanical movement of the fuel injection and the exhaust valve behave via the high-speed solenoid valves and hydraulic forces. High-speed switching solenoid valves in the fuel injection control unit and the exhaust valve control unit are driven by the electromagnetic force, while hydraulic force is generated by high pressure servo oil in the servo oil common rail pipe. On the basis of the varied working conditions of the marine diesel engine, WECS can offer the flexible control of the injection start angle, the fuel injection pulse width and the open and the close angles of the exhaust valve, besides the mechanical delay is also fully considered. Multi-injection system (multi-jet system) is employed with the assembly of many injectors per cylinder. Three injectors were mounted with each cylinder in bore 600 mm and above of RT-flex marine diesel engine, but two injectors mounted with each
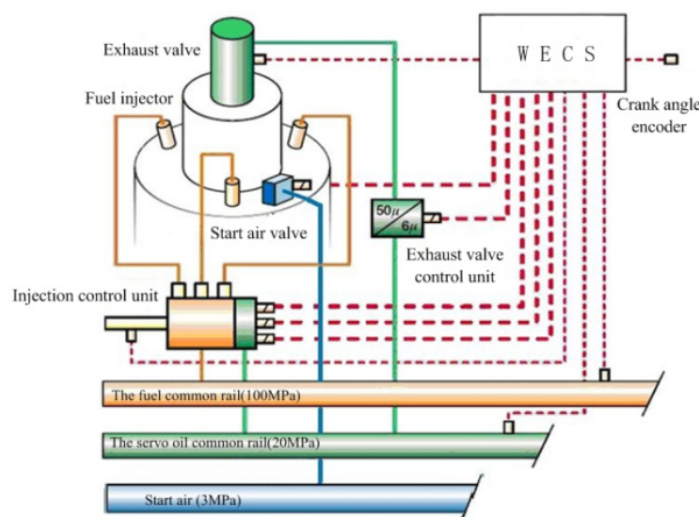

Figure 2. HPCR electronic control system of RT-flex

cylinder in bore 600 mm and below of RT-flex marine diesel engine. When the marine diesel engine starts, the start air valve on the upper cylinder is open and piston drives crankshaft rotating with the start air (3 MPa) via the 24V DC pulse control signals generated by WECS.

2.2.1. Oil supply unit

As is shown in Figure 3, the structure integrated is adopted in HPCR oil supply unit of RT-flex series marine diesel engine, with the features of a small footprint and easier installation and maintenance. The fuel pump, the servo oil pump, and the respective conduction-powered three-phase drive camshaft and the driving shaft are combined into a compact independent structure. The driving force for the oil supply unit is provided by gears set at the free end of the crankshaft. There is a fixed ratio between the speed of the camshaft, the crankshaft and the marine diesel engine crank (Speed of crank: Speed of Camshaft: Speed of the crankshaft=1:2.5:11.9).
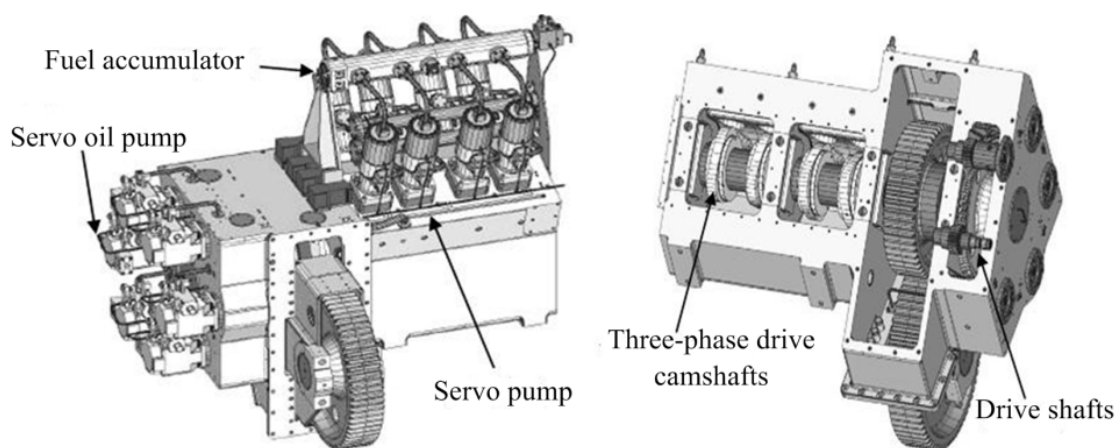

Figure 3. Oil supply unit of RT-flex

2.2.2. Common rail unit

As is shown in Figure 4, the common rail unit includes the fuel common rail pipe and the servo oil common rail pipe. The fuel common rail of 5RT-flex60C can afford the 100MPa pressure with the 5616 mm length and the peanut inwall. The operating pressure of the servo oil is the 20MPa with 5287 mm length and the round inwall. The fuel injection control unit is directly attached to the common rail fuel pipe, and the exhaust control unit is directly attached to the servo oil common rail. According to the fuel pressure signals of two pressure sensors set at the end of fuel common rail, the fuel pressure control signals are sent out to adjust the rotation angle of the fuel pump plunger cylinder from WECS, to accomplish a closed-loop control of the fuel common rail pressure. Similarly, a closed-loop control of the servo oil common rail pressure is still under WECS. Servo oil pressure signals are sent out by two pressure sensors set at the end of the servo oil common rail, and WECS adjust the servo pump swash plate tilt angle thus changing the fuel supply of the servo pump and accomplish a loop-locked control of the servo oil common rail pressure.
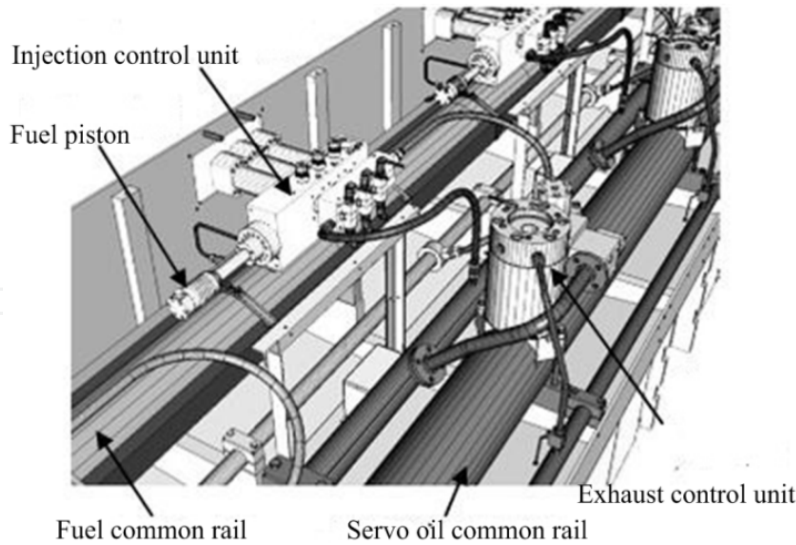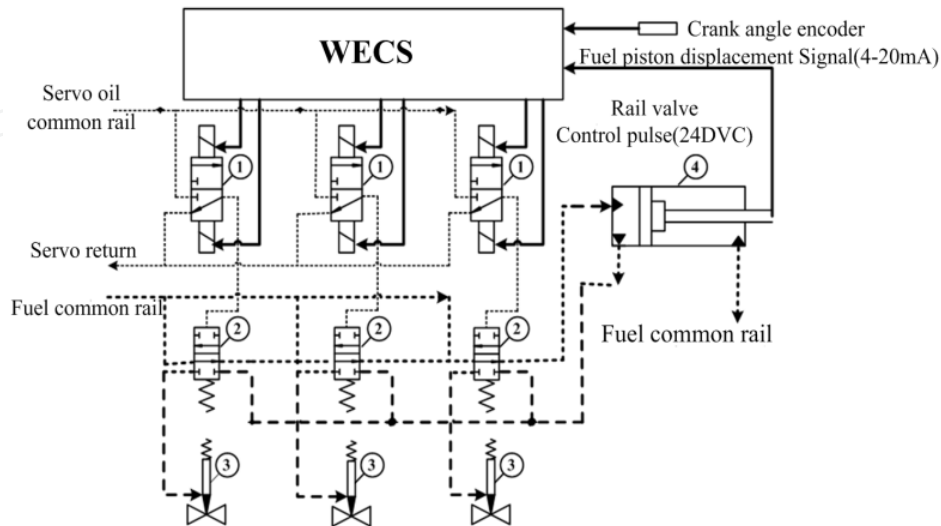
Figure 4. Common rail unit of RT-flex

2.2.3. Injection control unit

Figure 5 shows the injection control unit schematic structure of a single cylinder in no injection status. Injection rail valve converts to the open state under the excitation of the start pulse signals from WECS, then the high-pressure servo oil flow into the injection control valve. When the pressure produced by the servo oil is higher than the spring pre-tightening force, the state of the valve will change. The inlet oil line at the left side is closed, and the outlet oil line is opened. Because the pressure at the right side of the piston is higher than that at the left side, the piston is pushed to move to the left, meanwhile the high pressure fuel get into the injectors. The injection rail valve converts to the close state under the excitation of the close pulse signals from WECS, then the injection control valve will reset under the action of the spring. The outlet oil path at the left side of piston closes, while

① Injection rail valve ② Injection control valve ③ Injector ④ Fuel piston

Figure 5. Principle of injection control unit

the inlet oil path opens. Although the pressure on both sides of the piston is approximately same, the discrepancy of the area is existed. The piston moves right back under the pressure differential, and the fuel pressure in the injector reduces. Based on the current crank angle and the actual fuel quantity calculated from the displacement of the fuel piston, WECS triggers the fuel injection order and correct the injecting angle.

2.2.4. Exhaust control unit

Figure 6 shows the principle of exhaust valve control unit when a single-cylinder exhaust valve is closed. It is mainly consist of ① exhaust rail valve, ② exhaust control valve, ③ isolated transmission device, ④ exhaust valve and ⑤ air spring and so on. According to the current crank angle, WECS triggers the opening and closing orders of the exhaust valves. The exhaust valve lift is used to detect the open\ close process and the open\close angle of the exhaust valve.
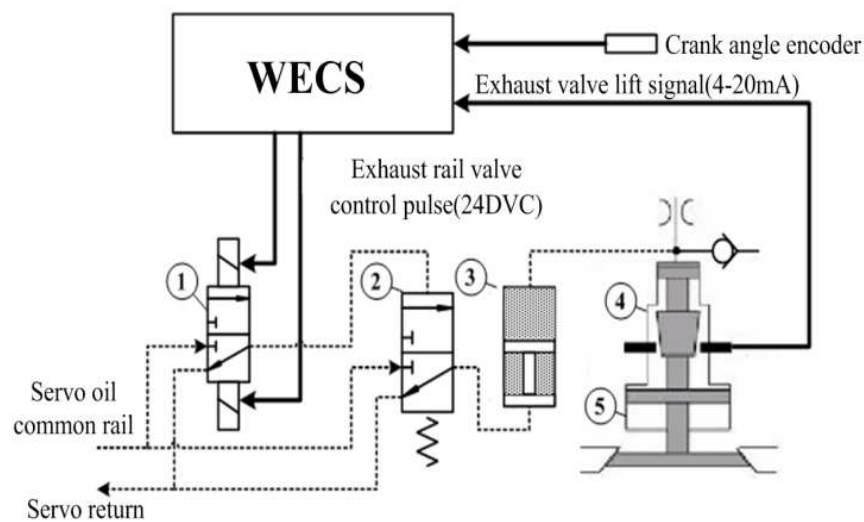

