пятна по вертикали которое зафиксировала камера, L - расстояние от гелиостата до концентратора,

$$\beta = arc\sin\left(\frac{\Delta s}{L}\right),$$

β - угол дефокусировки гелиостата по горизонтали, Δs-величина отклонения отраженного пятна по горизонтали, L-расстояние от гелиостата до концентратора.

С помощью данного метода производится оценка углов дефокусировки работы каждого гелиостата в течении всего дня, а именно в течении всего видимого движения солнца. Данный метод позволяет провести полную оценку работы гелиостата не зависимо от его способа управления –будь то программный, оптический или комбинированный.

ЛИТЕРАТУРА

[1] Гелиокомплекс «Солнце» (рус.) // журнал. Архитектура СССР — М., 1988. — Март (№ 2). — С. 37-43.

[2] Michalsky J. J. The Astronomical Almanac's Algorithm For Approximate Solar Position (1950-2050). Solar Energy. 1988. V 40

[3] A.A. Abdurachmanov, S.A.Orlov, S.A. Bahramov, A.V. Burbo, Sh.I. Klychev, Kh.K. Fazilov. On Sun Tracking Accuracy of Concentrators /APPLIED SOLAR ENERGY USA , 2010. Vol 46, №4б-P.316-318. [05.00.00;№4]

ENVIRONMENT CO & CO_2 EMISSIONS PROPOSED REDUCING MEASURES

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Annotation

The Diesel Engines (ICE) exhaust gas atmosphere noxious emissions reducing measures were introduced by the different editions and engine manufacturer publications already 25 years ago. Many of that have used up to present depend of its installation, usage and maintenance costs. For the mentioned above 25 years of emissions decreasing ways practical using on the vessels has identified it further usage consistency and profitability (efficiency). The atmosphere SO_X noxious emissions proposed decreasing way is directly connected with using fuel oil, i.e. at the fuel oil sulphur content decreasing the SO_X emission has decreasing too, that is task not for ship owners, but for petroleum-refining manufactures and bunkering companies. CO and CO_2 emissions decreasing is a corner task, as a fuel oil quality and lower calorific value are identified by the carbon & hydrogen content. Thus the fuel oil carbon and hydrogen content decreasing will bring to the decreasing of a quality and lower calorific value. Therefore all of this 25 years for the vessels diesel engines (ICE) exhaust gases CO & CO_2 emissions decreasing the energy efficiency task is stated. Our proposed way can allow to resolve the CO & CO_2 emissions decreasing task for the engines operation parts of loads and nominal loads.

Keywords: ICE (Diesel Engines) exhaust gas noxious emissions, carbon oxides, fuel oil Lower Calorific Value, emissions decreasing way, engine heat balance.

Introduction

The main reason of fuel oil incomplete combustion and exhaust gases toxicity increase, even at significant excess air ratio is bad mixture formation.

The fuel oil mixture failure is typical for the engine transient operating modes, specifically for ME running–in mode. Trial test data is showing that with engine load increasing a main constituent harmful substances concentration are listed above decreasing in exhaust gases. It is proved that with engine load increasing a carbon oxide concentration decreasing, afterwards it gets the stable condition before a certain limit value of mean effective pressure, but at overloading is slightly increases again. The nitrogen oxides concentration is continue to decreasing at mean effective pressure greater values.

Thereby, the exhaust gases minor toxicity is typical for full load mode. The engine operation experience shows that big amount of harmful substances escapes at engine starting, specially when it is not sufficiently warmed–up. But it is impossible go without starting, reverse and operation with low load. Thereby, environment contamination is inescapably during the operation with these modes, but it is possible to reduce the operation duration with these modes.

1. ATMOSPHERE SO_X EMISSIONS REDUCING MEASURES

Using the ULSMGO – Ultra Low Sulphure Marine Gasoil with sulphure content:

– < 0.5% for worldwide application.

– < 0,1% for application in SECA areas (Sulphure Special Emission Control Areas).

Using dual–fuel engines, therefore it is required:

– Purchasing or designing and production a modern dual–fuel engines.

– Development and designing the gas fuel storage, transfer and supply to Diesel Engines systems.

– Development and designing the gas fuel storage, transfer and bunkering coast and float facilities.

2. ATMOSPHERE CO & CO_2 EMISSIONS REDUCING MEASURES

Using the engines with the highest efficiency.

