Modeling and Simulating of Container Ship's Main Diesel Engine

Jianyuan Zhu

***Abstract*—The multi-order nonlinear model of container ship's main diesel engine is formulated based on Quasi Steady State Method, which allows simulating the dynamics of diesel engines with a relatively simple model. The whole engine system is divided into several functional blocks. The sub-models of individual blocks and the dynamic model of engine system are established according to engine working principles and experimental data. Some empirical equations describing the transient performance of engines are proposed to simplify the modeling. The real-time simulation of a container ship's main engine—MAN B&W 6L80MC low speed two-stroke turbo-charged diesel engine is presented. The simulation results show that the proposed model is satisfactory to meet the needs of engine real-time simulation with adequate accuracy.**

***Key Words*—Container Ship, Diesel Engine, Model, Simulation.**

I. Introduction

A few modeling approaches, such as Linear Method, Non-dimension Method and Quasi Steady State Method [1]-[3], are presently used to simulate the transient performance of diesel engine. With Linear Method, the diesel engine system is treated as a black box, and its input/output, i.e., the fuel rack position and engine speed relationship is represented by a transfer function. Since such simulation models consider the engine system as a black box, the internal events of engine working process are neglected, and thus various steady state and transient behaviors of the engine (such as exhaust temperature, air flow rate in turbocharger, etc.) can not be described by simulation. Furthermore, the speed changes of the engine must be assumed to be small to fit the linear approximation. When the speed changes are large, the accuracy of engine simulation falls significantly. With Non-Dimension Method, the accuracy of engine simulation can be high. However, to simulate all the internal events of the engine, the model may get very bulky and very complicated; and to determine all the constants within the model, a great amount of analytical and experimental effort will be required. Moreover, the calculating work for Non-Dimension Method is time-consuming and prolix for engine real-time simulation.

A multi-order nonlinear model of two-stroke marine diesel engine is discussed and formulated based on Quasi Steady State Method, which allows for simulating both the steady state and the transient performance of diesel engines with a relatively simple model. The overall engine system is divided into several functional blocks, namely, air compressor, air cooler, scavenging air receiver, engine cylinder, exhaust manifold, exhaust gas turbine, fuel injection and electronic governor etc. The approach adopted here is to model those events, which can be expressed by mathematical equations, such as in computation of the engine speed and the exhaust gas temperature, by the corresponding mathematical equations. Whereas for the events which are highly complicated and generally require a very bulky model and also a great number of experimental data not easily available from shop test to describe accurately, such as the combustion or gas exchange process, the Quasi Steady State Method is adopted. These kinds of models can demonstrate the events and the internal parameters of the engine with adequate accuracy and reasonable amount of calculating work for real-time simulation.

On the basis of analyzing the experimental data of engines, some empirical equations, which describe the transient characteristics of two-stroke diesel engines, such as mechanical efficiency, indicating efficiency, air cooler efficiency etc., are proposed and discussed.

The object for the simulation is a container ship's main engine—MAN B&W 6L80MC low speed two-stroke turbo-charged diesel engine. The main functions of the simulation software consist of data pre-processing (such as multiple parameters linear regression and nonlinear regression), whole engine system modeling and simulating, and simulation results analyzing etc. Various interfaces are developed to facilitate the engine real-time simulation. The simulation results show that the proposed multi-order nonlinear model based on Quasi Steady State Method can demonstrate the events and internal parameters of the engine with a relatively simple model and high degree of accuracy.

II. MODEL OF MARINE MAIN DIESEL ENGINE

The overall engine system is divided into several functional blocks. For each functional block, its dynamic characteristics are expressed in mathematics equations or graphs, and the dynamics of the overall engine system is thus expressed as a set of simultaneous algebraic and differential equations.

A. Air Compressor of Turbocharger

6L80MC marine diesel engine is equipped with two sets of VTR564 turbochargers. The inlet air temperature of air compressor T_{ci} (K) is considered to be equal to the ambient temperature T_0. Its inlet air pressure P_{ci} (Pa) is expressed as

$$P_{ci}=b\cdot P_0 \tag{1}$$

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Where b is the inlet loss factor. According to the regression analysis of experimental data, it can be represented as

$$b=1-0.000012G_c^2 \tag{2}$$

Where G_c (kg/s) is the air flow rate of compressor. The outlet temperature of compressor T_{co} is given by

$$T_{co}=\frac{T_{ci}}{\eta_c}[(\frac{p_{co}}{p_{ci}})^{\frac{k-1}{k}}-1]+T_{ci} \tag{3}$$

