DIESEL FUEL PROPERTY EFFECTS ON MARINE MEDIUM-SPEED ENGINE EXHAUST EMISSION

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Abstract

*The increasing demands for environmental protection in shipping are being recognised by **International Maritime Organisation**, through further development of the rules valid with respect to **sulphur and nitrogen oxides**. As the engine test-bed measurement is presently carried out mainly by **ISO procedure, the effects of diesel fuel property on exhaust emissions were evaluated for marine medium-speed engine. The popular diesel generator engine** was investigated - 8L32/40 MAN B&W. Exhaust emission and other **engine performance** measurement performed. A comparison with calculated combustion process was made. **Accumulation** of difference of load condition and measured value of emission characteristics gives basic **data and** prevail detailed studies by giving special attention to trend of fuel physical characteristics.*

1. Introduction

The ignition delay in a diesel engine was defined as the time interval between the start injection and the start of combustion. The start of injection is usually taken as the time when injector needle lifts of its seat – defined by a needle lift indicator. The start of combustion is more difficult to determine precisely. It is best identified from the change of heat release rate, determined from cylinder pressure data. Both physical and chemical process must take place before a significant fraction of the chemical energy of the injected liquid fuel is released. The physical process are: the atomization of the liquid fuel jet, the vaporization of the fuel droplets. The mixing of fuel vapor with air. These processes are affected by engine design, operating variables and fuel characteristics. Good atomization requires besides fuel injection pressure, cylinder air pressure and nozzle geometry – optimum fuel viscosity at the time of the injection. Accurate predictions of fuel behavior within the injection system require sophisticated hydraulic models. However, approximate estimates of the injection rate trough the injector nozzle flows that can be estimated or measured. If the flow through nozzle is quasi steady, incompressible, and one dimensional, the mass flow rate of injected fuel is given by [1]:

$$\dot{m}_f = C_D A_n \sqrt{2\rho_f \Delta p} \tag{1}$$

where:

A_n - nozzle minimum area,
C_D – discharge coefficient,
ρ_f – fuel density,
Δ_p – pressure drop across the nozzle.

When the pressure drop across the nozzle and the nozzle open area are essentially constant during the injection period, the mass of fuel injected per cycle is then:

$$m_f = C_D A_n \sqrt{2\rho_f \Delta p}\,\frac{\Delta\varphi}{360n} \tag{2}$$

where:

$\Delta\varphi$ – nozzle open period in crank angle degrees,
n – engine speed.

Above equations illustrate the dependence of injected amounts of fuel on injection system and engine parameters. Different spray configurations are used in the different diesel combustion systems. The common configuration used in marine engines design is the simplest and involves multiple sprays injected into quiescent air. Under diesel engine injection conditions, the fuel jet usually forms a cone-shaped spray at the nozzle exit. This type of behavior is classified as the atomization breakup regime and produces droplets with sizes very much less than the nozzle exit diameter. Since the ignition characteristics of the fuel affect the ignition delay, this property of the fuel is very important in determining diesel engine operating characteristic such as fuel conversion efficiency, smoothens of operation, misfire, emissions noise and ease to starting. Studies have shown that the temperature and pressure of the cylinder air are the most important variables for a given fuel composition. Many correlations have been proposed for predicting ignition delay as a function of engine and air charge variables. In the packet model, originally proposed by Hiroyasu [2] and later applied and extended by several other authors, the fuel jet is described by numerous discrete so-called packets in an attempt to model both, the global geometry of the penetrating spray and detailed local sub-processes such as fuel atomization and evaporation, fuel-air mixing, ignition, combustion and pollutant formation. Fuel atomization for a particular packet is assumed to occur instantaneously at the breakup time. Thereafter all droplets within the pocket are represented by a Sauter mean diameter (SMD), which characterizes a single droplet with the same volume to surface area ratio as the ratio of the respective quantities integrated over the whole droplet size distribution present in real spray [3]:

$$SMD = \frac{\sum_{i=1}^{N_{drops}} d_i^3}{\sum_{i=1}^{N_{drops}} d_i^2} \tag{3}$$

The initial value of the SMD after breakup is estimated by an empirical correlation fitted from experimental data. Frequently applied correlations:

$$SMD = 6156 \cdot 10^{-6} \cdot \nu_f^{0.385} \cdot \rho_f^{0.737} \cdot \rho_a^{0.06} \cdot \Delta p_{inj}^{-0.54} \tag{4}$$

where:

