ISSN: 2472-3703 DOI: 10.6911/WSRJ.202005_6(5).0027

Effect of Injection Strategy on Combustion and Emissions Performance of Marine Medium-Speed Dual-Fuel Engines

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Abstract

In order to find out the important factors affecting the combustion emission performance of marine medium-speed dual-fuel engines and their changing trends. In this paper, a MAN L23/30H medium-speed engine is used as a prototype, and the commercial CFD software FIRE is used to perform numerical calculations under certain gas operating conditions. The chemical kinetic mechanism constructed by importing CHEMKIN is combined with basic models such as spray breaking and turbulent flow The three-dimensional working process of the engine was constructed. Through numerical simulation, the analysis focused on the influence of natural gas replacement rate and fuel injection timing on the combustion emissions of Diesel /LNG dual fuel engines. The results show that, unlike the pure diesel mode, in the dual fuel mode, the injection timing has basically the same effect on the parameters in the cylinder at different natural gas replacement rates. In dual fuel mode, the engine's power is more sensitive to the injection timing, and the power reduction caused by the injection delay is greater than in pure diesel mode. Under the high natural gas replacement rate, the diffusion combustion of the pilot diesel will disappear, and the exothermic rate curve and the n-dodecane consumption curve show a single peak. With the increase of the natural gas replacement rate, the power of the marine medium-speed engine is gradually improved, and the injection timing with the best power is approaching the top dead center.

Keywords

Medium-speed marine engine; Numerical simulation; Injection strategy; Fuel substitution rate; Injection timing.

1. INTRODUCTION

As an important factor affecting combustion and emissions, the injection strategy, which is one of the methods for controlling the reaction in the combustion chamber, has gradually attracted more attention. Compared with the traditional combustion mode, two different fuels in the dual fuel mode will have a greater impact on combustion due to their respective physical and chemical properties. The purpose of studying the injection strategy is to better reveal its reaction principle and optimization significance. In the past ten years, scholars around the world have proposed various control methods based on this purpose. Many researchers have demonstrated that Homogeneous Charge Compression Ignition (HCCI) and Premixed Homogeneous Charge Compression Ignition (PCCI) are in-cylinder emission reduction technologies that reduce both NOx and soot [1-2]. Due to the characteristics of the combustion requirements of dual fuel engines, the PCCI mode is more widely used in dual fuel engines. PCCI combustion mode is usually used in combination with exhaust gas recirculation and fuel multi-

ISSN: 2472-3703 DOI: 10.6911/WSRJ.202005_6(5).0027

injection technology, which can effectively control ignition delay, combustion speed and phase [3]. The dual fuel premixed intake compression ignition concept is considered to be more positive in controlling the combustion characteristics of the engine. According to the appeal method, joint control can extend the working range to high load conditions in the dual fuel mode, and has a higher fuel replacement rate, reducing pollutant emissions to meet stricter emission regulations.

2. THE ESTABLISHMENT OF THEORETICAL MODEL

2.1. Mathematical Model

A variety of turbulence models, droplet breaking, evaporation models, and combustion models are built into FIRE software. This article follows the optimization model summarized by the predecessors. The turbulence model uses the four-equation k-ζ-f model [4]. The model has high accuracy and stability, and the calculation time is only 15% longer than the k-ε model of the double equation. The evaporation model uses the Dukowicz model [5], which assumes that: 1) the droplets are spherically symmetric; 2) a quasi-stable vapor film is wrapped around the droplets; 3) the watershed around the droplets have consistent physical properties; Liquid-Vapor Thermal Equilibrium on Drop Surface. Based on these assumptions, the temperature change rate of the droplet is determined by the energy balance equation:

$$m_d c_{pd} \frac{dT_d}{dt} = L \frac{dm_d}{dt} + Q \tag{1}$$

Q is the conductive heat flow from the working fluid in the cylinder to the surface of the droplet, and the expression is:

$$Q = \alpha A_s (T_\infty - T_s) \tag{2}$$

m_d : Droplet quality [kg]

c_{pd} : Droplet specific heat capacity [J/(kg•K)]

T_d : Droplet temperature [K]

A_s : Droplet surface area [m2]

T_∞ : Distant droplet temperature [K]

T_s : Drop surface temperature [K]

L : Latent heat of evaporation [J/kg]

α : Heat transfer coefficient [W/(m2•K)]

The crushing model uses the WAVE model [6], which considers that the growth rate of the initial disturbance on the droplet surface is determined by the physical properties of the incident fuel and its kinetic parameters and the physical properties of the watershed around the droplet.

The mathematical model of three-dimensional CFD coupled chemical reaction kinetics was used for further research on combustion. The diesel / natural gas dual fuel reaction mechanism is taken from the literature [7]. The mechanism contains 143 components and 746 chemical reactions. It has high accuracy in the prediction of combustion and emissions of dual fuel engines. Used to characterize natural gas, n-heptane / n-butylbenzene is used to characterize diesel oil, and the proportion of each component is determined according to the actual fuel.

Control equation of coupled chemical reaction kinetics:

(1) Component transport equation:

$$\frac{\partial}{\partial t}(\rho y_k) + \frac{\partial}{\partial x_j}(\rho u_j y_k) = \frac{\partial}{\partial x_j}(\Gamma_k \frac{\partial y_k}{\partial x_j}) + S_k \tag{3}$$

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In the formula: ρ is the density; y_k is the mass fraction of the k component; k = 1, 2 ⋯ K, where K is the total number of components; S_k is the chemical reaction quality source term of component k.

