UDC 621.436:629.5

Tomislav SENČIĆ[1]
Nikola RAČIĆ[2]
Bernard FRANKOVIĆ[1]

Influence of Low-Speed Marine Diesel Engine Settings on Waste Heat Availability

Original scientific paper

The low-speed marine diesel engine is the most effective of all the ship propulsion systems. On every ship there is a need for thermal energy besides mechanical power to drive the propeller. It is possible to install a heat exchanger in the exhaust system that makes use of waste heat of the exhaust gasses of the diesel engine. Such a combined mechanical and thermal energy generation is called cogeneration. Modern engines allow the variation of the fuel injection timing and the variation of the exhaust valve timing, which results in a great usage flexibility.

In the current work a computer simulation model of a low-speed marine diesel engine is presented. The exhaust gas heat energy available to power a heat exchanger was calculated. The time of the beginning of fuel injection and the time of the opening of the exhaust valve was varied. It was analyzed how these parameters influence the power, the fuel consumption, the engine efficiency, the exhaust gas temperature, the heat energy available in the exhaust gasses, the overall efficiency of the cogeneration system, and the power to heat ratio.

Keywords: *cogeneration, computer simulation model, low-speed marine diesel engine.*

Utjecaj regulacijskih parametara sporokretnog brodskog motora na otpadnu toplinu

Izvorni znanstveni rad

Sporohodni dizelski motori su najefikasniji pogonski sustav broda. Osim za mehaničkom snagom za pogon vijka, na brodu postoji i velika potreba za toplinskom energijom. U ispušni sustav moguće je ugraditi utilizacijski izmjenjivač topline koji iskorištava osjetnu toplinu ispušnih plinova. Takva kombinirana proizvodnja mehaničke i toplinske energije naziva se kogeneracija. Suvremeni motori omogućuju promjenu vremena ubrizgavanja te otvaranja ispušnog ventila što rezultira fleksibilnošću u eksploataciji.

U radu je prikazan računalno-simulacijski model sporokretnog brodskog motora. Izračunata je raspoloživa toplinska energija ispušnih plinova za pogon izmjenjivača topline. Variran je trenutak početka ubrizgavanja goriva te trenutak otvaranja ispušnog ventila. Analizirano je kako ovi parametri utječu na snagu, specifičnu potrošnju, iskoristivost motora, na temperaturu ispušnih plinova, na količinu topline u ispušnim plinovima koju je moguće iskoristiti, na ukupni stupanj djelovanja sustava te na omjer mehaničke i toplinske energije.

Ključne riječi: *kogeneracija, računalno-simulacijski model, sporokretni brodski motor.*

Authors' addresses (Adrese autora):
[1] *University of Rijeka, Faculty of Engineering,* Vukovarska 58, 51000 Rijeka, tsencic@riteh.hr
[2] *University of Split, Faculty of Maritime Studies*

Received (Primljeno): 2012-01-27
Accepted (Prihvaćeno): 2012-06-10
Open for discussion (Otvoreno za raspravu): 2013-12-31

Nomenclature

BDC bottom dead centre
COP coefficient of performance
c specific heat capacity, [J/kg K]
EVO exhaust valve open
h enthalpy, [J/kg K]
m mass, [kg]
P power, [W]
p pressure, [Pa]
PHR power to heat ratio
Q heat, [J]
R gas constant, [J/kg K]
SOI start of injection
T temperature, [K]
U internal energy, [J]
u specific internal energy, [J/kg]
V volume, [m^3]
v specific volume, [m^3/kg]
η efficiency
κ adiabatic exponent
λ excess air ratio
π pressure ratio
φ crank angle, [º]

Subscripts

3 before turbine
4 after turbine
b brake

bb	blow by
c	cylinder
cool	cooling
e	exhaust
f	fuel
g	gas
i	inlet
in	at inlet
out	at exit
T	turbine
p	constant pressure
us	useful
w	wall