– As far as possible with increased fuel injection timing.

– Using the engines with loads are closed to NCR = 85%MCR.

– The engine turbocharging modification for scavenging air excess supply at the engine operation under parts of load (forcing by scavenging air).

– Using the manufacturer original spare parts, influencing to the engine cylinders combustion process.

– To monitor on regular bases for the engine adjustment, which to be comply to manufacturer adjustment.

3. USING THE ENGINES WITH THE HIGHEST EFFICIENCY.

The given way can be proposed as idea, which can be proved only by the Diesel Engine preliminary heat calculation and its engine TC heat balance calculation, as well as touches one of listed above items such as – The engine turbocharging modification for scavenging air excess supply at the engine operation under parts of load (forcing by scavenging air).

1) Heightening the Diesel Engines efficiency by variation the values are influencing to the engine power:

$$N_{IND} = k \cdot P_{IND} \cdot n \cdot i \text{ (IP)}$$

where: $k = 1{,}745 \cdot D^2 \cdot S \cdot m$ – cylinder constant (–);

D – cylinder diameter (mtr);

S – piston stroke (mtr);

m – engine stroke factor (4–stroke m = 2, 2–stroke m = 1);

P_{IND} – mean – indicated pressure (kg/cm^2);

n – engine speed (rpm);

i – number of cylinders (–).

$$=> N_{IND} = k \cdot P_{IND} \cdot n \cdot i = 1{,}745 \cdot D^2 \cdot S \cdot P_{IND} \cdot m \cdot n \cdot i \text{ (IP)}$$

a) Heightening the power by the cylinder diameter increasing – D. The way have used around 50 years, that is bring to the largest diameter is 90cm for the engines MAN–B&W & SULZER and as a result to the engine weight increasing. Further cylinder diameter increasing has been not profitable.

b) Heightening the power by the piston stroke increasing – S. The way have used around 40 years, that is bring into generation the long stroke and super long stroke engines models such as LMC & SMC type of the MAN–B&W & SULZER manufacturer, and to the engine weight increasing too. Further piston stroke increasing has been not profitable.

c) Heightening the power by the engine speed increasing – n. The way is not logical for SSE & MSE (Slow speed engines & Medium speed engines).

d) Heightening the power by the cylinders number increasing – i. The way have used till the particular time, and bring to the engine weight increasing too. Further cylinders number increasing has been not profitable.

e) All above listed ways are possible to relate to energy efficiency increasing, as well as to increasing the engine indicated power, because of at constant mean–indicated pressure (a fuel oil constant consuption) it has increased an indicated power. Heightening the power by the mean–indicated pressure P_{IND} increasing can not relate to the energy efficiency increasing due to reason as follow. The mean–indicated pressure P_{IND} increasing can be achieved by the indicator diagram area increasing via building–up a maximum combustion pressure or via injection length and cylinder's fuel oil combustion duration prolongation (via fuel oil cycle dosage and consumption raising). And that and other ways are not unlimited: by the maximum combustion pressure – due to cylinder head and cylinder liner strength limitations, by the fuel oil injection length – due to exhaust gas temperatures increasing, i.e. due to exhaust gases heat loss, if not changing the valve timing and therefore the engine efficiency can remains as invarianted.

f) Will approach to the engine energy efficiency and efficiency factor increasing from another side – will try to reduce the fuel oil injection length and cylinder's fuel oil combustion process duration (to reduce the fuel oil cycle dosage and consumption) at constant mean–indicated pressure. Have achieved some positive results in this question solution, we will reached at the same time a reducing the emissions CO_2, CO и NO_X to the atmosphere due to fuel oil consumption reducing for the same power achievement. This way already 20 years ago has got its development via engine forcing by scavenging air pressure, have builded–up it from 1.8 bar to 2.9÷3 bar. It is clear, as much air as possible take part in the fuel oil combustion, as more perfect the fuel oil combustion, then less the exhaust gases heat losses, then more a heat is go for effective power, more the combustion velocity, and therefore less the combustion duration (less exhaust gas temperature). Continue our proposal about scavenging air charge ratio build–up and the results follows from it in example of preliminary theoretical conclusions without Diesel Engines heat calculation and presented engine TC heat balance calculation.

2) Idea of scavenging air ratio increasing.

