Affections of Turbine Nozzle Cross-Sectional Area to the Marine Diesel Engine Working
Utjecaj prostora presjeka turbo mlaznice na rad pomorskoga dizelskog stroja

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Summary
After a long period of use, some important technical parameters of the main marine diesel engines (MDE) gradually become worse, such as the turbine speed, intake pressure, exhaust temperature, engine power, and specific fuel oil consumption (SFOC). This paper studies the affections of the turbine nozzle cross-sectional area (AT) to MDE and presents a method of AT adjustment to improve the performances of MDE. A mathematical model of an engine was built based on the existent engine construction and the theory of the diesel engine working cycle and the simulation was programmed by Matlab/Simulink. This simulation model accuracy was evaluated through the comparison of simulation results and experimental data of the MDE. The accuracy testing results were acceptable (within 5%). The influences of AT on the engine working parameters and the finding optimization point were conducted by using the simulation program to study. The predicted optimization point of the nozzle was used to improve the engine’s performances on board. The integration of the simulation and experiment studies showed its effectiveness in the practical application of the marine diesel engine field.

1. INTRODUCTION / Uvod
On the commercial motor vessels, the main engines usually are high-power diesel engines (two-stroke or four-stroke), most of them turbocharged by axial turbochargers (TC) due to retaining their high efficiency at medium and larger-size compared to the radial turbocharger [1]. Therefore, there are many studies focused on these objects. The experimental research of Rahmke [2] showed that for the same assembly of the axial and radial turbocharger, the inertia moment of the first type is about half of the inertia moment of the second one. In the work [3], Pesiridis, Saccomanno, Tuccillo, and Capobianco modified the design of an axial turbine and simulated by CFD theory. Their simulation results showed that the dynamic energy of the gas flow could be regulated by controlling the nozzle cross-sectional area, such as a slide hub wall.

Some methods have been used in modeling marine diesel engines. Theotokatos [4] investigated the transient response of two-stroke MDE by the cycle mean value models. The combination studies of the mean value and zero-dimensional models to enhance the accuracy of the MDE models was carried out by the following authors: Baldi, Theotokatos, and Andersson [5] for a four-stroke MDE, and Tang, Zhang, Gan, Jia, and Xia [6] for a two-stroke MDE. Sun, Wang, Yang, and Wang [7] developed and validated a sequential turbocharging MDE combustion model by the partial least squares method with acceptable accuracy.

The main marine diesel engine operates in heavy load conditions and continuity, it may operate about 6000 hours of 8760 hours per year, many of those at full design load [8], the

Summary
Sažetak
Nakon dugoga razdoblja uporabe, neki važni tehnički parametri glavnega dizelskoga brodskog motora postupno so postali lošiji, kaj so tu: brzina turbine, ulazni pritisak, ispušna temperatura, snaga stroja in specifična potrošnja goriva. Ovaj članak proučava svojstva prostora presjeka turbo mlaznice (AT) na brodski dizelski stroj in predstavlja metode AT prilagodbe, kako bi se poboljšale izvedbe brodskoga dizelskog stroja. Matematični model motora postavljen je prema potočni konstrukciji stroja in teoriji rada radnega ciklusa dizelskega stroja, a simulacijo je programirali Matalab/Simulink. Preciznost simulacijskega modela evaluirata se na temelju usporedbe rezultatov simulacije in eksperimentalnih podatkov pomorskoga dizelskega motora. Rezultati preciznosti testiranja bili sta prihvatljivi (unutar 5%). Utjecaj AT na radne parametre motora in trženje optimizacijske točke prevedeni so koriščeni simulacijskim programom v studiji. Pretpostavljena optimizacijska točka štrkaljke uporabljena je kako bi poboljšala izvedbo brodskoga motora. Integracija simulacije in studije eksperimenta pokazala je dejotvornost pri praktičnem primeri v polju brodskoga dizelskega stroja.

KEY WORDS
main marine diesel engine
turbine speed
nozzle cross-sectional area

KLJUČNE RIJEČI
glavni dizelski stroj
brzina turbine
prostor presjeka mlaznice
SFOC and the emissions gradually increased. The main reason of the scavenging air amount deficiency for the normal engine operation is the bad working of the turbocharger. Therefore, the improvements of the gas distribution system can improve the performances of the engine. In the statistical research on the ABB turbocharger [9], Schieman showed that the turbine efficiency and charging pressure had been decreased so much due to the turbocharger dirty and found a way to clean compressor and turbine blades by water or blasting with ground nutshells.

