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ОСОБЕННОСТИ ИСПОЛЬЗОВАНИЯ ЭЛЕКТРОННОГО ИНДИКАТОРА ДЛЯ СУДОВЫХ ДВУХТАКТНЫХ ДВИГАТЕЛЕЙ ВНУТРЕННЕГО СГОРАНИЯ И ДВС БЕРЕГОВЫХ ЭЛЕКТРОСТАНЦИЙ

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Данная статья освещает следующие задачи: Правильное использование электронного индикатора во время индицирования двухтактных двигателей внутреннего сгорания; правильный перенос результатов индицирования и диаграмм на ПК; корректировка ВМТ индикаторной диаграммы и правильный расчёт выходных параметров индицирования, таких как P_{MI}–средне–индикаторное давление, P_{ME}–средне–эффективное давление, N_{IND}–индикаторная мощность и N_{EFF}–эффективная мощность для каждого цилиндра и двигателя в целом.
Ключевые слова: Индицирование двигателя, параметры индицирования, электронный индикатор, средне–индикаторное и средне–эффективное давление, индикаторная и эффективная мощность.
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 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.2);
- Cylinders mean–effective pressure P_{ME}^{CYL} (bar);
- Cylinders indicated power N_{IND}^{CYL} (IKW) (Fif.2);
- Cylinders effective power N_{EFF}^{CYL} (EKW);
- Engine average mean–indicated pressure P_{MI}^{ENG} (bar) (Fig.2);
- Engine average mean–effective pressure P_{ME}^{ENG} (bar);
- Engine indicated power N_{IND}^{ENG} (IKW) (Fif.2);
- Engine effective power N_{EFF}^{ENG} (EKW);
- Engine mechanical efficiency η_{MEC} (%).

2) The values are entered manually to the electronic equipment tables (Fig.2):
- 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);
- Exhaust gas temperature after turbocharger T_{EXH}^{ATC} (^{O}C);

– Turbocharger speed n_{TC} (rpm);
– Cylinders exhaust gas temperatures T_{EXH}^{CYL} (°C);
– Cylinders fuel rack position FRP (fuel pump index FPI) (mm);

Note: However, the mentioned above values are not enough for the engine technical condition full diagnostic and analysis (cylinder tightness, fuel injection equipment condition and turbocharger system condition).

In completion of indication data entering to the PC without TDC correction the engine average mean–indicated pressure & indicated power calculation can give tolerance up to ±10%, while the same values calculation from indicator diagrams are taken by mechanical indicator with usage of computerized technology gives toleranceup to ±0.5% only.

The engine average mean–indicated pressure and indicated power calculation tolerance up to ±10% is not satisfactory for the engine 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.

Thereby we suggest the engine (2–stroke engine) indicated power accurate calculation procedure, afterwards it is possible a TDC accurate correction for each cylinder, and then a cylinders mean–indicated pressure P_{MI}^{CYL}, cylinders indicated power N_{IND}^{CYL}& engine average mean–indicated pressure P_{MI}^{ENG} same accurate calculation within tolerance ±0.5%.

Work object

Thehighaccuracyobtaining inthe indicatordiagram treatmentand as resultshighaccuracyin the cylinder power calculation, determination of load distribution betweencylindersand cylinders/engine conditiondiagnostic & analysiswithout engine dismantling.

Ways of investigation

Investigations has carried out on the vessel's (with effective power from **736 EKW** up to **11900EKW**) with different kind of micro–processing gauging and systems (Doctor–Engine, Diesel–Doctor and Electronic indicator) & with mechanical indicators.

Investigation results and discussion about

1. The indicator diagrams TDC correction and each cylinder/total engine output data calculation after the 2–stroke Diesel Propulsion Engine MAN–B&W type 6S50MC–Mk indication by Electronic indicator type HLV–2005 MK.

1) The Diesel Propulsion Engine performance data some measurement readings are taken during the indication (table 1):

