# THE TOTAL $NO_X$ EMISSION CONTROL POSSIBILITY IN MODERN LARGE BORE, SLOW SPEED MARINE ENGINES

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**Abstract**

*The common demands for environmental protection in shipping are being recognized by IMO, through development of the rules with respect to $SO_x$ and $NO_x$. This paper investigates the results of fuel injection variables, affecting cylinder process performance and exhaust gas emission. The effect of injection assembly design attributed on NO emission was estimated. The similar slow speed, large bore MAN B&W engines performance were compared. The main factors used for evaluation were: fuel consumption, rotational speed, mean and maximum in-cylinder pressures and total concentrations of $NO_x$, $CO_2$, $O_2$, CO, and HC. The engines performance and NO emission were compared with conventional 6S60MC-C and latest – electronically controlled 7S60ME-C. The CFD simulation of combustion and pollutant formation was created to represent processes of a marine large bore engine. The SMC-C engine operation and standard injection nozzle design was taken as a baseline for comparison with measured NO concentration. The predicted cylinder pressures showed reasonable agreement with experimental measurements. The predictive ability of used CFD simulation package was found to be quantitatively insufficient in terms of $NO_x$ emission. However, the predicted NO emission exhibits similar sensitivity with experiment results. The quantitative set of reference data for the validation of simulation results needs to be substantially improved. The determination of fuel injection and cylinder pressure history can be identified as the most important source of inaccuracy. Finally, weighted $NO_x$ specific emission factor for both engines shows close value, sufficiently placed against IMO limit presented in Fig. 11. The results are promising and show that controlled combustion process is capable of fulfilling present and future $NO_x$ emission requirements. Modern marine engines are designed with adjustable components to allow the engine to be adjusted for maximum efficiency or $NO_x$ weighted emission factor when used in particular application.*

***Keywords:** marine diesel engine, injection assembly, exhaust emission*

## 1. Introduction

Main ship propulsion engines in ocean-going vessels are very large bore and slow speed. These engines are generally uniquely built and designed for a particular vessel used in a particular application. They are currently designed for maximum fuel efficiency and performance with special consideration of the impacts on exhaust emissions. However, these engines can have high $NO_x$ emissions. Marine engines are built using advanced controls of combustion which are considered to be emission control strategies [1]. However, $NO_x$ emission reductions can still be achieved through fuel injection control [2]. The most recent advances in fuel injection technology are the systems that use fuel rate shaping to vary the delivery of fuel. Injecting most of the fuel into an established flame allows for a steady burn that limits $NO_x$. Use of electronic controls enables to implement much more precise control of the fuel injection. In addition, electronic controls can be used to sense ambient ship and engine operation, to maximize performance and minimize emissions over a wide range of conditions such as transient operation. This paper investigates the results of an experiment for a test cycle range of engine load in terms of exhaust emission. The engine performance and NO emission were compared with conventional MAN B&W - 6S60MC-C and latest - electronically controlled 7S60ME-C slow speed, large bore engine. The experiment included 7S60ME-C engine test cycle performed with electronic fuel rate shaping, to reflect economic and low emission mode. By varying some engine cylinder characteristic concerning fuel injection, the effect of cylinder combustion attribute on NO emission was established. The main factors used for evaluation were: fuel consumption, mean and maximum in-cylinder pressures and total concentrations of $NO_x$, $CO_2$, $O_2$ CO, and HC. The objective of the present study was to develop an efficient methodology for performing engine and cylinder data analysis for maintenance purpose. The results of steady-state simulations performed were discussed.

## 2. Fuel oil injection system description

For conventional diesel engines, fuel oil injectors, exhaust valves and starting air distributors are driven and thus mechanically controlled by crankshafts, via cams and camshaft linkages. In electronically controlled marine engines they are powered by hydraulic oil unit and controlled by solenoid valves, which act according to CPU signals. This makes possible to achieve optimum operational conditions, at wide range engine rotational speed, and reduces specific fuel consumption. The amount and timing of fuel injected into a cylinder can be governed independently for every other cylinder. $NO_x$ reduction mode is one option, and fuel economy mode is another option. Outlined below is the basic structure of electronically controlled marine diesel engines [3], notably the MAN B&W "ME" model (see Fig. 1). Hydraulic oil is pressurized by engine-driven multi-piston high-pressure oil pumps and then fed to servo pistons for the fuel injectors and exhaust valves, respectively.