Figure 6. Principle of Exhaust Control Unit

2.2.5. ECU for RT-Flex diesel engine

WECS of RT-flex marine diesel engine includes six FCM-20 modules (Figure7). The FCM-20 module assembled on the cylinder is used to control start air valves, exhaust rail valves and fuel injection rail valves, and a single one is provided as online spare. Two FCM-20 modules of the cylinder 1 and the cylinder 2 are connected to AC20 speed control system. Three servo pumps are controlled separately by three FCM-20 modules of the cylinder 2, the cylinders 3 and the cylinders 4. And also a set of the fuel pumps (2-3per set) is controlled by two FCM-20 modules of the cylinder 3 and the cylinder 4. The functions of FCM-20 module are described in Table 2. Each FCM-20 has the same structure of hardware sub-systems. Some of them are indispensable for each cylinder, such as CYL-EU and VDM, while some of them are shared, such as COM-EU, MCM and CAN. Therefore, the functions and running parameters of each FCM-20 modules are set through the Flex-View software at the beginning configuration of WESC.
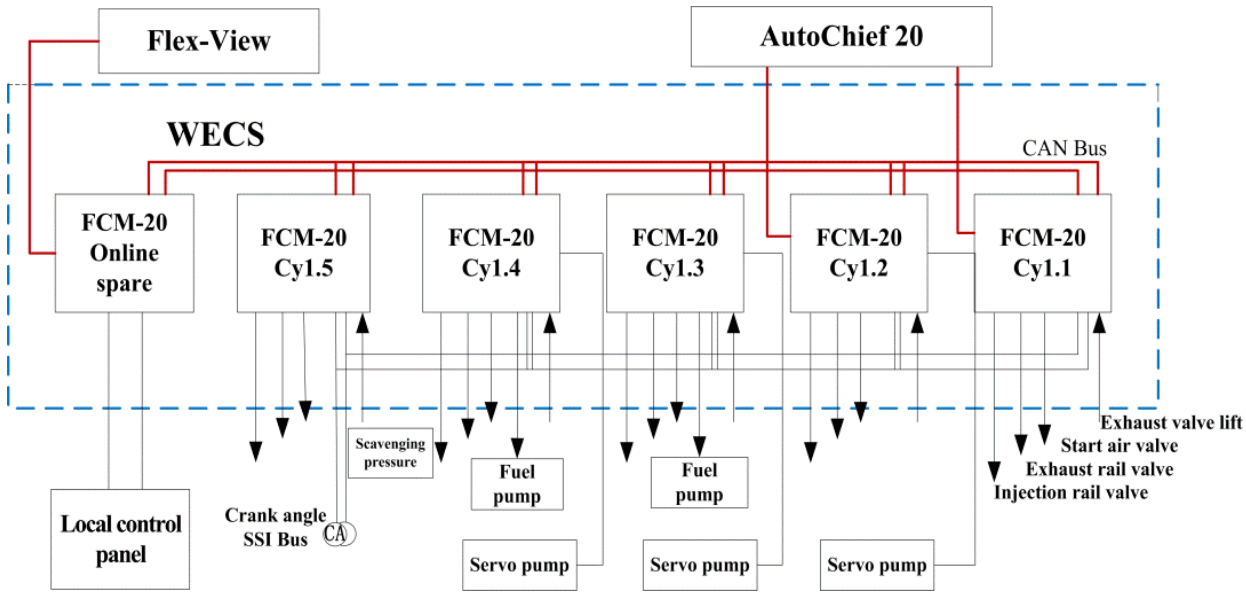
Figure 7. Structure of WECS

Number	Definition	Function
1	COM-EU	Communication and pressure control of the fuel common rail
2	ASM	Checking for the MCM
3	MCM	Communication and pressure control of the servo oil common rail
4	CAN	PWM signal output
5	CCM	Calculation for fuel quality, injection timing and exhaust valve timing
6	CYL-EU	Control for injection and exhaust valve
7	VDM	Amplifying the driven signal

Table 2. Function of the FCM-20 module

2.3. ME type marine diesel engine

ME marine diesel engine mainly consists of a hydraulic-machinery system for opening fuel injection valves and exhaust valves. The major executive components include two three-way solenoid valves, booster pumps, fuel injectors and electronic control exhaust valve, etc. As shown in Figure 8, there is only one high-pressure servo oil common rail in the HPCR system. The servo oil pump is driven by the diesel engine through the gear delivers high-pressure servo oil of 20MPa to the servo oil common rail. Then the high-pressure servo oil drives the booster pump and the exhaust valve piston through the solenoid valve. In the fuel oil system, each cylinder is assembled with a fuel oil pressure booster separately. With the help of the servo oil pressure, the fuel pressure rises from 1MPa to 75 - 120MPa. The solenoid valves control the movement of the plunger via the flow of the hydraulic oil, thus the controls of the fuel injection timing, the injection pressure and the fuel injection quantity are carried out.
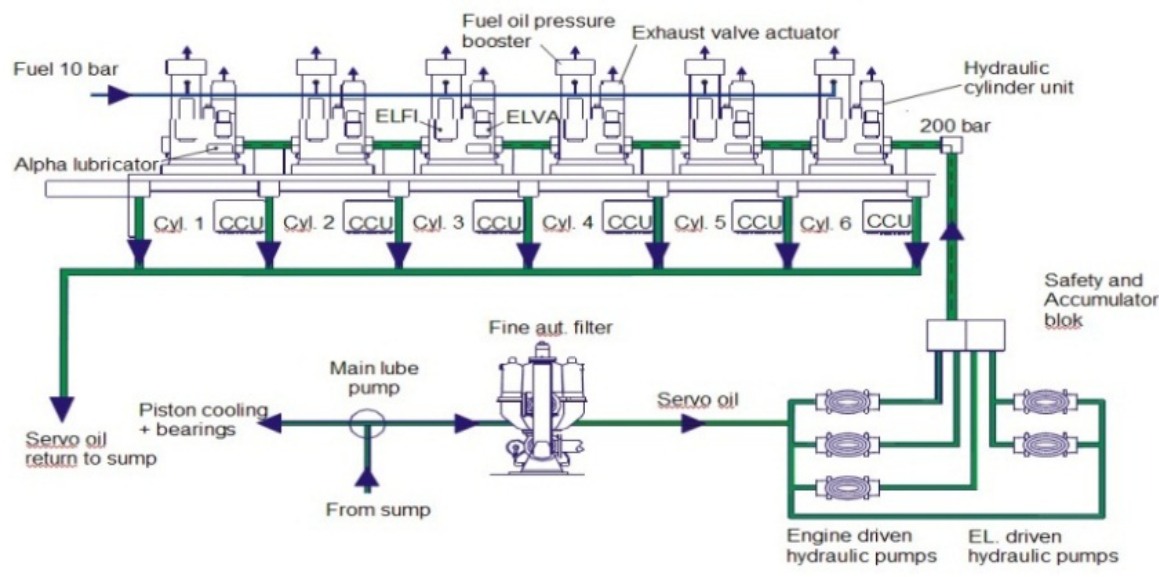
Figure 8. HPCR electronic control system of ME

2.3.1. Injection control unit

As is shown in Figure 9, the fuel injection control unit of ME-type marine diesel engine is called ELFI (Electronic Fuel Injection). The major component is a three way solenoid valve (NC Valve). The high-pressure servo oil flows into the fuel booster after the NC Valve opening. The area differential of the piston's two end caused high-pressure fuel injection. According to the fuel piston displacement, the actual fuel injection quantity is calculated and fuel injection angle is corrected in real time. The accurate control of the servo oil quantity can be achieved by the high-speed open and close action of NC valve. According to the different load conditions, it not only can achieve the change of the fuel injection timing angle and the cycle fuel injection quantity, but also can achieve fuel injection with the different fuel injection law.
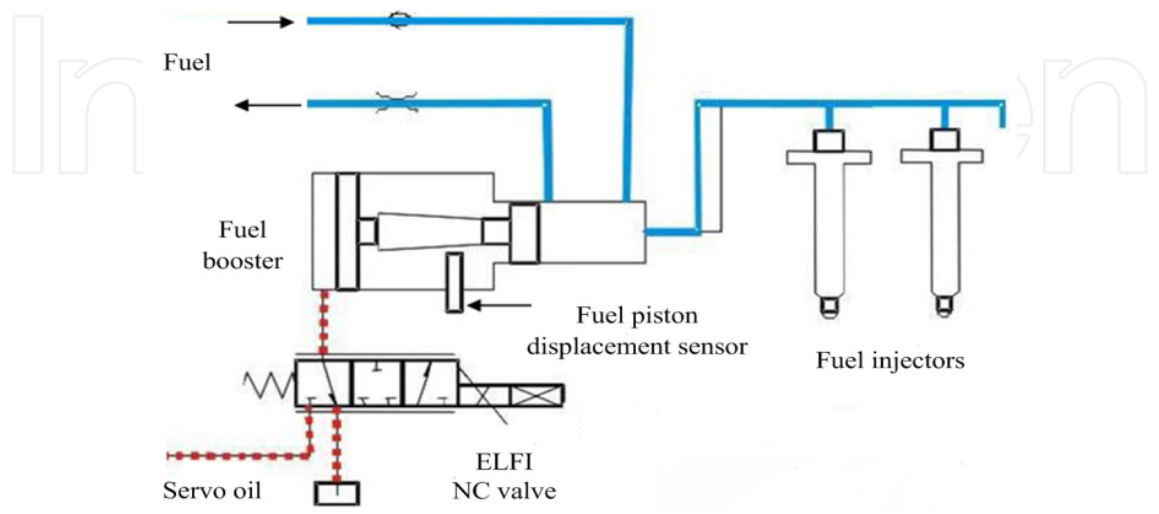

Figure 9. Injection control unit

2.3.2. ECU of ME diesel engine

As is shown in Figure 10, the electronic control system of ME diesel engine is divided into diverse function models in accordance with the different purposes.
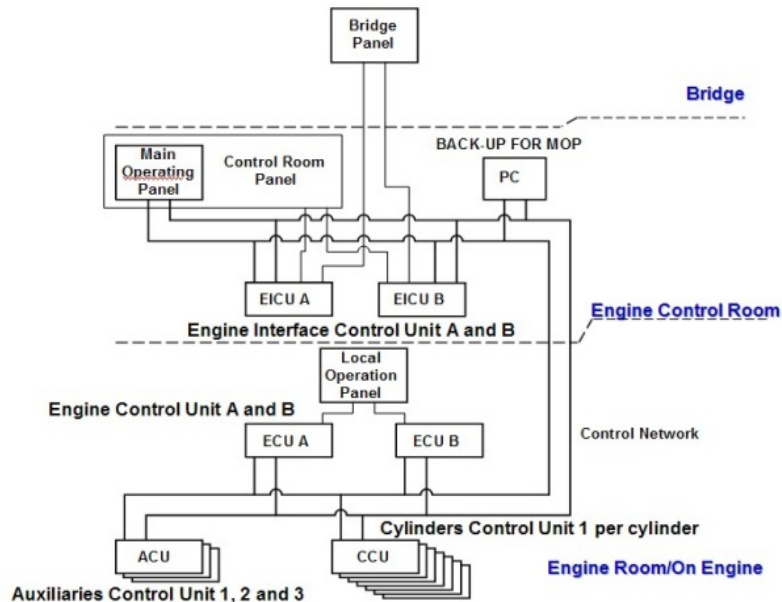

Figure 10. Structure of the electronic control unit

1. EICU (engine interface control unit) is responsible for interfacing with external system.
2. ECU (engine control unit) accomplishes the control functions of marine diesel engine, such as the speed, the operating mode and the starting sequence, etc.
3. ACU (auxiliary control unit) controls the hydraulic power supply unit of the pump and the auxiliary blower.
4. CCU (cylinder control unit) controls ELFI, ELVA (electronically controlled exhaust valve), SAV (air start valve), ALS (Alpha lubrication system) and HCU (hydraulic cylinder unit, including two parts of the fuel booster and exhaust valve actuator) of each cylinder.

Each module is installed in different parts of the marine diesel engine with the different control object and the function, as well as a set of backup module is set for the redundancy.

