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Possible Efficiency Increasing of Ship Propulsion and Marine Power Plant with the System Combined of Marine Diesel Engine, Gas Turbine and Steam Turbine

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1. Introduction

For years there has been, and still is, a tendency in the national economy to increase the efficiency of both the marine and inland propulsion systems. It is driven by economic motivations (rapid increase of fuel prices) and ecological aspects (the lower the fuel consumption, the lower the emission of noxious substances to the atmosphere). New design solutions are searched to increase the efficiency of the propulsion system via linking Diesel engines with other heat engines, such as gas and steam turbines. The combined systems implemented in marine propulsion systems in recent years are based mainly on gas and steam turbines (MAN, 2010). These systems can reach the efficiency exceeding 60% in inland applications. The first marine system of this type was applied on the passenger liner "Millenium". However, this is the only high-efficiency marine application of the combined propulsion system so far. Its disadvantage is that the system needs more expensive fuel, the marine Diesel oil, while the overwhelming majority of the merchant ships are driven by low-speed engines fed with relatively cheap heavy fuel oil. It seems that the above tendency will continue in the world's merchant navy for the next couple of years.

The compression-ignition engine (Diesel engine) is still most frequently used as the main engine in marine applications. It burns the cheapest heavy fuel oil and reveals the highest efficiency of all heat engines. The exhaust gas leaving the Diesel engine contains huge energy which can be utilised in another device (engine), thus increasing the efficiency of the entire system and reducing the emission of noxious substances to the atmosphere.

A possible solution here can be a system combined of a piston internal combustion engine and the gas and steam turbine circuit that utilises the heat contained in the exhaust gas from the Diesel engine. The leading engine in this system is the piston internal combustion engine. It seems that now, when fast container ships with transporting capacity of 8- 12 thousand TU are entering into service, the propulsion engines require very large power, exceeding 50-80 MW. On the other hand, increasing prices of fuel and restrictive ecological limits concerning the emission of NO_x and CO₂ to the atmosphere provoke the search for new solutions which will increase the efficiency of the propulsion and reduce the emission of gases to the atmosphere.

The ship main engines will be large low-speed piston engines that burn heavy fuel oil. At present, the efficiency of these engines nears 45 - 50%. For such a large power output

ranges, the exhaust gas leaving the engine contains huge amount of heat available for further utilisation.

The proposed combined system consisting of a piston internal combustion engine, a gas turbine and a steam turbine can also be used for engines of lower power, ranging between 400 ÷ 900 kW. For those power ranges a use of low-boiling media of organic-based refrigerant type instead of water (steam) in the steam cycle seems to be a reasonable solution. Piston internal combustion engines of this power range are used on coasting vessels, or in the inland water transport, for instance for driving cargo barges. On the other hand in inland applications the power blocks fired with solid, liquid, or gas fuels are in almost 100% the systems with steam or gas turbines.

In the Central Europe, Poland for instance, the basic fuel in power engineering is coal. Conventional electric power plants have the efficiency of an order of 38-42%, and emit large volumes of CO₂, NO_x and/or SO_x. In order to decrease the amount of noxious substances emitted to the atmosphere and reduce the cost of production of the electric energy, combined systems are in use - consisting of gas turbines with a steam turbine circuit.

On the other hand, the combined turbine power plants can be complemented by electric power plants with a Diesel engine as the main propulsion. The exhaust gas leaving the engine contains about 30-40% of the heat delivered to the engine in the fuel. Using the heat from the exhaust gas in the gas and steam turbine circuit will increase the efficiency of the entire combined system. For large powers of piston internal combustion engines, the additional gas and steam turbine circuit is a source of measurable economic savings in electric energy production. Moreover, in large-power piston internal combustion engines we can additionally use the low-temperature waste heat, for instance for heating the communal water (Dzida, 2009). In the seaside areas with no large electric power plants, a mobile power plant situated on a platform close to the coast reveals additional advantages:

- increasing production of electric energy in the so-called distributed system,
- diversification of primary energy sources which decreases the consumption of coal in favour of liquid fuels,
- possible combustion of residual heavy fuels from nearby oil refineries,
- reducing large-distance transport of solid fuels, the absence of slag and cinders,
- reducing the emission of CO₂ and NO_x due to the increased system efficiency,
- shorter time of plant erection compared to that of a conventional power plant, and possibility of opening it in stages: first with the Diesel engine alone, and then complementing it, during plant operation, with a combined steam/gas turbine system,
- no problems with the water cooling the condenser, small effect on the environment in water balance aspects,
- mobility of a combined power plant erected on the marine platform.

2. Concept of a combined system

Combined propulsion systems are used in marine engineering mostly in fast special-purpose ships and in the Navy, as the systems being a combination of a Diesel engine and gas turbines (CODAG, CODOG) or solely gas turbines (COGOG, COGAG). The propulsion system of the passenger liner "Millenium" uses a COGES-type system which improved the efficiency and operating abilities of the ship. The system consists of a gas turbine and a steam turbine which drive an electric current generator, while the propeller screws are driven by electric motors. In this system the steam turbine circuit is supplied with the steam generated in the waste heat boiler supplied with the exhaust gas from the gas turbines.

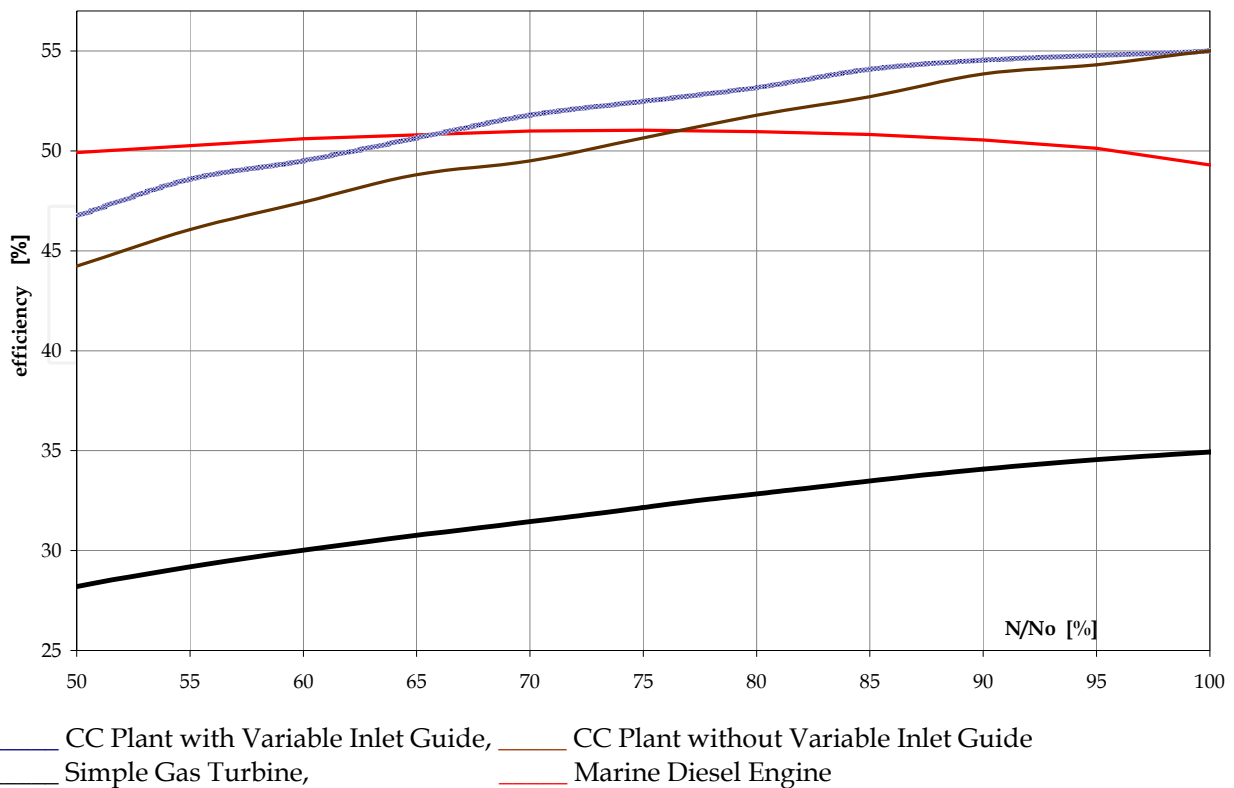


Fig. 1. Part - load Efficiency of a Combined - Cycle Plant (GT&ST), Simple Gas Turbine and Marine Diesel Engine

Combined systems used in inland power blocks base on a gas turbine as the main unit and a steam turbine that utilises the steam produced in a waste heat boiler using the heat recovered from the gas turbine exhaust gas. All this provides opportunities for reaching high efficiency of the combined block. The exhaust gas leaving a marine low-speed Diesel engine contains smaller amount of heat, of an order of 30-40% of the energy delivered to the engine.

Figure 1 shows sample efficiency curves of the combined gas turbine/steam turbine systems as functions of power plant load, compared to the gas turbine operating in a simple open circuit and the marine low-speed Diesel engine.

The efficiency curves in Fig. 1 show that the combined cycle gas turbine/steam turbine system has the highest efficiency for maximal loads (maximal efficiency levels for these circuits reach as much as 60%). Gas turbines operating in the simple open circuit have the lowest efficiency (average values of 33÷35%, and maximal values reaching 40%). Low-speed Diesel engines have the efficiency of an order of 47÷50%. It is also noticeable that the Diesel engine curve is relatively flat. This is of special importance in case of marine propulsion systems which operate at heavily changing loads. For the combined cycle gas turbine/steam turbine systems and the gas turbines operating in the simple open circuit the relative efficiency decrease $\Delta\eta/\eta_0$ is equal to 15÷20% when the load decreases from 100% to 50%. For the low-speed Diesel engine these numbers are equal to 1÷2%. This property of the Diesel engine, along with the ability to utilise additional heat contained in its exhaust gas, makes the engine the most applicable in marine propulsion systems operating in heavily changing load conditions. The amount of heat contained in the

exhaust gas from the gas turbine is approximately equal to 60÷65%, i.e. more than in piston engines, which results from lower exit temperature and less intensive flow of the exit gas leaving the Diesel engine, Figs. 2 and 3.