Introduction

Ship transport is the most efficient mode of transport, especially if the fuel consumption per cargo mass is considered. Of all ship propulsion systems, the most efficient is the one with the low-speed marine diesel engine. There were periods when steam turbines were the first choice for large ships, but only because such powerful diesel engines were not available [1]. Modern marine engines achieve the fuel conversion efficiency of around 0.5, which means the lowest specific fuel consumption. Besides this, which is very important today, it means the lowest specific emission of greenhouse gasses. Advanced fuel injection systems have been developed, and they allow the variation of the start and the duration of fuel injection. On certain engine models, the hydraulic exhaust valve drive allows variable valve opening [1]. Besides mechanical energy for ship propulsion, on ships there is the need for heat energy. The amount of such energy depends on the ship type. Some examples of heat applications are accommodation heating, steam generation, or cargo heating. A great amount of heat is needed on oil tankers. In fact, crude oil at lower temperatures changes from liquid to a state similar to fat, so it is necessary to heat it up during transport. Before the arrival to the destination port, the crude oil temperature has to be raised to allow pumping of the fluid. Furthermore, another possible application of waste heat is to use it as the power for an absorption refrigeration system [2] or air conditioning device.

A tendency towards pollutant emissions and fuel consumption reduction can be recognized in the world. One of the many strategies to achieve this goal is cogeneration or CHP (Combined Heat and Power) [3]-[7]. The comparison between the efficiencies of a separate production of heat and power and of a cogeneration system is shown in Figure 1. Cogeneration can be performed with a steam turbine, with a gas turbine or with a piston internal combustion engine. A low-speed marine diesel engine uses a big part of the chemical energy stored in the fuel and converts about a half of it to mechanical work, and dissipates the rest in the form of waste heat. Moreover, it operates with a low equivalence ratio (regarding fuel/air ratio), which results in lower exhaust gas temperatures in comparison with other engine types. These facts make it a less suitable choice for cogeneration when compared with other systems. However, if it is considered that its main goal is ship propulsion and that the value of the mechanical energy is higher than that of the heat energy, that it uses cheap heavy fuel oil and that it is often already installed on existing vessels of certain type, the cogeneration with the diesel engine is an option that has to be considered.

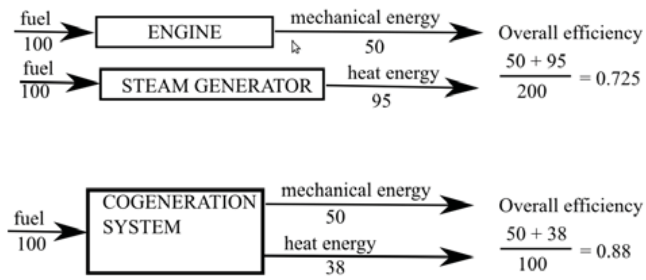

Figure 1 **Comparison of the overall efficiency for a classic system and for a cogeneration system**

Slika 1 **Usporedba ukupnog stupnja djelovanja za klasični i kogeneracijski sustav**

The marine engine settings are usually such that they allow the highest efficiency, the fulfilment of pollutant emission limits and ensuring of safe operation. However, modern engines with variable fuel injection systems and variable exhaust valve opening systems allow the change of these settings according to necessities. Hence, it is possible to optimize the operation for different goals: lower specific fuel consumption, higher maximum power or lower pollutant emission. If there is a waste heat utilization device installed onboard, there is another possible optimization goal: a larger quantity or a higher temperature of exhaust gasses, or in other words a higher amount of waste heat.

Among low-speed marine diesel engine manufacturers, MAN B&W offers the TES (Thermo Efficiency System) system, which allows the raising of the exhaust gas temperature so their heat could be used for steam production.

One of the tools that allow development and analysis of the processes in an engine is numerical simulation [8]-[12]. The mathematical model of engine components is based on conservation laws and on laws of thermodynamics and it consists of differential equations. Modern computers allow fast solution of such systems and a thermodynamic analysis of the influence of parameters on engine performances.

1 Mathematical model

The engine mathematical model is based on the first law of thermodynamics and on energy and mass conservation laws. The engine processes are described by nonlinear ordinary differential equations. The slow-speed marine engine system can be split in various elements, Figure 2, which can be analyzed separately or all together as a system. The main system elements are: engine cylinders, exhaust receiver, scavenging receiver, air cooler, turbine, compressor, engine governor and ship propeller. Besides these, a waste heat utilization heat exchanger has been added to the exhaust line.