Table 1

Engine indication start	T_{HS}	hrs	by observation	13
Engine indication start	T_{MS}	min	by observation	45
Engine indication stop	T_{HE}	hrs	by observation	14
Engine indication stop	T_{ME}	min	by observation	42
Engine indication period	T_{IND}	min	$T_{IND} = (T_{HE} - T_{HS}) \cdot 60 + T_{ME} - T_{MS}$	57
Eng.revolution counter at start	R_{CS}	revoluton	by observation	20344122
Eng.revolution counter at stop	R_{CE}	revoluton	by observation	20344788
Engine speed	n_{ENG}	rpm	$n_{ENG} = (R_{CE} - R_{CS}) \cdot 10 / T_{IND}$	116,80
Engine FO flowmeter at start	Q_{FOS}	ltrs	by observation	1711963
Engine FO flowmeter at stop	Q_{FOE}	ltrs	by observation	1713290
Engine FO consumption	Q_{FO}	ltrs / hr	$Q_{FO} = (Q_{FOE} - Q_{FOS}) \cdot 60 / T_{IND}$	1396,762
FO temperature inlet flowmete	T_{FO}	°C	by observation	130,3
FO specific gravity @ 15 °C	ρ_{FO}^{15}	kg / ltr	from FO bunker specification sertificate	0,9672
FO expansion factor	k_{FO}	kg/ltr.°C	$k_{FO} = 0,00183224 - 0,00131724 \cdot \rho_{FO}^{15}$	0,00056
FO specific gravity at flowmet	ρ_{FO}^{T}	kg / ltr	$\rho_{FO}^{T} = \rho_{FO}^{15} - k_{FO} \cdot (T_{FO} - 15)$	0,9028
FO sulfur content	S	%	from FO bunker specification sertificate	1,86
FO lower calorific value LCV	LCV	kcal / kg	$LCV = 12900 - 7095 \cdot S / 100 - 3162 \cdot \rho_{FO}^{15}$	9710
Engine FO consumption	G_{FO}	kg / hr	$G_{FO} = Q_{FO} \cdot \rho_{FO}^{T}$	1261,051
Engine average fuel rack posit.	FRP	mm	by observation	64,3
Turbocharger speed	N_{TC}	rpm	by observation	11000
Scavenging air pressure	P_{SC}	kg / cm^2	by observation	2,08
Air temperature air filter inlet	T_{INL}	°C	by observation	38,4
Scav.air temp.air cooler inlet	T_{SC}^{BC}	°C	by observation	177
Scav.air temp.air cooler outlet	T_{SC}^{AC}	°C	by observation	41,8
Scav.air temp.in scav.air manif	T_{SC}	°C	by observation	42,5
Exhaust gas temp.turbine inlet	T_{EXH}^{BTC}	°C	by observation	393
Exhaust gas temp.turbine outle	T_{EXH}^{ATC}	°C	by observation	263
FW temp.scav.air cooler inlet	T_{FW}^{BC}	°C	by observation	30,5
FW temp.scav.air cooler outlet	T_{FW}^{BC}	°C	by observation	44
Air cooler termoefficiency	η_T	°C	$\eta_T = (T_{SC}^{BC} - T_{SC}^{AC}) \cdot 100 / (T_{SC}^{BC} - T_{FW}^{BC})$	92,29
Atmospheric pressure	P_{ATM}	kg / cm^2	by observation	1,037

2) The Diesel Propulsion Engine ambient (reference) conditions and FO data from shop trial test results (table 2):

Table 2

Engine Room temperature	T_{ER}	^{O}C	from shop trial test results	23,9
Atmospheric pressure	P_{ATM}^{ST}	kg / cm^2	from shop trial test results	1,035
SW temp.scav.air cooler inlet	T_{SW}^{BC}	^{O}C	from shop trial test results	18,1
FO temperature inlet flowmete	T_{FO}^{ST}	^{O}C	from shop trial test results	34,3
FO specific gravity @ 15 ^{O}C	ρ_{ST}^{15}	kg / ltr	from shop trial test results	0,9136
FO expansion factor	k_{FO}^{ST}	kg/ltr.^{O}C	$k_{FO}^{ST} = 0{,}00183224 - 0{,}00131724 \cdot \rho_{ST}^{15}$	0,000629
FO specific gravity at flowmete	ρ_{ST}^{T}	kg / ltr	$\rho_{ST}^{T} = \rho_{ST}^{15} - k_{FO}^{ST} \cdot (T_{FO}^{ST} - 15)$	0,9015
FO sulfur content	S_{ST}	%	from shop trial test results	0,26
FO lower calorific value LCV	LCV_{ST}	kcal / kg	$LCV_{ST} = 12900 - 7095 \cdot S_{ST} / 100 - 3162 \cdot \rho_{ST}^{15}$	9993

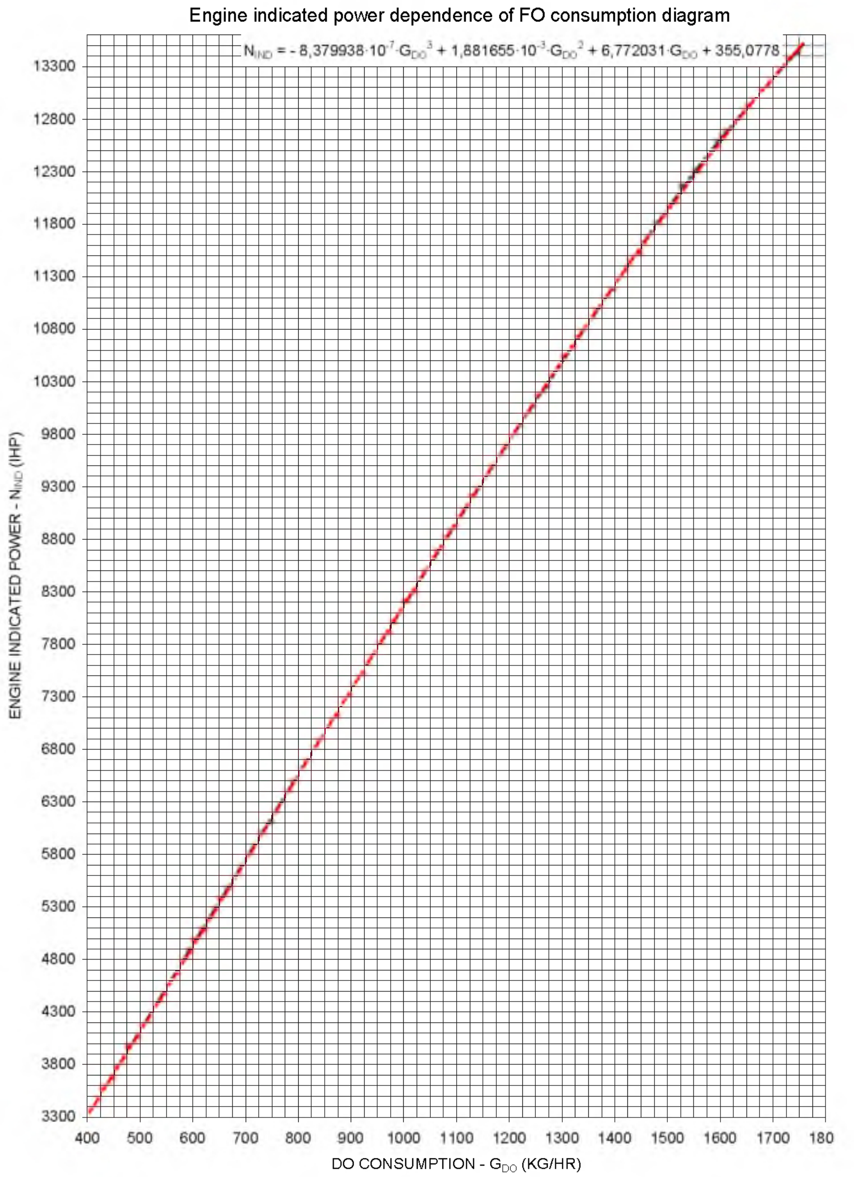
3) The Diesel Propulsion Engine FO consumption G_{FO} correction to the shop trial test reference conditions:

