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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DOI 10.17818/NM/2021/2.1
UDK 629.5:621.436
621.125
Original scientific paper / *Izvorni znanstveni rad*
Paper accepted / *Rukopis primljen:* 11. 2. 2020.

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.

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

Sažetak

Nakon dugoga razdoblja uporabe, neki važni tehnički parametri glavnoga dizelskoga brodskog motora postupno su postali lošiji, kao što su: brzina turbine, ulazni pritisak, ispušna temperatura, snaga stroja i specifična potrošnja goriva. Ovaj članak proučava svojstva prostora presjeka turbo mlaznice (AT) na brodski dizelski stroj i predstavlja metodu AT prilagodbe, kako bi se poboljšale izvedbe brodskoga dizelskog stroja. Matematički model motora postavljen je prema postojećoj konstrukciji stroja i teoriji rada radnoga ciklusa dizelskoga stroja, a simulaciju je programirao Matalab/Simulink. Preciznost simulacijskoga modela evaluira se na temelju usporedbe rezultata simulacije i eksperimentalnih podataka pomorskoga dizelskog motora. Rezultati preciznosti testiranja bili su prihvatljivi (unutar 5%). Utjecaji AT na radne parametre motora i traženje optimizacijske točke provedeni su korištenjem simulacijskim programom u studiji. Pretpostavljena optimizacijska točka štrcaljke upotrijebljena je kako bi poboljšala izvedbu brodskoga motora. Integracija simulacije i studije eksperimenta pokazala je djelotvornost pri praktičnoj primjeni u polju brodskoga dizelskog stroja.

KLJUČNE RIJEČI
glavni dizelski stroj
brzina turbine
prostor presjeka mlaznice

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 Rahnke [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

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 ($\dot{m}_{ei}$) and exhaust air mass flow rate ($\dot{m}_{eo}$). The principle scheme of the MDE –TC relationship was shown in Fig.1.

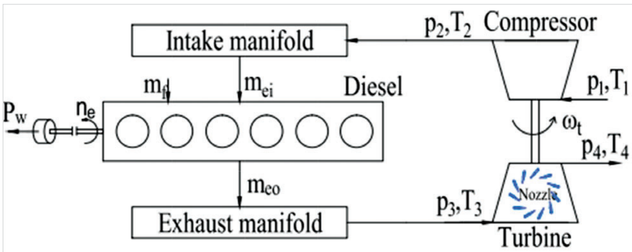


Figure 1 Principle scheme of MDE and turbocharger relationship

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

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

$$\dot{m}_{ei}=\eta_v \frac{p_s V_d n_e}{60zR_a T_s} \tag{1}$$

Where, n_e (rpm) – engine rotatory speed; V_d (m^3) – total swept volume; η_v (-) – volumetric efficiency; z – diesel stroke: z=1 for two-stroke, z=2 for four-stroke engine; R_a(kJ/kg.K) - gas constant; p_s(kN/m^2) – intake air pressure; T_s(^{0}K) – intake air temperature;

The total intake mass flow rate ($\dot{m}_{eo}$, kg/s) in to the cylinder is given as below:

$$\dot{m}_{eo}=\dot{m}_{ei}+\dot{m}_f \tag{2}$$

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

$$\dot{m}_f=\frac{m_{f,cycle} n_e n_{cyl}}{60z} \tag{3}$$

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

2.2. Cylinder model / *Model cilindra*

Intake process / *Proces unosa*

The intake process quality can be evaluated by the volumetric efficiency η_v in the following expression [10]:

$$\eta_v = c_{v1}\sqrt{p_s} + c_{v2}\sqrt{n_e} + c_{v3} \tag{4}$$

Where, c_{v1}, c_{v2}, c_{v3} 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_{v1}=0.006651; c_{v2}=0.7429; c_{v3}=-0.4093.

Combustion process / *Proces izgaranja*

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

$$dT=\frac{1}{mR}(pdV+Vdp) \tag{5}$$

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]:

$$\frac{dp}{d\varphi}=-\gamma\frac{p}{V}\frac{dV}{d\varphi}+\frac{(\gamma-1)}{V}\left(\frac{dQ_{in}}{d\varphi}-\frac{dQ_w}{d\varphi}\right) \tag{6}$$

Where, γ(-) is the specific heat capacity ratio of the mixed gas;

The heat release equation $dQ_{in}/d\varphi$ is shown as bellow:

$$\frac{dQ_{in}}{d\varphi}=Q_{in}\frac{dx_f}{d\varphi} \tag{7}$$

Where, Q_{in} is the total input heat in fuel combustion, $Q_{in}=m_f Q_H$ (Q_H: Low heating value of the used fuel); x_f - burn fraction.

