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An assessment method of influence of the dirtiness of turbocharger nozzle blades on the main operating parameters of diesel engine and turbocharger

SUMMARY

This paper presents the algorithm of calculation program intended for assessing influence of the dirtiness of turbocharger nozzle blades on the main operating parameters of diesel engine and turbocharger. Also, results of an experiment with 3AL25/30 Sulzer diesel engine equipped with VTR-160N turbocharger is briefly presented as well as their comparison with results of calculation by means of the program in question. From these comparisons it can be concluded that the dirtiness of nozzle blades can be very harmful for the operating conditions of diesel engine and turbocharger, and finding effective ways of its reduction is necessary.

INTRODUCTION

For many operational reasons the combustion process which takes place in the combustion chamber of a diesel engine in operation can be incomplete. In result, a.o., the products of incomplete combustion (soot and ash) stick to the surfaces of nozzle and movable blades of the turbochargers. When the products tightly stick to the surfaces of nozzle blades they make the cross sectional area of the nozzle blade channel decreasing and the surface of nozzle blades becoming rough. This affects the operating conditions of diesel engine and turbocharger, and their operating parameters can exceed their limit values.

Moreover, in service the non-uniform surface of nozzle blades can cause unbalance of the turbine rotor in a form of oscillations, and lead to increasing noise in the engine room.

It is difficult to study these complex phenomena by means of the experimental methods. Therefore the use of computational methods was selected to determine the influence in question.

The course of this study was arranged in the following way :

⇨ **1st** : basic expressions combining the main operating parameters of diesel engine and turbocharger with those of the turbine nozzle channels were selected from different literature sources (mainly handbooks listed in bibliography).

⇨ **2nd** : on their basis a calculation algorithm and computer program was elaborated.

⇨ **3rd** : the experimental data, measured during a relevant experiment with 3AL25/30 Sulzer diesel engine equipped with VTR-160N turbocharger at normal conditions of turbine nozzle blades, were selected to be the initial data for the calculations.

⇨ **4th** : the main operating parameters of the diesel engine and turbocharger were calulated with the use of the elaborated computer program at different operating conditions of turbine nozzle blades.

⇨ **5th** : the experiments with the above mentioned diesel engine and turbocharger were executed again at the same conditions of turbine nozzle blades as those used in the calculations.

⇨ **6th** : the calculation results were compared with the experimental data at the same engine load and the same conditions of turbine nozzle blades to assess correctness of the calculation program.

⇨ **7th** : Final conclusions are offered.

The study is based on the following assumptions :

- Only the simple supercharging system and a parallel one is taken into account.
- Dirtiness degree of nozzle blades is modelled by blanking-off some turbine nozzle channels with steel plates.
- Ambient conditions and technical conditions of the diesel engine and turbocharger do not affect their operating parameters.
- Values of the operating parameters do not change with operating time (i.e. they are constant).
- Revolutions of the diesel engine and turbocharger are regular.
- The diesel engine operates at a load level lower than limit load.
- The air compressor operates at a region sufficiently away from the surge area.

BASIC EXPRESSIONS

The following basic expressions were selected from the literature sources in order to model the considered phenomena :

Air compressor efficiency η_c [2]

$$G_{co} = B_1(n_{tco}) + B_2(n_{tco}) \cdot [B_3(n_{tco}) - \eta_c]^{\frac{1}{2}} \qquad (1)$$

where :

$$B_1(n_{tco}) = a_1 \cdot (n_{tco} - b_1)$$

$$B_2(n_{tco}) = a_2 \cdot (n_{tco} - b_2)^{-\frac{1}{2}}$$

$$B_3(n_{tco}) = a_3 \cdot n_{tco}^{2.5} + 1$$

Air compressor power N_c [7]

$$N_c = \frac{R}{\kappa - 1} \cdot \kappa \cdot T_{am} \cdot \left[\pi_c^{\frac{\kappa-1}{\kappa}} - 1 \right] \cdot \frac{1}{\eta_c} \cdot G_c \qquad (2)$$

Revolutions of turbocharger n_{tc} [7]

$$n_{tc} = A \cdot (L_c)^{0.5} \qquad (3)$$

Charger efficiency η_v [5]

$$\eta_v = \frac{\varepsilon}{\varepsilon - 1} \cdot \frac{P_{as}}{P_s} \cdot \frac{T_s}{T_{as}} \cdot \frac{1}{1 + \gamma_g} \qquad (4)$$

Scavenging coefficient φ [7]

$$\varphi = 1 + \frac{\sqrt{R \cdot T_s}}{V_{de} \cdot i \cdot z \cdot 6 \cdot n_{de} \cdot \eta_v} \cdot \psi\left(\frac{P_{tc}}{P_s} \right) \cdot \mu F_{de} \qquad (5)$$