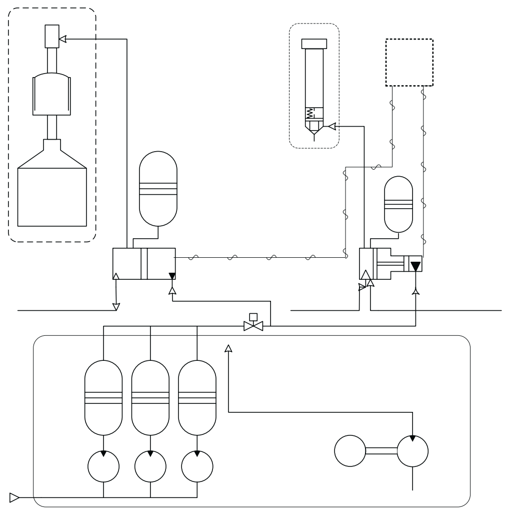

*Fig. 1. Outline of the ME engine fuel oil injection system*

In the fuel injection actuator, the amount, timing and injection pressure of fuel are controlled by proportional control valves, which admit hydraulic oil from intermediate accumulators. The PC valve is in turn, controlled by an electric linear motor that gets its control input from the cylinder control unit. This design concept is crucial for fuel economy, emissions and general engine performance. The optimum combustion (thus also the optimum thermal efficiency) requires an optimized fuel injection pattern which is generated by the fuel injection cam shape in a conventional engine. Common Rail injection systems with on/off control valves are becoming standard in many modern diesel engines at present. Such systems are relatively simple and will provide larger flexibility than the contemporary camshaft based injection systems. However, by nature the common rail system provides another rate shaping than, what is optimum for the slow speed engine combustion process. To counteract this, it has been proposed use proportional valve system that controls the injection pressure. The fuel mass flow and the total mass injected for each fuel valve is calculated by means of advanced dynamic fuel injection simulation computer code for a large bore engine with two or three fuel valves per cylinder. The system can perform as a single-injection system as well as a pre-injection system with a high degree of freedom to modulate the injection in terms of injection rate, timing, duration and pressure. In practical terms, a number of injection patterns will be stored in the computer and selected by the control system so as to operate the engine with optimal injection characteristics from dead slow to overload, as well as during astern running and crash stop. The capability of the fuel injection system is superior in terms of fuel consumption and very attractive trade-off between $NO_x$ reduction and SFOC increase.

## 3. Experiment details

General scope of performance procedure of marine engines on test beds, is the exhaust gas emission measurement, particularly nitrogen oxides and additional gaseous components that have to be registered using relevant gas analysers in accordance to Annex VI of Marpol 73/78 convention: $NO_x$, CO, $CO_2$, $O_2$ and HC. The exhaust gas methodology should comply with the specification given in the IMO's $NO_x$ Technical Code and ISO-8178 standard with regard to measurement method, accuracy and performance, sensitivity against other exhaust gas components. During the engine trials the exhaust gas sample was taken from the engine exhaust receiver after turbocharger, via a common probe and then distributed to all analysers (Fig. 2). The upper end of the sampling probe - a sintered ceramic filter, the probe, sample line, transfer pump and distribution box, were heated (191°C) by means of temperature controlled unit, to avoid water condensation and hydrocarbons.

