International Journal of Computer Information Systems and Industrial Management Applications.
ISSN 2150-7988 Volume 5 (2013) pp. 480-487

Modeling and Simulation of Working Process of Marine Diesel Engine with a Comprehensive Method

Cao Hui[1], Wu Peili[2] and Zhang Jundong[3]

[1] Marine Engineering College, Dalian Maritime University,
Linghai Road 1 Unit 1, Dalian, China
bingbinghui@sohu.com

[2] Marine Engineering College, Dalian Maritime University,
Linghai Road 1 Unit, Dalian, China
wupl121@163.com

[3] Marine Engineering College, Dalian Maritime University,
Linghai Road 1 Unit, Dalian, China
zhjundog@dlmu.edu.cn

***Abstract*: The working process of marine diesel engine is simulated by combining mean value engine model and volumetric model. It reflects some average parameters such as effective pressure, effective power and engine speed, as well as reflects real-time explosion pressure, maximum temperature in cylinder and indicator diagram. In order to accelerate the simulation, Crank Angle and cylinder pressure is imported into MATLAB workspace, and a diesel model based on BP neural networks is built. The cylinder pressure can be shown in marine simulator by using trained BP neural network.**

Keywords: Marine diesel engine, Volumetric model, MVEM, Neural network, Working Process**.**

I. Introduction

The energy shortage and more strict environmental standards make the energy efficiency and environmental protection become the most important issues to be considered by shipping industry and shipbuilding industry. So a good marine diesel engine not only has to keep good dynamic and steady characteristics, but also has to improve fuel economy and minimize the emissions. In order to meet these requirements, it is necessary for the marine diesel engine model to predict its properties in the severe environmental conditions, such as load changing quickly. In the other hand, the model is one of the most important parts of the marine simulator for dynamic simulation and fault diagnosis.

The working process simulation of diesel engine is known as using differential equations to carry on mathematics description on the working processes of the systems, solving the equations on a computer, and finding the changing regularity of parameters varying with Crank Angle or time. The simulation model of diesel engine can be generally classified as quasi-static model, volumetric model and feature model [1].

Mean value engine model (MVEM) [2] was presented by E. Hendricks in 1989, it combining quasi-static model and volumetric model. The diesel engine is divided into several relatively independent units, such as scavenging air receiver, diesel engine, exhaust pipe, turbocharger, intercooler and governor in MVEM with high speed and high precision.

Volumetric Model was only used in steady simulation but not for the dynamic change processes until Song Zhu [3] provided the crank-connecting rod model in 1991, which lay foundations for volumetric model used into dynamic simulation. The volumetric model divides the working process into compression, combustion, expansion and exhaust by which the working process can be simulated dynamically [4]. It needs more diesel structure characteristics, including air and location of the scavenging air ports, air, cone angle and lift curves of exhaust valve, and injection timing, etc.

Traditional MVEM can't simulate the working process of the cylinders dynamically, and that the simulation and calculation of the volumetric model can be time-consuming and error-prone. In order to avoid the problems before, a new model is put forward in this paper which combines two models using open-loop control to simulate the working process of the cylinder dynamically. BP neural network is used in the model to achieve co-simulation in the marine simulator.

II. Mean Value Engine Model

MVEM takes the time as the calculating unit with less computational effort and little time, and it less depends on the detail type of engine. MVEM divides the engine into several relatively independent volume units combines with the character of quasi-static model and volumetric model. Fig.1 shows the schematic diagram of a diesel engine. Turbine powered by exhaust air drives the compressor to compress fresh air. The compressed air cooled by intercooler is fed into scavenge box which can maintain a certain pressure. Fuel and air burns in the cylinder to push piston and produces torque.

The exhaust gas is cleared out of the cylinders after combustion is completed makes the temperature of exhaust pipe rise. MVEM is established based on the laws of mass, energy conservation and ideal gas state equation [5]. The block diagram of diesel engine is shown by Fig.2, and the block diagram of mean value model with SIMULINK is shown by Fig.3 and Fig.4. The specific mathematical model describes of as described as following.

A. Excess air coefficient

Excess air coefficient is an important parameter for combustion and emissions means the ratio of amount of scavenge air to amount of actual air for combustion. An appropriate excess air coefficient can improve its thermal efficiency, lower exhaust temperature, and reduce pollution [6]. Therefore, excess air coefficient has a great impact on the accurate modeling [7].