(2) Energy source term:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho u_j h) = \frac{\partial}{\partial x_j}(\frac{\lambda}{c_p}\frac{\partial h}{\partial x_j}) + \frac{\partial}{\partial x_j}(\tau_{ij} u_i) + \frac{\partial p}{\partial t} + S_h \tag{4}$$

In the formula: h is the total stagnation enthalpy; λ is the thermal conductivity; c_p is the specific heat capacity at constant pressure; τ_{ij} is the shear force tensor; S_h is the heat of chemical reaction. In the calculation of the three-dimensional fluid dynamics coupled chemical reaction, the source term S_k in the component transport equation (3) and the source term S_h in the energy equation (4) need to be additionally solved by The single-track chemical kinetic model was introduced into the CFD fluid calculation program. Treat each unit in the fluid calculation grid as a single-curve reactor, and add a single-zone chemical kinetic model with the same chemical calculation reaction time step in each flow calculation time step, and then output the calculation results to the flow control. Source term of the equation [8].

(3) Component source term:

$$S_k = \frac{\rho^{n+1} y_k^{n+1} - \rho^n y_k^n}{\Delta t} V_{cell} \tag{5}$$

In the formula: y_k^n is the initial mass fraction of the k component at the nth time step; y_k^{n+1} is the mass fraction of the k component after the time step Δt; y_k^{n+1} is obtained from a single-zone gas phase chemical kinetic model; V_{cell} is Calculate the volume of the grid.

(4) Energy source term:

$$S_k = \sum_k^K S_k H_k \frac{1}{W_k} \tag{6}$$

In the formula: S_h is the formation enthalpy of the k component at a given temperature; S_h is the relative molecular mass of the k-th component.

2.2. Introduction to Chemical Reaction Kinetic Models

2.2.1 Mechanism of diesel / methane dual fuel reaction

In the traditional diesel combustion simulation, the cetane number (CN) of n-heptane is about 55, which is equivalent to the cetane number of diesel in Europe and Japan. Therefore, various versions of the n-heptane oxidation model are widely used in the calculation of diesel ignition process. However, compared with diesel, another characteristic fuel, n-dodecane has a carbon chain structure closer to that of diesel than n-heptane, which has a shorter carbon chain than actual diesel. Diesel molecular clusters typically have between 10 and 25 carbon atoms. In addition, n-dodecane has the boiling characteristics of medium-range diesel fuel, so it can better simulate the fuel-air mixing process related to spray evaporation. Therefore, n-dodecane is more suitable as an alternative component of diesel than n-heptane. This is essential for numerical calculations. The physical and chemical properties of related fuels are shown in Table 1. At the same time, because diesel engine combustion is controlled in a hybrid manner. Engines using high volatile n-heptane cannot effectively capture the fuel mixing process, resulting in

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simulation results that are too short for the length of the liquid column of the real diesel fuel phase. The Engine Combustion Network (ECN) recently conducted a series of experiments on the spray and combustion characteristics of n-dodecane, carefully determined the boundary conditions required for three-dimensional spray combustion simulation, and noted uncertainty. These experiments further develop spray and combustion models for diesel engine applications. However, a chemically kinetic model of n-dodecane (combined with low-temperature chemistry) that can be coupled with the spray combustion model is still lacking.

Table 1. The physical and chemical properties of reaction substitutes and actual fuels

Fuel properties	#2 Diesel	DMA	N-heptane	N-dodecane
15°C Density	843kg/m3	890kg/m3	613kg/m3	752.1kg/m3
40°C Viscosity	2.35 mm2/s	1.5~6 mm2/s	1.76 mm2/s	1.5 mm2/s
Cetane number	47	40	56	87
Low calorific value	42.975MJ/kg	42.8MJ/kg	44.6MJ/kg	44.17MJ/kg
Sulfur content	9ppm	0.015%	0	0

In the past ten years, research on the kinetic model of n-dodecane has been carried out abroad. Westbrook [9] and others later developed the details of n-paraffins with low temperature chemistry from n-octane to n-hexadecane. Chemical kinetics mechanism, including 2755 components and 11,173 chemical reactions. These mechanisms are not suitable for analysis of combustion conditions, so different chemical kinetic models have been developed for different uses of the mechanism. The dual-fuel combustion mechanism of n-dodecane / methane blending in this paper is based on the semi-detailed mechanism proposed by You et al. [10]. The original semi-detailed mechanism includes 171 components and 1306 chemical reactions. Through reduction methods such as reaction flow analysis and isomer agglomeration, the mechanism is finally reduced to 54 components and 269 chemical reactions. The skeleton mechanism was verified by experiments such as ignition delay, fully stirred reactor, and laminar flame velocity. It was originally used in three-dimensional numerical simulation of compression ignition engines.

2.3.2 Optimization of the reaction mechanism

In this paper, the 13-step NOX emission sub-mechanism proposed by ProfessorGolovitchev of Chalmers University in Sweden is added on the basis of the Yao mechanism, and related parameters are modified. In order to verify that the addition of NOX-related sub-mechanisms has no effect on the ignition of n-dodecane, the ignition delay calculation of the integration mechanism was performed through CHEMKIN 4.1 and the experimental data were compared. It was found that the integration mechanism was basically consistent with the experimental data. Figures 1a and b are When the equivalent ratio is 0.5 and 1, the calculation of the integration mechanism is compared with the experimental ignition delay.

