THE NO_X EMISSION PREDICTION IN DIESEL EXHAUST FOR LARGE BORE MARINE ENGINES

Tadeusz Borkowski

Maritime University
ul. Wały Chrobrego 1/2, 70-500 Szczecin, Poland
tel.: +48 91 4809419, fax: +48 91 4809585
e-mail: tborkowski@am.szczecin.pl

Abstract

The general IMO procedure was adopted with the main aim to achieve compliant engine performance with current environmental legislation. The paper investigates possibility of multi-zone diesel combustion model use to estimate NO_x emission. The model calculation results are compared with the experimental data obtained for marine diesel engine - Sulzer A20/24. An experimental study is conducted to investigate the use of phenomenological multi-zone model to predict NO_x emission, based on marine diesel engine operation parameters. The predicted and experimental results comparison shows reasonable agreement only within high engine loads, while serious discrepancies were found in low range operating effective power.

Keywords: *marine diesel engine, exhaust emission, combustion modeling, performance*

1. Introduction

The ever more stringent exhaust emission standards have prompted the development of marine engines systems and associated controls in order to reduce the nitric oxide. While NO_x sensors exist, that provide robust continuous measurements in diesel engines at reasonable price, the present work focuses on indirect approach. Combustion pressure is analyzed for specific quantitative values (indicated mean effective pressure, maximum pressure, firing pressure angle, cycle variation, etc.), level during certain events (valve closures and opening, peak rate of pressure rise, etc.), as well as data variation between cylinders and within a specific cylinder over a certain number of cycles. The final aim is a calculation of indicated power for each cylinder as well as the total engine power derived from the measured cylinder pressures. Engine cylinder pressure analysis is also used to balance and tune the engine: valve and fuel injection timing as well as fuel rate and compression. Development of engine diagnostics calls for improving the reliability of the engine. One of the methods used to meet these requirements is a convenient and real-time cylinder pressure measuring system. Much effort has been spent developing combustion models that will adequately predict engine performance and pollutant formation. Both multi-zone and multi-dimensional models are computationally all-absorbing for utilization within an engine system simulation, so a need exists for simpler, faster, and reasonably predictive models associated with zero-dimensional (time dependent) approaches. Generally, zero-dimensional (time dependent) models tend to have a highly empirical nature and thus are limited to particular combustion system designs. Phenomenological multi-zone models take into account the spatial differences in temperature and chemical compositions by dividing a cylinder into two or more zones. Each zone is treated as a well-mixed open thermodynamic system. Because the model can give local temperature and compositions, it can be used to predict exhaust emissions. Their predictive ability and accuracy greatly depend on what methods and sub-models are used. Many aspects of multi-zone models, for example the accuracy of simultaneous prediction of exhaust emissions and cylinder pressure, still need to be improved.

The aim of this paper is to adopt a phenomenological multi-zone combustion model for use in steady condition analysis. The model should be able to predict some aspects of engine performance and cylinder-averaged quantities such as NO_x emissions. However, it is impossible for the phenomenological models give the detailed in-cylinder data flow and temperature fields. Nevertheless, compared with the CFD model, the phenomenological multi-zone model provides a simple way to obtain full cycle simulation.

2. Model aspects

Engine combustion model is based on the undisturbed turbulent gas jets, also referenced as the "Cummins model", where diesel spray is treated as quasi-steady gas jet penetrating into gaseous environment of combustion air [1], [2], [3]. The concept of the model assumes that the liquid fuel injected into a combustion chamber as several jets is divided into many small zones. In large marine diesel engines chamber, the air flow is essentially quiescent. All combustion events in each zone: droplet break-up, evaporation , air–fuel mixing, ignition, premixed heat release, mixing-controlled heat release, heat transfer and formation of exhaust emissions, are calculated in order to achieve zonal temperature and compositions. The zonal property is representing by the average state of temperature, air-fuel ratio and NO_x concentration. The model contains: spray development, air entrainment and mixing, droplet evaporation, heat transfer, combustion and NO_x formation sub-models. In order to estimate rates of combustion and pollutant formation, a set of evolving discrete combustion zones is superimposed on the continuous calculated fuel-air distribution (Fig. 1).