The cylinder is confined by the liner walls, cylinder head and by the piston which runs between the top dead centre to the bottom dead centre positions. By applying the first law of thermodynamics on the variable mass and composition fluid in the cylinder, the differential of heat energy is:

$$dQ = dU + p \cdot dV \tag{1}$$

$$dU = d(m \cdot u) = u \cdot dm + m \cdot du \tag{2}$$

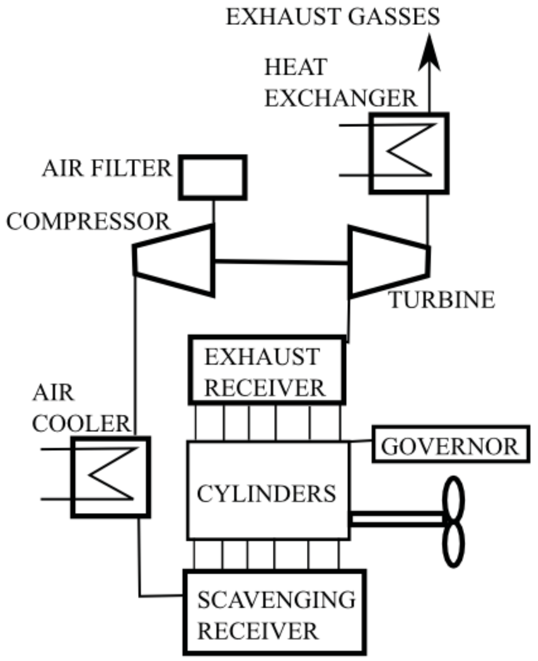

Figure 2 **Engine model scheme with exhaust gasses heat exchanger**
Slika 2 **Shema motora sa izmjenjivačem topline na ispušne plinove**

By inserting equation (2) in equation (1) and deriving by crankshaft angle it follows:

$$\frac{dQ}{d\varphi} = u\frac{dm}{d\varphi} + m\frac{du}{d\varphi} + p\frac{dV}{d\varphi} \tag{3}$$

which is the basic equation for the heat energy increase in the engine cylinder.

The heat energy exchange is defined by the energy released from fuel combustion, by the heat exchanged between the fluid and the cylinder walls, and by the heat that flows to the environment:

$$\frac{dQ}{d\varphi} = \frac{dQ_f}{d\varphi} + \frac{dQ_w}{d\varphi} + h_i\frac{dm_i}{d\varphi} + h_e\frac{dm_e}{d\varphi} + h_{bb}\frac{dm_{bb}}{d\varphi} \tag{4}$$

The gas mass exchange in the engine cylinder is defined by the mass of the fluid that flows through the scavenging ports and trough the exhaust valve, by the blow-by process, and by the injected fuel mass.

$$\frac{dm_c}{d\varphi} = \frac{dm_f}{d\varphi} + \frac{dm_i}{d\varphi} + \frac{dm_e}{d\varphi} + \frac{dm_{bb}}{d\varphi}. \tag{5}$$

The heat energy fluxes in the two-stroke, uniflow scavenged diesel engine cylinder is shown in Figure 3 where:

$\frac{dQ_f}{d\varphi}$ heat energy amount that is released from the fuel combustion process in the cylinder,

$\frac{dQ_w}{d\varphi}$ heat energy amount that flows to the cylinder through the walls,

$\frac{dm_i}{d\varphi}h_i$ heat energy of the inlet gas that flows through the scavenging ports,

$\frac{dm_e}{d\varphi}h_e$ heat energy of the gas that flows through the exhaust valve,

$\frac{dm_{bb}}{d\varphi}h_{bb}$ heat energy of the gas that blows by through the unsealed gaps (blowby).

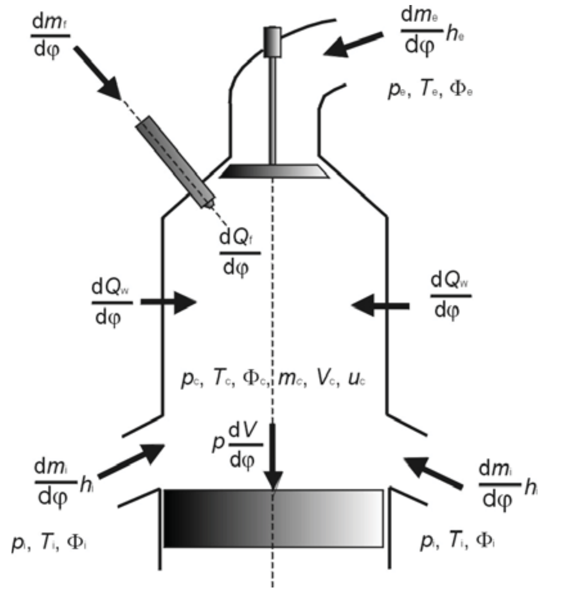

Figure 3 **Energy fluxes in the cylinder, source [8], [9]**
Slika 3 **Energijski tokovi u cilindru, izvor [8], [9]**

