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Possible use of combined Diesel engine / steam turbine systems in ship power plants

Możliwości zastosowania układów kombinowanych: silnik Diesla – turbina parowa w siłowniach okrętowych

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Key words: ship power plants, combined systems, piston internal combustion engine, gas turbine, steam turbine

Abstract
The article presents a combined system of a large power ship power plant. The system consists of a piston internal combustion engine and the steam turbine system which utilises the energy transported with the exhaust gas leaving the internal combustion engine. The analysed variant of the combined cycle includes a Diesel engine and a steam turbine with a single-pressure waste-heat boiler. The numerical calculations were performed for two low-speed internal combustion engines made by Wärtsilä and MAN Diesel & Turbo, each of an approximate power output of 54 MW. The assumptions and limits which were used in the calculations done using a specially worked out code are included. The energy optimisation of the entire combined ship power plant was done taking into account only the thermodynamic point of view, leaving aside the technical and economic aspects.

Słowa kluczowe: siłownie okrętowe, układy kombinowane, tłokowy silnik spalinowy, turbina gazowa, turbina parowa

Abstrakt
W artykule przedstawiono układ kombinowany siłowni okrętowej dużej mocy. Składa się on z silnika spalinowego tłokowego i skojarzonego z nim układu turbiny parowej, wykorzystującego energię zawartą w spalinach wylotowych silnika spalinowego. Rozpatrzono tylko wariant obiegu kombinowanego silnik Diesla – turbina parowa, z kotłem utylizacyjnym jednociśnieniowym. Podano założenia i ograniczenia oraz wyniki obliczeń rozpatrywanego układu z opracowanego programu komputerowego. Optymalizację energetyczną całej kombinowanej siłowni okrętowej przeprowadzono tylko z punktu termodynamicznego. Nie zajmowano się analizą techniczno-ekonomiczną. Obliczenia numeryczne wykonano dla silników spalinowych wolnoobrotowych firmy Wärtsilä oraz MAN Diesel & Turbo o mocy około 54 MW.

Nomenclature

b_{eD} – specific fuel consumption,
p – pressure,
m_{fD} – fuel mass flow rate,
m_{sD} – exhaust gas mass flow rate,
t_{wyl} – temperature of the exhaust gas leaving the waste-heat boiler,
t_4 – temperature of the exhaust gas leaving the Diesel engine,
t_{odg} – temperature of the degasifier,
N_{combi} – power output of the combined system,
N_D – power output of the Diesel engine,
N_{ST} – power output of the steam turbine,
Wu – caloric value of fuel,
η – efficiency.

Introduction

For years, a search has been made for technical solutions of ship power plants which would increase their thermodynamic efficiency and power output. Bearing this goal in mind, a decision was made to analyse the operation of a combined system composed of a Diesel engine, and a steam and gas turbines.

The internal combustion engines belong to a group of thermal engines in which the mechanical energy is converted from the thermal energy obtained during the combustion of the fuel delivered to the engine [1]. The thermal energy obtained in the above way is passed to the exhaust gas which acts as an intermediary working medium in the process of conversion of the thermal energy into mechanical energy. When it leaves the internal combustion engine, the exhausts gas still contains about 25% of the heat delivered to the engine in the fuel. Using the heat collected in the exhaust gas in the gas and/or steam turbine cycle we can increase the efficiency from about 40% for the classical steam cycle to about 60% in the combined cycle. For large-power piston internal combustion engines the additional gas and steam turbine cycle brings measurable economic savings during ship power plant operation. The proposed solution makes it possible to reduce the specific fuel consumption via increasing the power output of the system resulting from recovering the thermal energy from the exhaust gas leaving the piston engine. Another possibility of energy recovery in the piston internal combustion engine system is the use of so-called low-temperature waste heat, for instance, in an additional system covering general ship needs, including heating of the ship and/or heating of the utility water. The proportions of the waste heat related to the nominal engine power output are equal to [2]:

- heat from the supercharging air cooler system – 38%,
- heat in the cylinder cooling water – 16%,
- heat from the lubricating oil coolers – 11%.