To examine the scavenging air ratio increasing idea in example of engine HYUNDAI MAN–B&W 6S50MC (MCR 11640 BHP & MS 127 RPM). The presented ME indicator diagram and indication main variables summary table are taken during the operation have introduced on the figure 1.

a) Engine speed: 116,3 rpm = 91,58% MS (maximum speed);

b) Engine indicated power: 10103 IP = 7431 IKW = 86,8% MCR;

c) Cylinders compression pressures:

P_{COM}^{1} = 105,42 bar; P_{COM}^{2} = 104,39 bar; P_{COM}^{3} = 102,65 bar;

P_{COM}^{4} = 103,29 bar; P_{COM}^{5} = 102,94 bar; P_{COM}^{6} = 103 bar; P_{COM}^{AV} = 103,62 bar;

d) Cylinders maximum combustion pressures:

P_{MAX}^{1} = 124,27 bar; P_{MAX}^{2} = 121,91 bar; P_{MAX}^{3} = 120,21 bar;

P_{MAX}^{4} = 120,81 bar; P_{MAX}^{5} = 122,99 bar; P_{MAX}^{6} = 118,2 bar; P_{MAX}^{AV} = 121,4 bar;

e) Scavenging air pressure: P_{SC} = 2,01 bar;

f) Fuel ignition timing: $\varphi_{INJ} = 2^{O}$ after TDC;

g) Shall visualize the engine forcing by a charge air and then variables changing on the given operating mode: therefore a cylinders compression pressures average value has reached a maximum combustion pressures average value $P_{COM}^{REC} = P_{MAX}^{AV}$ = 121,4 bar (figure 2(b)):

– a required scavenging air pressure for estimated compression pressure achievement P_{COM}^{REC} = 121,4 bar:

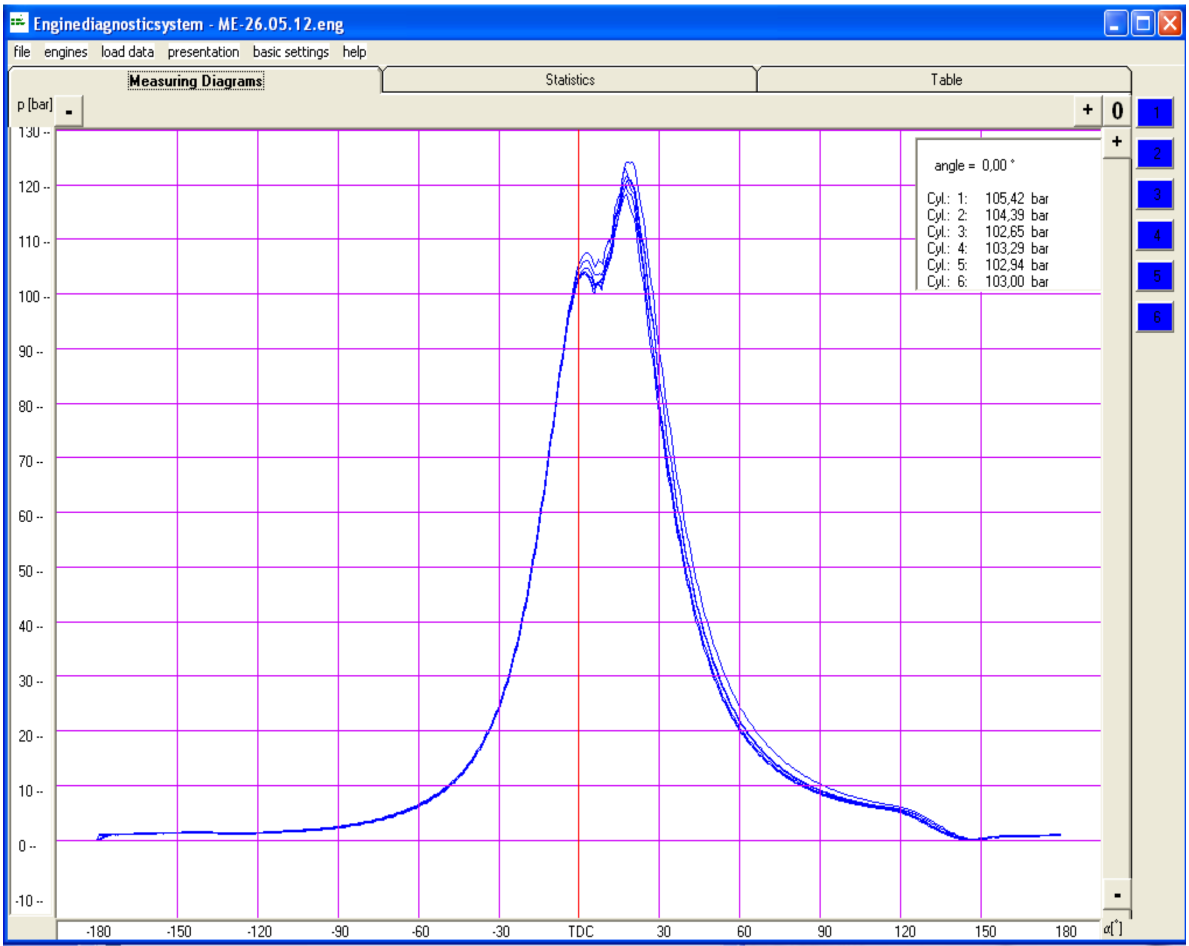
$$\varepsilon^{n_1} = \frac{P_{\mathrm{COM}}^{\mathrm{AV}} + \mathrm{P_{AMB}}}{P_{\mathrm{SC}} + \mathrm{P_{AMB}}} = \frac{103{,}62 + 1{,}017}{2{,}01 + 1{,}017} = 34{,}567889\ (-) - \text{ absolute pressures ratio}$$