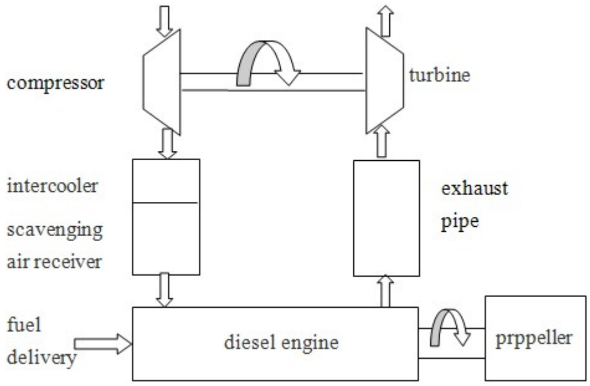

Figure 1. Schematic diagram of a diesel engine

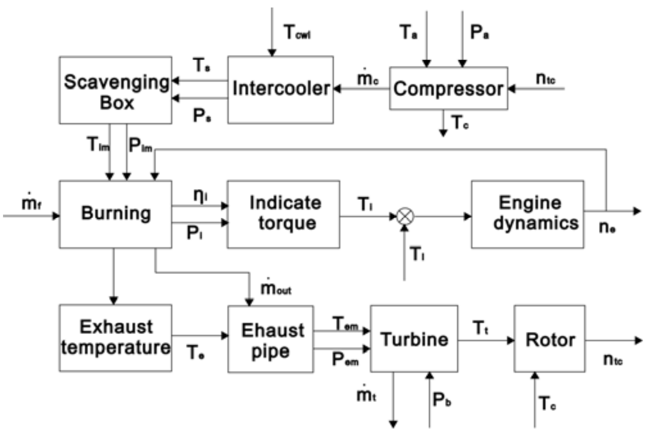

Figure 2. Block diagram of diesel engine

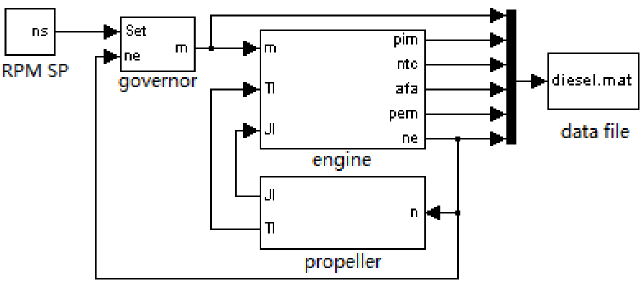

Figure 3. SIMULINK block diagram of diesel engine mean value model

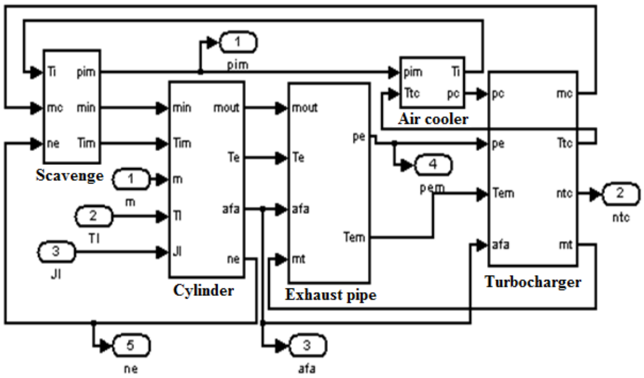

Figure 4. Diesel engine module details

Air mass flow into cylinder can be calculated as below:

$$\hat{m}_{in} = \eta_v \frac{p_{im} V_d N_{cyl} n_e}{60 N_{st} R T_{im}} \tag{1}$$

p_{im} Suction pipe pressure;

η_v Cylinder volume coefficient;

N_{cyl} Cylinder number;

V_d Empty volume in each cycle;

n_e Engine revolution;

N_{st} Number of stroke, 2 stroke engine N_{st} =1, four stroke engine N_{st} =2.

So, fuel oil average mass flow in each cycle is defined as below:

$$\hat{m}_f = \frac{m N_{cyl} n_e}{60 N_{st}} \tag{2}$$

On the basis of excess air coefficient definition, we can take average excess air coefficient as the ratio of air mass flow into cylinder and fuel oil average mass flow.

Because on the process of scavenging, a part of fresh air will enter the exhaust pipe with exhaust, so flow into cylinder air cannot burn fully. The ratio of air flow through scavenge port and burning air is called scavenging coefficient. So if we want calculate coefficient exactly, we should take scavenging coefficient into consideration. Scavenging coefficient is related to valve overlap angle, so excess air coefficient can be redefined as below:

$$\alpha = \frac{\hat{m}_{in}}{\hat{g}_f L_o \phi_s} \tag{3}$$

α Excess air coefficient;

ϕ_s Coefficient of scavenging;

$\hat{m}_{in}$ Air mass flow;

$\hat{g}_f$ Fuel mass flow;

$L_o = 14.3$, Minimum air for complete combustion of 1kg fuel.