Based on Wiebe's equation [13] [14], the differential equation dx_f/dt is:

$$\frac{dx_f}{dt}=\beta\frac{dx_{f,p}}{dt}+(1-\beta)\frac{dx_{f,d}}{dt} \tag{8}$$

$$\frac{dx_{f,p}}{dt}=a_p\frac{m_p}{\varphi_p}\left(\frac{\varphi-\varphi_s}{\varphi_p}\right)^{m_p-1}\exp\left(-a_p\left(\frac{\varphi-\varphi_s}{\varphi_p}\right)^{m_p}\right) \tag{9}$$

$$\frac{dx_{f,d}}{dt}=a_d\frac{m_d}{\varphi_d}\left(\frac{\varphi-\varphi_s}{\varphi_d}\right)^{m_d-1}\exp\left(-a_d\left(\frac{\varphi-\varphi_s}{\varphi_d}\right)^{m_d}\right) \tag{10}$$

Where a_p, a_d, m_p, m_d are shape factors; φ_p, φ_d are the duration of premixed and diffusion phases; φ_s is the start of combustion. Combustion factors (a_p, a_d, m_p, m_d, φ_p, φ_d) was estimated as [15], $a_p=a_d$=6.9; $\varphi_p=70^0$; m_p=3; m_d=0.5, and φ_d was calculated by least square regression method as [15], $\varphi_d=c_1p_e^2+c_2p_e+c_3p_e$, c_1=-0.0002931; c_2=0.05108; c_3=-1.313, with coefficient of determination R-squared=0.9956.

The heat transfer $dQ_w/d\varphi$ from the gases to the combustion surfaces is as bellow [16]:

$$\frac{dQ_w}{d\varphi}=\frac{h_t(\varphi)A_w(\varphi)(T-T_w)}{2\pi n_e} \tag{11}$$

Where, $A_w(\varphi)$ is the area of surfaces (head, cylinder, piston); T_w(^{0}K) is the average temperature of surfaces; $h_t(\varphi)$ is the heat transfer coefficient, estimated by Woschni [17] and corrected by Heywood [12], given as below:

$$h_t=3.26p^{0.8}U^{0.8}b^{-0.2}T^{-0.55} \tag{12}$$

Where, p(N/m²) - 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_{fp}(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:

$$p_{fp} = c_{f1} + c_{f2}\frac{n_e}{1000} + c_{f3}\overline{S}_p^2 \tag{13}$$

Where c_{fi} (i=1÷3) – tuning parameters, estimated as [12], c_{f1}=75, 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

$$P_i=\int p_i dV;\ P_w=\int p_e dV;\ p_e=p_i-p_{fm} \tag{14}$$

The total engine effective power, P_{Ew} (kW) is defined as bellow:

$$P_{Ew}=P_w.n_{cyl} \tag{15}$$

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,η_t, is a function of blade speed ratio (BSR) [1] as below:

$$\eta_t = \eta_{t,max} - c_t\left(BSR - BSR_{opt}\right)^2 \tag{16}$$

Where, $\eta_{t,max}$ - maximum of the turbine efficiency; BSR_{opt} - optimum of BSR ; c_t - tunning parameters.

The turbine mass flow, $\dot{m}_t$(kg/s), depends on the area of the nozzle and ratio of pressures as below [10]:

$$\dot{m}_t = \frac{p_3}{\sqrt{R_e T_3}} A_T\, f(\pi_t) \tag{17}$$

Where, R_e - gas constant of exhaust gas (J/kg.K); p_3, T_3 - pressure and temperature at the inlet turbine; A_T (m²) - nozzle cross-sectional area; f(π_t) - function of the turbine pressure ratio π_t , which can be modelled as following expressions below [18]:

$$\Pi = \max\left(\pi_t, \left(\frac{2}{\gamma_e+1}\right)^{\frac{\gamma_e}{\gamma_e-1}}\right) \tag{18}$$

$$f(\pi_t) = \sqrt{\frac{2\gamma_e}{\gamma_e+1}\left(\Pi^{\frac{2}{\gamma_e}} - \Pi^{\frac{\gamma_e+1}{\gamma_e}}\right)} \tag{19}$$

With γ_e(-) 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]

$$P_t=\eta_t\dot{m}_t c_{pe}(T_3-T_4)=\eta_t\dot{m}_t c_{pe}T_3\left(1-\left(\frac{p_4}{p_3}\right)^{\frac{\gamma_e-1}{\gamma_e}}\right) \tag{20}$$

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 η_c can be estimated as the quadratic function [20], given as below.

$$\eta_c = \eta_{c,max} - \chi^T Q_c \chi;\ \chi^T = [\dot{m}_c - \dot{m}_{c,max}, \pi_c - \pi_{c,max}] \tag{21}$$

Where π_c(-) - 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 Φ_c and energy transfer coefficient Ψ_c. The energy transfer coefficient Ψ_c [21] is:

$$\psi_c = \frac{2c_{pa}T_1\left(\pi_c^{1-1/\gamma_a} - 1\right)}{\omega_t^2 r_c^2} \tag{22}$$

Where ω_t (rad/s) – turbine speed, r_c (m) – compressor radius; c_{pa}(J/kg.K) – specific heat capacity at constant pressure; γ_a– specific heat capacity ratio of the inlet air; T_1 – inlet air temperature.