2.4. Comparative analysis between RT-Flex and ME

2.4.1. Similar points

In both types of the marine diesel engines, the mechanical control cams from traditional engine types are discard, which is used as the control core of the fuel injection and the exhaust valve timing, and replaced by the more precise and flexible ECU and electric-liquid-machine switching equipment[10]. The revolutionary improvement offers the low-speed marine diesel engine more flexibly features in adjusting control parameters, for instance the fuel injection timing, the fuel injection pulse width, the exhaust valve timing and etc. Air-

fuel ratio, the pressure of the combustion starting point, the fuel atomization rate can be optimized, so the reduced fuel consumption and emissions are realized under the different conditions. Especially in the low-load working condition, the influence of NO_x emission reductions is more obvious. The technique of high pressure fuel injection is adopted in the two types to improve the fuel atomization rate. The technique also guarantees the steady running of the diesel engine at the low speed and the low load. The minimum steady speed of the two types can reach 15 rounds per minute.

2.4.2. Dissimilar points

1. HPCR system of ME marine diesel engine is a pressure-charged system including the servo oil and the fuel. The fuel pressure of the common-rail is maintained at a medium level, for the pressure is set up by the booster oil pump. In the pressurized structure, the shape distortion of the common rail is small, and the sealing performance of the solenoid valve is less required, and the processing difficulty of exactitude parts is reduced. However, more fuel booster pumps are needed. The complexity and maintenance costs of the system increase.
2. HPCR system of RT-flex marine diesel engine also has two sets of oil ways. Fuel pressure is set up by oil supply units directly, so it is no necessary to boost pressure. The solenoid valves are used as the pilot valves with the lower flow rate and the response requirement. When the solenoid valves are opened, the piston is moved to make the fuel common rail and fuel atomizer connected. Because the pressure of the fuel common rail is kept at 100Mpa frequently, it is hard difficult to process the common rail.

3. Development of HIL simulation tested bench of HPCR electronic control system

3.1. Design of HIL simulation test bench of HPCR electronic control system

3.1.1. Structure of HIL simulation test bench

Various types of RT-flex series marine diesel engines are distinguished with the number of cylinders and the cylinder bore. There are not only 5 cylinders, 7 cylinders, 9 cylinders, 14 cylinders for RT-flex series marine diesel engines, but also the cylinder diameters cover 50 mm, 58 mm, 60 mm, 84 mm, 96 mm, etc. Although HPCR system structure, the length and the diameter of the common-rail pipe, the number of the fuel pumps and the servo oil pumps, and the number of injector per cylinder are slightly different for the specific type, the functions and principles are similar to each type including WECS and HPCR system, the common-rail technology with the fuel and the servo oil, the centralized supply unit for the high pressure fuel and the servo oil, the fuel injection control unit and the exhaust control unit, the system integration and the information communication technology. Those mentioned above are just the key parts that RT-flex type marine diesel engine is different with the traditional marine diesel engine. Based on the 5RT-flex60 CMKII HIL test bench is designed. The main technical parameters of the 5RT-flex60 CMKII are shown in Table 3.

Num.	Parameter	Unit	Value
1	Number of cylinder	—	5
2	Stroke	—	2
3	Bore	m	0.6
4	Pistonstroke	m	2.25
5	Rated power	kW	11800
6	Rated speed	r/min	114
7	Compression ratio	—	16.5
8	Fire order		1-4-3-2-5

Table 3. Main parameters of the 5RT-flex60CMKII

HPCR electronic control system of RT-flex series marine diesel engine is a cohesive whole according to the structure and function. The fuel supply unit, the common rail unit, the fuel injection control unit and the exhaust control unit are controlled by WECS. HIL simulation test bench is designed with integrated structure and the application type—Real-time simulation model of marine diesel engine, Control unit and executing mechanism. So the original structure characteristics and the functions of RT-flex series marine diesel engine are retained, and the system errors from the boundary conditions of the test bench are reduced. The structure integrated of HIL simulation test bench is divided into HIL simulation system and monitoring system [11], which is shown in Figure 11. Two parts are independent of each other, in addition the parameters and the data mainly are transferred via CAN bus. On HIL simulation test bench, the electronic control unit is WECS from the original machine, and the executive mechanism is properly simplified and improved based on the HPCR system of
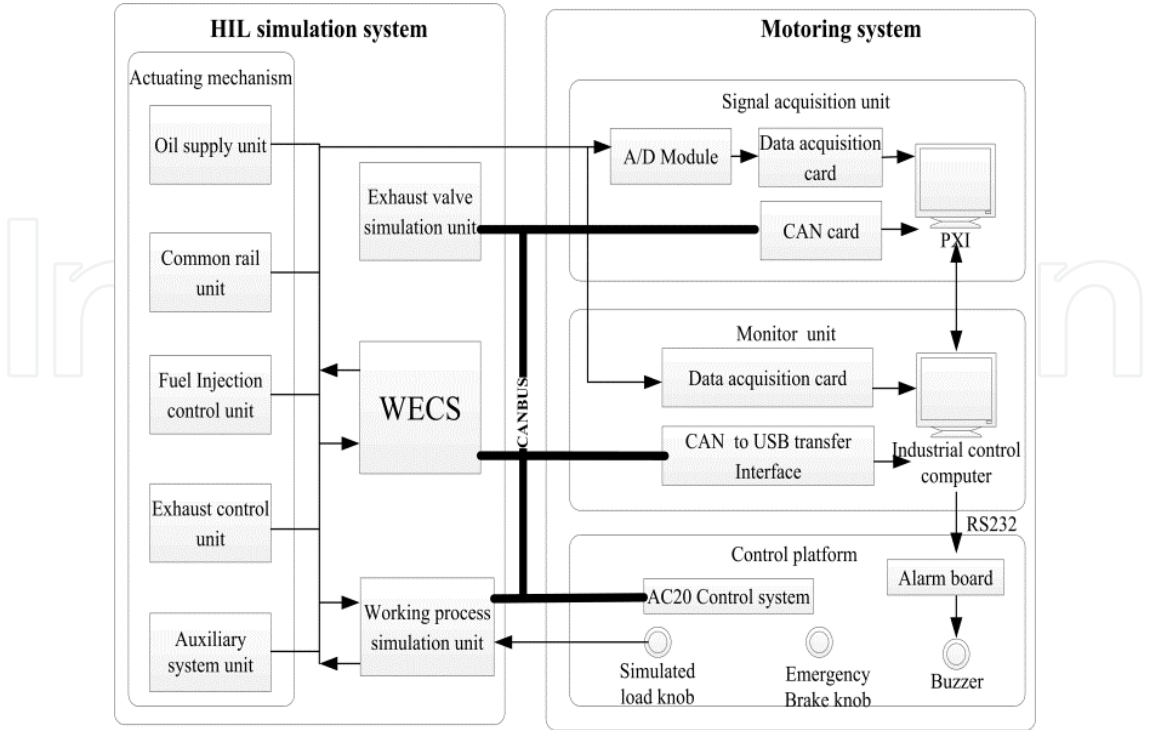

Figure 11. The structure of HIL simulation test bench

5RT-flex60C. Even more important, the real-time simulation model of the marine diesel engine which consists of the working process simulation unit and the exhaust valve simulation unit, is built up to keep the bench working well. The monitoring system of the bench is carried out for collecting, displaying and recording the test bench parameters. NI LabVIEW is served as the software development kit, besides the model and monitoring system are eployed separately in the compactRIO controller and PXI computer (from NI Corporation). The PXI is set in a console which including the Auto Chief 20 propulsion control system exporting the set point of the marine engine speed, and the load simulation device exporting physical signal for simulating the change of the engine load.

3.1.2. Functions of HIL simulation test bench

1. Performance tests of HPCR electronic control system

The different working states of the diesel engine including starting, accelerating, decelerating, mutating and steady working can be simulated throughout HIL simulation test bench. So the bench can be used for the research on the control strategies and the working characteristics of HPCR electronic control system. All parts of the bench work together harmonically. The common-rail pressure, the fuel injection and the exhaust valve movement are controlled by WECS, while the different working conditions of the diesel engine are simulated by the real-time simulation model. The signals of the crankshaft angle, the fuel pump rack position, the fuel common-rail pressure, the servo oil common-rail pressure, the fuel injection quantity, the needle valve lift, the exhaust valve lift, the control from WECS are measured by the monitoring system.

2. Fuel injection characteristic tests

The objective of the test is to conduct the dynamic working characteristics of the fuel control unit and injector. The fuel injection system is in running state independently, and some important parameter signals related to the fuel injection system, such as the crankshaft angle, the fuel common-rail pressure, the fuel injection quantity, the needle valve lift and reference injection signals from WECS are acquired and analyzed. However, the exhaust valves do not work, and the exhaust valve lift signals and various load working conditions of the diesel engine are simulated via the model.

3. Exhaust valve characteristic tests

The objective of the test is to conduct dynamic working of the exhaust control unit and the exhaust valve. The exhaust system is in running state independently, and some important parameter signals which include the crankshaft angles, the servo oil common-rail pressure, the exhaust valve lift and the reference control are acquired and analyzed. However, the fuel injection system does not work, and the various load working conditions of the diesel engine are simulated via the model.

4. Key executive mechanism tests

The key executive mechanisms of the high pressure common-rail system contain the fuel pump, the servo oil pump, the common-rail pipe, the fuel accumulator, the fuel injector, and

the exhaust valve, etc. The performance tests of the parameters comparison for the executive mechanism can be proceed by applying the replacement way, also the executive mechanism can be tested independently.

3.2. Executive mechanisms of HIL simulation test bench

The sketch map of executive mechanisms is shown in figure 12. It is no feasibility that the same executive mechanisms from the 5RT-flex60C diesel engine are used for the test bench due to the huge structure. In the premise condition that the test bench can realize the integrated original system function, the fuel supply unit and the exhaust valve unit are simplified based on HPCR system. The tubing length, the diameter, the bending angle of the connector of the executive mechanism are close to the original machine as much as possible, and the key components which need to be focused on are retained. The common and non-key components are simplified and improved. An auxiliary system unit is added for providing the boundary conditions for the test bench. The auxiliary system unit is composed of the low pressure oil supply unit, compressed air unit, crank angle unit and fuel weighing unit.
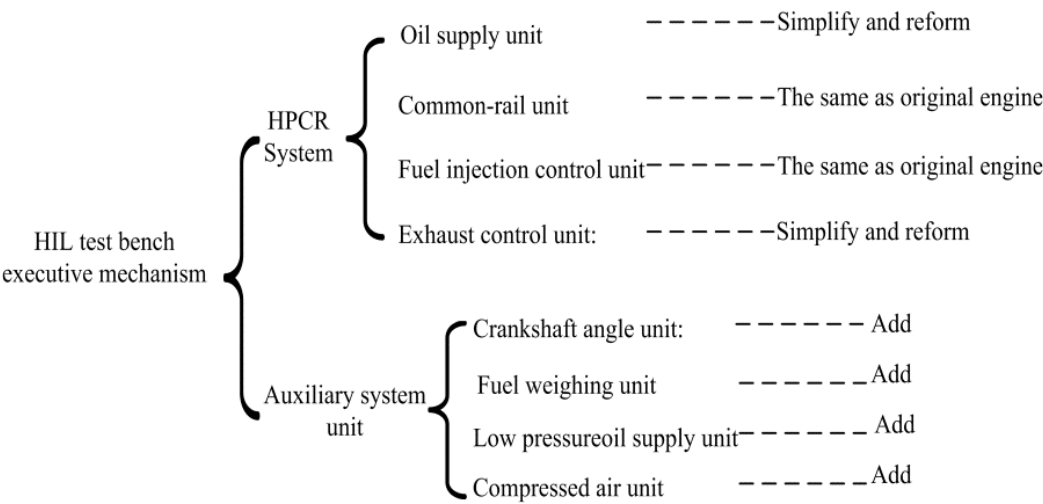

Figure 12. The sketch map of executive mechanism

1. Oil supply unit

The research on the fuel supply unit focuses on the oil supply way, the electronic control technology and the characteristics of the fuel pump and the servo oil pump, etc. The key components, which can reflect the functional structure characteristic, are kept such as the three-phase power cam and the fuel accumulator etc. To satisfy the function of HIL test bench and site layout, some simple components are replaced on the basis of the reliable equipment designed with the advanced and proven technique. The number of the fuel pump is 2, the one of which on the 5RT-flex60C diesel engine is 4. The number of the servo oil pumps is reduced from 3 to 1. Two sets of the variable-frequency adjustable-speed three-phase induction motors (be referred to as "fuel pump motor" and "servo oil pump motor") are served as the power of the oil supply unit. The speeds of motors are controlled by the real-time simulation model of the diesel engine, but the flow is controlled by WECS.