B. Indicated thermal efficiency

The equation (4) is obtained by Hendricks E., who got the indicated thermal efficiency through a lot of experimental data [8].

$$\eta_i = (a_1 + a_2 n_e + a_3 n_e^2)(1 - a_4 \alpha^{a_5}) \tag{4}$$

$a_i (i = 1,2,3,4,5)$ is constant related to different engines' structure. So, the average indicated torque and exhaust temperature can be calculated as flow:
are constants related to engine structure

$$T_i = \frac{30}{\pi}\frac{P_i}{n_e} = \frac{30}{\pi}\frac{10^3 \eta_i H_u \hat{m}_f}{n_e} \tag{5}$$

$$T_e = T_{im} + \frac{K}{1 + L_o \alpha} \tag{6}$$

C. Scavenging box

Scavenging box model depends on mass and energy conservation law and ideal gas state equation. Scavenging box heat dissipation affects less to diesel engine model by Sergey Edward Lyshevski's research. So, the heat dissipation affection can be ignored [9].
Air flows into intercooler from suction pipe then into cylinder. Based on ideal gas state equation, pressure in suction pipe can be indicated as equation (7).

$$\hat{p}_{im} = \frac{kR}{V_{im}}(\hat{m}_c T_s - \hat{\mathrm{m}}_{\mathrm{in}} T_{im}) \tag{7}$$

$\hat{p}_{im}$ Pressure change rate in suction pipe;
V_{im} Volume of suction pipe;
k Rate of specific heat;
R Gas constant;
T_{im} Temperature in suction pipe;
$\hat{m}_c$ Flow of intercooler outlet;
T_s Temperature of intercooler outlet;
$\hat{m}_{in}$ Air flow into cylinder.

D.Compressor

On the base of ideal gas adiabatic compression, the boost pressure rate, rotor speed, temperature of compressor outlet and torque are calculated as flow:

$$T_{tc} = T_a \left\{ 1 + \frac{1}{\eta_c}\left[(\pi_k)^{\mu} - 1 \right] \right\} \tag{8}$$

$$T_c = \frac{\hat{m}_c c_p T_a}{\eta_c n_{tc}} \left[(\pi_k)^{\mu} - 1 \right] \tag{9}$$

$\hat{m}_c$ Air mass flow of compressor;
η_c Efficiency of compressor;
n_{tc} Rotor speed;
π_k Boost pressure rate;
T_a Suction temperature of compressor;
T_{tc} Outlet temperature of compressor;
T_c Absorbed torque of compressor;
k Adiabatic index of air;
c_p Specific heat at constant pressure.

E. Turbine

The turbine of diesel engine can be simplified to a nozzle, which depend on compress rate ferrets out flow coefficient and turbine efficiency [10] [11]. So, the air mass flow can be calculated as below:

$$\hat{m}_t = \mu_t F_{TA} \psi \frac{p_{em}}{\sqrt{RT_{em}}} \tag{10}$$

μ_t Flow coefficient;
F_{TA} Equivalent area of turbine nozzle;
ψ Flow function.

If $\pi_t \le \left(\frac{k_e + 1}{2} \right)^{\frac{k_e}{k_e - 1}}$, then

$$\psi = \sqrt{\frac{2k_e}{k_e - 1}\left(\frac{p_b}{p_{em}} \right)^{\frac{2}{k_e}} \left[1 - \left(\frac{p_b}{p_{em}} \right)^{\frac{k_e - 1}{k_e}} \right]} \tag{11}$$

Else, ψ will not change as flow as pressure and reach the maximum $\psi_{\max}$:

$$\psi_{\max} = \left(\frac{2}{k_e + 1} \right)^{\frac{k_e}{k_e + 1}} \sqrt{\frac{2k_e}{k_e + 1}} \tag{12}$$

k_e Exhaust adiabatic index;
π_t Expand rate.

Turbine output torque can be calculated as below:

$$T_t = \frac{\hat{m}_t c_{pe} T_{em} \eta_t}{n_{tc}} \left[1 - \left(\frac{1}{\pi_t} \right)^{\frac{k_e}{k_e + 1}} \right] \tag{13}$$

T_t Compressor drive torque;
T_{em} Temperature of exhaust;
c_{pe} Specific heat at constant pressure of exhaust;
c_{pe} Turbine efficiency rotor.

F. Rotor

Depend on Newton's Second Law [12], rotor model can be described as below:

$$\hat{n}_{tc} = \frac{\eta_m T_t - T_c}{J_{tc}} \frac{60}{2\pi} \tag{14}$$

J_{tc} Rotational inertia of rotor;
η_m Efficiency of turbine.

G.Air cooler

Air cooler have more efficiency, it can be abstracted as throttling node, and the pressure drop of fluid can be described as below:

$$\Delta p_s = \eta_\gamma \frac{\hat{m}_c^2}{\rho_c} \tag{15}$$

Compressor outlet pressure and temperature of intercooler outlet can be described as follow equations (16) (17):

$$p_s = p_{im} + \Delta p_s \tag{16}$$

$$T_s = T_{tc} - \eta_s (T_{tc} - T_{cwi}) \tag{17}$$

T_{cwi} Inlet temperature of cooling medium;

η_s Cooling efficiency.