Where k is the ratio of Cp/Cv for air, Cp (J/kg.K) and Cv (J/kg.K) are the constant pressure and the constant volume specific heat of air respectively, η_c is the compressor efficiency and P_{co} is the outlet pressure of compressor. The compressor torque (Nm) can be written as

$$M_c=\frac{30C_p\cdot G_c\cdot T_{ci}}{\pi\cdot\eta_c\cdot n_{tc}}[(\frac{p_{co}}{p_{ci}})^{\frac{k-1}{k}}-1] \tag{4}$$

Where n_{tc} (r/min) is the speed of turbocharger. The relationships between G_c , P_{co} , n_{tc} and η_c are determined according to the maps of compressor characteristics furnished by the engine manufacturer.

B. Exhaust Gas Turbine

The outlet gas pressure of exhaust gas turbine P_{to} is given by

$$P_{to}=P_o+C(G_t/G_{tm})^2 \tag{5}$$

Where G_{tm} and G_t are the gas flow rate of turbine at rated condition and at simulation calculating condition respectively, P_0 is the atmospheric pressure and C is a constant. From the experimental data, it is found that C equals to 1.55×10^3 .

When calculating the turbine flow rate, we treat the turbine as a nozzle, thus it is expressed as

$$G_t=F_e\frac{p_{ti}}{\sqrt{R_e\cdot T_{ti}}}\sqrt{\frac{2k_e}{k_e-1}(\frac{p_{ti}}{p_{to}})^{\frac{-2}{k_e}}[1-(\frac{p_{ti}}{p_{to}})^{\frac{1-k_e}{k_e}}]} \tag{6}$$

Where F_e (m^2) is the effective equivalent nozzle area of the turbine, P_{ti} and T_{ti} are the turbine inlet gas pressure and inlet gas temperature respectively, R_e (J/kg.K) is the exhaust gas constant and k_e is the ratio of Cp/Cv for exhaust gas. The turbine torque is expressed as

$$M_t=\frac{30G_t\cdot\eta_t\cdot\eta_{mt}}{\pi\cdot n_{tc}}\cdot\frac{k_e\cdot R_e\cdot T_{ti}}{k_e-1}[1-(\frac{p_{to}}{p_{ti}})^{\frac{k_e-1}{k_e}}] \tag{7}$$

Where η_t is the turbine efficiency and η_{mt} is the mechanical efficiency of turbocharger. Then the speed of turbocharger can be represented as

$$\frac{dn_{tc}}{dt}=\frac{30}{\pi}\cdot\frac{M_t-M_c}{I_{tc}} \tag{8}$$

Where I_{tc} is the moment of inertia of turbocharger.

C. Air Cooler

The air cooler equipped for marine diesel engine has high efficiency, its pressure drop is small and the air temperature after the cooler is low. Thus the cooler can be expressed as a simple model. The air temperature after cooler, T_{ao} is given by

$$T_{ao}=T_{ai}-E(T_{ai}-T_{wi}) \tag{9}$$

Where T_{ai} and T_{wi} denote the inlet temperature of the air and the water respectively. The air cooler efficiency E is determined from the regression analysis of experimental data.

$$E=0.00097(T_{ai}-T_{wi})+0.803 \tag{10}$$

According to the experimental data, the outlet air pressure of cooler is expressed as

$$P_{ao}=0.9926P_{ai}+83.7 \tag{11}$$

Where P_{ai} is the inlet air pressure.

D. Scavenging Air Receiver

The air Temperature in scavenging air receiver, T_r is determined by

$$\frac{dT_r}{dt}=\frac{k}{M_r}[G_c\cdot T_{ri}-(G_s+\frac{1}{k}\cdot\frac{dM_r}{dt})T_r] \tag{12}$$

Where M_r (kg) is the amount of air in receiver and T_{ri} is the air temperature before receiver, G_s is the air flow rate into engine cylinder.

$$G_s=\frac{\varphi\cdot V_s\cdot P_r\cdot n}{60\cdot R\cdot T_r} \tag{13}$$

Where n (r/min) is the engine speed, V_s (m^3) is the swept volume of engine cylinder, φ is the scavenging excess air factor and P_r is the air pressure in receiver.

$$\frac{dP_r}{dt}=\frac{k\cdot R}{V_r}(G_c\cdot T_{ri}-G_s\cdot T_r) \tag{14}$$

Where V_r is the volume of scavenging air receiver.