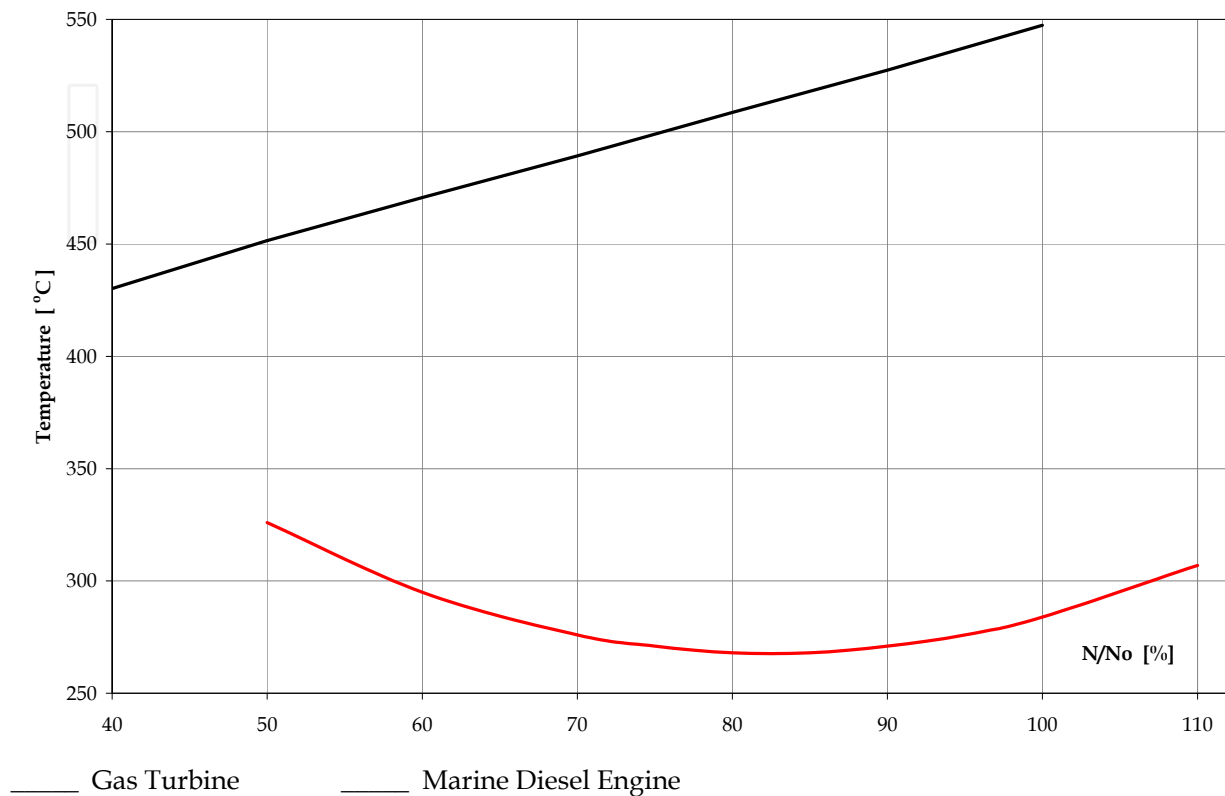


Fig. 2. Temperatures of the exhaust gas from the Diesel engine and the gas turbine as function of power plant load

The exit temperatures of the exhaust gas from the gas turbines range between 450÷600°C, on average, while those from the low-speed Diesel engines are of an order of 220÷300°C. In the gas turbines, decreasing the load remarkably decreases the temperature of the exhaust gas, while in the Diesel engine these changes are much smaller, and the temperature initially decreases and then starts to increase for low loads.

This property of the steam turbine circuit in the combined system with the Diesel engine for partial loads makes it possible to keep the live steam temperature at a constant level within a wide range of load. The related exhaust gas mass flow rate m_g/N [kg/kWh] changes only by about 5% in the Diesel engine when the load changes from 100% to 50%, while in the gas turbine this parameter changes by about 55% for the same load change, Fig. 3.

The combined propulsion system with the low-speed piston internal combustion engine used as the main engine and making use of the heat from the engine exhaust gas is shown in Fig. 4, (Dzida, 2009; Dzida & Mucharski, 2009; Dzida et al., 2009).

The exhaust gas flows leaving individual main engine cylinders are collected in the exhaust manifold and passed to the constant-pressure turbocharger. Due to high turbocharger efficiency ranges (MAN, 2010; Schrott, 1995), the scavenge air can be compressed using the energy contained only in part of the exhaust gas flow. The remaining part of the exhaust gas flow can be expanded in an additional gas turbine, the so-called power turbine, which additionally drives, via a gear, the propeller screw or the electric current generator.

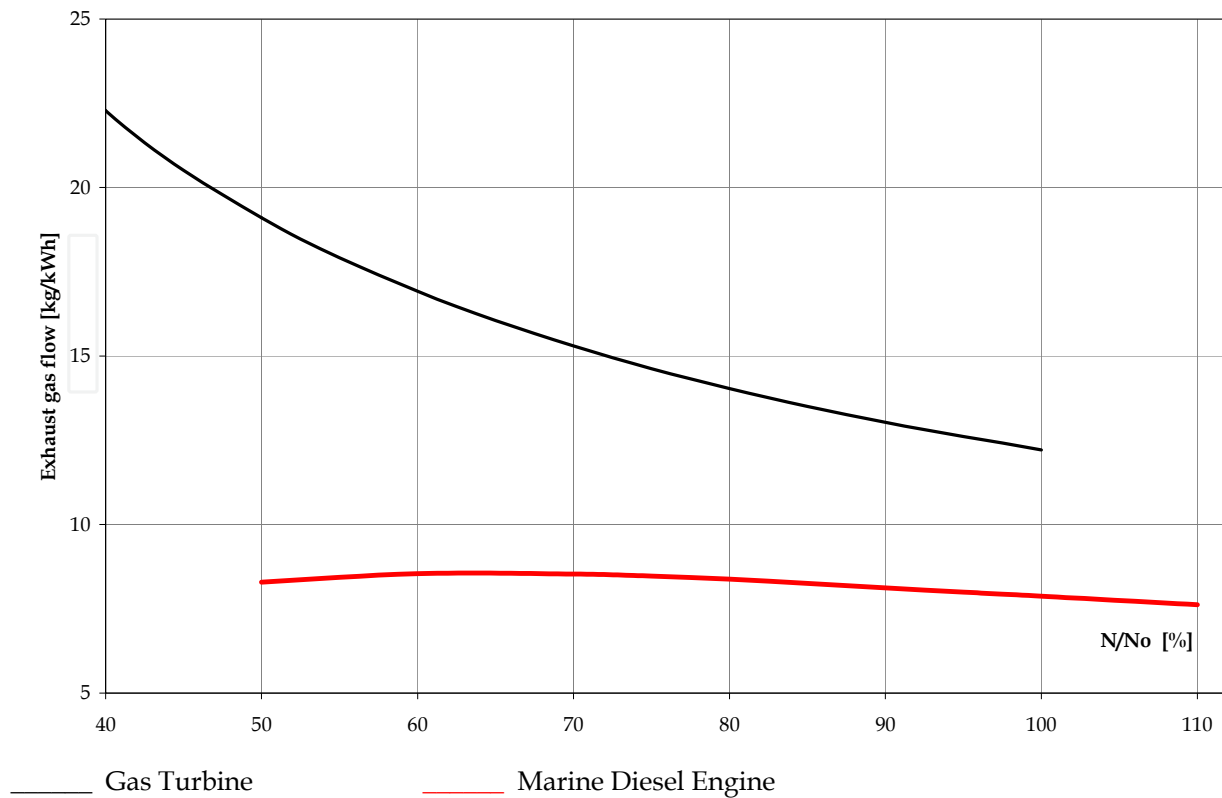


Fig. 3. Related exhaust gas mass flow rate as a function of power plant load

The exhaust gas from the turbocharger and the power turbine flows to the waste heat boiler installed in the main engine exhaust gas path, before the silencer. The waste heat boiler produces the steam used both for driving the steam turbine that passes its energy to the propeller screw, and for covering all-ship needs.

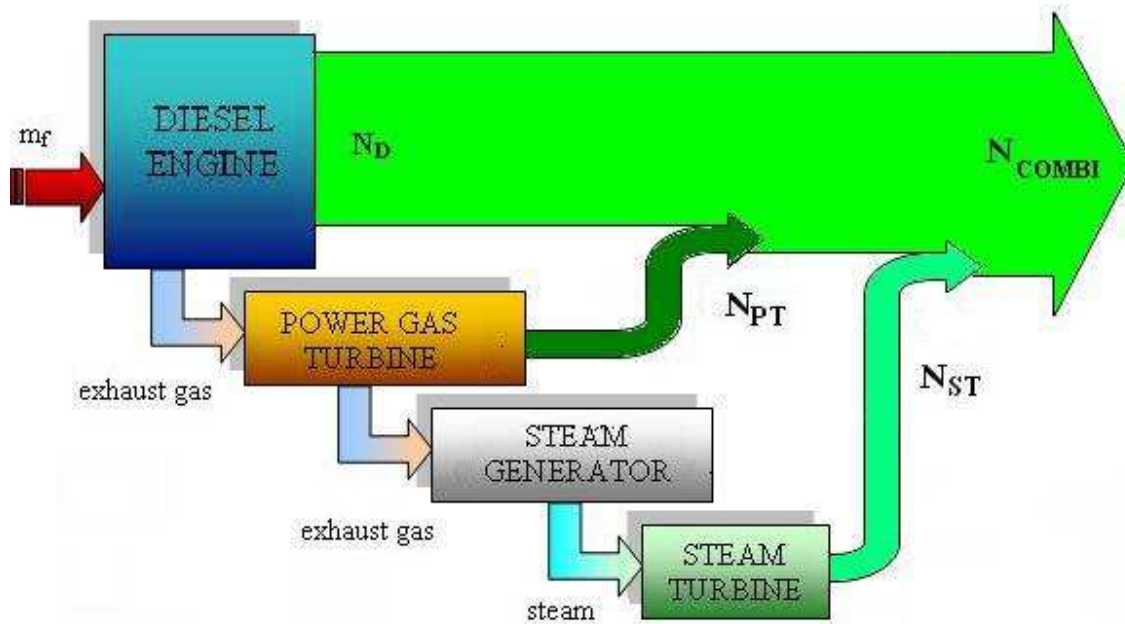


Fig. 4. Concept of the combined propulsion system

In the marine low-speed Diesel engines, another portion of energy that can be used along with the exhaust gas energy is a huge amount of so-called waste heat of relatively low temperature. In the low-speed engines the waste heat comprises the following components (with their proportions to the heat delivered to the engine in fuel):

- heat in the scavenge air cooler (17-20%), of an approximate temperature of about 200°C,
- heat in the lubricating oil cooler (3-5%), of an approximate temperature of about 50°.
- heat in the jacket water cooler (5-6%), of the temperature of an order of 100°C.

This shows that the amount of the waste heat that remains for our disposal is equal to about 25-30% of the heat delivered in fuel. Part of this heat can be used in the combined circuit with the Diesel engine.

2.1 Energy evaluation of the combined propulsion system

The adopted concept of the combined ship propulsion system requires energy evaluation, Fig. 4. Formulas defining the system efficiency are derived on the basis of the adopted scheme.

The power of the combined propulsion system is determined by summing up individual powers of system components (the main engine, the power gas turbine, and the steam turbine):

$$N_{combi} = N_D + N_{PT} + N_{ST} \quad (1)$$

hence the efficiency of the combined system is:

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

and the specific fuel consumption is:

$$b_{ecombi} = b_{eD} \cdot \frac{1}{\left(1 + \frac{N_{PT}}{N_D} + \frac{N_{ST}}{N_D} \right)} \text{ [g / kWh]} \quad (3)$$

where η_D , b_{eD} - is the efficiency and specific fuel consumption of the main engine.