$$P_{\mathrm{SC}}^{\mathrm{REC}} = \frac{P_{\mathrm{COM}}^{\mathrm{REC}} + \mathrm{P_{AMB}}}{\varepsilon^{n_1}} - \mathrm{P_{AMB}} = \frac{121{,}4 + 1{,}017}{34{,}567889} - 1{,}017 = 2{,}52\ (\text{бар}) -$$

recommended scavenging air pressure for the ME forcing

i.e. for compression pressure achievement from existing P_{COM}^{AV} = 103,62 bar up to recommended P_{COM}^{REC} = 121,4 bar, it is necessary to raise the scavenging air pressure at presented mode from P_{SC} = 2,01 bar up to P_{SC}^{REC} = 2,5 bar.

h) Will change the fuel oil injection timing, in order that ignition timing was not 2^{O} after TDC, but significantly late on the expansion line for achievement the maximum combustion pressure with the same value as compression pressure $P_{MAX}^{REC} = P_{COM}^{REC}$ = 121,4 bar (figure 2(b));



Enginediagnosticsystem - ME-26.05.12.eng

file engines load data basic settings help

Measuring Diagrams | Statistics | Table

Engine type:	6S50MC	**Stroke:**	2
Cylinder Count:	6	**Swept Volume [cmi]:**	375021
Lenght of Con.rod [mm]:	2170	**Half stroke [mm]:**	955
bore [mm]:	500	**Compression room [cmi]:**	21447
T before cooler 1 [°C]:	177	**T after Cooler 1 [°C]:**	41,8
P after cooler 1 [bar]:	2,01	**Blower revs [rpm]:**	11000
T after blower 1 [°C]	258,6		

Cylinder	n [rpm]	pmax [bar]	pi [bar]	Power [kW]	Rack Set [mm]	T.exhaust [°C]
1. 27.02.12 13:54	116,8	124,27	19,34	1 411,45	64,0	339,5
2. 27.02.12 13:58	116,9	121,91	17,15	1 253,04	65,0	330,8
3. 27.02.12 14:00	116,1	120,21	16,30	1 182,65	65,0	348,1
4. 27.02.12 14:03	116,4	120,82	17,52	1 274,76	65,5	339,5
5. 27.02.12 14:07	115,6	122,99	16,04	1 159,06	63,0	325,8
6. 27.02.12 14:10	116,3	118,20	15,82	1 149,82	63,0	337,6
	O 116,3	O 121,40	O 17,03	Σ 7□430,79	O 64,25	O 336,88

