MODELING TURBOCHARGER OF MARINE DIESEL GENERATOR ENGINE IN STEADY LOAD CONDITIONS

Nguyen Quang Vinh[1], Bui Hong Duong[2], Le Van Vang[3]

Abstract: *This paper presents a model of the turbocharger for a marine diesel generator engine which used on ships. Based on the laws of thermodynamics, mean value model and Matlab/Simulink computational environment, the submodelshave been built torepresent for components of theturbocharger, and acomplete model includingthe submodels and their relationshipsdescribes the turbocharger's internal performances. The diesel - turbocharger combination is simulated at some usual operation modes at different steady loads (0% - 100%) with constant speed. At each regime of load, the key characteristics are varied, for example mass flow, turbine speed, efficiency and temperature. In the practical operation, these parameters are important to evaluate engine conditions, however they are very difficult to measure. Therefore,this is meaningful in predicting turbocharger operations and maintenance activities.*

Keywords: Model, turbocharger, marine diesel generator engine, mass flow, efficiency.

1. INTRODUCTION

The turbocharger has been applied on internal combustion enginein the early 20th century, it is particularly installed on ships to increaserapidly the diesel engine power output. Nowadays turbocharging is also critical for emission control and downsizing engines.

It is obvious that turbocharging is the most useful technique to improve diesel engine efficiency and reduce emissions. Because of the complexity and high cost at laboratory, the modelling method is widely applied to research internal performance of the turbocharger and save the cost and time with affordable results.

Many researchers in universities and manufacturers have been studying and improvingthe turbocharger. Watson and Janota(Watson and Janota 1982) summarized the concept of turbocharging and built keyindividual equations for each kind of turbocharger that are the background of many researches. Lars Eriksson and Lars Niesel(Eriksson and Nielsen 2014)described models for the compressor and turbine that fit into the mean value model framework.Turbocharger modelling is considered complex and difficult, therefore, parameterization method based on compressor and turbine flow and efficiency maps are carried out in the study of (Jung et al. 2002) and (Bozza et al. 2011). In recent years, in response to highly emission regulations many technologies, for example VGT (variable geometry turbine), EGR (exhaust gas recirculation), twin turbo, have been developed by manufacturers ABB, Honeywell, BorgWarner, etc and achieved many excellent successes.

There are many different ways of modelling, here is a model starting by conveying the principle theories, then continue with the details of submodels to make the complete model. The main target of this paper is to build a model which can clearly describe thermodynamic relationships among internal components, based on that predict their characteristics at differentmodes of load (0%, 25%, 50%, 75%, 100% load). The object of simulation is a diesel generator engine which used on ships. The accuracy of model can be checked with the compressor map which is provided by the supplier.

2. BACKGROUND

For modelling in this paper, some main concepts are background that presents as below.

Quasi-steady method: Following this method, in

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specific system aerodynamic characteristics (velocity, temperature, pressure, density) only are the function of time and one space co-coordinate.

Laws of thermodynamics:They are widely applied in science and in this paper, they are the foundation to set up the formulas.

Mean value model: The parameters and variables are considered average values in the model.

3. MODEL STRUCTURE

3.1. Cylinder model

Pressure and temperature of gas through the cylinder:Based on the first law of thermodynamics of the gases inside the combustion chamber, equations of the fuel burning process (Miyamoto et al. 1985) and the thermo-exchange with the combustion chamber wall (Woschni 1967). The energy equation applied to the cylinder, canbe rewrited as:

$$\frac{dp}{d\varphi} = -k_\gamma \frac{p}{V}\frac{dV}{d\varphi} + \frac{(k_\gamma - 1)}{V_{cyl}}\left(\frac{\partial Q_{in}}{d\varphi} - \frac{\partial Q_{loss}}{d\varphi}\right) \quad (1)$$

$$\frac{dT}{d\varphi} = \frac{dp}{p\, d\varphi} + \frac{dV}{V\, d\varphi} \quad (2)$$

Where p (N/m^2), V(m^3)- gas pressure and cylinder volumetric correspond to crank angle φ, $\partial Q_{in}/d\varphi$ - heat release rate while burning fuel; $\partial Q_{loss}/d\varphi$ - exchange heat through cylinder wall; $k\gamma = c_p/c_v$ – ratio of specific heats.

To solve system of equations (1) &(2), some key boundary conditions used (heat law of fuel burning, law of heat transfer, intake and outlet mass flow of cylinders).

Law of fuel burning process.