Cylinder displacement air mass flow rate G_{kad} [1]

$$G_{kad} = V_{de} \cdot i \cdot z \cdot \frac{n_{de}}{60} \cdot \rho_s \cdot \eta_v \qquad (6)$$

Cylinder scavenging air mass flow rate G_{ks} [1]

$$G_{ks} = \mu F_{de} \cdot (\rho_s \cdot P_s)^{0.5} \cdot \psi\left(\frac{P_{tc}}{P_s} \right) \qquad (7)$$

Indicated efficiency η_i [4]

$$\eta_i = A \cdot (\alpha - 0.1)^{\frac{1}{\alpha + 0.2}} \qquad (8)$$

Mechanical losses in the form mean effective pressure P_m [1]

$$P_m = (c_1 + c_2 \cdot \bar{n}_{de} + c_3 \cdot \bar{n}_{de}^2) \cdot \left(\frac{P_s}{P_{am}} \right)^{0.1} + \\ + (P_{tc} - P_s) + \Delta P_n \cdot \bar{n}_{de}^2 \cdot \bar{\rho}_s \qquad (9)$$

Indicated pressure P_i [5]

$$P_i = \frac{Q_f}{M} \cdot \frac{P_s}{R \cdot T_s} \cdot \frac{\eta_i \cdot \eta_v}{\alpha} \qquad (10)$$

Specific fuel oil consumption g_{de} [1]

$$g_{de} = \frac{G_f}{N_e} \qquad (11)$$

Exhaust gases temperature before turbocharger T_{tc} [4]

$$T_{tc} = \frac{h_{pb}}{h_{pg}} \cdot T_s + \frac{Q_f \cdot (1 - \eta_i - q_{res})}{\alpha \cdot M \cdot \varphi \cdot h_{pgt}} \qquad (12)$$

Exhaust gases pressure before turbocharger P_{tc} [3]

$$P_{tc} = \frac{\mu F_{tcn} \cdot G_g \cdot P_{tcn} \cdot \sqrt{T_{tc}} \cdot \psi\left(\frac{P_{atcn}}{P_{tcn}} \right)}{\mu F_{tc} \cdot G_{gn} \cdot \sqrt{T_{tcn}} \cdot \psi\left(\frac{P_{atc}}{P_{tc}} \right)} \qquad (13)$$

Exhaust gases pressure behind turbocharger P_{atc} [3]

$$P_{atc} = \left[1 + \left(\frac{P_{atcn}}{P_{am}} - 1 \right) \cdot \left(\frac{G_g}{G_{gn}} \right)^2 \right] \cdot P_{am} \qquad (14)$$

Turbocharger efficiency η_{tc} [6]

$$\eta_{tc} = e_1 + e_2 \cdot \left(\frac{U_{tc}}{C_g} \right) + e_3 \cdot \left(\frac{U_{tc}}{C_g} \right)^2 + \\ + e_4 \cdot \left(\frac{P_{tc}}{P_{atc}} \right) + e_5 \cdot \left(\frac{P_{tc}}{P_{atc}} \right)^2 + e_6 \cdot \left(\frac{U_{tc}}{C_g} \right) \cdot \left(\frac{P_{tc}}{P_{atc}} \right) \qquad (15)$$

Turbocharger power N_{tc} [4]

$$N_{tc} = G_g \cdot \frac{R_g}{\kappa - 1} \cdot \kappa_g \cdot T_{tc} \cdot \left[1 - \left(\frac{P_{atc}}{P_{tc}} \right)^{\frac{\kappa_g - 1}{\kappa_g}} \right] \cdot \eta_{tc} \qquad (16)$$