Relations (2) and (3) show that each additional power in the propulsion system increases the system efficiency and, consequently, decreases the fuel consumption. And the higher the additional power achieved from the utilisation of the heat in the exhaust gas leaving the main engine, the lower the specific fuel consumption. Therefore the maximal available power levels are to be achieved from both the power gas turbine and the steam turbine. The power of the steam turbine mainly depends on the live steam and condenser parameters.

2.2 Variants of the combined ship propulsion systems or marine power plants

For large powers of low-speed engines, the exhaust gas leaving the engine contains huge amount of heat available for further utilisation. Marine Diesel engines are always supercharged. Portions of the exhaust gas leaving individual cylinders are collected in the exhaust gas collector, where the exhaust gas pressure $p_{exh,D} > p_{bar}$ is equalised. In standard solutions the constant-pressure turbocharger is supplied with the exhaust gas from the

exhaust manifold to generate the flow of the scavenge air for supercharging the internal combustion engine.

Present-day designs of turbochargers used in piston engines do not need large amounts of exhaust gas, therefore it seems reasonable to use a power gas turbine complementing the operation of the steam turbine in those cases. Here, two variants of power gas turbine supply with the exhaust gas are possible.

2.2.1 Parallel power gas turbine supply (variant A)

In this case part of the exhaust gas from the piston engine exhaust manifold supplies the Diesel engine turbocharger. The remaining part of the exhaust gas from the manifold is directed to the gas turbine, bearing the name of the power turbine (PT). The power turbine drives, via the reduction gear, the propeller screw or the electric current generator, thus additionally increasing the power of the entire system. Figure 5 shows a concept of this propulsion system, referred to as parallel power turbine supply. After the expansion in the turbocharger and the power turbine, the exhaust gas flowing from these two turbines is directed to the waste heat boiler in the steam circuit.

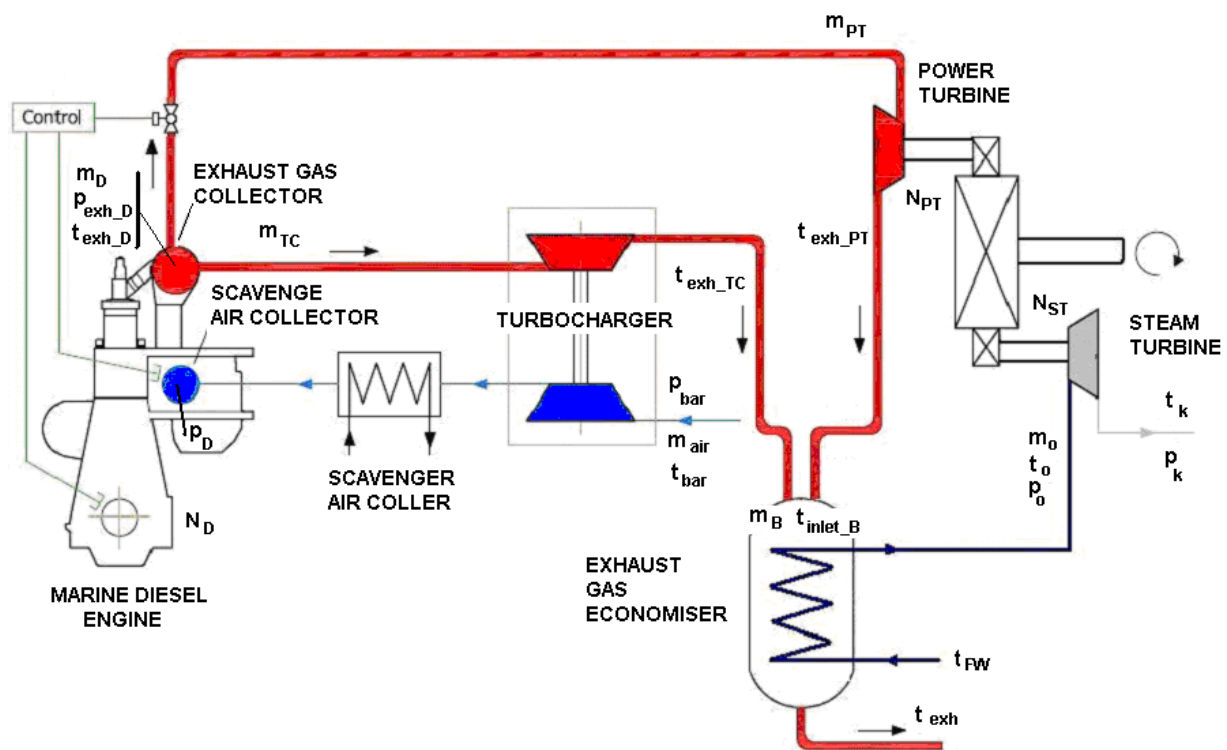


Fig. 5. Combined system with the Diesel main engine, the power turbine supplied in parallel, and the steam turbine (variant A)

In the proposed solution, at low load ranges the amount of the exhaust gas from the main engine is not sufficient to additionally supply the power turbine. In such case a control valve closes the exhaust gas flow to the power turbine, Figure 5. The operation of this valve is controlled by the control system using two signals: the scavenge air pressure signal, and the signal of the propeller shaft angular speed or torque. The waste heat boiler produces the steam which is then used both in the steam turbine and, in case of marine application, to

cover the all-ship needs. This system allows for independent operation of the Diesel engine, with the steam turbine or the power turbine switched off. The control system makes it possible to switch off the power turbine thus increasing the power of the turbocharger at partial load, and, on the other hand, direct part of the Diesel engine exhaust gas to supply the power turbine at large load.

Power turbine calculations are based on the Diesel engine parameters, i.e. the temperature of the exhaust gas in the exhaust gas collector, which in turn depends on the engine load and air parameters at the engine inlet. Marine engine producers most often deliver the data on two reference points for the atmospheric air (the ambient reference conditions):

ISO Conditions		Tropical Conditions
Ambient air temperature [$^{\circ}\text{C}$]	25	45
Barometric pressure [bar]	1	1

2.2.2 Series power gas turbine supply (variant B)

In this variant the exhaust gas from the exhaust manifold supplies first the piston engine turbocharger and then the power turbine, Fig.6.

After leaving the exhaust manifold, the exhaust gas expands in the turbocharger to the higher pressure than the atmospheric pressure, which leaves part of the exhaust gas enthalpy drop for utilisation in the power turbine. The exhaust gas leaving the power turbine passes its heat to the steam in the waste heat boiler, thus producing additional power in the steam turbine circuit.

Also in this combined system, the installed control valve makes it possible to switch off the power turbine at partial piston engine loads, thus increasing the power of the turbocharger by expanding the exhaust gas to lower pressure, Fig. 6. Unlike the parallel supply variant, here the entire mass of the exhaust gas from the piston engine manifold flows through the turbocharger. The exhaust gas pressure at the turbocharger outlet is higher than in variant A.

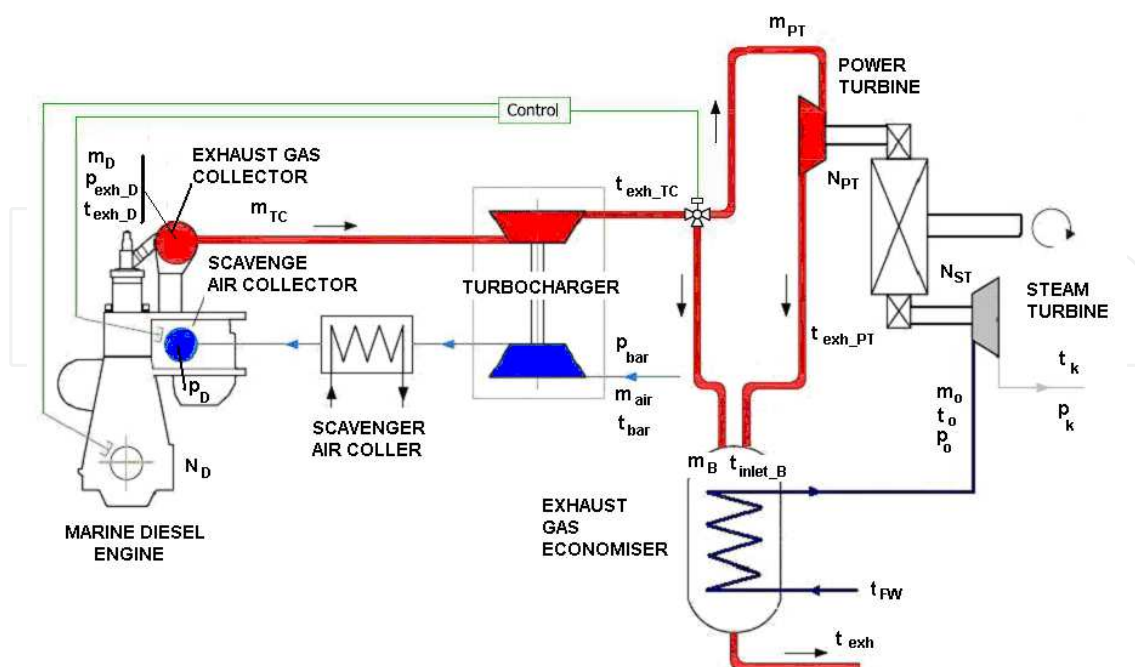


Fig. 6. Combined system with the Diesel main engine, the power turbine supplied in series, and the steam turbine (variant B)

3. Power turbine in the combined system

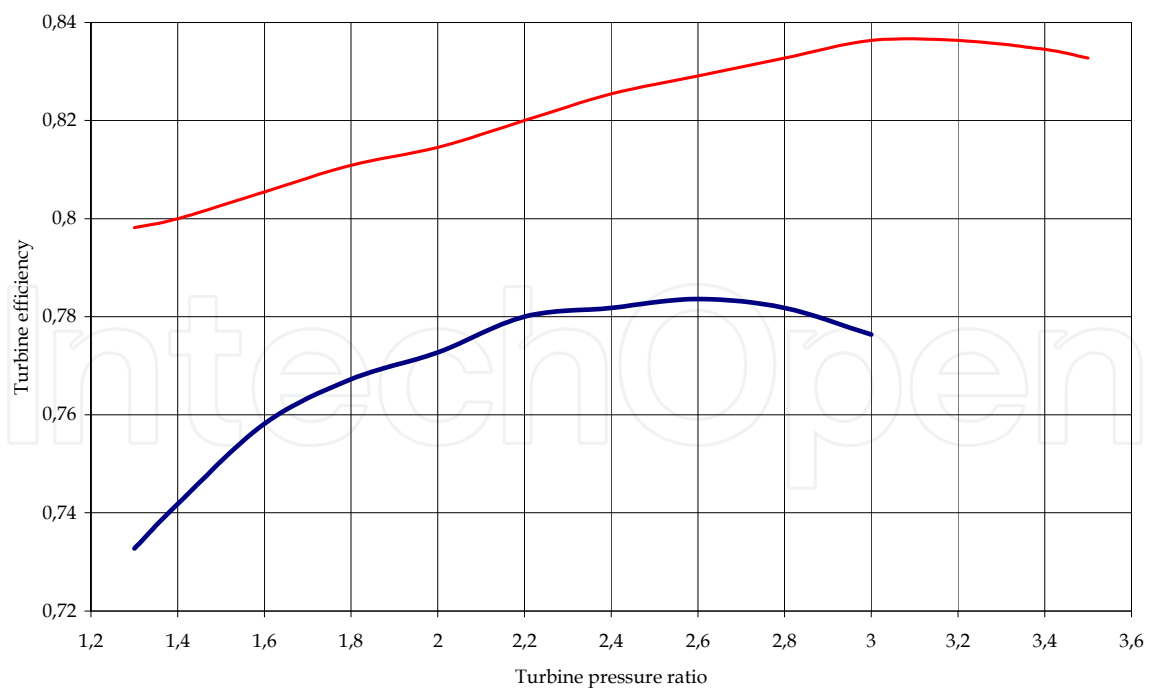
Calculating the power turbine in the combined system depends on the selected variant of power turbine supply. Usually, piston engine producers do not deliver the exhaust gas temperature in the exhaust manifold (which is equal to the exhaust gas temperature at turbocharger turbine inlet). Instead, they give the exhaust gas temperature at turbocharger turbine outlet (t_{exh_D}). The temperatures of the exhaust gas in the Diesel engine exhaust gas collector are calculated from the turbine power balance, according to the following formula:

$$t_{exh_D} = \frac{t_{exh_TC} + 273,15}{1 - \eta_T \cdot \left(1 - \frac{1}{\pi_T^{\frac{\kappa_g - 1}{\kappa_g}}} \right)} - 273,15 \text{ [}^\circ\text{C]} \quad (4)$$