2.   Common-rail unit

The common rail unit including the fuel common rail and the servo oil common rail is conformity to that of the original machine.

3.   Fuel injection control unit

The fuel injection control unit is consistent with that of the original machine, which has one fuel injection control unit and three injectors per cylinder.

4.   Exhaust control unit

Due to the similarity of the exhaust valves movements, only one cylinder exhaust valve is retained. The exhaust control units installed on each cylinder are conformity to that of the original machine. Furthermore a 40L nitrogen bottle is used to provide the pressure for the air spring.

5.   Crankshaft angle unit

Crankshaft angle signal is provided for WECS by the crankshaft coder driven by an AC servo motor (be referred to as "crankshaft motor"). The motor speed can be controlled by the real-time simulation model of the diesel engine.

6.   Fuel weighing unit

Fuel weighing unit is two movable carriages comprised by three electronic balances and container. It is used to weighing the fuel injection quantity of three fuel injectors in the cylinder during a certain time or a certain cycle.

7.   Low pressure fuel supply unit and compressed air unit

Low pressure fuel supply unit is taken for recycling the backflow or the leakage fuel, the servo oil and the lubricating oil after multi-stage filtration. Compressed air unit provides compressed air of 0.5 ~ 0.75MPa by using Ingersoll-Rand Company UP5-15-7 type air compressor, to simulate the back gas pressure in the cylinder.

3.3. Real-time diesel engine simulation model

As the half of the working process simulation unit, the real-time diesel engine simulation model is used to simulate the diesel engine working process for coordinating synchronous operation of the test bench. The closed-loop system (Figure13) is composed by the real-time model, HPCR system, WECS and the console. The features of the real-time model are:

1.   To start three motors of the crankshaft angle, the fuel pump and the servo oil pump simultaneously through the enabling signals after detecting the control pulse signals of the starting air valves from WECS.
2.   To calculate the motors speed and scavenging pressure with the displacement signals of the fuel quality piston.
3.   To generate 4-20 mA signals to control the speed of three motors respectively.

4. To simulate the load and propulsion characteristics of the diesel engine with the load torque set on the console.
5. To generate two consistent simulated scavenging pressure signals to WECS.
6. To monitor the crankshaft angle and simulate backpressure in the exhaust valve through opening the pneumatic valve with the exhaust valve at the closing statue, and allow the compressed air go into a seal space at the bottom of exhaust valve.
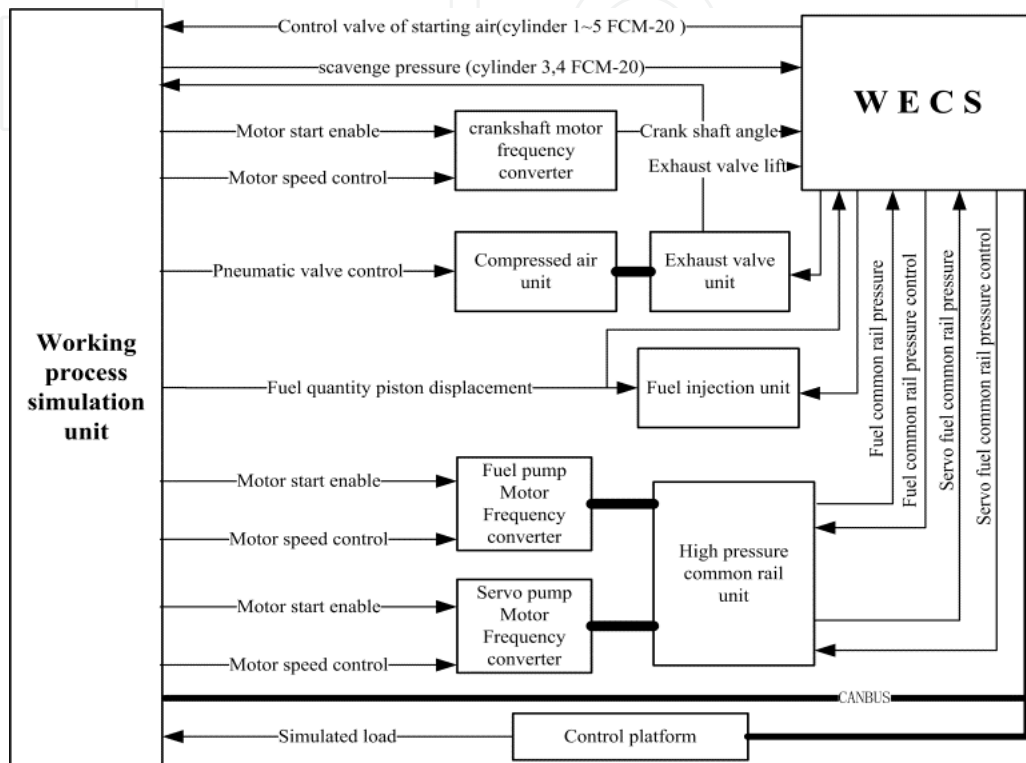

Figure 13. Schematic diagram of the working process simulation unit

Number	Crankshaft motor state	Control plus for staring air	Injecting	State
1	Stop	0V	No	Stopping
2	Stop	0V	Yes	–
3	Work	0V	Yes	Running stage
4	Work	0V	No	Stopping procedure
5	Stop	24V	No	Staring
6	Stop	24V	Yes	–
7	Work	24V	Yes	–
8	Work	24V	No	Staring procedure

Table 4. States of HIL simulation test bench

The executive mechanism status of HIL simulation test bench can be divided into eight forms shown in Table 4, according to crankshaft motor running, control pulse signal amplitude of starting air valve and the existence of injection process during a cycle of

crankshaft motor. The different algorithms are designed to deal with different possible situation. Because injection order from WECS cannot be sent out when crankshaft motor stops or the starting air valve is open, the number 2, 6 and 7 will not appear. The others are the contents focused on.

3.3.1. Staring and staring procedure

At the staring stage of RT-flex marine diesel engine, the crankshaft position is calculated and the control pulse is outputted to open the starting air valve by WECS. Then the compressed air is blown into the cylinder to make the crankshaft rotate by pushing the piston. After a period of time, if the engine speed and the fuel common-rail pressure both reach the expected values, starting is success. On the contrary it is fail. Staring process is shown in Figure 14.
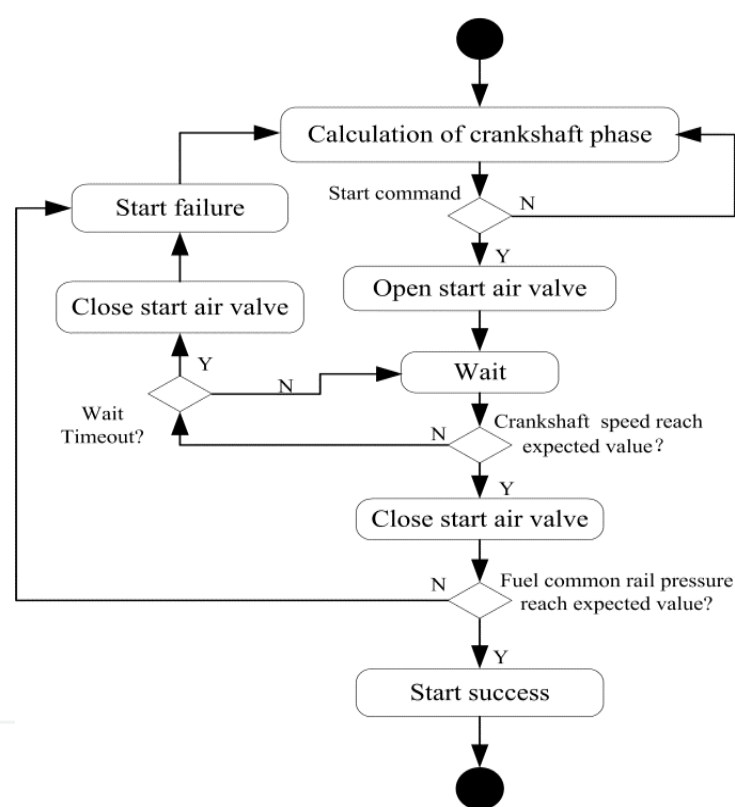

Figure 14. The flowchart of the starting

The primary control parameters in the process of diesel engine staring are shown in Table 5 and the opening angles of the starting air valve are shown in Figure 15, which come from experimental data of HIL simulation test bench.

Number	Category	Value
1	The waiting time(s)	15
2	The speed expected （r/min）	25
3	The pressure expected （MPa）	40

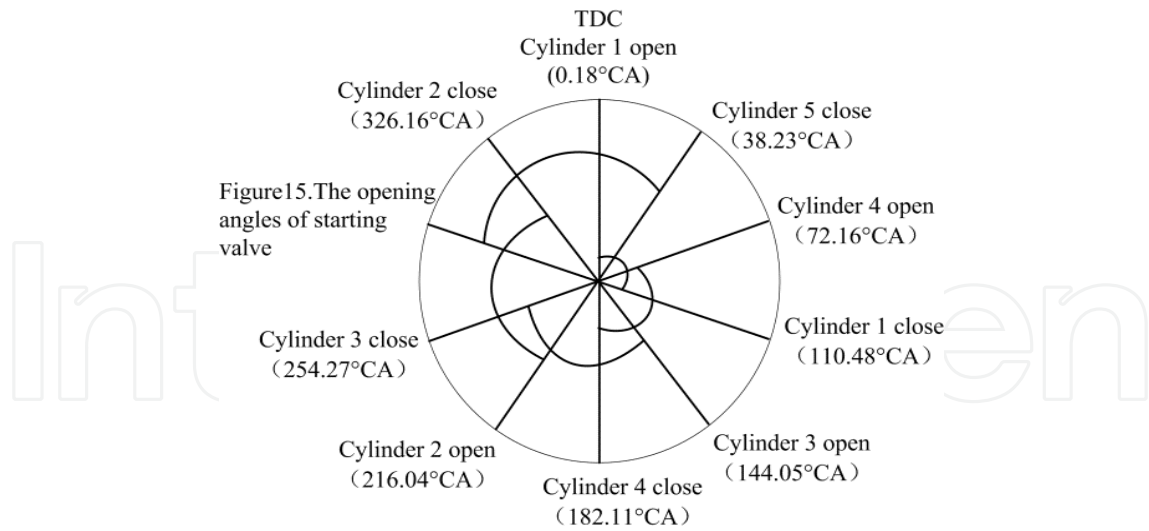**Table 5.** The primary parameters in the starting process

Figure 15. The opening angles of starting valve

As Figure16 is shown, the scavenging pressure (p_{sca}) is set 0.01MPa to simulate the real scavenging pressure of entity auxiliary fan. When the starting air valve control pulses are detected from any FCM-20 module, three motors will be started synchronously in the bound mode. However, in the non-bound mode only the crankshaft motor motor will be started.