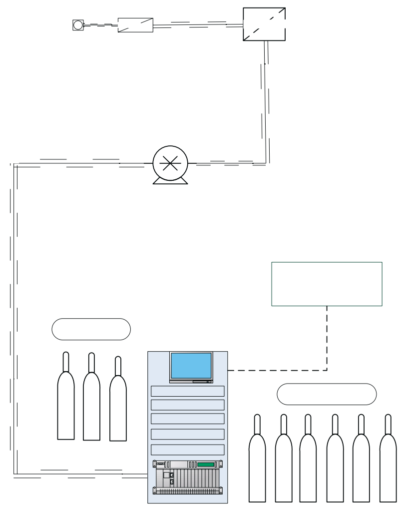

*Fig. 2. Test engine exhaust gas emission experimental setup*

Measurements were carried out on a test-bed for main ship propulsion engines (see Tab. 1), operating at steady speed and load conditions, over a range of power settings. The mode of operation was determined using the relevant, for ship propulsion engines test cycles, as specified in ISO standards 8178 part 4. The load of the engines was set at several levels - listed in Tab. 2, these being 25%, 50%, 75%, 100% of the nominal power. For each load and speed setting, the engine under test, was allowed to stabilise, prior to recording of the emissions. These confirmations were obtained from data logging profile that, the engine had reached a stable running conditions. Exhaust emission of individual engine cylinder is affected mainly by operational adjustment of fuel injection assembly, compression ratio and scavenge uniformity. Combustion process is specifically identified by quantitative values (indicated mean effective pressure, maximum pressure, etc). Variation can be observed and analyzed between cylinders or within a specific cylinder over a certain number of cycles.

*Tab. 1. Test engines details*

| Engine | | Nominal rate | |
|---|---|---|---|
| Designation | Maker/Type | Power [kW] | Speed [revs/min] |
| Ship propulsion | MAN B&W; 6S60MC-C | 13530 | 105 |
| Ship propulsion | MAN B&W; 7S60ME-C | 15820 | 105 |

*Tab. 2. Engines test program*

| 6S60MC-C | | | | 7S60ME-C | | | |
|---|---|---|---|---|---|---|---|
| Power | | Speed | | Power | | Speed | |
| kW | % | rpm | % | kW | % | rpm | % |
| 14911 | 110 | 108.3 | 102.85 | 15820 | 100 | 105 | 100 |
| 13559 | 100 | 105.1 | 100 | 11868 | 75 | 95.4 | 90.9 |
| 11951 | 88.3 | 100.8 | 96.0 | 7910 | 50 | 83.3 | 79.3 |
| 10191 | 75 | 95.5 | 90.95 | 3955 | 25 | 66.1 | 62.9 |
| 6780 | 50 | 83.5 | 79.5 | | | | |
| 3389 | 25 | 66.2 | 63.0 | | | | |

Since the nitric oxide formation process is strongly affected by the local - zonal gas temperature, each engine cylinder will produce cycles with different gas properties. Thus nozzle configuration can have potentially important effect on NO emission. This can possibly lead to improving emission characteristics in marine slow speed engines. Cylinder combustion pressure can also be used as a basic factor of injection assembly indication. Each combustion pressure element exhibits typical, distinctive characteristic regarding timing (in crank angle) and shape.

## 4. CFD analysis

Model geometry and mesh

A complete surface model of the combustion space consisting of piston crown, cylinder head combustion dome with axial operated exhaust valve and cylinder liner was created in CAD system [4]. The volume model generated in STL file format was then transferred to Vectis[1]. This format assumes the whole defined combustion chamber as a set of triangular faces. Fig. 3 shows the geometry of the cylinder head dome combined with exhaust valve, and piston crown surfaces and the volume model and mesh produced from TDC geometry.

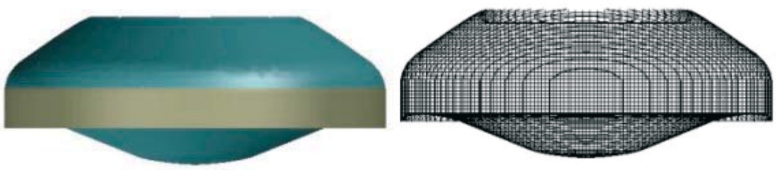

*Fig.3. Combustion chamber model and mesh domain at firing position*

All combustion chamber surfaces were classified as model boundaries with different temperatures. Piston motion was described with different time-dependant steps. The mesh set was automatically generated by Vectis, with improved local refinement around important boundaries in order to accurately capture geometry details. Throughout analysis the number of mesh cross-links varied as to maintain possibly highest quality of the moving mesh (minimal cell distortion). During the calculation, the mesh structure automatically deforms in a way that minimizes the individual cell distortion.

