THE EFFECT OF FUEL INJECTION NOZZLE ON COMBUSTION AND NO_X FORMATION OF MEDIUM SPEED MARINE DIESEL ENGINE

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Abstract

In recent years CFD calculation has been successfully used for mixture formation, fluid flow, combustion and pollutant formation investigations in diesel engines. Numerical simulation was carried out to investigate the effect fuel spray characteristics in marine medium speed diesel engine – Sulzer 6A20/24D. KIVA-3V code with a fuel spray model adopted for simulation atomization and combustion process. The fuel injection nozzle and other components of the engine were exchanged for combustion improvement. For better understanding, calculation and experimental test, all results are compared. Hence, standard marine engine test cycles performed to asses basic parameters caused by injection equipment retrofitting. Cylinder pressure, exhaust emission histories for full load range of the engine experimentally established.

Introduction

A decade ago, the marine industry paid little attention to air pollution that changed when the amended IMO exhaust emission regulations call for all engines installed on ships or those engines which have undergone a major conversion. Before the statement is given, the ship-owner must provide a Technical File which should include a NO_X measurement report. The Technical File is a record containing all details of parameters, including components and settings, which may influence the NO_X emissions of the engine. The Technical File contains among others: injection nozzle and pump details. Depending on engine type and age, different actions may be required to fulfil the NO_X emission level requirements from minor adjustments to upgrading. Whether or not the engine has to be adjusted, emission measurements, according to specific IMO regulations, have to be carried out and complete test protocol has to be produced specifying the engine parameters e.g. speed, load, etc.

Furthermore, in view of evolving engine management systems, injection profile could offer the flexibility of changing the emission characteristics during the ships operation. This could be helpful in entering area where emission may be restricted in the future by new legislation. Then, even if static injection timing is constant that specifies speed-torque combinations and defines significantly injection characteristics, thus consequently rate of heat release and exhaust emissions. However, the injection rate trough the injector nozzle flows that can be estimated or measured [1]. On a Sulzer A20 type engines the fuel injection was improved mainly to better engine behaviour, when running under part load conditions e.g. to avoid post injection, reducing the possibility of cavitation on the nozzle tip and also to reduce the smoke emissions. This improvement was introduce under the name of “Flow Controlled Injection” and entails modification of the fuel pumps as well as replacement of the fuel nozzles which are working with increased opening pressure of the fuel valve.

In recent years, mathematical models have been use for optimization process fuel injection components. CFD simulations also provide insight of fluid dynamics of fuel spray, and its interaction with the in-cylinder charge motion. It is a common practice to specify some injection parameters at the nozzle exit and initial droplet distribution. Velocity data can be produced using the steady state flow through the nozzle, whereas, the droplet distribution data are typically obtained from experiment. In the present study, parametric studies were carried out with varying nozzle exit diameter and different pump plunger concerned above-mentioned retrofitting.

1. Experimental details

1.1. Test bed engine operation

The experimental efforts described below were an attempt to quantify emissions associated with engine fuel injection equipment retrofitting. Using test bed engine, having the specification listed in Table 1, examinations were made of the effect of fuel nozzle and injection characteristic on exhaust emission.

Table 1. Test engine details

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

The static injection timing, for present experiment was kept constant by engine design. 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. A reasonable choice would be the injector opening pressure, this pressure should be measured at the moment the needle valve starts to open, for example it could be sensed by an injector needle lift measuring transducer. This method is complex and associated with technical difficulties. 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 before and after retrofitting presented in Table 2.

Table 2. Fuel oil injection equipment settings

Fuel pump details		Old design	New design
Commencement stroke	[m]	0.004	0.0042
Injection start	[deg]	-19.0	-18.5
Effective stroke	[m]	0.006	0.0075
Delivery completion	[deg]	+9.5	+18
Injection nozzle details			
Opening pressure	[MPa]	25.0	40.0
Spray angle	[deg]	159	159
Number of spray holes	[-]	7	9
Spray hole diameter	[m]	0.00026	0.00023
Needle lift	[m]	0.0005	0.0005

Subsequently the comprehensive series of trials were performed to assess the extent of these influences on engine operation and 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 [2]. 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-8178 standard [3], [4], [5].

All tests were covered by test-cycles procedure D-2 and E-2, which include generator and pitch propeller drive. To reduce emissions variability due to fuel variables, all tests performed with the selected marine distillate fuel DMX in accordance to ISO standard. During the test, engine was running on distillate fuel ISO-F-DMA [6]. Samples of the fuel being burnt were taken at the time of the trial for analysis and analyzed in accordance with standard industry procedure. An evaluation is given in Table 3.

