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تحليل مقارن بين وقود الغاز الطبيعي / الديزل (الوقود المزدوج) والديزل النقي على محركات الديزل البحرية

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الخلاصة

تلوث البيئة الخطير وأزمة الطاقة في جميع أنحاء العالم يحفز تطوير محركات الديزل البحرية ذات التلوث القليل والأقل استهلاكاً للطاقة، أصبحت الهدف الرئيسي للبحث العلمي. ومن أجل حساب كفاءة وإنبعاثات محركات الديزل البحرية والتي تعمل على الغاز الطبيعي / الديزل، تم إجراء تحليل مقارن من بين حالة الديزل النقي وحالة الوقود الغاز الطبيعي / الديزل (الوقود المزدوج) في شدة مطربة، عملية الاحتراق، ومعدل طرح الحرارة، والإنبعاثات. وتمت دراسة كفاءة حالة الوقود المختلفة من خلال دراسة العددية والتجريبية. النتيجة تظهر بأن محرك الديزل البحري وطبيعة حالة الوقود الغاز الطبيعي / الديزل يمكن من خلالها الحصول على الطاقة الحركية أقل اطرابا، وكذلك تقليل درجة الحرارة إلى حوالي 400 كلفن وتفعيل ميزة الاشتعال وتقليل من انبعاثات إلى 75% وانبعاث غاز CO_2 إلى 35%، وكذلك انبعاث غاز أكاسيد النيتروجين إلى 80%.

Comparative analysis between natural gas/diesel (dual fuel) and pure diesel on the marine diesel engine

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ABSTRACT

The serious environmental pollution and the energy crisis all over the world have caused the development of lower pollution and lower energy consumption marine diesel engine to become a major research goal. In order to obtain the performance and emission of marine diesel engine on natural gas/diesel, a comparative analysis was carried out between the pure diesel mode and natural gas/diesel (dual fuel) mode at turbulence intensity, combustion process, rate of heat release and emissions. The performance of different fuel modes was studied through numerical analysis and experiment. The results showed that marine diesel engine, in natural gas/diesel mode, can get a lower turbulent kinetic energy, temperature reduction of about 400K, ignition advance, soot emission reduction by 75%, reduction of CO_2 emission by 35% and NOx emission reduction by 80%.

Keywords: comparative analysis; emission; marine diesel engine; natural gas/diesel (dual fuel); turbulent kinetic energy.

INTRODUCTION

Energy is the building block of socio-economic development in any country (Chauhan *et al.*, 2010). China is a developing country, where oil is the major energy sources used in industry, agriculture and economical society. However, the escalating oil prices and scarcity of fuel oils coupled with exploding population have resulted inserious energy crisis. There is thus a pressing need to develop technology for utilizing renewable energy sources that can make significant contribution to the economy (Das *et al.*,2012). At the same time, because of stricter emissions regulations, many recent studies have focused on reduction of engine-out emissions.One representative set of marine emissions regulations, the Annex VI of MARPOL (IMO, 2010), requires that beginning in 2016, NO_x emissionsbe reduced by more than 80% of the level allowed by the current standard. However, the conventional liquid fuels (such as diesel) cannot meet these requirements (Lee *et al*., 2013).

Thus, the concept of using alternative gaseous fuel in diesel engines has gained worldwide attention. Natural gas is one such fuel available in large quantities in many parts of world at attractive prices (Paykani *et al*., 2012). It is a clean burning fuel as compared to the conventional liquid fuels like diesel or gasoline. It has a high octane number and therefore it is suitable for engines with relatively high compression ratio (Roy *et al*., 2014; Mustafi *et al*., 2013). Its self-ignition temperature is 730°C and it requires intense source of energy to enable combustion, i.e. glow plug, spark plug or pilot liquid fuel. It mixes rapidly with air to form homogeneous air fuel mixture for efficient combustion inside engine cylinder and substantial reduction in harmful emissions (Brynolf *et al*., 2014). Dual fuel engine is one of the possible short-term solutions to reduce emissions from traditional diesel engines; meantime, utilizing an alternative fuel like natural gas as primary fuel. It consequently results in not only an interesting technology to meet future emission regulations, but also a powerful solution to retrofit existing engines (Scarcelli, R. 2007). With the continued development of computing power, multidimensional modeling has become an increasingly feasible and economical tool in engine design and development processes (Sukumaran *et al*., 2013). Good computational models may be used to achieve a better research of the combustion process within engines, finding effective means to overcome operational problems, evaluating new design concepts, and reducing hardware prototype and development costs (Liu & Karim, 2009).