$$G_{DO} = \frac{G_{FO} \cdot LCV.}{LCV_{ST}} = \frac{1261.051 \cdot 9710}{9993} = 1225.3 \text{ kg / hr}$$

4) Draw the diagram of the engine indicated power dependence of FO consumption from shop trial test results table and found its dependence function by the trend line (Diagram 1):

5) The engine calculated indicated power by the function is founded from the diagram 1:

Diagram 1



$$N_{IND}^{1} = -8.379938\cdot10^{-7}\cdot G_{DO}^{3} + 1.881655\cdot10^{-3}\cdot G_{DO}^{2} + 6.772031\cdot G_{DO} + 355.0778 =$$
$$= -8.379938\cdot10^{-7}\cdot1225.3^{3} + 1.881655\cdot10^{-3}\cdot1225.3^{2} + 6.772031\cdot1225.3 +$$
$$+ 355.0778 = 9937\ IHP$$

6) The Diesel Propulsion Engine turbocharger speed N_{TC} correction to the shop trial test reference conditions:

$$N_{TC}^{ST} = N_{TC}\cdot\sqrt{\frac{(273+T_{INL})}{(273+T_{ER})}} = 11000\cdot\sqrt{\frac{(273+38.4)}{(273+23.9)}} = 11266\ rpm$$

7) Draw the diagram of the engine indicated power dependence of turbocharger speed from shop trial test results table and found its dependence function by the trend line (in the same way as Diagram 1):

8) The engine calculated indicated power by the function is founded from the diagram by item 7):

$$N_{IND}^{2} = -1.41411647\cdot10^{-12}\cdot {N^{ST}}_{TC}^{4} + 5.25309184\cdot10^{-8}\cdot {N^{ST}}_{TC}^{3} - 6.2157409\cdot10^{-4}\cdot {N^{ST}}_{TC}^{2} +$$
$$+ 3.79006967\cdot N^{ST}_{TC} - 5945.706 =$$
$$= -1.41411647\cdot10^{-12}\cdot11266^{4} + 5.25309184\cdot10^{-8}\cdot11266^{3} - 6.2157409\cdot10^{-4}\cdot11266^{2} +$$
$$+ 3.79006967\cdot11266 - 5945.706 = 10195\ IHP$$

9) The Diesel Propulsion Engine multiply FRP $\cdot$ n_{ENG} correction to the shop trial test reference conditions:

$$FRP_{ST}\cdot n_{ENG} = \frac{FRP\cdot n_{ENG}\cdot LCV\cdot \rho_{FO}^{T}}{LCV_{ST}\cdot \rho_{ST}^{T}} = \frac{64.3\cdot116.8\cdot9710\cdot0.9028}{9993\cdot0.9015} = 7303\ mm\cdot rpm$$

10) Draw the diagram of the engine indicated power dependence of multiply $FRP_{ST}\cdot n_{ENG}$ from shop trial test results table and found its dependence function by the trend line (in the same way as Diagram 1):

11) The engine calculated indicated power by the function is founded from the diagram by item 10):

$$N_{IND}^{3} = 2.48249632\cdot10^{-12}\cdot(FRP_{ST}\cdot n_{ENG})^{4} - 6.76738036\cdot10^{-8}\cdot(FRP_{ST}\cdot n_{ENG})^{3} +$$
$$+6.18921346\cdot10^{-4}\cdot(FRP_{ST}\cdot n_{ENG})^{2} - 0.769905624\cdot(FRP_{ST}\cdot n_{ENG}) + 2042.11999 =$$
$$= 2.48249632\cdot10^{-12}\cdot7303^{4} - 6.76738036\cdot10^{-8}\cdot7303^{3} + 6.18921346\cdot10^{-4}\cdot7303^{2} -$$
$$- 0.769905624\cdot7303 + 2042.11999 = 10132\ IHP$$