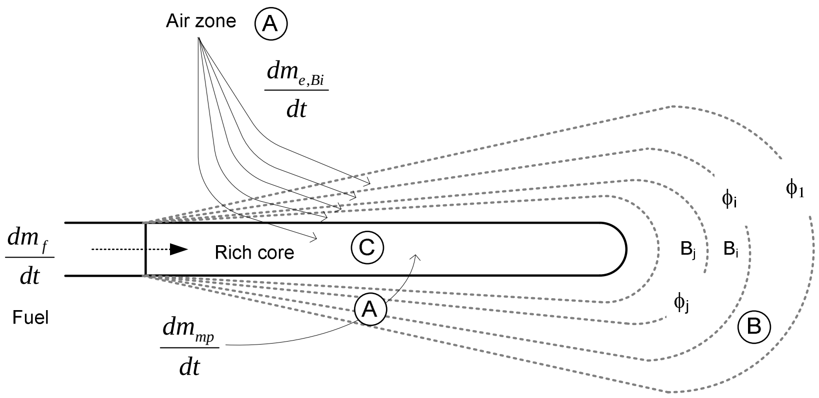

Fig. 1. The evolution of combustion zones and air entrainment

Entrainment of air into the jet is assumed to take place at each point along the jet surface at rate proportional to the velocity difference between the jet and surrounding air at that point. The considered chamber is divided into number of zones at incipient ignition, air zone corresponds to the medium into that the fuel is injected (d_{mf}/dt) – rich core and n zones B_i of combustible mixture between the lean and rich flammability limits. During the course of combustion the fuel mass in each zone B_i remains constant and the additional fuel that is diluted beyond rich flammability limit is assigned to an additional combustion zone B_j. In contrast to these constants, fuel masses in zones B_i, where air is continuously entrained ($dm_{e,Bi}/dt$) into the zones B_j and rich core. The entrainment rate into a zone is simply equal to the change of mass of this zone, because the fuel mass remains constant. The contribution of each zone to the total wall heat transfer is based on the product of zone mass and temperature:

$$\frac{dQ_{w,z}}{d\varphi} = \frac{m_z t_z}{m_A T_A + m_C T_C + \sum_{i=1}^{j} m_{Bi} T_{Bi}} \cdot \frac{dQ_{w,tot}}{d\varphi}, \tag{1}$$

The change of composition and internal energy of each zone, caused by the mixing between the zones can be determined similarly to energy and mass balance equations for various zones. Thus, NO_x concentration caused by the mixing can be calculated from the following equations [4]:

$$\frac{d[NO]_{Ci}}{dt} = \frac{d[NO]_{Ci,A}}{dt} + \sum_{i=1}^{j} \frac{d[NO]_{Ci,B}}{dt}, \tag{2}$$

$$\frac{d[NO]_{Ci,A}}{dt} = \frac{dm_{Ci,A}}{dt} \frac{[NO]_A}{m_A}, \tag{3}$$

$$\frac{d[NO]_{Ci,B}}{dt} = \frac{dm_{Ci,B}}{dt} \frac{[NO]_{Bj}}{m_{Bj}}, \tag{4}$$

In order to enable prediction an emissions formation in terms of NO_x, the model has been equipped with the widely accepted extended Zeldovich model [1]:

$$N_2 + O = NO + N \qquad k_{1,f} = 7.6 \cdot 10^{13} exp[-38000/T] cm^3/(mol \cdot s), \tag{5}$$

$$N + O_2 = NO + O \qquad k_{2,f} = 6.4 \cdot 10^{9} exp[-3150/T] cm^3/mol \cdot s, \tag{6}$$

$$N + OH + H \qquad k_{3,f} = 4.1 \cdot 10^{13} cm^3/(mol \cdot s), \tag{7}$$

The formation rate of NO can be written as:

$$\frac{d[NO]}{dt} = k_{1,f}[N_2][O] + k_{2,f}[N][O_2] + k_{3,f}[N][OH] - k_{1,r}[NO][N] - k_{2,r}[NO][O] - k_{3,r}[NO][H], \tag{8}$$

after simplification:

$$\frac{d[NO]}{dt} = 2k_{1,f}[N_2][O] - 2k_{1,r}[NO][N], \tag{9}$$

3. Experimental details

Basically, multi-zone models should be checked against experimentally derived heat-releases profiles and recalibrated if necessary. The main objective was to predict emission, with NO_x as the primary target, of the Sulzer 6AL 20/24 engine. This engine has been used as the test engine for examination. The first step of the experiment was to establish performance parameters with greatest possible accuracy. Further, comparing the results of model calculation against the experimental based results Using test bed engine and having the specification listed in Table 1, examinations were made to predict exhaust emission. The key injection parameter needed for combustion analysis is fuel injection pressure and time of the start, when fuel first enters the cylinder. Injection pressure data and dynamic start of injection were identified as one of the key parameters that have a great impact on the qualitative and quantitative aspects of the combustion process. Hence, a strain gauge transducer measurement technique which is readily available, has been adopted to determine injection pressure data. The task includes identification of the period of fuel injection as a function of crank angle, and the measured fuel consumption.