In the present contribution, dual fuel engines are relatively rare application in marine engines. Therefore, in this research, the computational fluid dynamics (CFD) model is established to simulate the combustion and emission processes of a marine natural gas/diesel dual fuel engine.

TEST EQUIPMENT AND TEST CONDITIONS

An experiment was carried out to ensure the reliability of the model and accuracy of the calculated results on the marine natural gas/diesel dual fuel engines. According to propulsion characteristic, the experiment measured engine operating data under operation at 25%, 50%, 75% and 100% load (corresponding to speeds 756, 952, 1090, and 1200 r/min) in pure diesel mode and dual fuel mode. The engine used for the investigation is asix-cylinder, four-stroke, water cooled and direct injection diesel engine. The technical specifications of the engine are given in Table 1.

Table 1. The specifications of marine diesel engine

Item	Specification
Type	Four stroke
Cylinder number	6
Cylinder configuration	Inline
Fuel system type	Direct injection
Coolant	water
Bore (mm)	190
Stroke (mm)	230
Compression ratio	14:1
Injection timing (^oCA)	34 BTDC
Rated rotate speed (r/min)	1200
Rated power (kW)	400
Method of aspiration	Turbo-charging

Mass flow rates of diesel and natural gas fuels are measured by volumetric flow meters. Temperature of cooling water, lubricating oil, inlet air, exhaust gas, pressure of exhaust gas, charge of air and atmosphere are also measured to ensure proper engine operating conditions. The experiment was carried out on the standard marine natural gas/diesel engines test cell. The information of test cellis given in Table 2.

Table 2. Information of test cell

Analyzer	Model	Measurement ranges	Deviation
CO (%)	AIA 240	0~16.0	0.16%
CO_2 (%)	AIA 240	0~16.0	0.12%
NO (10^{-6})	FAC 246	0~2000	0.11%
O_2 (%)	IMA 241	0~25	-0.10%
HC (10^{-6})	FAC 246	0~2000	0.15%
t (^oC)	FC 2022	0~200	0.1%
p (kPa)	FC 2022	-50~400	0.1%
Speed (r/min)	FC 2010	0~3000	0.1%
Torque (N·m)	Y 1900	0~12000	0.1%
Fuel flow (kg/h)	FC 2210	0~1000	0.1%
LNG flow (m^3/h)	CMF 200M	0~2000	0.2%

MODELING METHODOLOGY

Mathematical model

CFD has been shown to be an effective tool for design as well as improved understanding of behavior of actual combustion systems (Morsy *et al.*, 2012). There are number of commercial CFD software available in the market, including AVL-FIRE, which has been widely used for engine simulations. The flow in the cylinder of engines is a typical unsteady, strong compression, strong rotation and anisotropic turbulent flow. *K-ε* two equation turbulence model is selected to simulate cylinder flow field of the internal combustion engine in the CFD software (Jie Wang *et al.*, 2009; Gan Tian *et al.*, 2014). The complete standard, high-Re-number *k-ε* model, integrated over a control volume is:

$$\rho\frac{\partial k}{\partial t}+\rho U_j\frac{\partial k}{\partial x_j}=P+G-\varepsilon+\frac{\partial}{\partial x_j}\left(\mu+\frac{\mu_t}{\sigma_k}\frac{\partial k}{\partial x_j}\right) \tag{1}$$

$$\rho\frac{D\varepsilon}{Dt}=\left(C_{\varepsilon 1}P+C_{\varepsilon 3}G+C_{\varepsilon 4}k\frac{\partial U_k}{\partial x_k}-C_{\varepsilon 2}\varepsilon\right)+\frac{\partial}{\partial x_j}\left(\frac{\mu_t}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial x_j}\right) \tag{2}$$