E. Engine Cylinder

The engine torque equation is expressed as

$$\frac{\pi}{30}(I_e+I_L)\frac{dn}{dt}=M_e-M_L \tag{15}$$

Where I_e (Nms^2) and I_L are the equivalent moment of inertia of engine and load respectively, M_e (Nm) and M_L are brake torque of engine and load torque on engine respectively, The indicated torque of engine, M_i is given by

$$M_i=H_u \cdot g_c \cdot \eta_i/(2\pi)=f_1(g_c,\eta_i) \tag{16}$$

Where H_u (kJ/kg) is the low calorific value of fuel and g_c (kg) is the amount of fuel injected into cylinder per cycle. η_i is the engine indicated efficiency and it can be simplified as a function of engine speed ω (rad/s) and excess air coefficient α. According to the experimental data, it is expressed as

$$\eta_i = f_2[(\omega-\omega_0)^2+d^2(\alpha-\alpha_0)^2] \tag{17}$$

The engine's mechanical efficiency η_m mainly depends upon the engine speed, load (indicated power N_i) and cooling water temperature t_w (ºC), From the regression analysis of experimental data, it is found that

$$\eta_m =(75.347-0.010n+0.000477N_i+0.0872t_w)/100 \tag{18}$$

The exhaust temperature T_e is expressed as a function of engine speed and amount of fuel injected into cylinder per cycle. From the experimental data, it is given by

$$T_e=165.415g_c+41.878n^{0.325}+305.4 \tag{19}$$

F. Exhaust Manifold

The exhaust gas temperature in manifold T_m is determined by

$$\frac{dT_m}{dt}=\frac{k_e}{M_m}[G_e \cdot T_e-(G_t+\frac{1}{k_e}\cdot\frac{dM_m}{dt})T_m] \tag{20}$$

Where G_e is the gas flow rate from cylinder to manifold, G_t is the gas flow rate of turbine and M_m (kg) is the amount of gas in exhaust manifold. The gas pressure in manifold P_m is represented by

$$\frac{dp_m}{dt}=\frac{k_e \cdot R_e}{V_m}(G_e \cdot T_e-G_t \cdot T_{ti}) \tag{21}$$

Where V_m is the volume of exhaust manifold and T_{ti} is the inlet gas temperature of turbine.

G. Fuel Injection

According to regression analysis of experimental data, the amount of fuel injected into cylinder is expressed as

$$g_c=0.0356n^{0.335}+0.00557Z-0.19 \tag{22}$$

Where Z is the fuel rack position and n is the speed of fuel injection pump, which equals to the engine speed for a two-stroke diesel engine.

H. Electronic Governor

MAN B&W 6L80MC diesel engine is equipped with digital electronic governor DGS8800. The main purpose of the governor is to regulate the position of the engine fuel servo in order to maintain an engine speed equal to a reference setting. The governor system is composed of two separate parts: the speed regulating function and the fuel actuating function. The model of electronic governor is formulated based on PID control [4],[5], which is shown in Fig. 1.

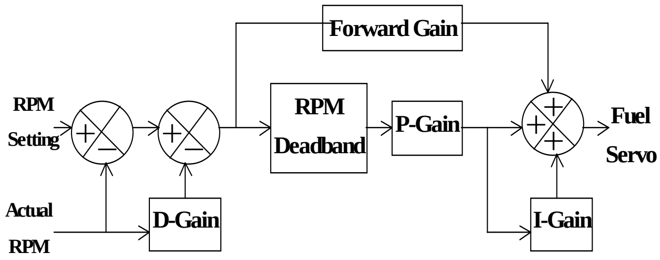

Fig.1 Electronic governor

III. SPECIFICATION OF THE DIESEL ENGINE

MAN B&W 6L80MC two-stroke diesel engine is chosen as the object for the simulation. The principal specifications of the engine are shown as follows.

Type: Marine single acting two-stroke
Cylinders: 6
Cylinder Diameter: 0.8 m
Stroke: 2.59 m
Nominal Output: 16670 kW
Nominal Speed: 88 r/min
Mean Effective Pressure: 1.33 Mpa

IV. SIMULATION RESULTS OF THE ENGINE

The proposed simulation model for diesel engine system is simulated and a computer program written in Visual Basic is developed to simulate both the steady state and the transient performance of diesel engine. The steady state performance of the engine as computed from the simulation model is compared with the experimental data as shown in Fig. 2.