ν_f - liquid fuel viscosity [m^2/s],

ρ_a – air density [kg/m^3],

ρ_f – fuel density [kg/m^3],

Δp_{inj} – fuel injection to chamber pressure difference [kPa],

and by Varde [4] who related the SMD to the diameter of the injection nozzle:

$$\frac{SMD}{d_{noz}} = 8.7\left(Re_f\, We_f\right)^{-0.28} \tag{5}$$

The Reynolds and Weber numbers refer to injection velocity, the nozzle diameter and the liquid fuel properties. An important factor in assessing the appropriateness of any correlation is how is to be used to predict the magnitude of the delay. The physical characteristics of the diesel fuel does not significantly affect the auto ignition delay in fully or partially warmed-up engine. Fuel viscosity variations in fuel atomization, spray penetration, and vaporization rate over reasonable ranges do not appear to influence the duration of the delay period significantly.

2. Calculation method

The model of an engine cycle is based on the emptying and filling approach. The structure of the engine is subdivided into several interactive components associated with compression, combustion, expansion and emissions formation. Under a steady mode of engine operation, the inputs to the model includes: engine speed, fuel supply and ambient conditions.

2. 1. Heat release

The basic algorithm was derived from the 1st law of thermodynamics. Applying equation to the system the energy rates are expressed with respect to the crank angle - φ

$$\frac{\delta Q_{ch}}{d\varphi} - \frac{\delta Q_{ht}}{d\varphi} - \frac{\delta W}{d\varphi} + \frac{dm_f}{d\varphi} h_f = \frac{dU}{d\varphi} \tag{6}$$

where:

$\frac{\delta Q_{ch}}{d\varphi}$ - apparent rate of chemical energy,

$\frac{\delta Q_{ht}}{d\varphi}$ - rate of heat transfer out of the system,

$\frac{\delta W}{d\varphi}$ - rate of work transfer out of the system,

$\frac{dm_f}{d\varphi} h_f$ - rate of enthalpy inflow with the fuel,

$\frac{dU}{d\varphi}$ - rate of change of internal energy of the system.

2. 2. NO_x formation

NO_x emissions are predicted on the basis of the widely accepted extended Zeldovich mechanism. In prediction, all the species except NO are considered to be in chemical equilibrium. The rate of NO formation is

$$\frac{d[NO]}{dt} = \frac{2R_1(1-\beta^2)}{1 + \frac{\beta R_1}{R_2 + R_3}} \tag{7}$$

$$R_1 = k_1[O]_e[N_2]_e \tag{8}$$

$$R_2 = k_2[N]_e[O_2]_e \tag{9}$$

$$R_3 = k_3[N]_e[OH]_e \tag{10}$$

$$\beta = \frac{[NO]}{[NO]_e} \tag{11}$$

where:

[ ] square brackets denote mole concentrations,

k_1, k_2, k_3 - reaction rate constant ($cm^3 \cdot mol \cdot s$).

2. 3. Exhaust gas mass flow

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, 6]. Exhaust gas composition depends on the relative proportions of fuel and air fed to the engine, fuel composition, and completeness of combustion. The overall combustion reaction can be written as;

$$\begin{aligned} &HC_x + A(O_2 + 3.7274N_2 + 0.0444Ar + 0.0014CO_2) + BH_2O \Rightarrow aCO + bCO_2 + cO_2 + dH_2O \\ &+ eHC_x + fH_2 + gN_2 + hNO + jAr \end{aligned} \tag{12}$$

where:

$B = 4.7733\,A\,(P_s/P_a - P_s)$,

P_s - saturation vapour pressure of the inlet air [N/m^2],

P_a – ambient barometric pressure [N/m^2].

2. 4. Exhaust gas emission factors

As the NO_x emission depends on ambient air conditions, measured NO_x concentration was corrected for ambient air temperature and humidity. Measured dry concentration of CO and CO_2 were converted on the wet basis. Mass emission and required factor calculations were based upon Technical Code of Annex VI recommendation and carbon – oxygen balance method (C-O) used. Negligible levels of pollutant in the engine intake air were assumed and no allowances were made in this respect in the calculations.