This work focused on the MDE that has a long using time, and at the practical operation, in the load range (Load Index, Li): Li= 60% ÷ 68%. The research methodology of this paper was presented in the following procedure. Firstly, the mathematical model was built and written its code (simulating) in MatLab / Simulink program. The model was presented accurately and reliably in accordance with the difference between the simulation results and test records provided by the manufacturer. Secondly, the optimum nozzle cross-sectional area was predicted to receive the optimal working values of the main important parameters of the MDE by simulation way. This step is very important for practice orientation to narrow the nozzle of the turbocharger. Thirdly, the real experiment study on a turbocharged marine diesel engine was conducted and analysed the experimental results.

2. MODEL STRUCTURE / Struktura modela

2.1. Marine diesel engine and turbocharger relationship / Brodski dizelski stroj u odnosu na turbopuhalo

The MDE and TC relationship was based on the intake air mass flow rate (\(\dot{m}_\text{a}\)) and exhaust air mass flow rate (\(\dot{m}_\text{o}\)). The principle scheme of the MDE –TC relationship was shown in Fig.1.

![Figure 1 Principle scheme of MDE and turbocharger relationship](Image)

**Slika 1. Osnovna shema brodskoga dizelskog stroja u odnosu na turbopuhalo**

The intake air mass flow rate (\(\dot{m}_\text{a}, \text{kg/s}\)) is given as below:

\[
\dot{m}_\text{a} = \eta_\text{a} \cdot \frac{p \cdot V_\text{e}}{60z \cdot R \cdot T_\text{i}}
\]  
(1)

Where, \(\eta_\text{a}\) (rpm) – engine rotatory speed; \(V_\text{e}\) (m³) – total swept volume; \(\eta_\text{a}\) (−) – volumetric efficiency; \(z\) – diesel stroke: \(z=1\) for two-stroke, \(z=2\) for four-stroke engine; \(R\) (l/kg/K) - gas constant; \(p\) (kN/m²) – intake air pressure; \(T_\text{i}\) (K) – intake air temperature;

The total intake mass flow rate (\(\dot{m}_{\text{a, tot}}, \text{kg/s}\)) can be calculated as equation (3):

\[
\dot{m}_{\text{a, tot}} = \dot{m}_\text{a} + \dot{m}_\text{f}
\]  
(2)

Where, \(\dot{m}_\text{f}\) (kg/s) is the rate of the feed fuel oil that is calculated as equation (3):

\[
\dot{m}_\text{f} = \frac{m_{\text{f, cycle}} \cdot n_{\text{cycle}}}{60z}
\]  
(3)

Where, \(m_{\text{f, cycle}}\) (kg/cycle) is the amount of fuel oil per cycle (in this work, it was determined by the experiment); \(n_{\text{cycle}}\) is the number of cylinders.

2.2. Cylinder model / Model cilindra

Intake process / Proces unosna

The intake process quality can be evaluated by the volumetric efficiency \(\eta_\text{v}\) in the following expression [10]:

\[
\eta_\text{v} = c_1 + c_2 \cdot \sqrt{\eta_\text{a}} + c_3
\]  
(4)

Where, \(c_1\), \(c_2\), \(c_3\) are adjustable parameters, and estimated by the least square regression method. In accordance with the determination coefficient of the model, R-squared = 0.9689, the model coefficients are defined [11]: \(c_1=0.006651\); \(c_2=0.7429\); \(c_3=-0.4093\).

Combustion process / Proces izgaranja

From the ideal gas, by differentiating of this equation: \(PV=mRT\), the temperature model is obtained as equation (5) bellow:

\[
\frac{dT}{dt} = \frac{1}{mR} (pdV + Vdp)
\]  
(5)

Where, \(R\) (J/kg.K) is the mixture gas constant; \(m\) (kg) is the amount of mixture gas.