[1] Ricardo Company

Initial and boundary conditions

Time dependent inlet boundary conditions were derived from one dimensional intake and exhaust system simulation model, using Ricardo Wave code. The model was analyzed with engine test-bed operation conditions matching to experimental data. For cylinder liner and combustion chamber the total instantaneous surface heat flux was calculated by solving the unsteady heat conduction equation. Heat transfer was assumed to be strictly one-dimensional, perpendicular to wall surface. The heat transfer between gas and walls was calculated on the basis of mean bulk gas temperature and heat transfer coefficient based on Woschni's heat transfer model. Original Woschni model can be rewritten as:

$$h_w = L^{m-1} p^m T^{0.75-1.62m} \left( C_1 \bar{S}_p + C_2 \frac{V_d T_r}{p_r V_r} \left( p - p_{mot} \right) \right)^m \tag{1}$$

where:

$$Nu = 0.035 \mathrm{Re}^m$$

The highest temperatures on liner surface were found at locations in upper part of cylinder liner-firing ring, close to the head. The in-cylinder condition initial values at scavenging completion were adjusted to match engine factory settings. The initial in-cylinder mean flow was assumed to have an axial velocity component varying from piston surface to the cylinder head surface. The initial fluctuating velocity field was set at the intake port closing time. Accuracy of initial pressure and gas temperature must be high as the total mass is determined by the predicted flow field.

Fuel injection

In order to determine the influence fuel shape of injection profile created by standard nozzle, investigations were conducted with typical nozzles, mounted in the S60MC-C engine cylinder. Each engine cylinder unit is equipped with two injection valves, mounted oppositely in cylinder cover. The standard nozzle, with five holes has the spatial arrangement presented in Fig. 4.

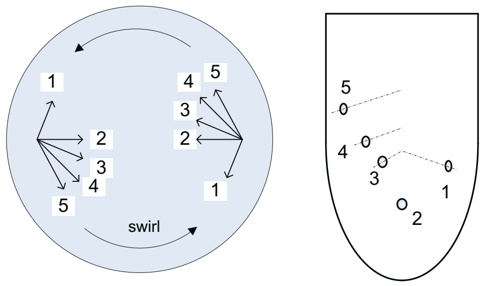


*Fig. 4. Fuel valve and nozzle holes arrangement*

Accurate predictions of fuel behavior within the injection system require sophisticated hydraulic models. However, to achieve only approximate estimates of the injection rate trough the injector nozzle flow a following method was assumed. In cases when flow through nozzle is quasi steady, incompressible and one dimensional, the mass flow rate of injected fuel is given by:

$$\dot{m}_f = C_D A_n \sqrt{2 \rho_f \Delta p} \frac{\Delta \varphi}{360n} \tag{2}$$

## 5. Results and discussion

The exhaust gas mass flow and combustion air consumption are based on exhaust gas concentration and fuel oil consumption measurement. Universal method, known as carbon/oxygen-balance, which is applicable for fuels containing H, C, S, O, N in known composition is used. As the $NO_x$ emission depends on ambient air conditions, measured $NO_x$ concentration was corrected for ambient air temperature and humidity. Fig. 5 shows the measured $NO_x$, CO, THC concentrations over a full spectrum of engine load - 6S60MC-C at steady state operating conditions, and similar records of 7S60ME-C engine in Fig. 6. Extracted and averaged values of $NO_x$ concentration were used as a comparison basis for calculation results.