3. Experimental details

General scope of exhaust performance measurement procedure of marine engines on test beds, beside the exhaust gas emission, particularly nitrogen oxides, are 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 analysers should comply with the specification given in the ISO-8178 standard. Measurements were carried out on a test-bed for ship propulsion engine - specification given in Table 1, operating at steady speed and load conditions, over a range of power settings [7].

Table 1. Test engine data

Engine			Nominal rate	
Designation	Maker	Type	Power [kW]	Speed [revs/min]
Ship propulsion	MAN B&W	8L32/40	3840	750

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 engine was set at four levels - scheduled in accordance to test programme D2. 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 (Figure 1). 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.

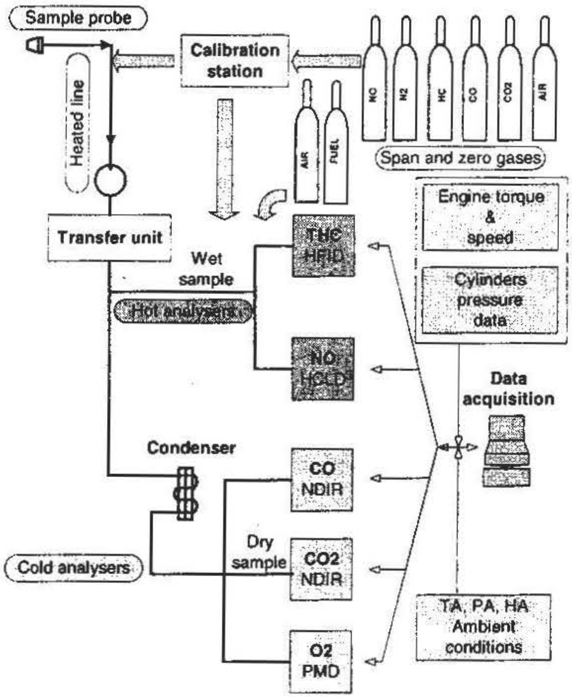

Figure 1. Experimental apparatus set-up

The test engine was running on distillate fuel ISO-F-DMA [8]. Samples of the fuel being burnt were taken at the time of the trial for analysis and analysed in accordance with standard industry procedure and an evaluation is given in Table 2.

Table 2. Fuel oil characteristic

Determination			Test results
1	Density @ 15 °C	kg/m^3	855.1
2	Viscosity @ 40 °C	mm^2/s	6.567
3	Calorific value	MJ/kg	42.60
4	C	%	85.2
5	H	%	13.8
6	N	%	0.004
7	S	%	0.85
8	O	%	0.1
9	Ash	%	0.001
10	Water	%	0.05

In addition to the exhaust emission, some essential operating engines data were measured to asses, the respective engine operating conditions – in accordance to ISO-3046 standard. Amongst other variables, this included: effective load, speed, fuel consumption, exhaust temperature, performance of the turbo blowers, together with the ambient conditions prevailing at the time of the measurement. Fuel oil viscosity was controlled by means of temperature loop of pressurized fuel oil supply circulating system. In order to ensure correct atomization, the fuel oil temperature must be adjusted according to the specific fuel oil viscosity used. Inadequate temperature will influence the combustion and may cause fuel pump damage. The recommended viscosity meter setting is 10÷12 cSt. However, for experiment purpose fuel low viscosity was set in accordance with manufacturer agreement to prevent potential risk of engine failure. Two engine test cycle D2, with different fuel oil viscosity settings – **a** and **b** performed (see figure 2).

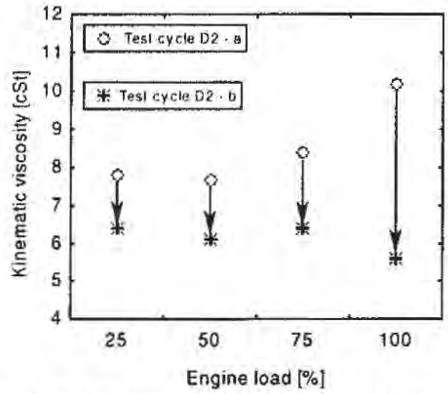

Figure 2. Fuel oil kinematics viscosity history during engine tests

4. Results and discussion

An investigation was conducted with the aim of identifying and quantifying the effects of fuel oil viscosity on exhaust emission – particularly nitrogen oxides. Exhaust mass emission and required factor calculations were based upon Technical Code of Annex VI recommendation and carbon – oxygen balance method (C-O) used. An examination of the measured data, particularly NO_x and CO concentrations - reveals, difference of level emissions associated with fuel oil temperature - viscosity. As shown in Figure 3 NO_x and CO emission differences are controlled by engine load. Detailed results of engine tests and calculation – emission factors are presented in table 3 and 4 (appendix).