The flow coefficient Φ_c [21] is:

$$\Phi_c = \sqrt{\frac{1-c_{\psi1}\left(\psi_c - c_{\psi2}\right)^2}{c_{\phi1}}} + c_{\phi2} \tag{23}$$

Where, $c_{\psi1}$, $c_{\psi2}$, $c_{\phi1}$, $c_{\phi2}$ – tunning parameters.

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

$$\dot{m}_c = \frac{p_1}{R_a T_1}\pi r_c^3 \omega_t \Phi_c \tag{24}$$

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

$$P_c = \frac{1}{\eta_c}\dot{m}_c c_{pa} T_1\left(\pi_c^{\frac{\gamma_a-1}{\gamma_a}} - 1\right) \tag{25}$$

The balance between the turbine and the compressor / *Ravnoteža između turbine i kompresora*

At the steady condition with a speed of turbocharger n_t(rev/min), the balancing between the turbine and the compressor powers is given in the following expression:

$$J_t\frac{d\omega_t}{dt}=\text{Moment}=\frac{\text{Power}}{\omega}=(P_t\eta_m-P_c)/\frac{30}{\pi n_t} \tag{26}$$

Where, J_t (kg.m²) - turbocharger inertia moment; η_m (-) - 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:

$$\frac{d}{dt}p_{im} = \frac{R_a T_{im}}{V_{im}}\left(\dot{m}_c - \dot{m}_{ei}\right) \tag{27}$$

$$\frac{d}{dt}p_{em} = \frac{R_e T_{em}}{V_{em}}\left(\dot{m}_{eo} - \dot{m}_t\right) \tag{28}$$

Where, subscripts *im, em* denote the intake and exhaust manifolds; R_a(J/kg.K), R_e(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 ε, compressor diameter D_c, turbine diameter D_t, inertia moment J_t..., and ambient conditions: temperature and pressure, low heat value Q_H, Cetan number CN.