The increment heat ∂Q_{in} is calculated as:

$$\frac{\partial Q_{in}}{d\varphi} = Q_{in}\frac{dx}{d\varphi} \quad (3)$$

Where, the cumulative burn fraction of fuel (x) is based on double Weibe's formula (Miyamoto et al. 1985)

$$\frac{dx}{d\varphi} = \frac{dx_1}{d\varphi} + \frac{dx_2}{d\varphi} \quad (4)$$

$$\frac{dx_1}{d\varphi} = \frac{Q_p}{Q_{in}} a(m_p + 1)\frac{1}{\varphi_p}\left(\frac{\varphi - \varphi_s}{\varphi_p}\right)^{-a(\frac{\varphi-\varphi_s}{\varphi_p})^{m_p+1}} \quad (5)$$

$$\frac{dx_2}{d\varphi} = \frac{Q_d}{Q_{in}} a(m_d + 1)\frac{1}{\varphi_d}\left(\frac{\varphi - \varphi_s}{\varphi_d}\right)^{-a(\frac{\varphi-\varphi_s}{\varphi_d})^{m_p+1}} \quad (6)$$

Where dx/dφ combustion law; $dx_1/d\varphi$ premix combustion law; $dx_2/d\varphi$ diffusion combustion law; aWeibe efficiency factor; m_p premix combustion quality factor; m_d diffusion combustion quality factor; φ_s start of combustion; φ_p duration of premix combustion; φ_d duration of diffusion combustion. φ_s is the start of ignition angle of injected fuel, which in turn is affected by the time of ignition delay τ_i (s).

Heat transfer law. Newton model (Ferguson and Kirkpatrick 2015)is used to calculate the heat transfer through surface of the combustion chamber:

$$Q_{loss} = hA(T - T_w) \quad (7)$$

Where, h- heat transfer coefficient (W/m^{2}K), A - exposed combustion chamber surface area (m^2); T- temperature of the cylinder gas (^{0}K); T_w - cylinder wall temperature (^{0}K).

Heat transfer coefficient h is given by Woschni(Woschni 1967):

$$h = 3.26p^{0.8}U^{0.8}b^{-0.2}T^{-0.55} \quad (8)$$

Where, p - gas pressure; b - cylinder bore; T - gas temperature; U - gas velocity.

During combustion and expansion, gas velocity is given by:

$$U = 2.28\bar{v}_p + 0.00324\frac{V_d T_{in}}{p_{in} V_a} p \quad (9)$$

Where, v_p - mean piston velocity (m/s); V_d - displacement volume (m^3); T_{in}(K), p_{in}(N/m^2), V_a (m^3)- gas temperature, gas pressure and cylinder volume at bottom dead center;

Intake and outlet mass flow. According to (Heywood 1988),the intake and outlet mass flowof cylinder calculated as

$$m_{air} = \rho_{air}\frac{p_{in}}{R_{air}T_{in}} \quad (10)$$

$$m_{out} = m_{air} + m_{fuel} \quad (11)$$

Where ρ_{air}(kg/m^3) the density of the intake air, R_{air}(J/kg.K)the gas constant of intake air, $m_{fuel}$the mass of injected fuel (kg).

3.2. Turbinemodel

The turbine is a kind of machine, in which the exhaust gas releases the energy on the turbine blades. The model of turbine includes submodels: efficiency, exhaust mass flow, nozzle vanes, and power.

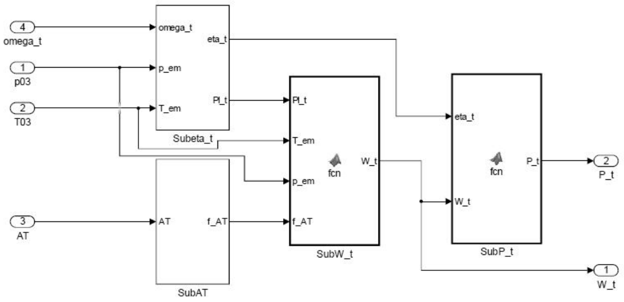

Figure 1. Turbine layout

Turbine efficiencysubmodel

The turbine efficiency depends on the blade speed ratio (BSR). The BSR is defined as the ratio of the rotor tip velocity to the velocity that would be achieved by the gas following isentropic expansion from the inlet conditions to the pressure at the exit from the turbine(Watson and Janota 1982).

$$BSR = \frac{\omega_t R_t}{\sqrt{2c_{pe}T_{03}\left(1-\pi_t^{\frac{k_e-1}{k_e}}\right)}} \tag{12}$$

Where R_t the turbine blade radius (m), ω_t is the turbine speed (rad/s), T_{03} the temperature at inlet turbine (K).