After a series of mathematical manipulations presented in [8], the equation for the temperature variation in function of the crankshaft angle is obtained:

$$\frac{dT_C}{d\varphi} = \frac{\frac{1}{m}\left[-\frac{p\cdot dV}{d\varphi} + \sum_i \frac{dQ_i}{d\varphi} + \sum_j h_j\frac{dm_j}{d\varphi} - u\frac{dm}{d\varphi} - m\left(\frac{\partial u}{\partial\lambda}\right)\frac{d\lambda}{d\varphi} - C\right]}{\frac{\partial u}{\partial T} + \frac{A}{B}\frac{\partial u}{\partial p}} \tag{6}$$

where

$$A = 1 + \frac{T}{R}\frac{\partial R}{\partial T}, \tag{7}$$

$$B = 1 - \frac{p}{R}\frac{\partial R}{\partial p} \tag{8}$$

and

$$C = \frac{p}{B}\frac{\partial u}{\partial p}\left[\frac{1}{m}\frac{dm}{d\varphi} - \frac{1}{V}\frac{dV}{d\varphi} + \frac{1}{R}\left(\frac{\partial R}{\partial\lambda}\right)\frac{d\lambda}{d\varphi}\right]. \tag{9}$$

By integrating equation (6), cylinder temperature is obtained. The cylinder pressure is obtained from the ideal gas equation of state (10):

$$p_C = \frac{m_C\cdot R_C\cdot T_C}{V_C} \tag{10}$$

Additional details about the engine mathematical model are given in [8] and [9].

The waste heat flux contained in the exhaust gases that can be utilized is calculated according to:

$$\dot{Q}_{us,g} = \dot{m}_g \cdot c_p \cdot \left(T_{in} - T_{out}\right) \tag{11}$$

where $\dot{m}_g$ is the mass flux of the exhaust gasses, c_p is the specific heat capacity of the exhaust gas, T_{out} is the supposed outlet temperature of the exhaust gases. The inlet temperature of the gases in the heat exchanger T_{in} is the outlet temperature of the gases that leave the turbine and is calculated according to:

$$T_4 = T_3 \left[1 - \eta_T \left(1 - \pi_T^{\frac{1-\kappa}{\kappa}}\right)\right], \tag{12}$$

where T_4 is the temperature of the exhaust gases after the turbine, T_3 is the temperature of the exhaust gases before the turbine, η_T is the turbine efficiency, π_T is the turbine pressure ratio, and κ is the adiabatic exponent for the conditions in the turbine.

2 Calculation settings

The mathematical model was built in MATLAB-SIMULINK environment. The model was applied on a low-speed, two stroke marine diesel engine MAN B&W 6S50MC. The main technical characteristics of the engine are presented in Table 1. A crosscut of the engine is shown in Figure 4. The validation of the model and the comparison with experimental results from the test bench are presented in [8] and [9].

Table 1 **Main engine characteristics**
Tablica 1 **Glavne značajke motora**

Bore	500 mm
Stroke	1910 mm
Cylinders	6 in line
Max. continuous power	8580 kW
Max. continuous speed	127 min^{-1}
Max. mean eff. pressure	18 bar
Max. combustion pressure	143 bar
Specific fuel consumption	171 g/(kW h)
Compression ratio	17.2
Engine mass	232 000 kg