Figure 1 – actual indicator diagram and indication data

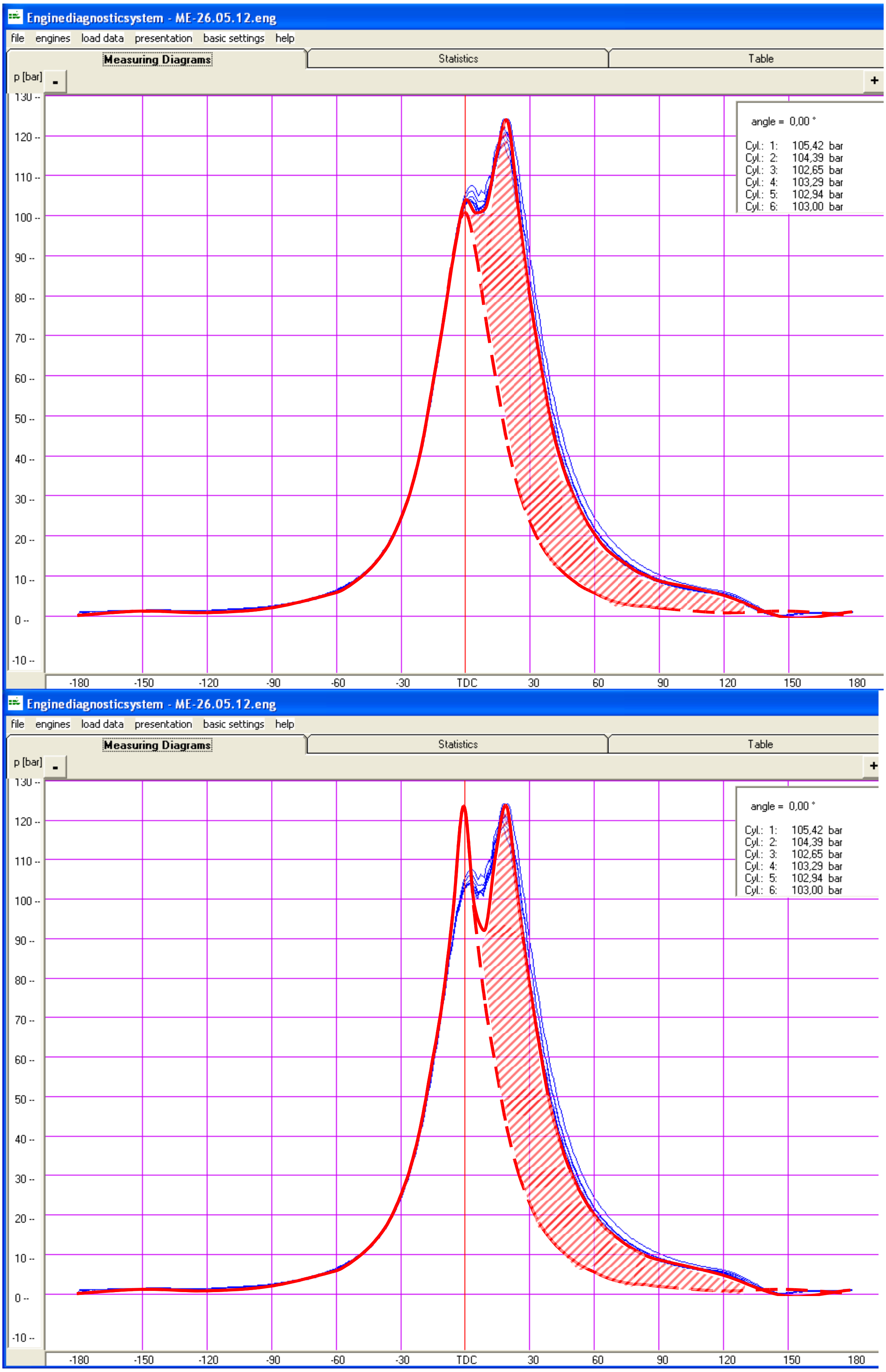


Figure 2(a) – actual indicator diagram; (b) – estimated indicator diagram

i) At the engine forcing by a scavenging air in that aspect that we proposes, it is possible to expect the effects as follows:

– indicated diagram area is specified the mean–indicated pressure depends on combustion gases quantity is consist of supplied fuel oil quantity and scavenging air quantity is involving in fuel oil mixture formation and mixture combustion per cycle: $G_{CG} = G_{FO} + G_{SCA}$;

– we can assume, that for the engine is operating by the external propeller line (at locked Fuel Rack), at increasing the involving in fuel oil mixture formation and mixture combustion scavenging air quantity and constant combustion gases quantity (constant indicator diagram area and mean–indicated pressure), a fuel oil consumption will reduced;