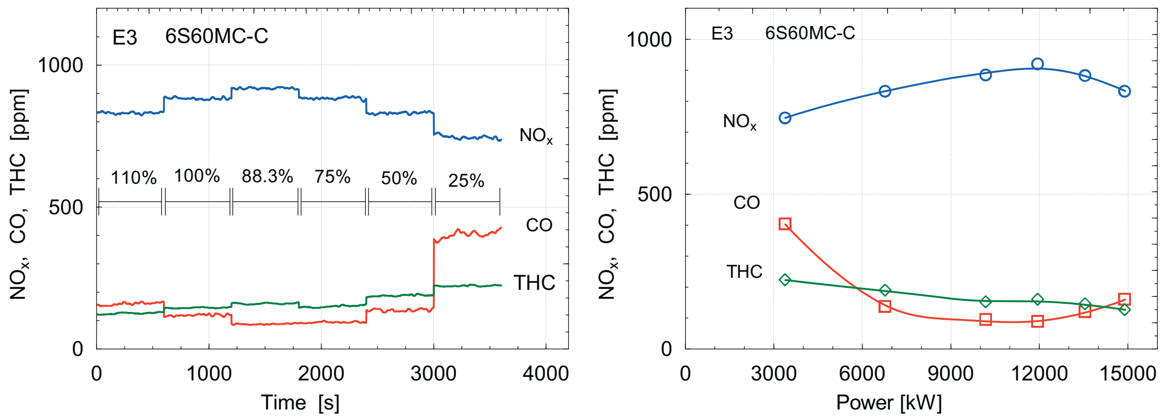


*Fig. 5. Measured raw exhaust gas emission component profile of 6S60MC-C engine*

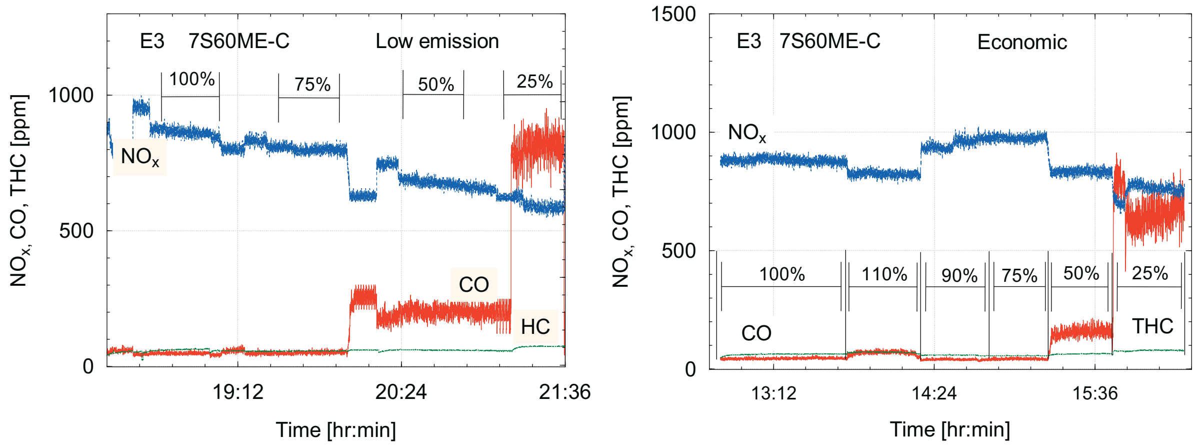


*Fig. 6. Measured raw exhaust gas emission component profile of 7S60ME-C engine in low emission and economic mode*

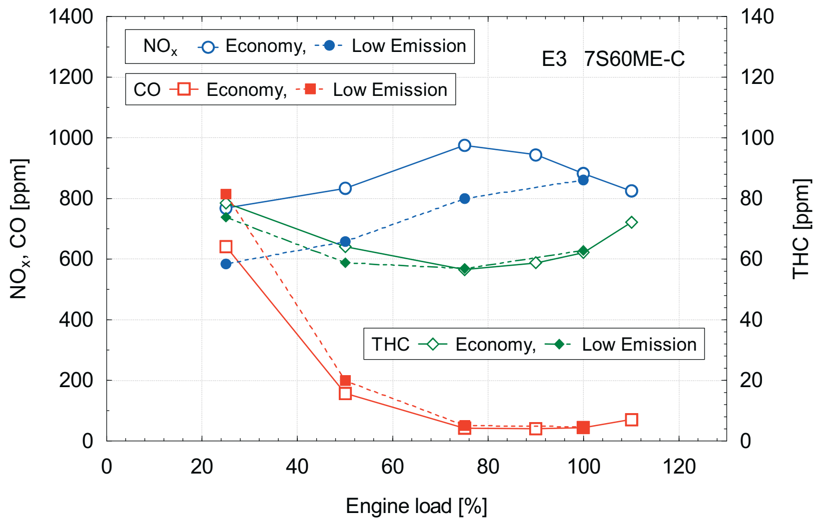


*Fig. 7. Average emission component profile comparison of 7S60ME-C engine in two modes of operation*