H.Exhaust pipe

The exhaust pipe model of diesel engine can be described by the equation (18)

$$\hat{p}_{em} = \frac{k_e R_e}{V_{em}} \left(\hat{m}_{out} T_e - \hat{m}_t T_{em} - \frac{\hat{Q}_{wem}}{c_{pe}} \right) \tag{18}$$

$\hat{p}_{em}$ Pressure change rate in exhaust;

R_e Exhaust gas constant;

k_e Exhaust adiabatic index;

V_{em} Volume of exhaust pipe;

$\hat{m}_{out}$ Mass flow of exhaust;

T_e Temperature of exhaust;

$\hat{m}_t$ Mass flow of turbine;

T_{em} Temperature in exhaust pipe;

$\hat{Q}_{wem}$ Thermo flow.

I. Crankshaft and connecting rod dynamic model

When the crankshaft and connecting rod move, the instantaneous position of crank is be described with the angle φ, it is shown in Figure 5.

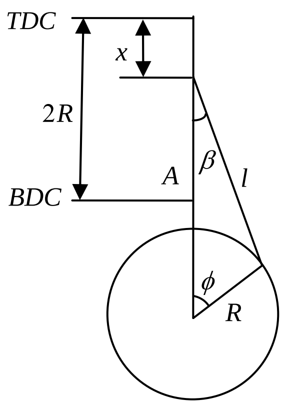

Figure 5. Schematic diagram of crank-connecting rod mechanism

$\sin\beta = \lambda \sin\varphi$, $\lambda = R/l$, so piston displacement and piston velocity can be shown as follow equations (19), (20):

$$x = R\left[\left(1+\frac{1}{\lambda}\right) - \left(\frac{\sqrt{1-\lambda^2 \sin^2\left(\frac{\pi n_e}{30}t\right)}}{\lambda} + \cos\left(\frac{\pi n_e}{30}t\right)\right)\right] \tag{19}$$

$$x' = R\frac{\pi n_e}{30}\left(\frac{\lambda \sin\left(\frac{\pi n_e}{15}t\right)}{2\sqrt{1-\lambda^2 \sin^2\left(\frac{\pi n_e}{15}t\right)}} + \sin\left(\frac{\pi n_e}{15}t\right)\right) \tag{20}$$

Piston's acceleration can be described as below:

$$x'' = R\left(\frac{\pi n_e}{30}\right)^2 \left[\frac{\lambda \cos\left(\frac{\pi n_e}{15}t\right)}{\sqrt{1-\lambda^2 \sin^2\left(\frac{\pi n_e}{30}t\right)}} + \frac{\lambda^3 \sin^2\left(\frac{\pi n_e}{15}t\right)}{4\left(1-\lambda^2 \sin^2\left(\frac{\pi n_e}{30}t\right)\right)^{3/2}} + \cos\left(\frac{\pi n_e}{30}t\right)\right] \tag{21}$$

Gas pressure on the piston is calculated as below:

$$F_g = \frac{\pi}{4} D^2 \left(p_z - p_s\right) \tag{22}$$

So, the reciprocating inertia force is as below:

$$F_j = -m_j x'' \tag{23}$$

m_j Crank and connecting rod reciprocating mass.

The force analysis is shown as follows:

$$F = F_g + F_j \tag{24}$$

By force analysis, the connecting rod can be regarded as rigid body, so the mechanical analyzing diagram of single-cylinder crankshaft and connecting rod is shown in Figure 6.

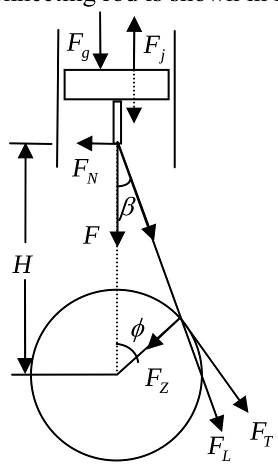

Figure 6. Analyzing diagram of force acting on the crankshaft

All kinds of force can be described as below equations:

$$F_L = F/\cos\beta = F\Big/\sqrt{1-\lambda^2 \sin^2\left(\frac{n\pi}{30}\right)t} \tag{25}$$

$$F_N = F\frac{\lambda \sin\left(\frac{n\pi}{30}\right)t}{\sqrt{1-\lambda^2 \sin^2\left(\frac{n\pi}{30}\right)t}} \tag{26}$$

$$F_z = F_L \cos(\alpha+\beta) = F\left[\sqrt{1-\sin^2\left(\frac{n\pi}{30}\right)t} - \lambda \sin^2\left(\frac{n\pi}{30}\right)t\right] \quad (27)$$

$$F_T = F\left(\sin\left(\frac{n\pi}{30}\right)t + \frac{\lambda \sin 2\left(\frac{n\pi}{30}\right)t}{2\sqrt{1-\lambda^2 \sin^2\left(\frac{n\pi}{30}\right)t}}\right) \quad (28)$$

F_L Connecting rod thrust;

F_N Side thrust;

F_Z Normal force;

F_T Tangential force.