The article discusses a combined cycle variant consisting of a Diesel engine and the steam turbine cycle with a single-pressure waste-heat boiler which utilises the energy collected in the exhaust gas.

Concept of the combined ship power plant system

The here adopted combined system of a Diesel engine ship power plant which utilises the exhaust gas heat in the steam turbine cycle is shown in figure 1.

The exhaust gas flows from the piston engine to the waste-heat boiler installed in the main engine exhaust gas duct. The exhaust heat boiler produces the steam which is used both for driving the steam turbine which transmits the power to the electric generator or propeller screw, and for covering general ship needs.

Energy evaluation of the combined ship power plant system

The adopted concept of the combined ship power plant system requires evaluating the energy balance of the analysed cycle. Based on the adopted scheme, see figure 1, formulas have been derived for calculating the power and efficiency of the propulsion system.

The power output of the combined power plant system was determined via summing up of particular powers of the system components (the main Diesel engine, the steam turbine):

$$N_{\mathrm{combi}} = N_D + N_{ST} = N_D\left(1+\frac{N_{ST}}{N_D}\right) \tag{1}$$

As a result, the efficiency of the combined system is:

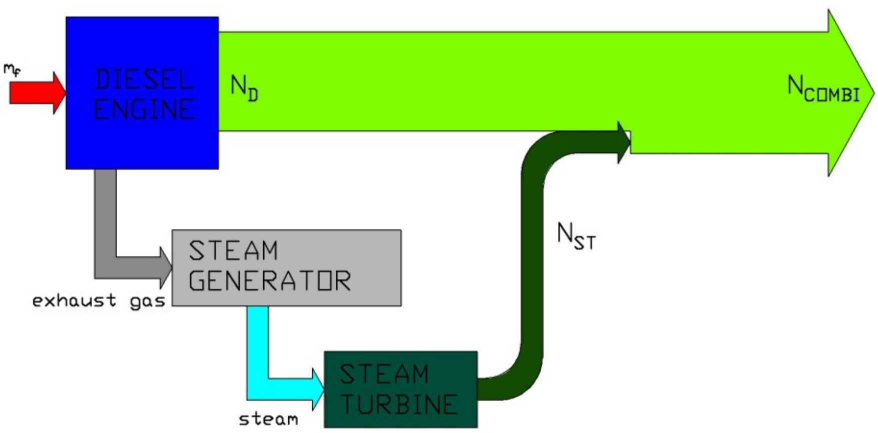


Fig. 1. Concept of a combined ship power plant system
Rys. 1. Koncepcja kombinowanego układu siłowni okrętowej

$$\eta_{\text{combi}} = \frac{N_{\text{combi}}}{m_{fD} \cdot Wu} = \eta_D \cdot \left(1 + \frac{N_{ST}}{N_D}\right) \qquad (2)$$

where:

$$\eta_D = \frac{N_D}{m_{fD} \cdot Wu} \qquad (3)$$

while the specific fuel consumption is:

$$b_{e\text{combi}} = b_{eD} \cdot \frac{1}{\left(1 + \frac{N_{ST}}{N_D}\right)} \quad \text{[g/kWh]} \qquad (4)$$