Once all the exhaust emission measured data had been collected and averaged for each engine load respectively. The two investigated engine modes of operation presents similar $NO_x$ formation rate under nominal load.

Since part load operation, particularly below 50% indicates the impact of air-fuel ratio, engine „eco-mode" gives higher average $NO_x$ concentrations with similar other properties composition (see Fig. 7). Current analysis concerns one cylinder unit with standard injection nozzle of the SMC-C engine. The baseline simulation of spray and combustion with fully-functional injection nozzles condition showed reasonable agreement between measured and calculated pressures Fig. 8. The main difference in cylinder pressure trace is located around TDC and was caused by combustion delay. The measured pressure after TDC shows slight decrease. This is due to the ignition delay period when colder fuel and hot air are mixing with each other, while predicted trace rises against measured trends. The auto ignition model has the probability scaling factor that modifies overall reaction rates. The sensitivity of the created model has been tested by prediction of combustion with short ignition delay (1) and one calibrated to experimental trace (2).

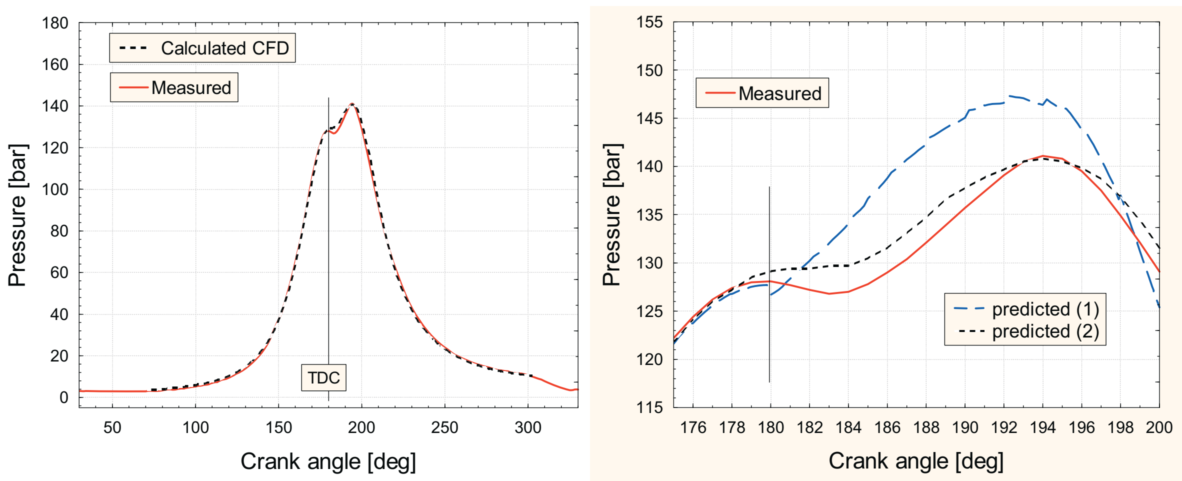


*Fig. 8. Comparison of calculated and measured cylinder pressure under 100% effective load*

Combustion process results expressed in NO distribution are presented on adequate cut plane arrangement diagrams - Fig. 9. The fuel is injected at a specified angle with respect to the horizontal axis. Due to flow created by piston upward motion it is forced to strike very close to cylinder axis and piston surface. Then, fuel-air mixture spreads along piston surface and gives distinctive temperature rise. Figure shows the baseline were NO highest densities are placed in post-flame regions. That is due to slow NO formation chemistry. NO develops within hot, post-combustion gases after combustion completion, in section where flame jet had traveled. The whole flame structure is equally spread out thanks to moving fuel-air mixture. Temperature is at maximum value within range 20-25 CA after TDC and it corresponds to highest NO mass fraction density. NO formation regions can be identified with fuel spray location and its dynamic actions.