J. Governor

The control mode of electronic governor is PI in 6S60MC, so the governor's output logic as below:

$$P(t) = K_p\left(e(t) + \frac{1}{T_i}\int e(t))\right) \quad (29)$$

K_p proportion coefficient;

T_i integral time;

$e(t)$ revolution deviation.

K. Torque and exhaust temperature

Cylinder combustion process is very complex, and the mathematical model of volume method cannot meet the requirements of real-time, so this part of the data obtained with the indicated data. So the mean indicate torque of diesel engine is calculated by:

$$T_i = \frac{30}{\pi} \cdot \frac{1000\eta_i H_u \hat{m}_f}{n_e} \quad (30)$$

$\hat{m}_f$ Injection flow of each cycle in one cylinder;

H_u Fuel lower calorific value;

η_i Indicated efficiency;

n_e RPM of diesel engine.

Depend on experience of Man B&W, friction loss pressure can be described as flow:

$$p_f = k_1 n_e + k_2 n_e^2 \quad (31)$$

k_1, k_2 are constants related to engine structure.

Average friction torque can be described as below:

$$T_f = \frac{10^3 V_d p_f}{2\pi N_{st}} \quad (32)$$

N_{st} Number of strokes of diesel engine.

Load torque can be described as below:

$$T_p = K_p \rho n_e^2 D^5 \quad (33)$$

K_p Propeller torque coefficient;

ρ Density of sea water;

D Diameter of propeller.

III. Volumetric Model

Volumetric model divides the diesel engine into several control volumes, including scavenging air receiver, cylinder, exhaust pipe, turbocharger and intercooler, and they are combined by energy and gas flow. To calculate the working process, the model made following hypothesis [13]:

a) Regarding working medium as ideal gas and consists of air and combustion products;
b) Working medium distributes uniformly, and fresh air and residual gas mixes well instantly;
c) The gas flows into and out of the cylinder is quasi-steady flow, not considering fluctuation.

A. Fundamental equation of the working process

The pressure of cylinder can be calculated by formula (34) and formula (35):

$$\frac{dT_z}{d\varphi} = \left(\frac{dQ_f}{d\varphi} + \frac{dm_s}{d\varphi}h_s - \frac{dm_s}{d\varphi}h_e - \frac{dQ_w}{d\varphi} - p_z\frac{dV_z}{d\varphi} - c_{vmz}T_z\frac{dm_z}{d\varphi}\right)\Bigg/ m_z c_{vz} \quad (34)$$

$$p_z V_z = m_z R_z T_z \quad (35)$$

$dQ_f/d\varphi$ Heat release rate;

$dm_s/d\varphi$ Mass flow of scavenge air;

$dm_e/d\varphi$ Mass flow of exhaust air;

$dQ_w/d\varphi$ Heating taken away by cooling medium;

$p_z\, dV_z/d\varphi$ Work on cylinder;

$dm_z/d\varphi$ Mass flow of gas in cylinder;

h_s Enthalpy of scavenge air;

h_e Enthalpy of exhaust air;

c_{vmz} Constant-volume specific heat of exhaust air;

c_{vz} Constant-volume specific heat of working medium in cylinder;

p_z Gas pressure of cylinder;

T_z Gas temperature of cylinder;

V_z Cylinder volume;

R_z Gas constant of cylinder;

m_z Gas quality of cylinder.

B. Rate of heat release

Combustion process is an important part of the model, and the understanding in the complex characteristics of the combustion is not mature. In the model, we assumed that fuel is injected into cylinder at one point, and ignore the injection gate. The rate of heat release is [4]:

$$\frac{dQ_f}{d\varphi} = 4.2 H_u \cdot g_f \cdot \eta_u \cdot \frac{dx}{d\varphi} \quad (36)$$

Double Vibe empirical formula is always used to simulate the rate of heat release as below:

$$\frac{dx}{d\varphi}=\frac{dx_1}{d\varphi}+\frac{dx_2}{d\varphi} \tag{37}$$

$$\frac{dx_1}{d\varphi}=\left[\left(m_1+1\right)\cdot 6.908\left(\frac{1}{2\tau}\right)^{(m_1+1)}\cdot \left(\varphi-\theta_z\right)^{m_1} e^{-6.908/(2\tau)^{(m_1+1)}(\varphi-\theta_z)^{(m_1+1)}}\right]\left(1-Q_d\right) \tag{38}$$

$$\frac{dx_2}{d\varphi}=\left[\left(m_2+1\right)\cdot 6.908\left(\frac{1}{\varphi_{zd}}\right)^{(m_2+1)}\cdot \left(\varphi-\theta_z-\tau\right)^{m_2} e^{-6.908/(\varphi_{zd})^{(m_2+1)}(\varphi-\theta_z-\tau)^{m_2}}\right]Q_d \tag{39}$$

H_u Fuel lower calorific value;

η_u Combustion efficiency;

$\frac{dx}{d\varphi}$ Combustion law;

$\frac{dx_1}{d\varphi}$ Premix combustion law;

$\frac{dx_2}{d\varphi}$ Diffusion combustion law;

m_1 Premix combustion quality factor;

m_2 Diffusion combustion quality factor;

τ Advance angle of premix combustion;

θ_z Start point of premix combustion;

φ_{zd} Start point of diffusion combustion.