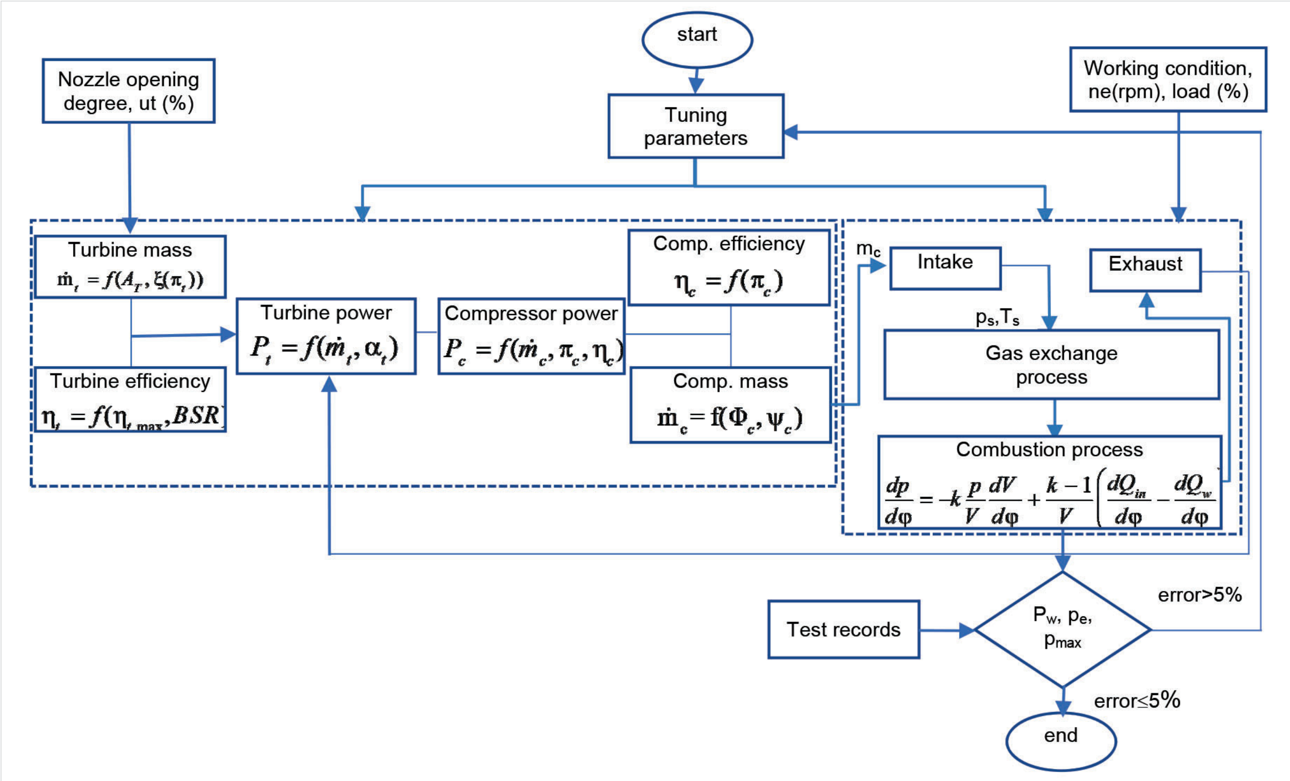


Figure 2 Algorithm of MDE and TC relationship

Slika 2. Algoritam brodskoga dizelskog stroja i odnos testnih podataka

- Tuning parameters include volumetric efficiency coefficients c_{v1},c_{v2}, c_{v3}; friction coefficients c_{fi}; and combustion factors a_p, a_d, φ_p, φ_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 ≤5%

$$\varepsilon_e = \frac{|x_{sim} - x_{mea}|}{x_{mea}}.100\% \tag{29}$$

With x_{sim} – simulation parameters; x_{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.

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; mf cyc ($m_{f,cycle}$) 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 prilagođenome 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.

Table 1 The fundamental parameters – Input of the main engine 8 MAK43 for simulation