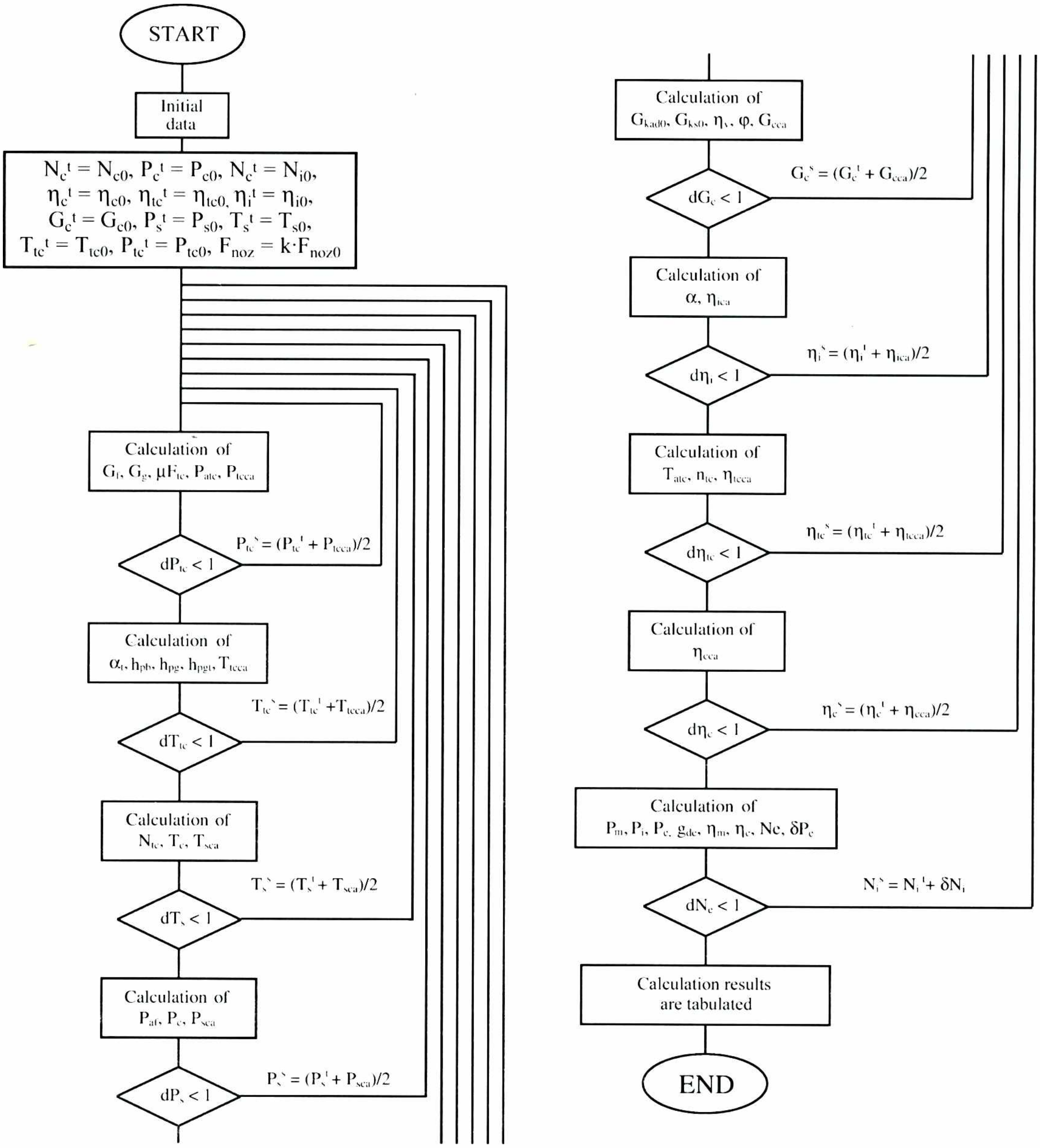


Fig. 1. Flow diagram of the calculation algorithm

CALCULATION ALGORITHM AND COMPUTER PROGRAM

The calculation flow diagram based on the above presented formulas is shown in Fig.1.

On its basis the calculation program was written in the Borland Pascal computer language with taking into account particulars and dimension parameters of the test stand system used for verification of the proposed calculation method. Parameter values of the diesel engine and turbocharger at normal operation conditions were assumed initial data for calculations.

EXPERIMENTAL TESTS AND EXAMPLE CALCULATIONS

In order to verify the proposed calculation method experimental tests were arranged with the use of the test stand of Gdynia Maritime Academy laboratory. The experimental system consisted of 3AL25/30 diesel engine equipped with VTR-160N turbocharger, driving DOLMEL GD8-500-50 electric power generator. Particulars of the test engine, turbocharger and generator are presented in Tab.1 and 2, and scheme of their subsystems taken into account, measured parameters and measuring equipment is shown in Fig.2 (the electric power generator is omitted).

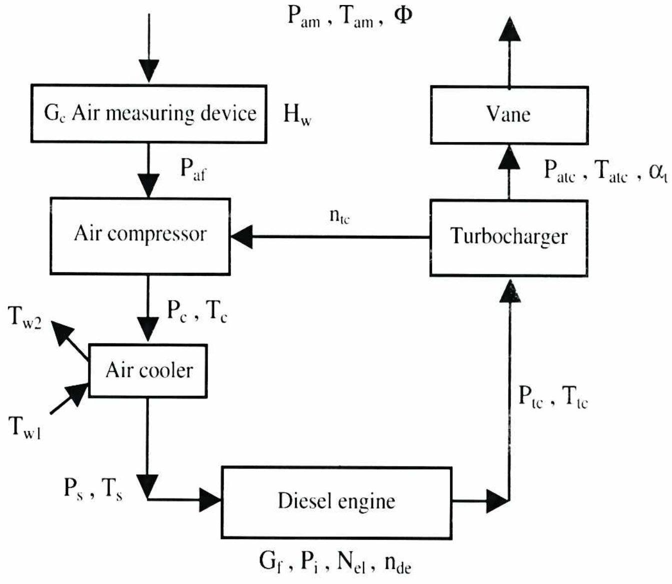


Fig.2. Scheme of the test stand subsystems taken into account, measuring equipment and measured parameters