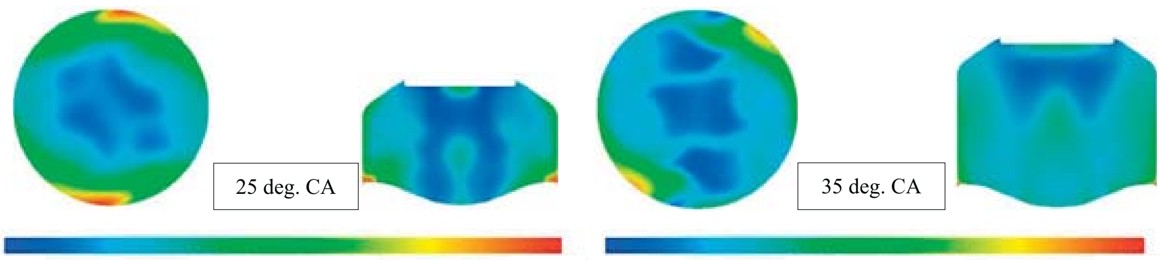


*Fig. 9. NO formation distribution with standard fuel injection valve operation*

## 6. Conclusions

The CFD simulation of combustion and pollutant formation was created to represent processes of a marine large bore engine. The SMC-C engine operation and standard injection nozzle design was taken as a baseline for comparison with measured NO concentration. The predicted cylinder pressures showed reasonable agreement with experimental measurements . The predictive ability of used CFD simulation package was found to be quantitatively insufficient in terms of $NO_x$ emission. However, the predicted NO emission exhibits similar sensitivity with experiment results. The quantitative set of reference data for the validation of simulation results needs to be substantially improved. The determination of fuel injection and cylinder pressure history can be identified as the most important source of inaccuracy. Finally, weighted $NO_x$ specific emission factor for both engines shows close value, sufficiently placed against IMO limit presented in Fig. 11. The results are promising and show that controlled combustion process is capable of fulfilling present and future $NO_x$ emission requirements. Modern marine engines are designed with adjustable components to allow the engine to be adjusted for maximum efficiency or $NO_x$ weighted emission factor when used in particular application.

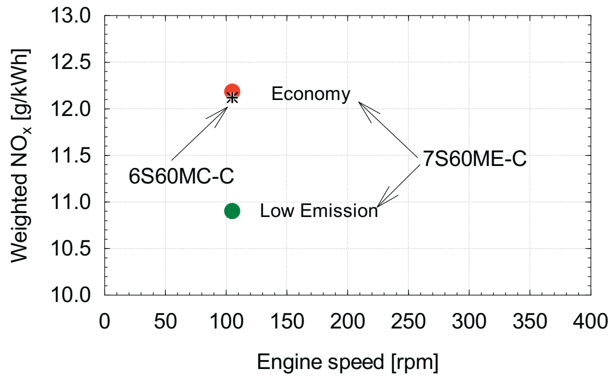


*Fig. 11. The total weighted $NO_x$ emission factor of 6S60MC-C and 7S60ME-C engines*

### Nomenclature

| | |
|---|---|
| $A_n$ | nozzle minimum area |
| $B$ | cylinder bore |
| $C_1$, $C_2$ | constants related to engine speed |
| $C_D$ | discharge coefficient |
| $\Delta\varphi$ | nozzle open period |
| $\Delta p$ | pressure drop across the nozzle |
| $n$ | engine speed |
| $p$ | instantaneous cylinder pressure |
| $p_{mot}$ | motored pressure |
| $p_r$ | working fluid pressure* |
| $\rho_f$ | fuel density |
| $T_r$ | working fluid temperature* |
| $w$ | local average gas velocity |
| $V_d$ | displaced volume |
| $V_r$ | working fluid volume* |

* at reference state (initial condition)

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