This formula needs the data on turbocharger turbine efficiency changes for partial loads. These data can be obtained from the producer of the turbocharger (as they are rarely made public), Fig. 7, or calculated based on the relation used in steam turbine stage calculations:

$$\frac{\eta_T}{\eta_{T0}} = 2 \cdot \bar{v} - \bar{v}^2 \quad (5)$$

where \bar{v} - related turbine speed indicator, η_{T0} - maximal turbine efficiency and the corresponding speed indicator.



— turbine of S - wheel type — turbine of R - wheel type

Fig. 7. Turbocharger turbine efficiency as a function of scavenge air pressure, acc. to (Schrott, 1995)

The turbine speed indicator is defined as:

$$v = \frac{u}{c_s} = \sqrt{\frac{u^2}{2 \cdot H_T}} \quad (6)$$

where u - circumferential velocity on the turbine stage pitch diameter, H_T - enthalpy drop in the turbine.

The calculations make use of static characteristics of the turbocharger compressor, with the marked line of cooperation with the Diesel engine, Fig.8.

Figure 9 shows the turbocharger efficiency curves calculated from the relation:

$$\eta_{TC} = \eta_T \cdot \eta_C \cdot \eta_m \quad (7)$$

where η_T - the turbocharger turbine efficiency is calculated from relation (5), while the compressor efficiency η_C is calculated from the line of Diesel engine/compressor cooperation, Fig. 8. In the same figure a comparison is made between the calculated turbocharger turbine efficiency with the producer's data as a function of the Diesel engine scavenge pressure. The differences between these curves do not exceed 1,5%.

For the presently available turbocharger efficiency ranges, the amount of the exhaust gas needed for driving the turbocharger turbine is smaller than the entire mass flow rate of the exhaust gas leaving the Diesel engine. Fig. 10 shows sample curves of exhaust gas

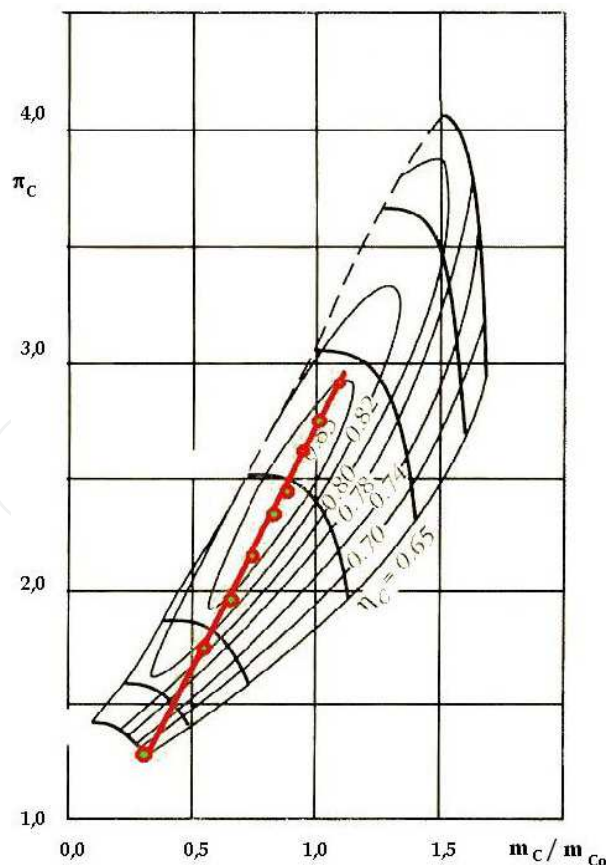


Fig. 8. Diesel engine cooperation line against turbocharger compressor characteristics

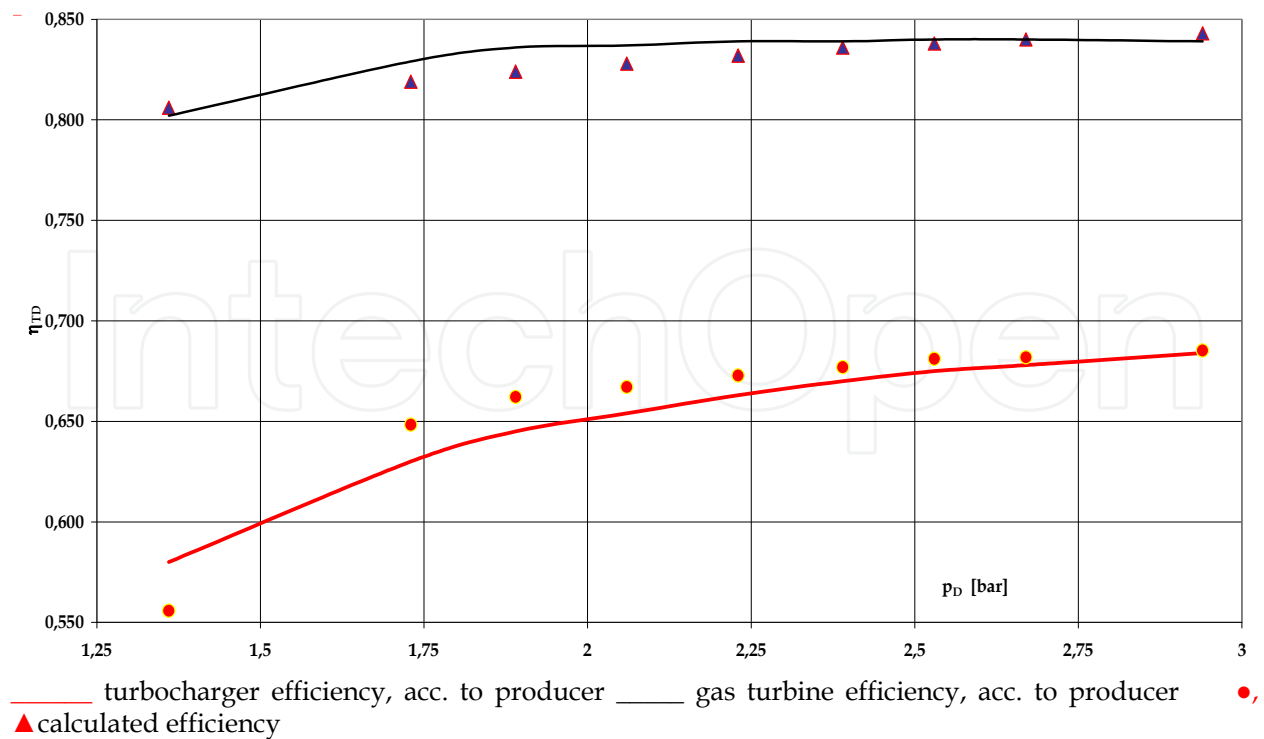


Fig. 9. Efficiency characteristics of the turbocharger and the turbocharger gas turbine as a function of scavenge air pressure

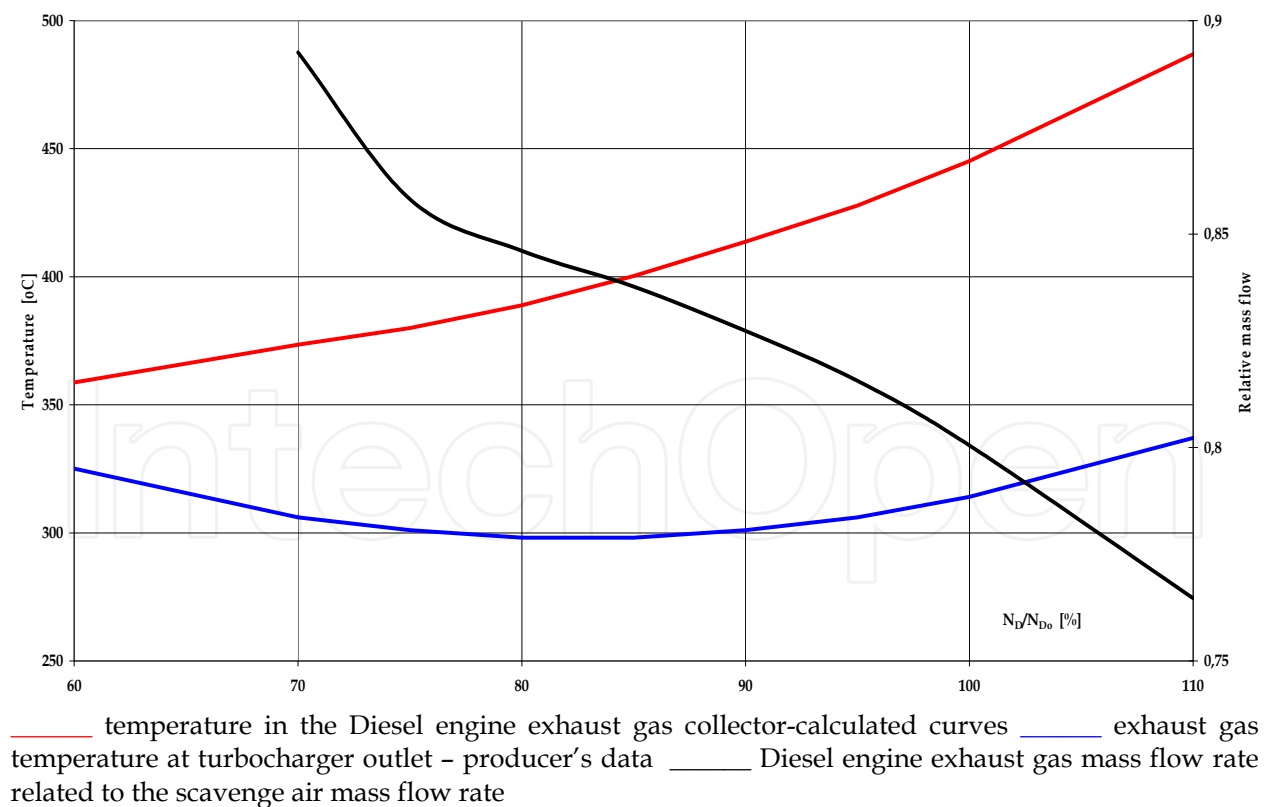


Fig. 10. Sample temperature characteristics of the turbocharger during gas expansion in the turbine to the atmospheric pressure and the related exhaust gas mass flow rates as functions of Diesel engine load

temperature changes in the engine manifold (calculated using the relation (4)) and the exhaust gas temperature at the turbocharger outlet (according to the data delivered by the producer) as functions of engine load, when the standard internal combustion engine exhaust gas is expanded to the barometric pressure. The figure also shows the Diesel engine exhaust gas flow rate related to the scavenge air flow rate, as a function of the engine load. This high efficiency of the turbocharger provides opportunities for installing a power gas turbine connected in parallel with the turbocharger (variant A).