C. Working volume

$\varphi=0$,when the crank at TDC. Change rule of the working volumes as below:

$$V_z=\frac{\pi D^2}{4}\left\{\frac{S}{\varepsilon-1}+\frac{S}{2}\left[\left(1+\frac{1}{\lambda}\right)-\left(\cos\left(\frac{\pi}{180}\varphi+\frac{1}{\lambda}\sqrt{1-\lambda^2\sin\left(\frac{\pi}{180}\varphi\right)}\right)\right)\right]\right\} \tag{40}$$

Changing rates of the working volume:

$$\frac{dV_z}{d\varphi}=\frac{\pi^2 D^2 S}{8\times 180}\left[\sin\left(\frac{\pi}{180}\varphi\right)+\frac{\lambda}{2}\cdot\frac{\sin\left(\frac{\pi}{180}\cdot 2\varphi\right)}{\sqrt{1-\lambda^2\sin^2\left(\frac{\pi}{180}\varphi\right)}}\right] \tag{41}$$

λ Connecting rod length ratio;

ε Compression ratio;

S Piston stroke;

D Cylinder diameter.

IV. Comprehensive Model

The working process of volumetric model uses CA as independent variable to establish equations, as the MVEM using time [14], so we have to convert angle to time form as follow formula (42):

$$dt=6n_e d\varphi \tag{42}$$

As the Fig.7 shown, the input data of cylinder working process are exhaust air pressure p_e, scavenging air pressure p_{im}, oil per cycle g_f, scavenging air temperature T_{im} and engine speed n_e.The data above can be simulated by the MVEM.

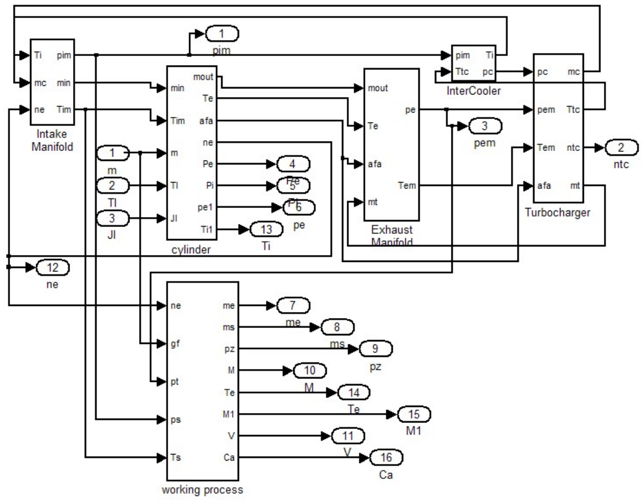

Figure 7. Working process of diesel engine

Working process can be divided into six modules according to piston stroke: compression, combustion, expansion, exhaust and post exhaust [15]. Each module will be invoked according to CA and timing, and different modules have different differential equations [16].

6S60MC marine diesel engine is taken as an example, which parameters are shown in Tab.1.

Number of cylinders	6
Bore(mm)	600
Stroke(mm)	2292
Rated speed(r/min)	105
Rated power(kW)	12240
Turbocharger	TPL80-B12

Table 1. Parameters of 6S60MC diesel engine

First, we set the change rule of speed, and in the paper, the diesel engine operated at rating condition 105r/min till 60s, then a step change of 83.3r/min in 50% load. At last, the speed changed to 95.5r/min in 75% load. The simulation sets time step as 0.002s and simulation time as 300s on 6S60MC shows the dynamic track performance as the Fig.8. From Tab.2, we can see the calculated results by the simulation are in good agreement with measure data. (1 is measured results, 2 is simulation results of the volumetric model, and 3 is the paper results)

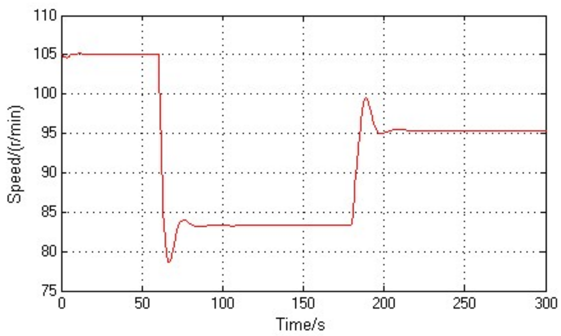

Figure 8. Change rule of speed

load/%	50	75	100
pmax/Mpa (1)	9.5	12.5	14.2
pmax/Mpa (2)	9.49	12.6	14.3
pmax/Mpa (3)	9.2	12.4	14.1
pcomp./Mpa (1)	7.4	10.4	12.9
pcomp./Mpa (2)	7.45	10.4	13.0
pcomp./Mpa (3)	7.4	10.8	12.9

Table 2. Result compares with measure data

Fig.9 shows the indicator diagrams in different loads.