Where $C_{\varepsilon 1}$, $C_{\varepsilon 2}$, $C_{\varepsilon 3}$ and $C_{\varepsilon 4}$ is constant ($C_{\varepsilon 1}$=1.44, $C_{\varepsilon 2}$=1.92, $C_{\varepsilon 3}$=0.8 and $C_{\varepsilon 4}$=-0.373), U(m/s) is the velocity component, k (m^2/s^2) is turbulent kinetic energy, ε(m^2/s^3) is the turbulent kinetic energy dissipation rate, X_j is the coordinate components, μ(Pa·s) is a kinetic viscosity coefficient, σ_k and σ_ε is the turbulent Prandtl number of turbulent kinetic energy and turbulent energy dissipation rate, P (m^2/s^2) is turbulent kinetic energy produced by velocity gradient, G (m^2/s^2) is turbulent kinetic energy generated by buoyancy.

In order to describe the injector atomization process, walljet1 model is used to simulate that the spray hit the wall spray process. This model, in principle is based on the spray/wall impingement model of Naber & Reitz (1988). The concept is that, under engine conditions a vapor cushion is formed under the droplets and that they rebound or slide along the walls. This model does not take into account the wallfilm physics. This allows spray/wall impingement calculations to beperformed without using the wallfilm module, which is sufficient in a variety of practical applications, where the wallfilm physics do not play an essential role within the wall interaction process (Wang *et al.*, 2006). In the currently implemented model, it is assumed that a droplet which hits the wall suffers one of the two consequences, namely rebound or reflection in the manner of a liquid jet, depending on the Weber number. The transition criterion between these two regimes is described by a critical Weber number, which is taken to be *Wec*=80(Wei *et al.*, 2013). Wave model is used to simulate droplet breakup (Abianeh *et al.*, 2014; Zsély *et al.*, 2011). The Multi-component model is used to simulate droplet evaporation. ECFM-3Z model of Coherent Flame Model is used to simulate the combustion process. The ECFM-3Z model is developed by the Groupement Scientifque Moteurs(GSM) consortium specifically for Diesel combustion (AVL, 2010). This is a combustion model based on flame surface density transport equation and a mixing model that can describe inhomogeneous turbulent premixed and diffusion combustion (Zhixia He *et al*., 2013). When using the ECFM-3Z model, it is possible to define the fuel as consisting of more than one chemical species. The fuel can be prescribed as a mixture of several components. By using this number of components, each group of hydrocarbons can be represented. This description is available for all types of combustion. The provided solution enables a connection to the multi-component spray capabilities. The ECFM-3Z combustion model uses the fuel components to combine them temporarily to a fuel mixture during the calculation. This means that effects like auto-ignition and flame propagation are handled for this combined fuel within the combustion model. The rate of reaction for each fuel component is finally split up. By doing this it is possible to calculate the consumption of each component separately. The development of the combustion products is based on the consumption of the single components.

In the ECFM-3Z model, the transport equations are solved for the averaged quantities of chemical species O_2, N_2, CO_2, CO, H_2, H_2O, O, H, N, OH and NO. This equation is classically modeled as:

$$\frac{\partial \bar{\rho}\tilde{y}_x}{\partial t} + \frac{\partial \bar{\rho}\tilde{u}_i\tilde{y}_x}{\partial x_i} - \frac{\partial}{\partial x_i}\left(\left(\frac{\mu}{Sc} + \frac{\mu_t}{Sc_t}\right)\frac{\partial \tilde{y}_x}{\partial x_i}\right) = \bar{\dot{\omega}}_x \tag{3}$$