The engine allows the variation of the fuel injection timing. It is assumed that there is a possibility of variation of the start of the exhaust valve opening, which is an option on some engines of similar characteristics. The possibilities of fuel injection timing variation and exhaust valve opening variation are integrated in the model. The parametric analysis was done for the start of fuel injection in the range from φ = -6° to +6° compared to the original value (φ = 177° after BDC – Bottom Dead Centre), that is from φ = 171° to 183° after BDC. As far as the beginning of the exhaust valve opening is concerned, simulations were done for cases from φ = -16° to +8° compared to the standard value (φ = 290° after BDC), that is from φ = 274° to 298° after BDC. The governor regulates the amount of fuel per process in function of the engine speed and of the position of the throttle handle. The engine load was 100% of the maximum continuous power according to the results obtained on the test bench. It means that the governor leads the engine in the way to obtain 8182 kW at the revolution speed of 121.4 min^{-1}. It was achieved for all the cases except for those with the latest injection timing, with the beginning of the injection at 183 °CA.

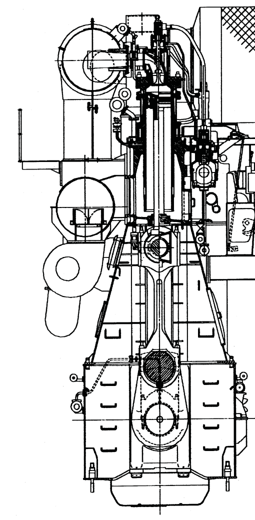

Figure 4 **Cross section of the engine, source [8], [9]**
Slika 4 **Poprečni presjek motora, izvor [8], [9]**

3 Results

The start of fuel injection (SOI) was varied from φ = 171° to 183°. The moment of the beginning of the exhaust valve lift (EVO) was also varied from φ = 274° to 298°. The duration of fuel injection and the duration of valve lift remained unchanged. It was observed how the change of these parameters influences the fuel amount per process (Figure 5), the engine efficiency (Figure 6), exhaust gas temperature (Figure 7), the heat available in the exhaust gasses (Figure 8), the overall efficiency of the cogeneration system (Figure 9), and the power to heat ratio (Fig. 10).

The governor tries to achieve the selected power by changing the amount of fuel per process. Since the load is determined by the propeller curve, each power level is related to a determined revolution speed. In Figure 5 it can be seen that the fuel mass per process increases for later injection and decreases for earlier. This is an expected behaviour, since for an earlier fuel injection higher cylinder pressures are reached for the same fuel amount, so for an earlier injection a smaller fuel quantity is enough to reach the same power. In the same figure it can be noticed that the beginning of exhaust valve lift has a weaker influence: an earlier opening and closing of the valve results in the need for a greater fuel mass to produce the same power on the crankshaft.

The engine efficiency is given by the following equation:

$$\eta_{f,b} = \frac{P_b}{\dot{Q}_f} \tag{13}$$

where P_b is the brake power while $\dot{Q}_f$ is the heat flux released by the combustion of the injected fuel. In Figure 6 the basic trend can be seen and this is that engine efficiency rises for an earlier fuel injection and for a later exhaust valve opening.

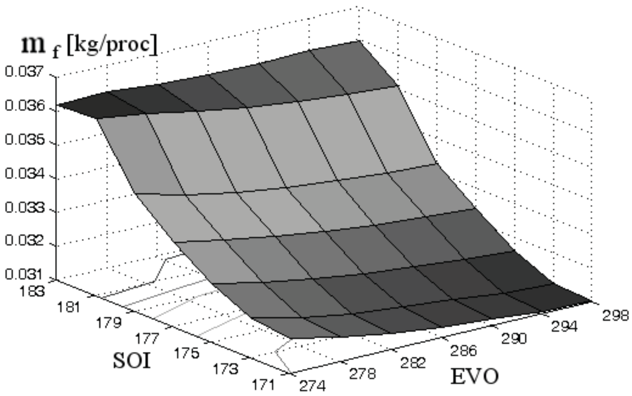

Figure 5 **Fuel mass per process**
Slika 5 **Masa goriva po procesu**

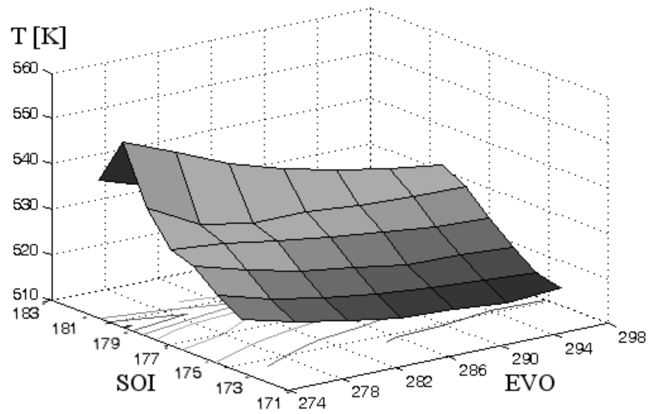

Figure 7 **Exhaust gas temperature after the turbine**
Slika 7 **Temperatura ispušnih plinova nakon turbine**