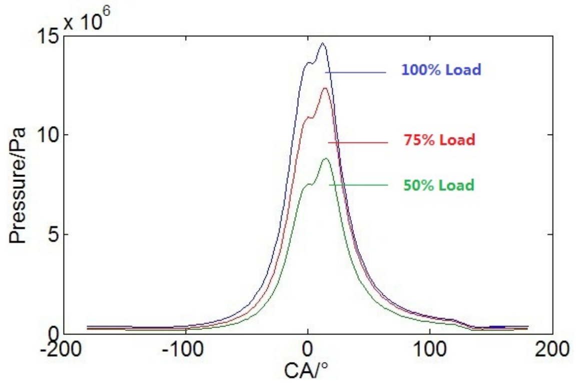

Figure 9. Indicator diagrams

V. P-Φ Indicator Diagram Based On BP Neural Networks

Indicator diagram is an important basis of the perfection degree of the working process as well as the indicator power, dynamic analysis and strength calculation of diesel engine [17]. The paper uses above simulation results and BP networks to design and calculate the p-φ indicator diagram.

A. Data selection and normalization

BP network is mainly used for function approach, pattern recognition, classification, and data compression [18]. The network has three layers: input layer, hidden layer and output layer. 312 data are normalized and distributed between -1 to 1 with premnmx function before training the network, and the outputs need demoralization with postmnmx function.

B. Network training and outputs

The network has one hidden layer with 60 hidden nodes. We use CA as the input layer and cylinder pressure as the output layer, and train the network with training function. According to the results shown in fig.10, the arithmetic has the advantages of high convergence, and the relative error is 0.73% in 100% load. Fig.11 shows an indicator diagram in 100% load.

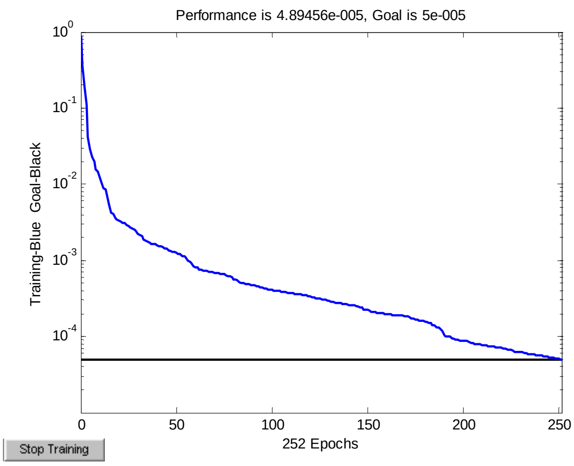

Figure 10. Network error performance at the end of training

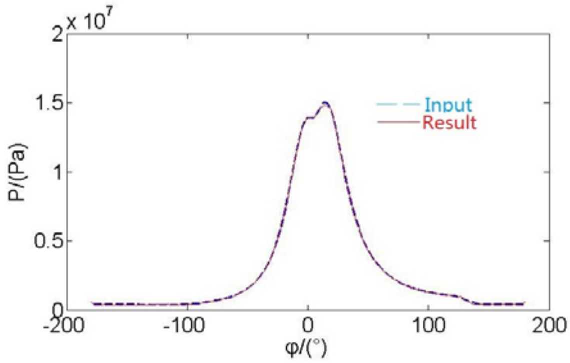

Figure 11. Indicator diagram in 100% Load

VI. Conclusion

The paper simulated working process of marine diesel engine by combining mean value engine model and volumetric model which can reflect not only some average parameters but also pressure and temperature of cylinder in real time with high performance and less error. The measured results was analyzed and compared to the simulation results, and the mathematic model of diesel engine was verified. And then we used output data from comprehensive model to establish a model of the diesel engine on BP neural network.

The application reveals that the network training speed and the precision won't necessarily be enhanced by increasing hidden nodes and hidden layers. The BP network with three layers is a priority when designing BP network and much lower error is achieved by increasing hidden nodes in a certain range sometime. The number of hidden nodes affects the network performance and may cause over-fitting directly. The trained BP neural network can be used into marine simulator to satisfy training requirements.