Where x is the distance, i is the quantity of species, μ is the dynamic viscosity, $\tilde{u}$ is the velocity, Sc is the Schmidt-number, $\bar{\rho}$ is the ensemble-averaged density, $\bar{\dot{\omega}}_x$ is the combustion source term and $\tilde{y}_x$ is the averaged mass fraction of species α.α is the index for chemical species. The fuel is divided in two parts: the fuel present in the fresh gases $\tilde{y}^u_{Fu}$ and the fuel present in the burnt gases, $\tilde{y}^b_{Fu}$

$$\tilde{y}^u_{Fu} = \frac{\bar{m}^u_{Fu}}{\bar{m}} = \frac{\bar{m}^u_{Fu}/V}{\bar{m}/V} = \frac{\bar{\rho}^u_{Fu}}{\bar{\rho}} \text{ and } \tilde{y}^b_{Fu} = \frac{\bar{m}^b_{Fu}}{\bar{m}} = \frac{\bar{m}^b_{Fu}/V}{\bar{m}/V} = \frac{\bar{\rho}^b_{Fu}}{\bar{\rho}} \tag{4}$$

$\tilde{y}_{Fu} = \tilde{y}^u_{Fu} + \tilde{y}^b_{Fu}$ is the mean fuel mass fraction in the computational cell (Colin *et al*., 2003). V is the volume. $\bar{m}^u_{Fu}$ (resp.$\bar{m}^b_{Fu}$) is the mass of the fuel contained in fresh gases (resp. burnt gases). A transport equation is used to compute $\tilde{y}^u_{Fu}$:

$$\frac{\partial \bar{\rho}\tilde{y}^u_{Fu}}{\partial t} + \frac{\partial \bar{\rho}\tilde{u}_i\tilde{y}^u_{Fu}}{\partial x_i} - \frac{\partial}{\partial x_i}\left(\left(\frac{\mu}{Sc} + \frac{\mu_t}{Sc_t}\right)\frac{\partial \tilde{y}^u_{Fu}}{\partial x_i}\right) = \bar{\rho}\tilde{\dot{S}}^u_{Fu} + \bar{\dot{\omega}}^u_{Fu} \tag{5}$$

Where $\tilde{\dot{S}}^u_{Fu}$ is the source term quantifying the fuel evaporation in fresh gases. $\bar{\dot{\omega}}^u_{Fu}$ is a source term taking auto-ignition, premixed flame and mixing between mixed unburned and mixed burnt areas into account.

Emission models selected were extended Zeldovich NOx emission model and Kennedy / Hiroyasu / Magussen soot emission model. The extended Zeldovich model is widely used to calculate the generation of nitric oxide (NO) in the soot emissions models. The reaction mechanism of nitric oxide can be expressed in terms of the so-called extended Zeldovich mechanism (AVL, 2010; Moranahalli & Gopalakrishnan, 2011):

$$N_2+O \underset{k_{1b}}{\overset{k_{1f}}{\leftrightarrow}} NO+N \tag{6}$$

$$N+O_2 \underset{k_{2b}}{\overset{k_{2f}}{\leftrightarrow}} NO+O \tag{7}$$

$$N+OH \underset{k_{3b}}{\overset{k_{3f}}{\leftrightarrow}} NO+H \tag{8}$$

The reaction mechanism (6-8) is known as the extended Zeldovich mechanism that considers the effect of oxygen, nitrogen and hydrogen radicals on NO formation (Hill & Smoot, 2000). It is very important to point out that all three chemical reactions that represent the Zeldovich mechanism exhibit strong temperature dependency.

The temporal change of NO concentration (or net rate of NO formation) via reactions (6-8) is given by:

$$\frac{\partial c_{NO}}{dt} = k_{1f}c_O c_{N_2} + k_{2f}c_N c_{O_2} + k_{3f}c_N c_{OH} - k_{1b}c_{NO}c_N - k_{2b}c_{NO}c_O - k_{3b}c_{NO}c_H \tag{9}$$

Considering forward and backward directions, where the concentration c is given in mol/cm^3.