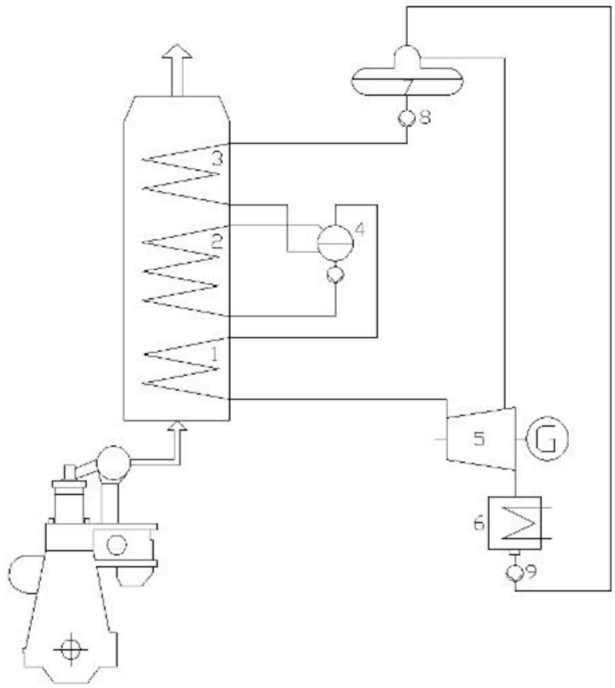
Relations (2) and (4) show that the additional power in the power plant cycle increases the system efficiency, i.e.: decreases the fuel consumption. The higher the additional power obtained from utilisation of the heat contained in the exhaust gas leaving the main engine, the lower the fuel consumption. Taking into account this fact, we should tend to obtain the maximum power from the steam turbine, as it does not increase the mass flow rate of the consumed fuel and the power output of the entire system is only increased via utilising the heat collected in the exhaust gas from the Diesel engine. Increasing the power of the steam turbine can be done by relevant selection of parameters of the live steam and the condenser.

Adopted variant of the combined ship power plant system

Depending on the adopted concept of the combined ship power plant system we can analyse a number of systems consisting of the main engine and the steam turbine. The here analysed variant is composed of a Diesel engine and a steam turbine with a single-pressure waste-heat boiler as a steam supply for the turbine (Fig. 2). The additional power of the steam turbine in the combined cycle is decisive about the efficiency increase of the entire system. Therefore we should select the steam cycle in a way which will provide opportunities for the steam turbine to generate the highest possible power output. It can be done by optimal selection of the parameters of the live steam, and the reduction of the regenerative water heating to the minimum by proper selection of the boiler feed water temperature. If no limits were imposed on the minimum temperature of the feed water, we could completely resign from the regeneration [3].

Figure 2 shows a solution of the combined ship power plant system including the steam cycle with a single-pressure waste-heat boiler. The exhaust gas leaving the Diesel engine is passed to the waste heat boiler installed in the exhaust gas duct of the main engine. The boiler produces single-pressure steam, and the turbine has only one regeneration bleeder for water heating in the mixing heater which also plays the role of a degasifier. The boiler which utilises the waste heat consists of three parts:

- feed water heater (economizer),
- evaporator,
- steam superheater.



Diesel engine

Fig. 2. Combined cycle with Diesel engine and single-pressure waste-heat boiler feeding the steam turbine: 1 – steam superheater, 2 – evaporator, 3 – economizer, 4 – drum, 5 – steam turbine, 6 – condenser, 7 – degasifier, 8 – main feed water pump, 9 – condensate pump

Rys. 2. Obieg kombinowany z silnikiem o zapłonie samoczynnym oraz kotłem utylizacyjnym jednociśnieniowym zasilającym turbinę parową: 1 – przegrzewacz pary, 2 – parownik, 3 – ekonomizer, 4 – walczak, 5 – turbina parowa, 6 – skraplacz, 7 – odgazowywacz, 8 – pompa główna wody zasilającej, 9 – pompa kondensatora

The steam produced in the waste heat boiler can be used for feeding the steam turbine which transmits the power to the generator, or to cover general ship needs.

Limits of parameters and the values assumed in combined cycle calculations

Analysing the proposed variant of the combined cycle required numerical calculations, before which particular cycle components were to be precisely defined. Then, the calculations were performed taking into account the assumed limits of the parameters and the values selected for the entire combined system utilising the exhaust gas energy and consisting of the Diesel engine and the steam turbine system.