Similarly, the differential pressure model is [11]:

\[
\frac{dQ}{dp} = \frac{Q}{V p^\gamma + (1-\gamma)}
\]  
(6)

Where, \(\gamma(\cdot)\) is the specific heat capacity ratio of the mixed gas;

The heat release equation \(\frac{dQ}{dt}\) is shown as bellow:

\[
\frac{dx_m}{dt} = \beta \cdot \frac{dx_{\text{a, tot}}}{dt} + (1-\beta) \frac{dx_{\text{f}}}{dt}
\]  
(8)

\[
\frac{dx_{\text{a, tot}}}{dt} = a_{\text{a}} \cdot \frac{m_{\text{a}}}{\varphi_p} \cdot \frac{\varphi - \varphi_p}{\varphi_p - \varphi_p} \exp\left(-a_{\text{a}} \cdot \frac{\varphi - \varphi_p}{\varphi_p - \varphi_p}\right)
\]  
(9)

\[
\frac{dx_{\text{f}}}{dt} = a_{\text{f}} \cdot \frac{m_{\text{f}}}{\varphi_p} \cdot \frac{\varphi - \varphi_p}{\varphi_p - \varphi_p} \exp\left(-a_{\text{f}} \cdot \frac{\varphi - \varphi_p}{\varphi_p - \varphi_p}\right)
\]  
(10)

Where \(a_{\text{a}}, a_{\text{f}}, m_{\text{a}}, m_{\text{f}}\) are shape factors; \(\varphi_p, \varphi_{p, f}\) are the duration of premixed and diffusion phases; \(\varphi_p, \varphi_{p, f}\) is the start of combustion.

Combustion factors \(a_{\text{a}}, a_{\text{f}}, m_{\text{a}}, m_{\text{f}}, \varphi_p, \varphi_{p, f}\) was estimated as [15], \(a_{\text{a}}=6.9; \varphi_p=70\); \(m_{\text{a}}=3; m_{\text{f}}=0.5\), and \(\varphi_{p, f}\) was calculated by least square regression method as [15], \(\varphi_{p, f} = c_{p, 2} p + c_{p, 3} p + c_{p, 4} p + c_{p, 5} c_{p, 7}\); \(c_{p, 2}=0.0002931; c_{p, 3}=0.05108; c_{p, 7}=-1.313\), with coefficient of determination R-squared=0.9956.

The heat transfer \(\frac{dQ}{dp}\) from the gases to the combustion surfaces is as bellow [16]:

\[
\frac{dQ}{dp} = h(\varphi) A_{\text{s, p}} (T_{p, s} - T_{\text{e}}) \frac{2\pi n}{2\pi n}
\]  
(11)

Where, \(A_{\text{s, p}}(\varphi)\) is the area of surfaces (head, cylinder, piston); \(T_{p, s}(\text{K})\) is the average temperature of surfaces; \(h(\varphi)\) is the heat transfer coefficient, estimated by Woschni [17] and corrected by Heywood [12], given as below:

\[
h = 3.26p^{0.14} U^{0.58} b^{4.2} T^{-0.55}
\]  
(12)
Where, $p(N/m^2)$ - gas pressure; $U(m/s)$ - gas velocity; $b(m)$ - cylinder bore; $T(K)$ - gas temperature.

**The total mechanical losses / Ukupni mehanički gubici**
The total mechanical friction pressure of the engine, $p_f(kPa)$, includes friction losses and pumping losses which was determined through the engine speed $n_e$(rpm), mean piston speed $(m/s)$ and is defined in [12] as below:

$$p_f = c_{f1} + c_{f2} \frac{n_e}{1000} + c_{f3} \epsilon^2 $$

Where $c_{f_i} (i=1-3)$ – tuning parameters, estimated as $[12]$, $c_{f1}=48; c_{f2}=48; c_{f3}=0.4$.

**Indicated and Effective (Brake) powers / Navedene i efektivne konjske snage**
The Powers (kW): Indicated $P_i$ and effective $P_w$ of one

$$P_i = \int p \, dV; \quad P_w = \int p \, dV; \quad P_{fr} = P_i - P_{bl}$$

The total engine effective power, $P_{em}(kW)$ is defined as below:

$$P_{em} = P_{w} \cdot n_{em}$$

**2.3. Turbocharger Model / Model turbopuhala**
In this research, the turbocharger includes a radial compressor and an axial turbine which mounted on a common shaft. The turbocharger model has two sub-models: turbine model compressor model.