With the curve fitting method, the crankshaft motor speed n_c *(r/min)* is regarded as the time logarithmic function according to the formula (1), and has to reach 25 *(r/min)* during 15 seconds. The current speed $n_c^{'}$ *(r/min)* is calculated by formula (2) according to the crankshaft angle difference ΔCA in 10 seconds. In the bound mode, the fuel pump motor speed n_f *(r/min)* and the servo oil pump motor speed n_s *(r/min)* are calculated respectively according to formula (3) and formula (4). Parameter a is the constant. Parameter t is the time *(ms)*. Parameter n_f is the fuel pump bound parameter referring to chapter 2. Parameter n_s is the servo oil pump bound parameter referring to chapter 2.
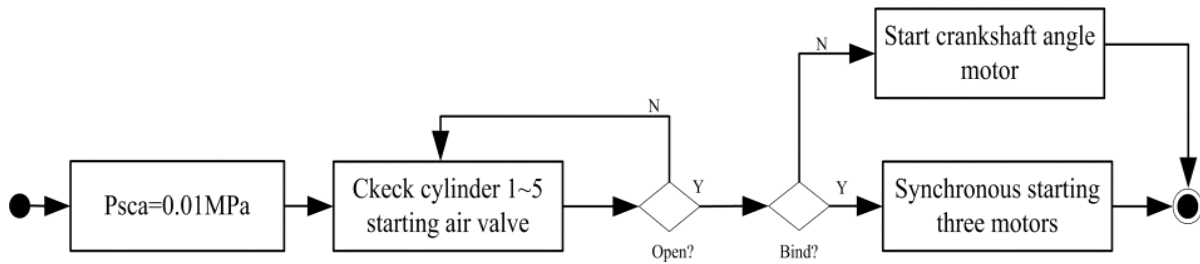

Figure 16. The control flowchart of the starting

$$n_c = \frac{\log_a(1+t)}{1000 \times 60} \tag{1}$$

$$n_c' = \frac{\Delta CA \times 100 \times 60}{360} \tag{2}$$

$$n_f = k_f \times n_c' \tag{3}$$

$$n_s = k_s \times n_c' \tag{4}$$

3.3.2. Running stage

The simulation model of 5RT-flex60C at the running stage is restricted by the following conditions and design requirements.

1. The filling and emptying method can't be applied to the mathematical model of the diesel engine, because key parameters such as the intake/exhaust pipe volume, the scavenging volume and the turbocharger characteristics can not be obtained.
2. MAP of the measured data, such as MAP of the fuel pump supply rate and the diesel engine output power from the diesel engine bench, can not be used as the input boundary the model.
3. The purpose of the model at running stage is to stability control the speed of three motors, but not to predict the cylinder pressure, the temperature and the flow changing process of the intake and exhaust pipe.
4. The model needs to be interacted with WECS and the executive mechanisms in the real time. It is possible to be interfered by the external instable factors, such as the noise signals and speed fluctuations from motors, etc.
5. The changed working conditions of the diesel engine have to be simulated by adjusting the load simulation device in the HIL simulation test bench.
6. The understanding of WECS control strategies is limited, which includes the rail pressure, the fuel injection timing and the exhaust timing.

Therefore, the objective of the simulation model is to keep the test bench safe and reliable. The simulation model at the running stage is applied with the mean value engine model (MVEM), and the ventilation process model is simplified appropriately. MVEM is commonly used for describing the diesel engine behavior, and it is based on the global energy balance of the diesel engine. The algorithm of MVEM is very simple and reliable, and its resolving cost is very low. So that it is very suitable for the real-time application.

The propulsive characteristics of the diesel engine can be used to assess its performance, so the tests of the propulsive characteristics are adopted by the manufacturer, and an "official test report" with the propulsive characteristics of the real diesel engine based on the test bench is provided to the ship-owner before delivery, which contains the mass experimental data such as the speed, the scavenging pressure, the effective fuel consumption etc. under the working condition of 25% load, 50% load, 75% load, 90% load, 100% load, 110% load. Some experimental data from the official test report are used in the simulation model.

As shown in Figure17, MVEM is improved in basis of the restrictions mentioned above. It is supposed that the exhaust loss, the heat exchange loss and the mechanism loss of 5RT-flex60C diesel engine simulated in MVEM is the same with that of 7Rt-flex60C diesel engine, which has the same cylinder diameter and the stroke of RT-flex60C diesel engine. The specific fuel oil consumption (SFOC) at the standard ambient temperature from the "Official Test Report" of 7RT-flex60C diesel engine is calculated the diesel engine effective power, and the scavenging pressure from the "Official Test Report" of 7RT-flex60C diesel engine is as the load function with the curve fitting method.
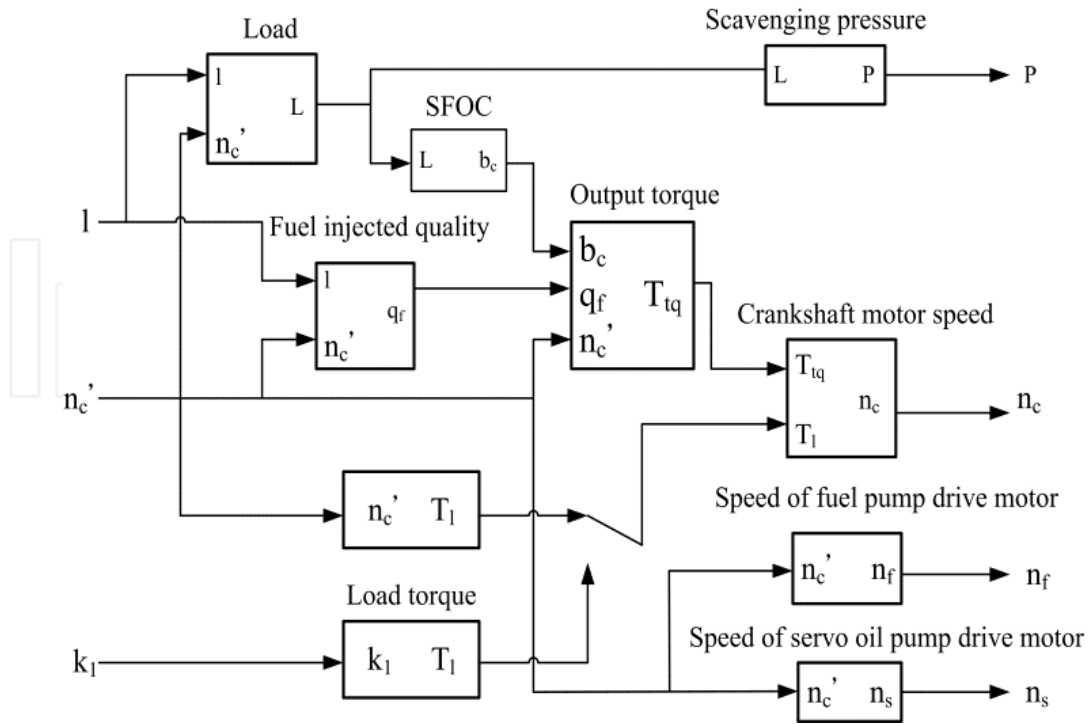
Figure 17. Schematic diagram of the simulation model at the running stage

The definition of the model boundary in this section refers to the following. The fuel quality displacement $l(mm)$ is acquired from the injection control unit. The current crankshaft motor speed n_c' (r/min) is calculated from formula (2). The load torque adjusting coefficient k_l is obtained from the load simulation device. The primary formulas in the improved model are as follows.

1. Load

$$L = \frac{n_c}{n_{MCR}} \times \frac{l}{l_{MCR}} \times 100\% \tag{5}$$

$l_{MCR}(mm)$ is the maximum displacement of fuel piston in the maximum continuous ratings (MCR) working condition. n_{MCR} (r/min) is crankshaft speed in MCR working condition.

2. Fuel injected quality per cycle

$$q_f = \begin{cases} \left(\frac{\pi \times r^2 \times l}{10^6} \times \rho - q_{leak}\right) \times N \\ \sum_{i=1}^{N} \left(\frac{\pi \times r^2 \times l_i}{10^6} \times \rho - q_{leak}\right) \end{cases} \tag{6}$$

$r(mm)$ is the radius of the fuel piston. $\rho(0.835kg/L)$ is the fuel density. N is the number of cylinder. $q_{leak}(kg)$ is the leak fuel quality per cycle, set as constant. The first equation in the equation (6) is used to calculate the fuel injected quality per cycle with the displacement of the fuel piston of any cylinder. And the second equation in the equation (6) is used to calculate the fuel injected quality per cycle with the displacement of fuel piston of N cylinders.

3. Load torque

$$T_l = \begin{cases} k_l \times T_{MCR} & k_l \in [0.2, 1.1] \\ (n_c'/n_{MCR})^2 \times T_{MCR} \end{cases} \quad (7)$$

T_{MCR} *(N*m)* is the load torque in MCR (maximum continuous rating) working condition. The first equation in the equation (7) is used in the condition of the load characteristic, and the second equation in the equation (7) is used in the condition of propulsion characteristics.

4. SFOC

$$b_e = f(n_c') \quad (8)$$

SFOC at standard ambient temperature from the "Official Test Report" of 7RT-flex60C diesel engine SFOC of the model is calculated throughout the formula (8) with the linear interpolation method.

5. Output torque

$$T_{tq} = P_e \times \frac{9550}{n_c'} \quad (9)$$

$$P_e = \frac{60 \times n_c' \times q_f}{b_e} \quad (10)$$

P_e *(kW)* is the effective power.

6. Scavenging pressure

$$p_{sca} = a_0 + a_1 L + a_2 L^2 + a_3 L^3 \quad (11)$$

a_0, a_1, a_2 and a_3 are constants.

7. Crankshaft motor speed

$$n_c = n_c' + \frac{T_{tq} - T_i}{J} \times \frac{30}{\pi} \quad (12)$$

J(kg/m) is the rotational inertia of 5RR-flex60C diesel engine.

8. Speeds of fuel pump motor and servo oil pump motor are calculated according to formula (3) and (4) respectively.

3.3.3. Stopping and stopping procedure

As Figure 18 is shown, the scavenging pressure is deceased to 0.01MPa in the stopping stage. The crankshaft motor speed is reduced by Δn_c*(r/min)* gradually. The speeds of the fuel pump motor and the servo oil pump motor are calculated according to formula (3) and (4) respectively. When the crankshaft motor speed is less than 5 *(r/min)*, all of three motors are stopped in the bound mode, but in the non-bound mode only the crankshaft motor is stopped.
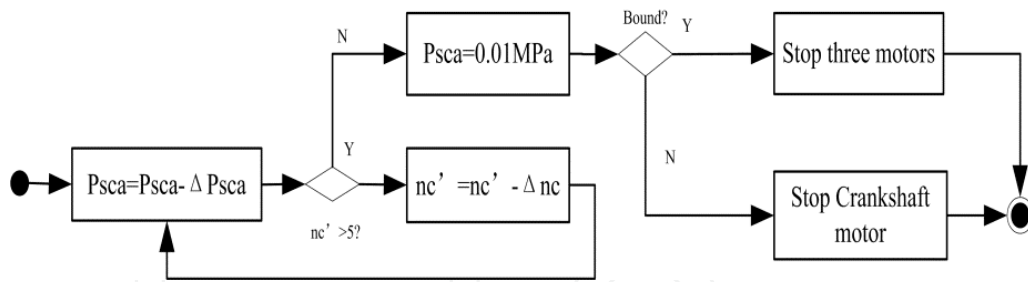
Figure 18. The control flowchart of the stopping

$$\Delta n_c = \Delta n_f / k_f \tag{13}$$

Because the fuel pump motor has the biggest inertia moment, to keep three motors actual speed rate steady in stopping stage, Δn_c is calculated from the Δn_f *(r/min)* by the formula (13). Δn_f is the different speed of the fuel pump motor in the free decelerate condition.

3.4. Exhaust valve simulation model

The exhaust valve simulation model is as the other working process simulation unit. The main functions of the exhaust valve simulation model are to provide the real-time simulated exhaust valve lift signals for WECS. The exhaust valve lift signals are triggered by the exhaust valve opening and closing pulses from FCM-20 modules. Because of the difference between HIL simulation test bench and the original machine, the exhaust valve of the test bench cannot work properly, if the load of the test bench exceeds 75%. So two approaches are designed in the exhaust valve simulation model.