ISSN: 2472-3703 DOI: 10.6911/WSRJ.202005_6(5).0027

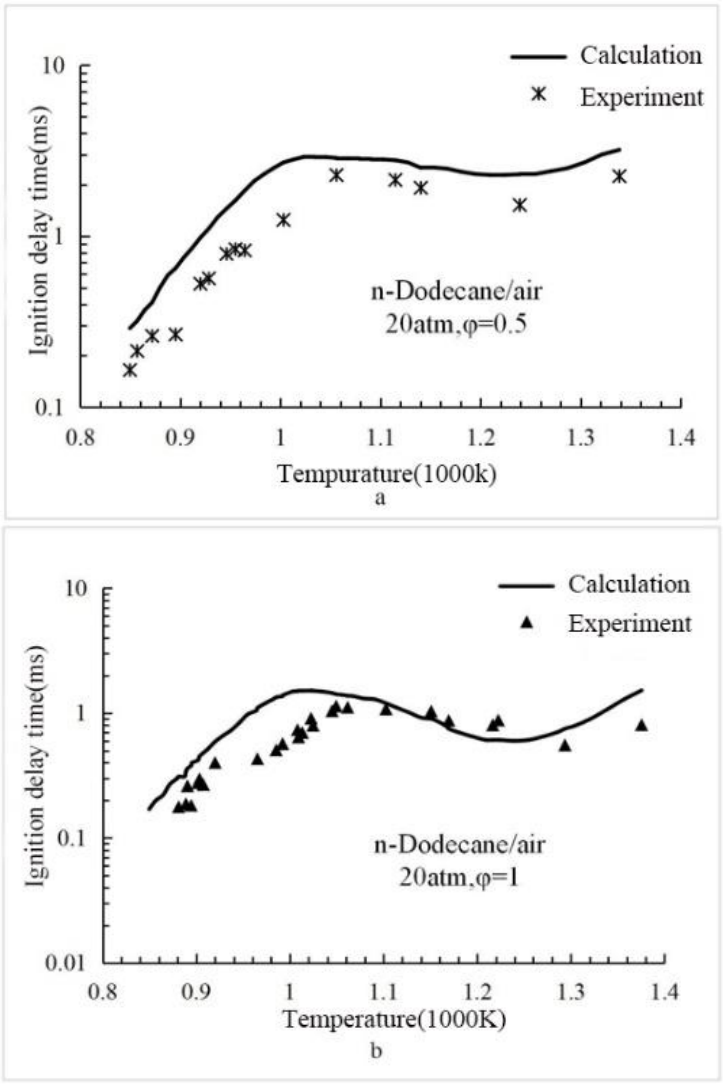

Figure 1. Comparison of integration mechanism calculation and experimental ignition delay under different equivalence ratios (a. Equivalence ratio is 0.5; b. Equivalence ratio is 1)

It can be seen from Figure 1 that the integration mechanism is in good agreement with the experimentally measured ignition delay, which proves that the integration mechanism is suitable for calculation of n-dodecane / methane dual fuel combustion.

3. ENGINE TECHNICAL PARAMETERS AND MODEL CONSTRUCTION

3.1. Structural Parameters of Prototype Engine

Since the relevant data of the MAN L23 / 30DF marine dual-fuel engine has not been made public, the calculation model of this article is based on the L23 / 30H diesel engine, and the MCR = 900rpm and full load conditions are selected as the calibration conditions. Table 2 is the relevant structural parameters of the original diesel engine under this operating condition.

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Table 2. Structural parameters of prototype diesel engine

Parameter name	Parameter value
Strokes	4
Bore/Stroke(mm)	225/300
MCR power(kW)	175
MCR speed(rpm)	900
Compression ratio	13.5
Single cylinder piston area(cm2)	398
Single-cylinder piston sweep volume(L)	11.9
Piston speed(m/s)	9.0
Maximum combustion pressure (bar)	150
MCR specific fuel consumption(g/kWh)	193
Indicated mean effective pressure(bar)	19.6

The injector uses a structure of 8 × Φ0.33mm. Since the injector is located in the center of the cylinder head and the injection holes are evenly distributed in the axial direction, the calculation model can be simplified to a 1/8 combustion chamber model.

3.2. Meshing and Initial Conditions

Before numerical simulation, the combustion chamber needs to be meshed. The geometric model of the combustion chamber is constructed by AUTO CAD software, and the structural parameters of the combustion chamber are shown in Figure 2. The quality of the grid determines the convergence of the calculation and the speed of the computer's calculation speed. Since the simulated injector uses a circumferential distribution of 8 nozzles, a one-eighth model of the combustion chamber is used for the calculation. . In this paper, after the grid independence test, the FAME tool in Fire software was used to mesh the combustion chamber. The total number of meshes after the division is 54304. There is no poor quality mesh. See Figure 3 for details.

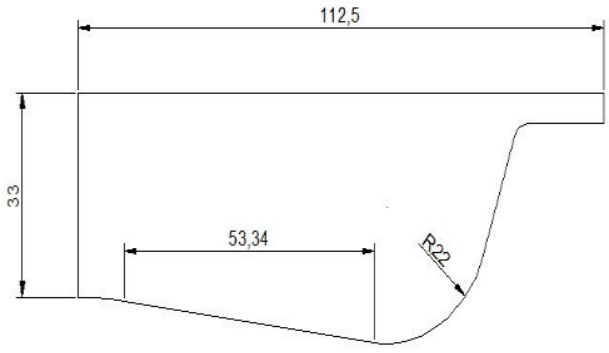

Figure 2. Combustion chamber structure

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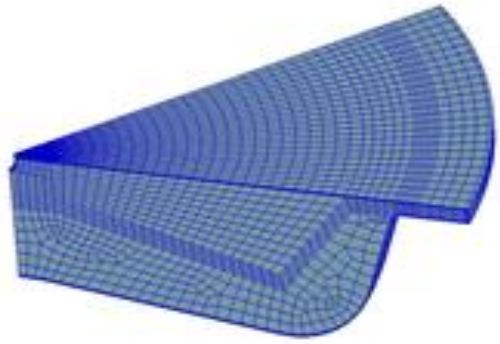