Today on conventional, commercial diesel engines there are no sensors available that can be mounted directly into the combustion chamber. Therefore a compromise has to be found between existing cylinder pressure sensors, mounting possibilities and accuracy of pressure measurement. A multitude of commercial equipment and software packages are available to facilitate the cylinder combustion pressure data. For this project a marine diesel engine electronic indicator (Premet-Lemag*) was chosen. The usual way to measure cylinder pressure is to use an indicator pipe with indicator cock for transient measurement. This gas passage in turn has an influence on the accuracy of the pressure measurement at the location where the sensor is mounted. The pressure transducer was mounted on the indicator cock for measurement and then moved from one cylinder to another in order to complete the measurement on all cylinders.

Table 3. Fuel oil characteristic

Determination			Test results
1	Density @ 15 °C	kg/m^3	852
2	Viscosity @ 40 °C	mm^2/s	4.8
3	Flash point	°C	62
4	Conradson Carbon (CCR)	%	0.003
5	Calorific value	MJ/kg	42.70
6	C	%	84.40
7	H	%	13.34
8	N	%	0.14
9	S	%	0.43
10	O	%	1.69
11	Ash	%	0.002

1.2 Calculation method

The computer code used in the study was KIVAII-3V which is an updated version to resolve moving valve problems [7]. KIVA codes can solve unsteady compressible turbulent flows with combustion and fuel spray, and have been used for the computation of various internal combustion engines. It uses a standard k-epsilon turbulence model with wall functions and a quasi-second-order upwind scheme for convection. The package includes a basic grid generator, that is not intended to generate the really sophisticated geometries, but it can define a useful block, allowing moderately complex geometries to be constructed in a reasonable amount of computer time. Further, the user can modify the various subroutines in to tailor it to specific needs. Then, an automatic grid generation program used to generate a block structured hexahedron grid and typical computational coarse mesh - total number of computational cells was about 180,000. In discussed application, the initial pressures and

* Lehmann & Michels GmbH & Co. KG, Germany

temperatures in the cylinder and ports were experimentally established. Given this information, along with the equivalence ratio for fuel-air mixtures allowed the subroutine calculate the initial species densities as a check the input data.
Later, in reverse re-calculated using these densities. Proper program subroutine calculates a variety of quantities as a function of engine crank angle. All data is averaged over the cylinder and information is written onto several separate files for post-processing which contain cylinder dynamic data crank angle, swirl ratio and total kinetic energy. Thermo file contains cylinder thermodynamic data: crank angle, average pressure, average temperature, average density, total volume, and total mass. Injection file contains cylinder fuel injection data: crank angle, total fuel mass injected, total vapour mass, particle mass, vapour mass with equivalence ratio. These are normalized by the instantaneous cylinder volume.

For combustion runs, the data include energy balance data and emissions data. The energy balance line contains: the crank angle, the cumulative heat release from all chemical reactions, the cumulative change in total energy in the cylinder, the cumulative wall heat loss in the cylinder, and the cumulative pressure work done by the piston on the gas. The emissions data contains the crank angle and the amounts of CO_2, CO, and NO in the cylinder in parts per million. Computation started at the closing inlet port position and ended at the exhaust valve opening. Initial thermodynamic and turbulence quantities were specified to be uniform in the ports and the cylinder. In order to enable an emissions formation in terms of NO_x, the model has been coupled with the extended Zeldovich model. In order to enable an emissions formation in terms of NO_x, the model has been coupled with the extended Zeldovich model [8].

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

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

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

The formation rate of NO can be written as

$$\frac{d[NO]}{dt} = k_{1,f}[N_2][O] + k_{2,f}[N][O_2] + k_{3,f}[N][OH] - k_{1,r}[NO][N] - k_{2,r}[NO][O] - k_{3,r}[NO][H]. \tag{4}$$

After simplification

$$\frac{d[NO]}{dt} = 2k_{1,f}[N_2][O] - 2k_{1,r}[NO][N]. \tag{5}$$

The exhaust gas mass flow and combustion air consumption are based on exhaust gas concentration and fuel 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 [5].

1. Results and discussion

In order to determine the influence of different fuel shape of injection profile created by improved nozzle and plunger design investigations were conducted on two types of injection equipment mounted in the same engine. The investigated operating conditions of engine tests were similar and calculations done for each test has been carefully corrected by proper standard procedure [5]. The histories of measured combustion and injection pressure used to create fuel flow rate file of boundary conditions set are shown in Fig.1.