The above calculations performed for two low-speed piston engines made by Wärtsilä 9RTA96C and MAN Diesel & Turbo 9K98MC-C7.1-TII have made the basis for comparing their operation as combined cycle components. For both engines working as the main engines the combined cycle calculations were performed for the load corresponding to 90% CMCR (Contract Maximum Continuous Rating).

The limits assumed for the steam cycle parameters resulted from technical and strength conditions, the required durability of particular system components, as well as from economic and constructional restrictions. The combined cycle calculations took into account all those limits and restrictions.

The temperature difference Δt between the exhaust gas temperature at Diesel engine exit and the live steam temperature in the waste heat boiler in a combined system was assumed equal to $\Delta t = 10°C$. The "pitch point" value recommended by MAN for boilers is equal to $\delta t \approx 10°C$. The limit for the steam dryness fraction x behind the steam turbine was assumed equal to $x_{\lim it} = 0.88$. The calculations were preformed assuming that the feed water temperature is within $t_{wz} \in \langle 102–152 \rangle°C$. It was also assumed that the exhaust gas temperature at boiler exit should be higher by 15°C that the feed water temperature, i.e. $t_4 > t_{wz} + 15°C$. This assumption was justified by the fact that the outer surfaces of the heater pipes, on the exhaust gas side, have the temperature higher by 15°C than the feed water temperature. Moreover, the use of materials revealing extended resistance to acid corrosion is recommended. The degasifier of a boiling type was assumed with different feed water temperatures in the degasifier $t_{odgz} = t_{wz} - 2°C$. The efficiency of the expansion line in the turbine in the analysed cycle was assumed at the level of $\eta_i = 0.85$. The caloric value of the fuel was assumed equal to $W_u = 42\ 700$ kJ/kg, following producer's recommendations [2, 4].

Table 1 collects the parameters of the analysed marine low-speed engines.

Table 1. Basic parameters of the marine low-speed Diesel engines
Tabela 1. Podstawowe parametry okrętowych wolnoobrotowych silników Diesla

Parameter	Unit	9RTA96C [2] WÄRTSILÄ	9K98MC-C7.1-TII [4] MAN DIESEL & TURBO
N_D	kW	46 332	48 762
m_{sD}	kg/s	104.504	134.25
m_{fD}	kg/s	2.146	2.369
t_4	°C	271	232.8
b_{eD}	g/kWh	166.8	174.9

Comparing the analysed combined ship power plant cycles

In practical application, the steam turbine system requires some limits imposed on certain parameters resulting from design, constructional and operating assumptions [5]. Using the assumptions adopted in the previous section, the operation of a combined ship power plant was analysed for two low-speed piston engines 9RTA96C and 9K98MC-C7.1-TII working as the main engines. For each engine an optimal solution was searched in which the power output of the steam turbine reached its maximum. The calculations were performed for three different medium temperatures in the degasifier, which were equal to 100°C, 120°C and 150°C.

The curves shown in figures 3 and 4 illustrate the effect of particular parameters on the steam turbine power output. We can observe large differences in the maximum steam turbine power depending on the selected low-speed piston engines: Wärtsilä 9RTA96C or MAN Diesel & Turbo 9K98MC-C7.1-TII. Moreover, a tendency is observed that the lower the feed water temperature, the higher the steam turbine power. Here, we should take into account the fact that the exhaust gas temperature limit at the waste heat boiler exit was equal to $t_{spal} = 137°C$ due to the sulphur content in the fuel.