Figure 3. Geometric grid mechanism

Taking into account the computer's operating capacity, the numerical calculations in this paper ignore the inlet and exhaust ducts. The starting and ending points of the numerical calculations are the closing time of the intake valve and the opening time of the exhaust valve. The time measurement method uses the crank angle (°CA), The top dead center time corresponds to 720°CA, the intake valve closing time is 156°CA before the top dead center, and the exhaust valve opening time is 126°CA after the top dead center. When the intake valve is closed, the temperature of the gas in the cylinder is 328K and the pressure is 3.1bar. After calculation, the original single-cylinder circulating fuel supply is 1.25g/cycle, so that the fuel supply per cycle is 0.156g in the calculation domain.

4. MODEL VERIFICATION AND PARAMETER ANALYSIS OF PURE DIESEL OPERATING CONDITIONS

4.1. Model Validation

The simulation calculation uses the in-cylinder pressure curve of the L23/30H diesel engine under 900r/min and full load conditions as the calibration condition. Adjust the heat transfer on the wall according to the marine diesel water cooling system. The initial conditions such as the intake air temperature and intake air pressure used in this calculation were obtained through experiments. The correction is performed according to the experimental model setting requirements. The comparison between the simulation calculation results and the in-cylinder pressure curve measured in the test is shown in Figure 4. The pressure trend in the cylinder is completely consistent, and the peak error is within 5%. The good matching performance demonstrates the validity of this calculation model.

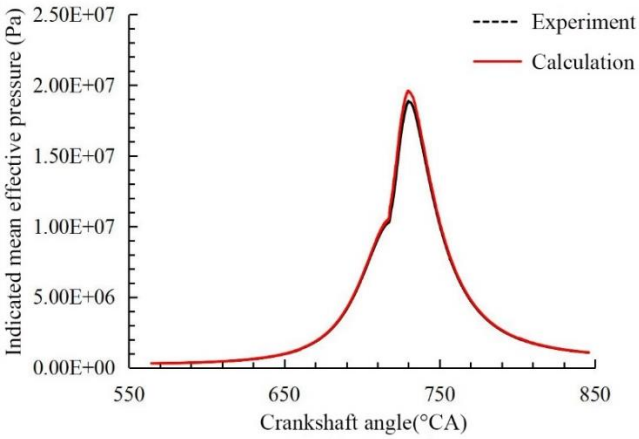

Figure 4. The verification of calculation model of L23/30H diesel engine

ISSN: 2472-3703 DOI: 10.6911/WSRJ.202005_6(5).0027

4.2. Effects of Different Injection Timings on the Combustion Performance of Diesel Engines

In order to study the effects of different injection timing (Start of Injection, SOI) on the combustion and emissions of marine diesel engines. This section analyzes and discusses the change trends of the temperature, pressure, and other parameters about the prototype diesel engine cylinder at four different injection timings (SOI=710,715,720,725˚CA). Figure 5 shows the changes in the parameters of the cylinder with the crank angle (a, pressure; b, temperature; c, heat release rate; d, NO emissions). Figure 6 shows the comparison of the power and fuel consumption of the original diesel engine at different injection timings. It can be seen from the change trend of each parameter in FIG. 5. With the delay of injection timing, the pressure and temperature in the cylinder decrease, the heat release rate lags, and NO emissions decrease. In the cylinder, the decrease in temperature and pressure is basically the same. However, with the delay of the injection timing, the NO emissions which generate before the top dead center change more significantly than that which after the top dead center. It can be seen from Figure 6 that the engine power and fuel consumption at the injection timings of 710˚CA and 715˚CA are basically the same. When the injection time is greater than 715˚CA, the engine power and fuel consumption decreases significantly with the delay of the injection time. The power and fuel consumption of each injection timing are higher than the power of the original engine single cylinder, and the fuel consumption is lower. The main reason for this error is that the calculated n-dodecane is a single-component fuel, which has a higher heating value than diesel, and the calculation ignores the fact that there is still residual exhaust in the cylinder when the engine is running stably. Since the initial calculation conditions are consistent, the above errors don't affect the validity of the calculation. From the calculation results obtained at each injection timing, it is very important to select the injection timing before the top dead center for the engine NOX emission reduction. For this model, the optimal injection timing is around 715˚CA, and The data (712-718˚CA) provided by engine manufacturers are basically the same.