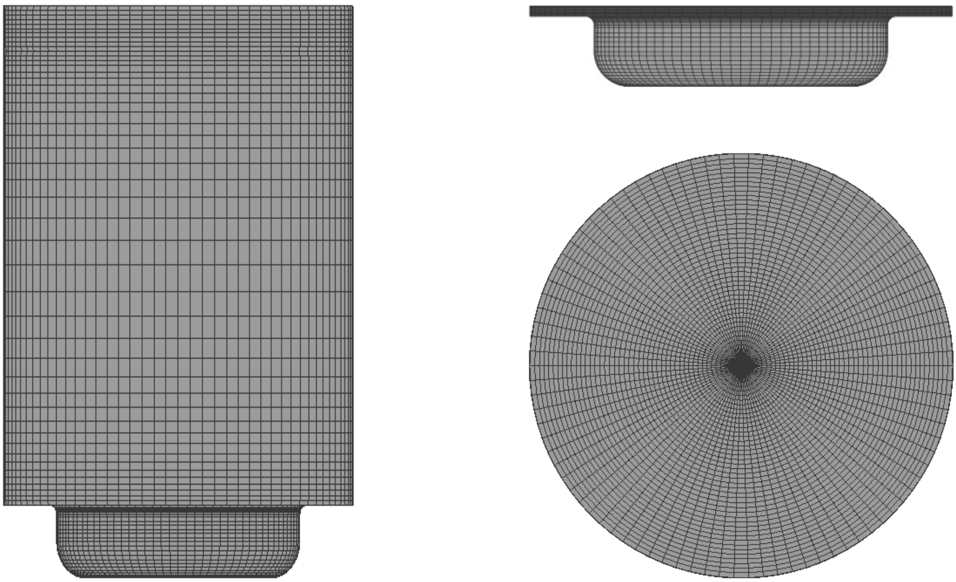
The thermal NO reactions are highly dependent on temperature, residence time and atomic oxygen concentration. The first reaction (6) has very high activation energy and it is usually accepted as being the rate-limiting step of the thermal NO formation. Due to the high activation energy required to split the strong N_2 triple bond, the rate of formation of NO is significant only at high temperatures (greater than 1800 K).

Calculation model

The calculation model is the combustion chamber of marine 6190 direct injection diesel engine. The technical specifications of the engine are given in Table 1. The calculation has taken into account the integral compression and power stroke, beginning with the intake valve closed time (215^{o}CA) and end with the exhaust valve open time (479^{o}CA). The parameters of CFD are given in Table 3. The model and grid of combustion chamber are shown in Figure 1.

Table 3. The parameters of CFD

Item	Parameter	Item	Parameter
Crank angle (deg)	215~479	Cylinder temperature (K)	475.15
Natural gas injection timing (°CA)	216~250	Pistons fire shore temperature (K)	550.15
Diesel injection timing (°CA)	326~360	Piston head temperature (K)	575.15
Natural gas flow (kg/h)	82.5	Initial in-cylinder pressure (kPa)	135
Diesel flow (kg/h)	17.2	Initial in-cylinder temperature (K)	372.7



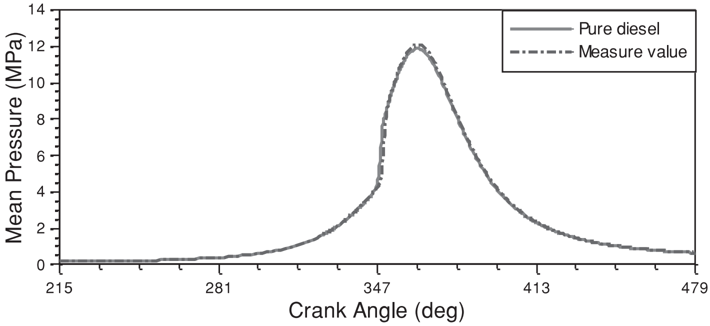
(a) 215°CA (b) 360°CA (TDC)