12) The Diesel Propulsion Engine scavenging air pressure correction to the shop trial test reference conditions:

$$P_{SC}^{ST} = P_{SC} + 0.002856\cdot(T_{INL} - T_{ER})\cdot(P_{ATM} + P_{SC}) - 0.00222\cdot(T_{FW}^{BC} - T_{SW}^{BC})\cdot(P_{ATM} + P_{SC}) =$$
$$= 2.08 + 0.002856\cdot(38.4 - 23.9)\cdot(1.037 + 2.08) - 0.00222\cdot(30.5 - 18.1)\cdot(1.037 + 2.08) =$$
$$= 2.123\ kg/cm^{2}$$

13) Draw the diagram of the engine indicated power dependence of scavenging air pressure from shop trial test results table and found its dependence function by the trend line (in the same way as Diagram 1):

14) The engine calculated indicated power by the function is founded from the diagram by item 13):

$$N_{IND}^{4} = 44.4567458\cdot {P^{ST}}_{SC}^{3} - 527.060152\cdot {P^{ST}}_{SC}^{2} + 5032.75628\cdot P^{ST}_{SC} + 1441.75234 =$$
$$= 44.4567458\cdot2.123^{3} - 527.060152\cdot2.123^{2} + 5032.75628\cdot2.123 + 1441.75234 =$$
$$= 10177\ IHP$$

15) The engine average indicated power is calculated by the indirect values:

$$N_{IND} = \frac{N_{IND}^{1} + N_{IND}^{2} + N_{IND}^{3} + N_{IND}^{4}}{4} = \frac{9937 + 10195 + 10132 + 10177}{4} = 10110\ IHP =$$
$$= 7436\ IKW$$

16) Enter the engine indication and performance data to the PC (Fig.1, Fig.2):

Conclusion: As we have seen from the Fig.1 the engine all cylinders indicator diagrams compression lines are in different position (arrow 1), that is what can not be for the same designed cylinders. They are should be in one line, that is can be adjusted by cylinders TDC correction individually (arrow 2). As we have seen from the Fig.2 the engine indicated power is 6464 IKW instead of calculated in item 15 – 7436 IKW, that is become 13.1% tolerance, which is not acceptable for the engine technical condition diagnostic and analyses. We have to correct the engine cylinders TDC totally.

17) The engine cylinders TDC angles (Fig.1) in degreases of crank angle CA:

Cylinder 1 TDC = – 1.5 $^{\circ}$ CA; Cylinder 2 TDC = – 1.5 $^{\circ}$ CA; Cylinder 3 TDC = – 2.5 $^{\circ}$ CA;
Cylinder 4 TDC = – 2 $^{\circ}$ CA; Cylinder 5 TDC = – 2.5 $^{\circ}$ CA; Cylinder 6 TDC = – 4 $^{\circ}$ CA;

18) Correct the engine cylinders TDC first of all individually for making the diagrams compression lines in one line (arrow 1), then totally for making the engine indicated power same as calculated in item 15 (arrow 2), (Fig.3, Fig.4):

Cylinder 1 TDC = – 4 ° CA; Cylinder 2 TDC = – 3.5 ° CA; Cylinder 3 TDC = – 4 ° CA;
Cylinder 4 TDC = – 4 ° CA; Cylinder 5 TDC = – 4 ° CA; Cylinder 6 TDC = – 5.5 ° CA;