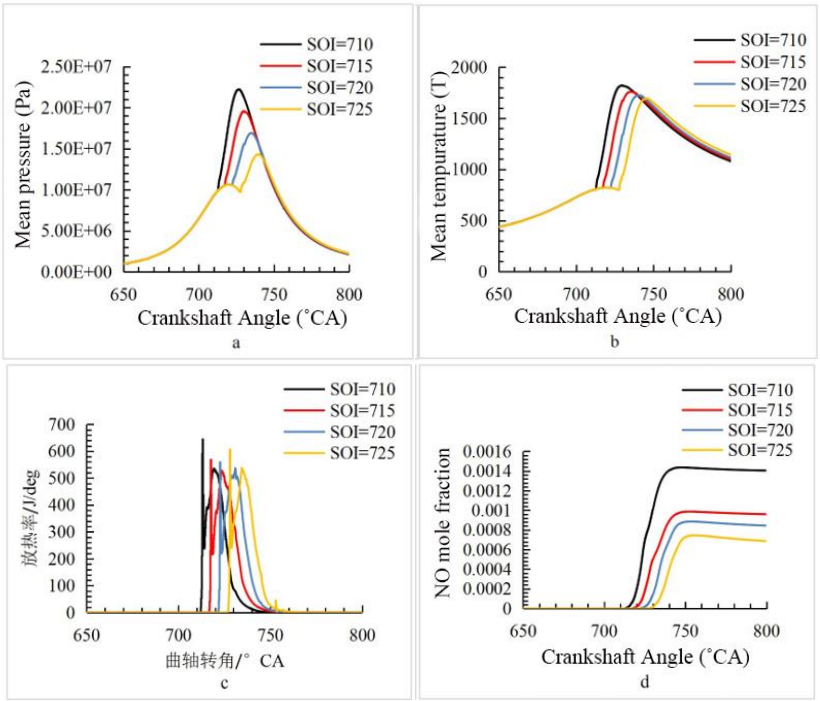

Figure 5. The variation of parameters in the cylinder with crankshaft angle (a, pressure; b, temperature; c, heat release rate; d, NO emissions)

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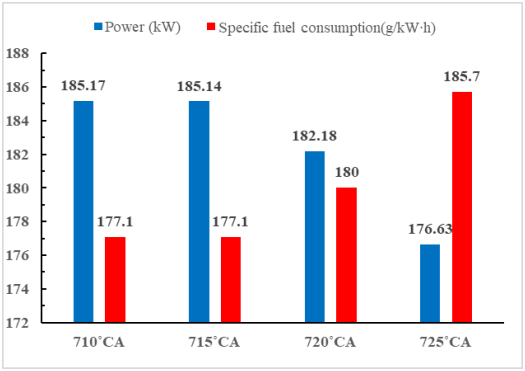

Figure 6. Comparison of engine single-cylinder power and fuel consumption at different injection timings

5. COMBUSTION CHARACTERISTICS AND PARAMETER ANALYSIS IN DUAL FUEL MODE

5.1. Effect of Injection Timing on Combustion Performance at 25% Natural Gas Replacement Rate

This section analyzes that the fuel injection timing of low natural gas replacement rates affects combustion characteristics in the cylinder. Figure 5-1 shows the changes in the parameters of the cylinder with the crankshaft angle (a, pressure; b, temperature; c, n-dodecane consumption rate; d, methane consumption rate; e, heat release rate; f, NO mole fraction). As the injection timing is delayed, it can be seen from the figure that the temperature and pressure are decreasing, and the decrease is basically the same. Taking the crankshaft angle corresponding to the maximum n-dodecane consumption rate as the start of combustion (Start of Combustion, SoC), SoC 710=714.2°CA, SoC 715=718.8°CA, SoC 720=724°CA, SoC 725=729.4°CA, the corresponding ignition delay/flame retardation periods are 4.2°CA, 3.8°CA, 4.0°CA, 4.4°CA. Because the injection regulation and working fluid state haven changed much before and after each injection time, their flame retardation periods are basically the same. At the beginning of combustion, the n-dodecane accumulated during the flame retardation period is rapidly cracked and oxidized. During this period, n-dodecane experienced a chemical reaction process from medium temperature to high temperature. In the medium temperature reaction zone (temperature ≥ 1100K), the β-cracking of free radicals reacts more quickly than the cracked products. The cracking mainly forms H2, C2-C4 olefins and methane. This is also the reason why methane will increase in the initial stage of combustion. Multiple ignitions can significantly improve fuel efficiency. From the content distribution of NO, it can be seen that NO is mainly concentrated in the high temperature region of the front end of the flame. Since NO is sensitive to temperature, NOx emissions can be greatly reduced by controlling the temperature in the cylinder. Figure 5-5 shows the comparison of the single cylinder power of the engine and the indicated mean effective pressure at different fuel injection timings when the natural gas substitution rate is 25%. It can be seen that when the natural gas replacement rate is 25%, the power of the dual fuel engine is equal to that of the prototype diesel engine. The power is the best at 710°CA, but it is more sensitive to the injection timing, and the power loss caused by the injection delay is greater than that of the prototype diesel engine.