Tab.1. Particulars of the test engine

Builder	SULZER	Piston stroke	300 mm
Engine	3AL25/30	rpm at MCR	750
Type	Marine Single Acting 4 Stroke	MCR	408 kW
Number of cylinders	3	Compression ratio	13
Cylinder bore	250 mm	Fire order	1-3-2

Tab.2. Particulars of the test turbocharger and electric power generator

Turbocharger		Electric power generator	
Builder	BBC	Builder	DOLMEL
Type	VTR-160N	Type	GD8-500-50
Max speed	43,000 rpm	Power	500 kVA
Max temperature	670°C	Voltage	400 V
Supercharging system	simple	Current	723 A (3 phase)

The test measurements were carried out at 270 kW electric power output of the generator and at the following three values of the cross-section area of the nozzle blade channels :

- $F_{noz} = 100\% \;\; F_{nozO}$
- $F_{noz} = 95.83\% F_{nozO}$ (the situation shown in Fig.3a)
- $F_{noz} = 91.66\% F_{nozO}$ (the situation shown in Fig.3b)

The example calculations were run for the same test stand diesel engine and turbocharger system and at the same conditions which were applied during the described experiments.

Results of the example calculations are compared with the experimental data in Tab.3. as well as in the form of the diagrams (Fig.4.).

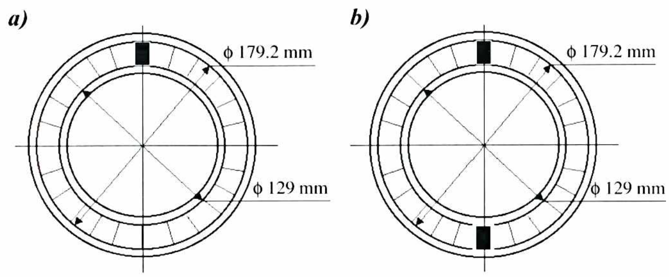
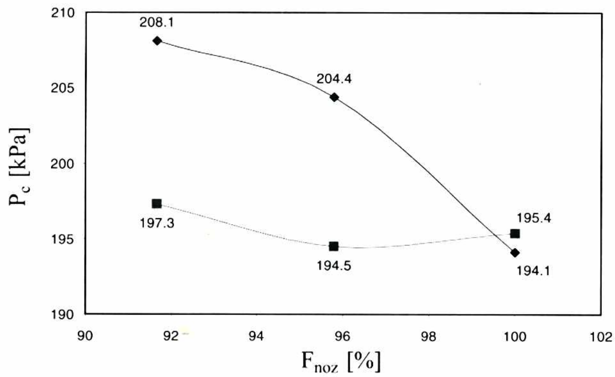


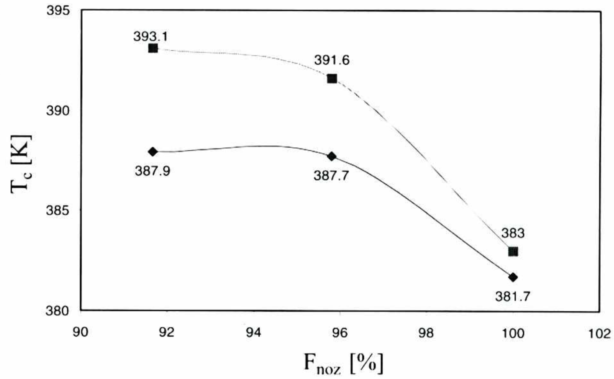
Fig.3. Position of blanking-off steel plates :
a - One turbine nozzle channel blanked - off
b - Two turbine nozzle channels blanked - off

Tab.3. Comparison of example calculation results and the experimental data obtained at 270 kW electric power output of the test stand generator