Fig. 1. The model and grid of combustion

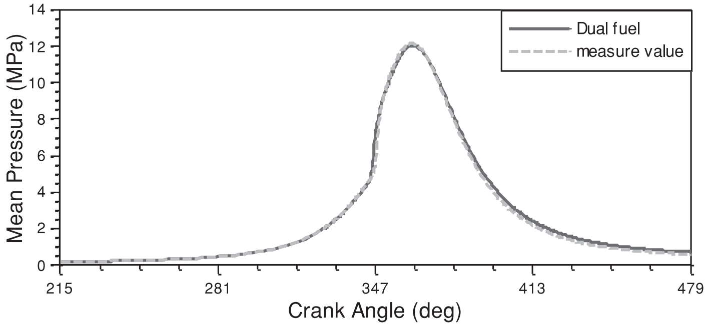
RESULTS AND DISCUSSION

Model validation

The results data is obtained by simulation of AVL FIRE software. To ensure the accuracy and reliability of the results of the calculation model, the data used for initial conditions of calculation model is obtained in the experiment. The comparison of cylinder pressure is shown in the Figure 2, between measure value and simulation value in the pure diesel mode and dual fuel mode.



Pure diesel mode



Dual fuel mode

Fig. 2. Comparison chart of cylinder pressure

As shown in Figure 2, the maximum cylinder pressure of simulation value is lower than the measure value, but the cylinder pressure curves of calculation and experiment are basically same. Thus, the calculation model of combustion had some practical value, and it could be used to analyze some factors on the combustion process and emissions on diesel engine on dual fuel.

Turbulence intensity analysis

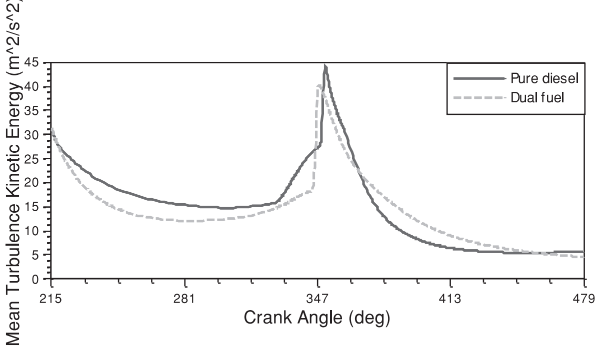


Fig. 3. Comparison of mean turbulence kinetic energy

As shown in Figure 3, the curves of mean turbulence kinetic energy in pure diesel and dual fuel have four regularity processes, which are decreased slowly, increased slowly, increased rapidly and decreased gradually. After the intake valve closing (215°CA), though the shear layer caused by the valve throat disappears, turbulence caused by shear layer still exists (Xie, 2005). As the piston moves upward, large-scale and small-scale eddy circulation convection occurs and the small-scale eddies are constantly broken. So the two curves of mean turbulent kinetic energy decrease slowly. Meanwhile, as the piston continues moving upward, many localized turbulences are produced by compression effect. This causes subsequent turbulent kinetic energy to increase slowly in the cylinder. As diesel oil is injected into cylinder (326°CA), the turbulence intensity of pure diesel mode increases faster than dual fuel mode. The large-scale circulation will be strengthened by the ignition and combustion, so that the turbulent kinetic energy in cylinder increases rapidly. The maximum turbulence kinetic energy of pure diesel mode is higher than turbulence kinetic energy of dual fuel mode. With the continued combustion, the turbulence is weakened gradually.

Combustion process analysis

Figure 4 and Figure 5 show mean pressure and mean temperature curve in cylinder. As shown in Figure 4 and Figure 5, the mean pressure and mean temperature curve have two peaks, as shown with the solid arrows. The first peak in cylinder is caused by ignition diesel injected in the compression stroke. The second peak is caused by the combustion of main fuel, such as diesel in pure diesel mode and natural gas in dual fuel mode. As shown in Figure 4, there is an ignition advance on the mean pressure curve in dual fuel mode, and the power in dual fuel mode is larger than pure diesel mode. As shown in Figure 5, there is an ignition advance on the mean temperature curve in dual fuel mode too. The maximum temperature in dual fuel mode is lower than pure diesel mode, and this is the real reason of low NO_x emissions in dual fuel mode.