It can be seen from Fig. 2 that for a wide range of engine operation conditions there is a good agreement with the experiment data, and the error margin of performance parameters is well within 5%. This shows that the proposed simulation model is appropriate and the simulation accuracy is high enough.

Various transient characteristics were simulated, and simulations in which engine speed setting or load changes were carried out. The case of the engine running initially at 8400 kW, 70.1 r/min with a sudden load increase of 25% nominal output was chosen as an example. The dynamic response of engine speed n, fuel rack position Z, turbocharger speed n_{tc}, turbine inlet temperature T_{ti} and inlet pressure P_{ti},

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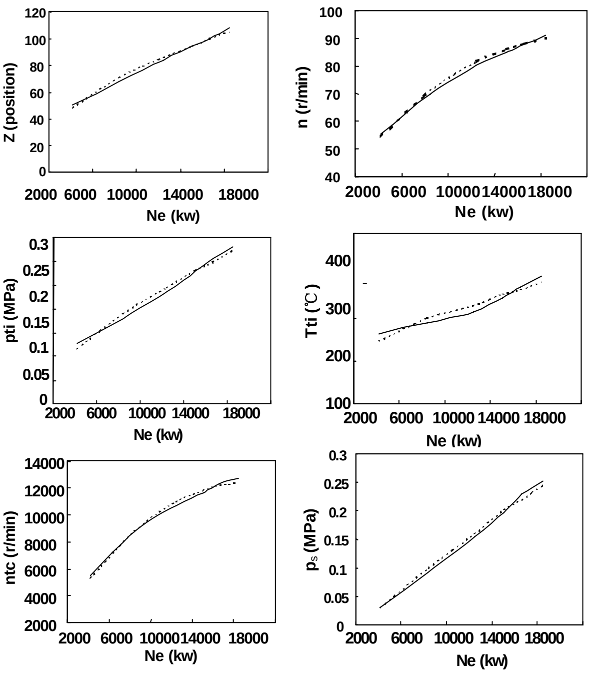
and scavenging air pressure P_s are illustrated respectively, as shown in Fig. 3.



——Experimental Results; ------ Simulation Results

Fig. 2 Comparison of simulation results to experimental data: steady state performance

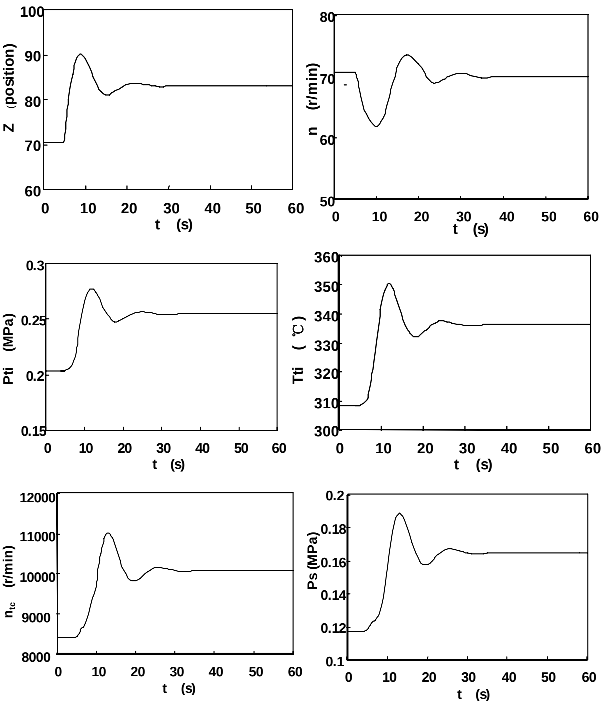


Fig. 3 Simulation results: Case of sudden load increase

The simulation results show that the transient characteristics of the engine are accurately simulated and the developed simulation program is satisfactory for marine diesel engine real-time simulation.

V. CONCLUSION

The simulation model for simulating both the steady state and the transient characteristics of two-stroke marine diesel engine is proposed in this paper. The performance of the simulation has been verified by comparing simulation results with experimental data in various operation conditions. The simulation results show that the proposed multi-order nonlinear model based on Quasi Steady State Method can demonstrate the events and internal parameters of the engine with a relatively simple model and high degree of accuracy.

On the basis of analyzing the experimental data of engines, some empirical equations, which describe the transient characteristics of marine diesel engines, are proposed in order to simplify the modeling, by which the accuracy and speed of the simulation are significantly improved. It can be concluded that the developed simulation software is satisfactory to meet the requirements of marine main diesel engine real-time simulation.

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