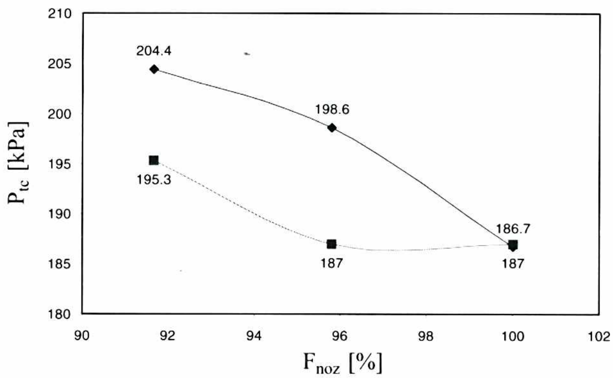
Parameters	Experimental data			Example Calculation results		
F_{noz} [%]	100	95.83	91.66	100	95.83	91.66
N_e [kW]	285.400	285.400	285.400	286.961	287.813	288.181
P_{am} [kPa]	101.1	100.4	101.6	101.1	101.1	101.1
T_{am} [K]	297.0	299.0	301.0	297.0	297.0	297.0
T_w [K]	297.5	295.0	295.5	297.5	297.5	297.5
P_c [kPa]	195.4	194.5	197.3	194.1	204.4	208.1
P_s [kPa]	195.0	194.1	196.9	193.7	204.0	207.7
P_{tc} [kPa]	187.0	187.0	195.3	186.7	198.6	204.4
P_{atc} [kPa]	102.1	101.4	102.6	102.1	102.2	102.1
T_c [K]	383.0	391.6	393.1	381.7	387.7	387.9
T_s [K]	305.0	312.0	316.0	304.9	305.4	305.4
T_{tc} [K]	812.0	816.0	815.0	801.9	787.8	797.2
T_{atc} [K]	679.0	688.0	686.0	688.0	665.5	668.5
G_c [kg/s]	0.6990	0.7006	0.6993	0.70042	0.72215	0.70761
G_f [kg/s]	0.0201	0.0202	0.0201	0.01964	0.01955	0.01952
G_g [kg/s]	0.7191	0.7208	0.7194	0.72007	0.74170	0.72712
η_i [-]	0.496	0.461	0.491	0.498	0.501	0.502
η_e [-]	0.339	0.337	0.339	0.351	0.354	0.355
η_v [-]	0.935	0.938	0.939	0.936	0.937	0.936
φ [-]	1.216	1.249	1.242	1.214	1.188	1.148
α_t [-]	2.610	2.520	2.560	2.676	2.772	2.721
P_i [kPa]	1510.0	1413.3	1496.6	1472.9	1475.3	1478.2
N_{tc} [kW]	65.659	70.854	70.348	64.773	71.521	70.251
η_c [-]	0.721	0.676	0.687	0.714	0.725	0.744
η_{tc} [-]	0.700	0.742	0.705	0.700	0.699	0.665
μF_{tc} [m^2]	0.0087	0.0087	0.0083	0.00870	0.00834	0.00797
g_{de} [kg/kW.h]	0.254	0.255	0.254	0.24643	0.24453	0.24380
n_{tc} [rpm]	34150.0	34300.0	34560.0	33904.9	35087.1	35129.6



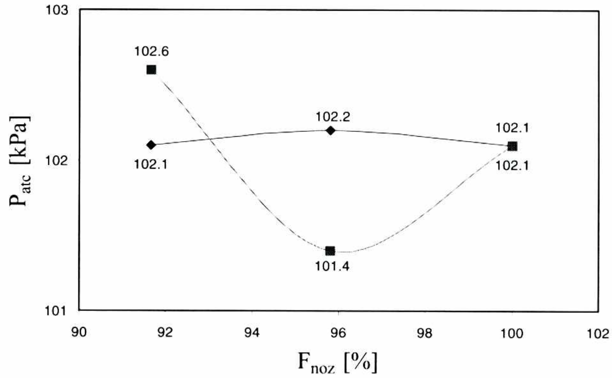
P_c [kPa]
F_noz [%]
208.1
204.4
197.3
194.5
195.4
194.1



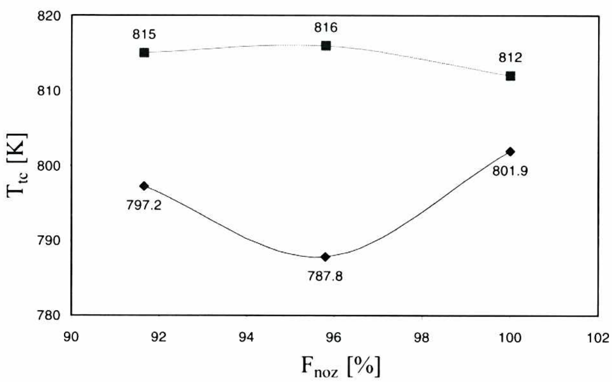
T_c [K]
F_noz [%]
393.1
391.6
387.9
387.7
383
381.7