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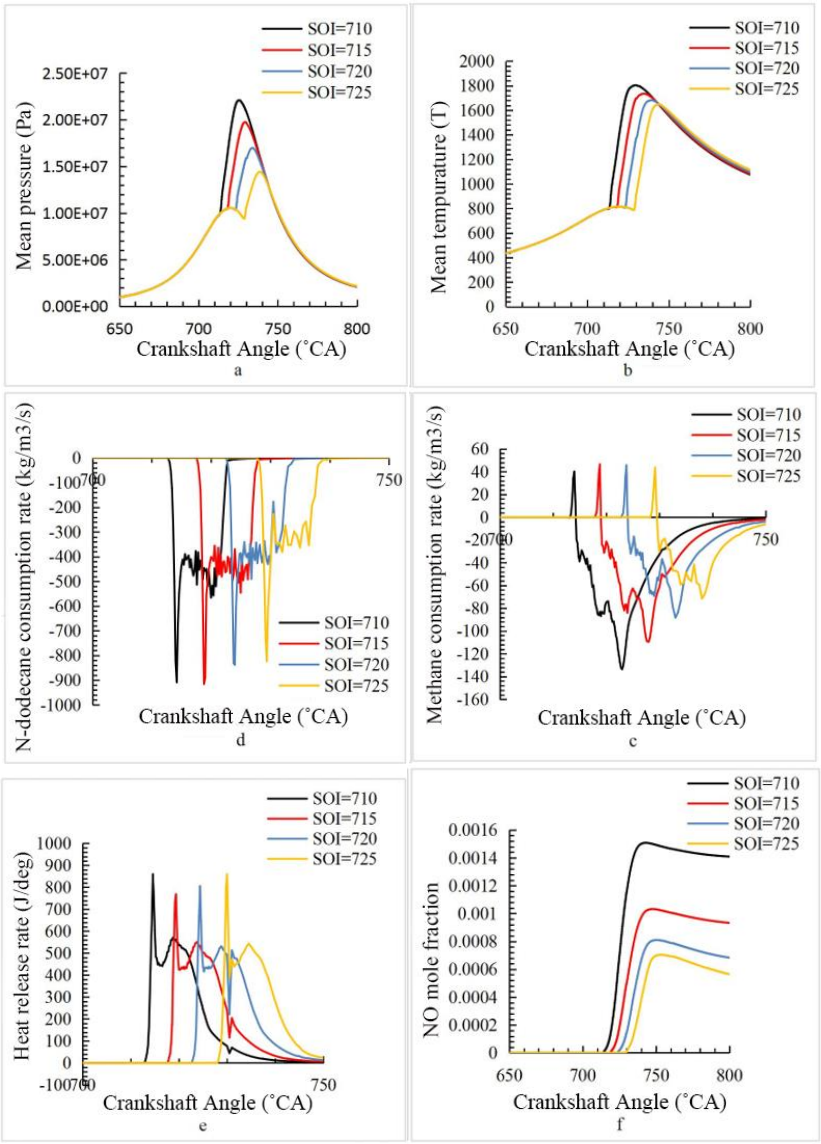

Figure 7. The Variation of parameters with crankshaft angle at 25% natural gas replacement rate (a. Pressure; b. Temperature; c. N-dodecane consumption rate; d. Methane consumption rate; e. Exothermic rate; f. NO mole fraction)

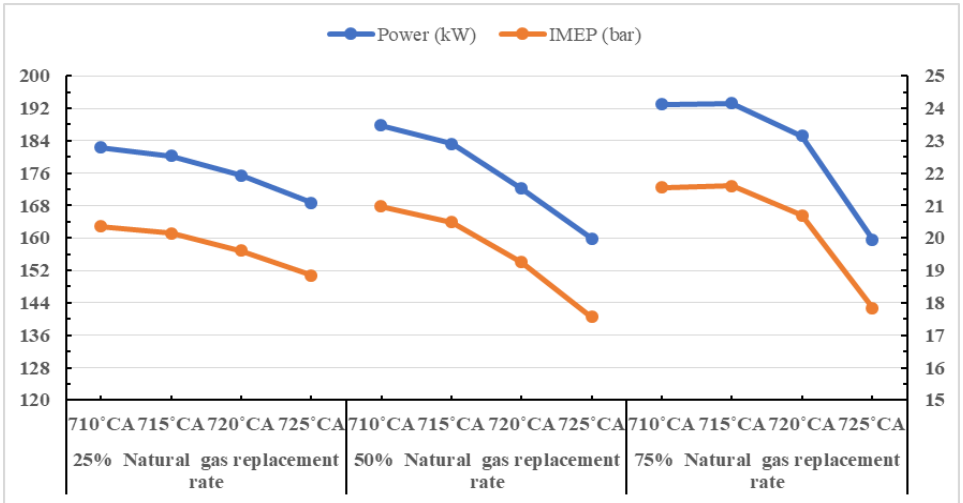

Figure 8. The variation of power and Indicated mean effective pressure at different injection timings and natural gas replacement rates

ISSN: 2472-3703 DOI: 10.6911/WSRJ.202005_6(5).0027

5.2. Effect of Injection Timing on Combustion Performance at 50% Natural Gas Replacement Rate

When the natural gas replacement rate is increased to 50%, the change trend of the pressure and temperature in the cylinder with the crankshaft angle at different injection timings is basically the same as the conditions of 25% natural gas replacement rate. The increase of natural gas replacement rate leads to a reduction in the amount of pilot diesel injection and a shorter injection duration. Because the fuel supply regulation has not changed, the fuel supply amount per unit crankshaft angle is basically the same. The n-dodecane content in the flame retardation period is about the same, so the consumption rate is basically the same. Because of the increase in natural gas content, methane consumption is faster than at 25% replacement rate. As the injection timing is delayed, the methane consumption rate will decrease. Theoretically, when the methane content increases, the flame retardation period will be extended. However, the change in methane content has little effect on the flame retardance period of diesel under the high-pressure oxygen-enriched environment of marine engines, When the natural gas replacement rate is increased to 50% and the fuel injection timing is 710˚CA, the single-cylinder rated power of the engine at this replacement rate is equivalent to that of the original diesel engine. When the timing is postponed to 715, 720, 725˚CA, the indicated power of the single cylinder of the engine drops rapidly, which is lower than that of pure diesel at the same time.