1. If the load of the test bench is below 75% load, the lift exhaust valve signals send to WECS, are from the real exhaust valve of the test bench. The lift exhaust valve signals of the real exhaust valve are sampled and saved by the exhaust valve simulation model, when exhaust valve opening/closing order is triggered. Then, the signals collected are
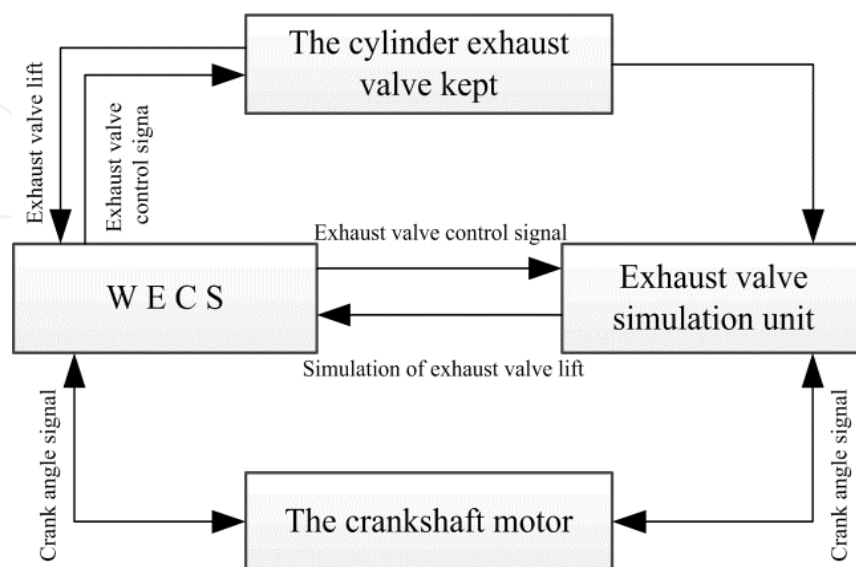

Figure 19. Schematic diagram of cylinder moving method

sent to the FCM-20 modules based on the "cylinder moving" method[13].The program flow of "cylinder moving" is shown in Figure 19. The "cylinder moving" method assumes that the working ways of all cylinders are all the same, and the working heterogeneity of the different cylinders is ignored. The working status of other cylinders is not directly calculated, but is obtained by the state recursive with the firing order. Therefore, not only the simulation speed is improved, but also the contradiction between the diesel engine's model accuracy and real-time is solved.

2. If the load of the test bench exceeds 75%, the emulator exhaust valve lift signals, simulated by the curve fitting method, are outputted by the exhaust valve simulation model. The emulator curve $l_{exh}(mm)$ is simplified for a trapezoidal , which is similar to the measured curve. The emulator curve is divided into four parts, including the closing status, the opening process, the opening status and the closing process[14]. The formulas for calculating the curve are shown below:

1. Closing status

$$l_{exh} = l_{min} \tag{14}$$

$l_{min}(mm)$is the displacement with minimum exhaust valve lift,.

2. Opening status

$$l_{exh} = l_{max} \tag{15}$$

$l_{max}(mm)$is the displacement with maximum exhaust valve lift.

3. Opening process

Simulated exhaust valve lift signals are calculated according to the formula (16), when exhaust valve opening order from FCM-20 module is delayed $t_{od}(ms)$.

$$\frac{dl_{exh}}{dt} = \begin{cases} \Delta l_{open}(l_{exh} < l_{min}) \\ 0 \qquad (l_{exh} \geq l_{min}) \end{cases} \tag{16}$$

$l_{open}(mm/ms)$is the exhaust valve opening rate.

4. Closing process

Simulated exhaust valve lift signals are calculated according to formula (17), when exhaust valve closing order from FCM-20 module is delayed $t_{cd}(ms)$.

$$\frac{dl_{exh}}{dt} = \begin{cases} -\Delta l_{close}(l_{exh} > l_{min}) \\ 0 \qquad (l_{exh} \leq l_{min}) \end{cases} \tag{17}$$

$l_{close}(mm/ms)$is the exhaust valve closing rate.

According to test analysis of exhaust valve, the initial values of model parameters are shown in Table 6.

Number	Parameter	Unit	Initial value	Number	Parameter	Unit	Initial value
1	l_{min}	mm	0	4	l_{max}	mm	73
2	t_{od}	ms	10	5	t_{cd}	ms	40
3	Δl_{open}	mm/ms	1.825	6	Δl_{close}	mm/ms	0.608

Table 6. Initial values of model parameters

3.5. Test verification

3.5.1. Starting process

HIL simulation test bench is started in the bound mode, and the start-up command comes from the Auto chief 20 system with the 51*(r/min)* setting speed. Figure 20 is shown that three motors start simultaneously and reach the setting speed at the same time in about 20 seconds. During the starting process, the three motors' speeds increase smoothly with the setting scale factors and achieve the desired objective.
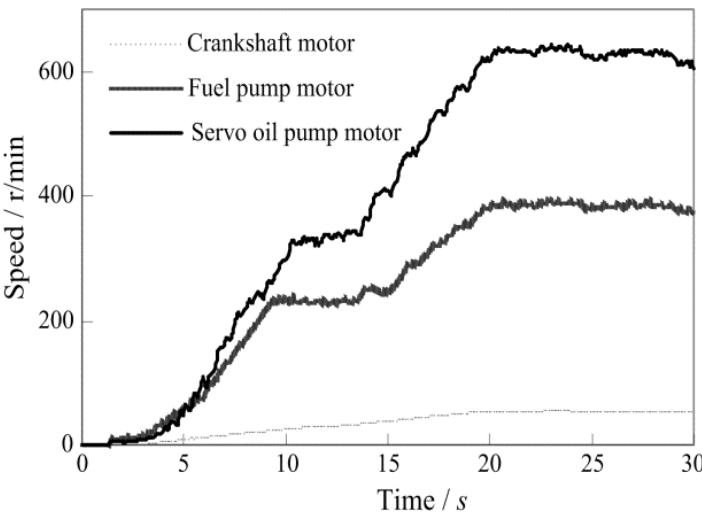

Figure 20. The motors speed in HIL simulation bench

3.5.2. Propulsion characteristic tests

The experimental data on HIL simulation test bench, including the crankshaft speed, the scavenging air pressure, the fuel consumption, the fuel indicator, the fuel rail pressure, the servo fuel rail pressure, the exhaust valve opening and closing angle, is compared with the values from the "Official Test Report" under the typical working conditions in propulsion characteristics test. The recorded curves of the crankshaft motor speeds in 5 seconds are shown in Figure 21(a). The speeds are relatively stable with less fluctuation. The speeds contrasts with the test bench and the report are shown in Figure 21(b). The relative errors of the speed from different data source are less than 1%.
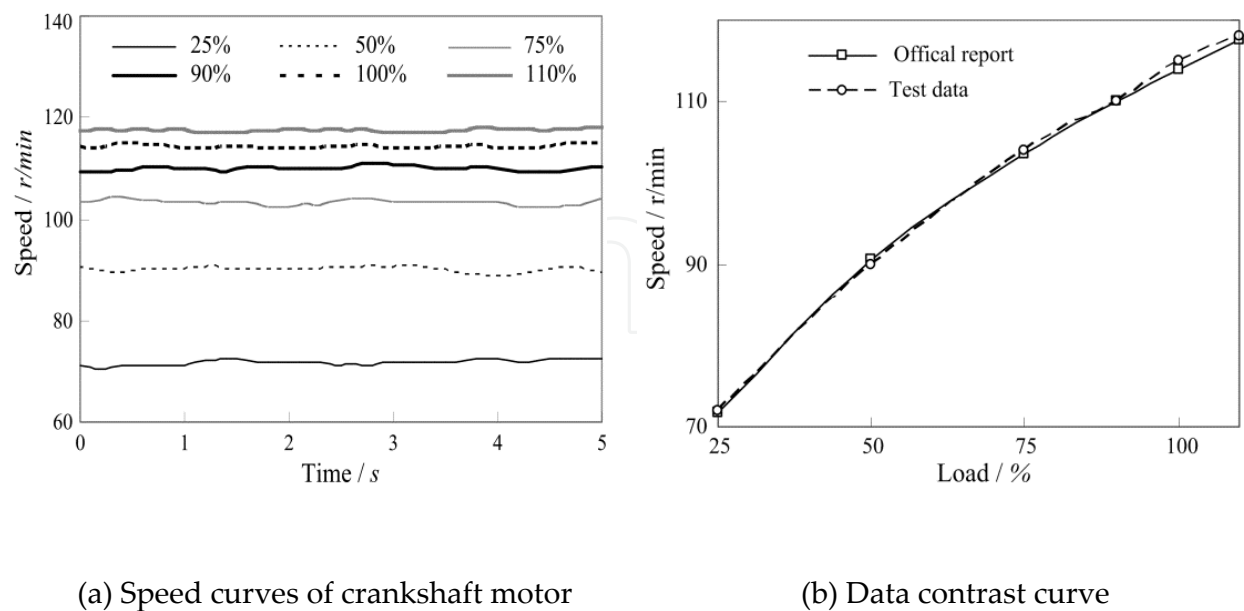
(a) Speed curves of crankshaft motor   (b) Data contrast curve

Figure 21. Speed curves of crankshaft motor in HIL simulation bench

Figure 22 is shown the scavenging air pressure. The relative errors between the test bench and report are less than 1.5%.
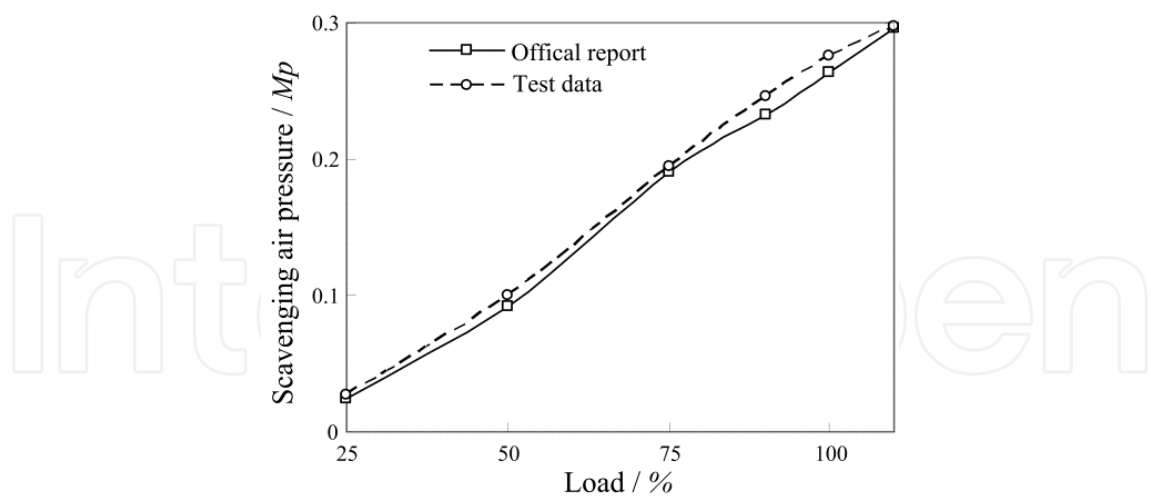