The turbocharger power balance indicates that in the power gas turbine we can utilise between 10 and 24% of the flow rate of the exhaust gas leaving the exhaust manifold of the piston engine. The power gas turbine can be switched on when the main engine power output exceeds 60%. For lower power outputs the entire exhaust gas flow leaving the Diesel engine is to be used for driving the turbocharger.

In variant B of the combined system with the power turbine, the turbocharger is connected in series with the power gas turbine. Here, the entire amount of the exhaust gas flows through the turbocharger turbine. Due to the excess of the power needed for driving the turbocharger, the final expansion pressure at turbocharger turbine output can be higher than the exhaust gas pressure at waste heat boiler inlet. In this case the expansion ratio in the turbocharger turbine is given by the relation:

$$\pi_T = \left[\frac{1}{1 - \frac{1}{\eta_{TC}} \cdot \frac{m_a}{m_D} \cdot \frac{c_a}{c_g} \cdot \frac{t_a}{t_{exh_D}} \cdot \left(\pi_C \frac{\kappa_a - 1}{\kappa_a} \right)} \right]^{\frac{\kappa_g}{\kappa_g - 1}} \quad (8)$$

where: π_C - compression ratio of the turbocharger compressor.

The exhaust gas temperature at turbocharger outlet is calculated from the formula:

$$t_{exh_TC} = (t_{exh_D} + 273,15) \cdot \left[1 - \eta_T \left(1 - \frac{1}{\pi_T \frac{\kappa_g}{\kappa_g - 1}} \right) \right] - 273,15 \text{ [}^\circ\text{C]} \quad (9)$$

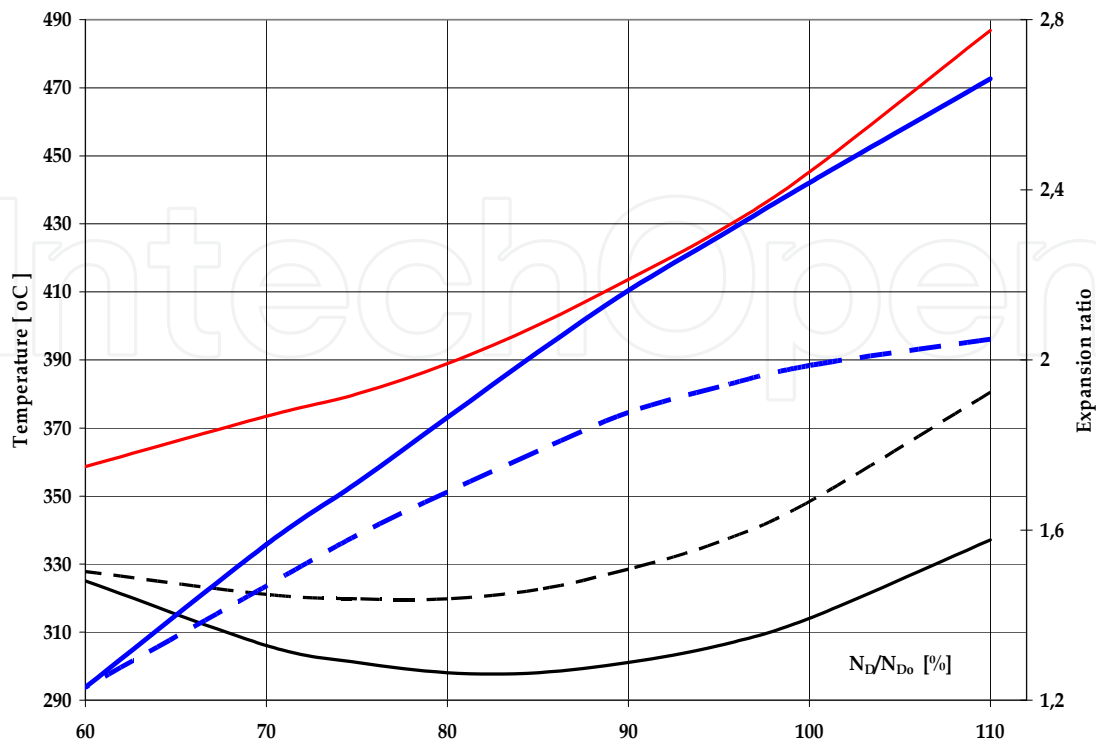
Figure 11 shows sample curves of temperature, compression and expansion rate changes in the turbocharger for variant B: series power turbine supply.

This case provides opportunities for utilising the enthalpy drop of the expanding exhaust gas in the power turbine. The operation of the power turbine is possible when the Diesel engine power exceeds 60%.

3.1 Power turbine in parallel supply system (variant A)

The power turbine (Fig.5) is supplied with the exhaust gas from the exhaust manifold. The exhaust gas mass flow rate m_{PT} and temperature t_{exh_D} are identical as those at turbocharger outlet: the mass flow rate of the exhaust gas flowing through the power turbine results from the difference between the mass flow rate of the Diesel engine exhaust gas and of that expanding in the turbocharger:

$$m_{TD} = m_a \cdot (1 - \bar{m}) + m_{pD} \quad (10)$$



— expansion ratio in the turbocharger turbine (standard arrangement - without power turbine) - - - expansion ratio in the turbocharger turbine with power turbine — exhaust gas temperature in the Diesel engine exhaust gas collector — exhaust gas temperature at turbocharger outlet without power turbine - - - exhaust gas temperature at turbocharger outlet with power turbine

Fig. 11. Changes of temperature and expansion ratio of the turbocharger in the combined system with series power turbine supply (variant B)

The mass flow rate of the exhaust gas needed by the turbocharger is calculated from the turbocharger power balance using the following formula:

$$\dot{m} \equiv \frac{m_{TC}}{m_a} = \frac{1 - \frac{1}{\pi_T^{\frac{\kappa_g - 1}{\kappa_g}}}}{\pi_C^{\frac{\kappa_a - 1}{\kappa_a}} - 1} \cdot \frac{T_{exh_D}}{T_a} \cdot \frac{c_g}{c_p} \cdot \eta_{TC} \quad (10.1)$$

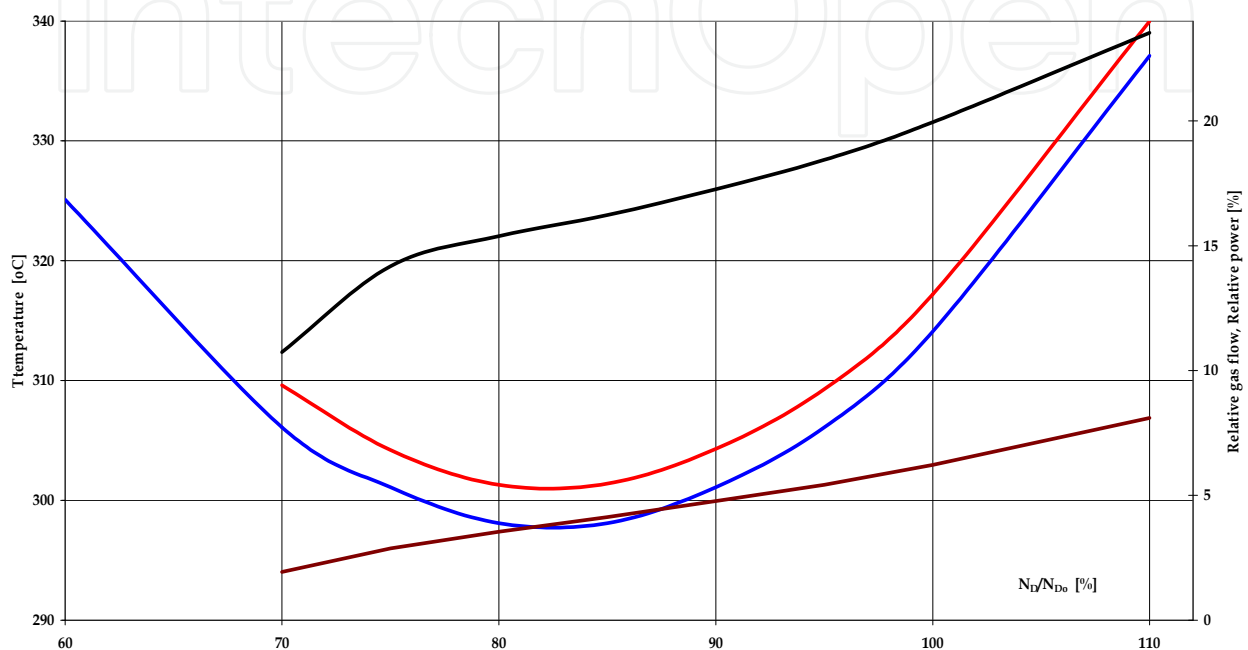
The exhaust gas expanding in the power turbine has the inlet and outlet pressures identical to those of the exhaust gas flowing through the turbocharger. The power of the power turbine is given by the relation:

$$N_{PT} = \eta_m \cdot \eta_{PT} \cdot m_{PT} \cdot H_{PT} \quad (11)$$

where η_m - mechanical efficiency of the power turbine, H_{PT} - iso-entropic enthalpy drop in the power turbine.

The power turbine efficiency η_{PT} is assumed in the same way as for the turbocharger turbine, Fig. 9, or using the relation (5). In the shipbuilding, the gas turbines used in combined Diesel engine systems with power turbines are those adopted from turbochargers.

The power turbine system calculations show that the exhaust gas temperature at the power turbine outlet is slightly higher than that at the turbocharger outlet, Fig.12. The increase of the main engine load results in the increase of both the exhaust gas temperature in the exhaust gas collector and the mass flow rate of the exhaust gas flowing through the power turbine. The increase in power of the combined system with additional power turbine ranges from about 2% for Diesel engine loads of an order of 70% up to over 8% for maximal loads, Fig.12.