Tablica 1. Fundamentalni parametri – unos glavnoga stroja 8MAK43 za simulaciju

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

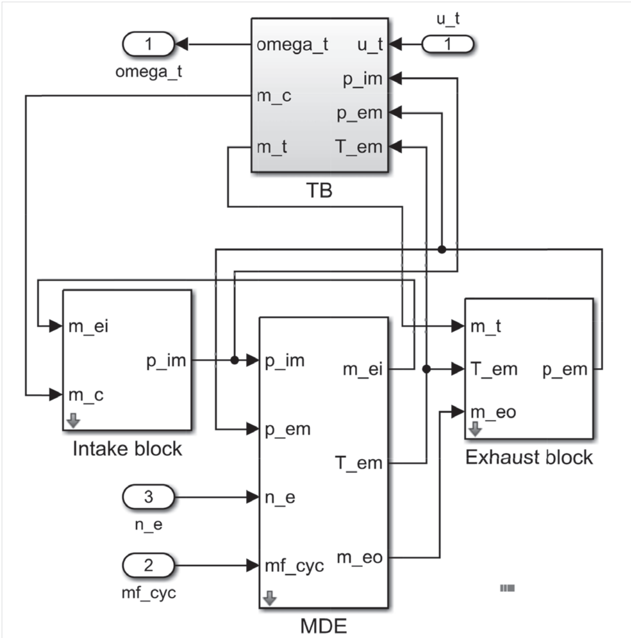


Figure 3 Simulink model of MDE and TB

Slika 3. Simulink model brodskoga dizelskog motora i bloka turbopuhala

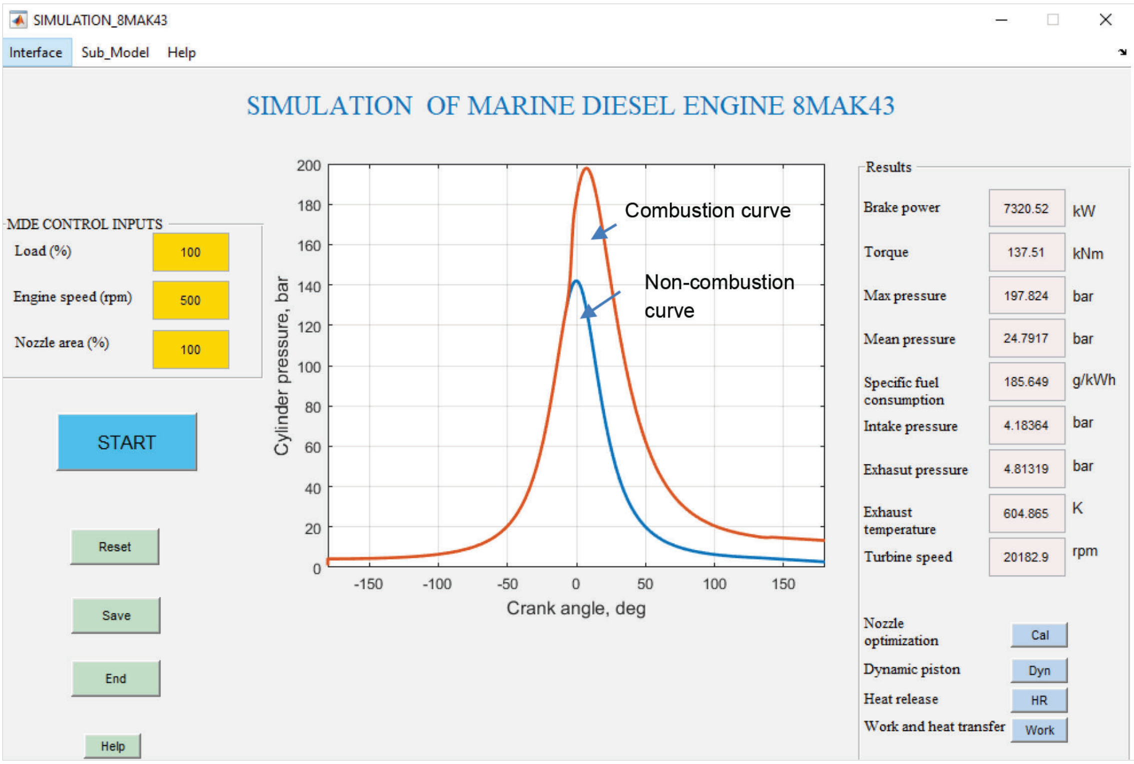


Figure 4 Simulation at the rated operation (100% load, 500 rpm)

Slika 4. Simulacija prema prilagođenom djelovanju

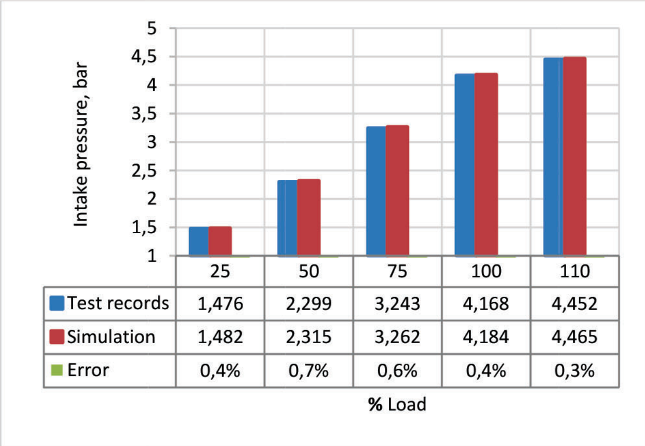
The comparison of the simulation results with the MDE data in accordance with the technical documents / *Usporedba rezultata simulacije s podacima o brodskome stroju u skladu s tehničkom dokumentacijom*

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*

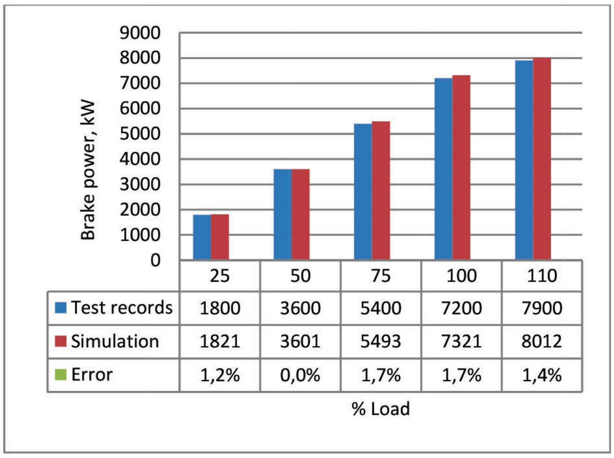
In accordance with the MDE book log, the engine 8MAK43 operated at the speed 412 rpm and LI range of (60% ÷68%). The regime with the LI 65% and speed 412 rpm was regularly used. Therefore, this mode was simulated to find the optimization point of the nozzle cross-sectional area for improving the engine's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

The interface and main results of the simulation was presented in Fig. 9.



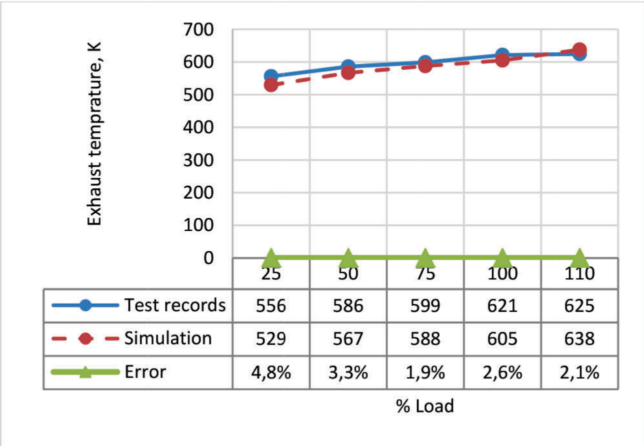
	25	50	75	100	110
Test records	1,476	2,299	3,243	4,168	4,452
Simulation	1,482	2,315	3,262	4,184	4,465
Error	0,4%	0,7%	0,6%	0,4%	0,3%