Tab. 1. Test engine details

Engine type	Sulzer 6A20/24, non-reverse.
Number of cylinder	6
Bore/Stroke [mm]	200/240 [mm]
Rated engine speed	720 [rpm]
Output	397 [kW]
Compression ratio	14
Brake mean effective pressure	1.47 [MPa]

The combustion pressure as well as injection pressure was continuously monitored and recorded by the fast data acquisition system. It is known that the injector chamber pressure (the fuel space around the needle valve seat) varies widely during the period the injector valve is open, so the representative value is difficult to define. 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:

$$\dot{m}_f = C_D A_n \sqrt{2\rho_f \Delta p}\,\frac{\Delta\varphi}{360n}, \tag{10}$$

where:

C_D - discharge coefficient,
A_n - nozzle minimum area,
ρ_f - fuel density,
Δp - pressure drop across the nozzle,
$\Delta\varphi$ - nozzle open period,
n - engine speed.

Defined heat release in functional form was chosen to match experimentally observed heat-release profile. The static injection timing, for present experiment was kept constant by engine design. The combustion pressure as well as injection pressure was recorded by the fast data acquisition system. It is known that the injector chamber pressure (the fuel space around the needle valve seat) varies widely during the period the injector valve is open, so the representative value is difficult to define. Therefore a different definition is given, it requires the use only one pressure transducer located in the fuel line near the injector and is easily used in practice. Basic fuel injection equipment characteristic and settings for nominal load presented in Table 2.

Tab. 2. Fuel oil injection equipment settings

Fuel pump setting		
Commencement stroke	[m]	0.004
Injection start	[deg]	-19.0
Effective stroke	[m]	0.006
Delivery completion	[deg]	+9.5
Injection nozzle		
Opening pressure	[MPa]	25.0
Spray angle	[deg]	159
Number of spray holes	[-]	7
Spray hole diameter	[m]	0.00026
Needle lift	[m]	0.0005

Subsequently, the trials were performed to assess the engine operation on consequent emission profiles. Emission measurements were carried out on engine at steady-state operation. All engine performances were continuously, together with exhaust gas components concentration recorded [5]. The performance measurement procedure of marine engines on test beds, performed in accordance to Annex VI of Marpol 73/78 convention - with the specification given in the IMO NO_x Technical Code and ISO standards [4]. To reduce emissions variability, all tests performed with the selected marine distillate fuel DMX [6]. Today on conventional, commercial diesel engines there are no sensors available that can be mounted directly into the combustion chamber. Therefore for this project a marine engine electronic indicator (Premet-Lemag*) was chosen.

4. Results and discussion

There are two prerequisites for being able to perform combustion experimental analysis, the fuel injection period and cylinder pressure. Both needs to be determined with great accuracy and has to be measured during the same cycle. Figure 2 illustrates the set of measured injection and cylinder pressures taken within the test cycle engine load range. Subsequent analysis of the in-cylinder pressure history, that will produce the rate of heat release profile. The pressure history displayed in Fig. 2. represents the experimental data and the simulation results obtained with the measured fuel injection input files It is shown that there is a certain discrepancy between the model and measured results. As the engine load and in-cylinder temperature decrease, the process of droplet evaporation becomes more important and is dominant.