**Turbine model / Model turbine**
Turbine efficiency, $\eta_t$, is a function of blade speed ratio (BSR) [1] as below:

$$\eta_t = \eta_{t,\max} - c_1 \left( \text{BSR} - \text{BSR}_{\text{opt}} \right)^2$$

Where, $\eta_{t,\max}$ - maximum of the turbine efficiency; $\text{BSR}_{\text{opt}}$ - optimum of BSR; $c_1$ - tuning parameters. The turbine mass flow, $\dot{m}_t(kg/s)$, depends on the area of the nozzle and ratio of pressures as below [10]:

$$\dot{m}_t = \frac{p_{in}}{R_k T_1} A_1 f(\pi_t)$$

Where, $R_k$ - gas constant of exhaust gas $(J/kg.K); p_{in}, T_1$ - pressure and temperature at the inlet turbine; $A_1 (m^2)$ - nozzle cross-sectional area; $f(\pi_t)$ - function of the turbine pressure ratio $\pi_t$, which can be modelled as following expressions below [18]:

$$\Pi = \max \left\{ \pi_t \left( \frac{2}{\gamma -1} \right) ^{\frac{\gamma -1}{\gamma}} \right\}$$

$$f(\pi) = \sqrt{\frac{2 \gamma -2}{\gamma -1}} \left( \Pi^2 - \Pi \right)$$

With $\gamma(\cdot)$ is the specific heat capacity ratio of exhaust gas. Turbine power, $P_t$(kW), can be calculated by the isentropic enthalpy drop in the turbine stage, given as below [19]

$$P_t = \dot{m}_t \cdot c_{pe}(T_1 - T_2) = \dot{m}_t \cdot c_{pe} T_1 \left( 1 - \frac{P_{in}}{P_{1}} \right)$$

Where, subscripts 3, 4 refer to the inlet and outlet of the turbine; $c_{pe}(J/kg.K)$ - specific heat value at constant pressure.

**Compressor model / Model kompresora**
The compressor efficiency $\eta_c$ can be estimated as the quadratic function [20], given as below.

$$\eta_c = \eta_{c,\max} - \chi^2 \eta_c ; \quad \chi^2 \left[ \dot{m}_c - \dot{m}_{c,\max} - \pi_c - \pi_{c,\max} \right]$$

Where $\pi_c(\cdot)$ - pressure ratio; $\eta_{c,\max}; \pi_{c,\max}$ - maximum of efficiency and pressure ratio which are taken from the compressor map; $Q_c$ - tuning parameter. The compressor mass flow model was based on two dimensionless parameters: flow coefficient $\phi_t$ and energy transfer coefficient $\psi_t$.

The energy transfer coefficient $\psi_t$ [21] is:

$$\psi_t = 2c_{ps} T_1 \left( \frac{\gamma-1}{\gamma} \right) - 1)$$

Where $\alpha_1$ - (rad/s) - turbine speed; $t$ - compressor radius; $c_{ps}(J/kg.K) -$ specific heat capacity at constant pressure; $\gamma$ - specific heat capacity ratio of the inlet air; $T_1$ - inlet air temperature.

The flow coefficient $\phi_t$ [21] is:

$$\phi_t = \sqrt{\left(1-c_{ps} \left( \psi_{c} - \psi_{c,\max} \right) \right)} + c_{s2}$$

Where, $c_{ps}, c_{ps,\max}, c_{s1}, c_{s2}$ - tuning parameters.

From equations $\phi_t$ and $\psi_t$, the compressor flow equation is $\dot{m}_c$ given below [21]:

$$\dot{m}_c = \frac{p_{in} \cdot \pi_1 \cdot \pi_{c,\max} \cdot \psi_{c,\max} \cdot \phi_t}{R_k T_1}$$

The compressor power $P_c$ is calculated according to the expression below [1]:

$$P_c = \frac{1}{\eta_c} \dot{m}_c T_1 \left( \frac{\gamma-1}{\gamma} \right) - 1)$$

The balance between the turbine and the compressor / Ravnovazni izmedju turbine i kompresora
At the steady condition with a speed of turbocharger $n_{im}$(rev/min), the balancing between the turbine and the compressor powers is given in the following expression:

$$J_{im} \frac{d\omega}{dt} = \text{Moment} = \frac{\text{Power}}{\omega} = \left( P_{ei} - P_{fr} \right) \frac{30}{\pi n_{im}}$$

Where, $J_{im} (kg.m^2)$ - turbocharger inertia moment; $\eta_m(\cdot)$ - friction coefficient.

**2.4. Intake manifold and exhaust manifold model / Model ulazne i izlazne mlaznice**
The intake air and exhaust gas pressures are defined in the following equations:

$$\frac{d}{dt} P_{in} = \frac{R_k T_{in}}{V_{in}} \left( \dot{m}_c - \dot{m}_a \right)$$

$$\frac{d}{dt} P_{ex} = \frac{R_k T_{ex}}{V_{ex}} \left( \dot{m}_e - \dot{m}_c \right)$$

Where, subscripts $in, ex$ denote the intake and exhaust manifolds; $R_{in}(J/kg.K), R_{ex}(J/kg.K)$ - gas constant of the intake and exhaust gas.