Figure 5 Intake pressure comparison
Slika 5. Usporedba ulaznoga tlaka



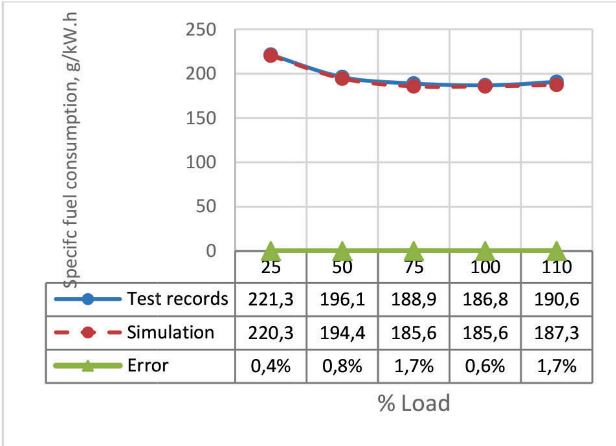
	25	50	75	100	110
Test records	1800	3600	5400	7200	7900
Simulation	1821	3601	5493	7321	8012
Error	1,2%	0,0%	1,7%	1,7%	1,4%

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



	25	50	75	100	110
Test records	556	586	599	621	625
Simulation	529	567	588	605	638
Error	4,8%	3,3%	1,9%	2,6%	2,1%

Figure 7 Exhaust temperature comparison
Slika 7. Usporedba ispušne temperature



	25	50	75	100	110
Test records	221,3	196,1	188,9	186,8	190,6
Simulation	220,3	194,4	185,6	185,6	187,3
Error	0,4%	0,8%	1,7%	0,6%	1,7%

Figure 8 SFOC comparison
Slika 8. Usporedba specifične potrošnje goriva

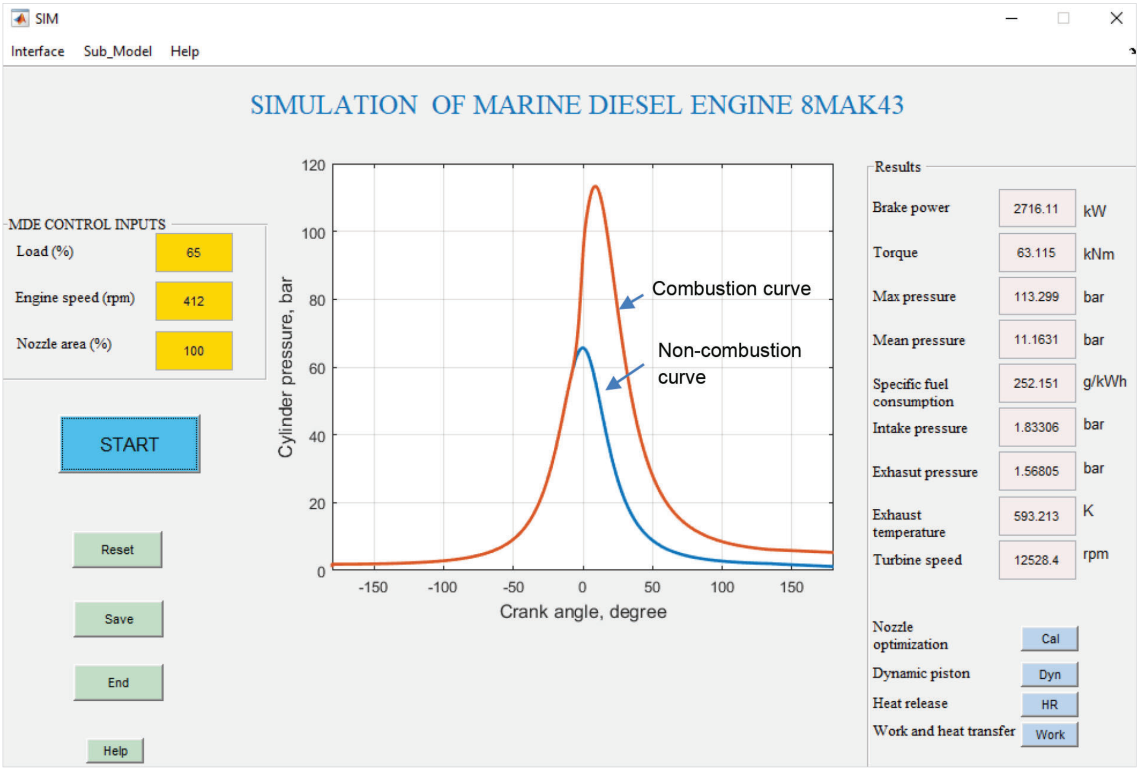


Figure 9 Simulation at the practical operation mode (65% load, 412 rpm)
Slika 9. Simulacija pri praktičnome modelu djelovanja (65% opterećenja, 412 okretaja u minuti)