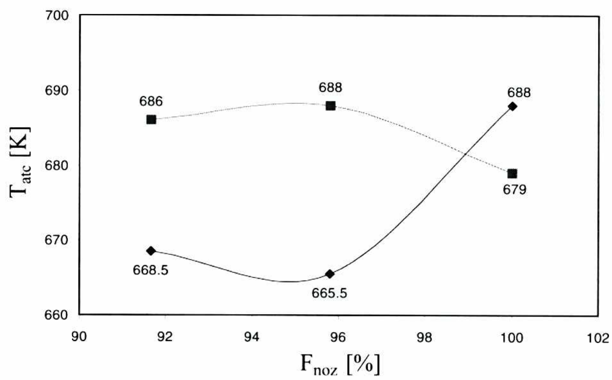
P_tc [kPa]
F_noz [%]
204.4
198.6
195.3
186.7
187
187



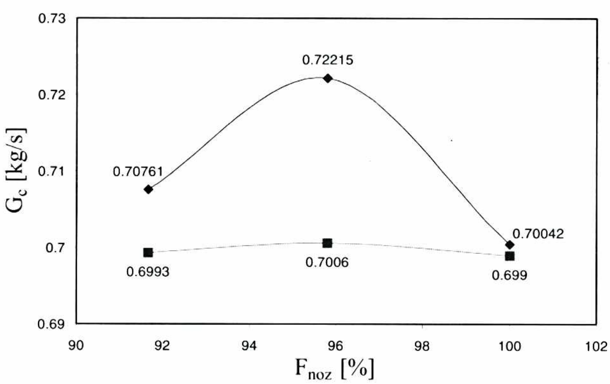
P_atc [kPa]
F_noz [%]
102.6
102.2
102.1
102.1
102.1
101.4



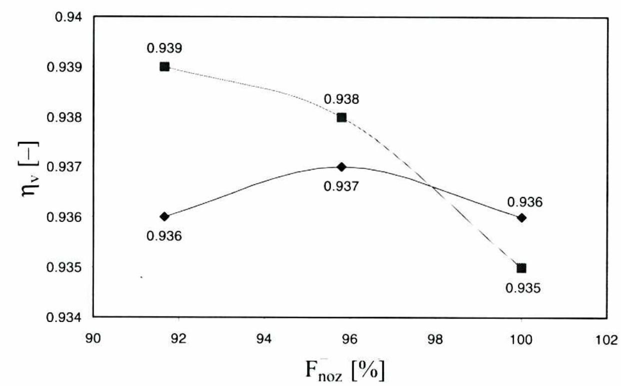
T_tc [K]
F_noz [%]
815
816
812
797.2
801.9
787.8



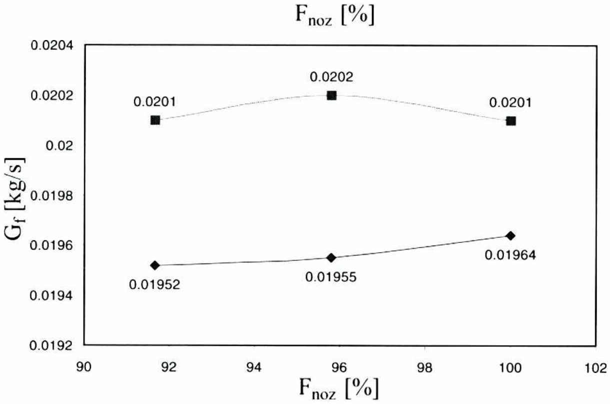
T_atc [K]
F_noz [%]
686
688
688
679
668.5
665.5



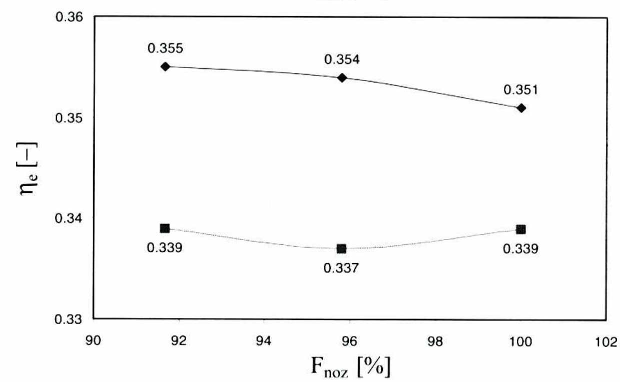
G_c [kg/s]
F_noz [%]
0.72215
0.70761
0.70042
0.6993
0.7006
0.699