**2.5. Algorithm / Algoritam**
Parameter estimation / Provjera parametara
There were two types of parameters, fixing and tuning parameters.
- Fixing parameters include MDE and TC structure parameters: cylinder bore $b$, stroke $s$, compression ratio $\epsilon$, compressor diameter $D_c$, turbine diameter $D_t$, inertia moment $J_{im}$ and ambient conditions: temperature and pressure, low heat value $Q_v$, Cetan number CN.
- Tuning parameters include volumetric efficiency coefficients $c_{v1}, c_{v2}, c_{v3}$; friction coefficients $c_f$; and combustion factors $a_p, a_d, \phi_p, \phi_d, m_p, m_d$.
- The output parameters: Effective power, mean pressure, and maximum pressure.
- The errors: To evaluate the accuracy, the relative deviation between the simulation parameters and measured parameters from test records was calculated, and loops ensured these errors were $\leq 5\%$.

\[ \varepsilon_r = \frac{|x_{\text{mea}} - x_{\text{sim}}|}{x_{\text{mea}}} \times 100\% \]  

With $x_{\text{sim}}$ – simulation parameters; $x_{\text{mea}}$ – measured parameters from test records.

The algorithm chart. The algorithm was built based on the above-mentioned equations (Fig. 2).

### 3. CASE STUDY / Studija slučaja

#### 3.1. The object for theoretical and experimental study / Predmet teoretske i eksperimentalne studije

The object for simulation is the diesel engine 8MAK43 installed on MV PhucHung of GLS company, Viet Nam. Some technical engine parameters are shown in Table 1.

| Parameters                                | Values   |
|-------------------------------------------|----------|
| Bore x stroke (mm)                        | 430 x 610 |
| Nominal speed (rev/min)                   | 500      |
| Number of cylinders                       | 8        |
| Max. pressure, bar                        | 193      |
| Mean pressure, bar                        | 26,4     |
| Specific fuel consumption (g/kWh)         | 186,8    |
| Nominal brake power (kW)                  | 7200     |

#### 3.2. Simulink model / Simulink model

The model was made in MatLab Simulink and shown in Fig. 3. In the figure were described the following:

- **Input controls**: $u_t(u_t)$ was the signal of the nozzle cross-sectional control; $m_f cycle (m_f,cycle)$ was the quality of fuel per cycle; $n_e(n_e)$ - the engine speed.

- **Function blocks**: Turbocharger block - TB; Cylinder block - MDE; Intake block; Exhaust block.

- **Output results**: the indicated pressure ($p_{cyl}$) which was used to calculate the engine performances (brake power, specific fuel consumption...)

#### 3.3. Simulation of a new engine at the rated operation mode / Simulacija novoga stroja prema prilagođenom modelu

The program simulated a new engine at the rated operation mode (100% load and 500 rpm).

**Interface of simulation / Sučelje simulacije**

The interface of this simulation was shown in Fig. 4. There were described: the input controls; Indicated cylinder pressure and key results.
The comparison of the simulation results with the MDE data in accordance with the technical documents / Usporedba rezultata simulacije s podacima o brodskome stroju u skladu s tehničkom dokumentacijom
At the LI=(25; 50;75;100;110)%, the simulation results and the reference data that are given in the engine technical documents [22] were compared and shown in Fig. 5 ÷ Fig. 8.

3.4. Simulation at the practical operation mode / Simulacija pri praktičnome modelu djelovanja
In accordance with the MDE book log, the engine 8MAK43 operated at the speed 412 rpm and LI range of (60% ÷68%). The regime with the LI 65% and speed 412 rpm was regularly used. Therefore, this mode was simulated to find the optimization point of the nozzle cross-sectional area for improving the engine's performances.

The interface and main results of the simulation was presented in Fig. 9.
Figure 5 Intake pressure comparison
Slika 5. Usporedba ulaznoga tlaka

Figure 6 Engine brake power comparison
Slika 6. Usporedba konjskih snaga stroja

Figure 7 Exhaust temperature comparison
Slika 7. Usporedba ispušne temperature

Figure 8 SFOC comparison
Slika 8. Usporedba specifične potrošnje goriva

Figure 9 Simulation at the practical operation mode (65% load, 412 rpm)
Slika 9. Simulacija pri praktičnom modelu djelovanja (65% opterećenja, 412 okretaja u minuti)
Prediction of the influence of the nozzle cross-sectional area / Predviđanje utjecaja prostora presjeka mlaznice

The variation of engine performance parameters (turbine speed \( n_t \), SFOC, exhaust temperature \( T_e \), brake power \( P_w \)) via the nozzle cross-sectional area (\( %A_t \)) were modelled by regressive models (equations) (30), (31), (32), and (33) with the confidence testing in accordance with the statistic F criterion: \( F(\beta=0.99; n_1; n_2) \), where, \( \beta=0.99 \) is confidence; \( n_1 \) and \( n_2 \) are freedom degrees [24].