The efficiency turbineη_t varies with BSR following parabolic curves, therefore, it can be expressed by a quadric function in BSR(Eriksson and Nielsen 2014)

$$\eta_t = \eta_{t,max}\left(1-\left(\frac{BSR - BSR_{t,max}}{BSR_{t,max}}\right)\right) \tag{13}$$

Where $\eta_{t,max}$ is the maximum of efficiency at $BSR_{t,max}$ ($\eta_{t,max}$, $BSR_{t,max}$ are tuning parameters)

The exhaust mass flowsubmodel

The exhaust mass flow rate can be expressed in a function of the temperature, pressure at the inlet p_{03}, the function of pressure ratio $f(\pi_t)$, the function of the nozzles cross sectional area $f(A_T)$. It is assumed that the mass flow rate through the nozzles is the same as passing through the turbine, the equation of turbine mass flow rate presents as below

$$\dot{m}_t = A_{Tmax}\frac{p_{03}}{\sqrt{R_e T_{03}}} f_{\pi_t}(\pi_t) f(A_T) \tag{14}$$

Where R_e is the gas constant; $k_e = c_p/c_v$; A_{Tmax} is the maximum of nozzles cross sectional area;$f_{\pi t}(\pi_t)$, $f(A_T)$ are sub functions of π_t, A_T which are calculated below.

The nozzlessubmodel

The nozzles are the components of turbine, the exhaust gas must pass through it before impacting on the blade. To describe behaviour of nozzles, we use the sub model A_{eff} ($A_{eff}$is the cross section of nozzles), $A_{eff} = A_{Tmax}.f(A_T)$.According to(Jung et al. 2002), the equation $f(A_T)$ can be modelled as:

$$f(A_T) = c_1 + c_2\sqrt{1-\left[\frac{A_T - c_3}{c_4}\right]^2} \tag{15}$$

Where c_1, c_2, c_3, c_4 are tuning parameters.

3.3. Compressor

The model of centrifugal compressor is mainly evaluated in terms of pressure p_{02}, efficiency η_c, turbine speed $\omega_t$and air mass flow rate $\dot{m}_c$ (Figure 2).

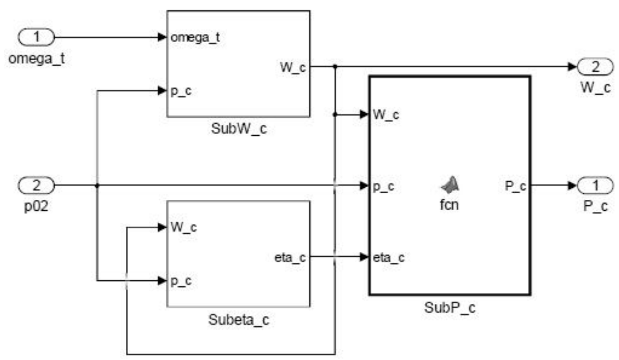

Figure 2. Compressor layout in Simulink environment

The outlet temperature of the compressor T_2 is modelled as (Yin et al. 2017)

$$T_2 = \frac{1}{\eta_c} T_{amb}\left(\pi_c^{\frac{k_a-1}{k_a}} - 1\right) \tag{16}$$

Where, $T_{amb}$is the ambient temperature, η_c is the compressor efficiency, k_a is the intake gas specific heat ratio, π_c is the compressor ratio.

Compressor efficiency submodel

In the practice, compressor efficiency is modelled through the mass flow coefficient dimensionless Φ(Eriksson and Nielsen 2014)

$$\Phi = \frac{\dot{m}_c}{n_t D^3}\frac{RT_{amb}}{p_{amb}} \quad (17)$$

Quadratic form Compressor Efficiency

$$\chi(\Phi, n_c) = \begin{bmatrix} \Phi - \Phi_{max} \\ n_c - n_{c,max} \end{bmatrix} \quad (18)$$

$$\eta_c(\chi) = \eta_{c,max} - \chi^T Q_\eta \chi$$

Where $Q\eta \in \Re 2x2$ is a symmetric and positive definite matrix. Φ_{max},n_c, n_{max} and the elements in Q are tuning parameters; T_{amb}, p_{amb} are the ambient temperature and pressure .

3.4. Turbine and Compressor balance

Figure 3 shows the pressures, temperatures, mass flow rate at the inlet and outlet of turbocharger and describes the relationships in Simulink environment.