Figure 22. Data contrast curve of the scavenging air pressure

Figure 23 is shown the contrasts of the fuel consumption and fuel indicator. The relative errors of the fuel consumption are less than 1%, and the relative errors of the fuel indicator are less than 4%.
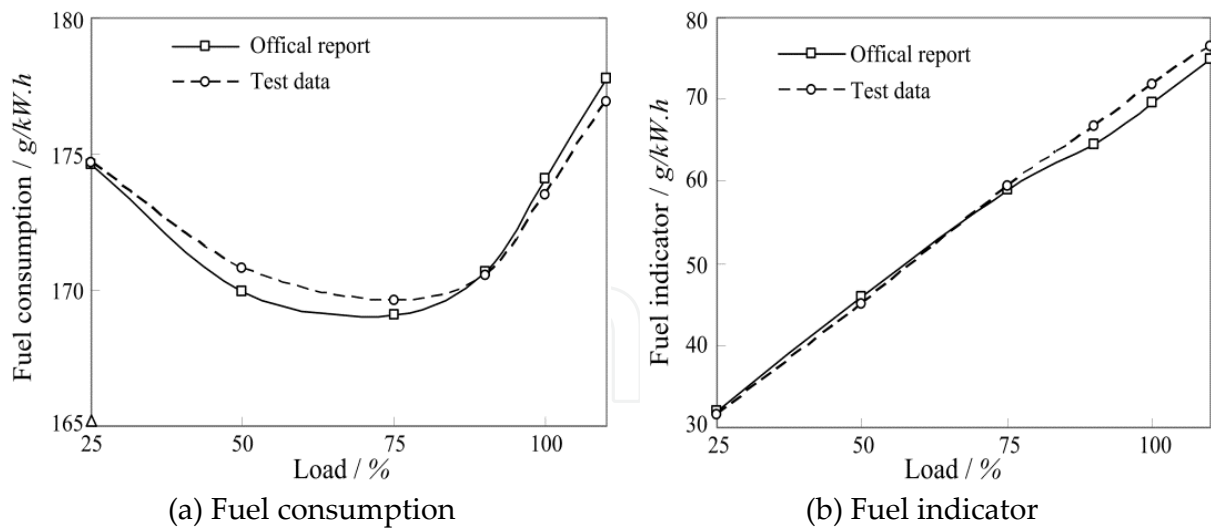
(a) Fuel consumption

(b) Fuel indicator

Figure 23. Data contrast curve of the fuel consumption and fuel indicator

Figure 24 is shown the contrasts of the exhaust valve opening and closing angles. The relative errors of opening angles are less than 0.2° CA (crank angle), and the relative errors of the closing angles are less than 0.1°CA.
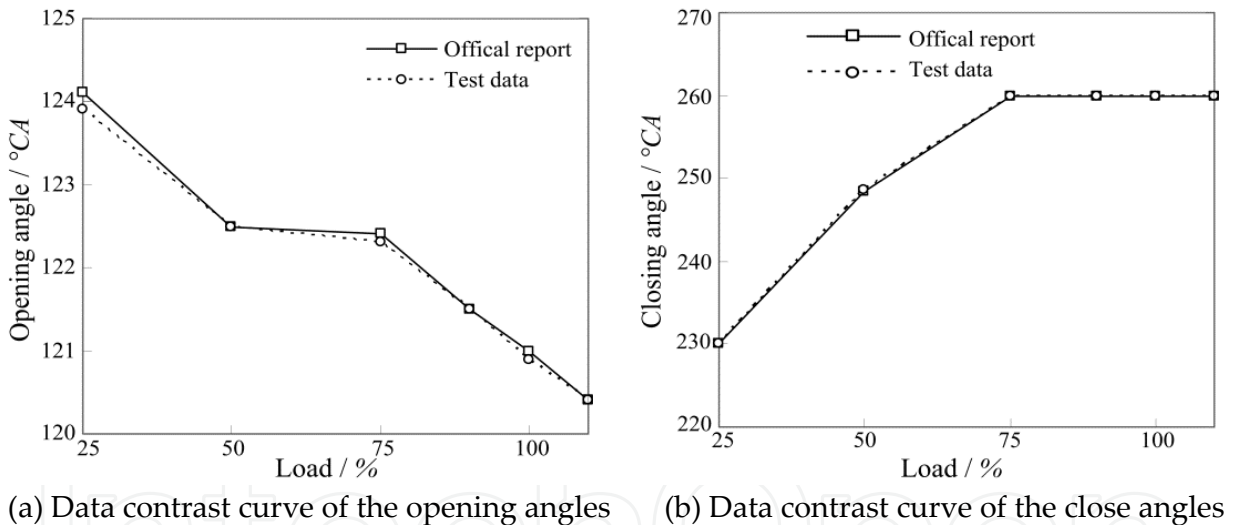

(a) Data contrast curve of the opening angles   (b) Data contrast curve of the close angles

Figure 24. Data contrast curve of the exhaust valve

Due to the existence of the engineering errors during diesel engine manufacturing, the performance data of the same type of low-speed marine diesel engine may be significantly different, let alone the measurement errors of the signals. Therefore, the existing errors on HIL simulation test bench are within the allowable range.

3.5.3. MCR full load shutdown test

During MCR full load shutdown test, the crankshaft motor speed is increased to the maximum instantaneous speed (n_{max}), then recoveries to MCR speed after an elapsed time (t_s) with the speed regulation (δ_1) according to the Formula (18).

$$\delta_1 = \frac{n_{max} - n}{n} \times 100\% \tag{18}$$

n_{max}, t_s and δ_1 in MCR full load shut down test on HIL simulation test bench is compared with the values provided in "Official Test Report" in Table 7, as a result the error is very small and could be allowed.

Number	Category	n_{max}/(r/min)	t_s/s	t_s
1	Official report	120	20.6	5.3
2	Test data	119.1	18.2	4.5

Table 7. Data contrast of the MCR

3.5.4. The minimum steady speed tests

The data contrast between the test bench and report is shown in Table 8. The data contain the diesel engine speed, the fuel indicator and the load in minimum steady speed test. As a result the errors are very small and could be allowed.

Number	Category	Speed/(r/min)	Fuel indicator/(%)	Load/(%)
1	Official report	16.0	12.5	3.5
2	Test data	15.3	13.0	2.7

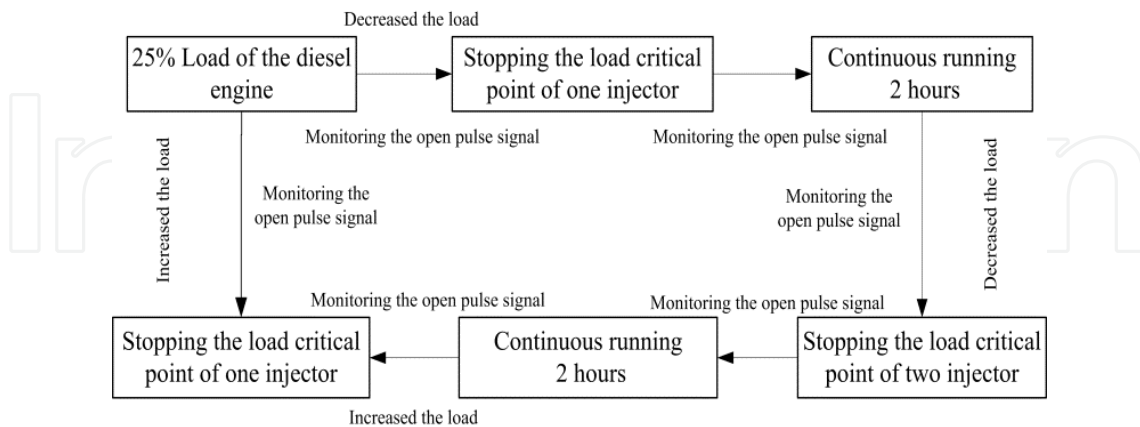**Table 8.** Data contrast of the minimum steady speed

4. Analysis of experiment result

The fuel common-rail pressure, the injection timing, the fuel injection pulse width and the fuel-injected quantity have an great influence on the fuel spray quality and the fuel injection law. Furthermore the combustion process and emissions of the diesel engines are also suffered the impact. The fuel injection mold of RT-flex diesel engine can be divided into three types, VIT ON (variable injection timing open), VIT OFF (variable injection timing off) and HEAVY SEA (diverse sea conditions). Based on the experimental data of the typical operating points in HIL simulation test bench, the control strategies of WECS are analyzed and investigated. The focus of analysis is the strategies of the fuel common rail pressure, the injection timing of VIT ON, VIT OFF and HEAVY SEA model, the control laws of the fuel injection pulse width and the fuel injected quantity, etc.

4.1. Injection control strategies in the low load condition

As shown in Figure 28, the tests are carried on in the HIL simulation test bench to figure out the control regulations of the injector in the low load condition. 25% load of the propulsive characteristics is set as the starting experiment point. Then, the load is decreased in accordance with the propulsive characteristics, and the injecting orders of the first cylinder from the FCM-20 module are measured. The load critical point, on which the number of the working injector is fallen from 3 to 2, can be determined with the amplitude variation. Additionally the test bench is kept operating in the current state for finding out the rotation law. In the same way

the load critical point can be catch, on which the number of the working injector is fallen from 2 to 1. Similarly, the load critical points in the increased process of the load also can be found. The special laws and parameters are described as follows.

Testing scheme of the injection control strategy.

Figure 25. In the decreased process of the load, the test results show that when the load is down to 7%, the average alternating interval time is 1153.6s with the number of the working injector from 3 to 2. When the load is down to 3%, there is only one injector working. And the average alternating interval times change to 1153.4s. In the increased process of the load, 10% load is the turning point of the working injector number from 1 to 2. In the 15% load, the number of the working injectors recovers to 3. And the average alternating interval times change to 1154.5s.

4.1.1. Control strategies analysis of the fuel common rail pressure

4.1.1.1. Starting process

In the start-up phase, the common rail pressure is quickly established. In order to achieve the rapidity and stability, the open-loop control strategy is applied in WECS. The experimental curves of the fuel common rail pressure and control signals are shown in Figure 26. To ensure the maximum fuel delivery, the fuel pump control signal is kept with
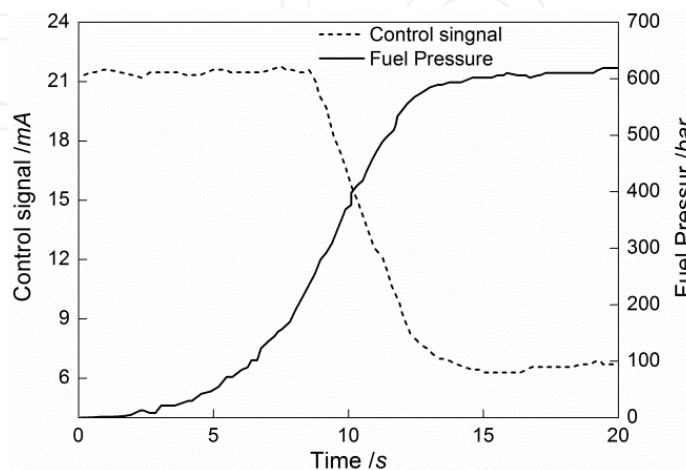

Figure 26. Curves of the fuel pressure and control signals

the 20 mA at the beginning, then the fuel pressure increases gradually with maintained control signal. When the actual pressures get to 25MPa, the control signal reduces to 7mA. While the actual pressures rise to the 60MPa, the control mode is translated into the PID (Proportion Integration Differentiation) closed-loop control state. To void pressure fluctuations, the current control signal is regarded as the initial value in the transition process.