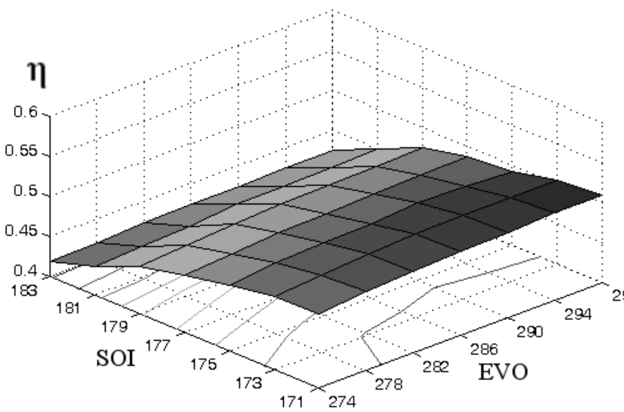

Figure 6 Engine efficiency
Slika 6 **Stupanj djelovanja motora**

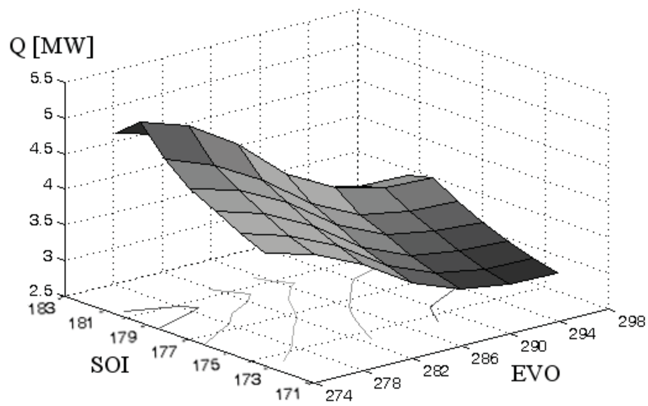

Figure 8 **Heat available in the exhaust gasses**
Slika 8 **Raspoloživost topline u ispušnim plinovima**

Figure 7 shows the exhaust gas temperature. It changes similarly as the fuel mass per process. It rises for later fuel injection and earlier exhaust valve opening. It can be noticed that the temperature decreases for the latest fuel injection checked, the one that occurs 6ºCA after the standard setting. For these settings the selected power could not be reached, the regulator cuts off the required fuel mass, and as a consequence the power and exhaust gas temperature drop.

The exhaust gas temperature influences strongly the heat contained in the exhaust gases, as it can be seen in Figure 8. The highest value of 5.317 MW is obtained for the valve opening at 274ºCA and for the beginning of fuel injection at 181ºCA.

If the heat is used to power an absorption refrigeration system, with an assumed coefficient of performance of 0.7, it means that in the optimal case from the waste heat it is possible to obtain the cooling capacity of:

$$P_{cool} = COP \cdot \dot{Q}_{us,g} = 0.7 \cdot 5.317 = 3.72 \text{ MW} \qquad (14)$$

where $\dot{Q}_{us,g}$ is the used heat flux from the exhaust gases.

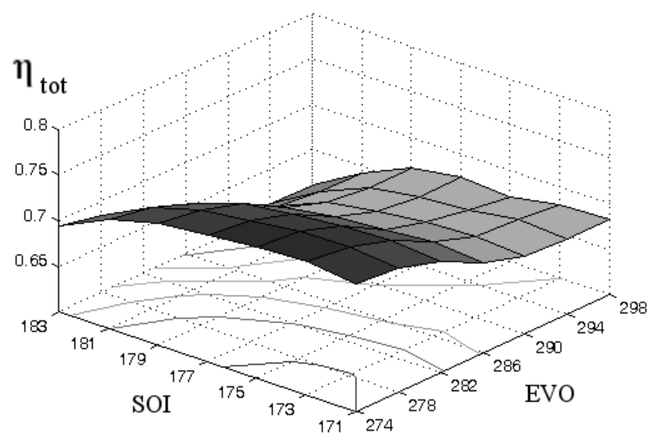

Figure 9 **Overall efficiency of the cogeneration system**
Slika 9 **Ukupni stupanj djelovanja kogeneracijskog sustava**