5.3. Effect of Injection Timing on Combustion Performance at 75% Natural Gas Replacement Rate

When the natural gas replacement rate is increased to 75%, the pressure and temperature in the cylinder still decrease with the delay of the injection timing, and the reduction is getting larger and larger. A small amount of n-dodecane is injected into the combustion chamber as a ignition source within a short duration. The fuel injection duration is even shorter than the flame retardation period of n-dodecane, which eventually causes n-dodecane to burn completely in premixed mode. Both the n-dodecane consumption curve and the heat release rate curve showed a single peak. When the fuel is injected before the top dead center (710˚CA, 715˚CA), the methane consumption rate is fast and concentrated, and the reduction of the n-dodecane content also reduces the methane generation rate in the initial combustion. Because the temperature in the cylinder drops, NO emissions gradually decrease with the delay of the injection timing. When the injection timing is 710˚CA, although the average temperature in the cylinder is higher than 25% and 50%, the NO content is lower than them. However, when the natural gas replacement rate is 75%, the diffusion combustion of n-dodecane disappears, there is no high temperature zone in the front end of the diffusion combustion flame, and NO emissions are relatively small. From the perspective of engine single-cylinder power and average in-cylinder pressure, when the natural gas replacement rate reaches 75%, the engine single-cylinder power and average in-cylinder pressure do not decrease monotonically with the delay of injection timing, The power and indicated pressure at 715˚CA are even higher than the injection timing at 710˚CA. As the natural gas replacement rate increases, it can be seen that the injection moment with the best power is gradually rising to the top dead center. Approaching. When the injection timing exceeds the top dead center, the power loss caused by the injection delay is also more serious than at 25% and 50% replacement rates

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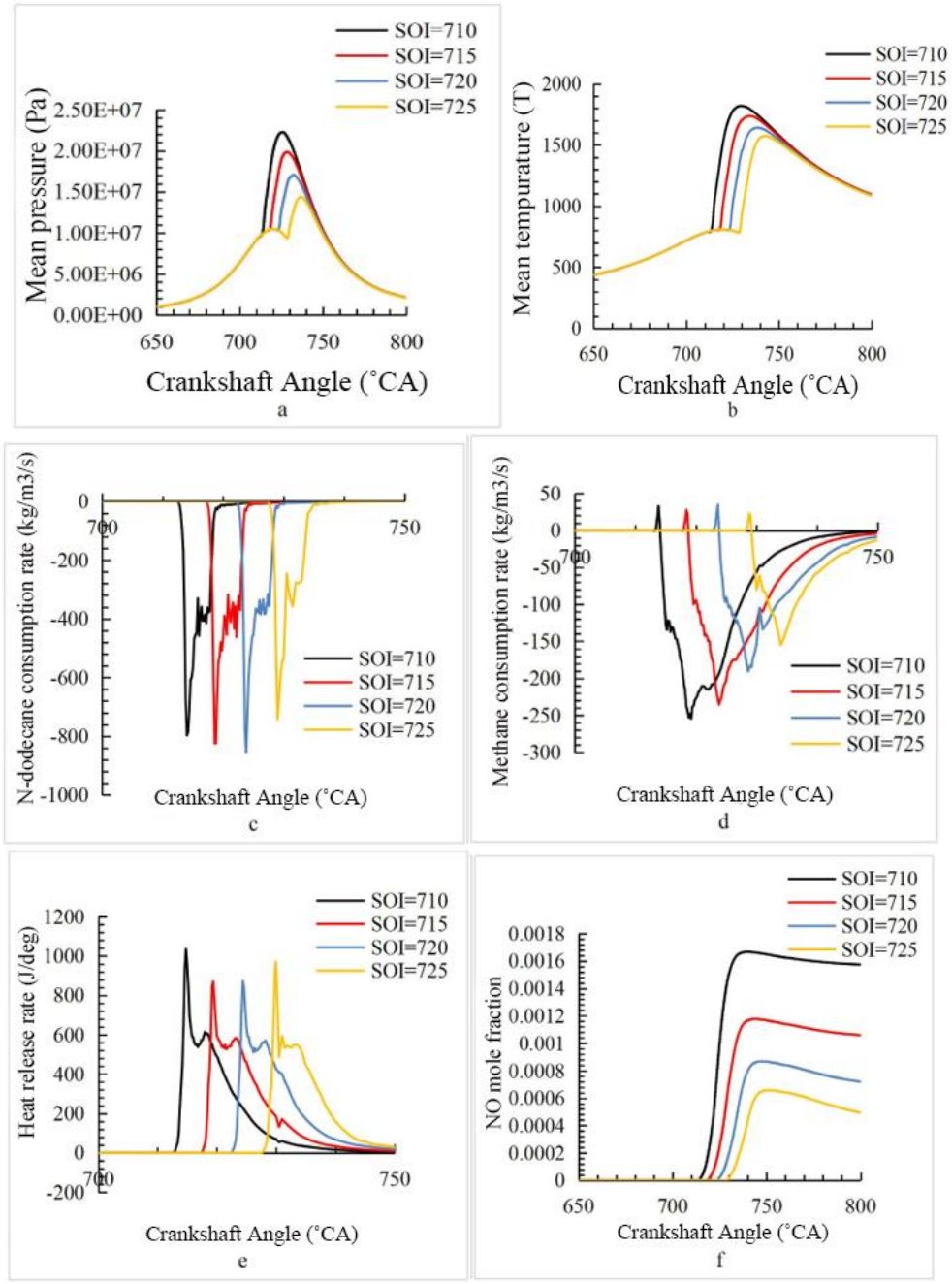

Figure 9. The Variation of parameters with crankshaft angle at 50% natural gas replacement rate (a. Pressure; b. Temperature; c. N-dodecane consumption rate; d. Methane consumption rate; e. Exothermic rate; f. NO mole fraction)