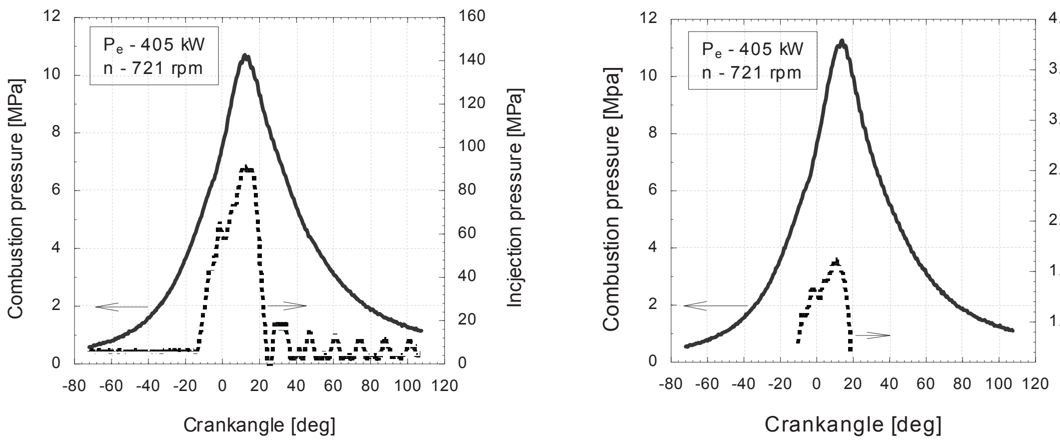

Fig. 1. Measured combustion and injection profile at nominal engine load

The procedure of checking the CFD program results was based on using the nominal load of the engine and the global parameters, these are: fuel consumption, total heat released, mean gas in-cylinder pressure versus crank angle and total exhaust NO_x, CO_2 and O_2 concentrations.

The Figure 2 shows rough comparison between the measured and predicted cylinder mean pressure versus crank-angle. The predicted cylinder pressures show very good agreement with the measurements, as this was the first step on the verification process. The agreement was examined and confirmed to apply the mass of fuel injected.

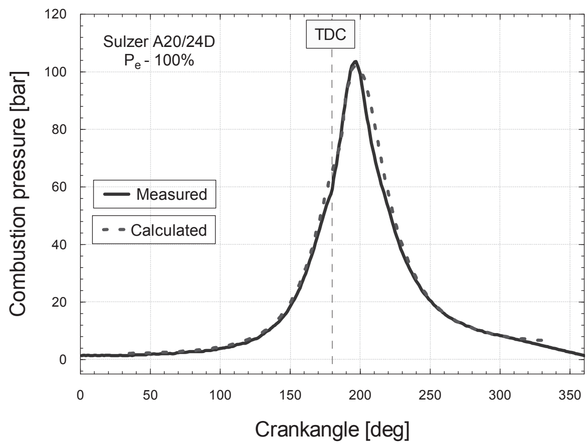

Fig. 2. Measured and predicted combustion pressure history at nominal engine load with old design of fuel injection equipment

The calculations have taken into account the formation of the liquid fuel spray and its interaction with the atomization, break-up, vaporization and combustion. The full load case calculation with start of injection at 19 (old design) and 18.5 (new design) degrees accordingly performed. The important output of this CFD investigation is represented by the predicted parameters shown in Fig 3. The injection pressure was modified to required start and stop of injection period. Since the mass of injected fuel was constant and the nozzle hole diameter smaller - it increases level of mixing at the early stage of combustion, resulting in increased NO_x emission as presented. As may be expected the cylinder pressure in both cases showed reasonable degree of reproduction. Judging from the predicted temperature and heat release curves the effect of injection duration is quite obvious.