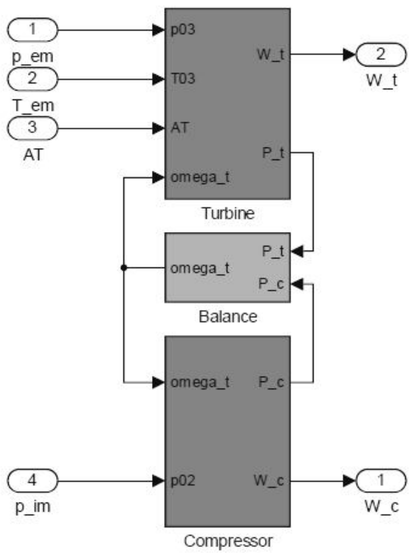

Figure 3. Turbocharger layout in Simulink environment

The laws of thermodynamics are used in this situation. The turbine and the compressor are mounted on a common shaft, the energy is transferred between two components. The balance of the turbine and the compressor is expressed as(19)

$$P_t \eta_m = P_c \quad (19)$$

Where P_t, P_c are turbine and compressor power respectively; η_m is the friction coefficient.

3.5. Engine Load (OutputPower)

The engine load is the output power (P_b), which can be calculated through indicated and friction power. Indicated Power is defined as the power developed by combustion of fuel inside the engine cylinder

$$W_i = \int p\, dV \quad (20)$$

For diesel engine, indicated power is

$$P_i = i.W_i.n/2 \quad (21)$$

Where i is the number of cylinders; n is the engine speed (rpm)

Outputpower, P_w, is the rate which work is done; therefore, it is less than the indicated power due to frictional loss.

$$P_w = P_i - P_f \quad (22)$$

Where P_f is the friction power (loss power).

4. RESULTS AND DISCUSSIONS

Simulink model:

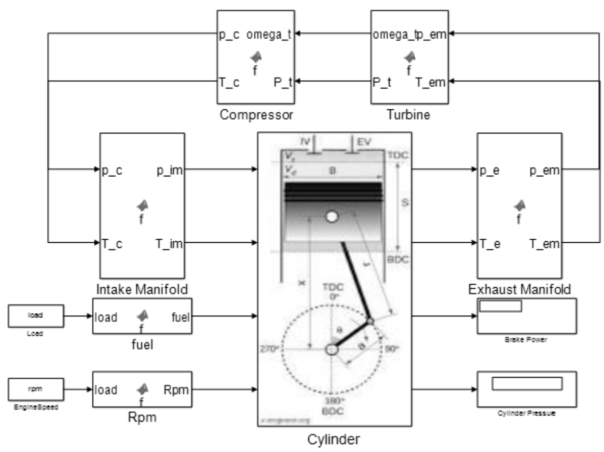

Figure 4. Simulink model of diesel and turbocharger

Object of modelling: The object is a marine diesel generatorengine,Yanmar S185L, which used on ships, assuming that engine speed is unchanged, 900 rev/min. Its turbocharger is VTR160, manufactured by ABB Group.

Table 1. Engineand turbochargerparameters

Parameters	**Values**
Numbers of cylinder	6
Bore x stroke (mmxmm)	185 x 230
Nominal speed (rpm)	900
Nominal power (kW)	447
Compressor diameter (mm)	170
Turbine diameter (mm)	172

4.1. Speed and mass flow rate

The turbocharger speeds (rev/min)and mass flow varyaccording to load modes, whichis shown in Figure 5 and Figure 6. The speed of turbine varies from 30.000 rev/min at idle load to 57.000 rev/min at full load.

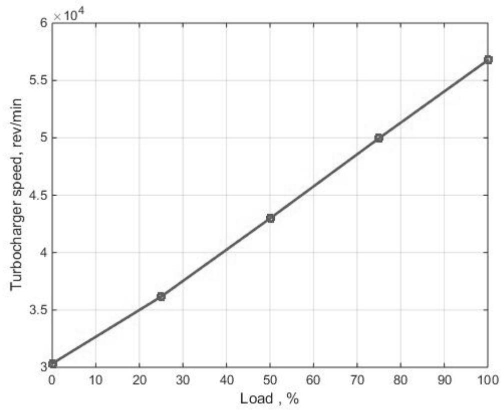

Figure 5. Turbospeed varies to loads

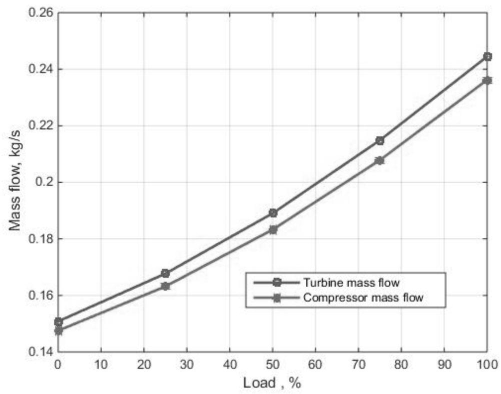

Figure 6. Mass flow rates diagram

4.2. Efficiency and temperature

The efficiencies and temperatures of turbine and compressor are shown in Figure 7 and Figure 7. At the low load conditions, the compressor efficiency is very low, it increases when the load increases. The turbine efficiency is not changed much at every load.The range of turbine's inlettemperatureis (620 $\div 830^0$K), the compressor's outlet temperature is not changed much (300 $\div 345^0$K)