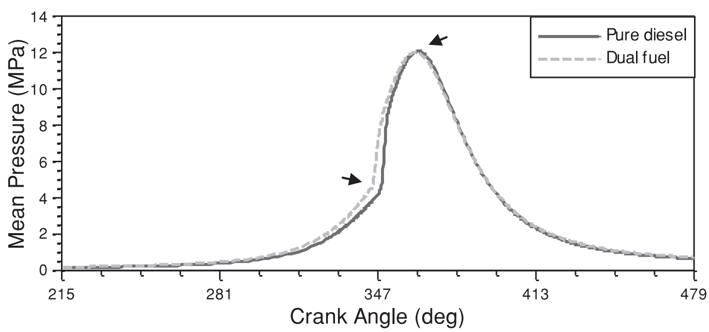


Fig. 4. Comparison of mean pressure

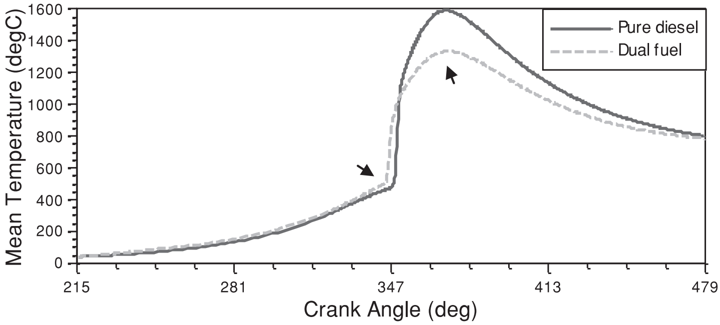


Fig. 5. Comparison of mean temperature

Rate of heat release analysis

As shown in Figure 6, single peak appears at rate of heat release curves in pure diesel mode and dual fuel mode. The initial heat release rate in dual fuel mode is higher than the original machine in pure diesel mode, and the starting time is earlier in dual fuel mode than in pure diesel mode. In dual fuel mode, natural gas atomization is better than diesel fuel in cylinder, so the starting time is earlier in dual fuel mode. Because natural gas has higher octane than diesel, so the peak of rate of heat release in dual fuel mode is higher than in pure diesel mode.

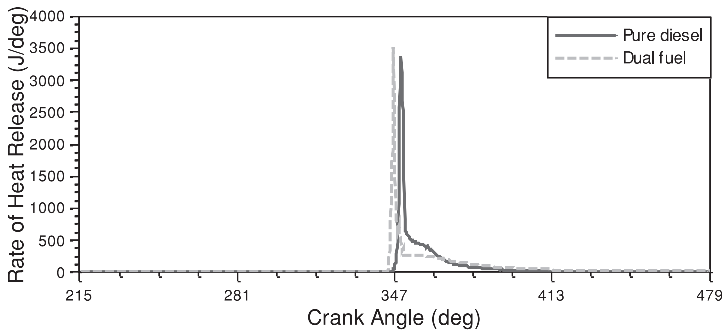


Fig. 6. Comparison of rate of heat release

Emissions analysis

As shown in Figure 7, less NO_X emission appears in dual fuel mode, because combustion temperature of natural gas is lower than pure diesel.

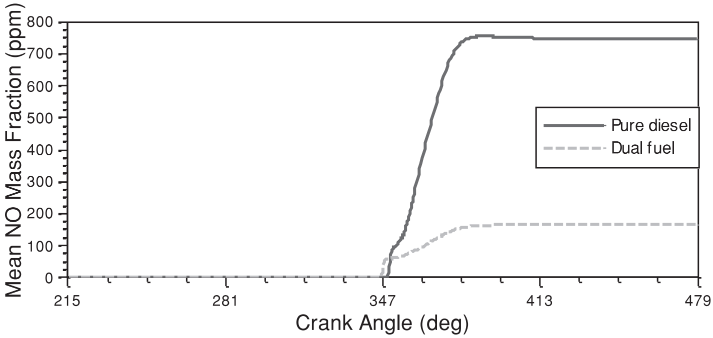


Fig. 7. Comparison of NO emission

As shown in Figure 8, when engine burn in dual fuel mode, soot emission of engine is reduced by 75% compared to pure diesel mode. Natural gas and air is gas phase, so that is more uniform mixing and more complete combustion. The main component of natural gas is methane; methane only have C-H bond in the molecular structure, no C-C bond, which can reduce soot emission.