4.1.1.2. Running process

The actual common rail pressures have to follow the target pressures at the engine working, so the closed-loop feedback control algorithm is used for the fuel rail pressure control. At the VIT ON and VIT OFF injection modes, the closed-loop control algorithm is used by WECS, and it is shown in Figure 27. The target pressures are get by looking up the rail pressure MAP chart according to the diesel engine load, and PID feedback control algorithm is carried out based on the difference value between the actual and the target pressures. What's more, the feed forward control is used to improve the system response performance.
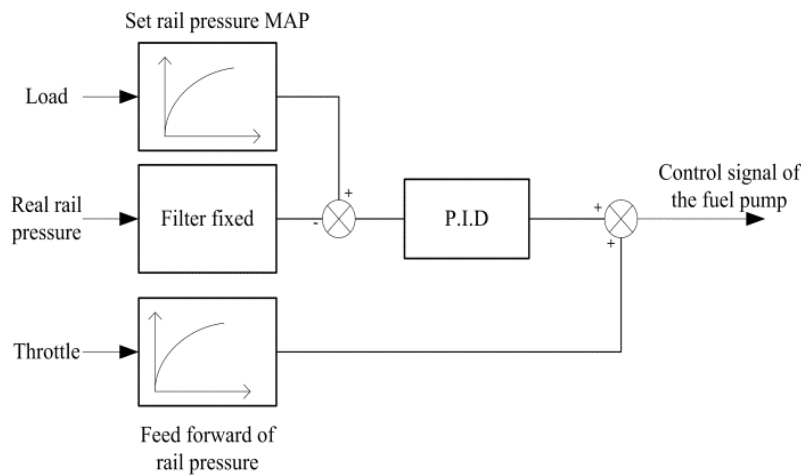

Figure 27. Closed-loop control algorithm

The actual pressure is unavoidable fluctuate in the fuel injection process. Also the pressure signals may be disturbed susceptibly. Therefore the signals need to be filtered to avoid the sharp pressure fluctuations caused by the mutations of control current signals. In addition, the target pressure may fluctuate wildly with the load changing when the diesel engine is working at the transient transition conditions. PID closed-loop control algorithm may result in a longer transition time of the actual rail pressure, which will impact on the fuel injecting and combusting. The feed forward control is added to the control algorithm to improve the response of the control system, and the feed forward control MAP of the common rail pressure is looked up via the fuel instruction. At VIT ON and VIT OFF injection modes, in the low load range of 0 to 15%, the target pressure value is 70MPa. In the load range of 15% to 25%, the target pressure value decreases to 60MPa. In the load range of 25% to 77%, the target value maintains 60MPa. In the load range of 77% to 90%, the target value gradually increases to 90MPa. In the load range more than 90%, the target value maintains 90MPa, and the feed forward control current signals is increased with the fuel indicate at the same percentage.

When the fuel injection mode is VIT ON or VIT OFF, the control strategies of the fuel common rail pressure prefer to reduce emissions at below 77% load, while prefer to improve the fuel economy when the load is more than 77% load. When the diesel engine load is less than 15% load, WECS will cut off parts of the injectors. Taking account of both the fuel economy and the emissions, the target value of the fuel common rail pressure is set as 70MPa. Since it can improve the combustion heat release rate, but not cause the NOx substantial increase. At the HEAVY SEA mode, PID closed-loop control algorithm is still active, but the target value of common rail pressure maintains 70MPa under various loads. The purpose is to avoid the actual pressure sharp fluctuation resulting in the mechanical components damaged.

4.1.2. Control strategies analysis of injection timing

The adjustment parameters of the injection timing angle will be freely set within a certain range by WECS according to the different fuel quality and the balance condition of each cylinder in the whole working situation. At VIT ON mode, in order to achieve optimal balance between economy and emissions of diesel engines, WECS adjusts the injection timing angle according to the scavenging pressure, the diesel engine speed and the change of the fuel common rail pressure. Variable injection timing angle is calculated as follows:

$$VIT \angle^\circ = inj.std + IT_{DEL} + FQS + VIT_A + VIT_B + VIT_c \tag{19}$$

The parameter $inj.std$ is set as the standard angle of the fuel injection timing, and the default value is 2 ° CA. The parameter IT_{DEL} is used for adjusting the imbalance working condition of each cylinder, caused by the tolerances of the manufacturing and the turbocharger matching. The parameter FQS is also the compensation of the fuel injection timing, which can be adjusted according to the fuel quality. If heavy oil is used, it will change the combustion lagging period of the diesel engine, then cause the cylinder pressure deviations in combustion process. The adjustment values of three injection timing mentioned above are set by user based on actual conditions. It will not change with the load in the operation process of the diesel engine. At VIT OFF and HEAVY SEA mode, the injection timing angle is the sum of three adjusted values.

At the VIT ON mode, VIT_A, VIT_B and VIT_c will be involved in the calculation of the variable injection timing angles. The parameter VIT_A is the fuel injection timing angle, which will be adjusted sectionally based on the scavenging pressure. When the scavenging pressure is lower than 0.35bar, the VIT_A value is set to 0. It is due to the low load of the diesel engine, and the auxiliary fan with the turbocharger does not work right now, so there is little significance to adjust the injection timing angle. When the scavenging pressure gradually increases from 0.35bar to 0.85bar, the value of VIT_A should gradually reduce from $0°CA$ to -$2.5°CA$. Since the ahead of the injection timing angle is benefited to improve the fuel economy at the low scavenging pressure. When the scavenging pressure continues rising to 0.85bar, the value of VIT_A should gradually rise from -2.5°CA to 1°CA. Since increased

compression pressure and the delayed injection timing angle help to reduce NOx emissions with the low combustion temperature.

The parameter VIT_B is used to adjust injection timing angle according to the diesel engine speed. When the diesel engine load is the constant, the lower average effective pressure caused by the high speed results in the reduce of the combustion pressure. When the diesel engine speed is in the region of 70% to 100%, the value of VIT_B reduces gradually from $3°CA$ to $-1°CA$.

The parameter VIT_c is used to adjust the injection timing angle according to the fuel common rail pressure. The lower of the fuel common rail pressure would cause longer injection time and poor fuel atomization. The advancement of fuel injection timing angle through VIT_c is in favor of promoting combustion in order to improve the fuel economy of the diesel engine. When the fuel rail pressure increases gradually and exceeds the operating point pressure of MCR, the delay of fuel injection timing angle will compensate for the increased NOx emissions caused by too high fuel injection pressure via the VIT_c.

4.1.3. The main conclusions

1. If the load of diesel engine gradually reduces from high load to 7% or 3%, the number of the actual working injectors of each cylinder will reduce from three to two or one. If the load of diesel engine gradually rises from low load to 10% or 15%, the number of the actual working injectors of each cylinder will increase from one to two or three. The average alternating times of the injectors are approximate 1154s.
2. At VIT ON and VIT OFF mode, the control strategies of the fuel common rail pressure prefer to reducing emissions at the load below 77%. The control strategies prefer to improve the fuel economy when the load rises to more than 77%. WECS will cut off some of the injectors when the diesel engine load is less than 15% load. Taking account of both the fuel economy and the emissions, the target value of the fuel common rail pressure is set to 70MPa, and the objects are to improve the combustion heat release rate and avoid NOx increasing. The fuel injection mode of HEAVY SEA is set to prevent the mutations of the diesel engine load caused by adverse sea conditions. At the HEAVY SEA mode, the pressure of each load maintains 70MPa, and the PID closed-loop control algorithm is still used by WECS. Because the rough sea conditions could lead to the actual rail pressure sharp fluctuation resulting in the mechanical components damaged.
3. At VIT ON mode, when the diesel engine is working in low load working condition, fuel injection control strategies of WECS prefer to reducing emissions by taking a delay of injection timing angle. When the diesel engine is working in 75% load working condition, the engine speed is 90% of MCR speed, which is the commonly working condition of the actual operation of the marine diesel engine. Therefore, if the diesel engine is working at the 75% load or nearby, the fuel injection timing angle is set to advance to improve the fuel economy in priority.

4.2. Control strategies and characteristics of the exhaust valve system

4.2.1. Characteristics of the exhaust valve

The duration opening angles of the exhaust valve in difference working condition of the diesel engine are obtained in the test bench, which are calculated by the different angle between the corresponding angle at 15% full lift of the exhaust valve opening and the one at 85% full lift of the exhaust valve closing according to the experimental data. The duration opening angle increases along with the engine load. The open degree of the servo oil pump is taken to regulate the servo oil common rail pressure. And the open degree is controlled by the duty cycle of PWM (Pulse-Width Modulation) from WECS.

The delay times and angles of the exhaust valve opening are compared with the test data under different operating conditions. There is a hydraulic mechanism delay from the opening signal sending to the exhaust valve moving. When the exhaust valve is turned on, the spring is hit by the stem to produce the rebound from the spring. Along with the increase of the diesel engine load, the common rail pressure increases, but the delay time of the exhaust valve opening reduces. If the crank angle is set as the abscissa, the delay angle of the exhaust valve opening becomes large with the increase of the engine speed.

The delay times and the angles of the exhaust valve closing are compared with different operating conditions from the test data. There is a delay between the closing signal of the exhaust valve sent by ECU and the exhaust valve closing fully. The air spring pushes the exhaust valve until closed, and the delay time of the exhaust valve doesn't change significantly, but the speed and the time of delaying angle of the exhaust valve closing increase along with the working load. Therefore, the opening and closing of the exhaust valve are related to not only the system characteristics, but also the working conditions of the diesel engine.

4.2.2. The main conclusion

The setting angles of the exhaust valve opening and closing are confirmed by WECS based on the working condition of the diesel engine. The angles of the exhaust valve opening and closing are calculated through the measured lift curve of the exhaust valve. The control signals phase is adjusted by the difference between the setting value and the calculated one. The closed-loop control is used to make the exhaust valves opening / closing at a specified angle.

Some characteristics of control strategies of the exhaust valve are listed as follows[15]:

1. Servo oil common rail pressure rises together with the increasing of the working load of diesel engine.
2. The corresponding angle of the opening control signal of the exhaust valve reduces with the increasing of the working load of diesel engine.

3. The corresponding angle of the closing control signal of the exhaust valve tends to increase with the working load of the diesel increases from 25% load to 75% load, and then the angle descends.

4. The angle difference between the exhaust valve opening and closing increases with the working load of the diesel increasing from 25% load to 75% load, and then the angle descends.

5. The corresponding angle of the 15% full lift of the exhaust valve opening drops with the increasing of the working load of the diesel engine.

6. The corresponding angle of the 85% full lift of the exhaust valve closing increases with the working load from 25% to 75%, and the delay angle of the exhaust valve closing becomes large. So the corresponding angle is amended by WECS through reducing the control signal angle.

7. The different corresponding angles between the 15% full lift of the exhaust valve opening and the 85% full lift of the exhaust valve closing generally increase with the increasing of the working load of the diesel engine.

5. Conclusion

Hardware-in-loop simulation test bench of the HPCR electronic control system for low-speed marine diesel engine is developed basing on the Wärtsilä marine diesel engine. The working characteristics of the fuel injection system and the exhaust valve, and the control strategies under difference injection model are analyzed by the method of the experimental research together with simulation analysis. The main conclusions are listed as follows:

1. Typical structural and functional characteristics of two type low-speed intelligent marine diesel engine are analyzed. HPCR system of RT-flex marine diesel engine is the hydraulic-mechanical system which functions are independent but structure is inseparable from each other. The system is consisted of the oil supply unit, the common-rail unit, the fuel injection control unit, the exhaust control unit and so on. WECS is the control center of the HPCR system. The control signals and the rail pressure regulator signals from WECS impact on the working process and the state of the HPCR system directly. The common rail unit of ME marine diesel engine adopts the form of a single-cylinder assembled with a fuel supercharger. With the help of the servo oil pressure, NC valve can conduct the fuel injection with the various injection laws. The combustion process under difference working conditions is improved. The reduced fuel consumption and emissions will be achieved.

2. HIL simulation test bench is developed on the base of the analysis of the HPCR electronic control system of 5RT-flex60c marine diesel engine. The test bench contains the real time simulation model of the diesel engine unit and the monitoring system. The experiment results are shown that: The working characteristics of the HPCR system is in conformity with that of the original machine, and WECS control strategies are reflected distinctly in the test bench. The experimental conditions are provided for the research on the system characteristics, the control strategies and the performance of HPCR of high-power marine diesel engine.

3. Different typical working conditions of the propulsive characteristics of the diesel engine are simulated throughout the HIL test bench. The feature data of HPCR electronic control system are obtained, including the fuel pressure, the needle lift, the control pulse signals of the fuel injection, the exhaust valve opening and closing signals and the exhaust valve lifts and so on. Based on the experimental data, the investigations into control strategies of WECS, the characteristic feature of the fuel injection unit and exhaust valve control unit are focused on.

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