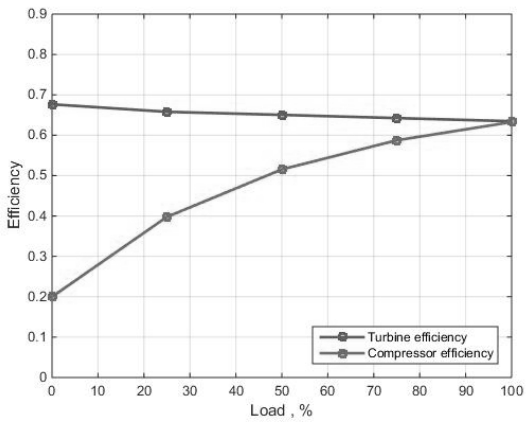

Figure 7. Turbochagerefficiency

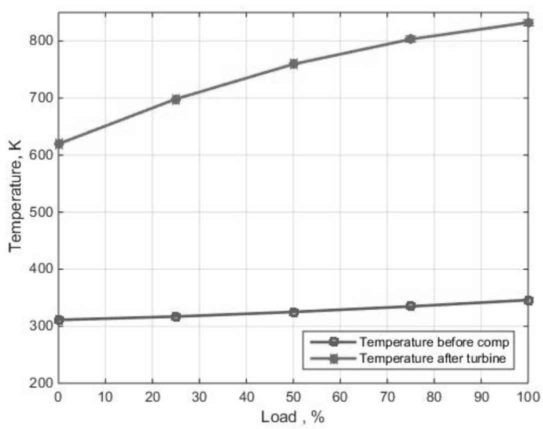

Figure 8. Inlet and outlet temperature

5. CONCLUSION

The paper has achieved some results:

Summarize the key thermodynamic characteristics of a diesel engine and its turbocharger.

Build a model in Matlab/simulink environment, which combines the diesel and the turbocharger. The relationship between them is very complex, at this situation we considered their working at the steady load modes. The model has described the contact between internal characteristics and simulated their thermodynamic processes.

The results are presented and discussed enlightening the key parameters for the turbocharger operation at different of steady loads, for example(speed, efficiency, mass flow, temperature).In the practical, these parameters are very difficult to measure, therefore it is meaningfulto save time and cost inmaintenance activities and operations.

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Tóm tắt:

MÔ HÌNH CỤM TĂNG ÁP TUA BIN KHÍ CỦA ĐỘNG CƠ DIESEL TÀU THỦY LAI MÁY PHÁT ĐIỆN TRONG CÁC ĐIỀU KIỆN TẢI ỔN ĐỊNH

Bài báo trình bày một mô hình cụm tua bin tăng áp của động cơ diesel tàu thủy lai máy phát điện. Dựa trên các định luật nhiệt động, mô hình giá trị trung bình và môi trường tính toán Matlab/Simulink, các mô hình con được thiết lập đặc trưng cho các bộ phận của bộ tăng áp. Sau đó một mô hình hoàn chỉnh bao gồm các mô hình con và mối liên hệ giữa chúng được xây dựng để mô tả các đặc tính bên trong bộ tăng áp. Tổ hợp diesel – turbocharger được mô phỏng tại các chế độ hoạt động thông thường với các điều kiện tải khác nhau (0% - 100%) và tốc độ không thay đổi. Với mỗi chế độ của tải, các đặc trưng cơ bản thay đổi theo, ví dụ như lưu lượng, tốc độ tua bin, hiệu suất và nhiệt độ. Trong thực tế khai thác, những thông số này rất quan trọng để đánh giá tình trạng động cơ, tuy nhiên lại rất khó đo kiểm được. Vì vậy, điều này rất có ý nghĩa trong dự đoán hoạt động tăng áp và quá trình bảo dưỡng nó.

Keywords: Mô hình, tăng áp, động cơ diesel lai máy phát điện, lưu lượng, hiệu suất.

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