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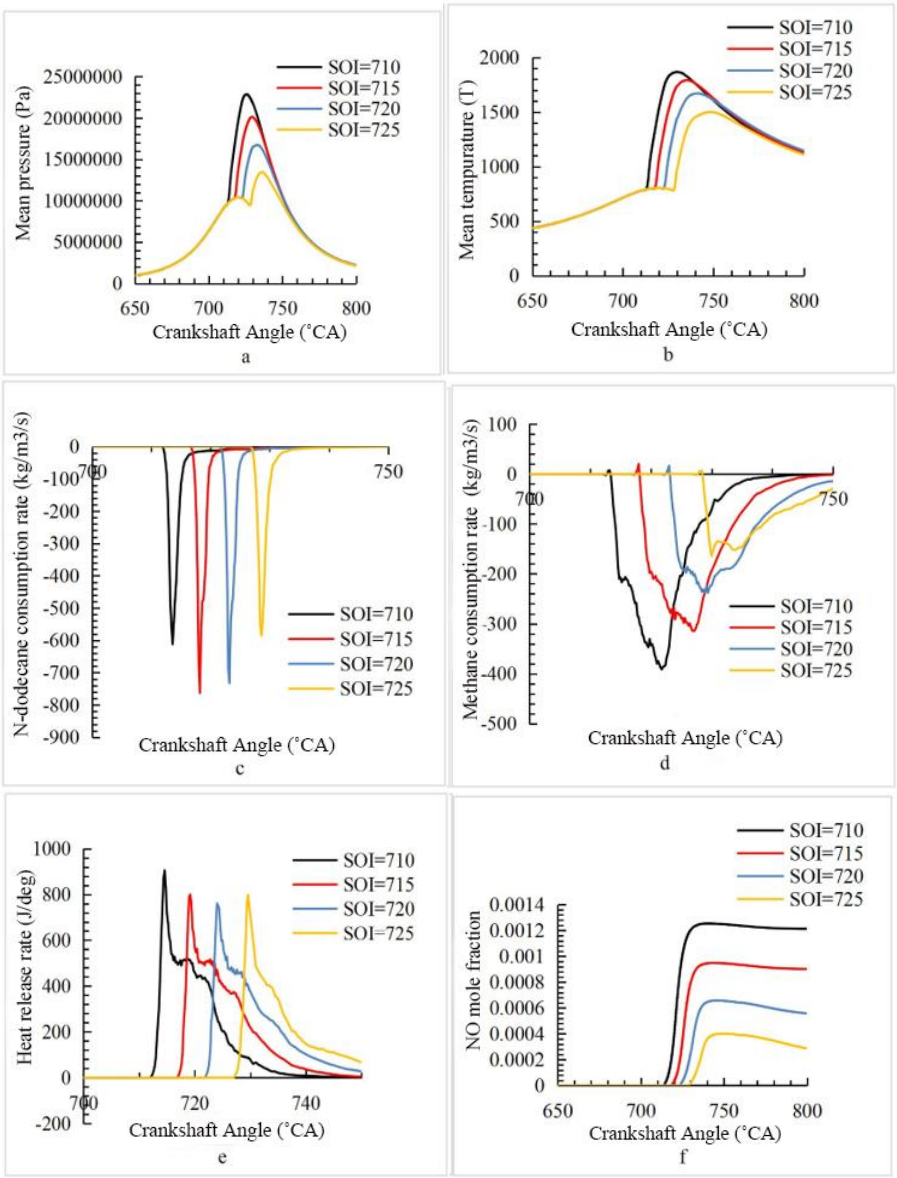

Figure 10. The Variation of parameters with crankshaft angle at 75% natural gas replacement rate (a. Pressure; b. Temperature; c. N-dodecane consumption rate; d. Methane consumption rate; e. Exothermic rate; f. NO mole fraction)

6. CONCLUDING REMARKS

The research object is based on the MAN L23/30H medium-speed engine. The calculation models of diesel and dual fuel engines were established by numerical simulation technology. Based on the experimental data of the original engine, the n-dodecane skeleton mechanism was coupled with the 13-step NOX emission sub-mechanism to analyze the ignition delay. The feasibility of the chemical kinetic model was demonstrated in three-dimensional CFD calculation. After the model calibration is completed, the combustion performance of the marine medium-speed engine is studied by adjusting the calculation parameters. By changing the injection timing of the pilot diesel, the effect of different injection timings on the combustion of the marine medium-speed engine cylinder is studied and the following conclusion are obtained. The results of the study can be summarized as follows:

(1) In the pure diesel mode, as the injection timing is delayed, the indicated power of the single cylinder of the engine and the average effective indicated pressure decrease, and NO emissions decrease. Among them, before the top dead center, the power change is small with

ISSN: 2472-3703 DOI: 10.6911/WSRJ.202005_6(5).0027

the delay of the injection timing; after the top point, the change is large with the delay of the injection timing, and NO emissions are reversed.

(2) In the dual fuel mode, the influence of injection timing on the parameters in the cylinder is basically the same, under different natural gas replacement rates.

(3) In the dual fuel mode, the engine's power is more sensitive to the injection timing, and the power reduction caused by the injection delay is greater than the pure diesel mode.

(4) Under the high natural gas replacement rate, the diffusion combustion of the pilot diesel will disappear, and the exothermic rate curve and the n-dodecane consumption curve show a single peak.

(5) With the increase of natural gas replacement rate, the power of the engine is gradually improved, and the injection timing with the best power is approaching the top dead center.

(6) In the dual fuel mode, the high-pressure direct injection in the cylinder has a lower methane utilization rate than the new combustion modes such as pre-injection and multiple injection.

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