— temperature at turbocharger outlet — temperature at power turbine outlet — related exhaust gas mass flow rate in power turbine — related power turbine power

Fig. 12. Parameters of parallel supplied power turbine as functions of the main engine load – variant A (calculations for tropical conditions)

When the Diesel engine power is lower than 60-70% of the nominal value the entire exhaust gas flow from the exhaust manifold is directed to the turbocharger drive. In this case the control system closes the valve controlling the exhaust gas flow to the power turbine, Fig. 5.

3.2 Power turbine in series supply system (variant B)

In this variant the power turbine is supplied with the full amount of the exhaust gas leaving the Diesel engine exhaust manifold. The power turbine is installed after the turbocharger. The exhaust gas pressure at the power turbine inlet depends on the pressure of the exhaust gas leaving the turbocharger turbine, Fig.11.

In this case the power of the power turbine is calculated as:

$$N_{PT} = \eta_{PT} \cdot m_D \cdot c_g \cdot t_{in_PT} \cdot \left(1 - \frac{1}{\pi_{PT}^{\frac{\kappa_g - 1}{\kappa_g}}} \right) \quad (12)$$

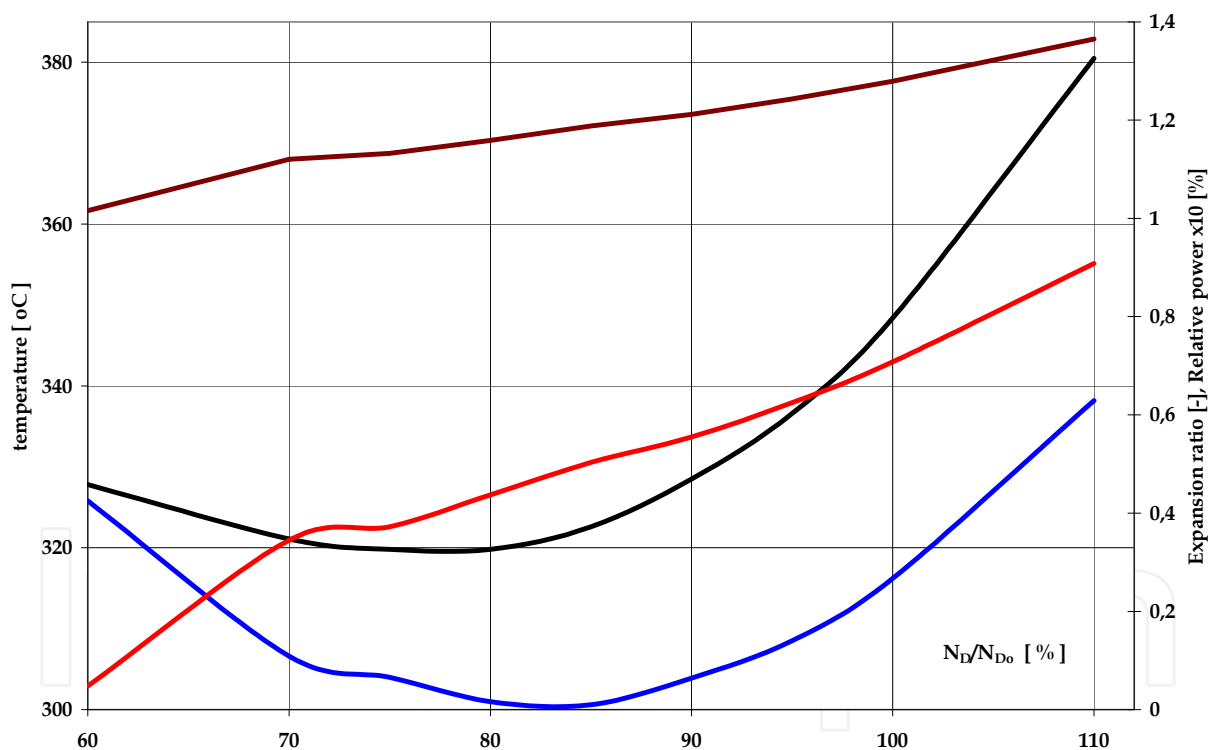
where t_{in_PT} - exhaust gas temperature at the power turbine inlet, π_{PT} - expansion ratio in the power turbine, η_{PT} - power turbine efficiency. The power turbine efficiency is assumed in the same way as in variant A.

In formula (12) the exhaust gas temperature at the power turbine inlet is assumed equal to that of the exhaust gas leaving the turbocharger, Fig. 13.

The exhaust gas temperature at the power turbine output is calculated from the formula:

$$t_{exh_PT} = (t_{in_PT} + 273,15) \cdot \left[1 - \eta_{PT} \left(1 - \frac{1}{\pi_{PT}^{\frac{\kappa_g - 1}{\kappa_g}}} \right) \right] - 273,15 [^{\circ}C] \quad (13)$$

Figure 13 also shows the expansion ratio, the power of the power turbine, and the exhaust gas temperatures at the turbocharger and the power turbine outlets for partial engine loads. The power turbine in this variant increases the power of the combined system by 3% to 9% with respect to that of a standard engine. The turbine power increases with increasing Diesel engine load.



— temperature at turbocharger outlet — temperature at power turbine outlet
 — expansion ratio in power turbine — related power of power turbine

Fig. 13. Parameters of series supplied power turbine as functions of the main engine load - variant B (calculations for tropical conditions)

3.3 Comparing the two power turbine supply variants

The analysis of the two examined variants shows that the power of the combined system increases depending on the Diesel engine load. For both variants the power turbine can be

used after exceeding about 65% of the Diesel engine power. The exhaust gas leaving the power turbine is directed to the waste heat boiler, where together with steam turbine it can additionally increase the overall power of the combined system.

In both cases the temperatures of the exhaust gas leaving the power turbine are comparable. The exhaust gas pressure at power turbine outlet depends on the losses generated when the gas flows through the waste heat boiler and outlet silencers. Following practical experience, the exhaust gas back pressure is assumed higher than the barometric pressure by 300 mmWC, i.e. about 3%. Taking into account powers of the power turbines for the above variants, Fig. 14, it shows that for the same Diesel engine parameters the series supply of the power turbine results in higher turbine power. For lower loads, the power of the series supplied power turbine increases, compared to the parallel supply variant.

4. Steam turbine circuit

The combined system makes use of the waste heat from the Diesel engine. In modern Diesel engines the temperatures of the waste heat are at the advantageous levels for the steam turbine circuit. This circuit makes use of water that can be utilised in a low-temperature process. Adding the steam circuit to the combined Diesel engine/power gas turbine system provides good opportunities for increasing the power of the combined system, and consequently, also the system efficiency, see formula (2).

In the examined combined system the exhaust gas leaving the turbocharger and the power turbine (variant A, Fig. 5) or only the power turbine (variant B, Fig. 6) flows to the waste heat boiler where it is used for producing superheated steam for driving the steam turbine.

The mass flow rate of the exhaust gas reaching the waste heat boiler is equal to that leaving the Diesel engine exhaust gas collector. The exhaust gas temperature at waste heat boiler inlet depends on the adopted solution of power turbine supply. For variant A with parallel supply it is calculated from the balance of mixing of the gases leaving the turbocharger and the power turbine:

$$t_{inl_B} = \frac{m_{TC} \cdot i_{exh_TC} + m_{PT} \cdot i_{exh_PT}}{m_D \cdot c_g} - 273,15 \text{ [}^\circ\text{C]} \quad (14)$$

while for the series power turbine supply (variant B) it is assumed equal to that at the power turbine outlet, formula (13).

In combined steam turbine systems for small power ranges and low live steam temperatures the single pressure systems are used, Fig. 15, (Kehlhofer, 1991).

Such system consists of a single-pressure waste heat boiler, a condensing steam turbine, a water-cooled condenser, and a single stage feed water preheater in the deaerator.

The main disadvantage of the systems of this type is poor utilisation of the heat contained in the exhaust gas (the waste heat energy). The steam superheater is relatively large, as the entire mass of the steam produced by the boiler flows through it. However, costs of this steam system are the lowest, as poor utilisation of the exhaust gas energy results in high temperature of the exhaust gas leaving the boiler. The deaerator is supplied with the steam extracted from the steam turbine. The application of the single pressure system does not secure optimal utilisation of the exhaust gas energy.

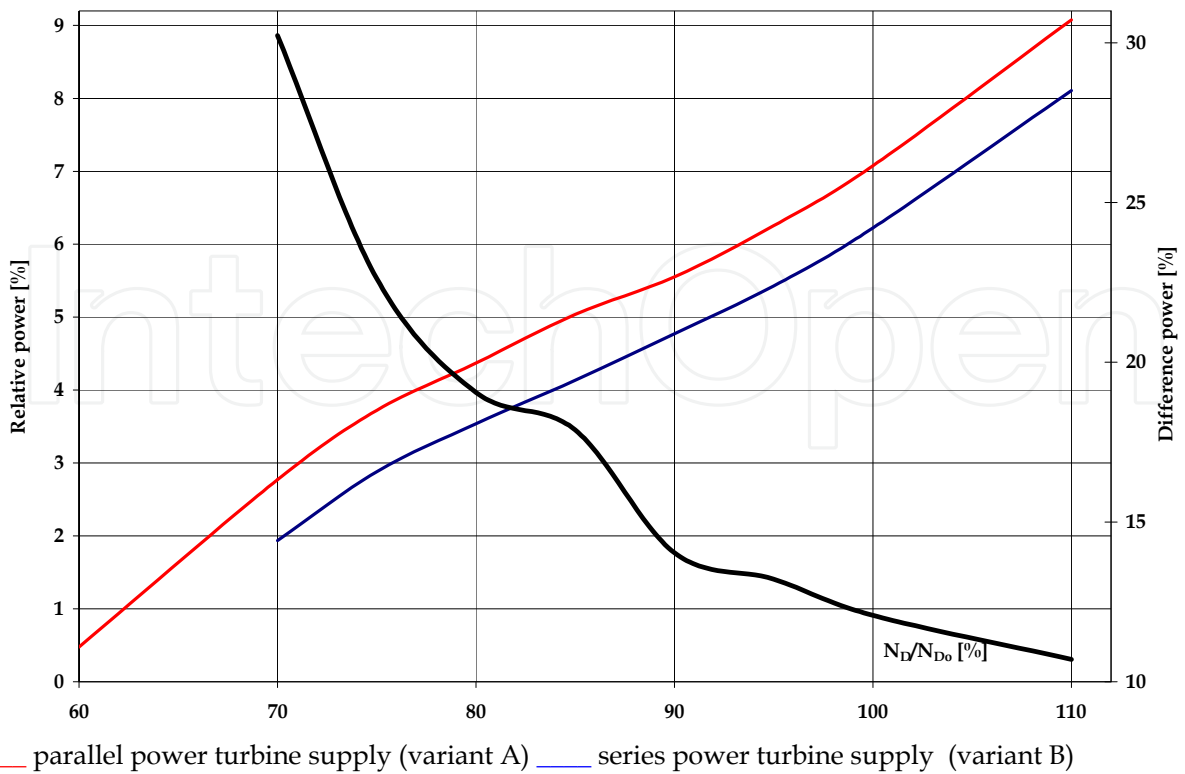
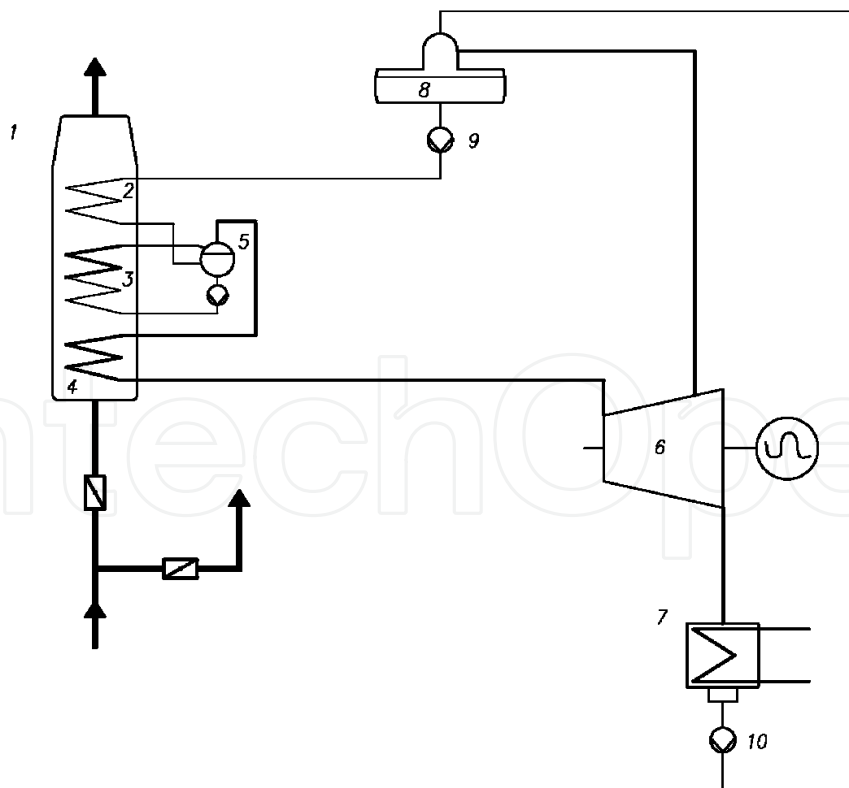