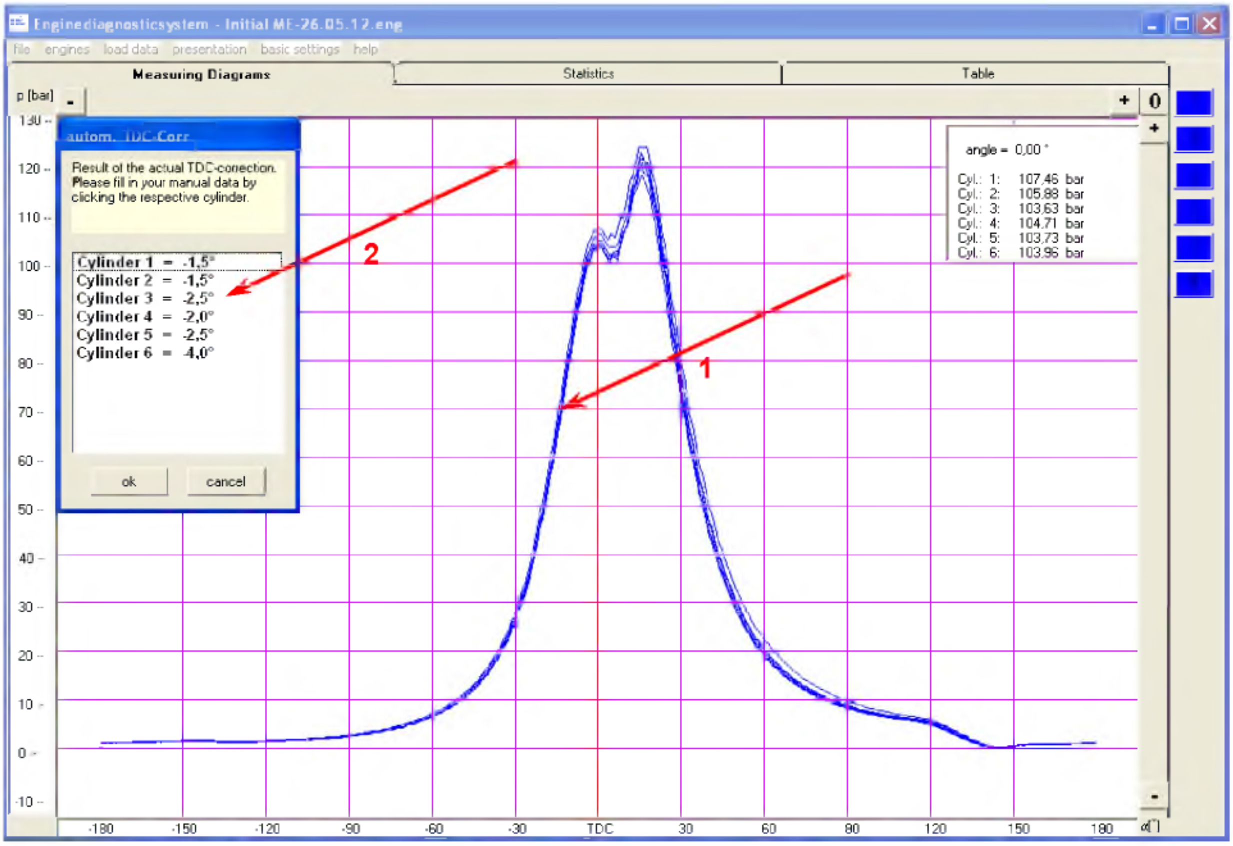
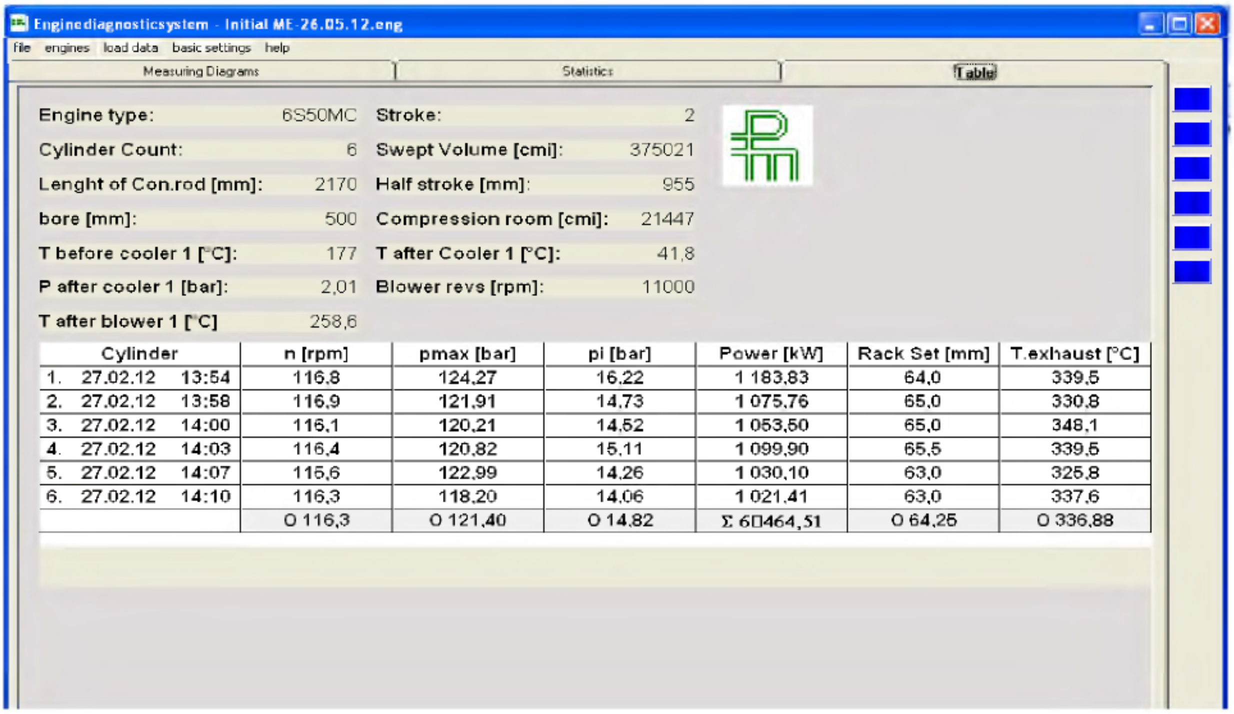


Figure 1 – Cylinders open indicator diagrams before TDC correction



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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	16,22	1 183,83	64,0	339,5
2. 27.02.12 13:58	116,9	121,91	14,73	1 075,76	65,0	330,8
3. 27.02.12 14:00	116,1	120,21	14,52	1 053,50	65,0	348,1
4. 27.02.12 14:03	116,4	120,82	15,11	1 099,90	65,5	339,5
5. 27.02.12 14:07	115,6	122,99	14,26	1 030,10	63,0	325,8
6. 27.02.12 14:10	116,3	118,20	14,06	1 021,41	63,0	337,6
	Ø 116,3	Ø 121,40	Ø 14,82	Σ 6 464,51	Ø 64,25	Ø 336,88

Figure 2 – Cylinders indication & performance data results table before TDC correction