Regressive model of the turbine speed \( n_t \) (\( A_t \)) was received with the 99%-confidence:

\[
n_t = -13.51A_t^2 + 2471A_t - 9944 \tag{30}
\]

Regressive model of the exhaust temperature \( T_e \) (\( A_t \)) was received with the 99%-confidence:

\[
T_e = 0.1968A_t^2 - 36.01A_t + 2226 \tag{31}
\]

Regressive model of the SFOC (\( A_t \)) was received with the 99%-confidence:

\[
SFOC = 0.3156A_t^2 - 57.71A_t + 2867 \tag{32}
\]

Regressive model of the brake power \( P_w \) (\( A_t \)) was received with the 99%-confidence:

\[
P_w = -3.516A_t^2 + 642.4A_t - 26360 \tag{33}
\]

From the simulation, the optimization point of the nozzle cross-sectional area (\( %A_t \)) was found out at the practical operation mode (LI 65% and \( n= 412 \) rpm). With the \( A_t=91\% \), the turbine speed \( n_t \) and brake power \( P_w \) reached the maximum (\( n_t=13545 \) rpm, \( P_w=2982 \) kW); at the same time, the exhaust temperature and specific fuel consumption \( g_e \) were minimum (\( T_e=5790K, g_e=229 \) g/kW.h).

4. EXPERIMENTAL STUDYING / Eksperimentalno proučavanje

According to the above simulation results the experiment was carried out. The nozzle of the TC was removed and then changed it sizes as Fig.14. The remaining percentage of nozzle cross-sectional area after narrowing is defined

\[
%A_t = \frac{13}{14.25} = 91\%
\]
The comparison results / Rezultati usporedbe
The MDE with improved nozzle has well operated. At the same regime operation (LI 60%÷68%, n = 412 rpm), the main output parameters before and after improving were shown in Fig.15 ÷ Fig.17.

Table 2 compares the measured output parameters at the most regularly mode (load 65% and n= 412 rpm), with before and after narrowed the nozzle to evaluate the effectiveness of improvement.

Table 2 Measured MDE parameters at LI= 65% and n= 412 rpm, before and after narrowing the nozzle

| Parameter                        | A =100% | A =91% | Change                  |
|----------------------------------|---------|--------|-------------------------|
| Max pressure (bar)               | 114     | 126,9  | Δp_z =13; (+11.3%)      |
| Turbine speed (rpm)              | 11165   | 13341  | Δn =2176; (+19.5%)      |
| Intake pressure (bar)            | 1.83    | 2.29   | Δp_im=0.46; (+25.1%)    |
| Exhaust temperature (°C)         | 379.1   | 343.6  | ΔT_e=35.5 0C; (-9.4%)   |

The engine's performances have significantly improved, the maximum combustion pressure increased by 13 bar (+11.3%), turbine speed increased by 2176 rpm (+19.5%), the intake pressure increased by 0.46 bar (+25.5%), and the exhaust temperature decreased by 35.5 °C (-9.4%).

5. CONCLUSION / Zaključak
In this research, there were synthesized the mathematical fundaments and made the simulation software in MatLab / Simulink for studying the working cycles of the turbocharged MDE. By using the made simulation software, the affections of nozzle cross-sectional area to the engine, were evaluated and an optimum point was determined to improve the engine brake power, SFOC, and exhaust temperature.

The experimental study was carried out on a marine diesel engine on board. At the practical operation with the LI=65% and speed n = 412 rpm, the nozzle cross-sectional area was narrowed with 91% of maximum area, same as in the simulation study, the results were positive. The turbine speed improved by 19.5%, at the same time, the intake pressure increases by 25.1%, and exhaust temperature reduced by 9,4%. Therefore, this method may be useful for improving old turbocharged engines with a fixed nozzle. However, it depends on the kinds of engine and the time of use, the nozzle cross-sectional area would be adjusted suitably.

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