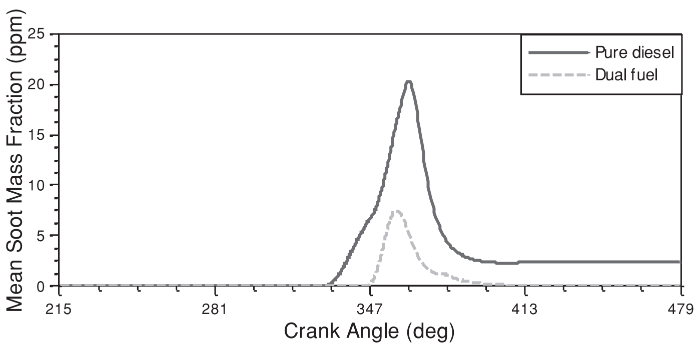


Fig. 8. Comparison of soot emission

As shown in Figure 9, when engine burn in dual fuel mode, CO_2 emission of engine is reduced by 35% compared to pure diesel mode. Because natural gas has H/C maximum among all hydrocarbon fuels, it release per unit of energy and only generates 20-30% lower CO_2 emissions compared to gasoline. Thus, the development of marine natural gas/diesel dual fuel engineis an important measure to alleviate the problem of global warming caused by CO_2 emissions.

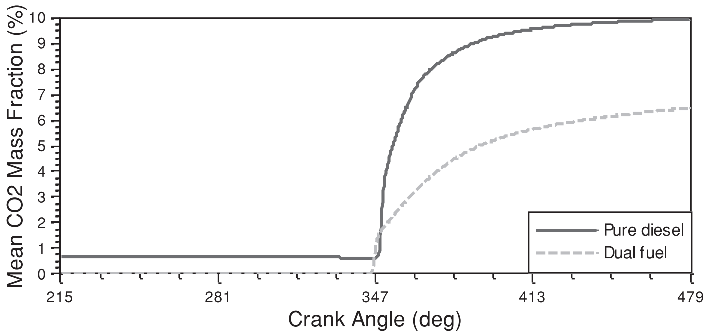


Fig. 9. Comparison of CO_2 emission

CONCLUSION

The turbulent kinetic energy of dual fuel mode is 11% less, compared to pure diesel model, resulting in reduced cylinder temperature of about 400K. There is an ignition advance in the combustion process of dual fuel mode, which cause the rate of heat to release increased rapidly. So N_2 residence time at high temperature and oxygen-rich condition is shortened and NO_X emission reduces 80%.

A smaller proportion of diesel plays the ignition role in the dual fuel mode, while natural gas is injected directly into cylinder at gaseous form and air diffusion mixes faster than diffusion rate of diesel atomization, so soot emission reduces by 75% and CO_2 emission reduces by 35%. Therefore, marine diesel engine on natural gas/diesel mode has important significance in protection of port and marine environment.

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NOMENCLATURE

BTDC	before top dead center
CA	crankshaft angle (deg)
CFD	computational fluid dynamics
CR	compression ratio
G	turbulent kinetic energy generated by buoyancy (m^2/s^2)
i	the quantity of species
k	turbulent kinetic energy (m^2/s^2)
LNG	liquefied natural gas

MARPOL	international convention for the prevention of pollution from ship
P	turbulent kinetic energy produced by velocity gradient (m^2/s^2)
RPM	revolutions per minute (r/min)
S_c	Schmidt number
TDC	top dead center
U	velocity component (m/s)
V	Volume (m^3)
x	Distance (m)
X_j	the coordinate components
α	the index for chemical species
ε	the turbulent kinetic energy dissipation rate (m^2/s^3)
μ	kinetic viscosity coefficient (Pa·s)
σ_k	turbulent kinetic energy
σ_ε	turbulent Prandtl number of turbulent energy dissipation rate

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