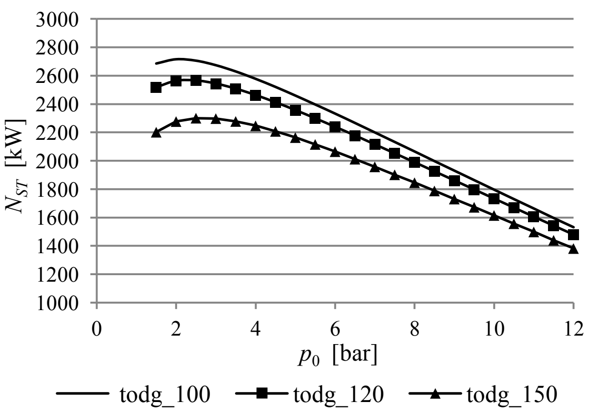


Fig. 3. Steam turbine power output as a function of the live steam pressure and the feed water temperature in the degasifier for the engine 9K98MC-C7.1-TII (MAN Diesel & Turbo)
Rys. 3. Moc turbiny parowej w funkcji ciśnienia pary świeżej oraz temperatury wody zasilającej w odgazowywaczu dla silnika 9K98MC-C7.1-TII (MAN Diesel & Turbo)

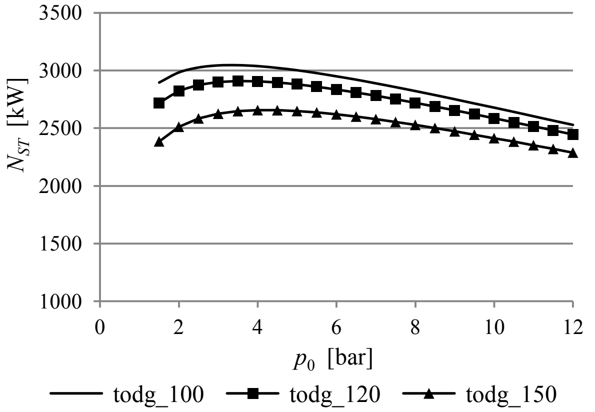


Fig. 4. Steam turbine power output as a function of the live steam pressure and the feed water temperature in the degasifier for the engine 9RTA96C (Wärtsilä)
Rys. 4. Moc turbiny parowej w funkcji ciśnienia pary świeżej oraz temperatury wody zasilającej w odgazowywaczu dla silnika 9RTA96C (Wärtsilä)

The curves shown in two next figures (Figs 5 and 6) illustrate the effect of particular parameters on the efficiency of the combined system. Like in case of the power output we can observe the difference in the obtained maximal efficiency of the system depending on the selected low-speed piston engines 9RTA96C and 9K98MC-C7.1-TII. Here, we should take into account the restrictions concerning the temperature of the feed water in the degasifier and the efficiency of the Diesel engines working as the main engines in the ship power plant.

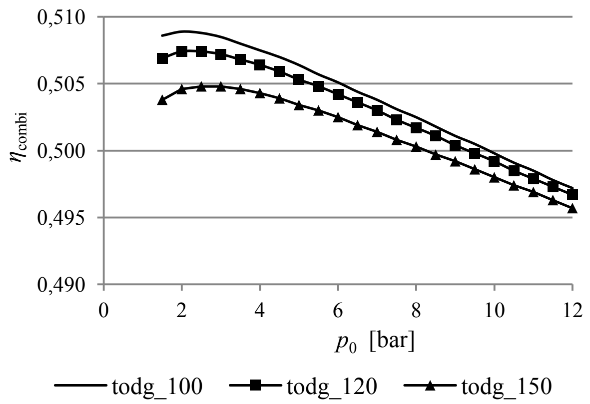


Fig. 5. Efficiency of the combined system as a function of the live steam pressure and the feed water temperature in the degasifier for the engine 9K98MC-C7.1-TII (MAN Diesel & Turbo)
Rys. 5. Sprawność obiegu kombinowanego w zależności od ciśnienia pary świeżej oraz temperatury wody zasilającej odgazowywacz dla silnika 9K98MC-C7.1-TII (MAN Diesel & Turbo)

The curves shown in the last series of figures (Figs 7 and 8) illustrate the effect of particular parameters on the exhaust gas temperature behind the waste heat boiler. We can observe that for some