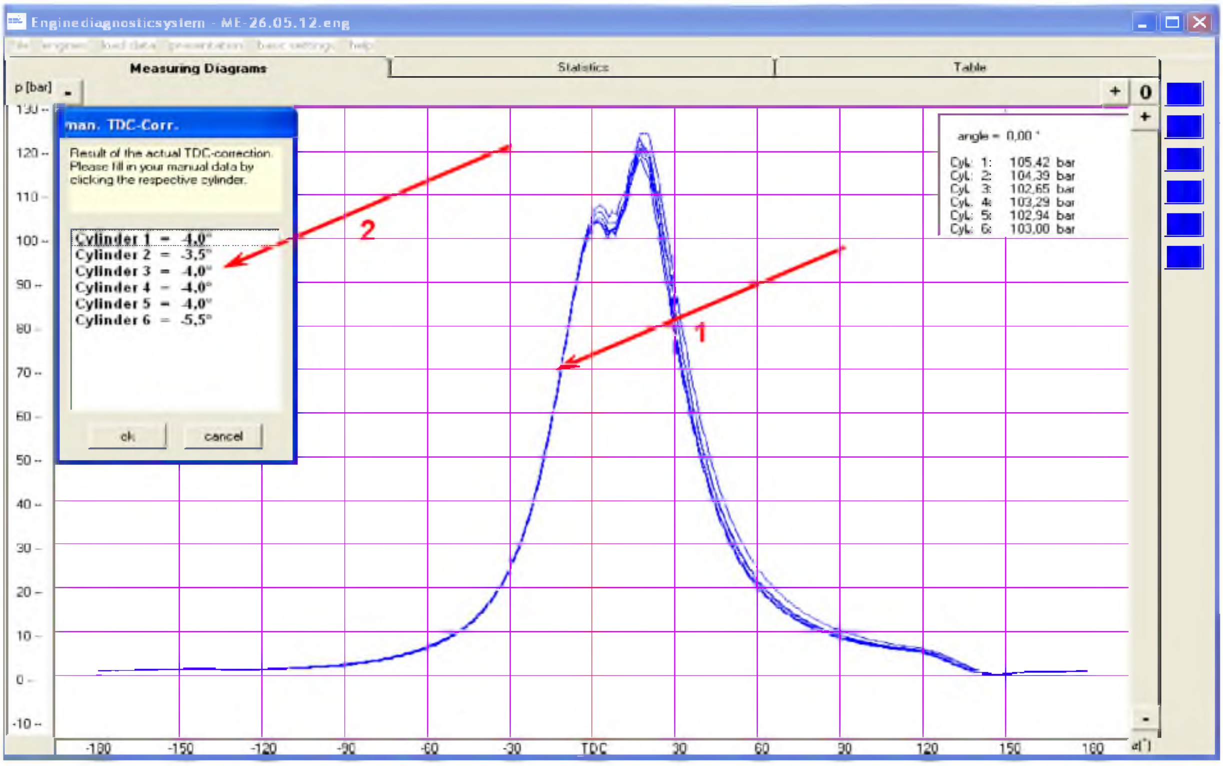
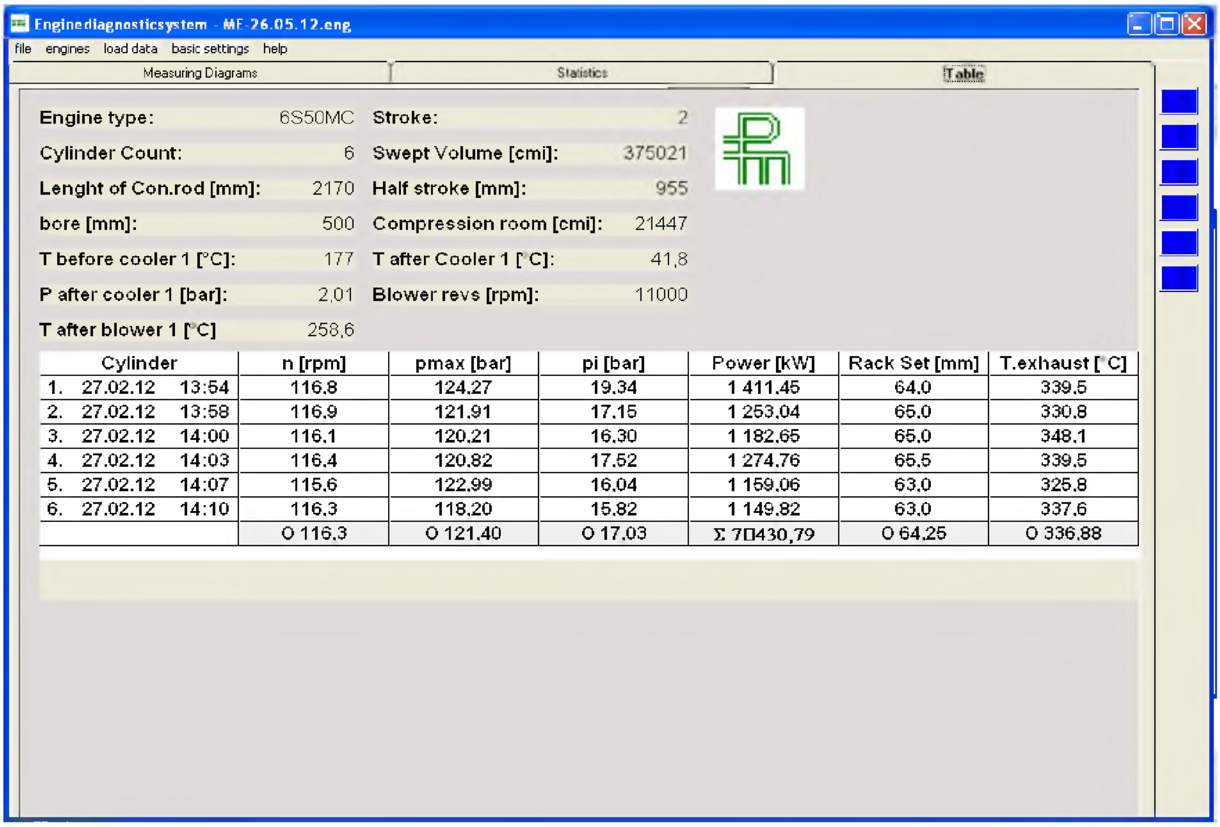