Fig. 14. Powers of the power turbine as functions of main engine load



1-Waste Heat Boiler 2-Superheater 3- Evaporator 4-Ekonomizer 5-Boiler drum 6-Steam turbine 7- Condenser 8-Deaerator 9-Feed water pump 10-Condensate pump

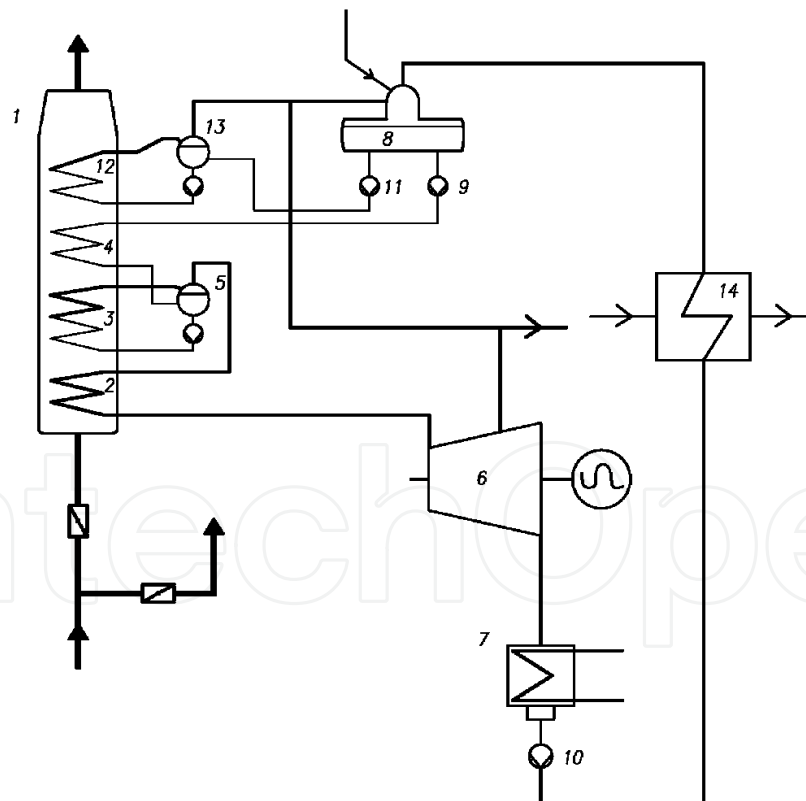
Fig. 15. Flow Diagram of the Single Pressure System

Those steam turbine systems frequently make use of an additional low-pressure evaporator, Fig. 16, which leads not only to more intensive utilisation of the waste heat contained in the exhaust gas, but also to better thermodynamic use of the low-pressure steam.

In this solution the high pressure superheater is relatively small, compared to the single pressure boiler. The deaerator is heated with the saturated steam from the low-pressure evaporator. The power of the main high-pressure feeding pump is also smaller. The excess steam from the low-pressure evaporator can be used for supplying the low-pressure part of the steam turbine, thus increasing its power, or, alternatively, for covering all-ship needs.

Figure 16 shows possible use of the temperature waste heat from the scavenge air cooler, the lubricating oil cooler, and from the jacket water cooler in the low-pressure water pre-heater.

The additional low-pressure exchanger in the steam circuit, Fig. 16, makes it possible to increase the temperature of the water in the deaerator. Higher water temperature is required due to the presence of sulphur in the fuel (water dew-point in the exhaust gas) – it is favourable for systems fed with a high sulphur content fuel. If the temperature of the feedwater is low when the system is fed with fuel without sulphur, the heat exchanger 14 in Fig. 16 is not necessary and the waste heat from the coolers can be used in the deaerator. For a low feedwater temperature the deaerator works at the pressure below atmospheric (under the vacuum).

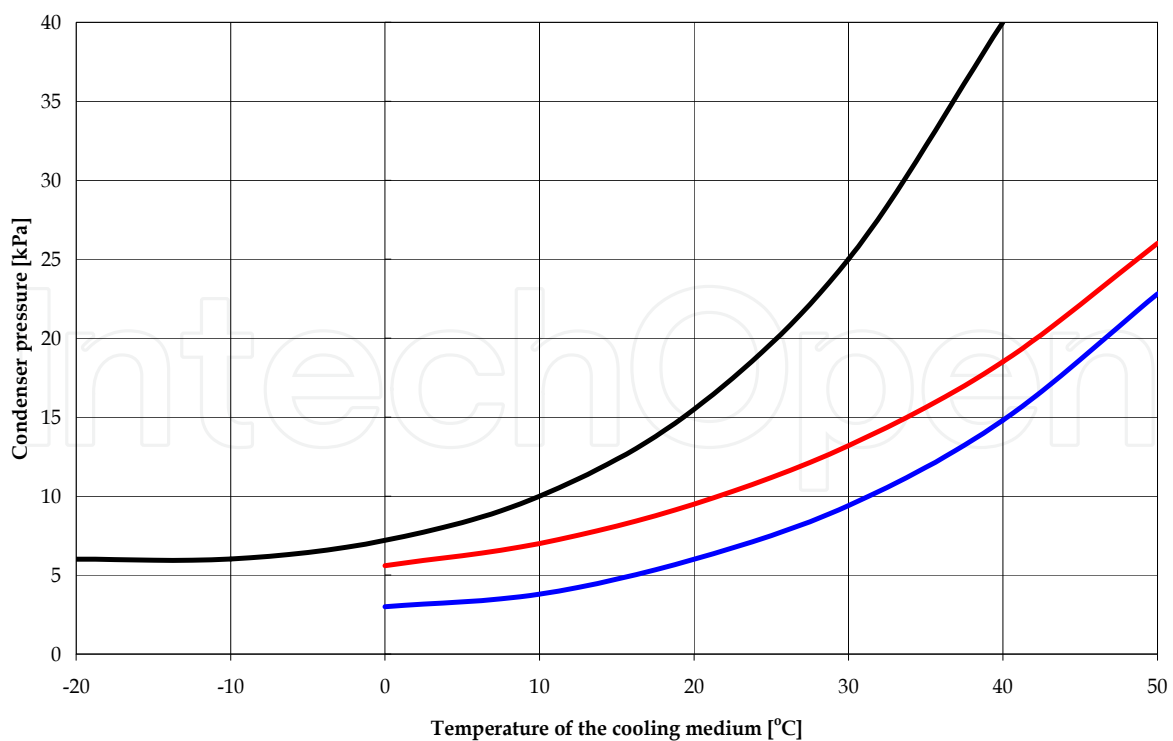


1-Waste Heat Boiler 2-High pressure superheater 3- High pressure evaporator 4- High pressure economizer 5- High pressure boiler drum 6 -Steam turbine 7-Condenser 8- Deaerator 9-High pressure feed water pump 10-Condensate pump 11-Low pressure feed pump 12-Low pressure evaporator 13- Low pressure boiler drum 14-Low pressure pre-heater

Fig. 16. Flow Diagram for a Two - Pressure System

4.1 Limits for steam circuit parameters

The limits for the values of the steam circuit parameters result from strength and technical requirements concerning the durability of particular system components, but also from design and economic restrictions. The difference between the exhaust gas temperature and the live steam temperature, Δt , for waste heat boilers used in shipbuilding is assumed as $\Delta t = 10\text{-}15^\circ\text{C}$, according to (MAN, 1985; Kehlhofer, 1991). The “pitch point” value recommended by MAN B&W (MAN, 1985) for marine boilers is $\delta t = 8\text{-}12^\circ\text{C}$. The limiting dryness factor x of the steam downstream of the steam turbine is assumed as $x_{\text{limit}} = 0,86\text{-}0,88$. For marine condensers cooled with sea water, MAN recommends the condenser pressure $p_K = 0,065$ bar. This pressure depends on the B&W (MAN, 1985) temperature of the cooling medium in the condenser. Figure 17 shows the dependence of the condenser pressure on the cooling medium temperature. The temperature of the boiler feed water is of high importance for the life time of the feed water heater in the boiler. The value of this temperature is connected with a so-called exhaust gas dew-point temperature. Below this temperature the water condensates on heater tubes and reacts with the sulphur trioxide SO_3 producing the sulphuric acid, which is the source of low-temperature corrosion. That is why boiler producers give minimal feed water temperatures below which boiler operation is highly not recommended. The dew-point temperature is connected with the content of sulphur in the fuel and depends on the excess air coefficient in the piston engine. Figure 18 shows the dew-point temperature as the function of: sulphur content in the fuel, SO_2 conversion to SO_3 , and the excess air coefficient in the engine. In inland power installations burning fuels with sulphur content higher than 2%, the recommended level of feed water temperature is $t_{\text{FW}} > 140\text{-}145^\circ\text{C}$ (Kehlhofer, 1991).



— Fresh Water Cooling — Wet Cooling Tower — Direct Air Condensation

Fig. 17. Condenser pressure as a function of temperature of the cooling medium

In marine propulsion (MAN, 1985) recommends that the feed water temperature should not be lower than 120°C when the sulphur content is higher than 2%. This is justified by the fact that the outer surface of the heater tubes on the exhaust gas side has the temperature higher by 8-15 °C than the feed water temperature, and that the materials used in those heaters reveal enhanced resistance to acid corrosion.