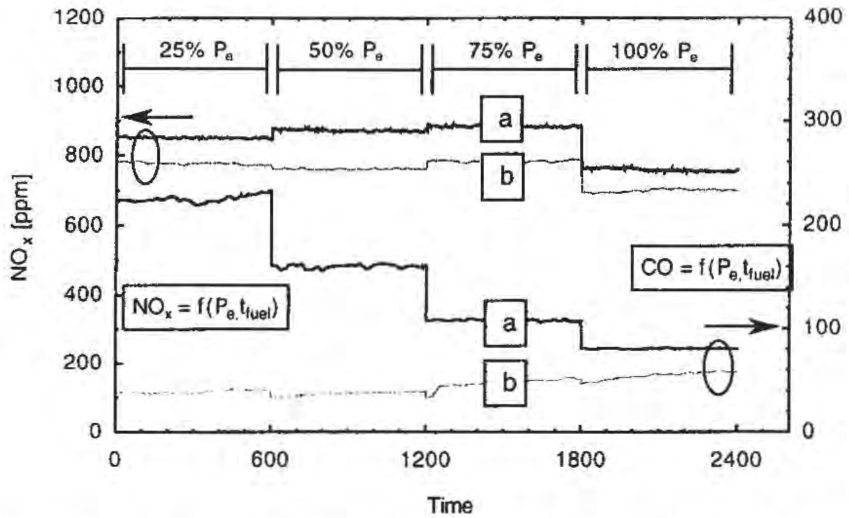

Figure 3.- Measured exhaust gas emission components profile of 8L32/40 engine during D2 - a and b test cycle

For a given engine load, the baseline cylinder pressure versus angle history can be established. Specifically, calculated combustion pressure vary for different fuel oil viscosity value. Because of the engine load influence, the combustion data and NO_x emission were evaluated. Rough comparison of calculated NO_x concentrations in test engines, with different fuel oil temperature shows substantial variations combustion rate under low load, and keeps steadily (see Figure 4). It also can be seen that NO_x and CO emission profile decreased with lower fuel oil viscosity achieved in second test cycle – b, under full range of engine load. Finally, NO_x weighted specific emission factor decreased up to 1.0 g/kWh that gives 9.5%.