– from the above saying we will beg to make conclusion, that at the scavenging air quantity rise and fuel oil quantity reduction are involving in mixture formation and mixture combustion and at constant combustion gases quantity (constant indicator diagram area and mean–indicated pressure), a combustion efficiency increases, exhaust gas temperature comes down, and that and other has bring to reduction of CO_2, CO & NO_X emissions to atmosphere. A prove of the above saying is indicated power equation at the engine constant load condition:

$$N_{IND} = LCV_{FO} \cdot G_{FO} - Q_{EXH} - Q_{CW} - Q_{LO} = \text{const}$$

where: LCV_{FO} – lower calorific value;

G_{FO} – fuel oil consumption (flow);

Q_{EXH} – exhaust gases heat (energy) losses;

Q_{CW} – cooling water heat (energy) losses;

Q_{LO} – lubricating oil heat (energy) losses.

Conclusion**:** At the exhaust gas temperature reduction, and thereafter an exhaust gases energy (heat) losses too Q_{EXH}, for keeping the condition N_{IND} = const, to reduce the fuel oil consumption G_{FO} it is required.

j) At the engine forcing by a scavenging air, in that aspect that we proposes, it is possible to expect, that the engine cylinder's air admission factor before closing the scavenging air ports will rised. In that case also can propose the latest opening of exhaust valve, ipso facto have increased the piston stroke efficiency, and the earliest closing of exhaust valve, ipso facto have increased compression ratio, have constructively changed exhaust valve driving cam profile.

3) Initial actions for stated idea approval:

a) Diesel Engines preliminary theoretical heat calculation and presented engine TC heat balance calculation;

b) Without any additional expenses to test the engine operation with already known manufacturer shop trial test results (to prove the stated idea) during its forcing by scavenging air on the repetitive test bed, have created for selected load the proposed scavenging air constant pressure in scavenging air receiver by any external source, for example from starting air bottles via reducing valve;

c) After expected positive result to calculate an estimated scavenging air constant pressures, has created by the same external source in the scavenging air receiver and estimated VIT racks for parts of load sequence and to carry out the trial tests for selected sequence;

d) In all likelihood VIT system to be operated by inverse proportionality dependence of the load, i.e. VIT index decreasing at the load increasing, in contrast to classical dependence – VIT index increasing at the load increasing from 0 up to 75%, and its further decreasing at the loads more then 75%.

e) To test the engine operation with already known manufacturer shop trial test results (to prove the stated idea) during its forcing by scavenging air on the repetitive test bed, have created by any external source (for example from starting air bottles via reducing valve) the proposed scavenging air constant pressure in scavenging air receiver equal to scavenging air pressure at MCR (100% of load) and keep it pressure at all parts of loads. In that case at any part of load scavenging air pressure, thereafter cylinders compression pressures and maximum combustion pressures will be constant, but the engine load will be changed by changing the fuel oil injection end, thereafter by changing the fuel injection length (due to constant fuel injection timing), by changing the fuel oil cycle dosage and consumption. Assumed that the VIT system will be not required for this particular case. How to operate the engine at this particular expected measure:

– to develop the highest capacity TC for achievement the proposed scavenging air constant pressure in scavenging air receiver equal to scavenging air pressure at MCR (100% of load), i.e. to 2.75 bar for this particular engine (for our presented engine 6S50MC) and to install it on engine;

– to fabricate the engine TC air inlet filter easy moved flap and keep it closed for all parts of load till NCR (85% of MCR);

– to change the fuel oil injection timing from 12.5^{O} before TDC to 12.5^{O} after TDC (for our presented engine 6S50MC);

– to set the VIT system rack to «0» in constant bases;

– to create by any external source (for example from starting air bottles via reducing valve) the proposed scavenging air constant pressure in scavenging air receiver equal to scavenging air pressure at MCR (100% of load), i.e. to 2.75 bar (for our presented engine 6S50MC);

– to start, reverse, maneuver and run–up the engine till the NCR (85% of MCR) with closed TC air inlet filter flap (for avoid the TC heavy surging) and created scavenging air constant pressure is 2.75 bar in scavenging air manifold;

– at the engine reaching a NCR (85% of MCR) to reduce the created scavenging air pressure in scavenging air manifold down to value is less then pressure at NCR (from manufacturer shop trial test results) by reducing valve (for avoid the TC heavy surging) and to open the TC air inlet filter flap;

– at the last to close the reducing valve totally.