The exhaust gas temperature at the boiler outlet is assumed higher by 15-20°C than the feed water temperature, i.e. $t_{\text{exh}} > t_{\text{FW}} + (15 - 20^\circ\text{C})$.

Each ship burning heavy fuel in its power plant uses the mass flow rate m_{SS} of the saturated steam taken from the waste heat boiler for fuel pre-heating and all-ship purposes. According to the recommendations (MAN, 1985) the pressure of the steam used for these purposes should range between $p_{\text{SS}} = 7-9$ bar. This pressure is also assumed equal to the pressure in the boiler low-pressure circuit. The back temperature of the above steam flow in the heat box is within 50 - 60°C.

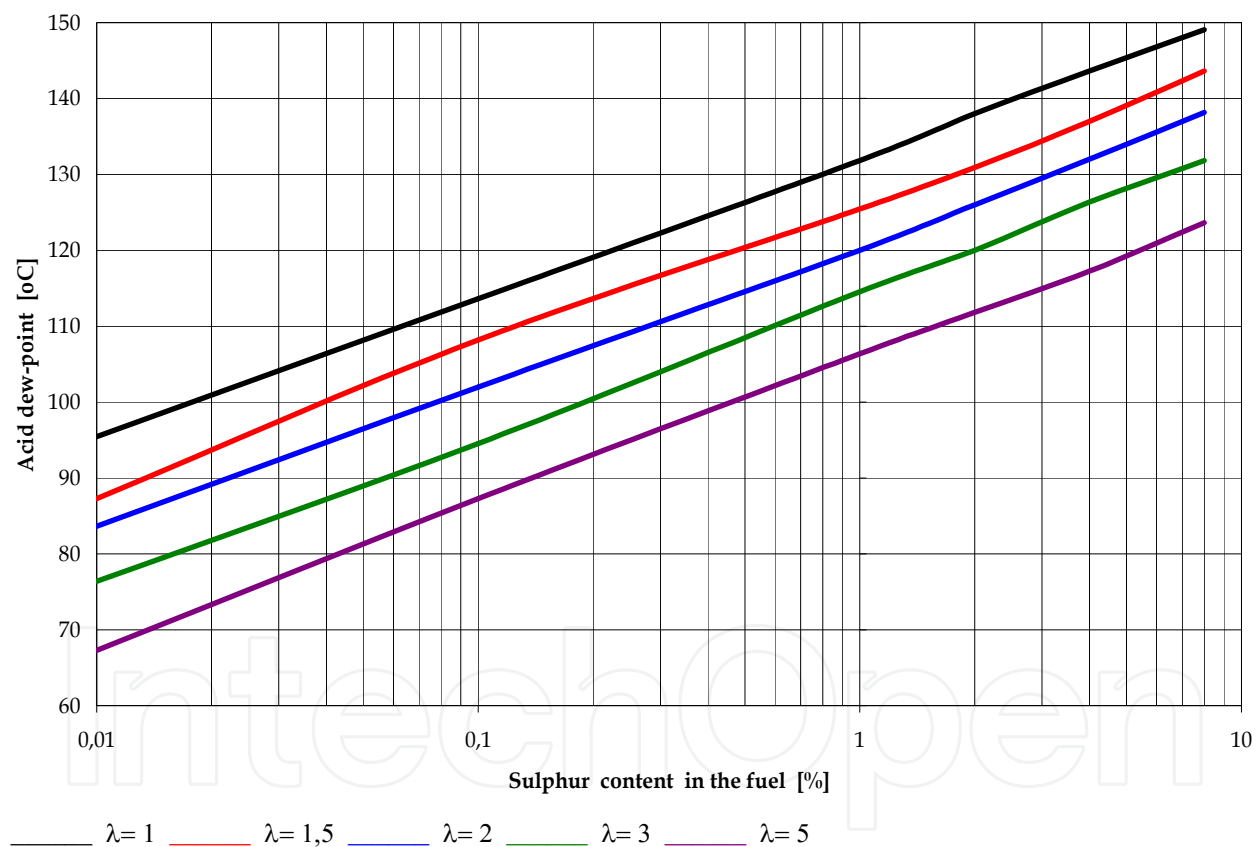


Fig. 18. Acid dew-point as a function of the sulphur content in the fuel and the excess air coefficient λ

4.2 Optimising the steam circuit

Optimisation of the steam system is to be done in such a way so as to reach the maximal possible utilisation of the heat contained in the exhaust gas. In this sense the optimisation is reduced to selecting the steam circuit parameters for which the steam turbine reaches the highest power. The area of search for optimal steam circuit parameters is to be narrowed to

the sub-area where the earlier discussed limits imposed on the steam system are met. The use of the steam system with the waste heat boiler increases the power of the propulsion system within the entire range of the main engine load.

Adding a steam turbine to the Diesel engine system increases the power of the propulsion system by $\Delta N_{ST}/N_D = 6,5 - 7,5\%$ for main engine loads ranging from 90 to 100%. The power of the steam turbine for both examined variants of power turbine supply are comparable, and slightly higher power, by about 2-4%, is obtained by the steam turbine in the variant with series power turbine supply.

The analysis of the system with an additional exchanger utilising the low-temperature waste heat from the Diesel engine to heat the condensate from the condenser before the deaerator, Fig.16, shows that the steam turbine power increases by 7- 11% with respect to that of the steam turbine without this exchanger.

The requirements concerning the waste heat boiler refer to low loss of the exhaust gas flow (which reduces the final expansion pressure in the power turbine) and small temperature concentrations (pitch points) in the boiler evaporators. There is a remarkable impact of the sulphur content in the fuel on the permissible exhaust gas temperature and the lower feed water temperature limit. In the steam turbine circuit, a minimal number of exchangers should be used (optimally: none). The optimal parameters of this circuit also depend on the piston engine load.

5. Conclusions

It is possible to implement a combined system consisting of a Diesel engine as the leading engine, a power gas turbine, and a steam turbine circuit utilising the heat contained in the Diesel engine exhaust gas. Such systems can reveal thermodynamic efficiencies comparable with combined gas turbine circuits connected with steam turbines.

5.1 Power range of combined systems

Depending on the adopted variant and the main engine load, the use of the combined system makes it possible to increase the power of the power plant by 7 to 15 % with respect to the conventional power plant burning the same rate of fuel. Additional power is obtained by the system due to the recovery of the energy contained in the exhaust gas leaving the piston internal combustion engine. Thus the combined system decreases the specific fuel consumption by 6,4 - 12,8 % compared to the conventional power plant.

In the examined systems the power of the steam turbine is higher than that of the power turbine by 6-29 %, depending on the system variant and the main engine load.

5.2 Efficiency of combined systems

The use of the combined system for ship propulsion increases the efficiency of the propulsion system, and decreases the specific fuel consumption. Additionally, it increases the propulsion power without additional fuel consumption.

Like the power, the efficiency of the combined system increases with respect to the conventional power plant by 7 to 15% reaching the level of 53 - 56% for maximal power ranges. These efficiency levels are comparable with the combined systems based on the steam/gas turbines, Fig. 1. For partial loads the efficiency curves of the combined system

with the Diesel engine are more flat than those for the combined turbine systems (smaller efficiency decrease following the load decrease).

In the combined system the maximal efficiency is reached using particular system components:

- the piston internal combustion engine with the maximal efficiency;
- Turbocharger. The turbocharger with the maximal efficiency should be used as it provides opportunities for decreasing the exhaust gas enthalpy drop in the turbine in case of the series supply variant, or exhaust gas mass flow rate in case of the parallel supply variant, which in both cases results in higher power of the power turbine;
- Power turbine. High efficiency is required to increase its power;
- Steam turbine circuit. The requirement is to obtain the maximal power of the steam turbine from the heat delivered in the exhaust gas flowing through the boiler.

5.3 Ecology

Along with the thermodynamic profits, having the form of efficiency increase, and the economic gains, reducing the fuel consumption for the same power output of the propulsion system, the use of the combined system brings also ecological profits. A typical new-generation low-speed piston engine fed with heavy fuel oil with the sulphur content of 3% emits 17g/kWh NO_x, 12g/kWh SO_x and 600g/kWhCO₂ to the atmosphere. The use of the combined system reduces the emission of the noxious substances by, respectively, g/kWh NO_x, g/kWh SO_x and g/kWhCO₂. The emission decreases by % with respect to the standard engine, solely because of the increased system efficiency, without any additional installations.

Depending on the adopted solution, the combined power plant provides opportunities for reaching the assumed power of the propulsion system at a lower load of the main Diesel engine, at the same time also reducing the fuel consumption.

The article presents the thermodynamic analysis of the combined system consisting of the Diesel engine, the power gas turbine, and the steam turbine, without additional technical and economic analysis which will fully justify the application of this type of propulsion systems in power conversion systems.

6. Nomenclature

b_e	- specific fuel oil consumption
c_g, c_a	- specific heat of exhaust gas and air, respectively
i	- specific enthalpy
m	- mass flow rate
N	- power
p	- pressure
T, t	- temperature
W_u	- calorific value of fuel oil
η	- efficiency
κ_g, κ_a	- isentropic exponent of exhaust gas and air, respectively

Indices:

a	- air
bar	- barometric conditions
B	- Boiler
C	- Compressor
combi	- combined system
D	- Diesel engine
d	- supercharging
exh	- exhaust passage
f	- fuel
FW	- feet water
g	- exhaust gas
inlet	- inlet passage
k	- parameters in a condenser
o	- live steam, calculation point
PT	- Power turbine
ST	- Steam turbine
ss	- ship living purposes
T	- Turbine
TC	- Turbocharger
π	- compression ratio in a compressor, expansion ratio in a turbine

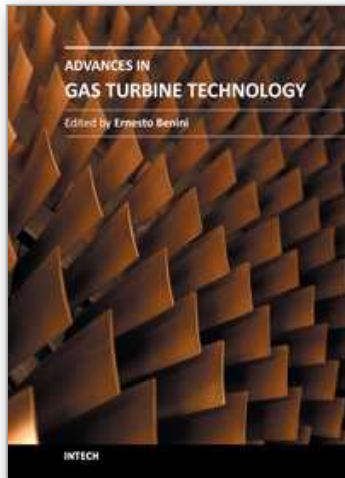
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Gas turbine engines will still represent a key technology in the next 20-year energy scenarios, either in stand-alone applications or in combination with other power generation equipment. This book intends in fact to provide an updated picture as well as a perspective vision of some of the major improvements that characterize the gas turbine technology in different applications, from marine and aircraft propulsion to industrial and stationary power generation. Therefore, the target audience for it involves design, analyst, materials and maintenance engineers. Also manufacturers, researchers and scientists will benefit from the timely and accurate information provided in this volume. The book is organized into five main sections including 21 chapters overall: (I) Aero and Marine Gas Turbines, (II) Gas Turbine Systems, (III) Heat Transfer, (IV) Combustion and (V) Materials and Fabrication.

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