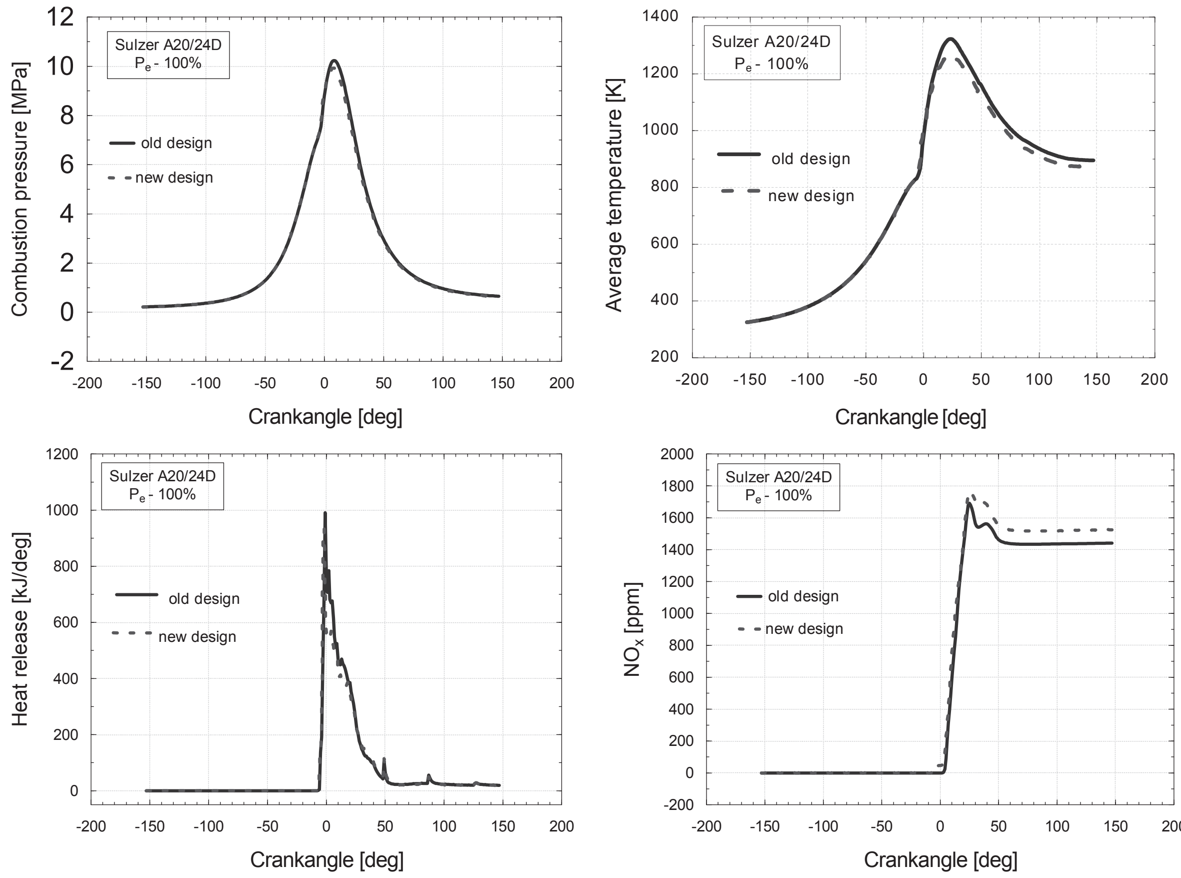

Fig. 3. Predicted combustion variables and NO_x average emission at nominal engine load with old and new design of fuel injection equipment

A comparison of the main measured engine variables, that consists: fuel consumption, cylinder pressure history, exhaust emission results with calculated flow rate presented in Tables 4 and 5 (Appendix). As a whole model calculation succeeds in predicting NO_x emission at full engine load, but a certain discrepancy in combustion pressure has been found. Measured NO_x emission profile and combustion characteristics caused by retrofitting of injection assembly are presented in Fig.4. The predicated combustion pressure has changed in opposite to experimental data under considered engine conditions; it supposed to be a largest error.

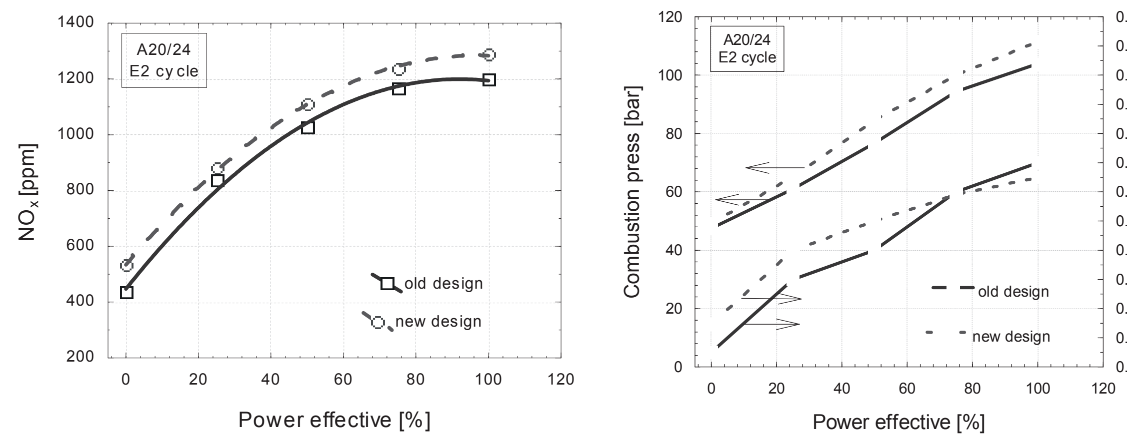

Fig. 4. Measured NO_x average emission and combustion profiles and at nominal engine load with old and new design of fuel injection equipment