η_lv [–]
F_noz [%]
0.939
0.938
0.937
0.936
0.936
0.935



G_f [kg/s]
F_noz [%]
0.0201
0.0202
0.0201
0.01964
0.01952
0.01955



η_e [–]
F_noz [%]
0.355
0.354
0.351
0.339
0.339
0.337

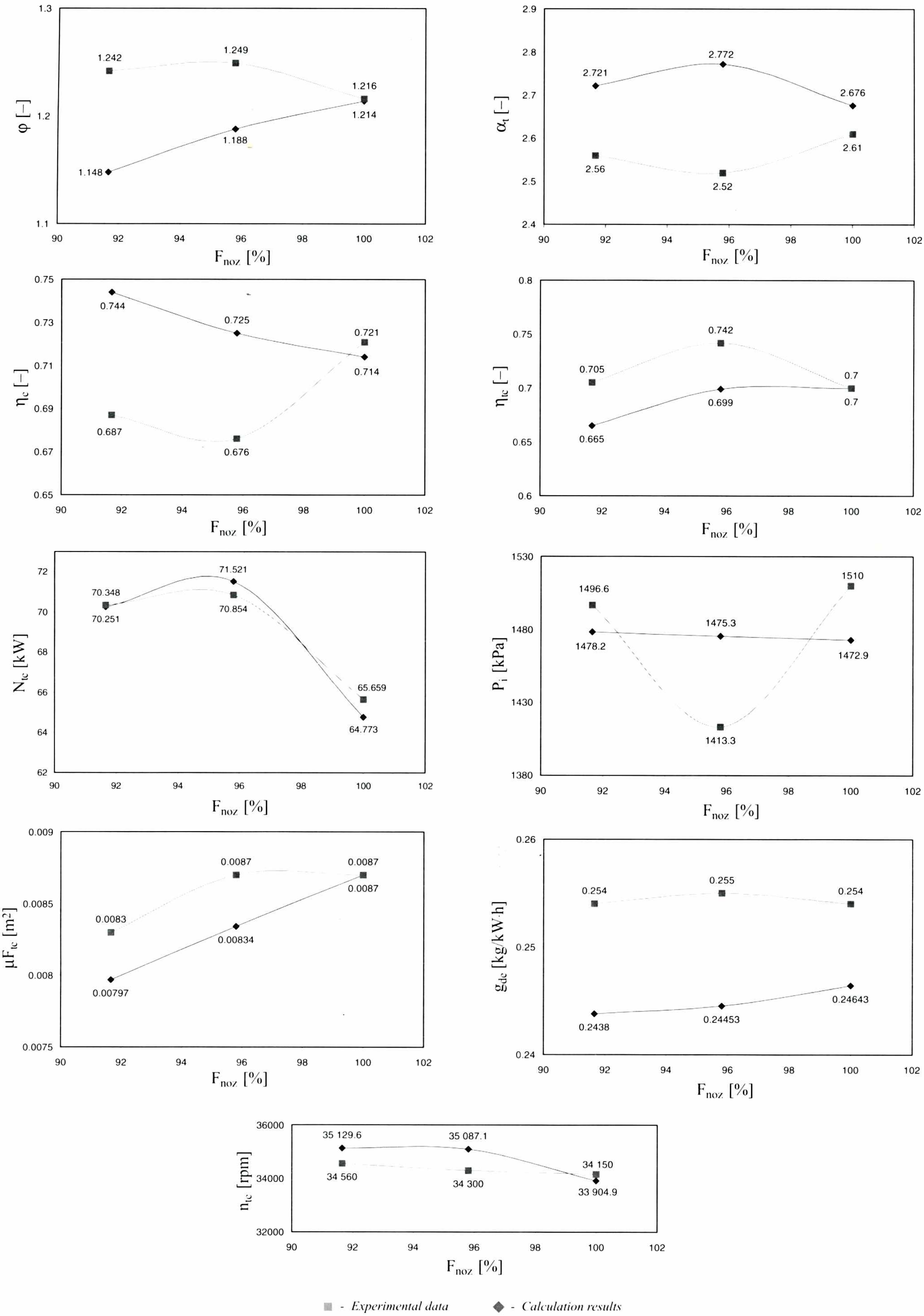


Fig.4. *Values of operating parameters of the diesel engine and turbocharger in function of the cross-section area of turbine nozzle channel, F_{noz} (Comparison of experimental data and calculation results)*

CONCLUSIONS

From the presented comparison of the experimental data and results of the example calculations performed with the use of the elaborated computer program the following conclusions can be offered:

- The corresponding values of the investigated parameters of the diesel engine and turbocharger are satisfactorily close to each other, and are supposed to be contained within their measurement and calculation error ranges.
- Therefore the computer program would be considered suitable for assessment of influence of the dirtiness of turbocharger nozzle blades on the main operating parameters of the system in question at least within the range of cross-section area variability as that investigated in this study, and provided the initial data corresponding to the normal operating conditions of a system to be investigated are known.
- However the change course of the parameters presented in Fig.4 is rather slight within the considered range of cross-section area variability. In order to investigate the above mentioned problem more thoroughly in the next phase of this work the range of the study will be extended to greater values of cross-section area of the nozzle blade channels, occupied by combustion products.