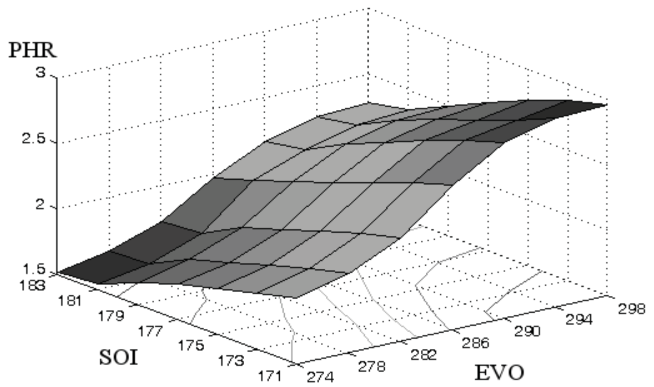

Figure 10 **Ratio between mechanical and heat energy (PHR)**
Slika 10 **Omjer mehaničke i toplinske energije (PHR)**

The overall efficiency of a cogeneration system takes into account that the heat from the exhaust gases is being used. It is calculated according the equation

$$\eta_{tot} = \frac{P_b + \dot{Q}_{us,g}}{\dot{Q}_f} \tag{15}$$

In Figure 9 it can be noticed that the highest overall efficiency is found for earlier injection timing and for earlier exhaust valve opening. The ratio between the mechanical power and the useful heat in the exhaust gases or PHR (Power to Heat Ratio) is calculated according to the equation

$$PHR = \frac{P_b}{\dot{Q}_{us,g}}. \tag{16}$$

Usually, it is desirable to have the PHR as high as possible since it is more difficult to obtain the mechanical power, and hence it has a higher value. However, it is possible to imagine the situation in an operational cogeneration system in which a greater demand for heat arises. For example, a tanker before arriving to the destination port has to raise the temperature of the carried crude oil in order to pump it. In such a situation it is possible to change the engine settings in order to obtain a higher quantity of heat. In such a case a lower PHR could be welcome. In Figure 10 it can be seen that the PHR rises for earlier fuel injection and for later exhaust valve opening. It means that for the above described situation of higher heat demand, the engine parameters should be set towards later fuel injection and earlier valve opening.

4 Conclusion

A slow-speed marine engine model was developed. The influence of the start of fuel injection and the influence of the start of the exhaust valve opening on the engine efficiency, on the exhaust gas temperature, on the amount of heat in the exhaust gases and on the cogeneration system properties: overall efficiency and power to heat ratio was analyzed.

The mechanical power output was constant since the governor dosed the fuel mass per process in order to achieve the selected engine revolution speed. It can be concluded on the basis of the results that a greater amount of fuel is injected as a consequence of a later start of injection. Furthermore, a greater amount of fuel is injected as a consequence of an earlier valve opening. A higher exhaust gas temperature and a grater amount of heat energy in the exhaust gases is manifested for the same engine settings as for those for which a greater amount of fuel was injected. Hence, greater heat and higher temperatures of the exhaust gases result from the larger fuel amounts injected. Since the crankshaft power remains constant, the engine efficiency decreases for these settings. However, the overall efficiency of the cogeneration system is the highest for earlier injection and especially for earlier exhaust valve opening. The power to heat ratio increases for earlier injection and for later exhaust valve opening.

For a more effective cogeneration application on a low-speed marine diesel engine powered vessel, the key would be a specifically developed governor. The target parameter of a standard governor is the engine revolution speed (or engine power, which is a simple function of the revolution speed), while the input parameters are the current revolution speed and the throttle handle position. A regulator specifically developed for the use with a cogeneration system should also have as a target value the exhaust gases heat or the exhaust gases temperature.