Conclusions:

Have submitted to your attention CO and CO_2 emissions reducing measure is required theoretically calculated and experimental confirmations. Last can be carry out at availability of Diesel Engine laboratory –

mini ER or by association with Diesel Engines manufacturer.

References

1. V.I. Korolev, A.G. Taranin, Training of engineers on watch with usage of the engine room simulator «DIESELSIM DPS–100». Parts 1 & 2, Novorossiysk, Admiral F.F. Ushakov State Maritime University, 2010.
2. V.I. Korolev, A.G. Taranin, Unattended machine service of a ship's power plant with simulator «DIESELSIM DPS–100». Parts 1 & 2, Novorossiysk, Admiral F.F. Ushakov State Maritime University, 2010.
3. A.G. Taranin, The ship's equipment operational instructions elements with usage of the ER simulator «DIESELSIM DPS–100», Novorossiysk, Admiral F.F. Ushakov State Maritime University, 2020.
4. A.G. Taranin, The ship's equipment operational instructions elements with usage of the ER simulator «NEPTUNE MC90–IV», Novorossiysk, Admiral F.F. Ushakov State Maritime University, 2020.

USAGE FEATURES OF THE ELECTRONIC INDICATORS FOR SHIP'S AND SHORE POWER SUPPLY FOUR–STROKE INTERNAL COMBUSTION ENGINES (DIESEL ENGINES)

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Annotation

The present publication illuminate the tasks as follows: Electronic indicator proper usage at four–stroke internal combustion engines (diesel engines) indication; Indication results & diagram proper transfer to PC; indicator diagram top dead center TDC correction and engine performance data output values such as P_{MI}–mean indicated pressure, P_{ME}–mean effective pressure, N_{IND}–indicated power and N_{EFF}–effective power proper calculations for each cylinder and engine total.

Keywords: Engine indication, performance data, electronic indicator, mean–indicated & mean–effective pressure, indicated & effective power.

Introduction

Currently on the worldwide fleet motor–vessels and shore diesel power plants for internal combustion engines–diesel engines indication and performance data measurement readings carrying–out the micro–processing gauging and systems, such as Doctor–Engine, Diesel–Doctor and Electronic indicators (different kind of brands and manufacturers) are used in most of cases. However, actually they are not carrying–out the functions of the engines technical condition (cylinder tightness, fuel injection equipment condition and turbocharger system condition) diagnostic and analysis, overload/download analysis and load distribution between the cylinders analysis, but they are electronic gauges for compression pressures P_{COM}, maximum combustion pressures P_{MAX} measurement by open indicator diagrams (Fig.1) and closed indicator diagrams (Fig.2) for each cylinder and for engine speed measurement at each cylinder indication. All others values are required for the engine technical condition diagnostic and analysis has determined by calculation from indicator diagrams or entered manually to the electronic equipment tables.

Examine the engine indication results from Electronic indicator type HLV–2005 MK (Praezisionsmesstechnik Beawert GMBH, Germany):

1) The values are calculated from the indicator diagrams:

– Cylinders indicator diagrams area A_D (mm^2);
– Cylinders mean–indicated pressure P_{MI}^{CYL} (bar) (Fif.3);
– Cylinders mean–effective pressure P_{ME}^{CYL} (bar);
– Cylinders indicated power N_{IND}^{CYL} (IKW) (Fif.3);
– Cylinders effective power N_{EFF}^{CYL} (EKW);
– Engine average mean–indicated pressure P_{MI}^{ENG} (bar) (Fig.3);
– Engine average mean–effective pressure P_{ME}^{ENG} (bar);
– Engine indicated power N_{IND}^{ENG} (IKW) (Fif.3);
– Engine effective power N_{EFF}^{ENG} (EKW);
– Engine mechanical efficiency η_{MEC} (%).

2) The values are entered manually to the electronic equipment tables (Fig.3):

– Scavenging air temperature after turbocharger or before scavenging air cooler T_{SC}^{BC} (^{O}C);
– Scavenging air temperature after scavenging air cooler T_{SC}^{AC} (^{O}C);
– Scavenging air pressure after scavenging air cooler P_{SC}^{AC} (bar);