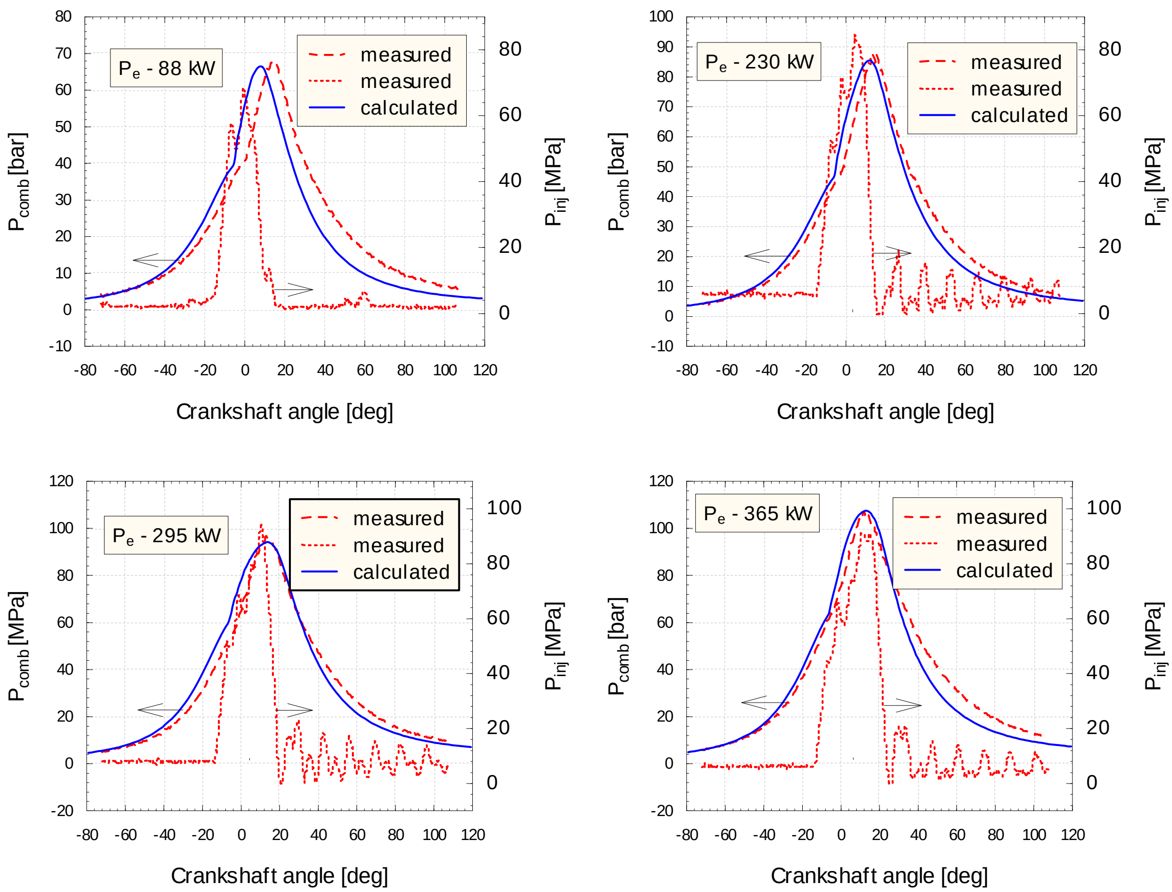

Fig. 2. Comparison of calculated and measured cylinder and injection profiles at different engine load

* Lehmann & Michels GmbH & Co. KG, Germany

Figure 3 shows rate of heat release profile comparison – model and experimental (resulted from cylinder pressure) illustrating the premixed spike during the first phase of burning and the smaller gradients during the diffusion part of burning. From the figure, it can be seen that there is only one peak in the heat release diagram for model results, while measured traces have not exhibit such occurrence. The rate of air-fuel mixing in a quiescent combustion chamber depends mainly on the fuel injection process. The key injection parameter needed for combustion studies is the actual dynamic start of injection, when fuel first enters the cylinder. During an ignition delay period, a substantial amount of combustible mixture will be formed and a significant part of the fuel will burn rapidly in the premixed mode, immediately after ignition. Once the premixed fuel is used-up, combustion becomes diffusion-controlled. If the delay is very small, depending on the conditions at the end of compression and the mixing process itself, burning is largely diffusion-controlled. The ignition delay and the rate of heat release are the key phenomena that need to be analyzed in order to characterize and quantify the burning process. Therefore, the influence of individual engine or fuel parameters can not be investigated without considering secondary influences. The fuel spray on break-up atomizes to a large number of droplets with a diameter equal to the Sauter mean diameter (SMD). The droplet heating and evaporation take place under different zone conditions according to the correlations of Hiroyasu [6]. The calculation have taken into account the formation of the liquid fuel spray and its interaction with the atomization, break-up, vaporization and combustion. In addition, the testing at low temperature in engines (low load) proved to be difficult. As already observed model results profiles – RHR, has clearly sketched first stage of combustion, that is advanced 2-4 CA degrees to the experimental traces. The largest discrepancy covers low engine load. Furthermore, it can be deduced that ignition delay of real diesel engine requires more deflected curve based on exponential equation [8].