2. Conclusions

An experimental investigation of fuel injection nozzle and other components contribution to NO_x emission has been tested and confronted with CFD calculation. Basic, relevant and important values of comparison presented in Table 6. It is found that predictive ability of the used model is proved by experiment results on a medium speed marine diesel engine. Because there is 8.9% error in maximum combustion pressure prediction, the model calculation and experimental data needs to be explaining as a further step. There is realistic problem to overcome – accuracy of combustion pressure measurement by means of standard passage and indication valve- cock.

Table 6. Comparison of measured and predicted combustion pressure and NOx emission

Operating condition/ trend		P_{max}[bar]		ΔNO_x [%]	
		Prediction	Measurement	Prediction	Measurement
100% P_e	Change	-1.3	+8.0	+5.8	+7.3
	Error [%]	8.9		1.5	

Finally, weighted NO_x emission factor for tested engine were found well above the limit, what can be found in uncontrolled marine diesel engine. Retrofitting of the injection nozzle and pump plunger does not change exhaust emission profile of the engine significantly.

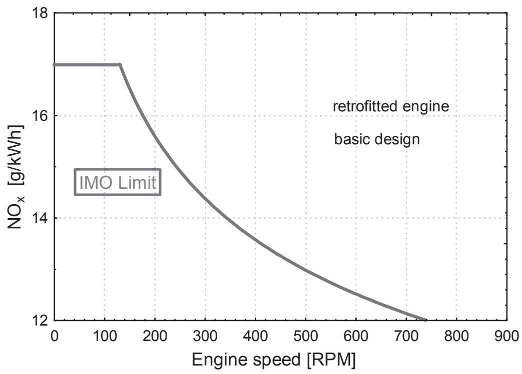

Fig. 4. Weighted NOx emission factor of Sulzer 6a20/24D engine against IMO limit before and after the fuel injection retrofitting

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Appendix

Table 4. Measured and calculated emission factors of engine equipped with old injection assembly

Test cycle mode	E2	1	2	3	4
Power	%	100.76	75.57	50.38	25.19
Speed	%	100.00	100.00	100.00	100.00
Comb. pressure - average	bar	103.9	94.4	76.7	61.0
Uncorrected spec. fuel cons.	g/kWh	216.3	223.7	231.5	261.0
NO_x (wet)	ppm	1200	1168	1023	833
CO (dry)	ppm	38	32	20	57
CO_2 (dry)	%	6.96	6.36	5.76	4.70
O_2 (dry)	%	10.70	11.50	12.50	13.90
HC (wet)	ppm	152	145	120	125
NO_x	kg/h	5.37	4.42	2.97	1.67
CO	kg/h	0.09	0.07	0.03	0.06
CO_2	kg/h	270	209	144	81
O_2	kg/h	299	273	226	173
HC	kg/h	0.232	0.187	0.117	0.083
SO_2	kg/h	1.47	1.14	0.79	0.44
NO_x specific	g/kWh	13.44	14.72	14.87	16.67
NO_x weighted	g/kWh	14.469			
CO weighted	g/kWh	0.241			
HC weighted	g/kWh	0.617			

Table 5. Measured and calculated emission factors of engine equipped with new injection assembly

Test cycle mode	E3	1	2	3	4
Power	%	100.76	75.57	50.38	25.19
Speed	%	100.00	100.00	100.00	100.00
Comb. pressure - average	bar	111.5	100.2	84.5	65.2
Uncorrected spec. fuel cons.	g/kWh	220.0	225.0	232.0	245.0
NO_x (wet)	ppm	1 288	1 239	1 107	881
CO (dry)	ppm	101	83	77	84
CO_2 (dry)	%	7.02	6.47	5.88	4.95
O_2 (dry)	%	10.63	11.41	12.23	13.53
HC (wet)	ppm	272	268	272	233
NO_x	kg/h	5.62	4.48	2.95	1.49
CO	kg/h	0.25	0.17	0.12	0.08
CO_2	kg/h	273	210	144	76
O_2	kg/h	299	267	216	150
HC	kg/h	0.415	0.339	0.259	0.138
SO_2	kg/h	1.41	1.08	0.74	0.39
NO_x specific	g/kWh	14.04	14.94	14.75	14.93
NO_x weighted	g/kWh	14.658			
CO weighted	g/kWh	0.602			
HC weighted	g/kWh	1.135			