Appraised by Romuald Cwilewicz, Assoc.Prof.,D.Sc.,M.E.

NOMENCLATURE

a - constant in the formula for calculating air compressor efficiency
A - constant in the formula for calculating revolutions of turbocharger
b - constant in the formula for calculating air compressor efficiency
B(n_{tco}) - coefficient depending on turbocharger revolutions at standard ambient conditions
c - constant in the formula for calculating mechanical losses in the form of mean effective pressure
C - absolute velocity of exhaust gas at turbocharger outlet [m/s]
d - loop parameter percentage
e - constant in the formula for calculating turbocharger efficiency
F - cross sectional area [m^2]
g - specific fuel oil consumption [kg/kW·h]
G - mass flow rate [kg/s]
h - specific heat [kJ/kg·K]
H_w - water head at measuring device of air mass flow rate [mmH_2O]
i - number of cylinders
k - ratio of the cross sectional area of turbocharger nozzle blade channels at changed condition to that at normal condition
L - work of adiabatic compression [kJ/kg]
M - theoretical air mass necessary for the complete combustion of one kilogram of liquid fuel [kg of air/kg of liquid fuel]
n - revolutions [rpm]
N - power [kW]
P - pressure [mmHg], [kPa]
q - percentage of heat losses
Q - lower heat value [kJ/kg]
R - characteristic gas constant [kJ/kg·K]
T - temperature [°C], [K]
U - peripheral velocity [m/s]
V - cylinder volume [m^3]
z - stroke factor
α - excess air coefficient during combustion
γ - residual gas coefficient
δ - difference
ΔP - mean pressure loss at valves of the diesel engine [mmHg], [kPa]
ε - actual compression ratio
η - efficiency
κ - adiabatic exponent
μF - equivalent cross sectional area [m^2]
π - pressure ratio
ρ - specific density [kg/m^3]
φ - scavenging coefficient
Φ - ambient humidity [%]
ψ - flow function

Upper indices:

- - relative value
s - parameter value at the next loop of calculations
t - parameter value at the previous loop of calculations

Lower indices:

0 - initial parameter
1, 2, 3, 4, 5, 6 - order number of constant coefficients
af - behind the measuring device of air mass flow rate
am - ambient condition
as - at end of the admission stroke in diesel engine cylinder
atc - behind turbocharger
ca - calculated value
e - effective
f - of fuel oil
i - indicated
ks - of cylinder scavenging
c - of compressor
de - of diesel engine
el - electric
g - of exhaust gas
kad - of cylinder displacement
m - mechanical
n - at rated load of diesel engine
noz - of turbocharger nozzle blade channel
o - at standard ambient conditions
pb, pg, pgt - of fresh air, exhaust gas and air-fuel mixture at constant pressure, respectively
res - of the heat losses (that transferred to the cooling water, to the atmosphere through radiation, lost through incomplete combustion, part of the friction heat, as well as due to errors in determining the heat balance components in diesel engine)
s - behind air cooler
t - total
tc - of turbocharger
v - of charger
w1, w2, w - of cooling water at the air cooler inlet, outlet and average, respectively

BIBLIOGRAPHY

1. Baikow B. P., Bordukow V. G., Ivanow P. V., Deich R. S.: *Turbocompressors for supercharging the diesel engines* (in Russian). Mashinostroenie. Leningrad, 1975
2. Ghaemi M. H.: *Zero-Dimensional, Instantanoeus Value, Quasi-Steady Mathematical Model of Marine Diesel Engine*. Technical University of Gdańsk. Gdańsk, 199.
3. Kamkin S. V.: *Gas replacement and supercharging of the ship diesel engines* (in Russian). Sudostroenie. Leningrad, 1972
4. Mezheritskiy A. D., *Turbocompressors of the ship diesel engine supercharging systems* (in Russian). Sudostroenie. Leningrad, 1986
5. Samsonow V. I., Khudow N. I.: *Internal combustion engines for ship* (in Russian). Transport. Moscow, 1990
6. Tolshin V. I., Mokerow.L. F., Devyanin S. N.: *Computer calculation of the internal combustion engine systems and units* (in Russian). Moscow, 1986
7. Vasilev~YU Zhin R. M.: *Ship diesel engine operation at non-specific conditions* (in Russian). Sudostroenie. Leningrad, 1967