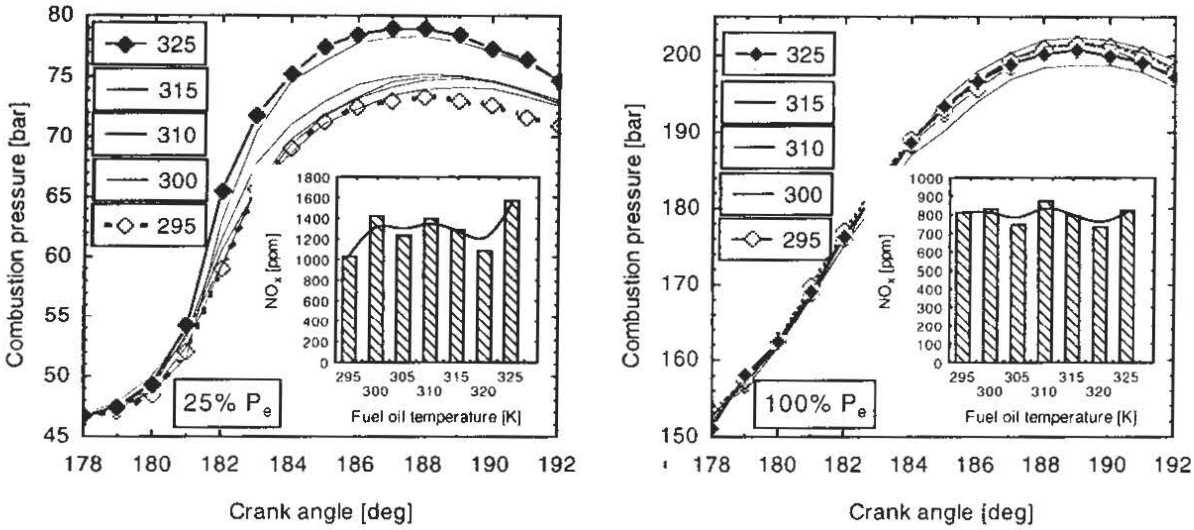

Figure 4. Calculated combustion pressure and NO_x emission profile of 8L32/40 engine fed by fuel oil with progressive arise of temperature (K) and for two loads 25% and 100%

Figure 4 shows comparison of calculated combustion pressure and NO_x emission caused by fuel oil temperature for lowest and highest test cycle load. The diagram exhibits a certain relation of NO formation rate to variation of pressure values. However, estimated NO emission do not reflect experimental results, in the same way. From this figure, it can be seen that calculated NO_x concentration fluctuation and value are not affected significantly by fuel oil property. It determines calculation sensitivity to in-cylinder temperature history.
The increase in fuel temperature affects combustion in a peak combustion pressure and temperature, in consequence which increases NO_x production and exhaust emissions.; the ignition delay can also be reduced, with consequent effects on several exhaust pollutant emissions.

Acknowledgements

The authors would like to express their gratitude to the Research & Development Center of HCP Cegielski, Poznań in Poland for the possibility of creating this paper. The presented work was carried out during the factory and acceptance tests of marine engines as part of the arranged program.

Appendix

*Table 3. Measured and calculated emission factors of engine MAN B&W 8L32/40 –cycle test **a***

Power effective	%	100	75	50	25
Mean effective pressure	bar	23.9	19.7	11.9	5.9
Specific fuel consumption	g/kWh	193	191	203	231
Scavenge air pressure	bar	3.082	2.084	1.132	0.480
Temp. after air cooler	°C	49.5	44.9	41.6	40.3
Fuel temp. before engine	°C	33.3	33.6	32.1	26.1
Barometric pressure	kPa	101.5	101.5	101.5	101.5
Intake air temperature	°C	28.0	27.5	27.0	26.5
Intake air humidity	%	41.5	42.0	44.0	46.5
Air flow - wet	kg/h	29020	21930	15200	8896
Exhaust flow - wet	kg/h	29770	22490	15590	9119
NO_x conc. wet	ppm	850	871	883	758
CO conc. dry	ppm	80	109	161	225
CO2 conc. dry	%	5.45	5.34	5.46	5.28
O2 conc. wet	%	12.76	12.84	12.70	12.89
HC conc. wet	ppm	90	101	126	116
NO_x specific	g/kWh	10.12	10.59	11.25	11.37
NO_x weighted	g/kWh	10.567			
CO weighted	g/kWh	0.817			
HC weighted	g/kWh	0.442			

*Table 4. Measured and calculated emission factors of engine MAN B&W 8L32/40 – cycle test **b***

Power effective	%	100	75	50	25
Mean effective pressure	bar	23.9	17.9	11.9	6.0
Specific fuel consumption	g/kWh	192	192	203	235
Scavenge air pressure	bar	3.156	2.231	1.251	0.473
Temp. after air cooler	°C	48.7	43.9	42.3	39.8
Fuel temp. before engine	°C	39.6	41.8	40.2	44.8
Barometric pressure	kPa	100.4	100.5	100.5	100.5
Intake air temperature	°C	23.8	23.8	23.5	22.8
Intake air humidity	%	60.7	60.8	60.9	62.0
Air flow - wet	kg/h	27860	22330	15320	8990
Exhaust flow - wet	kg/h	28600	22880	15710	9216
NO_x conc. wet	ppm	777	761	783	699
CO conc. dry	ppm	74	83	113	179
CO2 conc. dry	%	5.59	5.22	5.35	5.27
O2 conc. wet	%	13.35	13.85	13.66	13.76
HC conc. wet	ppm	39	37	47	54
NO_x specific	g/kWh	9.08	9.57	10.18	10.72
NO_x weighted	g/kWh	9.557			
CO weighted	g/kWh	0.653			
HC weighted	g/kWh	0.175			

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