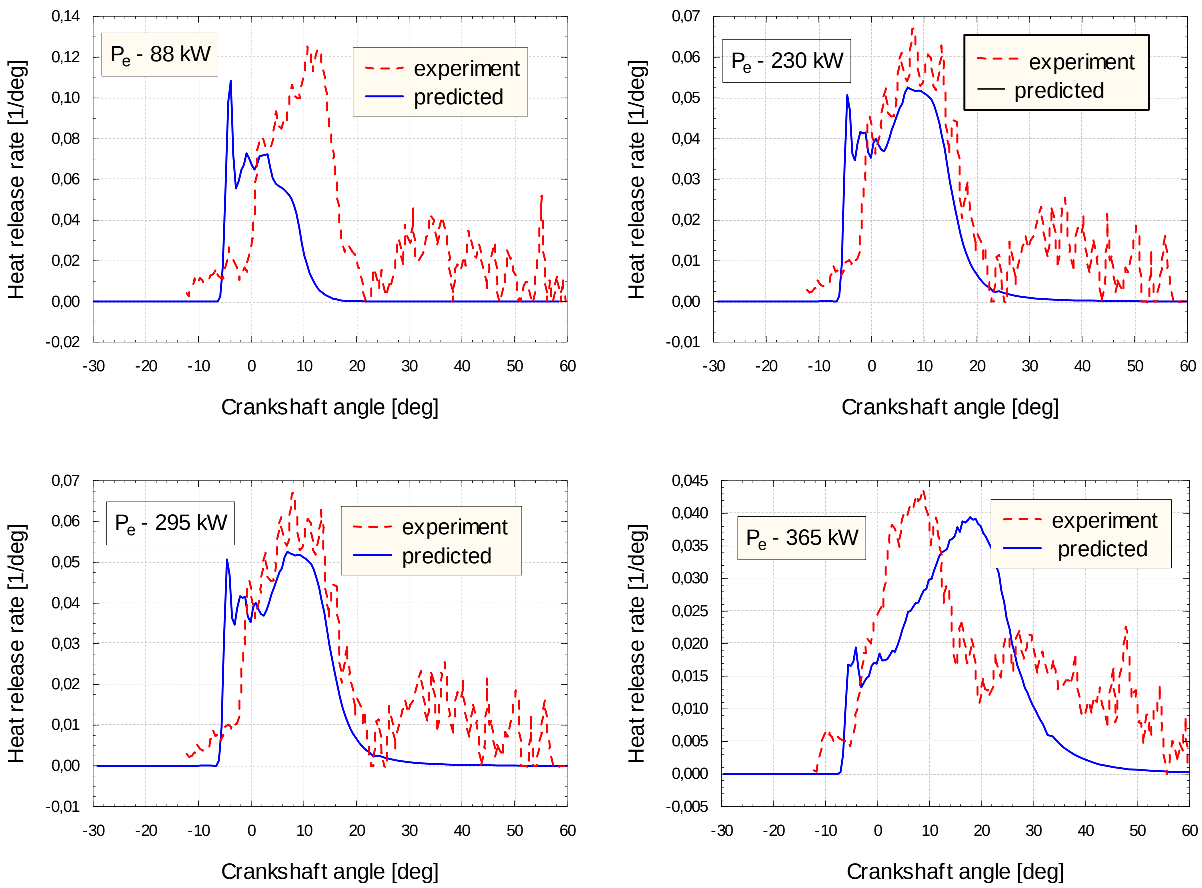

Fig. 3. Comparison of rate of heat release history comparison at different engine load

Figure 4 shows the baseline picture were NO highest densities are placed in post-flame regions - not in the highest temperature zones. 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 along chamber surface. Temperature is at maximum value within range 20-25 CRA 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. It should also be noted that space in central part of the cylinder exhibit lower NO mass fraction density. During the expansion the NO chemistry freezes as effect of temperature decrease and its concentration remains constant.