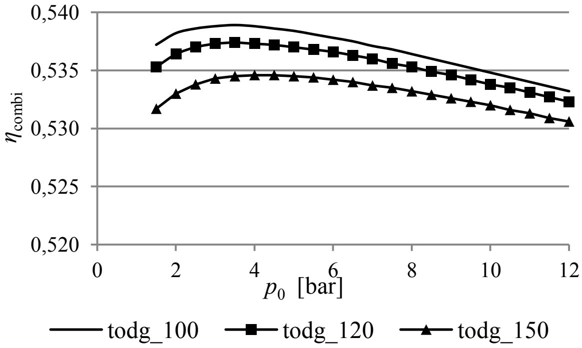


Fig. 6. Efficiency of the combined system as a function of the live steam pressure and the feed water temperature in the degasifier for the engine 9RTA96C (Wärtsilä)
Rys. 6. Sprawność obiegu kombinowanego w zależności od ciśnienia pary świeżej oraz temperatury wody zasilającej odgazowywacz dla silnika 9RTA96C (Wärtsilä)

parameters the temperature t_{wyl} is lower than the limit resulting from the sulphur content.

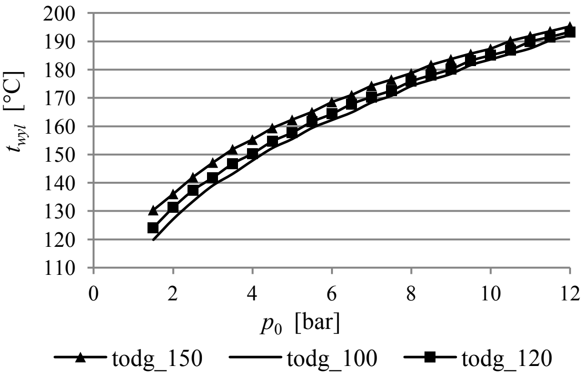


Fig. 7. Exhaust gas temperature at waste heat boiler exit as a function of the live steam pressure and the feed water temperature in the degasifier for the engine 9K98MC-C7.1-TII (MAN Diesel & Turbo)
Rys. 7. Temperatura spalin wylotowych z kotła utylizacyjnego w zależności od ciśnienia pary świeżej oraz temperatury odgazowywacza dla silnika 9K98MC-C7.1-TII (MAN Diesel & Turbo)

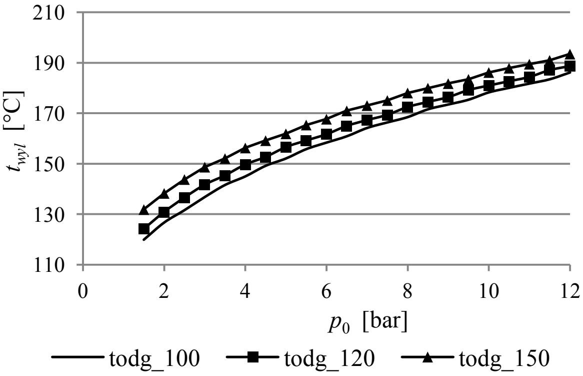


Fig. 8. Exhaust gas temperature at waste heat boiler exit as a function of the live steam pressure and the feed water temperature in the degasifier for the engine 9RTA96C (Wärtsilä)
Rys. 8. Temperatura spalin wylotowych z kotła utylizacyjnego w zależności od ciśnienia pary świeżej oraz temperatury odgazowywacza dla silnika 9RTA96C (Wärtsilä)

Figure 9 shows the effect of the live steam pressure on the steam dryness fraction in the two combined Diesel engine / steam turbine systems, taking into account the restrictions presented in Section IV. We can observe that the lower the pressure, the higher the steam dryness fraction is reached in the analysed system.