Figure 3 – Cylinders open indicator diagrams after TDC correction



Engine type:	6S50MC	Stroke:	2
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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
	Ø 116,3	Ø 121,40	Ø 17,03	Σ 7☐430,79	Ø 64,25	Ø 336,88

Figure 4 – Cylinders indication & performance data results table after TDC correction

Conclusion: As we have seen from the Fig.3 the engine all cylinders indicator diagrams compression lines are in one line (arrow 1) after TDC correction (arrow 2), that is what to be for the same designed cylinders. As we have seen from the Fig.4 the engine indicated power is 7431 IKW and almost the same with calculated in item

15 – 7436 IKW, that is become – 0.007% tolerance, which is perfect for the engine technical condition diagnostic and analyses.

19) The Diesel Propulsion Engine mechanical loss pressure calculation:

a) ME Turning Gear technical data from instruction manual (Table 3):

Table 3

Turning gear electromotor amperage	I^{ELM}	A	from turning gear technical data	4,9
Turning gear electromotor voltage	U^{ELM}	V	from turning gear technical data	440
Turning gear electromotor phases Nos	m	-	from turning gear technical data	3
Turning gear electromotor active load	P^{ELM}	HP	from turning gear technical data	3
Turning gear electromotor total load	S^{ELM}	HP	$S^{ELM} = 1.3596 \cdot m^{0,5} \cdot U^{ELM} \cdot I^{ELM} / 10^3$	5,077
Turning gear electromotor power factor	$\cos\varphi^{ELM}$	-	$\cos\varphi^{ELM} = P^{ELM} / S^{ELM}$	0,59088
Turning gear electromotor frequency	F^{ELM}	Hz	from turning gear technical data	60
Turning gear electromotor pole's pairs No	p	-	from turning gear technical data	3
Turning gear electromotor speed	n^{ELM}	rpm	$n^{ELM} = 60 \cdot F^{ELM} / p$	1200
Turning gear electromotor speed	n^{ELM}	rpm	from turning gear technical data	1155
Turning gear speed	n^{TG}	rpm	from turning gear technical data	1,04
Turning gear angular velocity	ω^{TG}	1/sec	$\omega^{TG} = \pi \cdot n^{TG} / 30$	0,10891
Turning gear output shaft torque	M^{TG}	N · mtr	from turning gear technical data	15696
Turning gear output shaft power	N^{TG}	HP	$N^{TG} = 1.3596 \cdot M^{TG} \cdot \omega^{TG} / 1000$	2,32414
Turning gear mechanical loss power	N_{MEC}^{TG}	HP	$N_{MEC}^{TG} = P_{ELM} - N^{TG}$	0,67586
Turning gear mechanical efficiency	η_{MEC}	-	$\eta_{ME}^{TG} = N^{TG} / P^{ELM}$	0,7747

b) ME mechanical loss pressure calculation by the turning gear operation data at ME opened indicator cocks (Table 4):

Table 4

Turning gear electromotor amperage	I	A	by observation	2,75
Turning gear electromotor voltage	U	V	by observation	446
Turning gear electromotor active load	P	HP	$P = 1.3596 \cdot m^{0,5} \cdot U \cdot I \cdot \cos\varphi / 10^3$	1,707
Turning gear output shaft power	N	HP	$N = P - N_{MEC}^{TG}$	1,031
ME turning time for 1 rev.by turning gear	t	sec	by observation	298
ME speed by turning gear	n^{ME}	rpm	$n^{ME} = 60 / t$	0,20134
ME mechanical loss pressure	P_{MEC}^{ME}	kg / cm^2	$P_{MEC} = N / (K \cdot n^{ME} \cdot i)$	1,024
ME mechanical loss pressure	P_{MEC}^{ME}	bar	$P_{MEC}^{BAR} = P_{MEC} / 1.0197$	1,004

or

c) Draw the diagram of the engine mechanical loss pressure dependence of the engine speed from shop trial test results table and found its dependence function by the trend line (Diagram 2):

d) The engine calculated mechanical loss pressure by the function is founded from the diagram 2:

$$P_{MEC} = 1.15598 \cdot 10^{-5} \cdot n_{ENG}^2 - 1.96628 \cdot 10^{-3} \cdot n_{ENG} + 1.13493 =$$
$$= 1.15598 \cdot 10^{-5} \cdot 116.8^2 - 1.96628 \cdot 10^{-3} \cdot 116.8 + 1.13493 = 1.063 \text{ kg / cm}^2 =$$
$$= 1.0425 \text{ bar}$$

20) The Diesel Propulsion Engine mean–effective pressure calculation:

$$P_{ME} = P_{MI} - P_{MEC} = 17.03 - 1.0425 = 15.9875 \text{ bar}$$

where: P_{MI} = 17.03 bar – from the engine performance data results table (Fig.4);
P_{MEC} = 1.0425 bar – from item 19), sub–item d) or 1.004 bar from table 4.