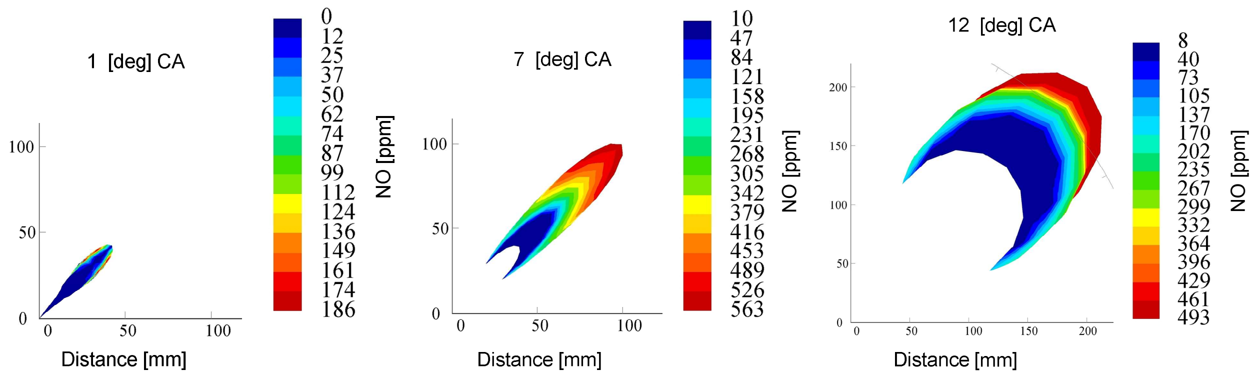

Fig.4. NO formation distribution with fuel injection spray

The spray simulation during normal operation showed droplet flow located in combustion chamber. However, a risk has arisen that some part of spray droplets can reach combustion chamber wall. Generally, the critical NO equivalence ratio formation in high-temperature and high-pressure burned gases is close to stoichiometric. The crucial time for its formation is when the burned gas temperature is at maximum, between the start of combustion and close to peak pressure occurrence. Early combustion process causes higher NO formation rate as combustion proceeds. Nearly final NO concentration is formed within the 25 CA following the start of combustion.

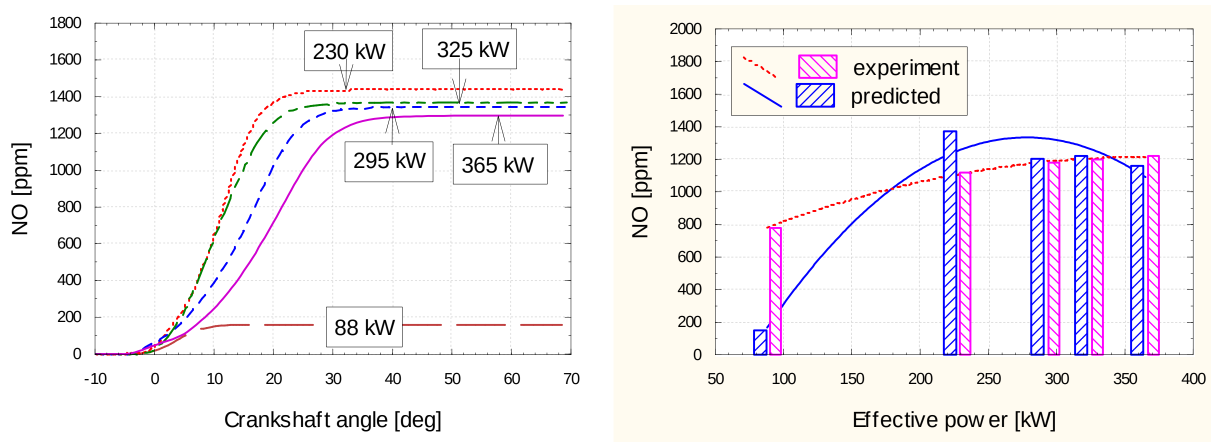

Fig. 5. Comparison of predicted (in cylinder) and measured (common outlet) NO_x emission

The investigated operating conditions expressed as the histories of NO_x concentration are displayed in Fig. 5. It can be seen that the NO_x concentrations level starts to rise after the start of combustion and continues to a maximum value. It has been shown that there are several CA delays of the start NO_x formation process, with the combustion commencing.

Later, while combustion proceeds NO_x concentration increases, up to maximum level at 10-20 CA with respect to highest cylinder pressure. However, the predicted value of NO_x emission is lower than experimental data under low part of the engine load. Coincidentally, along with the decrease of engine load, the discrepancy between the calculated and the measured cylinder pressure results becomes large.

5. Conclusions

An experimental investigation of diesel combustion contribution to NO_x emission has been tested and confronted with phenomenological multi-zone combustion model calculation results. It is found that predictive ability of the used model is partly proved – high engine load, by experiment results. There is an serious error concerning NO_x emission at low load, even if the maximum combustion pressure prediction and measurement shows reasonable agreement. Thus, model calculation influenced by ignition delay, specifically at low engine load, it needs to be modified as a further step. There is also a realistic problem to overcome – accuracy of combustion and injection pressure measurement [9]. The sub-models of spray development and air entrainment have to be improved to increase model predictive ability. Anyhow, the results are promising and shows that the phenomenological multi-zone combustion model is capable of predicting the NO_x emission.

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