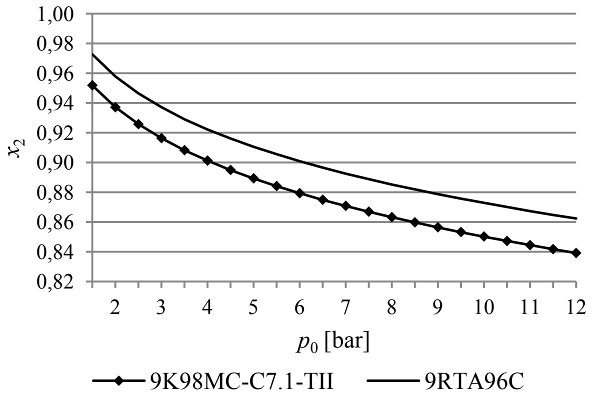


Fig. 9. Steam dryness fraction behind the turbine as a function of the live steam pressure

Rys. 9. Stopień suchości pary za turbiną w funkcji ciśnienia pary świeżej dla rozpatrywanych silników

The above calculations were performed taking into account the restrictions listed in Section IV, and additionally that:

- the exhaust gas temperature limit at the waste heat boiler exit is $t_{spal} > 137°C$ (due to the sulphur contest in the fuel) [6],
- the mechanical efficiency is $\eta_m = 0.98$,
- the temperature of the medium in the degasifier, assumed according to the recommendations of MAN Diesel & Turbo, is $t_{odg} = 120°C$ [7].

Table 2 collects optimal results for the analysed combined cycle of the ship power plant with, alternatively, two low-speed piston engines 9RTA96C and 9K98MC-C7.1-TII.

The calculations aimed at determining optimal parameters for the analysed combined cycle with the two above-named engines. The presented results include the maximal power output of the steam turbine which can be reached in the adopted simplest variant of the combined system with a single-pressure waste heat boiler. Their analysis reveals that this maximal power output equals 6.27 and 5.26% of the piston engine power output, depending on the type of engine.

Higher efficiency was reached in the combined cycle with the 9RTA96C engine, as compared to that with the 9K98MC-C7.1-TII engine. Moreover, the live steam pressure is higher in the combined cycle with the 9RTA96C Diesel engine than that in the cycle with the 9K98MC-C7.1-TII engine.

Table 2. Optimal parameters of the combined Diesel engine / steam turbine cycle

Tabela 2. Optymalne parametry układu kombinowanego silnik spalinowy Diesla – turbina parowa

Optimal parameters of cycle calculations taking into account the adopted restrictions			
Parameter	Unit	9RTA96C	9K98MC-C7.1-TII
N_D	kW	46 332	48 762
η_D	%	50.56	48.2
t_4	°C	271	232.8
t_0	°C	261	222.8
t_{wz}	°C	122	122
p_0	bar	3.5	2.5
m_0	kg/h	21 888	20 160
m_{odg}	kg/h	482.4	478.8
x_2		0.9292	0.9258
t_{wyl}	°C	145.4	137.4
N_{ST}	kW	2908	2566
N_{ST} / N_D	%	6.27	5.26
η_{combi}	%	53.74	50.74
$\Delta\eta_{combi}$ / η_D	%	6.28	5.27

Conclusions

The performed numerical calculations of the combined Diesel engine/steam turbine cycle have made it possible to formulate the following conclusions:

- the power output of the combined cycle can be increased by 6.27% when the Diesel engine 9RTA96C is used as the main engine, and by 5.26% for the engine 9K98MC-C7.1-TII with respect to the conventional power plant, and no additional fuel consumption is needed for this increase;
- the use of the combined cycle can increase the efficiency of the power plant operating at optimal parameters up to 53.74% in case of the Diesel engine 9RTA96C and up to 50.74% for the engine 9K98MC-C7.1-TII;
- a main engine of lower power output can be used in ship power plant systems in which the additional power is obtained in the combined cycle.

The article only presents the thermodynamic analysis of the combined ship power plant system. It should be complemented by additional technical and economic analysis to fully justify the use of such combined systems in ship power plants.

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