21) The Diesel Propulsion Engine effective power calculation:

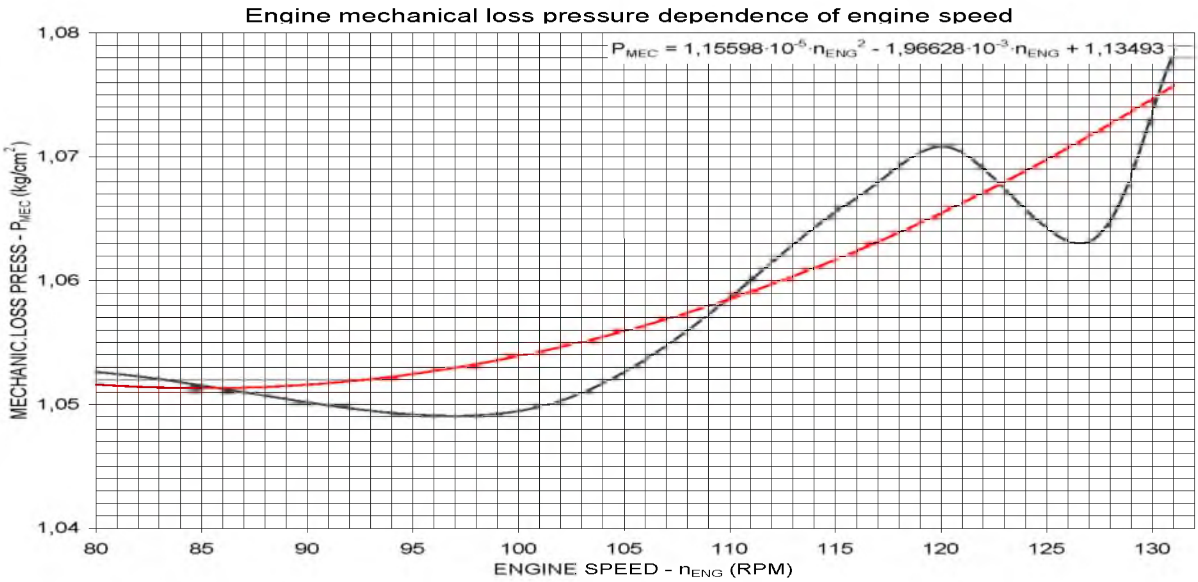
$$N_{EFF} = k \cdot P_{ME} \cdot n \cdot i = 0.624761 \cdot 15.9875 \cdot 116.8 \cdot 6 = 7000 \text{ EKW}$$

where: $k = 1.3084 \cdot D^2 \cdot S \cdot m = 1.3084 \cdot 0.5^2 \cdot 1.91 \cdot 1 = 0.624761$ – cylinder constant;
D = 0.5 mtr – cylinder diameter;
S = 1.91 mtr – piston stroke;
m = 1 – stroke factor (4–strike engine m = 2, 2–stroke engine m = 1).

Conclusion

As we have seen from mentioned above information for Diesel Propulsion Engines indicator diagrams TDC correction the ME indirect values measurement readings to be taken, recorded & output data have effected to the TDC correction to be calculated.

Diagram 2



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ПРЕДЛАГАЕМЫЕ МЕТОДЫ СНИЖЕНИЯ ВЫБРОСОВ SO_X, CO И CO_2 В АТМОСФЕРУ

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Методы снижения вредных выбросов в атмосферу с выпускными газами ДВС были предложены различными изданиями и публикациями заводов–изготовителей ещё 25 лет назад. Многие из них используются по настоящее время в зависимости от затрат на их установку, использование и техническое обслуживание. За 25 лет практического применения методов снижения выбросов на судах определило состоятельность и рентабельность дальнейшего их использования. Предлагаемый методснижениявыбросов SO_Xв атмосферу напрямую связан с используемым топливом, т.е. при снижении содержания серы в топливе снижаются и выбросы SO_X, что является задачей не судовладельцев а нефтеперерабатывающих производств и бункеровочных компаний.Снижение выбросов CO и CO_2 является краеугольной задачей, т.к. качество и низшая теплотворная способность топлива определяются содержанием углерода и водорода. Таким образом, снижение содержания углерода и водорода приведёт к снижению качества и низшей теплотворной способности топлива. А потому все эти 25 лет для снижения выбросов CO и CO_2 с выпускными газами судовых ДВС ставилась задача их энергоэффективности. Предлагаемый нами метод может позволить решить задачу снижения выбросов CO и CO_2 на частичных и номинальных режимах работы дизелей.

Ключевые слова:Вредные выбросы с выпускными газами ДВС, окислы углерода, низшая теплотворная способность топлива, метод снижения выбросов, тепловой баланс двигателя.

The Diesel Engines (ICE) exhaust gases atmosphere noxious emission decreasing ways 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 emission decreasing ways practical using on the vessels has identified it further usage consistency and profitability