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:

$$n_t = -13.51A_T^2 + 2471A_T - 9944 \tag{30}$$

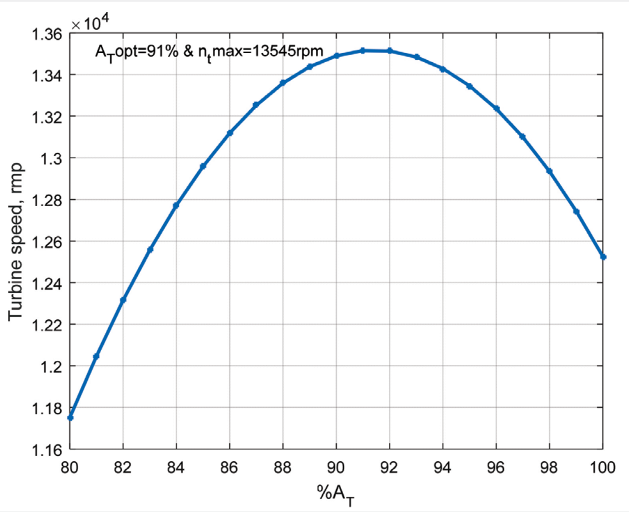


Figure 10 Regressive analysis of the simulated results of the relation between $n_t - A_T$

Slika 10. Regresivna analiza simuliranih rezultata odnosa $n_t - A_T$

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

$$T_e = 0.1968A_T^2\text{-}36.01A_T\text{+}2226 \tag{31}$$

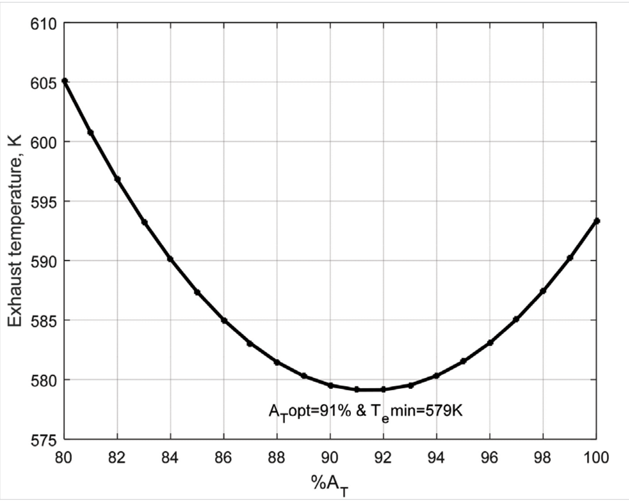


Figure 11 Regressive analysis of the simulated results of the relation between $T_e - A_T$

Slika 11. Regresivna analiza simuliranih rezultata odnosa $T_e - A_T$

SFOC is written by the symbol g_e. The SFOC model SFOC (A_T) was received with the 99%-confidence:

$$\text{SFOC} = 0.3156A_T^2\text{-}57.71A_T + 2867 \tag{32}$$

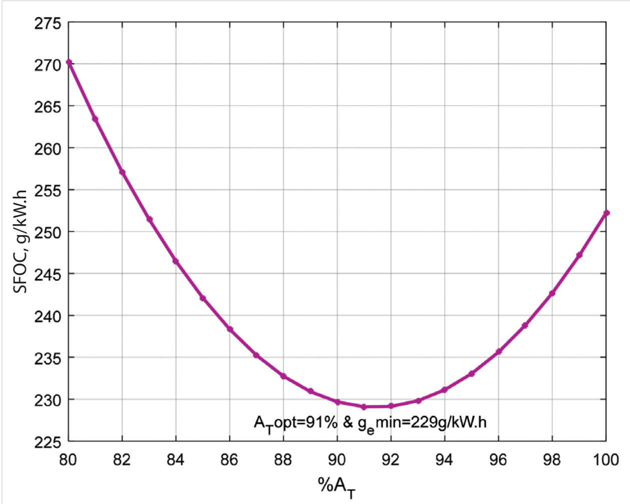


Figure12 Regressive analysis of the simulated results of the relation between $g_e - A_T$