References

[1] Minghui Kao, John J. Moskwa. “Turbocharged Diesel Engine Modeling for Nonlinear Engine Control and State Estimation”, *ASME Journal of Dynamic Systems Measurement and Control*, 117 (1), pp. 20-30, 1995.
[2] E. Hendricks. “Mean Value Modeling of Large Turbocharged Two-stroke Diesel Engines”, *SAE Paper*, 890564, 1989.
[3] Song Zhu. “A dynamic Model for Automotive Diesel Engines (Thesis for PhD)”, *The University of Wisconsin-Madison*, 1991.
[4] Wang Haiyan, Ren Guang, Zhang Jundong. “Dynamic Modeling of Large Two-Stroke Marine Diesel Engine”, *Transactions of CSICE*, 24(5), pp. 453-458, 2006.
[5] Wang Haiyan, Zhang Jumdong, Zeng Hong. “Modeling and Simulation of a Large-Scale Low-Speed Marine Diesel Engine”, *Journal of Dalian Maritime University*, 32(2), pp. 1-4, 2006.
[6] Piero Azzoni, Giorgio Minelli, Davide Moro. “Air-fuel Ratio Control for A High Performance Engine Using Throttle Angle Information”, *SAE Paper*, 1999-01-1169, 1999.
[7] John R. Wagner, Darren M. Dawson, Liu Zeyu. “Nonlinear Air-to-fuel Ratio and Engine Speed Control for Hybrid Vehicles”, *IEEE Transactions on Vehicular Technology*, 52(1), pp. 184-195, 2003.
[8] J. P. Jensen, A. F. Kristensen, S. C. Sorenson, et al. “Mean Value Modeling of A Small Turbocharged Diesel Engine”, *SAE Paper*: No.910070, 1991
[9] S. E. Lyshevski, A. S. C. Sinha, J. P. Seger. “Modeling and Control of Turbocharged Diesels for Medium and Heavy Vehicles”, *The American Control Conference*, pp. 2688-2692, 1999.
[10] M. Venturini. “Development and Experimental Validation of A Compressor Dynamic Model”, *Journal of Turbomachinery*, 127, pp. 599-608, 2005.
[11] D. T. Hountalas, D. A. Kouremenos. “A Diagnostic Method for Heavy-duty Diesel Engines Used in Stationary Applications”, *Journal of Engineering for Gas Turbines and Power*, 126, pp. 886-898, 2004.
[12] M. Taburri, F. Chiara, M. Canova, Y.-Y. Wang. “A Model-based Methodology to Predict the Compressor Behaviour for the Simulation of Turbocharged Engines”, *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 226(4), pp. 560-574, 2012.
[13] E. Streit, G.L. Borman. “Mathematical SIMULINK of a Large Turbocharged Two-Stroke Diesel Engine”, SAE Paper, 710276, 1971.
[14] Y. H. Zweiri, J. F. Whidborne, L. D. Seneviratne. “Dynamic Simulation of A Single-cylinder Diesel Engine Including Dynamometer Modeling and Friction”, *Proceedings of the Institution of Mechanical Engineers*, 213(4), pp. 239-402, 1999.
[15] Xuedong Wen, Guo He, Baihui Xu. “Modeling and Simulation on Static Characteristics of Pressure Reducing Valve for Ship's Diesel Engine”, *Journal of Shanghai Jiaotong University*, 45(4), pp. 481-485, 2011.
[16] H. Teng. “A Thermodynamic Model for a Single Cylinder Engine with Its Intake/Exhaust Systems Simulating a Turbo-Charged V8 Diesel Engine”, *SAE International Journal of Engines*, 4(1), pp. 1385-1392, 2011.
[17] Tao Wang. “An Improved BP Neural Network Algorithm Embedded With Logistic Mapping and Its Application”, *Advances in Intelligent and Soft Computing*, 115(2), pp. 951-957, 2012.
[18] Kaisheng Huang, Dongliang Wang, Zhihua Lin, Xiangrui Zeng. “Engine Torque Estimation Based on BP Neural Network”, *Advanced Materials Research*, v 403-418, pp. 2848-2851, 2012.

Author Biographies

Cao Hui was born in North China on the 26th of February 1979 and received the Ph.D. in marine engineering automation and computer science from Dalian Maritime University during 2008. Now, he has become a teacher at the same university. His research interests include marine automation and intelligence, computer processing, system modeling and simulation.

Wu Peili was born in South China on the 21th of July 1986 and received the Ph.D. in marine engineering automation and computer science from Dalian Maritime University during 2010. Now, she has become a teacher at the same university. His research interests include marine automation and intelligence, system modeling and simulation, complex network analysis.

Zhang Jundong was born in South China on the 11th of December 1967 and received the Ph.D. in marine engineering automation and computer science from Dalian Maritime University during 1996. Now, he has become a teacher at the same university. His research interests include marine automation and intelligence, computer network, system modeling and simulation.

* This work was supported by “the Fundamental Research Funds for the Central Universities” (Grant No. 2012QN018).