The variable injection timing and the hydraulic valve drive allow the variation of the beginning of the injection and of the opening of the exhaust valve. However, such systems would allow even a greater flexibility, such as different injection patterns, split injection and different exhaust valve lift curves. These options are not analyzed in the current work.

In turbocharged engines boost air is cooled after raising its pressure in the compressor and before the intake. If the utilization of the heat retrieved from the boost air it is taken into account, the overall efficiency of the cogeneration system would increase even further. However, it would be necessary to do an economic analysis of the convenience of the recuperation of this heat.

Four-stroke diesel engines or other propulsion systems have higher exhaust gas temperatures. Hence it is possible to utilize a greater part of waste heat. However, it has to be taken into account that mechanical energy has a higher value than heat energy and that the main purpose of the here analyzed engine is ship propulsion. Furthermore, low-speed marine diesel engines are fuelled by cheap heavy fuel oil. It can be concluded that the system with a diesel engine is irreplaceable as ship propulsion system on most ship types.

The heat exchanger installation in the exhaust pipes reduces exhaust gas temperatures, so it is necessary to take care to avoid condensation and low temperature corrosion. Besides, the heat exchanger configuration has to be such to avoid soot deposition or to allow its removal.

For further investigation it would be interesting to model a different governor which would allow a more flexible control of the fuel quantity. It would be necessary to analyze the influence of greater fuel amounts which the new regulator would allow. Furthermore, the influence of different injection patterns, longer injection durations and different exhaust valve curve lifts on engine and cogeneration system performance should be also analyzed. It is supposed that in this way a much higher amount of heat in the exhaust gas could be generated and utilized in a heat exchanger. This could maybe allow savings on a separately fuelled steam generator. It would also be good to develop the engine model to allow the calculation of pollutants emissions and to check what the described variation of parameters means for pollutant emission.

5 References

[1] BERNEČIĆ, D.: "Analiza suvremenih sistema ubrizgavanja i sistema upravljanja ispušnim ventilom na sporohodnim brodskim motorima", Master Thesis, University of Rijeka, 2005.

[2] PERANIĆ, J., BOŽUNOVIĆ, A., PAVKOVIĆ, B.: "The feasibility of ammonia – water absorption refrigeration for fishing vessels, Proceedings of IIR Conference Ammonia Refrigeration Technology for Today and Tomorrow", Ohrid, Macedonia, 2007.

[3] ABUSOGLU, A., KANOGLU, M.: "Exergetic and thermoeconomic analyses of diesel engine powered cogeneration: Part 1 – Formulations", Applied Thermal Engineering 29 (2009) p. 234–241.

[4] ABUSOGLU, A., KANOGLU, M.: "Exergetic and thermoeconomic analyses of diesel engine powered cogeneration: Part 2 – Application", Applied Thermal Engineering 29 (2009) p. 242-249.

[5] ABUSOGLU, A., KANOGLU, M.: "First and second law analysis of diesel engine powered cogeneration systems", Energy Conversion and Management 49 (2008) p. 2026-2031.

[6] …:"An Educational Tool for Cogeneration", COGEN Europe, EDUCOGEN Project, National Technical University of Athens, 2001.

[7] …: "A Guide to Cogeneration", COGEN Europe, EDUCOGEN Project, Brussels, 2001.

[8] RAČIĆ, N.: "Simulacija rada brodskog propulzijskog sustava sa sporohodnim dizelskim motorom u otežanim uvjetima", PhD Thesis (in Croatian), University of Rijeka, 2008.

[9] MEDICA, V., RAČIĆ, N., RADICA, G.: "Performance Simulation of Marine Slow-Speed Diesel Propulsion Engine with Turbocharger under Aggravated Conditions", Strojarstvo, Zagreb, 51 (2009), p. 199-212.

[10] MEDICA, V.: "Simulation of turbocharged diesel engine driving electrical generator under dynamic working conditions", PhD Thesis, University of Rijeka 1988.

[11] CHOW, A., WYSZYNSKI, M. L.: "Thermodynamic modelling of complete engine systems - a review", PhD Thesis, MSAE School of Manufacturing and Mechanical Engineering, University of Birmingham, 1998.

[12] MRAKOVČIĆ, T.: "Osnivanje i vođenje brodskog pogonskog postrojenja primjenom numeričke simulacije", PhD Thesis (in Croatian), University of Rijeka, 2003.