Slika 12. Regresivna analiza simuliranih rezultata odnosa $g_e - A_T$

Engine brake power model P_{Ew} (A_T) was received with the 99%-confidence:

$$P_{Ew} = -3.516A_T^2 + 642.4A_T - 26360 \tag{33}$$

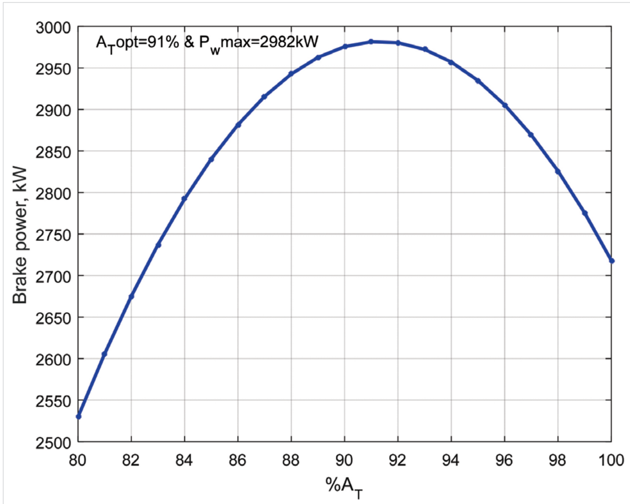


Figure 13 Regressive analysis of the simulated results of the relation between $P_{Ew} - A_T$

Slika13. Regresivna analiza simuliranih rezultata odnosa $P_{EW} - A_T$

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_{Ew}=2982 kW); at the same time, the exhaust temperature and specific fuel consumption g_e were minimum (T_w=579[0]K, 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

$$\%A_T = \frac{13}{14,25}.100\% \approx 91\%$$

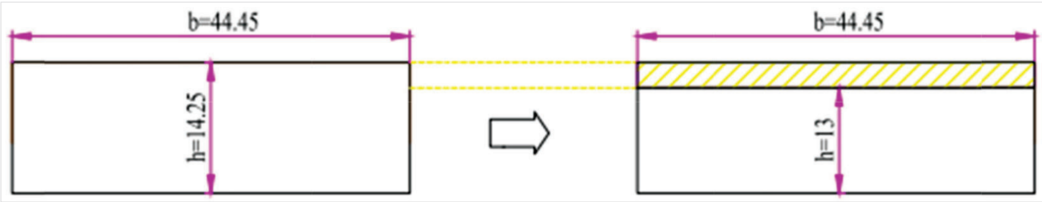


Figure 14 Narrowed nozzle area of the turbine
Slika 14. Suženi prostor mlaznice turbine

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.

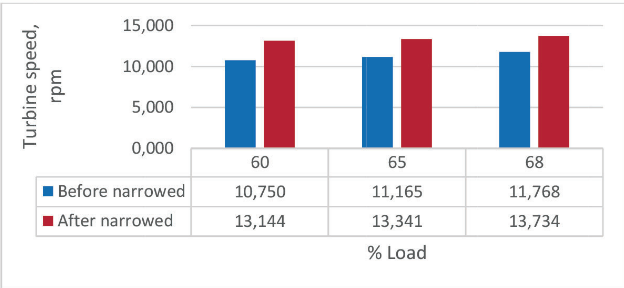


Figure15 Turbine speed
Slika 15. Brzina turbine

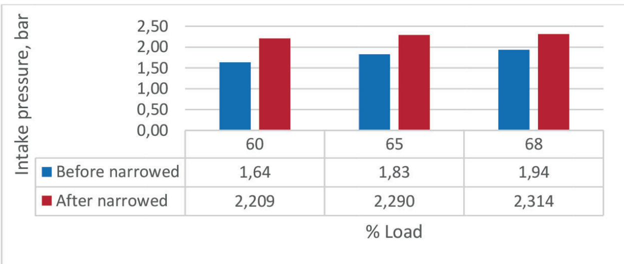


Figure16 Intake pressure
Slika 16. Ulazni pritisak

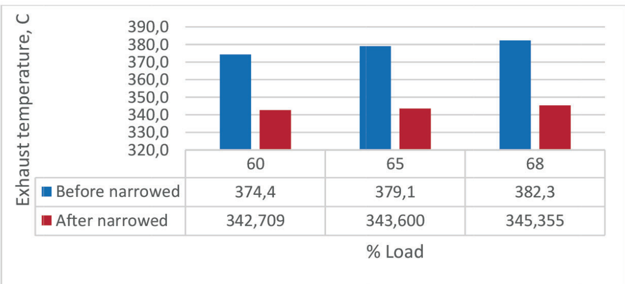


Figure17 Exhaust temperature
Slika 17. Temperatura ispuha

Table 2 compares the measured output parameters at the most regularly mode (load 65% and speed 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
Tablica 2. Izmjereni parametri brodskoga dizelskog stroja pri LI = 65% i n = 412 okretaja u minuti, prije i poslije sužavanja mlaznice

Parameter	A_T=100%	A_T=91%	Change
Max pressure (bar)	114	126,9	Δp_z =13; (+11.3%)
Turbine speed (rpm)	11165	13341	Δn_t=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 °C; (-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.

Acknowledgment / *Zahvala*

The authors would like to thank the MV Phuc Hung of GLS Shipping Co. LTD Viet Nam for supporting us to complete this work.

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